Effect of Injection Pressure on Performance, Emission and Combustion Characteristics of Direct Injection Diesel Engine Running on Blends of Pongamia Pinnata Linn Oil (Honge oil) and Diesel Fuel

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ABSTRACT

The growing concerns on the environment and the effect of green house gases have received more and more interests in the use of triglycerides and their derivatives as a substitute of fossil fuel. Few literatures are available on the use of neat methyl esters of honge oil and blend with diesel fuel in diesel engines. However, use of raw honge oil blend with diesel fuel in diesel engines with enhanced injection opening pressure appears to be scarce. This research work presents some findings of the use of honge oil and diesel fuel blend in direct injection diesel engine with increased injection opening pressure (IOP). The performance, emissions and combustion parameters of 20% honge oil and 80% diesel fuel (volume basis) were found very close to neat diesel fuel where as higher blend ratios were found inferior compared to neat diesel fuel. Improved premixed heat release rate were noticed with H30 when the IOP is enhanced. Performance and emissions with H30 are even better than neat diesel fuel at enhanced IOP. With increased injection pressure amount of honge oil in blend can be increased from 20% to 30%.

Key words: Vegetable oil, honge oil, blending, performance, emissions, combustion characteristics, injector opening pressure, India

1. INTRODUCTION

Compression ignition engines enjoy importance among internal combustion engines because of relatively better fuel economy, lower emissions of HC and CO compared to spark ignition engines**.** However, PM and NO_x emissions are high. The fossil fuel reserves are limited. The continuing rise in prices, growing concern on environment from exhaust emissions, global warming and threat of supply instabilities have led to a growing concern for alternative fuel for diesel engines throughout the world, more so in the petroleum importing countries like India. Dependence on foreign sources of energy has been always a bane for any country, particularly for countries like India. It is the single biggest drain on the foreign exchange reserves of the country. An increase of $1 per barrel of crude oil prices adds $425 million to Indian oil import bill (Anonymous, 2005). From the point of view of long term energy security, it is necessary to develop new alternative fuels with properties comparable to petroleum based fuels. Considering the fuel transport and utilization equipment, vegetable oils easily qualify for use in diesel engines as alternative fuels for diesel engines. The diesel fuel can not be replaced totally by the vegetable oils (in conventional diesel engines) because of the low cetane number, high ignition temperature, high viscosity and incomplete combustion leading to increased emissions and decreased performance. Hence, only a partial replacement of diesel fuel is possible in conventional DI diesel engine. Some of these problems are mentioned in the literatures (Agarwal et al., 2001; Altin et al., 2001; Agarwal et al., 2003; Ramadhas et al., 2004).

Some of these problems can be reduced by reducing the viscosity of fuel by four methods: dilution, micro emulsification, pyrolysis and transesterification in order to improve injection performance. Vegetable oil and diesel fuel blending (dilution) is one of the methods to reduce their viscosity (Herche et al., 2001; Silvo et al., 2002; Murat et al., 2007; Nwafor, 2004; Agarwal et al., 2007). The flash point of honge oil is higher than neat diesel; hence it is safe to use it in the engine. Gaseous emissions and performance of DI diesel engines are primarily influenced by combustion processes in turn controlled by fuel properties, injector opening pressure, atomization and also oxygen concentration in the fuel mixture (Heywood, 1998).

Nomenclatures

Brake specific fuel consumption	bsfc	Hydrocarbon	HC
Brake specific energy consumption	bsec	Ignition delay period	IDP
Carbon monoxide	CO	Injection opening pressure	IOP
Combustion duration	CD	Mass fraction burned	MFB
Cylinder pressure	CP	Oxides of sulphur	Sox
Direct injection	DI	Oxides of nitrogen	NOx
Estimated end of combustion	EEOC	Particulate matter	PM
Exhaust gas temperature	EGT	Polycyclic aromatic hydrocarbons	PAH
Fuel line pressure	FP	Static injection timing	SIT
Fuel consumption	FC	Start of combustion	SOC

1.1 Literature on Use of Vegetable Oils in Diesel Engines

Triglycerides (vegetable oils or animal fats for non-edible and non-industrial application), and their derivatives are considered as viable alternatives for diesel fuels. The effect of raw vegetable oils as alternative diesel engine fuels has been reported by many investigators (Agarwal et al., 2001; Altin et al., 2001; Agarwal et al., 2003; Ramadahas et al., 2004). The result showed increase in bsfc and emissions such as CO, HC and smoke opacity compared to neat diesel fuel. This was attributed to the lower heating value, high viscosity, poor atomization, low volatility and polyunsaturated characteristic of neat vegetable oils. Some studies demonstrated that smoke, NO_x lowered and CO, HC increased (Herche et al., 2001). It has been reported that use of vegetable oil results in increased bsfc (He, 2003). Emissions of CO, HC and SO_x were increased, where as NO_x and PM emissions were decreased compared to diesel fuel (Senthil et al., 2003; Huzayyin et al., 2004). Utilization of 100% vegetable oil is also possible and emissions of PAH and PM were reduced (Silvo et al., 2002). The properties of the raw honge oil, diesel fuel and blends of honge oil and diesel fuel have been determined and summarized in Table 1.

Table 1. Properties of the fuel under consideration

Properties	Unit	Method IS:1448	D100	H100	H10	H20	H30	H40	H50
Density at 30^0C	kg/m^3	P:16	830.0	924.0	838.4	848.8	862.2	860.6	871.0
Flash point	^0C	P:69	56	235	63	65	67	67	69
Kinematic viscosity at 40^0C	cSt	P:25	3.12	45.23	6.11	6.66	7.73	10.27	13.58
Kinematic viscosity at 100^0C	cSt	P:25		9.09	1.37	1.83	5.17	3.16	4.39
Calorific Value	kJ/kg	P:6	43000	39440	42689	42288	42388	41506	41165

1.2 Present Work

Many investigators have used rapeseed oil, jojoba oil, soyabean oil, palm oil, coconut oil, cotton seed oil, mahua oil and jatropha oil in diesel engines. Few literatures are available on the use of neat methyl esters of honge oil and blend with diesel fuel in diesel engines. However, a detailed analysis of combustion characteristics and emissions of honge oil blend with diesel fuel in diesel engines appears to be scarce, especially on a scientific level. The objective of this work is to improve the use of vegetable oil with minimal fuel processing and without engine modification. The present research work is aimed to investigate experimentally the effect of injection pressure on performance, combustion characteristics and the exhaust emissions by exploring technical feasibility of honge oil and diesel fuel blends in **a** direct injection compression ignition engine.

2. MATERIALS AND METHODS

2.1 Experimental Set Up

Experiments and tests were conducted on a DI diesel engine. The specifications of the engine under consideration are given in Table 2. The photograph of the engine test rig is shown in Figure 1. Three fuel tanks are used in the present investigation with switch over arrangement, so that supply of fuel can be changed without stopping the engine operation. The engine is started with diesel and data were noted down after attaining steady state. Then the experiment is switched over to blend of diesel fuel and honge oil. Arrangement is made to change IOP without stopping the engine.

2.2 Emission Analyzer

The exhaust gas composition was measured using exhaust gas analyzer and smoke opacity was measured using smoke opacity meter. Make, model and principle of measurement of these emissions are given in Table 3.

Table 2. Engine specifications

Manufacturer	Kirloskar Oil Engines Ltd., India.
Model	TV_SR II, naturally aspirated
Engine	Single cylinder, direct injection diesel engine
Bore / stroke / compression ratio	80 mm / 110 mm / 16.5:1
Speed	1500 rpm, constant
Injection pressure / advance	200 bar / 23^0 before top dead center
Dynamometer	Eddy current (Make: Dynaspeed)
Type of starting	Manually
Air flow measurement	Air box with ‘U’ tube
Exhaust gas temperature	RTD thermocouple
Fuel flow measurement	Burette with digital stop watch
Governor	Mechanical governing (centrifugal type)
Cylinder pressure / fuel line pressure	0-200 bar / 0-2000 bar (fixed 45 mm before the injector)
Resolution	0.1 bar for CP / 1 bar for FP
Type of sensor	Piezo electric (5000 PSI for CP and 10000 PSI for FP
Response time	4 micro seconds
Sampling resolution	1 degree crank angle
Crank angle sensor	360 degree encoder with resolution of 1 degree

Table 3. Make, model and principle of measurement of emissions

Exhaust gas analyzer	Make - MRU, DELTA 1600 S, Germany, O_2 - Electrochemical- Resolution: 0.01% NOx-Electrochemical- Resolution:1 ppm CO - NDIR - Resolution: 0.01% CO_2 - NDIR - Resolution: 0.1% HC - NDIR- Resolution: 1 ppm
Smoke meter	Make - MRU, Optrans 1600, Germany

2.3 Experimental Plan

The engine tests were conducted for the entire load range (0 to 100% i.e., 0 to 5 hp) at constant speed of 1500 rpm and the performance parameters, such as FC, EGT, torque and exhaust gas emissions were measured after attaining the steady state. The cooling water temperature was maintained constant (70 to 75^0C). The fuel test matrix is given in Table 4.

3. RESULTS AND DISCUSSION

The engine performance and emission parameters such as FC, bsfc, bsec, smoke opacity, CO, HC, NO_x and combustion characteristics at standard injector opening pressure are presented in Figures 4 to 13 against percentage by volume of honge oil in honge oil and diesel fuel blend. Brake thermal efficiency, heat release rate and emissions of H30 at higher injector opening pressure are presented in Figures14 to18.

3.1 Effect on Performance Parameters

Fuel consumption for the blend ratios under consideration for the entire load range is given in Figure 2. The FC increased with the increase of engine load as expected. FC increases with increasing amount of honge oil in the fuel blend compared to neat diesel fuel throughout the entire load range. This is possible because of the lower heating value of honge oil as compared to neat diesel fuel.

Table 4. Fuel test matrix under consideration

Trial	Fuel composition
01	Neat diesel fuel (ND or D100) at 200 bar
02	Honge oil - Diesel fuel blends at 200 bar
	The blends were 10-90 (H10), 20-80 (H20), 30-70 (H30),
	40-60 (H40), 50-50(H50) honge oil – diesel fuel volume ratios
03	H30 at 225 bar, 250 bar and 275 bar

Figure 3 shows the variation of bsfc with blend ratios under consideration at different loads. Brake specific fuel consumption of the fuel blends follows the trend similar to the neat diesel. The bsfc decreased with the increase of engine load as expected. Brake specific fuel consumption of blend up to H20 was very close to neat diesel fuel. The reason may be due to higher density even though calorific value is lower compared to petroleum-diesel fuel. Above H20 bsfc was found to increase with increase in honge oil in the blend compared to neat diesel fuel throughout the entire load range. This is attributed to the combined effect of viscosity and lower mass based heating values of blends required larger FC in order to release same energy as that of diesel fuel.

Generally bsfc is not used to compare the two different fuels, because their calorific value, density, chemical and physical parameters are different. Performance parameter, bsec, is used to compare two different fuels by normalizing bsec in terms of the amount of energy released with the given amount of fuel. Brake specific energy consumption of H10 and H20 is almost the same as that of neat diesel fuel as shown in Figure 4. Even though viscosity of H10 and H20 is slightly higher than that of neat diesel, inherent oxygen of the fuel molecules improves the combustion characteristics. This is an indication of relatively more complete combustion (Agarwal et al., 2007; Huzayyin et al., 2004).

Figure 1. Photograph of the engine set up

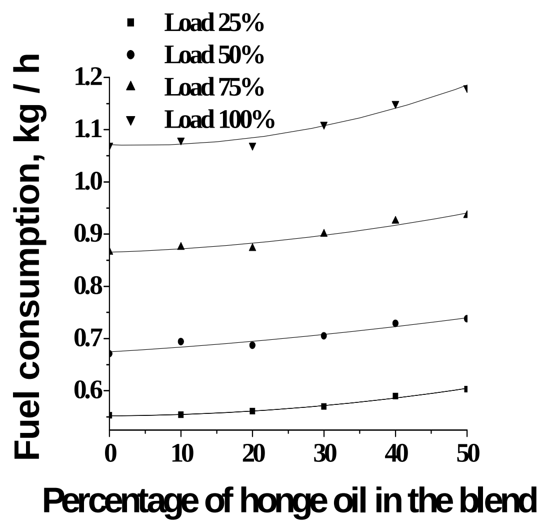

Figure 2. Effect of the volume of honge oil in the fuel blend on FC

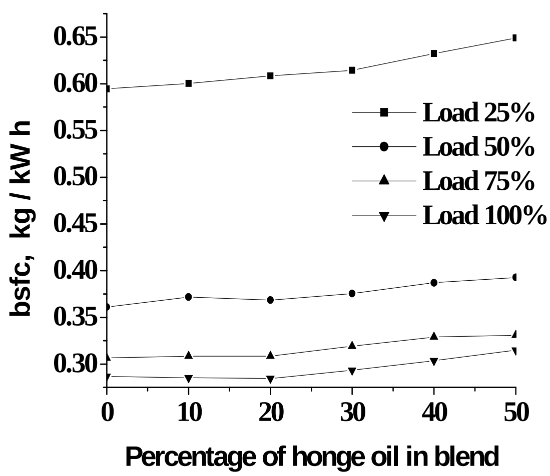

Figure 3. Effect of the volume of honge in the fuel blend on bsfc

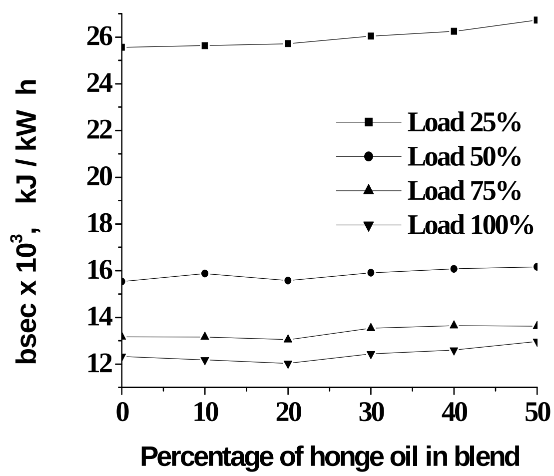

Figure 4. Effect of the volume of honge oil in the fuel blend on bsec

Blends higher than 20% showed higher bsec compared to ND at all engine loads. The reason may be due to higher viscosity, poor volatility and reduction in heating value lead to their poor atomization and combustion characteristics. With blends higher than 20%, viscosity effect, in turn atomization is more predominant than the oxygen availability in the blend leads to lower volatile characteristics and affects combustion process**.**

3.2 Effect on Emission Parameters

Diesel combustion is heterogeneous in nature. Smoke formation mainly takes place in the rich zone of fuel at high temperature and pressure, specifically within the core region of each fuel spray. The smoke opacity emissions increased with the increase of the engine load as expected and this is shown in Figure 5. Smoke opacity emissions first decrease up to H20 and then increase with further addition of honge oil in the blend compared to neat diesel fuel. The reasons for smoke opacity decreasing up to 20% of the honge oil in the blend are: (i) dilution of aromatics and (ii) presence of oxygen molecules in the honge oil. Aromatics are known to contribute to soot formation, while the inherent oxygen molecule in the fuel, which helps to promote combustion processes by delivering oxygen to the pyrolisis zone of the burning fuel by reducing locally over rich areas and limit primary smoke formation, results in lower smoke emissions. These results agree with the results of Agarwal et al., 2007 and Herchel et al., 2001**,** but the degree of reduction is not same.

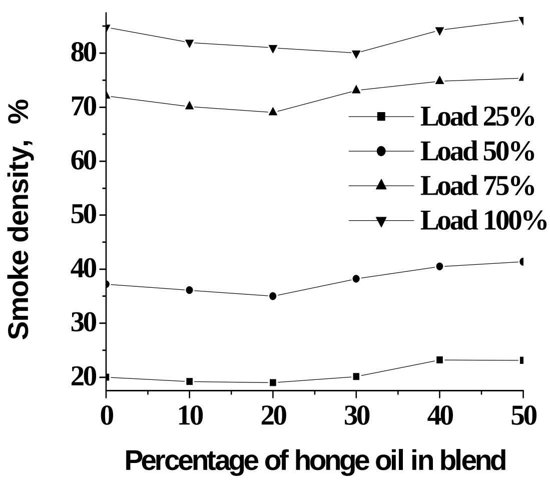

Figure 5. Effect of the volume of honge oil in the fuel blend on smoke density emissions

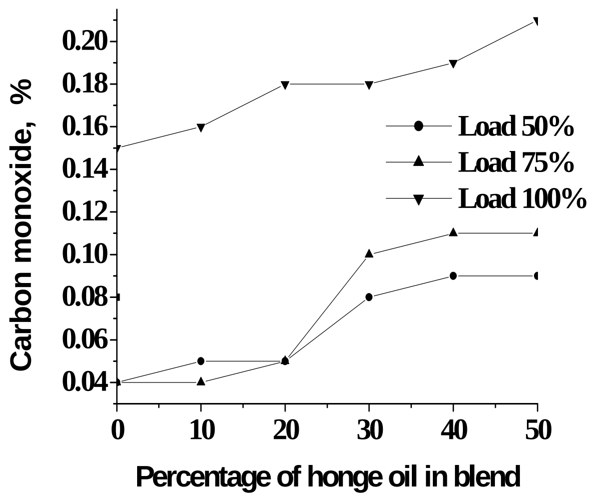

Figure 6. Effect of the volume of honge oil in the fuel blend on CO emissions

Higher smoke emissions above H20 may be due to poor atomization. Higher viscosity and bigger size fuel molecules result in poor atomization of fuel blends. In the present investigation a maximum of 6% reduction in smoke opacity emissions is observed. Smoke opacity decreased throughout the entire load range for the entire blend ratio and a maximum of 36% reduction is observed (Herchel et al., 2001).

CO formation depends upon mixture strength i.e., oxygen quantity and fuel viscosity, in turn atomization. At low load range (up to 50% of the maximum load) air fuel ratio is high, availability of oxygen is more, and hence formation of CO emissions is also low. Figure 6 shows the CO emissions of the fuel blends under consideration at different engine load. Up to H20 formation of CO is close to that of neat diesel fuel. This is due to the presence of vegetable oil i.e., availability of inherent oxygen in the fuel molecules**.**

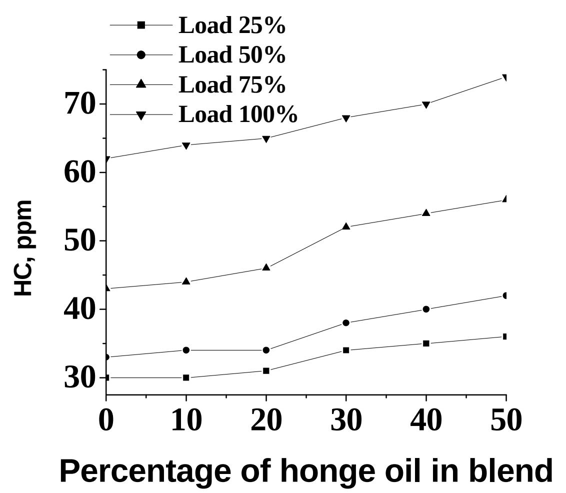

Figure 7. Effect of the volume of honge oil in the fuel blend on HC emissions

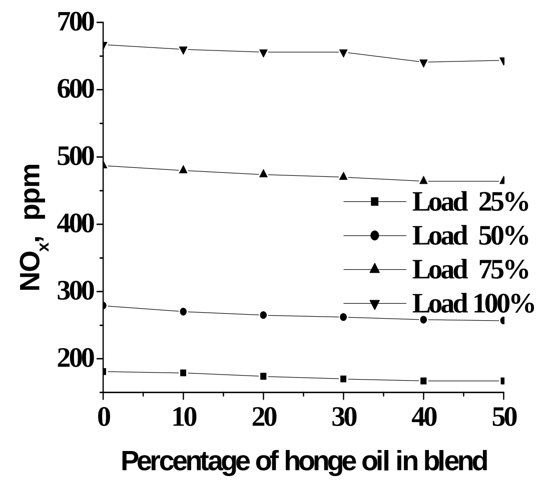

Figure 8. Effect of the volume of honge oil in the fuel blend on NO_x emissions

Blends higher than 20% showed higher CO emissions compared to ND over the entire load range (CO formation for 25% load not shown), and are significant. The results are similar to that in reference (Agarwal et al., 2007). Lower heating value of blends leads to higher quantity of fuel injection (Figure 2) as compared to neat diesel for the same load conditions. Viscosity effect, in turn atomization, is more predominant than the oxygen availability. CO emissions increase with load as well as increasing amount of vegetable oil in the fuel blend (Herchel et al., 2001; Senthil et al., 2003). The results (Wang et al., 2005) show the entirely opposite trend. Entirely opposite trend was obtained (Wang et al., 2006) compared to the present result, that is CO emissions from the vegetable oil and its blends are lower than that of the diesel fuel at the engine full load, but in case of lower engine loads, the CO emissions are all slightly higher. The present results agree with the results at higher load range (Silvo et al., 2002).

Figure 7 shows the comparison of the HC emissions over the entire load range for all the fuel blends. Over the entire load range, HC emissions in case of H10 and H20 were close to ND, thereafter (blends higher than 20%) HC emissions increased with increasing amount of honge oil in fuel blends compared to ND. This is due to relatively less oxygen available for the reaction when more fuel is injected in to the engine cylinder at higher engine load. Similar observations on HC emissions have been reported (Herchel et al., 2001; Silvo et al., 2002; Agarwal et al., 2007).

The increase in emissions (CO, smoke opacity and HC) for H30 to H50 could be due to relatively incomplete combustion. This is also a reason for increase in bsec (Figure 4) at higher load and higher percentage of honge oil in the fuel. Otherwise bsec would be a straight line. Soot, CO and HC compete for the available oxygen in the rich combustion zone.

Figure 8 shows the NO_x emissions for all the blend ratios under consideration. The important factor that causes NO_x formation is due to high combustion temperatures and availability of oxygen. The NO_x emissions increased as the engine load increased, due to the increase in combustion temperature. It is seen that within the range of tests, the NO_x emissions of H10 - H50 were lower than that of diesel fuel. Similar results are found in references (Herchel et al., 2001; Silvo et al., 2002 Wang et al., 2005; Rakopoulos et al., 2006). The reason may be due to (i) smaller calorific value of the blends (ii) lower localized gas temperature in the cylinder, (iii) oxidation rate, (iv) high viscosity leads to poor atomization.

3.3 Combustion Characteristics

The effect of crank angle on cylinder pressure and rate of pressure rise at 75% load are shown in Figures 9 and 10. H20 follows the trend similar to the neat diesel pressure diagram and rate of pressure rise diagram. Same trends obtained for other blend ratios of the entire load under consideration. H20 results in lower peak pressure and lower maximum rate of pressure rise as compared to neat diesel. The occurrence of peak pressure and maximum rate of pressure rises of H20 moved away compared to neat diesel. In a CI engine, the peak pressure depends on the combustion rate in the initial stages, which is influenced by the amount of fuel taking part in the uncontrolled combustion phase, which in turn governed by the delay period (Heywood, 1998). Thus, the slight higher viscosity and poor volatility of the H20 results in lower peak pressure and maximum rate of pressure rise as compared to neat diesel.

The effect of crank angle on second derivative of the rate of cylinder pressure rise at 75% load is shown in Figure 11. The IDP is calculated based on the SIT. The duration of the point of SIT to the point of SOC is taken as the IDP. According to ASTM D613, the SOC is defined as the point at which the second derivative of the combustion chamber pressure versus time becomes positive. This also shows a sudden rise in the slope at the point of ignition due to the high premixed heat release rate. IDP of H20 is slightly higher as compared to neat diesel due to low cetane number and higher self ignition temperature of the H20 (even though it is not measured, self ignition temperature and cetane number is higher and lower respectively). The reason may be due to the inferior atomization and vaporization, physical delay of H20 becoming larger as compared to neat diesel fuel.

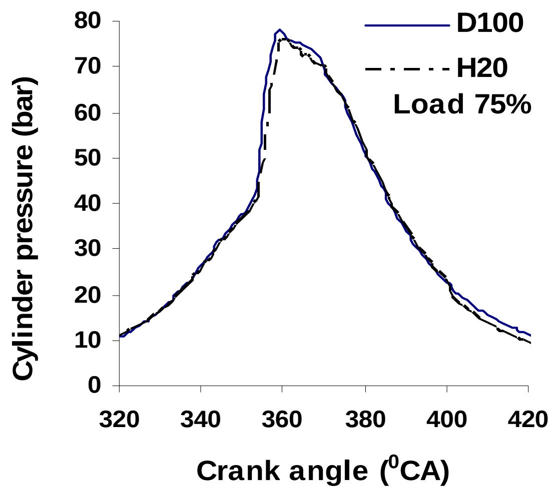

Figure 9. Effect of the volume of honge oil in the fuel blend on cylinder pressure

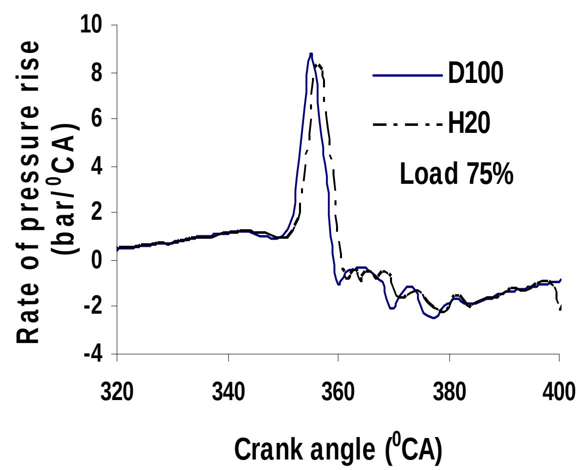

Figure 10. Effect of the volume of honge oil in the fuel blend on rate of pressure rise

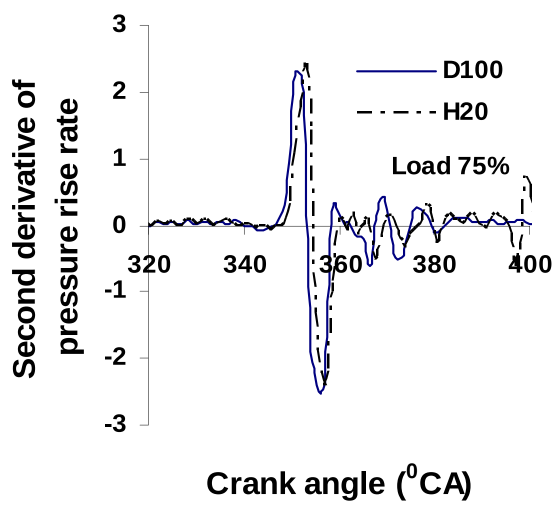

Figure 11.Effect of the volume of honge oil in the fuel blend on second derivative of pressure rise

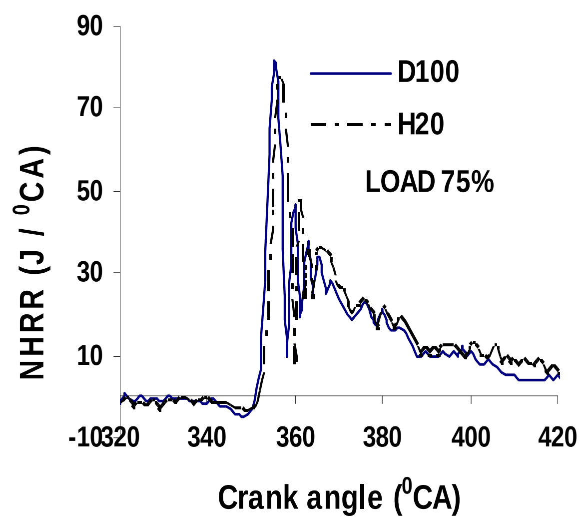

Figure **12.** Effect of the volume of honge oil in the fuel blend on net heat release rate

Figure 12 shows the comparison of the net heat release rate curves for neat diesel and H20 for 75% load. It is clear that H20 changes the combustion pattern that is peak of the heat release rate curves for H20 is rather lower than that of neat diesel fuel. It is seen that the premixed combustion region is lesser for H20 indicating that with neat diesel fuel greater mixing is enhanced. The slightly high viscosity and density of H20 result in inferior atomization and vaporization and lead to reduction in fuel air mixing rates. Hence, more burning occurs in the diffusion phase (Figure 13). This could explain the lower NO_x, higher CO and HC emissions with H20. However, this does not explain the decrease in smoke as shown in Figure 5. Table 5 shows the combustion duration, peak pressure, occurrence of peak pressure, maximum rate of pressure rise, occurrence of maximum rate of pressure rise, maximum net heat release rate, occurrence of maximum net heat release rate, SIT, SOC, IDP and estimated end of combustion.

The duration of SOC to the point of EEOC is taken as the CD. Higher CD was obtained with H20 compared to neat diesel fuel. This is due to the injection of more amount of H20 than diesel fuel as shown in Fig. 2. Also due to the more diffusion burning with H20, more amount of H20 burns in the expansion stroke resulting in longer CD. This also indicates that with H20 the crank angle at which MFB attains maximum is longer than diesel fuel. This is shown in Fig. 13.

Table 5. Important combustion parameters and their occurrence at 75% load

Combustion parameters	Units	Combustion parameters			
		D100	H20	H30	H30
Injector opening pressure	Bar	200	200	200	225
Peak pressure	Bar	78.3	75.7	77.4	78.9
Occurrence of peak pressure	^{0}CA	359	361	362	359
Maximum pressure rise rate	bar/^{0}CA	8.8	8.2	7.45	7.55
Occurrence of maximum pressure rise rate	^{0}CA	355	356	355	355
Maximum net heat release rate	J/ ^{0}CA	81.88	77.12	77.5	78.38
Occurrence of maximum net heat release rate	^{0}CA	355	356	357	356
Static injection timing	^{0}CA	337	337	337	337
Start of combustion	^{0}CA	350	352	353	352
Ignition delay period	^{0}CA	13	15	16	15
Estimated end of combustion	^{0}CA	394	398	400	400
Combustion duration	^{0}CA	44	46	47	48

3.4 Performance, Emission and Combustion Characteristics of H30 at Different IOPs

Brake thermal efficiency of H30 for the entire load range at different IOPs is shown in Fig. 14. This clearly indicates that a maximum brake thermal efficiency is obtained corresponding to 225 bar and it is slightly higher compared to neat diesel fuel at 200 bar. This is due to improved spray characteristics, better atomization and mixing with air is good. This will enhance combustion process and in turn improve efficiency. Too high IOP leads to smallest diameter of fuel droplet and affects the spray pattern and penetration. Brake thermal efficiency of H30 at 225 bar and 250 bar is almost the same. Better combustion i.e., high heat release rates were obtained in case of H30 at IOP 225 bar compared to diesel fuel and H30 at IOP 200 bar as shown in Fig. 15. Figure 16 shows that smoke opacity emissions steadily decreased with increase in the IOP due to better atomized spray leading to improved mixture formation. Lowest smoke opacity is obtained with the IOP of 275 bar. Better smoke emissions were seen for all the IOPs under consideration compared to neat diesel fuel at standard IOP of 200 bar. Variation of the CO and HC emissions at different IOPs are given in Figures 17 and 18. Least CO and HC emissions were observed at 275 bar and 250 bar respectively and this is better than neat diesel fuel at standard IOP of 200 bar. This is due to improvement in the spray pattern improving atomization and leading to a lower physical delay in turn to a lower IDP. This will enhance performance and emissions with H30. Figure 19 shows the NO_x emissions at different IOPs under considerations. The NO_x emissions increased as the IOP increased due to improved combustion in turn lead to higher temperature.

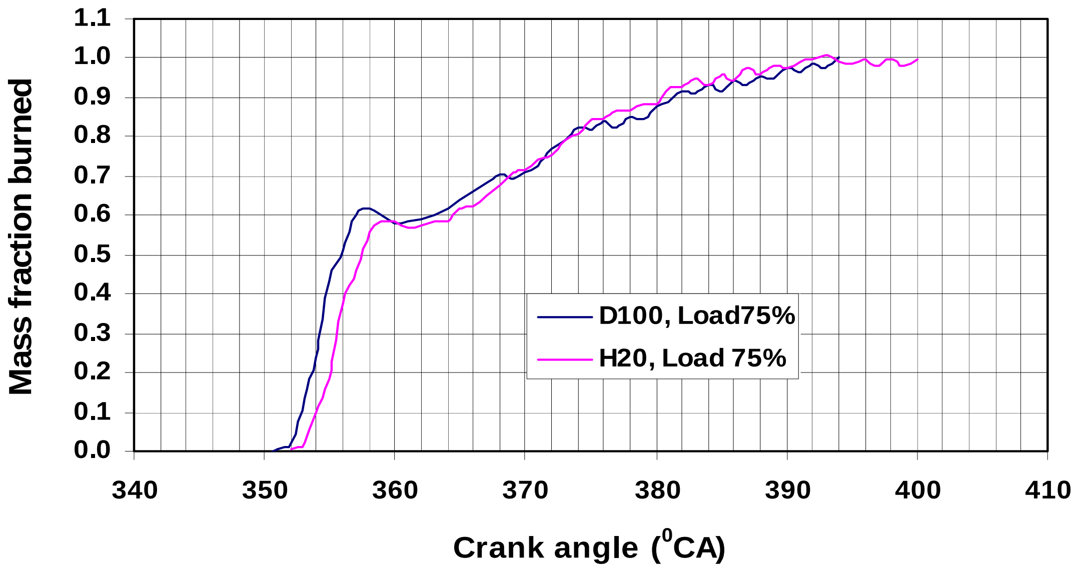

Figure 13. Effect of crank angle on MFB for D100 and H20 at load 75%

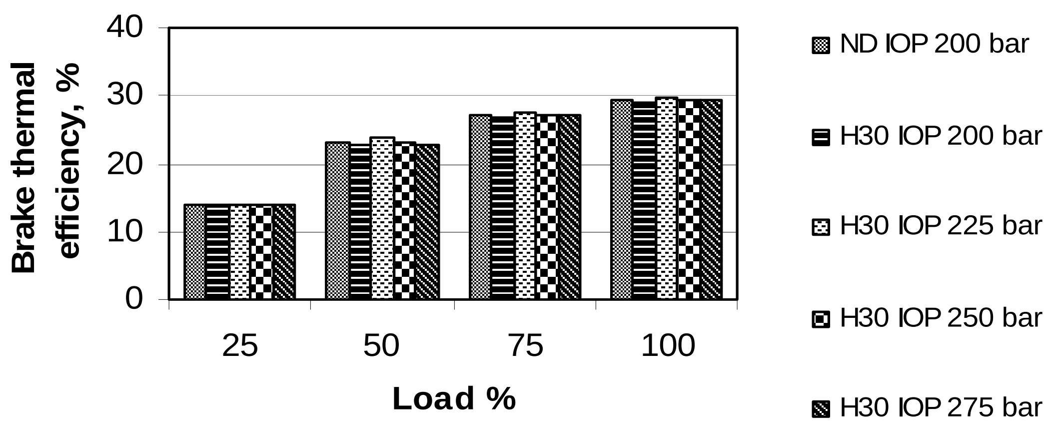

Figure 14. Effect of load on brake thermal efficiency at different IOPs for ND and H30

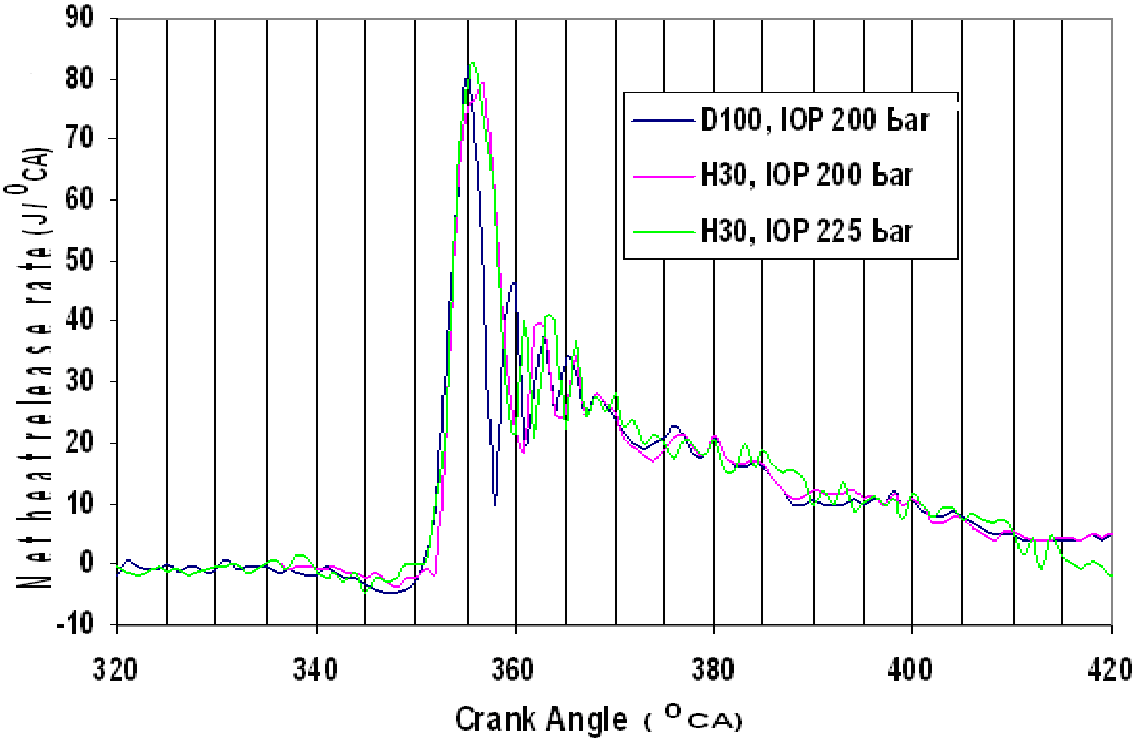

Figure 15. Effect of crank angle on net heat release rate
at different IOPs for ND and H30 at 75% load

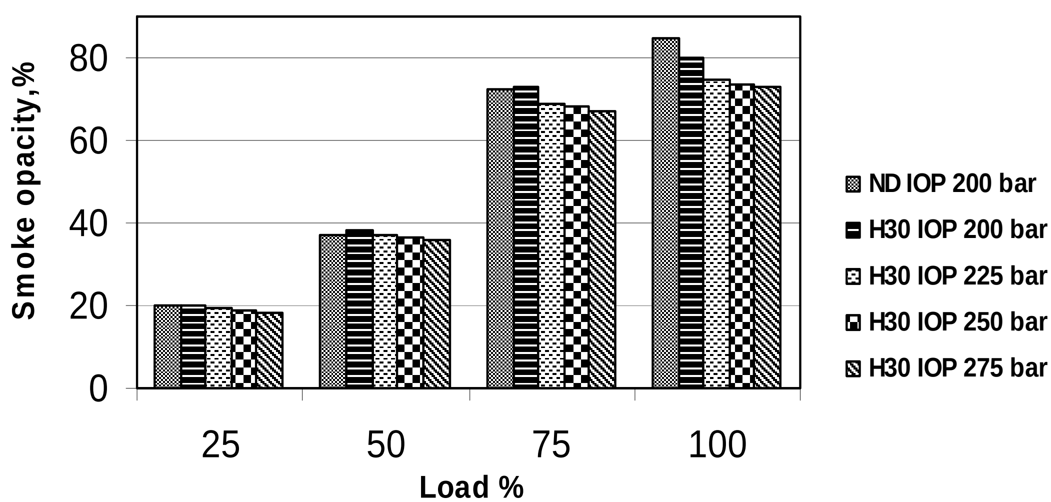

Figure 16.Effect of load on smoke opacity at different IOPs for ND and H30

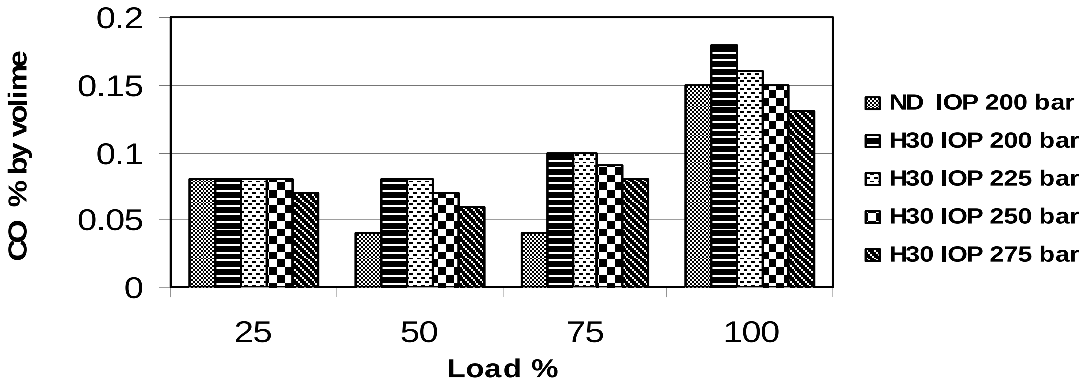

Figure 17. Effect of load on CO at different IOPs for ND and H30

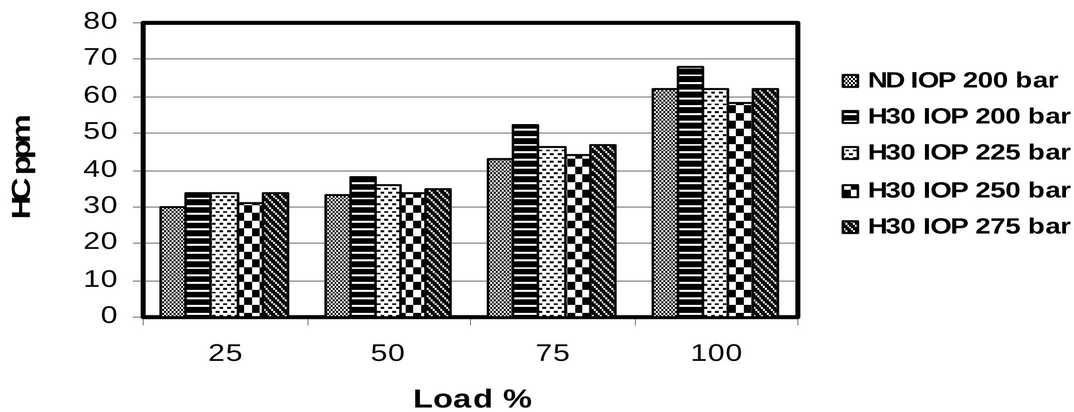

Figure 18. Effect of load on HC at different IOPs for ND and H30

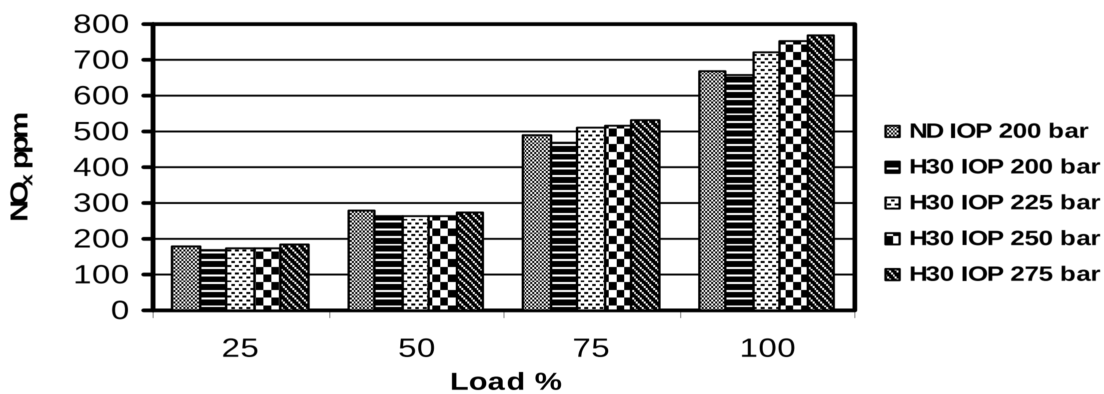

Figure 19. Effect of Load on NO_x at different IOPs for ND and H30

4. CONCLUSIONS

Based on the experimental results of this work over the range of honge oil percentage in fuel blend the following conclusions are drawn.

1. No problem was faced at the time of starting the DI diesel engine and the engine ran smoothly over the range of honge oil percentage in fuel blend.
2. FC rate increased with increase of honge oil in the blend.
3. bsfc of H20 is close to that of diesel fuel. Thereafter increased compared to diesel fuel.
4. bsec of H20 is close to that of diesel fuel. Thereafter increased compared to diesel fuel.
5. Emission parameters such as smoke opacity, CO, HC and NO_x up to H20 were found to be close to that of diesel fuel. Thereafter increased compared to diesel fuel.
6. H20 exhibits; slightly higher IDP, slightly lower maximum rate of pressure rise, slightly lower peak pressure, slightly lower maximum net heat release rate, slightly higher CD compared to neat diesel fuel.
7. Performance, emission and combustion parameters of H20 are almost similar to that of neat diesel fuel.
8. The performance and emission characteristics of H30 to H50 are inferior compared to neat diesel fuel.
9. Improved premixed heat release rate were noticed with H30 when IOP is enhanced.
10. Performance and emissions of H30 were improved at IOP 225 bars.
11. This work establishes that 20% honge oil in the fuel blend can be used in direct injection diesel engine without any modification and without any adverse effect.
12. This work also establishes that 30% honge oil in the fuel blend can be used in direct injection diesel engine by increasing injection pressure from 200 bar to 225 bar

5. ACKNOWLEDGMENTS

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