JCARME
Journal of Computational and Applied Research in Mechanical Engineering
SRTTU
http://jcarme.srttu.edu
ISSN: 2228-7922
Vol. 1, No. 2, March 2012

Performance evaluation of a low heat rejection diesel engine with carbureted ethanol and jatropha oil

M. V. S. Murali Krishna[a,*] , V. V. R. Seshagiri Rao[a], P. V. K. Murthy[b] and T. K. K. Reddy[c]

[a] *Department of Mechanical Engineering, Chaitanya Bharathi Institute of Technology, Gandipet, Hyderabad-500 075. India.*
[b] *Vivekananda Institute of Science and Information Technology, Shadnaga, Mahabubnagar-509216, India*
[c] *Department of Mechanical Engineering, J.N.T.U. College of Engineering, Kukatpally, Hyderabad- 500 085, India.*

Article info:
Received: 31/08/2011
Accepted: 31/01/2012
Online: 03/03/2012

Keywords:
Crude jatropha oil,
Ethanol,
LHR engine,
Performance,
Pollution levels,
Combustion characteristics

Abstract

Experiments were conducted to evaluate the performance of a low heat rejection (LHR) diesel engine. Performance parameters and emission levels were determined at various magnitudes of brake mean effective pressure. Combustion characteristics of the engine were measured with TDC (top dead centre) encoder, pressure transducer, console and special pressure-crank angle software package at peak load operation of the engine. Conventional engine (CE) and LHR engine showed improved performance at recommended injection timing of 27°bTDC and recommended injection pressure of 190 bar, when compared with CE with pure diesel operation. Peak brake thermal efficiency increased by 18%, smoke levels decreased by 48% and NOx levels decreased by 38% with LHR engine relatively at its optimum injection timing and maximum induction of ethanol when compared with pure diesel operation of CE at manufacturer's recommended injection timing.

Nomenclature

BMEP	Brake mean effective pressure in bar	HPLC	High performance liquid chromatography
BSEC	Brake specific energy consumption in kW/kW	HSU	Hartridge Smoke Unit
BSFC	Brake specific fuel consumption in kg/hr·kW	LHR	Low heat rejection
bTDC	Before top dead centre	MRPR	Maximum rate of pressure rise in bar/deg
BTE	Brake thermal efficiency in %	NOx	Oxides of nitrogen in ppm
CE	Conventional engine	PP	Peak pressure in bar
CJO	Crude jatropha oil	ppm	Parts per million
CL	Coolant load in kW	SFC	Specific fuel consumption in kg/h·kW
CO	Carbon monoxide in %	TDC	Top dead centre
DI	Direct injection	TOPP	Time of occurrence of peak pressure in degrees
DNPH	Dinitrophenyl hydrazine solution	VE	Volumetric efficiency in %
EGT	Exhaust gas temperature in Centigrade degrees		

* *Corresponding author*
Email address: maddalivs@gmail.com

1. Introduction

In the context of depletion of fossil fuels, the increase of pollution levels with fossil fuels and ever increase of prices of crude petroleum in International Market, the search for alternate fuels has become pertinent for the engine manufacturers, users and researchers involved in the combustion research. Diesel fuel was being consumed not only in transport sector but also agricultural sector. Hence much emphasis was given for an alternate fuel, which could substitute for diesel fuel. Alcohols and vegetable oils are probable candidates as alternate fuels for diesel engine. Vegetable oils have Cetane number compatible to diesel fuel. Alcohols have high volatility. However, high viscosity and poor volatility of vegetable oils and low Cetane number of alcohols call for LHR engine. It is well known fact that small amount of energy is left for useful purpose in engine. The remaining energy is wasted through friction and other losses. Hence the concept of LHR engine is to minimize the heat loss to the coolant; by providing resistance in the path of heat flow to the coolant thereby gains the heat energy. Hence combustion chamber is maintained very hot for LHR engine which is useful for burning cheap, renewable and alternate fuels. Several methods adopted for achieving LHR to the coolant are:
i) using ceramic coatings on piston, liner and cylinder head ii) creating air gap in the piston and other components and using low-thermal conductivity materials like superni, cast iron and mild steel. Investigations were carried out on bio-diesel by many researchers [1-4] they coated with low thermal conductivity materials like ceramics on engine components like cylinder head, cylinder liner, valves and piston crown, and it was reported that ceramic coated engines improved SFC and decreased pollution levels. However, due to low degree of insulation provided by these researchers, highly viscous crude vegetable oil was not effectively burnt.
Creating an air gap in the piston involves the complications of joining two different metals. LHR engine [5] with an air gap insulated piston resulted in a reduction in total coolant heat rejection ranging from 3% at light load to 5 to 7% at full load. BFSC decreased in LHR engine at part loads in comparison with conventional piston engine. However no reduction in BSFC was noticed at full load operation of the engine. The piston crown was made of Inconel-750x, which has high temperature strength and relatively poor thermal conductivity. The crown was fitted to the body of the piston through four-screws employing disc springs to maintain sufficient clamping load despite dimensional changes due to thermal expansion. The effective thickness of air gap was worked out to be about 4 mm. Though the sealing area showed no evidence of combustion gas leakage, some carbon was found in the space on disassembly because of the leakage through a crack developed during testing. Air gap insulation extending over the full crown was tested by Moffattet et al. [6] in a pressure charged single cylinder DI diesel engine. In this test an air gap was formed between a thin heat resistant crown made from nimonic nickel alloy and an aluminum piston body. The crown was supported clear of the body by a broad ring at the piston periphery and a groove in the ring provided a secondary air gap. The crown was also supported directly on the body by an annular seat beneath the bowl and by a support at the bowl center. Attachment was made by a single steel bolt through the under crown into the bowl center and by six short titanium bolts through the crown flange and supporting into the body. The piston could withstand 77 hours of running at full load with periodic stops before failure of two bolts attaching the crown occurred. At the same time carbon deposits noticed within the air gap, indicated an imperfect sealing against combustion gas pressure before failure of the bolts occur. The reduction of heat loss to the piston was estimated to be in the range of 25 to 30 % at full load when the exhaust gas temperature was raised by 24 ^{0}C. **A**n air gap insulated piston [7] was developed employing bolted and welded/roll bonded designs. While the bolted design demonstrated the effectiveness of air gap insulation, the roll bonded and welded designs were found to be more robust and succeeded to a great extent to provide the complicate sealing

of the air gap necessary for conventional insulation. Engine testing of more than 200 hours at full load was achieved for this air gap piston. The heat flow to the crown was reduced by 33%. An improved design giving 41% reduction of heat flow was tested for 78 hours at full load with no evident deterioration. It was also expected to have a more durable piston achieving up to 50% reduction in heat flow. However, attempts were not fully successful in perfecting the design as some of the piston crowns got detached due to failure in aluminum weld directly below the roll bond. Some of the pistons were found to have suffered cracking of the aluminum weld while in some other cases the crown piece had exerted such a force that a radial displacement was noticed.

Rama Mohan et al. [8] made an air gap insulated piston in which perfect sealing was achieved between the crown and the rest of the body of the piston. The design was entirely different from the bolted or welded designs of the other researchers who faced failures and leakage. In this novel design the crown as a cap was screwed to the piston body having a desired air gap extending over most of the under crown area. The exhaust gap temperature increased and brake specific fuel consumption decreased up to 80% of the full load condition, but the investigations were restricted to pure diesel operation. It was reported from these investigations that SFC improved and pollution levels of smoke decreased at advanced injection timing Krishna [9] used an air gap insulated piston with superni crown and air gap insulated liner with superni insert. He used pure diesel and alternate fuels of alcohol and vegetable oils (in crude and esterified form) at different operating conditions of vegetable oil and varied injection timing and injection pressure and it was reported that LHR engine improved the performance and decreased pollution levels with alternate fuels. Krishna Murty [10] Experiments conducted on high grade LHR engine consisting of air gap insulated piston, air gap insulated liner and ceramic coated cylinder head. He used bio-diesel with varied injection timing and injection pressure and reported that bio-diesel operation on LHR engine improved the performance. Experiments were conducted [11-13] on CE with vegetable oils and it was reported that these blends improved the efficiency of the engine and decreased the pollution levels. Rajesh et al. [14] found out that compression ratio was also increased with CE with vegetable oil based bio-diesel. He reported that poor performance was obtained at lower compression ratio and performance of the engine was improved at compression ratio of 18:1. Experiments were conducted [15-19] on bio-diesel on CE. An improvement in BTE, NOx emissions was increased and brake specific fuel consumption (BSFC) was slightly increased. In [20], investigations were carried out on waste fried vegetale oil collected from restaruents. It was reported that CO and smoke emissions were reduced using preheated waste frying oil at 135°C. In another investigation, [21], carbureted ethanol along with crude jatropha oil was used in CE and in LHR engine with air gap in the insulated piston and insulated liner. It was reported that performance improved with both versions of the engine, when compared with pure diesel operation on CE as combustion improved with adiabatic conditions.

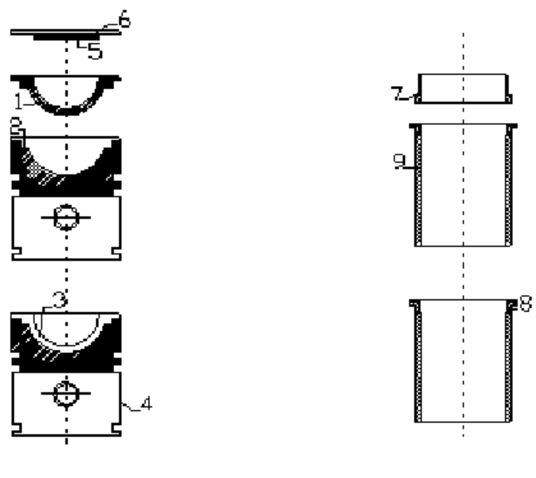
Table 1. Properties of the vegetable oil and diesel.

Test Fuel	Viscosity at 25°C (centi-poise)	Density at 25 °C	Cetane number	Calorific value (kJ/kg)
Diesel	12.5	0.84	55	42000
Jatropha oil (crude)	125	0.90	45	36000

In order to take advantage from high Cetane number of vegetable oils and high volatility of alcohols, both vegetable oils and alcohols have to be used in LHR engine.
In this work, it is attempted to evaluate the performance of LHR engine, which has an air gap piston, air gap liner and ceramic coated cylinder head crude jatropha oil and carbureted ethanol are used with varying engine parameters of injection pressure and injection timing. The results are compared with pure diesel operation on CE at recommended injection timing and injection pressure.

2. Experimental Program

Figure 1 shows the details of insulated piston, insulated liner and ceramic coated cylinder head employed in the experimentation. LHR diesel engine contains a two-part piston; the top crown made of a low thermal conductivity material, superni-90, threaded to aluminum body of the piston, providing a 3mm-air gap between the crown and the body of the piston. The optimum thickness of air gap in the air gap piston was found [8] to be 3 mm, for better performance of the engine with superni inserts and diesel fuel. A superni-90 insert was screwed the top portion of the liner in such a manner that an air gap of 3-mm was maintained between the insert and the liner body. At 500^{o}C the thermal conductivity of superni-90 and air are 20.92 and 0.057 W/m.K respectively. Partially stabilized zirconium (PSZ) of thickness 500 microns was coated by means of plasma coating technique. The properties of vegetable oil and diesel fuel are given in Table 1.
The experimental setup used for the investigations of LHR diesel engine with CJO and carbureted ethanol was shown in Fig. 2. CE has an aluminum alloy piston with a bore of 80 mm and a stroke of 110 mm. The rated



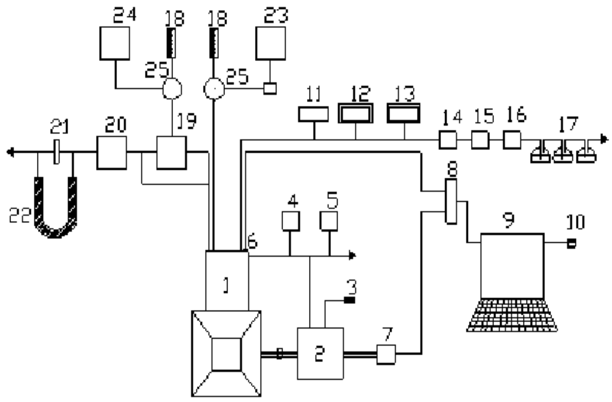
(a) Insulated piston & Ceramic coated cylinder head

(b) Insulated liner

1. Crown	4. Body	7. Insert
2. Gasket	5. Ceramic coating	8. Air gap
3. Air gap	6. Cylinder head	9. Liner

Fig. 1. Assembly details of insulated piston and ceramic coated cylinder head (a), and insulated liner (b).

power output of the single cylinder engine at 1500 rpm is 3.68kW. The compression ratio is 16:1 and the manufacturer’s recommended injection timing and injection pressures are 27^{o}bTDC and 190 bar respectively. The fuel injector has 3 holes of size 0.25 mm. The combustion chamber is of a direct injection type no temperature control were incorporated, for measuring the lube oil temperature. Copper shims of suitable size were provided in between the pump body and the engine frame, to vary the injection timing. Effect of injection timing on the performance of the engine was studied.



1.Engine, 2.Electical Dynamo meter, 3.Load Box, 4. Outlet jacket water temperature indicator,5.Outlet-jacket water flow meter Orifice meter, 6. Piezo-electric pressure transducer, 7. TDC encoder 8.Console, 9. Pentium Personal Computer, 10. Printer, 11.Exhaust gas temperature indicator, 12.AVL Smoke meter, 13. Netel Chromatograph NOx Analyzer, 14. Filter, 15.Rotometer, 16.Hetaer,17. Round bottom flask containing DNPH solution, 18.Burette, 19. Variable jet carburetor, 20. Air box, 21.Orifice meter, 22. U-tube water manometer, 23.Vegetable oil tank, 24.Ethanol tank, 25. Three-way valve.

Fig. 2. Experimental Set-up.

and no special arrangement for swirling motion of air is made. The engine is connected to an electric dynamometer for measuring its brake power. A variable jet carburetor was fitted on the inlet manifold of the engine for inducting ethanol at different percentages of diesel flow rate during the suction stroke of the engine and the CJO was injected through the injector. Two separate fuel tanks and burette arrangements were made for measuring consumption of the vegetable oil and ethanol. Air-consumption of the engine was measured by air-box method. The naturally aspirated engine was provided with water-cooling system in which the inlet temperature of water was maintained at 60^{o}C by adjusting the water flow rate. The engine oil was provided with a pressure feed system and Injection pressures from 190 bars to 270 bars (in steps of 40 bars) using nozzle testing device. The maximum injection pressure is restricted to 270 bars due to the practical difficulties involved. EGT is measured with thermocouples made of iron and iron-constantan. Emissions of smoke and NO_x are recorded by AVL smoke meter and Netel Chromatograph NOx analyzer respectively at various magnitudes of BMEP. With alcohol-vegetable mixture operation, the major pollutant emitted from the engine is aldehydes. These aldehydes are carcinogenic in nature, and are harmful to human beings. The measure of the aldehydes is not sufficiently reported in the literature. DNPH method was employed for measuring aldehydes in the experimentation [9]. The exhaust of the engine

is bubbled through 2,4 dinitrophenyl hydrazine (2,4 DNPH) solution. The hydrazones formed are extracted into chloroform and are analyzed by employing high performance liquid chromatography (HPLC) to find the percentage concentration of formaldehyde and acetaldehyde in the exhaust of the engine.
Piezo electric transducer, fitted on the cylinder head to measure pressure in the combustion chamber is connected to the console which in turn was connected to a personal computer. TDC encoder provided at the extended shaft of the dynamometer is connected to the console to measure the crank angle of the engine.

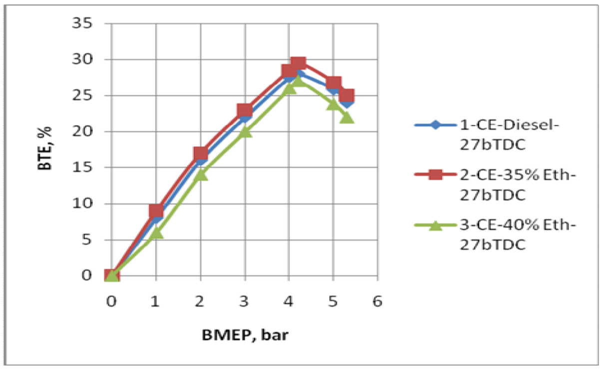


Fig. 3. Variation of BTE with BMEP in CE at different percentages of ethanol induction.

A special P-θ software package evaluates the combustion characteristics such as peak pressure (PP), time of occurrence of peak pressure (TOPP) and maximum rate of pressure rise (MRPR) from evaluated the combustion the signals of pressure and crank angle at the peak load operation of the engine.

3. Results and Discussions

3.1 Performance Parameters

Investigations were carried out with the objective of determining the factors that would allow maximum use of ethanol in diesel engine with best possible efficiency at all loads.
Figure 3 shows the variation of BTE with BMEP with pure diesel operation on CE is also shown for comparison purpose. BTE increased at all loads with 35% ethanol induction but for the increase of ethanol induction beyond 35%, it is decreased at all loads in CE when compared with CE with diesel operation (standard diesel). The reason for improving the efficiency with the 35% ethanol induction is that due to lower combustion temperature, dissociated losses, specific heat losses and cooling losses are decreased. Lower temperature is due to high heat of evaporation of ethanol. Induction of the ethanol also results in more moles of working gas, which causes

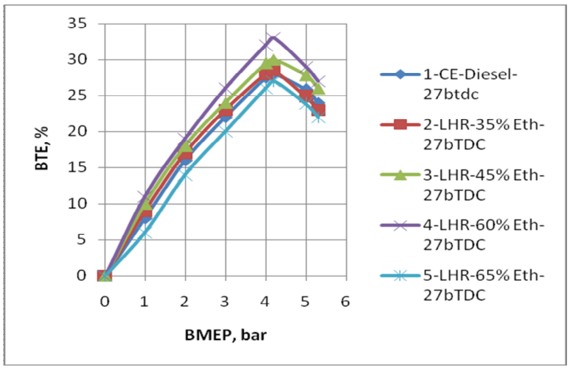


Fig. 4. Variation of BTE with BMEP in LHR engine at different percentages of ethanol induction.

higher pressure in the cylinder. The observed increase in the ignition delay period would allow more time for fuel to vaporize before ignition started. This leads to a higher burning rate and more heat release rate at constant volume, which means a more efficient conversion process of heat into work.
From Fig. 4, it could be observed that LHR engine showed an improvement in the performance with the carbureted ethanol at all loads when compared to the standard diesel engine. This is due to recovery of heat from the hot insulated components of LHR engine and high latent heat of evaporation of ethanol. From the figure it is observed that the maximum induction of ethanol, which shows improvement in the performance, is 60%.

According to Krishna [10] the optimum injection timings are at 33°bTDC for CE and at 32°bTDC for LHR engine with pure diesel operation. Similar trends are observed from Fig. 5 for CE and LHR engine with alcohol-vegetable oil operation when the injection timings were advanced to 32°bTDC in LHR engine and to 33°bTDC in CE. However, the maximum induction of alcohol is limited to 55% in the LHR engine at 32°bTDC against the 60% induction at 27°bTDC. For CE the maximum induction of alcohol is the for both 33°bTDC and 27°bTDC.

From Fig. 5, it could be noticed that LHR engine with 55% ethanol induction at its optimum injection timing showed improved performance at all loads when compared with other versions of the engine. This was due to higher amount of ethanol substitution and improved combustion at advanced injection timing caused better evaporation leading to produce higher BTE.

There is a limitation for use of ethanol due to its low Cetane number and scine it has a higher self-ignition temperature compared to vegetable oil. When percentage of ethanol is increased, more heat is utilized to evaporate ethanol and less heat is available to evaporate crude jatropha oil. Therefore, a major quantity of ethanol which burns late in the expansion stroke, would not be fully utilized. In order to avert this, injection pressure was increased. Therefore fuel droplet size decreased and surface to volume ratio was increased. As a result, less heat was required to evaporate droplets of vegetable oil. Poor performance at lower injection pressures indicates slow mixing probably because of insufficient spray penetration with consequent slow mixing during diffusion burning.

The trend exhibited by both versions of the engine with dual fuel operation at higher injection pressure of 270 bars is similar that for the injection pressure of 190 bars. However, the maximum induction of ethanol is 40% in CE at an injection pressure of 270 bars against 35% at 190 bars. Maximum ethanol induction remained same with LHR engine at 270 bars as in the case of 190 bars.

From Fig. 6, it could be observed that BSEC is decreased with increase of ethanol induction. The reason is that higher amount of ethanol substitution leads to a better evaporation and produces lower BSEC in both versions of the engine. BSEC is lower in LHR engine at its optimum injection timing, which shows the suitability of the engine for alternate fuels. It also decreases with increase of injection pressures in both versions of the engine. This is due to improved fuel spray characteristics.

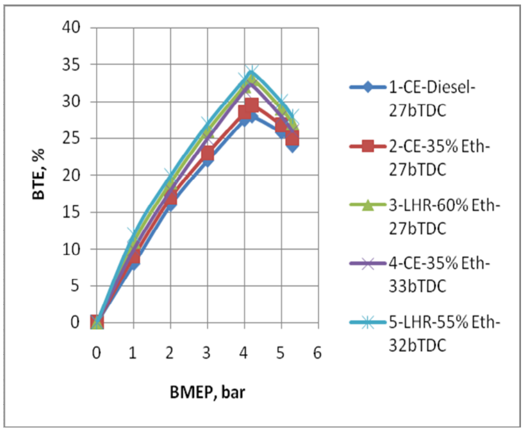


Fig. 5. Variation of BTE with BMEP with maximum percentage of ethanol induction in CE and LHR engine at recommended and optimum injection timings.

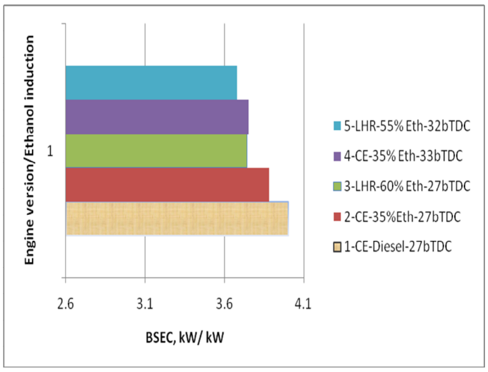


Fig. 6. Bar chart showing the variation of BSEC with maximum percentage of ethanol induction in CE and LHR engine at recommended and optimum injection timings.

From Fig. 7, it is seen that the magnitude of EGT decreased with increase of percentage of ethanol induction in both versions of the engine. Lower exhaust gas temperatures are observed in the LHR engine with 60% ethanol induction when compared with the CE with 35% ethanol induction. This shows that the performance of the LHR engine improves with 60% ethanol induction over the CE with 35% ethanol induction. EGT further decreases, when the injection timings are advanced in both versions of the engine. This is because, when the injection timing is advanced, the work transfer from piston to the gases in the cylinder at the end of the compression stroke is too large, leading to reduce in magnitude in EGT.

Figure 8, shows that coolant load decrease in both versions of the engine at different percentages of ethanol induction at all loads when compared with the pure diesel operation on CE. This is due to the reduction of gas temperatures with ethanol induction. Cooling load is less in the LHR engine with 60% ethanol induction when compared with the CE with 35% ethanol induction at all loads. This is due to the insulation provided in LHR engine. Cooling load decreases in both versions of the engine with advancing of injection timing and increase of injection pressure. This is due to decrease of gas temperatures and improving of air fuel mixing.

From the Fig. 9, it is observed that VE decreases marginally in both versions of the engine with dual fuel operation when compared with pure diesel operation on CE. As percentage of alcohol induction increases, amount of air admitted into cylinder is reduced. However, CE with different percentage of ethanol induction showed higher VE when compared with LHR engine. This is because of the increase of temperatures of the insulated components in LHR engine, which heat the incoming charge to higher temperatures and consequently the mass of air inducted in each cycle is lower. VE increases marginally with the increase of injection pressure in both versions of the engine. This is due to the improvement of air utilization and combustion with the increase of injection pressure. However, these variations are very small.

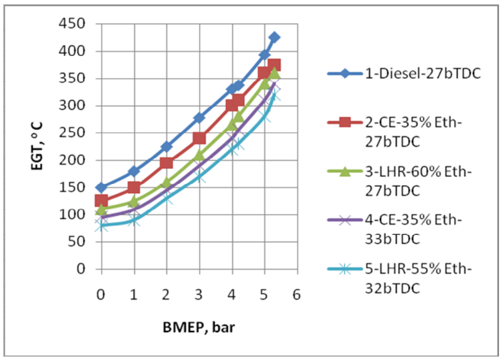


Fig. 7. Variation of EGT with BMEP in CE and LHR engine at recommended injection timing and optimized injection timings with maximum induction of ethanol.

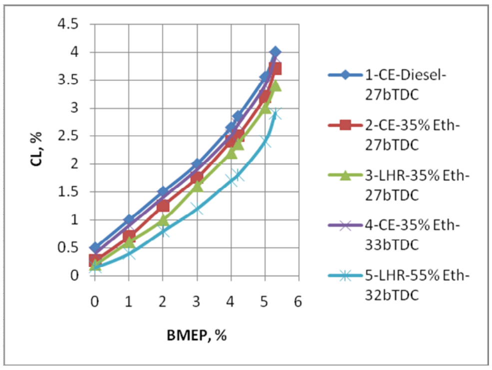


Fig. 8. Variation of CL with BMEP in CE and LHR engine at recommended injection timing and optimized injection timings with maximum induction of ethanol.

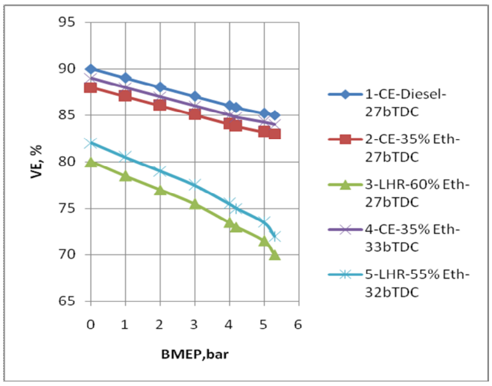


Fig. 9. Variation of VE with BMEP in CE and LHR engine at recommended injection timing and optimized injection timings with maximum induction of ethanol.

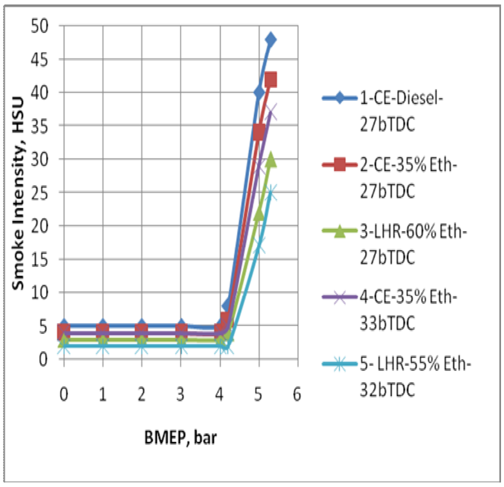


Fig. 10. Variation of Smoke levels in HSU with BMEP in CE and LHR engine at recommended injection timing and optimized injection timings with maximum induction of ethanol.

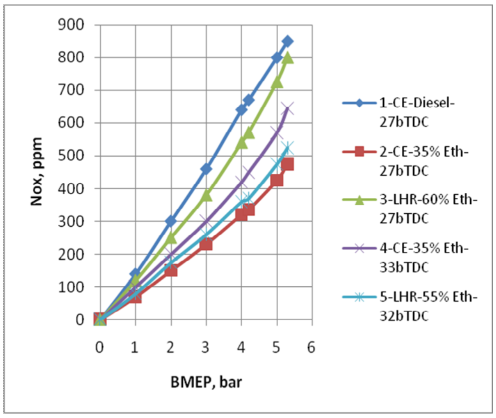
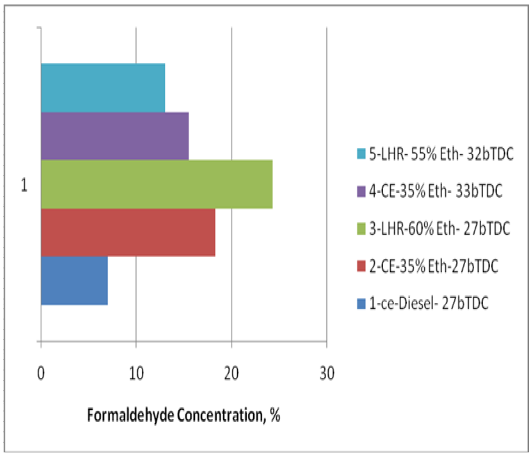
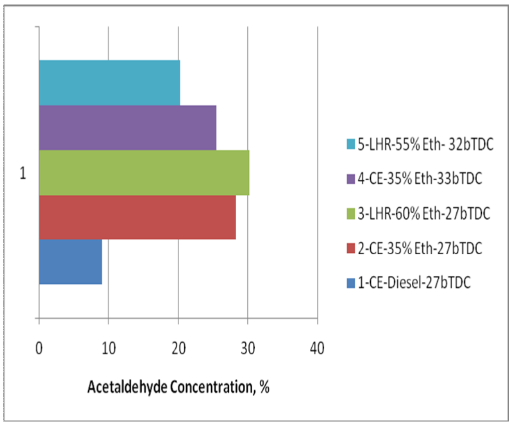


Fig. 11. Variation of NOx levels with BMEP in CE and LHR engine at recommended injection timing and optimized injection timings with maximum induction of ethanol.



(a) Formaldehyde concentraiton



(b) Acetaldehyde concentraiton

Fig. 12. Variation of aldehyde concentration in CE and LHR engine at recommended injection timing and optimized injection timings with maximum induction of ethanol, (a) formaldehyde concentraiton, and (b) acetaldehyde concentraiton.

3.2. Emission Levels

From Fig. 10, it could be seen that for the same load, the smoke density remains constant with increase of BMEP. The magnitude of smoke intensity increases from no load to full load in both versions of the engine. During the first part, the smoke level is more or less constant, as there is always excess air present. However, in the higher load range there is an abrupt rise in smoke levels due to less available oxygen, causing the decrease of air-fuel ratio, leading to incomplete combustion, producing more soot density. The variation of smoke levels with the brake power or BMEP, typically shows a U-shaped behavior due to the pre-dominance of hydrocarbons in their composition at light load and of carbon at high loads. Smoke levels decreased with induction of alcohol. The combustion of injected fuel in case of pure vegetable oil operation is predominantly one of oxidation of products of destructive decomposition. In this case, there are more chances of fuel cracking and forming carbon particles. On the other hand, the combustion of ethanol is predominantly a process of hydroxylation and the chance of fuel cracking is negligible. Ethanol does not have carbon-carbon bonds and therefore cannot form any un-oxidized carbon particles or precursor to soot particles. One of the promising factors for reducing smoke level with the alcohols is that they contain oxygen in their composition, which helps to the reduction of soot density. The soot emission increased linearly with increase of carbon to hydrogen (C/H) ratio if the equivalence ratio is not changed. This is because higher C/H leads to more concentration of carbon dioxide, which would be further, reduced to carbon. Consequently, induction of alcohol reduces the quantity of carbon particles in the exhaust gases as the magnitudes of C/H for diesel fuel, vegetable oil and ethanol are 0.45, 0.83 and 0.25, respectively. Lower smoke levels are observed in both versions of the engine in dual fuel mode when compared with pure diesel operation on CE. The LHR engine with 60% ethanol induction shows lower smoke levels when comparing with the CE with 35% ethanol induction. Smoke level decreases with increase of ethanol induction in both versions of the engine. In dual fuel operation, smoke level further decreases with advancing of the injection timing and with increase of injection pressure in both versions of the engine. This is due to efficient combustion at higher injection pressures, which improves the atomization. Hence, faster rate of combustion and shorter combustion duration at the advanced injection timings leads to the reduction of the smoke density in both versions of the engine.

From Fig. 11, it is observed that NOx emission decreases with increase of percentage of ethanol induction in both versions of the engine, due to lower combustion temperatures. The low value of C/H ratio in ethanol has indirect effect in reducing oxygen availability in the gases, which leads to the reduction of NOx. However, LHR engine with different percentages of ethanol induction shows higher NOx levels compared with the CE with 35% ethanol induction. This is due to increase of gas temperature in LHR engine. NOx level further decreases with increase of ethanol induction in both versions of the engine. NOx level increases marginally in the CE while it decreases in LHR engine, with advancing of the injection timing and with increase of injection pressure. This is due to reduction of gas temperature in LHR engine at 32^{o}bTDC and increase of residence time in CE.

From Fig. 12 (a), it is seen that aldehyde emissions are low with pure diesel operation in both CE and LHR engine. Formaldehyde emissions increase drastically with ethanol induction in both CE and LHR engine. With increased induction of ethanol up to 60%, CE registered very high value of formaldehyde emission in the exhaust, while it must lower in LHR engine. Hot environment of LHR engine completes combustion reactions and reduces emission of intermediate compounds, aldehydes. Figure 12 (b) shows the same trend as Fig. 12(a). Hence it is concluded that LHR engine was more suitable for alcohol engines in comparison with pure diesel operation. Advanced injection timing and increase of injection pressure also improves the combustion performance in LHR engine by reducing the intermediate compounds like formaldehyde and acetaldehydes.

3.3. Combustion Characteristics

From Fig. 13(a), it is obvious that the magnitude of PP increased with increases of ethanol induction in both versions of the engine. The magnitude of PP also increases with advancing of the injection timing in both versions of the engine with ethanol induction. LHR engine with 60% of ethanol induction exhibits higher PP compared with CE at 27^{o}bTDC and at injection pressure of 190 bar. This is due to increased amount of ethanol with LHR engine. With maximum induction of ethanol, LHR engine at 32^{o}bTDC produced higher PP compared with CE at 33^{o}bTDC.

From Fig. 13(b), it could be noticed that magnitude of TOPP decreases with increase of ethanol induction in both versions of the engine. When the ethanol induction is increased to 60% in LHR engine, the magnitude of TOPP is lower (shifted towards TDC) when compared with CE with 35% ethanol induction. This was once again confirmed by the observation of higher in LHR engine with dual fuel mode. The magnitude of TOPP decreases with advancing of the injection timing with both versions of the engine. From Fig. 13(c), it could be observed that LHR engine showed higher MRPR when compared with CE at different injection timing. This was due to higher amount of ethanol induction in LHR engine. MRPR increased with the advancing of the injection timing in both versions of the engine. These combustion characteristics also improve with increase of injection pressure.

4. Conclusions

Maximum induction of alcohol was 35% on mass basis with best possible efficiency at all loads in CE while it is 60% in the LHR engine. LHR engine with 60% alcohol induction showed improved performance when compared to CE with 35% alcohol induction**.** The maximum induction of alcohol is 35% in CE at 33^{o}bTDC, while it is 55% in LHR engine at 32^{o}bTDC. Performance, emission levels (smoke, NOx and aldehyde levels) and combustion characteristics are improved in both versions of the engine with maximum induction of ethanol, with advanced injection timings and with increase of injection pressure.

Acknowledgment

Authors thank authorities of Chaitanya Bharathi Institute of Technology, Hyderabad for providing facilities for carrying out research work. Financial assistance provided by All India Council for Technical Education (AICTE), New Delhi, is greatly acknowledged.

References

[1] P. Ramu and C. G. Saravanan, “Investigation of combustion and emission characteristics of a diesel engine with oxygenated fuels and thermal barrier coating”, *Energy Fuels*, Vol. 23, No. 2, pp. 653-656, (2009).

[2] H. Hazar, “Effects of bio-diesel on a low heat loss diesel engine”, *Renewable Energy*, Vol. 34**,** pp. 1533-1537, (2009).

[3] B. Rajendra Prasath, P. Tamilporai, And F. Mohd. Shabir,”Analysis of combustion, performance and emission characteristics of low heat rejection engine using bio-diesel”, *International Journal of Thermal Sciences,* Vol. 49, pp. 2483-2490, (2010).

[4] M. Mohamed Musthafa, S. P. Sivapirakasam, and M. Udayakumar, “Comparative studies on fly ash coated low heat rejection diesel engine on performance and emission characteristics fueled by rice bran and pongamia methyl ester and their blend with diesel”, *Energy*, Vol. 86, pp. 2343-2351, (2011).

[5] R. M. Cole, and A. C. Alkidas, “Evaluation of an Air-Gap-Insulated piston in a divided-chamber diesel engine”, *SAE Paper*, No. 850359, (1985).

[6] R. J. Moffatt, W. J. Griffiths, G. M. Dinnison, and Brunil, “Low heat loss pistons”, *SAE Paper*, No. 8, AE Group symposium, (1986).

[7] D. A. Parker, and G. M. Dennison, “The development of an air gap insulated piston”, *SAE Paper*, No. 870652, (1987).

[8] K. Rama Mohan, C. M. Vara Prasad, and M. V. S. Murali Krishna, “Performance of a low heat rejection diesel engine with air gap insulated piston”, *ASME Journal of Engineering for Gas Turbines and Power,*Vol. 121, pp. 530-540, (1999).

[9] M. V. S. Murali Krishna, *Performance evaluation of low heat rejection diesel engine with alternate fuels*, Ph. D. Thesis, J. N. T. University, Hyderabad, (2004).

[10] P. V. Krishna Murthy, *Studies on bio-diesel with LHR engine*, Ph. D. Thesis, J. N. T. University, Hyderabad, (2010).

[11] S. Sudhakar Babu, G. Ramachandra, V. Nagamalleswara Rao, Naga Prasad Naidu, "Investigations on the suitability of rapeseed oil in CI engine", *Proc. National Conf. on Advanced Trends in Mechanical Engg. Research & Development, JNT College of Engineering*, Anantapur, pp. 593-596, (2001).

[12] K. Pramanik, "Properties and use of jatropha curcas oil and diesel fuel blends in compression ignition engine", *Journal of Renewable Energy*, Vol. 28 (2), pp. 239- 48 (2003).

[13] S. Choudhury, P. K. Bose, Karanja or Jatropha, "A better option for an alternative fuel in C.I.Engine", *Proc. of the International Conf. & XX National Conf. on I.C. Engines and Combustion*, pp. 321-326 (2007).

[14] Rajesh Kumar Pandey, A. Rehman, and R. M. Sarvivy, "Karanja oil: The possibility of using as fuel in CI engine", *Proc. of the International Conf. & XX National Conference on I.C. Engines and Combustion*, pp. 235-239, (2007).

[15] S. Sinha and A. K. Agarawal, "Performance evaluation of a biodiesel (rice bran oil methyl ester) fuelled transportation diesel engine", *SAE. Paper*, No. 2005-01-1730, (2005).

[16] M. K. Gajendra Babu, C. Kumar, and M. Das. Lalit, "Experimental investigations on a karanja oil methyl ester fuelled DI diesel engine", *SAE Paper*, No. 2006-01-0238, (2006).

[17] J. Suryawanshi, "Performance and emission characteristics of CI engine fueled by coconut oil methyl ester", *SAE Paper*, No. 2006-32-0077, (2006)

[18] T. Ganapathy, R. Balasubranabian, R. P. Gakkhar, and K. Urgesan, "Use of Jatropha based biodiesel in DI diesel engines: critical Review", *Proc. of the International Conf. & XX National Conf. on I.C. Engines and Combustion*, pp. 229-234, (2007).

[19] A. Siva Kumar, D. Maheswar, K. Vijaya Kumar Reddy, "Comparison of performance parameters by using jatropha and fish oil as biodiesel", *Proc. of the International Conf. & XX National Conf. on I.C. Engines and Combustion*, pp. 235-239, (2007).

[20] M. Pugazhvadivu, and K. Jayachandran, "Investigations on the performance and exhaust emissions of a diesel engine using preheated waste frying oil as fuel", *Renewable Energy*, Vol. 30, No. 14, pp. 2189-2202, (2005).

[21] M. V. S. Murali Krishna and P. V. K. Murthy, "Studies on exhaust emissions from low heat rejection diesel engine with carbureted ethanol and jatropha oil", *Ecology, Environment &Conservation*, Vol. 14, No. 2, pp. 263-268, (2008).