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Investigations on the Use of Rubber Seed Oil in a Diesel Engine Using Waste Exhaust Heat Energy

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Abstract – *Rubber seed oil can be used as an alternative fuel for diesel engine operation. But its high viscosity causes improper atomization of fuel during injection, resulting in incomplete combustion. This leads to smoky exhaust in diesel engines. The high viscosity of vegetable oils can be reduced by preheating. The heat energy from the exhaust, which is otherwise wasted, can be used to preheat the vegetable oil. In the present work a heat exchanger was designed and fabricated to preheat the rubber seed oil (RSO) for use in diesel engines. It was observed that the RSO requires a temperature of 155°C and 133°C to bring down its viscosity to that of diesel and rubber seed oil methyl ester (RSOME). The work is related to the study of exhaust preheated RSO and its ester on the performance and emission characteristics of a single cylinder DI diesel engine. Experimental results indicated that there is a marginal increase in brake thermal efficiency when the fuel is preheated to a temperature of 155°C. Smoke level of preheated RSO is significantly reduced, compared with RSO (without preheating). Preheated fuel improves the fuel spray pattern that leads to better mixing; hence the ignition delay and combustion duration decreases with preheated oil, which indicates an increase in combustion rate. The overall result shows that by preheating the RSO, the engine performance and exhaust emission characteristics improves significantly and approaches the performance of RSOME in a diesel engine.*

Keywords – Combustion, emissions, exhaust gas preheating, heat exchanger, rubber seed oil.

1. INTRODUCTION

Diesel engines have gained its name and fame by serving society in many ways. Its main attractions are ruggedness in construction, simplicity in operation and ease of maintenance. However, due to shortage of fossil fuels, they may not be able to avail their services for a long time. Hence, efforts are being taken all over the world, to find alternative fuels for diesel engines. Vegetable oils can be used as an alternative to diesel. The main advantages of using vegetable oils are that many of their properties are quite close to diesel.

The inventor of the diesel engine, Rudolf Diesel confidently predicted that plant based oils would be widely used to operate his engine. In fact he used peanut oil as fuel for demonstrating his engine in 1890. In this regard vegetable oil can be a partial replacement for diesel oil, which forms the basis for our transportation and agricultural operations. Over a few decades a lot of research has gone into the use of alternative fuels in IC engines. Vegetable oils seem to be a forerunner as they are renewable and easily available. In an agricultural country like India, the use of vegetable oil would be economical because of larger productivity and reduced dependability on import of petroleum products. Use of these oils is also beneficial, as they do not contain sulphur, aromatic hydrocarbon, metals and crude oil residues. Only non-edible oils can be used as fuels in India, since edible oils are in great demand and expensive for their applications.

2. BACKGROUND

Work has been done earlier to evaluate the performance of different vegetable oils viz, palm oil, sunflower oil, rapeseed oil, cotton seed oil, rice bran oil, jatropha oil, neem oil, karanji oil, etc. [1]-[3]. Research has revealed that engine durability is adversely affected by the use of vegetable oils in the long-term. These problems can be traced back to the high viscosity, low volatility, and reactivity of unsaturated hydrocarbon chains. Of these factors, high viscosity of vegetable oils poses the major hurdle as it causes problems like excessive carbon deposits, ring sticking, plugging of injector orifices, improper injection, atomization and incomplete combustion [4]-[5]. Therefore, a reduction in viscosity is of prime importance to make vegetable oils a suitable alternative fuel for diesel engines. The problem of high viscosity of vegetable oils has been studied in several ways, such as preheating the oils, blending with other fuels, transesterification and thermal cracking/pyrolysis [6]-[8].

Among these, transesterification is found to be the best method to reduce the viscosity of vegetable oil. The free fatty acid content of RSO was about 12.91%. If the FFA content is more than 3%, two step transesterification processes (consisting of acid and alkaline process) is needed to convert the raw vegetable oil to its esters, which will increase the production cost of biodiesel. And also higher percentage of FFA in the oil reduces the yield of the esterification process [9]. So exhaust preheating can be considered as an alternate and effective method to utilize the high FFA RSO in diesel engines for power generation and agriculture purpose in rural areas.

Bari *et al.* [10] used preheated crude palm oil as a fuel in a diesel engine. They demonstrated that preheating

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is essential for the smooth flow of fuel through the injection system. However, excessive heating of crude palm oil reduces its viscosity to too low values and it deteriorates fuel pump delivery efficiency and engine power output. Senthil Kumar *et al.* [11] have indicated that the peak pressure and rate of pressure rise are high with animal fat at higher fuel inlet temperature. Improvement in premixed combustion phase with preheated animal fat is also reported. Nazar *et al.* [12] reported that preheated karanji oil and neem oil have significant improvement in engine performance and reduction in CO and HC emissions when compared to crude oils.

Earlier work on preheating of vegetable oil was done with an electrical heater [10]-[12]. This shows improvement in many aspects but engine efficiency reduces due to extra source of heating. From the experiment, it is shown that the temperature of exhaust coming out from the engine when it is run by vegetable oil is higher than that of diesel. Higher temperature of the exhaust, which is otherwise wasted, can be used to preheat the vegetable oil using a properly designed heat exchanger for this purpose.

The present work is mainly concerned with the study of the performance and emissions of a naturally aspirated direct injection diesel engine using exhaust gas preheated vegetable oil. It was found that while heating, viscosity of vegetable oils reduced rapidly. The performance of diesel engine was evaluated by using a heat exchanger to preheat the rubber seed oil at the temperature where the viscosity of this vegetable oil is equal to the viscosity of RSOME and diesel at room temperature and the results are compared with RSOME and diesel fuels.

3. DESIGN OF HEAT EXCHANGER

In this experimental work, the heat exchanger was designed to exchange heat between the exhaust gas and vegetable oil (RSO). Heat exchanger should be indirect contact type, time for heat transfer from one fluid to another should be less, mass flow rate of tube and shell side should be less and heat exchanger should be simple in construction. Based on the above requirements a double pipe counter flow heat exchanger was designed [13]. To determine the size of the heat exchanger the following steps were adopted:

Step 1

Based on the engine requirement a shell made of stainless steel of inner diameter 25.4mm was chosen for the exhaust gas flow. The shell was directly fixed to the engine exhaust manifold. Tube made of copper with inner and outer diameter of 4.76mm and 5.76mm respectively were chosen based on the injection pipe diameter of the engine for the fuel flow.

Step 2

The Reynolds Number (Re) was calculated and found to be greater than 2300, hence the flow inside the shell was assumed to be turbulent. Dittus-Boelter correlation $Nu_D = 0.023\,\mathrm{Re}_D^{0.8}\,\mathrm{Pr}^n$ where n is 0.4. Then the heat transfer coefficient (h) was calculated on the shell side by using the relation $Nu_D = \frac{hD}{k}$, where k is the thermal conductivity (218 W/m^2-K) and D is the shell diameter.

In the similar manner the tube side heat transfer coefficient was calculated and found to be 127W/m^2-K. The flow inside the tube was assumed laminar.

The overall heat transfer coefficient (U) was calculated using Equation 1:

$$\frac{1}{U} = \frac{1}{(r_i / r_o)h_i} + \frac{F_i}{r_i / r_o} + \frac{r_o \ln(r_i / r_o)}{k} + F_o + \frac{1}{h_o} \qquad (1)$$

Where, r is the tube radius, F is the fouling factor and k is the thermal conductivity of tube material. The overall heat transfer coefficient is found to be 62.8W/m^2-k.

Step 3

The required area was calculated from Equation 2:

$$A_{total} = Q / U_{total} \Delta T_m$$

$$\Delta T_m = F \Delta T_{\ln}$$

$$Q = U * A * F(LMTD)$$

$$LMTD = \frac{(T_1 - t_1) - (T_2 - t_2)}{\ln(\frac{T_1 - t_1}{T_2 - t_2})} \qquad (2)$$

$$R = \frac{T_1 - T_2}{t_2 - t_1}$$

$$P = \frac{t_2 - t_1}{T_2 - t_1}$$

Where Q is the total heat load to be transferred, U is the overall heat transfer coefficient, F is the correction factor based on R and P, LMTD is the log mean temperature difference and $\Delta T_{\ln}$ is the log mean temperature. The total area is found to be 11.12 *10^{-2}m$^{2.}$

Step 4

The length of the heat exchanger was calculated by using Equation 3:

$$L_f = \frac{A_{total}}{P_{fs}} \qquad (3)$$

Where P_{fs} is the surface tube perimeter in cm, A_{total} is the total area of the heat exchanger in cm^2 and L_f is the total length of the heat exchanger that comes out as 760 cm.

Step 5

The pressure drop using friction factor f for the tube side was calculated using Equation 4:

$$\Delta P = f(L/D)(\rho V^2{}_m / 2) + 0.5(\rho V^2{}_m / 2) \qquad (4)$$

where V_m is the mean velocity on the tube side, f is the friction factor. The pressure drop was found to be 42.47*10^3 N/m^2. To compensate for this pressure drop the heat exchanger was fixed one meter below the oil tank. The pressure drop was not computed on the shell side because the exhaust was let out into the atmosphere. The photographic view of the fabricated exhaust gas heat exchanger is shown in Figure 1.

4. EXPERIMENTAL SET-UP AND PROCEDURE

A single cylinder, 4 stroke, air cooled, direct injection, constant speed, compression ignition engine developing power output of 4.4 kW was used for this work. The test engine specifications are given in Table 1. An electrical dynamometer was used for loading the engine. An orifice meter connected to an air surge tank was attached to inlet manifold of the engine to measure the airflow. A digital data acquisition system in conjunction with a piezo electric pressure transducer was used for the measurement of engine cylinder pressure at every one-degree crank angle. A software was used to compute the average pressure values for 100 consecutive cycles. From this the peak pressure, occurrence of peak pressure, maximum rate of pressure rise and heat release rate were calculated. The carbon monoxide (CO), unburnt hydrocarbon (HC) and oxides of nitrogen (NO_x) emissions were measured by QROTECH, QRO-401 exhaust gas analyzer. This analyzer performs the measurement by applying Non dispersive infrared (NDIR) method for CO and HC and electrochemical method for NO_x. Smoke density was measured from the smoke paper by using Bosch smoke meter. The schematic diagram of the experimental set up is shown in Figure 2.

Fig. 1. Photographic view of exhaust gas heat exchanger.

Table 1. Specifications of engine used

Make and model	Kirloskar TAF-1
Type	4-stroke, air-cooled, direct injection, compression ignition
Bore × stroke	87.5mm ×110 mm
Compression ratio	17.5:1
Rated output	4.4 kW at 1500 rpm
Fuel injection pressure	210 bar
Static injection timing	23° bTDC

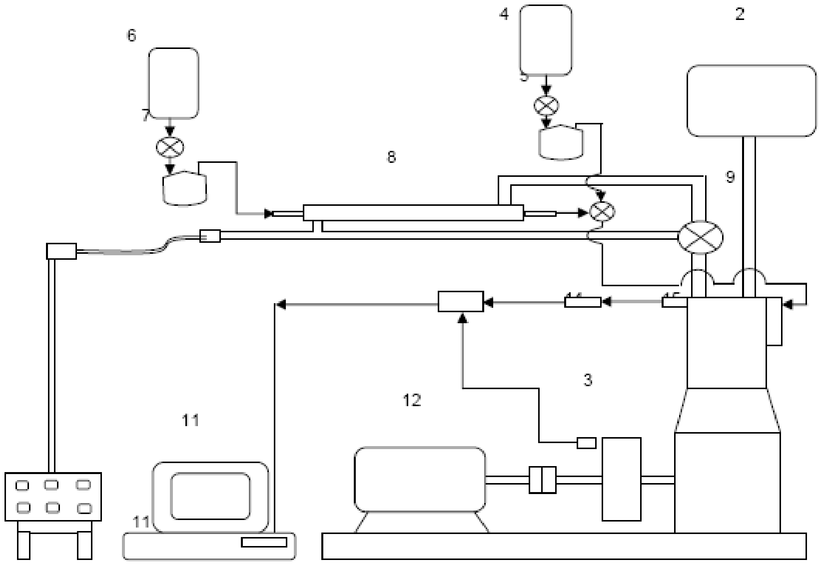

Fig. 2. Schematic diagram of the experimental setup.

1.Engine, 2.Air surge tank, 3.Crank angle encoder, 4.Diesel fuel tank, 5.Diesel fuel filter, 6.RSO fuel tank, 7.RSO fuel filter, 8.Exhaust gas heat exchanger, 9.Exhaust gas by pass valve, 10.Exhaust gas analyzer, 11.Computer, 12.Elecric generator, 13.Analog to digital converter card, 14.Charge amplifier, 15.Cylinder pressure transducer

In the present experimental work, rubber seed oil (RSO) was used as a fuel in a diesel engine. The main yield of rubber tree is rubber milk (natural rubber) which is used in several ways in India. At present rubber seed oil is used for the manufacture of soap and lubrication oil. The production of rubber seed is 150 kg/hectare in India. The oil content in the rubber seed is 40-60% [9]. The properties of RSO and RSOME are given in Table 2.

Initially experiments were conducted on the engine using diesel, RSOME and RSO as fuels for baseline data. The experiment was repeated with RSO at different fuel temperatures. It was found that RSO has to be heated using exhaust gas to 133°C and 155°C to bring down its viscosity as that of RSOME and diesel. The engine gives the maximum efficiency with diesel, so the experiment has been conducted with 155ºC to achieve the maximum possible efficiency with RSO. RSOME gives higher efficiency than that of raw RSO, so the experiment has been conducted with 133ºC just to compare the various results. The exhaust gas temperature increases from 167°C to 410°C (no load to full load) for RSO operation. Hence it was not possible to run the engine above 155ºC at no load. Also at temperatures beyond 155ºC, the viscosity of RSO is very less and impurities present in the RSO starts vaporizing which causes the problem of vapor lock in the injection system.

Table 2. Properties of diesel, rubber seed oil methyl ester and rubber seed oil.

Properties	Diese	RSOME	RSO
Specific gravity	0.83	0.88	0.91
Acid value	0.062	0.18	24.8
Net calorific value, kJ/kg	42,900	41,070	37,500
Cetane number	45	49	42
Kinematic Viscosity at 40°C (c St)	3.01	5.96	58.76

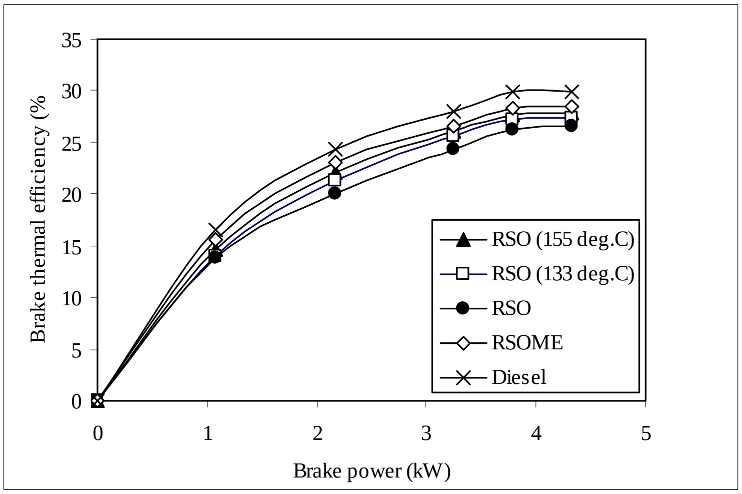

Fig. 3. Variation of brake thermal efficiency with brake power.

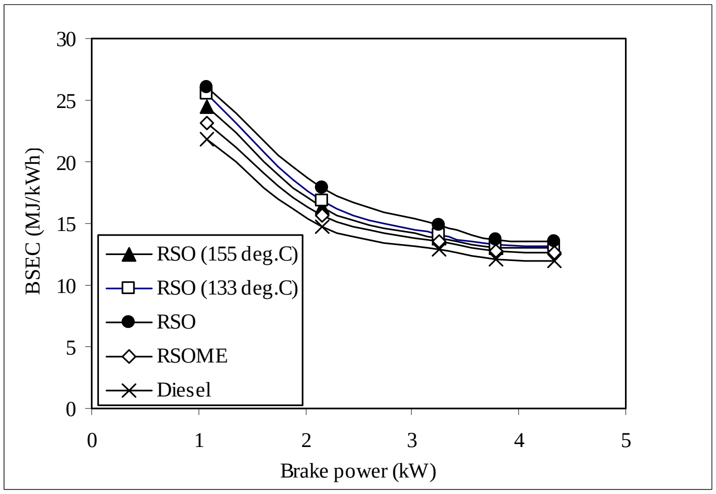

Fig. 4. Variation of brake specific energy consumption with brake power.

Figure 4 presents the comparison of brake specific energy consumption. The brake specific energy consumption of raw RSO (without preheating) is higher compared with other fuels. When it is converted to RSOME the brake specific energy consumption decreases. The increase in BSEC of the engine with RSO is due to low volatility, low heat value and high viscosity, which affect the mixture formation resulting in poor combustion. In addition lower heating value of RSO requires larger fuel flow rates to maintain constant energy input to the engine. The BSEC reduces with RSO at 133°C and 155°C compared to raw RSO. The reduction in BSEC for preheated RSO compared to RSO is due to improvement in viscosity and better atomization of fuel.

The variation of exhaust gas temperature with brake power for different modes of operation is shown in Figure 5. RSO shows the highest exhaust temperature than RSOME. Higher viscosity and poor volatility of RSO lead to poor mixture formation and hence more dominant diffusion combustion phase which prolongs the heat release process. This leads to higher exhaust gas temperature. The longer heat release will lead to lesser thermal efficiency with RSO. The exhaust gas temperature decreases marginally with increase in fuel inlet temperature compared to RSO. This is because reduction in viscosity improves the combustion of RSO.

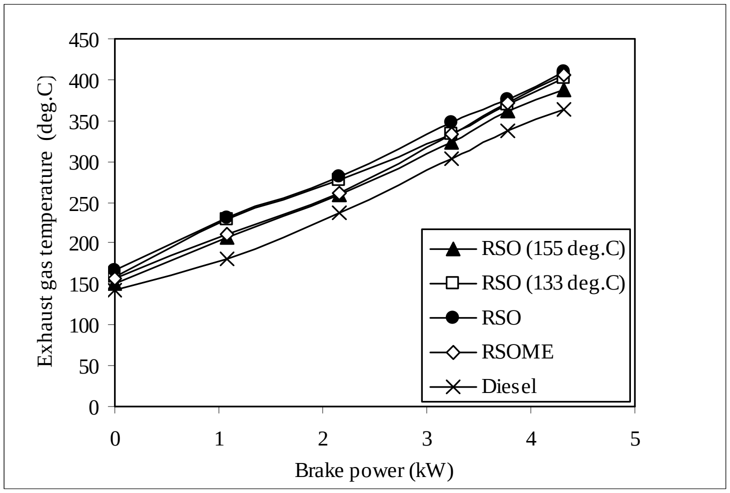

Fig. 5. Variation of exhaust gas temperature with brake power.

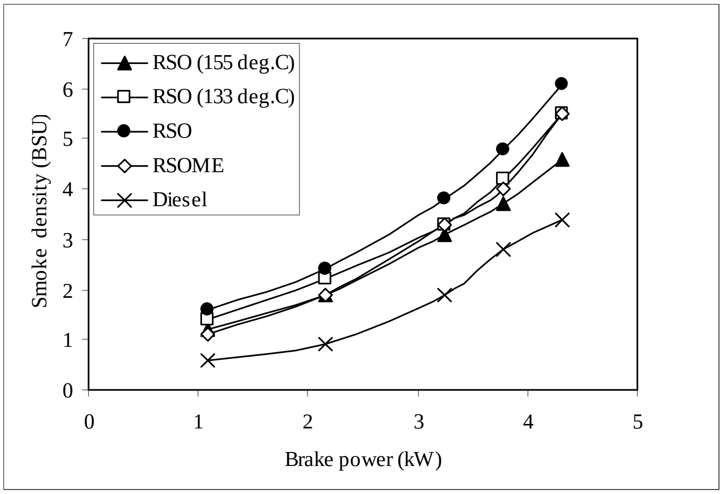

Fig. 6. Variation of smoke density with brake power.

Figure 6 shows the smoke emission characteristics of the engine operated with diesel, RSO, preheated RSO and RSOME fuel. The smoke density of raw RSO is very high compared to diesel. Due to heavier molecular structure and higher viscosity, atomization becomes poor and this leads to larger droplet sizes consequently sluggish combustion which results in higher smoke emission for raw RSO. However, the smoke density decreases with preheated RSO. With preheated RSO, reduction in smoke level and better combustion can be achieved due to the reduction in viscosity and a better fuel air mixing rates.

The variation of CO emission with brake power for the test fuels is shown in Figure 7. RSO results in higher CO emission compared to RSOME and diesel. Due to the poor spray characteristics as a result of higher viscosity of RSO, some of the fuel droplets will not get burned. When these droplets mix with the hot gases in the later part of the power stroke and early exhaust stroke, oxidation reaction of the fuel occurs but do not have enough time for complete combustion. As the fuel temperature increases, the CO emission decreases. The fall in CO emission is an outcome of improved oxidation of carbon monoxide to carbon dioxide. The increase in fuel temperature of preheated oil will result in finer spray and thus good oxidation of fuel and over rich pockets will be minimized. This is the reason for the reduced CO with high fuel temperature.

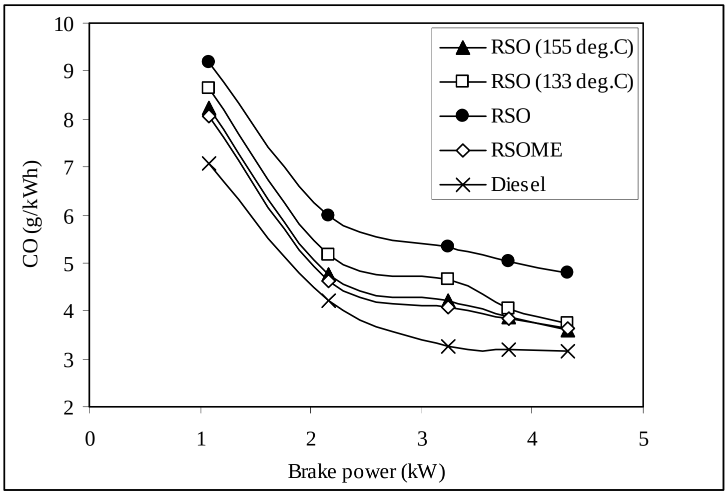

Fig. 7. Variation of carbon monoxide emission with brake power.

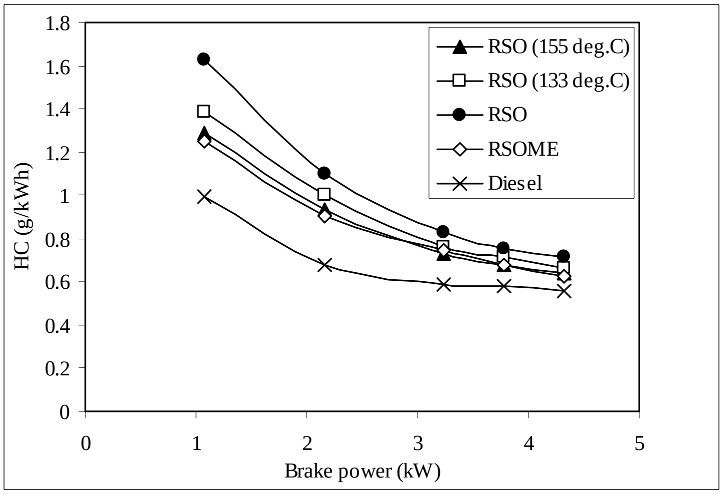

Fig. 8. Variation of hydrocarbon emission with brake power.

Figure 8 shows the variation of HC emissions with different fuels. The HC emission increases marginally with raw RSO. Higher density and viscosity of RSO cause poor mixture formation which results in partially burned hydrocarbons during combustion process. So hydrocarbon emissions are higher for RSO. The HC emissions of the vegetable oil decrease with increase in fuel temperature. The reduction in HC emission at higher fuel temperature is due to better atomization and evaporation. It may be noted that heating leads to a larger cone angle of the spray and helps fuel dispersion. Thus, over rich fuel pockets are avoided.

NO_x variation with brake power is shown in Figure 9. NO_x is generated mostly from the nitrogen present in incoming air to engine and also from the nitrogen present in the fuel due to high combustion temperature. NOx emissions for the RSO operation is very less compared to diesel. This reduction in NO_x with RSO operation is due to less intensity of premixed combustion compared with RSOME and diesel. It can be seen that the NOx emission increases with increase in fuel inlet temperature, but still it is lower by 11% and 14% (RSO at 155°C and 133°C) than that of diesel operation. The increase in NO_x with preheated fuel is due to the rapid burning of RSO with increase in fuel inlet temperature.

The cylinder pressure - crank angle diagram obtained for different fuels at full load is shown in Figure 10. It can be seen that RSO has lower peak pressure than diesel and RSOME. It can also be seen that the maximum peak pressure occurs at 13° CA from TDC with RSO and RSO at 133°C and 12° CA with RSOME and RSO at 155°C. While for diesel the peak occurs at 10° CA from TDC. The shift in peak pressure in the case of RSO and RSO at 133°C and 155°C may be due to the reduction in volumetric efficiency and higher ignition delay (compared with diesel) that lead to increased combustion duration and more late burning of fuel.

The variation of cylinder peak pressure and maximum rate of pressure rise with crank angle for different fuels are shown in Figures 11 and 12. RSO produces a lower peak pressure and maximum rate of pressure rise compared to RSOME and diesel on account of the lower combustion rates. In compression ignition engines the peak pressure depends strongly on the initial combustion rate, which in turn depends on the amount of fuel taking part in the uncontrolled or premixed combustion phase. A marginal variation in peak pressure at different inlet temperatures of the fuel than RSO can be noticed. This is due to the increase in heat release rate (Figure 13) of preheated RSO in the premixed phase than RSO.

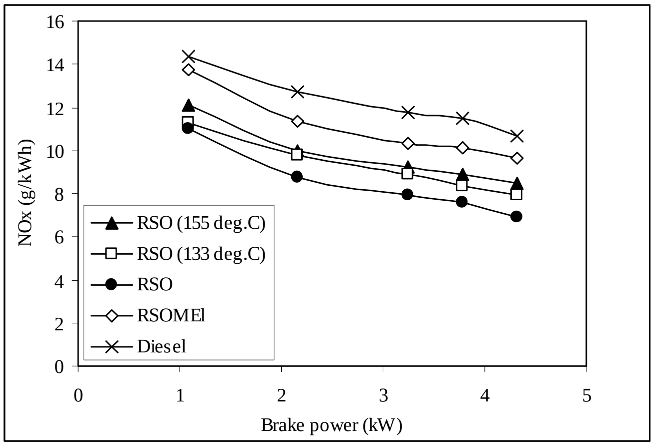

Fig. 9. Variation of nitrogen oxides emission with brake power.

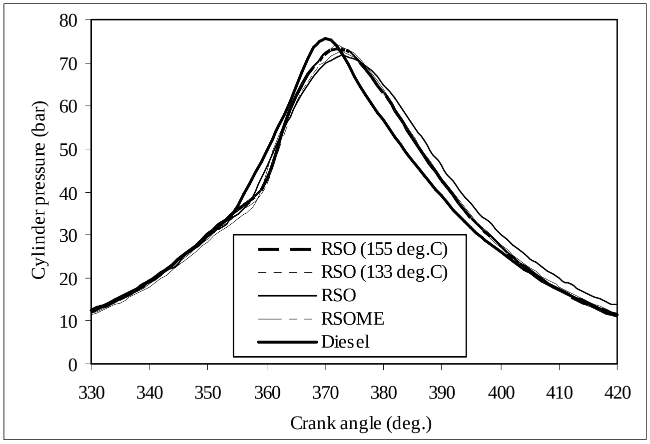

Fig. 10. Variation of cylinder pressure with crank angle.

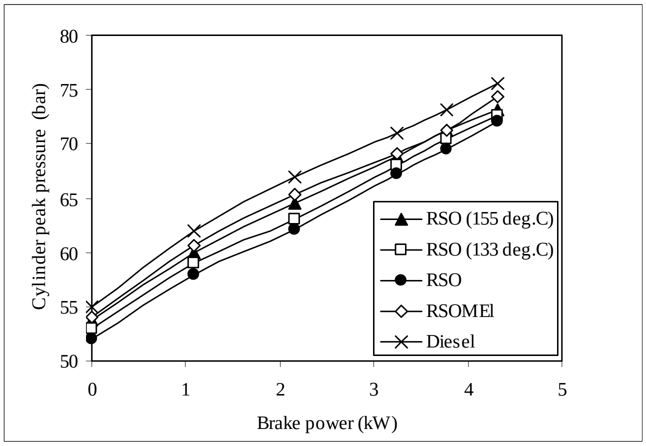

Fig. 11. Variation of cylinder peak pressure with crank angle at full load.

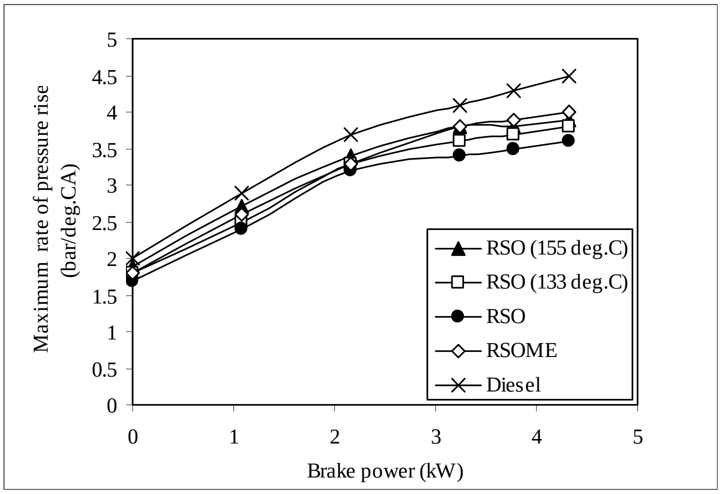

Fig. 12. Variation of maximum rate of pressure rise with crank angle.

The variation of heat release rate with crank angle is shown in Figure 13. It can be observed that the premixed combustion phase is associated with a high heat release rate in diesel. The diffusion combustion phase indicated by the second peak is higher for RSO and preheated RSO compared to RSOME. This is due to high viscosity of fuel spray leading to a reduction in air entrainment and fuel air mixing rates. This leads to less fuel being prepared for rapid combustion with RSO during ignition delay. Therefore more burning occurs in the diffusion phase rather than in the premixed phase with RSO. The significantly higher combustion rates during the later stages with RSO leads to high exhaust temperature and lower thermal efficiency. Preheating of fuel reduces the delay period and hence combustion starts earlier than RSO.

The variation of ignition delay with brake power at various loading conditions is shown in Figure 14. It can be observed that the ignition delay is higher in the case of raw RSO. The increase in ignition delay with raw RSO is due to its high viscosity and poor volatility, which cause slow vaporization and fuel-air mixing. It is also seen that the ignition delay decreases with increase in fuel inlet temperature. The vaporization of fuel is better due to the increase in fuel inlet temperature, which decreases the ignition delay.

Figure 15 shows the variation of combustion duration with brake power for various fuels. Combustion duration generally increases with increase in power output of the engine with all fuels due to increase in the quantity of fuel injected. The combustion duration is higher for raw RSO compared with RSOME and diesel. The increase in combustion duration is due to longer ignition delay followed by longer diffusion combustion phase of RSO. The combustion duration decreases with preheated RSO, which indicates higher heat release rate leading to higher thermal efficiency with preheated RSO.

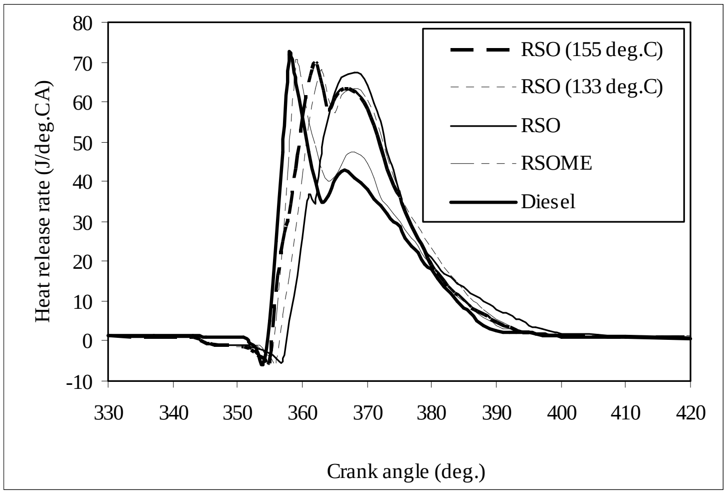

Fig. 13. Variation of heat release rate with crank angle at full load.

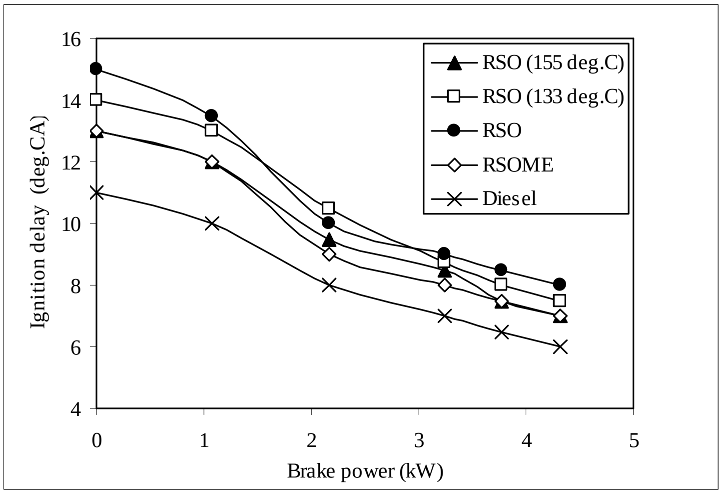

Fig. 14. Variation of ignition delay with brake power.

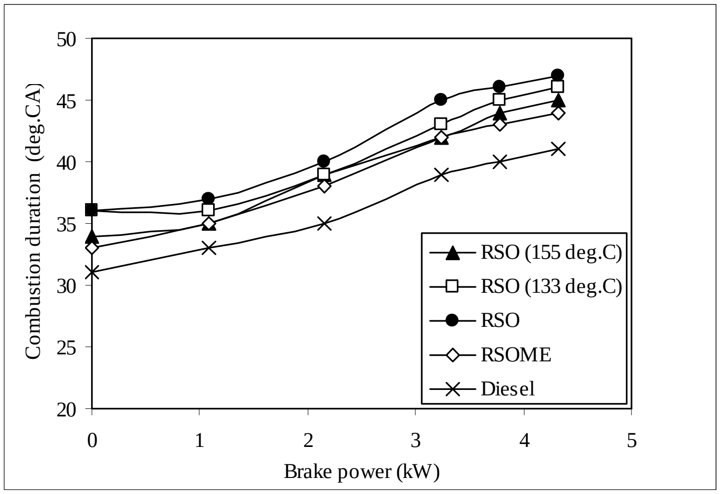

Fig. 15. Variation of combustion duration with brake power.

6. ESTIMATION OF UNCERTAINTY

In measuring any quantity, the results will always differ from the true value even with careful experimentation. This error in measurement may be either random or systematic. By adding a correction value, the systematic error can be removed. Random error can only be estimated statistically and cannot be predicted in advance[14]. Its presence can be detected only when the same quantity is measured again and again under the same conditions. The estimated uncertainty values at typical operating conditions in the present work are:

Speed	:	± 1.5%
Brake power	:	± 1.5%
HC	:	± 3%
CO	:	± 3%
NOx	:	± 4%
Exhaust temperature	:	± 0.6%
Smoke	:	± 3.5%

7. CONCLUSIONS

A single cylinder compression ignition engine was operated successfully on preheated and raw rubber seed oil. The following conclusions are drawn based on the experimental results:

- Brake thermal efficiencies are higher with preheated RSO compared to raw RSO.
- The exhaust gas temperature is higher for RSO compared to RSOME. This is due to the late burning of RSO. It is further reduced with preheated RSO.
- The intensity of premixed combustion phase is lower for RSO compared to RSOME. However, heat release rate improves with preheated RSO.
- Combustion duration and ignition delay are longer with RSO as compared with RSOME at all loads. There is a reduction in combustion duration and ignition delay with preheated RSO.
- A marginal increase in NO_x emissions with preheated rubber seed oil than RSO is observed because of increased fuel temperature that leads to high combustion temperature.
- The increase in fuel temperature reduces the HC emissions. The reduced viscosity of RSO will lead to better atomization resulting in marginal reduction in HC emissions than raw RSO.
- Significant reduction in smoke level with preheated RSO compared to raw RSO is observed. The

improvement in volatility of preheated RSO is beneficial in improving the fuel evaporation resulting in smoke reduction.

On the whole, it is concluded that preheating by exhaust gas is a very effective way to utilize raw rubber seed oil in a diesel engine without any external energy input for heating the fuel in order to reduce the viscosity.

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