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Comparison of Jatropha Biodiesel with Pure Diesel an Approach for Alternative and Environmental Friendly Energy Resource

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ABSTRACT

Energy is a valuable asset to our economy. Ever since there has been growing demand for energy consumption during the last two decades. Mostly the energy consumption is based on the fossil fuels like coal, oil, natural gas, petroleum, etc. These are conventional sources of energy which are no longer expected to cater the needs of the economy as they are already in a depleting state. Hence, the exploration for renewable sources of energy has become a striking phenomenon in the recent years. Wind energy is an example, but it is not so economical since it is site-specific. Also, the hydroelectric power generation is considerable only where construction of dams are possible without having any effects on the environment such as flooding of rivers. We also know that the oceans are a rich source of energy (Tidal energy), but one cannot utilize the energy possessed by it since the procedure has not been fully developed. Hence, the only alternative left to the common masses are bio-fuels. various blends of Jatropha biodiesel were tested and its properties were compared with that of 100% diesel which was fond to be more environmental friendly than pure diesel.

KEY WORDS: Biodisel, Jatropha oil, non conventional energy.

1. INTRODUCTION

Biodiesel refers to a vegetable oil- or animal fat-based diesel fuel consisting of long chain alkyl (methyl, propyl or ethyl) esters. Biodiesel is typically made by chemically reacting lipids (e.g., vegetable oil, animal fat (tallow)) with an alcohol. Biodiesel is meant to be used in standard diesel engines and is thus distinct from the vegetable and waste oils used to fuel *converted* diesel engines. Biodiesel can be used alone, or blended with petro-diesel.Biodiesel can be used in pure form (B100) or may be blended with petroleum diesel any concentration in most injection pump diesel engines. New extreme high pressure (29,000 psi) common rail engines have strict factory limits of B5 or B20 depending on manufacturer. Biodiesel has virtually no sulphur content, and it is often used as an additive to Ultra-Low Sulfur Diesel (ULSD) fuel. Reddy and Ramesh (2006) studied the effect of air swirls on the performance, emissions and combustion with Jatropha oil, which included a detailed analytical determination of NO emissions and smoke level. Purpose of this work was to achieve the best brake thermal efficiency using biodiesel, but was still lower than the diesel values.Sahoo and Das (2009) made a detailed observation of the fuel properties (like Flashpoint, density, calorific value, viscosity) of Jatropha, Karanja and Polanga based biodiesel and their blends with comparison to a diesel. A striking feature of their study is that the comparison of these properties with diesel shows that the methyl esters of Jatropha, Karanja and Polanga oil have relatively closer fuel property values to that of diesel. Hence, no hardware modifications are required for handling these fuels (biodiesel and their blends) in the existing engine. Xingcai Lu, Junjun Ma, Libin Ji and Zhen Huang (2007) proposed the introduction of premixed DMM (Dimethoxymethane) on the base of neat biodiesel .It is very interesting to find that at PI (premixed ratio) values from 15% to 20%, both the NO and smoke emissions decreases substantially. Also, hydrocarbon emissions and CO (Carbon monoxide emissions) begin to decrease remarkably with the port injection of DMM (Dimethoxymethane) fuel. Hamasaki et al (2004) studied the combustion characteristics of a turbocharged direct injection automotive diesel engine fuelled by waste vegetable oil biodiesel and conventional diesel fuel, reporting slight differences in the shape of heat release rate curves and in the main combustion parameters. Similar results have been obtained by other researchers testing biodiesel from several vegetable oils and animal fats. The development of similar stages in the combustion process using biodiesel and diesel fuel resulted in small differences on thermal efficiency, and so the lower energy content of biodiesel led to an increase in specific fuel consumption. Ryan et al. (1984) characterized injection and combustion properties of several vegetable oils. The atomization and injection characteristics of vegetable oils were significantly different from that of diesel fuel due to the higher viscosity of the vegetable oils. Engine performance tests showed that power output slightly decreased when using vegetable oil fuel blends. Injector coking and lubricating oil contamination appeared to be a more dominate problem for oil-based fuels having higher viscosities. Pramanik K.(2002) presented the effect of Brake Horse Power (BHP) on exhaust gas temperature with variation in the load in the range of 0-3.74 BKW for diesel, jatropha curcas and various blends. The exhaust temperature with the blends having higher percentage of jatropha curcas oil was found to be higher at the entire load in comparison to diesel oil, but the deviation was observed to be greater at higher BKW, particularly above 2.32. The higher exhaust temperature with blends containing above 50%v of jatropha oil is indicative of lower thermal efficiencies of the engine. At lower thermal efficiency, less of the energy input in the fuel is converted to work, thereby increasing exhaust temperature. The real merit lies in the fact that the properties of the blends may be further improved to make use of higher percentage of jatropha oil in the blend using jatropha oil of purer grade which may be obtained by pretreatment of the oil. Moreover, the long term durability of the engine using bio-diesel as fuel requires further study. Ravi et al. conducted an extensive study on the blending of methanol with jatropha oil fuelled diesel combustion engine. The methanol was carburetted with different jet openings and observed that the rates of pressure rise and peak pressures were reduced considerably.The findings of Wirawan et al. (2008) and Knothe et al. (2004) suggested that the engine power rating exhibited by biodiesel blends is no doubt affected by lower viscosity profile, higher fuel injection and combustion of the tested pure bio-diesel. On the other hand, high fuel viscosity reduces fuel injection efficiency and atomization, and this lead to poor fuel combustion and power losses in engine.

2.EXPERIMENTAL SET-UP

The Experiment included preliminary testing of n-hexane with different blends of IOCL grade diesel (100%pure) and NRL diesel(100%pure) where performances were studied, being carried out on a rope brake dynamometer. The second experimental setup was carried out for performance evaluation of the compression ignition engine using an eddy current dynamometer. Jatropha bio-diesel was tested along with its blends. Pure diesel was used as the pilot fuel. Care was taken to ensure that no technical snag should be developed during its testing and evaluation.

2.1 Experimental arrangement

The setup consists of single cylinder, four stroke Diesel engine connected to eddy current type dynamometer for loading. Provision is also made for interfacing airflow, fuel flow, temperatures and load measurement. The set up has stand-alone panel box consisting of air box, two fuel tanks for duel fuel test, manometer, fuel measuring unit, transmitters for air and fuel flow measurements, process indicator and engine indicator. Rota meters are provided for cooling water and calorimeter water flow measurement. Mechanical efficiency, volumetric efficiency, specific fuel consumption, A/F ratio and heat balance. Lab view based Engine Performance Analysis software package “Engine soft LV” is provided for on line performance evaluation.

3.DATA COLLECTION

3.1Characteristic properties of NRL-High Speed Diesel

Sl. No.	CHARACTERISTIC	REQUIREMENTS
1	Ash, percent by mass, Max	0.01
2	Carbon residue (in %), Max	0.30
3	Cetane number, Min	51
4	Cetane index, Min	46
5	Pour point, Max(in °C) a) Winter b) Summer	 3 15
6	Distillation, percent v/v, recovered at 360ºC, Min	95
7	Flash Point: a) Abel, °C, Min b) Pensky Martin closed cup, °C, Min	 35 66
8	Kinematic viscosity cSt, at 40ºC	2.0 to 4.5
9	Sediment, percent by mass,Max	Nil
10	Total contamination, mg/kg. Max	24
11	Specific Gravity	8400
12	Total sulphur , mg/kg, Max	350
13	Water content, mg/kg, Max	200
14	Oxidation stability, g/m^3, Max	25
15	Polycyclic Aromatic Hydrocarbon (PAH) percent by mass, Max	11
16	Lubricity corrected wear scar diameter (wsd 1.4) at 60ºC, microns, Max	460
17	Oxygen content, percent by mass, Max	0.6
18	Calorific value, MJ/kg	47.46

3.2 Characteristic properties of IOCL (High Speed Diesel)

01	Ash, percent by mass, Max	0.01
02	Carbon residue (Rams bottom) on 10 percent residue1), percent by mass, Max	0.30
03	Cetane number, Min	51
04	Cetane index, Min	46
05	Pour point3), Max: a) Winter b) Summer	 3°C 15°C
06	Distillation, percent v/v, recovered at 360°C, Min	95
07	Flash point: a) Abel, °C, Min b) Pensky Martens closed cup, °C, Min	 35 66
08	Kinematic viscosity, cSt, at 40°C	2.0 to 4.5
09	Sediment, percent by mass, Max	–
10	Total contamination, mg/kg, Max	24
11	Specific gravity	8371
12	Total sulphur , mg/kg, Max	50
13	Water content,mg/kg, Max	200
14	Oxidation stability, g/m3, Max	25
15	Polycyclic aromatic hydrocarbon ,percentage by mass,Max	11
16	Lubricity corrected wear scar diameter (wsd 1.4) at 60°C, microns, Max	460
17	Oxygen content percentage by mass,Max	0.6
18	Calorific value, kJ/kg	45733.8

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3.3Performance analysis of pure NRL diesel:

Sl no.	N(rpm)	Brake load (in %)	BThE (in %)	VolE (in %)	Te (in °C)	Wsfc (in kg/kW-s)	A/F ratio
1	1530	56.20	19.09	98.02	132	0.818	7:1
2	1500	60..40	16.04	99.26	156	0.974	6.57:1
3	1498	77	15.09	97.20	193	0.976	5.43:1
4	1528	62.50	17.96	96.32	176	0.870	6.25:1
5	1502	78.30	19.28	98.71	181	0.810	5.68:1

Performance of pure IOCL diesel:

Sl. No.	N(rpm)	Brake Load (in %)	BThE (in %)	VolE (in %)	Te (in °C)	Wsfc (in kg/kW-s)	A/F ratio
1	1450	45.80	5.02	89.80	233	0.4348	4.8:1
2	1430	50	11.24	89.71	252	0.195	5.46:1
3	1448	54	4.83	89.83	264	0.452	4.64:1
4	1428	66.60	13.14	89.72	293	0.166	4.48:1
5	1413	83.33	5.04	89.59	322	0.4378	3.69:1

Performance analysis of 5% hexane-NRL:

Sl no.	N(rpm)	Brake load (In%)	BThE (In %)	VolE (In %)	Wsfc (In kg/KWs)	Te (In ◦c)	A/F ratio
1	1524	37.5	13.04	99.14	1.197	194	5:1
2	1546	54.16	38.15	97.05	4.097	234	4.807:1
3	1500	66.66	47.06	99.27	3.321	246	4.629:1

Performance analysis of 10% NRL HSD-Hexane blend:-

Sl.No	N(rpm)	Brake load(In%)	η_{bth} (In %)	η_v (In %)	W_{sfc} (In kg/KWs)	T_e (In ◦C)	A/F ratio
1	1524	37.5	12.54	97.7	1.246	264	4.807:1
2	1542	54.16	39.61	97.28	3.9458	262	5:1
3	1509	66.66	52.66	99.07	2.9682	255	5.208:1

Performance analysis of 15% NRL HSD-Hexane blend:-

Sl.No	N(rpm)	Brake load(In%)	η_{bth} (In %)	η_{v} (In %)	W_{sfc} (In kg/KWs)	T_{e} (In ◦C)	A/F ratio
1	1538	37.5	12.65	97.20	1.253	252	4.807:1
2	1553	54.16	39.89	95.93	3.917	257	5:1
3	1500	66.66	46.81	98.59	3.321	263	4.629:1

Performance analysis of 5% IOCL HSD-Hexane blend:-

Sl.No	N(rpm)	Brake load(In%)	η_{bth} (In %)	η_{v} (In %)	W_{sfc} (In kg/KWs)	T_{e} (In ◦C)	A/F ratio
1	1522	37.5	17.49	99.61	1.25	220	4.807:1
2	1518	54.16	48.89	96.72	4.472	271	4.481:1
3	1500	66.66	70.99	96.80	3.08	308	5.019:1

Performance analysis of 10% IOCL HSD-Hexane blend:-

Sl.No	N(rpm)	Brake load(In%)	η_{bth} (In %)	η_{v} (In %)	W_{sfc} (In kg/KWs)	T_{e} (In ◦C)	A/F ratio
1	1518	37.5	19.81	97.38	1.103	304	5.453:1
2	1565	54.16	50.40	93.82	4.338	290	4.353:1
3	1515	66.66	66.37	97.97	3.294	295	4.513:1

Performance analysis of 15% IOCL HSD-Hexane blend:-

Sl.No	N(rpm)	Brake load(In%)	η_{bth} (In %)	η_{v} (In %)	W_{sfc} (In kg/KWs)	T_{e} (In ◦C)	A/F ratio
1	1514	37.5	17.48	98.04	1.25	265	4.678:1
2	1528	54.16	49.20	96.29	4.443	246	4.353:1
3	1506	66.66	65.92	98.62	3.316	252	4.513:1

Properties of NRL diesel and 20% blend used:

Calorific value: 43.2 MJ/kg(20% blend) and 47.86 MJ/kg(pure NRL)

Specific gravity:0.84(pure) and 0.875(blended).

Observation Data:

Speed rpm	Load kg	Comp Ratio	T1 deg C	T2 deg C	T3 deg C	T4 deg C
1586.00	3.09	18.00	26.09	34.28	26.09	34.82

1568.00	6.08	18.00	26.42	35.80	26.42	38.14
1552.00	9.10	18.00	27.13	37.29	27.13	42.57
1532.00	12.02	18.00	27.23	38.96	27.23	46.90

Observation Data:

T5 deg C	T6 deg C	Air mmWC	Fuel cc/min	WFlow Eng lph	WFlow Cal lph
200.63	139.07	87.05	14.62	300.00	70.00
243.37	165.12	84.16	17.05	300.00	70.00
288.94	190.18	79.82	21.17	300.00	70.00
356.21	222.40	81.98	23.94	300.00	70.00

Result Data:

Torque Nm	BP Kw	FP Kw	IP Kw	BMEP bar	IMEP bar	BTHE %	ITHE %	MechE ff %
5.60	0.93	3.66	4.59	1.06	5.27	10.95	54.01	20.28
11.03	1.81	3.49	5.30	2.09	6.01	18.28	53.47	34.19
16.52	2.69	3.44	6.13	3.14	7.15	21.83	49.82	43.82
21.81	3.50	3.33	6.83	4.14	7.91	25.16	49.07	51.26

Result Data:

Air Flow kg/hr	Fuel Flow kg/hr	SFC Kg/Kw -Hr	VOLE %	A/F	HBP %	HGas %	HJW %	RAD %
30.28	0.74	0.79	82.52	41.20	10.95	21.76	33.62	33.67
29.78	0.86	0.47	82.07	34.74	18.28	23.02	33.03	25.67
29.00	1.06	0.40	80.75	27.24	21.83	22.06	28.83	27.29
29.39	1.20	0.34	82.90	24.42	25.16	24.99	29.42	20.44

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4.1 RESULTS AND DISCUSSIONS

By comparing the volumetric efficiency against load of the NRL HSD and IOCL HSD we observed that in case of IOCL diesel at first the volumetric efficiency goes on increasing from 98% to a point of point 99.48% and then decreased linearly up to 90.58% with increases in load of 65.66%.After then it goes on increasing again since there is a decreases in load occur. Furthermore in case of IOCL HSD volumetric efficiency goes on decreasing from the point 89.8% to 89.71% with increases in load of 50%.And then it increases gradually to a point 89.84% and then it decreases again. The all above occurrence is due to that when load increases A/F decreases and due to this rich mixture will inject to the cylinder as per stroke. And so the intake temperature increases causing the chances of turbulence more, as a result of this volumetric efficiency decreases. Rate of pressure is lowered in the above cases. But from the point of view of selection, NRL HSD has a higher volumetric efficiency than IOCL HSD, which will permit more air suction per stroke and hence power output.

In terms of brake thermal efficiency, we observe that when the engine load increases, the brake thermal efficiency decreases upto a certain load, which is 77% in the case of NRL HSD blend. Similar case is also observed in the case of IOCL HSD at an arbitrary brake load of 54%, which then increases linearly in both the cases. The main striking reason is that when load increases, the brake power due to an increase in the resultant torque. Also, the brake thermal efficiency at a given load was found to be higher in the case of NRL HSD than IOCL HSD. This is because the calorific value of NRL HSD is 63970.452 kJ/kg, which is much higher than IOCL HSD, where calorific value is 45733.8 kJ/kg. Surprisingly, this coincides with the theoretical deduction as η_{bth}=B.P./ ($W_f \times C_V$), where C_v is the calorific value of the fuel.

With load increase, the exhaust gas temperature increases. In the case of NRL HSD, the exhaust gas temperature rises dramatically from 132°C to 193°C, which is the peak temperature as evident from the graph. Similarly, for IOCL HSD, the peak temperature occurs at 66.6% brake load (293°C). The reason for increase is that the dissociation effects and losses due to variation of specific heats become more prominent at higher loads. Specific -heat increases at higher loads. This explains the practical anomalous deviation from the actual fuel-air cycle. Also, the graphs provide evidence of the fact that exhaust gas temperature for IOCL HSD is more than that of NRL HSD. The dissociation effects are more than that of NRL, thus lowering of peak pressure will be higher and lower power will be delivered per cycle in the long run.

In terms of specific fuel consumption, as load increases, the specific fuel consumption increases. This is significant since A /F ratio decreases, thus enriching the desired mixture to be introduced per stroke. Flame speed becomes low and the timing losses increases. Hence, in comparing both IOCL and NRL HSD grades, we observe that the specific fuel consumption is much lower for IOCL HSD than that of IOCL HSD. For example, we may take the case of NRL HSD, which has the peak specific fuel consumption is 0.976 kg/kW-s at an A/F ratio of 5.43:1., while for IOCL HSD, the peak specific fuel consumption of 0.452

kg/kW-s at an A/F ratio of 4.64:1. Thus, in the long run for feasibility of commercial vehicles, IOCL HSD is considered as an optimum as compared to that of NRL HSD in terms of reduced specific fuel consumption, reduced friction losses and reduced pumping losses.

At constant blends of n-hexane (including 5%, 10%, 15% blends), for both NRL and IOCL grades of diesel as brake load increases, thermal efficiency increases. This is significant because brake power increase with increase in load as evident from the graph. In the case of 5% blend, with increase in brake load, thermal efficiency rises subsequently from 13.04% to 47.06%, whereas in the case of 10% blend, it rises from 12.54% to 52.66% but in the case of 15% blend, it rises from 12.65% to 46.81%. But there is slight reduction in volumetric efficiency, since A/F ratio goes on decreasing with increase in load as the intake temperature increases. But one advantage is of the fact that the delay period will decrease since increase in intake temperature at suction will increase the temperature of compressed air inside the cylinder, thus reducing the tendency to knock.

Also, at constant blends of 5%, 10% and 15% blends of n-hexane, the specific fuel consumption seemed to increase up to a certain extent but it again decreases. This is possible since the brake load increases producing more brake power, but a maximum point is reached on the fuel consumption curve. This may indicate that at the given readings for 5% blend, as the air-fuel ratio decreases, exhaust gas temperature and dissociation effects will be produced. As the mixture goes on enriching from 5:1 to 4.629:1 on the test cylinder, a certain point is reached after which the effects of dissociation will decrease , indication greater formation of CO as white smoke comes out from the exhaust . This may explain the anomalous behaviour of specific fuel consumption. Similar is the case for 10% and 15% blend with both NRL and IOCL grades. So, on behalf of constant blends, the optimum mixture is obtained at 66.6% brake load, as the specific fuel consumption is quite fair(3.321 kg/kW-s for 5% NRL blend,3.08kg/kW-s for 5% IOCL blend and similar cases for 10% and 15% blends). For the same reason, optimum mixture is obtained at 66.6% brake load IOCL-hexane blend.

Now, comparing on basis of blends of same grades and at corresponding brake loads, we observe that with the same brake load of 37.5%, brake thermal efficiency decreases with increase in blending up to 10% blend, and again increases with 15% blend. Similar is the case for the other two loads, 54.16% and 66.6% respectively. Increase in blends of hexane may be responsible for this, since the efficiency of higher blends decreases as the calorific value of the mixture decreases with the increase in hexane ratio. But a sharp increase with 15% blend at the corresponding load indicates that there must be a minimum value at which this deviation occurs. The lower heat value of the hexane makes heating value of the mixture to decrease and hence the BSFC to increase for higher blends. But volumetric efficiency decreases with increase in the percentage of blend. With increase in the percentage of blend, causes a reduction in the A/F ratio. Similar is the case for loads at 54.16% and 66.6%, individually for 10% and 15% blends. Also, at a given load of 54.16%, with increase with the percentage of blends, the dissociation temperature is maximum for 10% blend (262°C) beyond which it again starts to decrease. This is significant with enrichment of mixture and partial combustion of fuel with low flame speed. Similar results are also obtained for IOCL grades. So, on the basis of a given load and at different blends, one can conclude that 5% blend of n-hexane may be taken as the optimum mixture for IOCL and NRL grades as low dissociation and hence low NO_x emissions, optimum fuel consumption is achieved.

Again we compare different grades of diesel with variation in the percentage of blends. At 5% blend, brake thermal efficiencies are slightly higher for IOCL hexane blends as compared to NRL hexane blends. This is evident at corresponding brake loads. For 37.5% brake load, it is 13.04% for while it is 17.49%. At 54.16% brake load, it is 38.15% for NRL hexane blend and it is 48.89% for IOCL hexane blend. At 66.6% brake load, it is 47.06% while it is 70.99% for IOCL grade hexane blends. Similar investigations were carried out for 10% blends as well as 15% blends for both IOCL and NRL blends. The striking reason for decrease is that calorific value of IOCL grade HSD(high speed diesel) is 45733.8 kJ/kg, as per the data obtained with permission from the Quality Control Laboratory, IOCL BONGAIGAON REFINERY, while that for NRL it is 63970.452 kJ/kg, which decreases the brake thermal efficiency. Surprisingly, this also clearly coincides with the theoretical deduction η_{bth}=(b.p.)/(m_f×C.V.), where C.V. is the calorific value of the fuel.

Also, from the graphs of volumetric efficiency plotted against brake load, we find that both the curves of 5% blend for NRL-IOCL hexane blends decreases sharply at 54.16% brake load but after that, the steep increase is prominent for the NRL blend. This indicates that increase in brake load beyond 54.16%, the volume of air at suction will be more, so the power output of the cylinder will be more and hence hexane-fuel blend may be injected per stroke. This ensures higher peak pressures to be achieved with NRL-hexane blends.

Interestingly, if we observe the variation of specific fuel consumption with load on the curve, we can imply that it varies proportionally for both 5% blends of IOCL-hexane and NRL-hexane up to a brake load of brake load of 54.16%, then it decreases sharply for IOCL-hexane blend with further increase in brake load. This is indicates that the blending at 5% will be more effective for IOCL HSD (High Speed Diesel) than the NRL one, since at the same time it will reduce the specific fuel consumption and at the same time it will reduce the friction losses in the cylinder that decreases with increase in engine load.

One further notice is that at a given brake load, the exhaust gas temperature is more pronounced for 5% IOCL HSD hexane blend than 5% NRL HSD- hexane blend. This may be considerable with the fact that A/F ratio is less for the IOCL blending (4.807:1) as against the NRL blend (5:1). Also, the high exhaust gas temperature indicates that a higher amount of unburnt particles will be released at exhaust. Similar cases are also obtained from the graph for 10% blend and 15% blends with both NRL -HSD and IOCL-HSD. Thus, one can conclude that while 5%, 10%, 15% blends with NRL-HSD will take care of the environmental degradation and volumetric efficiency, but owing to factors like specific fuel consumption and friction losses in the bearing and cylinder, IOCL HSD-hexane blend may be applicable in the long run.

Graphical analysis:

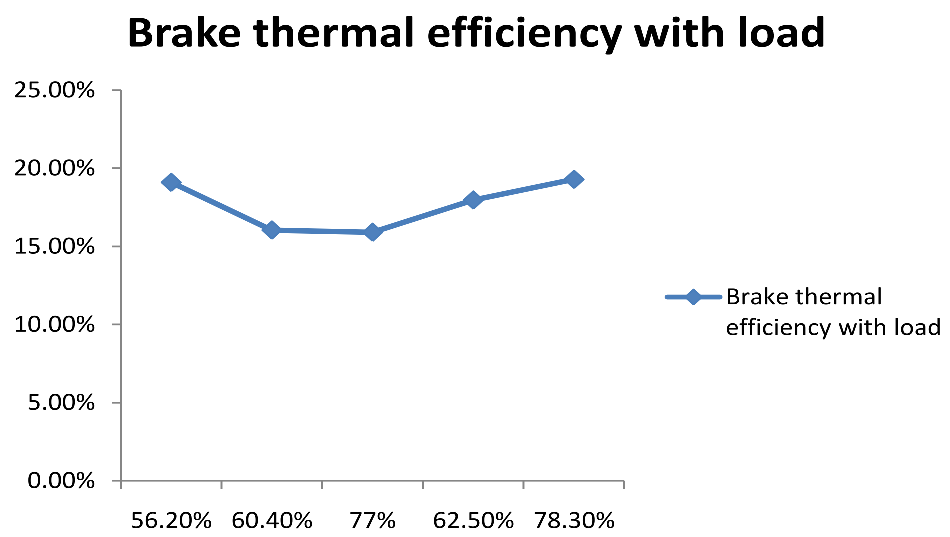

Fig. 1 Variation of brake thermal efficiency with load (pure NRL HSD)

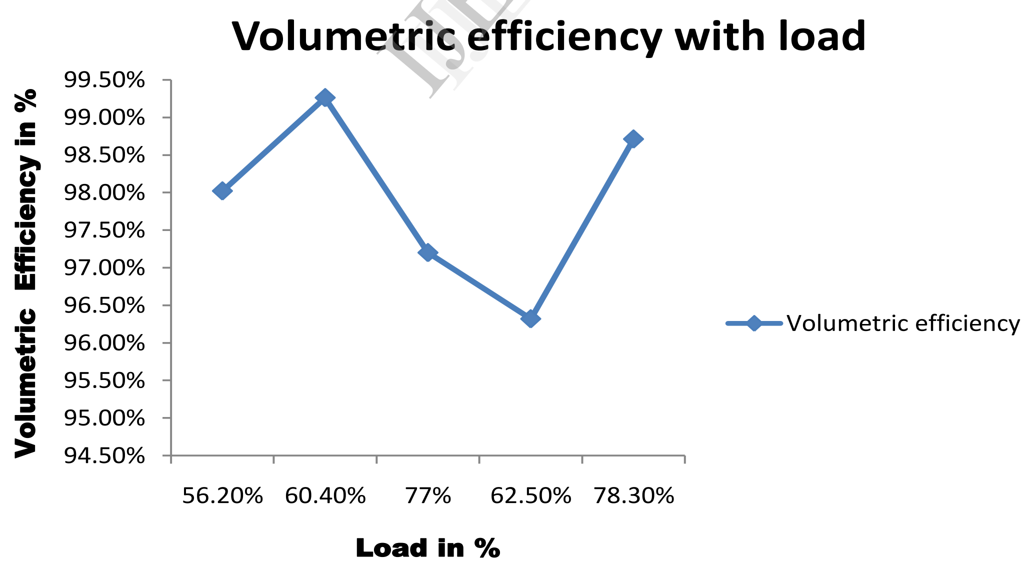

Fig. 2 Variation of volumetric efficiency with load (pure NRL HSD)

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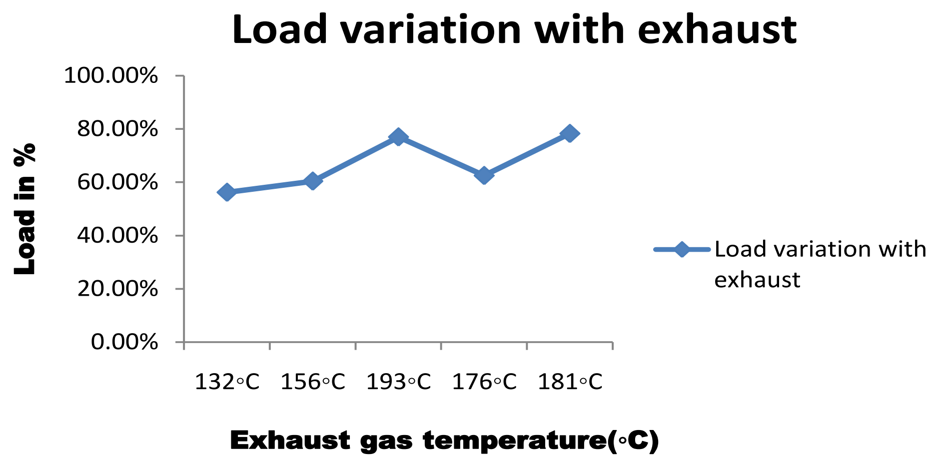

Fig. 3 Load variation with exhaust (pure NRL HSD)

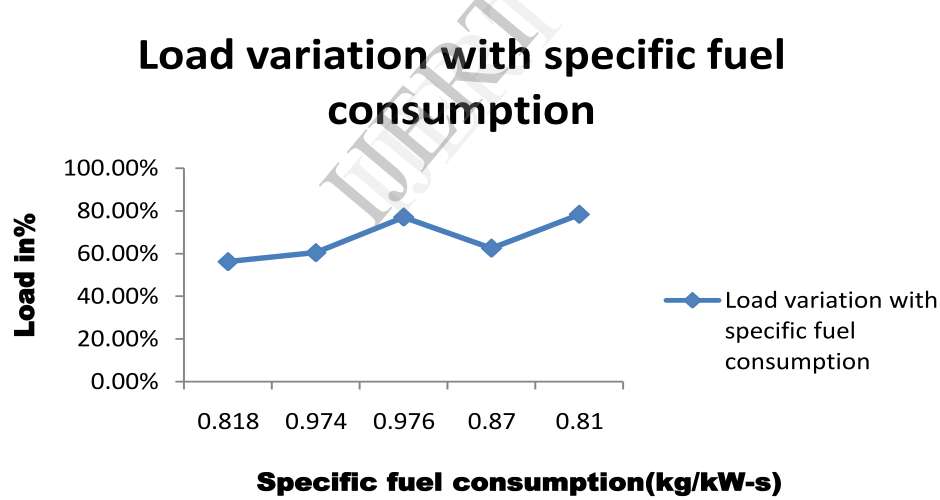

Fig. 4 Load variation with specific fuel consumption (pure NRL HSD)

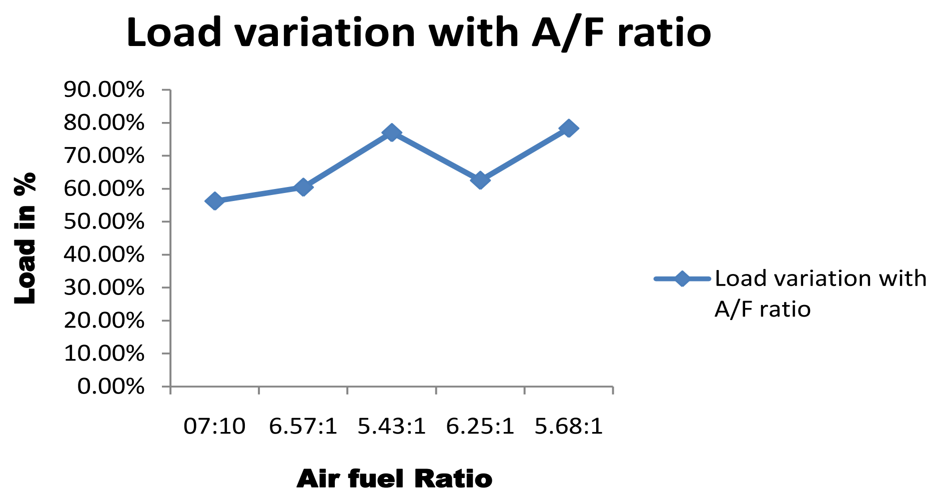

Fig. 5 Load variation with A/F ratio (pure NRL HSD)

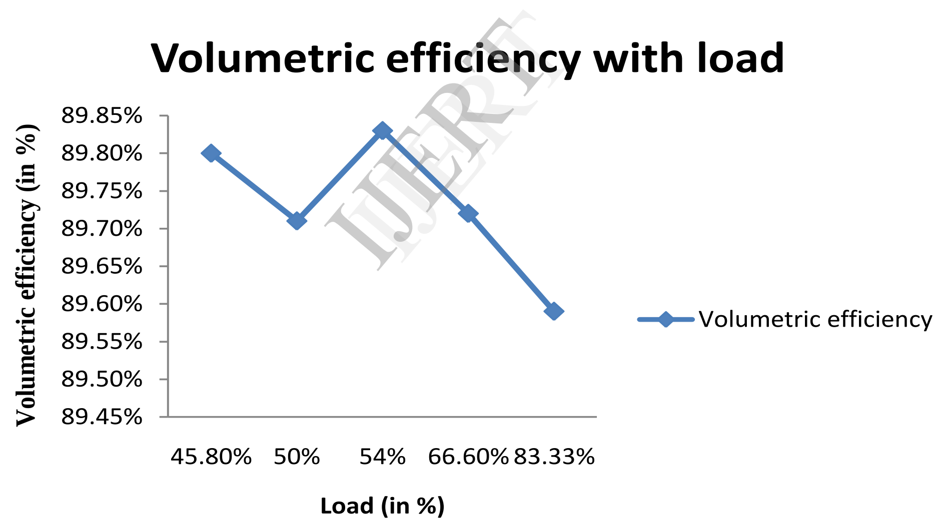

Fig. 6 Variation of volumetric efficiency with load (pure IOCL HSD)

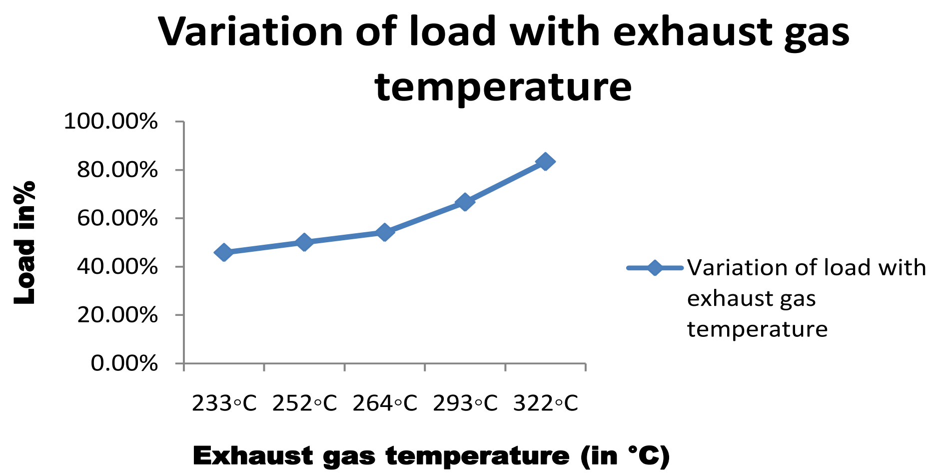

Fig. 7 Effect of load on exhaust gas temperature (pure IOCL HSD)

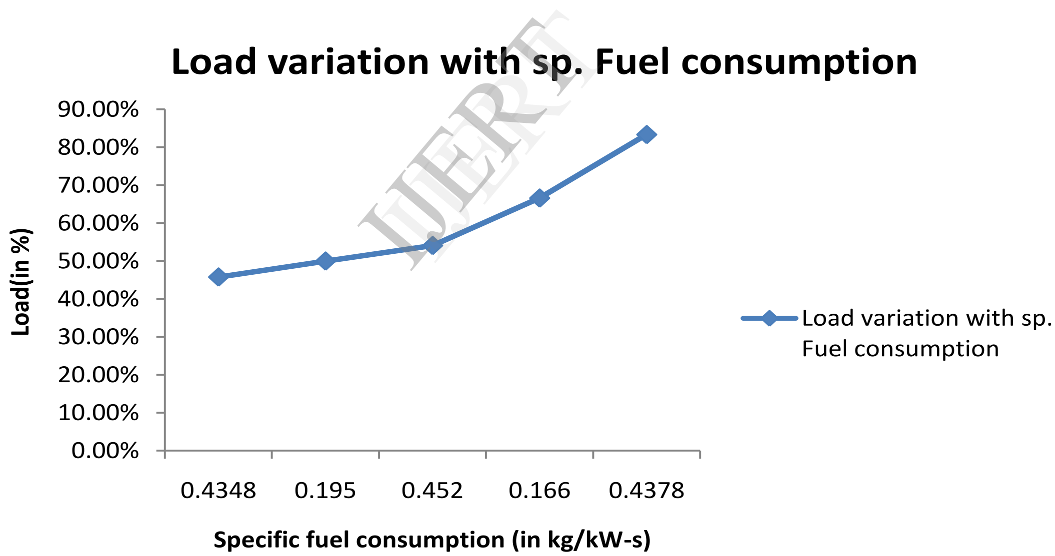

Fig. 8 Effect of load variation on specific fuel consumption (pure IOCL HSD)

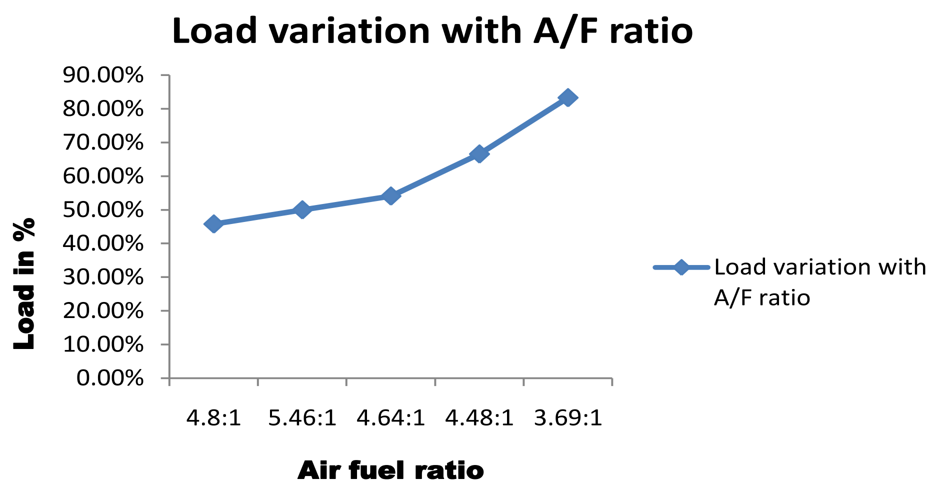

Fig. 9 Effect of load variation on A/F ratio(pure IOCL HSD)

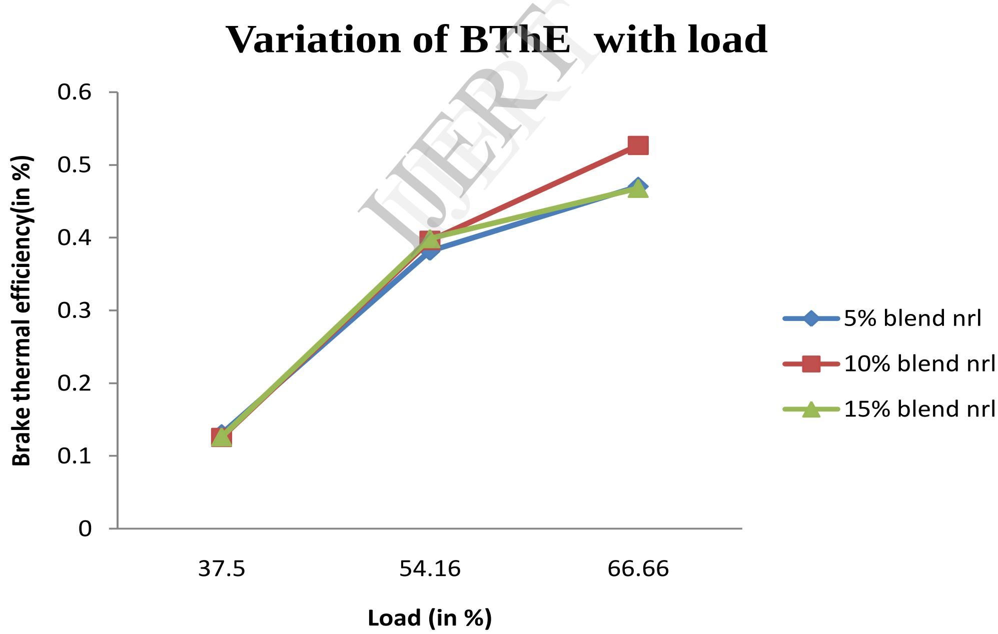

Fig. 10 Variation of brake thermal efficiency with load at different blends

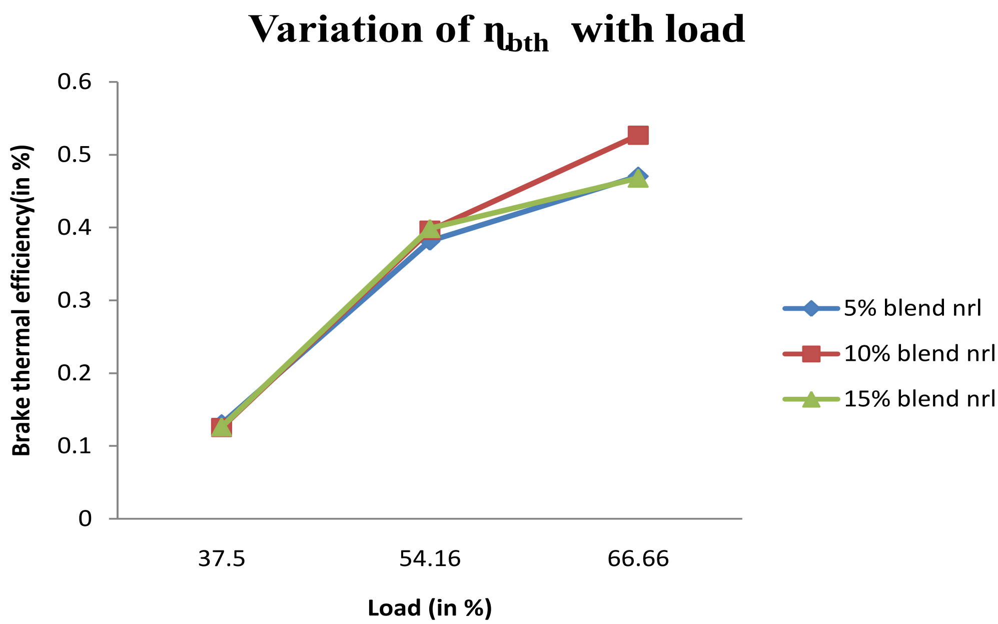

Fig. 6 Variation of brake thermal efficiency with load at different blends

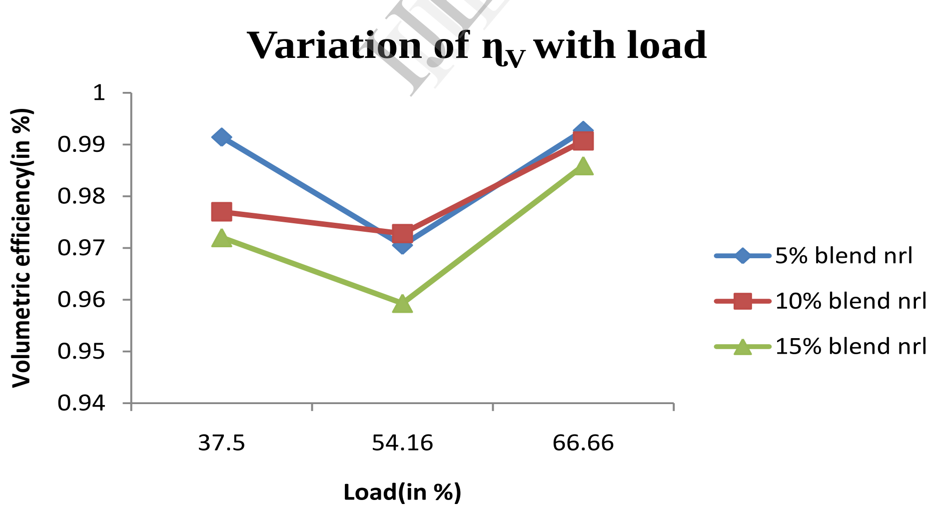

Fig. 7 Variation of volumetric efficiency with load at different blends

CONCLUSIONS

It was seen that the NRL HSD is more efficient than IOCL HSD in terms of volumetric efficiency. Reduction in brake thermal efficiency with an increase in load is upto a certain point of load in case of both of the diesel brands after which a linear increase in brake thermal efficiency occurs. This variation is due to the fact that as the load increases, the resultant torque increases. After our experimental analysis the brake thermal efficiency of the NRL HSD is found to be more than that of IOCL HSD. And it is due to the fact that the calorific value of NRL HSD is more than that of IOCL HSD. The exhaust gas temperature increases with an increase in load. This is due to the dissociation effects and losses due to variation of specific heats which becomes more prominent at higher loads. From our experimental analysis we have found that the exhaust gas temperature of IOCL HSD is more than NRL HSD. The dissociation effects of IOCL HSD are more than that of NRL, thus lowering of peak pressure will be higher and lower power will be delivered per cycle in the long run. As the load increases the air-fuel ratio decreases, thus enriching the desired mixture to be introduced per stroke and hence the specific fuel consumption increases. From our experimental analysis we have found that the specific fuel consumption of IOCL HSD is more than that of NRL HSD.At constant blends of n hexane (including 5%, 10%, 15% blends), for both NRL and IOCL grades of diesel as brake load increases, thermal efficiency increases. This is significant because brake power increase with increase in load as is evident from our experimental analysis. At constant blends of n hexane (including 5%, 10%, 15% blends), for both NRL and IOCL grades of diesel, the specific fuel consumption increases upto a certain extent, but it again decreases. This is due to an increase in brake load which produces more brake power. This anamolous behaviour is due to exhaust gas temperature and dissociation effects which gives rise to greater formation of white gases called CO.

On comparing the blends of three different proportions 5%, 10%, 15% blends with the two aforesaid grades of diesel we observe some striking variations in the parameters of our project importance. Such as, the brake thermal efficiency decreases with an increase in blending upto 10% blend, but increases with 15% blend. This may be due to increase in blends of hexane, since the efficiency of higher blends decreases as the calorific value of the mixture decreases with the increase in hexane ratio.

With an increase in the percentage of blend, a reduction in the A/F ratio occurs. Similar is the case observed for the other two loads with the other two blends. With the increase in blends the dissociation temperature. So, on the basis of a given load and at different blends, one can conclude that 5% blend of n-hexane may be taken as the optimum mixture for IOCL and NRL grades as low dissociation and hence low NO_xemissions, optimum fuel consumption is achieved.

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