Performance evaluation of Single Cylinder 4-Stroke electronically assisted Gasoline Direct Injection System –

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ABSTRACT:

The basic goals of the automotive industry; a high power, low specific fuel consumption, low emissions, low noise and better drive comfort. With increasing the vehicle number, the role of the vehicles in air pollution has been increasing significantly day by day. The environment protection agencies have drawn down the emission limits annually. Furthermore, continuously increasing price of the fuel necessitates improving the engine efficiency. Since the engines with carburetor do not hold the air fuel ratio close to the stoichiometric at different working conditions, catalytic converter cannot be used in these engines. Therefore these engines have high emission values and low efficiency. Electronic controlled Port Fuel Injection (PFI) systems instead of fuel system with carburetor have been used since 1980's. In fuel injection systems, induced air can be metered precisely and the fuel is injected in the manifold to air amount. By using the lambda sensor in exhaust system, air/fuel ratio is held of stable value. Fuel systems without electronic controlled it is impossible to comply with the increasingly emissions legislation.

The Direct Injection spark ignition (DISI) engines give a number of features, which could not be realized with port injected engines: avoiding fuel wall film in the manifold, improved accuracy of air/fuel ratio during dynamics, reducing throttling losses of the gas exchange by stratified and homogeneous lean operation, higher thermal efficiency by stratified operation and increased compression ratio, decreasing the fuel consumption and CO2 emissions, lower heat losses, fast heating of the catalyst by injection during the gas expansion phase, increased performance and volumetric efficiency due to cooling of air charge, better cold- start performance and better the drive comfort.

Added to the problems of fast dwindling resources of petroleum fuels and political factors, associated with their procurement, environmental pollution is another major problem with the petroleum fuels. Thus, the global oil crises, environment degradation, economic factors and the total human life dependence on the non-renewable fossil fuels, have created serious concern for alternative fuel research.

This in turn leads to search for alternative fuels that they themselves can produce. These alternative fuels preferably available from renewable sources. Therefore, attention is mainly focused towards the fuel made from waste engine oil called as LDO. When light diesel oil (LDO) is added with methanol and ethanol in proper proportion so that its properties will remain nearly to that of diesel, then it behaves like biomass based fuels.

Experimental results shown that diesel engine shows poor performance at lower compression ratio while running on LDO and its blends with diesel. Better performance of engine can be obtained.

KEYWORDS:

Engine efficiency, Direct Injection spark ignition (DISI) engine, CO2 emissions, LDO, Biomass based fuels, engine performance, emission parameters, engine performance etc.

INTRODUCTION

1.1 DIRECT INJECTION SPARK IGNITION ENGINE AND ALTERNATIVE FUELS.

Electronic controlled Port Fuel Injection (PFI) systems instead of fuel system with carburetor have been used. In fuel injection systems, induced air can be metered precisely and the fuel is injected in the manifold to air amount. By using the lambda sensor in exhaust system, air/fuel ratio is held of stable value. Fuel systems without electronic controlled it is impossible to comply with the increasingly emissions legislation.

If port fuel injection system is compared with carburetor system, it is seen that has some advantages. These are-

1. Lower exhaust emissions.
2. Increased volumetric efficiency and therefore increased output power and torque. The carburetor venturi prevents air and, in turn, volumetric efficiency decrease.
3. Low specific fuel consumption. In the engine with carburetor, fuel cannot be delivered the same amount and the same air/fuel ratio per cycle, for each cylinder.
4. The more rapid engine response to changes in throttle position. This increases the drive comfort.
5. For less rotation components in fuel injection system, the noise decreases.

Though the port fuel injection system has some advantages, it cannot be meet continuously increased the demands about performance, emission legislation and fuel economy, at the present day. The electronic controlled gasoline direct injection systems were started to be used instead of port fuel injection system.

The Direct Injection spark ignition (DISI) engines give a number of features, which could not be realized with port

injected engines: avoiding fuel wall film in the manifold, improved accuracy of air/fuel ratio during dynamics, reducing throttling losses of the gas exchange by stratified and homogeneous lean operation, higher thermal efficiency by stratified operation and increased compression ratio, decreasing the fuel consumption and CO2 emissions, lower heat losses, fast heating of the catalyst by injection during the gas expansion phase, increased performance and volumetric efficiency due to cooling of air charge, better cold- start performance and better the drive comfort.These alternative fuels preferably available from renewable sources. Therefore, attention is mainly focused towards the fuel made from waste engine oil called as LDO. When light diesel oil (LDO) is added with methanol and ethanol in proper proportion so that its properties will remain nearly to that of diesel, then it behaves like biomass based fuels.
LDO is one such promising fuel for direct for DISI engines, which has characteristics very close to diesel. They are also missible with diesel fuel in any proportion and can be used as diesel fuel extenders.

1.2 LIGHT DIESEL OIL AS FUEL FOR DISI ENGINE

Light Diesel Oil falls under class C category fuel having flash point above 66˚C. It is a blend of distillate components and a small amount of residual components. It is marketed under BIS 1460-2000 specification for Diesel fuels.LDO is used in lower RPM engines. It is used in lift irrigation pump sets, DG Sets and as a fuel in certain boilers and furnaces. It is Condensate of waste engine oil formed by heating it at red hot temperature in a boiler. Cold condensate then stored in a large tank so that impurities get settled down at Bottom and remaining will be the LDO. Market Price of LDO is 1KL = 55482 Rs (i.e. 1 litre = 55.48 Rs,A kilolitre (kL) is equivalent to one thousand litres.)Its characteristics and carbon content are nearly closer to that of Diesel.

2.2 COMPONENTS OF DISI ENGINE

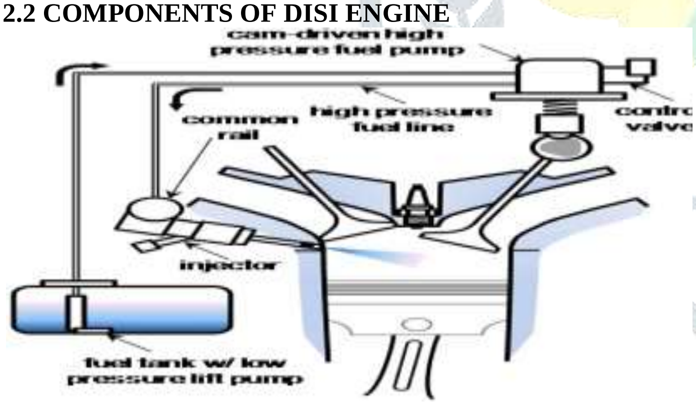


Fig. 2.1 Components of DISI engine

HIGH PRESSURE PUMP
INJECTOR
ENGINE SENSORS
2.2.1 HIGH PRESSURE PUMPS:

Fig. 2.2 shows fuel pump is a frequently (but not always) essential component on a car or other internal combustion engine device. Fuel has to be pumped from the fuel tank to the engine and delivered under low pressure to the carburetor or under high pressure to the fuel injection system. Fuel injected engines often use electric fuel pumps that are mounted inside the fuel.

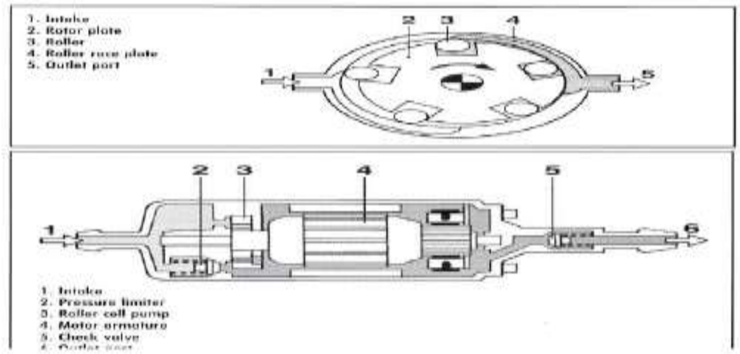
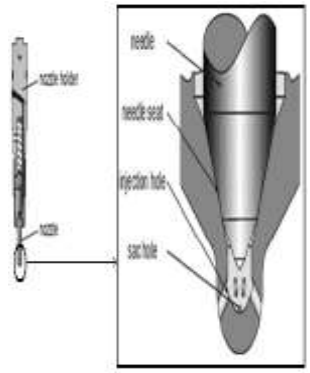


Fig. 2.2 High Pressure Pump

In many modern cars the fuel pump is usually electric and located inside of the fuel tank. The pump creates positive pressure in the fuel lines, pushing the gasoline to the engine .Placing the pump in the tank puts the component least likely to handle gasoline vapor well farthest from the engine, submersed in cool liquid. Another benefit to placing the pump inside the tank is that it is less likely to start a fire. Though electrical components can spark and ignite fuel vapors, liquid fuel will not explode. And therefore submerging the pump in the tank is one of the safest places to put it.

2.2.2 INJECTORS:

The solenoid-operated fuel injector is shown in the figure 2.3 below. It consists of a valve body and needle valve to which the solenoid plunger is rigidly attached. The fuel is supplied to the injector under pressure from the electric fuel pump passing through the filter. The needle valve is pressed against a seat in the valve body by a helical spring to keep it closed until the solenoid winding is energized. When the current pulse is received from the electronic control unit, a magnetic field builds up in the solenoid which attracts a plunger and lifts the needle valve from its seat. This opens the path to pressurized fuel to emerge as a finely atomized spray. The amount of fuel supplied to the engine is determined by the amount of time the fuel injector stays open. This is called the pulse width, and it is controlled by the ECU. The injectors are mounted in the intake manifold so that they spray fuel directly to cylinder. A pipe called the fuel rail supplies pressurized fuel to all of the injectors.



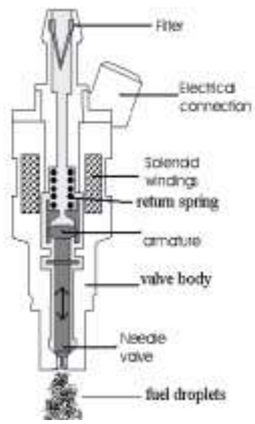


Fig 2.3 Cut section of the injector. Fig 2.4 Cross section of the nozzle tip.

2.2.3 ENGINE SENSORS:

In order to provide the correct amount of fuel for every operating condition, the engine control unit (ECU) has to monitor a huge number of input sensors.
Mass air flow sensor - Tells the ECU the mass of air entering the engine.

Oxygen sensor - The device measures the amount of oxygen in the exhaust gas and sends this information to the electronic control unit. If there is too much oxygen, the mixture is too lean. If there is too little, the mixture is too rich. In either case, the electronic control unit adjusts the air fuel ratio by changing the fuel injected. It is usually used with closed loop mode of the ECU.
Throttle position sensor- Monitors the throttle valve position (which determines how much air goes into the engine) so the ECU can respond quickly to changes, increasing or decreasing the fuel rate as necessary.
Coolant temperature sensor- Allows the ECU to determine when the engine has reached its proper operating temperature.
Voltage sensor- Monitors the system voltage in the car so the ECU can raise the idle speed if voltage is dropping (which would indicate a high electrical load).
Manifold absolute pressure sensor- Monitors the pressure of the air in the intake manifold. The amount of air being drawn into the engine is a good indication of how much power it is producing; and the more air that goes into the engine, the lower the manifold pressure, so this reading is used to gauge how much power is being produced.
Engine speed sensor - Monitors engine speed, which is one of the factors used to calculate the pulse width.
Crank angle sensor - Monitors the position of the piston and gives the information to the ECU. Accordingly the ECU adjusts the valve timing.

2.3 COMPOSITION OF LDO

Nomenclature of LDO as given by
LDO is a blend of distillate fuel with a small proportion of residual fuel.
Cetane number:-The most accurate method of assessing the ignition quality of a diesel fuel is by measuring its cetane number in a test engine, the higher the cetane number the higher the ignition quality.The cetane number of a fuel is defined as the percentage of cetane, arbitrarily given a cetane number of 100, in a blend with alphamethyl-naphthaline (cetane number -0), which is equivalent in ignition quality to that of the test fuel. Typical cetane number of LDO is around 35 – 38.
Viscosity:-Defined simply, viscosity means resistance to flow or movement. In metric system, centistoke is the unit for its measurement. It is function of time taken in seconds for a given volume of oil to flow through a calibrated viscometer under specified conditions. Viscosity depends on temperature and decreases as the temperature increases, so no numerical value has any meaning unless the temperature is specified.
Carbon residue:-Different fuels have different tendencies to crack and leave carbon deposits when heated under similar conditions. This property is normally measured by the Conradson or the Ramsbottom coke tests. In these tests, a sample of the fuel is heated without contact with air under specified conditions and the weight of carbon residue remaining after the test is expressed as a percentage of the weight of the sample.
Volatility:-As a rule, the higher the viscosity of a liquid fuel, the lower its volatility. Therefore provided the viscosity lies within specified limits, a satisfactory volatility is automatically ensured. However, the percentage recovered at some particular temperature e.g. 366˚ C, is specified in the case of HSD mainly to control engine fouling due to incomplete combustion of the higher boiling components.
Total sulphur:-This is significant because it governs the amount of sulphur oxides formed during combustion. Water from combustion of fuel collects on the cylinder walls, whenever the engine operates at low jacket temperatures. Under such conditions, sulphurous and sulphuric acids are formed, which attack the cylinder walls and piston rings, promote corrosion, and thus cause increased engine wear and deposits.
Typical Sulphur content in LDO is 1.8% w/w.
Acidity:-This should be low in order that corrosion of metals in contact with the fuel during storage and distribution is minimised.
Inorganic or mineral acidity: Where diesel fuels are treated with mineral acid as part of the refining procedure, traces of mineral acid remaining in the final product would obviously be undesirable. Hence, zero limit is usually specified for this property.
Organic acidity:This is due to the naphthenic type which are constituents of crude petroleum. Their presence in small amounts is not necessarily an indication of improper refining or poor quality. Although much weaker than mineral acids, they may attack galvanised metal and this is why the use of galvanised containers for the storage of diesel fuels is not recommended.

Ash content:-Ash is a measure of the incombustible material present in a fuel and is expressed as a percentage of the weight of the fuel sample. In the case of distillate fuels, it usually consists of rust, tank scale or sand, which settles out readily. Blends of distillate and residual fuel, e.g. LDO may additionally contain metal oxide derived from oil soluble and insoluble metallic compounds. Ash is significant because it can give rise to deposit problems such as abrasion, malfunctioning of injectors and high temperature corrosion, particularly with residual fuels. Typical Ash content is 0.02% w/w.

Sediment and water:-These are absolutely undesirable contaminants and should be as low as possible. The higher the specific gravity and viscosity of a fuel, the greater the quantities of water and sediment it can hold in suspension. Large quantities of sediment can affect the combustion of the fuel, and if abrasive, may cause excessive wear of closely fitting parts of fuel pumps and injectors. It may also clog filters and build up deposits in tanks and piping. Typical Water content in LDO should not exceed 0.25% w/w and sediment content should not exceed 0.1% w/w.

Calorific value:-Calorific value of a fuel is the quantity of heat generated in kilocalories by complete burning of one-kilogram weight of fuel. Gross calorific value is higher than net calorific value to the extent of heat required to change water formed by combustion into water vapours

Typical Gross Calorific Value of LDO varies between 10200-10800 Kcal/Kg.

Typical composition:

LDO has higher C/H ratio than Furnace Oil.

C- 85.5,H2-11.5,S-2,H2O-0.25,ASH-0.02

Properties:

Sr no.	Property	Gasoline	Diesel	LDO	Ethanol	Methanol
1	Composition by Wt in % Carbon Hydrogen Oxygen	84 16 Nil	86.1 11.7 0.3	85.5 11.5 Nil	52 13 35	37.5 12.5 50.0
2	Density Kg/m3	700-750	850-860	850-870	794	796
3	Gross Calorific Value (KJ/Kg)	42000	43200	42800	6400	4700
4	Flash point	30	40	66	78	65
5	Self Ignition Temp (˚C)	300-450	210	263	420	478
6	Octane Number	82	15-25	_	94	94
7	Cetane Number	8.14	41-45	35-38	8	3

2. 5 THE MIXTURE FORMATION AND OPERATION MODES IN THE DISI ENGINE

2.5.1 The Mixture Formation

An important operating criterion of a well-designed DISI engine shown in fig. 2.7 is that the fuel must be vaporized before the spark event occurs in order to limit UBHC emissions to an acceptable level. Moreover, the complete evaporation of the fuel can make the ignition process more robust. For a gasoline droplet with a diameter of 80 mm, vaporization under typical compression conditions takes tens of milliseconds, corresponding to more than a hundred crank angle degrees at an engine speed of 1500 rpm. By contrast, the vaporization of a 25 mm SMD droplet requires only several milliseconds, Mechanisms of air entrainment and spray contraction at elevated ambient pressure. Schematic of the outwardly opening, single-fluid, high pressure, swirl injector .corresponding to tens of crank angle degrees. This is the essence of the degradation of DISI engine combustion characteristics for sprays in which the droplets outer mean diameter exceeds 25 mm. The rapid vaporization of very small droplets helps to make the direct gasoline injection concept feasible. Therefore, many techniques have been proposed for enhancing the spray atomization of DISI injectors. The most common technique for DISI combustion systems is to use an elevated fuel pressure in combination with a swirl nozzle. The required fuel rail pressure level is generally on the order of 5.0 MPa, or in some cases up to 13 MPa, in order to atomize the fuel to the acceptable range of 15–25 mm SMD or less. The required spray characteristics and the minimum thresholds change significantly with the GDI engine operating conditions. In the case of fuel injection during the induction event, a widely dispersed fuel spray is generally required in order to achieve good air utilization for the homogeneous mixture. The impingement of the fuel spray on the cylinder wall should be avoided. For injection that occurs during the compression stroke, a compact spray with a reduced penetration rate is preferred in order to achieve a stratified mixture distribution. At the same time, the spray should be very well atomized since the fuel must vaporize in a very short time, even though fuel impingement on the bowl surface of a hot piston may promote vaporization. It may be seen that a suitable control of spray cone angle and penetration over the engine-operating map is advantageous, but is very difficult to achieve in practice. The in-cylinder droplet evaporation process was evaluated by Dodge using a spray model, and it was recommended that a mean droplet size of 15 mm SMD or smaller be utilized for DISI combustion systems. Based on calculation, a differential fuel pressure of at least 4.9 MPa is required for a pressure-swirl atomizer to achieve the required degree of fuel atomization. It was noted from the calculations that the additional time available with early injection does not significantly advance the crank angle positions at which complete droplet vaporization is achieved. This is because the high compression temperatures are very influential in vaporizing the droplets, and these temperatures occur near the end of the compression stroke. It was also noted that the atomization level that is utilized in some widely studied DISI engines may not be sufficient to avoid some excessive UBHC emissions due to reduced fuel evaporation rates that are associated with fuel impingement on solid surfaces. For homogeneous combustion in the SI engine, the combination of high turbulence intensity and low mean velocity at the spark gap is desirable. This is generally achieved for PFI engines, and also for DISI engines that operate exclusively in the early injection mode.

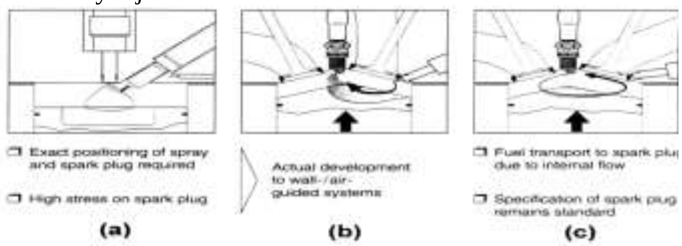


Fig 2.7 Classification of DISI combustion systems (a) spray-guided system (b) wall-guided system and (c) Air guided system.

2.6 OPERATION MODES

In response to driving conditions, the DISI engine changes the timing of the fuel spray injection, alternating between two distinctive combustion modes- stratified charge (Ultra-Lean combustion), and homogenous charge (Superior Output combustion).

2.9 VARIABLE COMPRESSION RATIO

Methods of reducing compression ratio

Low compression pistons : This seems to be the way to go. The pistons are much shorter than conventional ones. A small plus is that they are also often lighter so the engine will rev a little more freely. We would recommend combining low compression pistons with a shorter stroke to get the most benefit. The shape of the piston crown will also have a bearing on the amount of compression that takes place in the engine. This will require a strip down of the engine and whilst the engine is apart you may just as well perform some of the other modes listed below.

Shorter rods & reducing the stroke : A shorter stroke will have a dramatic effect on the compression ratio. By combining this method with low compression pistons one can start to think about running very high boost pressures when adding a turbo. The crank will also have some impact on the throw of the engine and the crank, piston crowns and rods should ideally all be matched up.

Head work : Again increases the volume of the cylinder but the effectiveness depends a lot on how the intake and exhaust valves are sited, and how much space there is for you to work with. Removing the head is relatively simple and does not require as much effort as other compression lowering modes, but it requires great skill to do a proper job on the head and achieve the lower compression ratio you are seeking.

Thicker head gaskets: This option is a bit of a budge, but we should mention it as a lot of people do run thicker gaskets to achieve a lower compression ratio. We have also seen people using 2 gaskets (or more) to achieve a lower compression ratio! Using multiple gaskets is certainly not recommended and introduces a major weak spot in an engine. A thicker gasket will reduce the compression ratio by a small fraction, probably only by .1 or .2. This is by far the easiest method of reducing compression but the risk is gasket failure and the gains in lower compression are minimal.

Decompression plates : Decompression plates are essentially an extension to the head and can be very effective at reducing the compression ratio. The block side needs a conventional gasket seal but the head side generally only requires a non setting high temperature sealant (in the case of aluminum decompression plates). Plates can be made of a variety of metals and we suggest you talk to a specialist about your options here. The decompression plates may fail prematurely in high boost applications where high temperatures are involved. Many view this as a good thing as replacing a decompression plate is a lot easier to do than replacing pistons and heads should they go, and in these extreme conditions this can be quite likely and the plate failure will have flagged up the potential problem for you.

OBJECTIVES

The functional objectives for fuel injection systems can vary. All share the central task of supplying fuel to the combustion process, but it is a design decision how a particular system is optimized. There are several competing parameters on which performance of modified engine (DISI engine) using LDO and its blends are compared with the conventional CI engine are-

- Power output
- Fuel efficiency
- Reliability
- Drivability and smooth operation
- Initial cost

WORKING AND CONSTRUCTION

3.1 SELECTED DIESEL ENGINE SPECIFICATION:

4 –Stroke Single Cylinder Diesel Engine

Manufacturers Name : Crown Engines, Power Developed : 3.5 H.P Or 2.611 KW

Speed : 1300 rpm, Bore Diameter : 85mm

Stroke Length : 78 mm, Compression Ratio :16.6 : 1

Capacity : 442.46cc (π/4 *d2 *L)

Fig. 3.1 Crown diesel engine.

3.2 VARIABLE COMPRESSION RATIO

Implemented method and procedure:

Fig.6.3 shows for reducing C.R (from 17 to 12) we select two method of them mention above, i.e. thicker head gas kit and decompression plate. Before fabricating decompression plate we calculate the actual thickness of plate we calculate thickness of plate for required C.R. calculation for required thickness of plate:

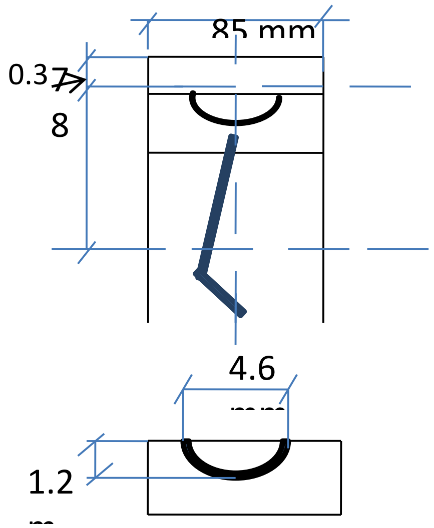


Fig. 3.2 Representation of cylinder & piston specification

L= Stroke Length = 7.8 cm
D= Diameter of cylinder = 8.5 cm
Lc= Length of compressed gasket in cm = 0 .3 cm
d = Cavity diameter= 4.6 cm
h = Height of cavity =1.8 cm
Where,

C.R = Compression ratio= $\frac{Vs+Vc}{Vc}$

$Vc = Vc1 + Vc2$

h = Height of cavity

$Vs = \frac{\pi}{4}\times D^2\times L = \frac{\pi}{4}\times 8.5^2\times 7.8$

$Vs = 442.61\ cm^3$

Vc1=Clearance volume of head gasket= $\frac{\pi}{4}\times D^2\times Lc$

$= \frac{\pi}{4}\times 8.5^2\times 0.1$

$Vc1 = 5.67\ cm^3$

Vc2= Clearance volume of dished type (hemispherical) cavity

$= \frac{\pi}{3}\times r^2\times (h+r) = \frac{\pi}{3}\times 2.3^2\times (1.8+2.3)$

$Vc2 = 22.71\ cm^3$

$Vc = Vc1 + Vc2 = 5.67 + 22.71 = 28.38\ cm^3$

Now,

$C.R = \frac{Vs+Vc}{Vc} = \frac{442.61+28.38}{28.38}$

C.R. = 16.54

Now,

We derived C.R. for Lc=0 .1 cm + 0.1 cm = 0.2 cm

Vc1=Clearance volume of head gasket= $\frac{\pi}{4}\times D^2\times L = \frac{\pi}{4}\times 8.5^2\times 0.2$

$Vc1 = 11.35\ cm^3$

$Vc = Vc1 + Vc2 = 11.35 + 22.71 = 34.06\ cm^3$

As

$C.R = \frac{Vs+Vc}{Vc}$

$= \frac{442.61+34.06}{34.06}$

C.R. = 13.99

Similarily, we can find out Various C.R. by increasing compressed gasket length by 0.1cm.

3.5 EXPERIMENTAL SET UP

RESULTS AND DISCUSSION

4.1 EFFECT ON ENGINE PERFORMANCE

Engine performance is an indication of the degree of success with which it is doing its assigned job i.e. conversion of chemical energy contained in the fuel into the useful mechanical work. The short term tests were performed on variable compression engine for diesel, Light Diesel Oil and their blends at two different compression ratios 16.5 and 14 and results are compared. The degree of success was compared on the basis of the following parameters at varying load (constant speed) and at compression ratio 16.5 & 14

1) Power output.
2) Thermal Efficiency.
3) Specific fuel consumption.
4) Brake mean effective pressure.

In this way, we can calculate the effect of load on parameters and also plot the graph of parameter at varying Load. Which are as follows.

4.1.1 EFFECT OF LOAD ON BRAKE POWER AND BRAKE MEAN EFFECTIVE PRESSURE AT COMPRESSION RATIO 16.5.

Since it's a constant speed engine, Brake power will remain constant for aparticular load but it increases with the increase in loading which are 0.6, 1.0, 1.6, & 2.0 KW and brake mean effective pressure are 1.25,2.08,3.34 and 4.17 bar for loading 3, 5, 8 & 10 respectively shown in figure 4.1 which validates with the theoretical concept and engine specification which is due to increase in fuel combustion with increase in load.

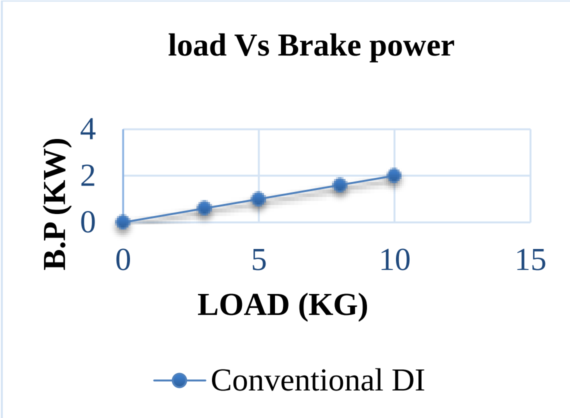


Fig : 4.1EFFECT OF LOAD ON BRAKE POWER AT COMPRESSION RATIO 16.5.

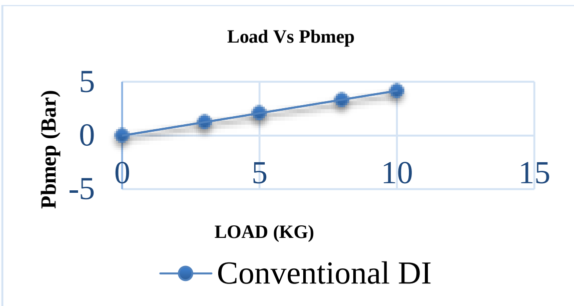


Fig :4.2EFFECT OF LOAD ON BRAKE MEAN EFFECTIVE PRESSURE AT COMPRESSION RATIO 16.5.

4.1.2 EFFECT OF LOAD ON B.S.F.C FOR LDO AT COMPRESSION RATIO 16.5

Variation in brake specific fuel consumption with variation in load for different fuels presented in figure 4.3.The specific fuel consumption for diesel was 0.56 ,0.38, 0.39 and 0.38 Kg/Kw-hr for load 3,5,8 and 10 respectively.This is the basis of calculation on which performance of LDO blends are compared.The specific fuel consumption is higher for pure LDO at C.R.16.5 and other blends than diesel. The increase in specific fuel consumption for pure LDO than that of diesel was 0.07, 0.06 and 0.1Kg/Kw-hr for load 3, 5 and 8 respectively. It was observed that specific fuel consumption increases with increase in load for pure LDO than that of diesel. The percentage increase in specific fuel consumption for pure LDO was 11.11, 11.36 and 26.5 for load 3,5 and 8 respectively. The maximum increased BSFC for pure LDO was 26.5 at 8 kg load than that of diesel. Similar trends are observed for 10%, 20% and 30% LDO. The increased specific fuel consumption for pure LDO and its blends may be due to higher calorific value of pure diesel .Higher density of pure LDO and its blends lead to more discharge of fuel for same plunger displacement in the fuel injection pump may be another reason for higher specific fuel consumption.

As there is no research with LDO and its blend thus we cannot exactly validate our results,but when compared with other research performance using other Biofuels such as esterified jatropha oil found very similar to our results which is around 20% increase in specific fuel consumption reported by S.P. Chincholkar [8]

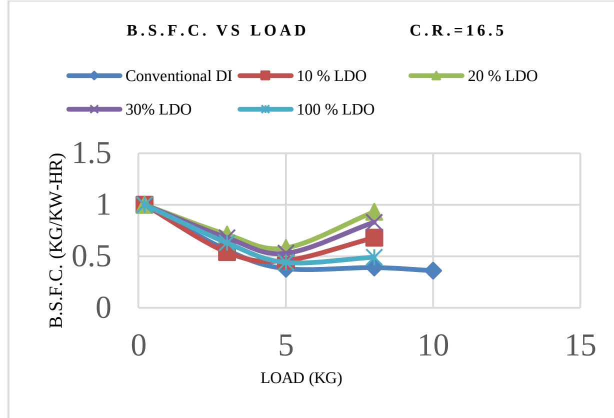


Fig: 4.3EFFECT OF LOAD ON B.S.F.C FOR LDO AT COMPRESSION RATIO 16.5

4.1.3 EFFECT OF LOAD ON BRAKE THERMAL EFFECIENCY FOR LDO AT COMPRESSION RATIO 16.5.

Variation in brake thermal efficiency with variation in load for different fuels presented in figure 4.4.The brake thermal efficiency for diesel was 15.13%, 22.41%, 21.51 and 23.53% for load 3, 5, 8 and 10 respectively. This is the basis of calculation on which performance of LDO blends are compared. Brake thermal efficiency is higher for pure LDO at C.R.16.5 and other blends than diesel. The decrease in brake thermal efficiency for pure LDO than that of diesel was 1.69%, 3.08% and 4.26%. for load 3, 5 and 8 respectively. It was observed that brake thermal efficiency decreases with increase in load for pure LDO than that of diesel. The percentage decrease in brake thermal efficiency for pure LDO was 12.57, 15.95 and 24.69 for load 3, 5 and 8 respectively. The maximum decreased brake thermal efficiency for pure LDO was 24.69 at 8 kg load than that of diesel. Similar trends are observed for 10% , 20% and 30% LDO. The decreased brake thermal efficiency for pure LDO and its blends may be due to higher calorific value of pure diesel .Higher density of pure LDO and its blends lead to more discharge of fuel for same plunger displacement in the fuel injection pump may be another reason for lower brake thermal efficiency. As there is no research with LDO and its blend thus we cannot exactly validate our results, but when compared with other research performance using other Biofuels such as esterified jatropha oil found very similar to our results which is around 13% decrease in brake thermal efficiency reported by S.P. Chincholkar [8].

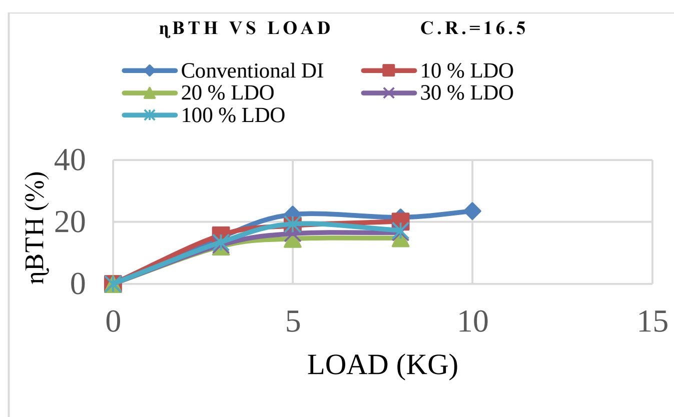


Fig : 4.4 EFFECT OF LOAD ON BRAKE THERMAL EFFECIENCY FOR LDO AT COMPRESSION RATIO 16.5.

1.4 EFFECT OF LOAD ON B.S.F.C FOR RAW OIL AT COMPRESSION RATIO 16.5.

Variation in brake specific fuel consumption with variation in load for different fuels presented in figure 4.5.The specific fuel consumption for diesel was 0.56, 0.38, 0.39 and 0.38 Kg/Kw-hr for load 3,5,8 and 10 respectively. This is the basis of calculation on which performance of LDO blends are compared. The specific fuel consumption is higher for pure LDO at C.R.16.5 and other blends than diesel. The increase in specific fuel consumption for pure LDO than that of diesel were 0.02, 0.02 and 0.06 Kg/Kw-hr for load 3, 5 and 8 respectively. It was observed that specific fuel consumption increases with increase in load for pure LDO than that of diesel. The percentage increase in specific fuel consumption for pure LDO was 3.44, 5.0 and 13.3 for load 3,5 and 8 respectively. The maximum increased BSFC for pure LDO was 13.3 at 8 kg load than that of diesel. Similar trends are observed for 10% , 20% and 30% LDO. The increased specific fuel consumption for pure LDO and its blends may be due to higher calorific value of pure diesel. Higher density of pure LDO and its blends lead to more discharge of fuel for same plunger displacement in the fuel injection pump may be another reason for higher specific fuel consumption.

As there is no research with LDO and its blend thus we cannot exactly validate our results, but when compared with other research performance using other Biofuels such as esterified jatropha oil found very similar to our results which is around 20% increase in specific fuel consumption reported by S.P. Chincholkar [8]

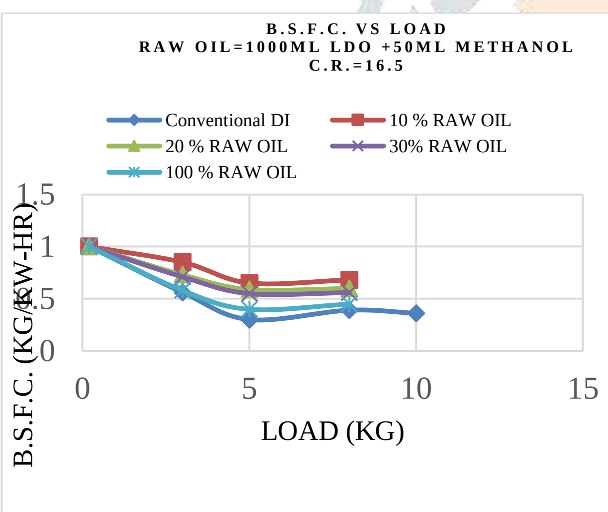


Fig :4.5EFFECT OF LOAD ON B.S.F.C FOR RAW OIL AT COMPRESSION RATIO 16.5.

4.1.5 EFFECT OF LOAD ON BRAKE THERMAL EFFECIENCY FOR RAW OIL AT COMPRESSION RATIO 16.5.

Variation in brake thermal efficiencywith variation in load for different fuels presented in figure 4.6.The brake thermal efficiencyfor diesel was 15.13%, 22.41%, 21.51 and 23.53% for load 3, 5, 8 and 10 respectively. This is the basis of calculation on which performance of LDO blends are compared. Brake thermal efficiencyis higher for pure LDO at C.R.16.5 and other blends than diesel. The decrease in brake thermal efficiencyfor pure LDO than that of diesel was 0.42%, 0.98% and 2.69%. for load 3, 5 and 8 respectively. It was observed that brake thermal efficiency decreases with increase in load for pure LDO than that of diesel. The percentage decrease in brake thermal efficiencyfor pure LDO was 2.8, 4.5 and 14.29 for load 3,5 and 8 respectively. The maximum decreased brake thermal efficiencyfor pure LDO was 14.29 at 8 kg load than that of diesel. Similar trends are observed for 10% , 20% and 30% LDO. The decreased brake thermal efficiencyfor pure LDO and its blends may be due to higher calorific value of pure diesel.Higher density of pure LDO and its blends lead to more discharge of fuel for same plunger displacement in the fuel injection pump may be another reason for lower brake thermal efficiency.

As there is no research with LDO and its blend thus we cannot exactly validate our results,but when compared with other research performance using other Biofuels such as esterified jatropha oil found very similar to our results which is around 13% decrease in brake thermalefficiency reported by S.P. Chincholkar [8]

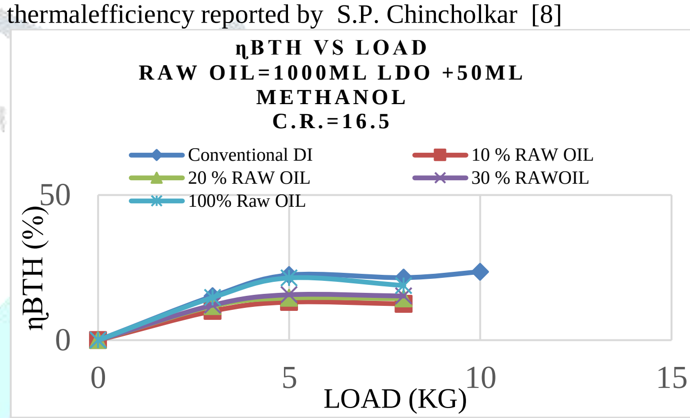


Fig :4.6 EFFECT OF LOAD ON BRAKE THERMAL EFF. FOR RAW OIL AT COMPRESSION RATIO 16.5.

4.1.8 EFFECT OF LOAD ON B.S.F.C FOR LDO AT COMPRESSION RATIO 14.

Variation in brake specific fuel consumption with variation in load for different fuels presented in figure 4.9.The specific fuel consumption for diesel was 0.56 ,0.38, 0.39 and 0.36 Kg/Kw-hr for load 3, 5, 8 and 10 respectively at C.R. 16.5.This is the basis of calculation on which performance of LDO blends are compared. The specific fuel consumption is higher for pure LDO at C.R.16.5 and other blends than diesel. The increase in specific fuel consumption for pure LDO than that of diesel were 0.11, 0.12 and 0.16 Kg/Kw-hr for load 3, 5 and 8 respectively. It was observed that specific fuel consumption increases with increase in load for pure LDO than that of diesel. The percentage increase in specific fuel consumption for pure LDO was 16.42, 31.58 and 29.01 for load 3,5 and 8 respectively. The maximum increased BSFC for pure LDO was 31.58 at 5 kg load than that of diesel. Similar trends are observed for 10% , 20% and 30% LDO. The increased specific fuel consumption for pure LDO and its blends may be due to higher calorific value of pure diesel .Higher density of pure LDO and its blends lead to more discharge of fuel for same plunger displacement in the fuel injection pump may be another reason for higher specific fuel consumption.

The increased BSFC at lower C.R. for pure LDO may be due to the incomplete combustion of fuel. Less combustion time and less compression temperature achieved may be the other reason for higher specific fuel consumption at C.R. 14.and it is well known fact that engine gives better performance for higher compression ratio.

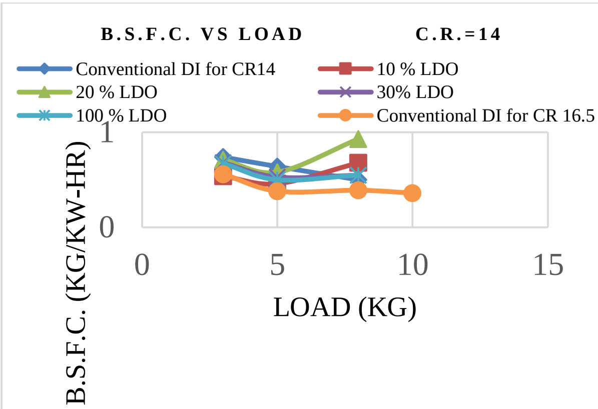


Fig :4.9EFFECT OF LOAD ON B.S.F.C FOR LDO AT COMPRESSION RATIO 14

4.1.9 EFFECT OF LOAD ON BRAKE THERMAL EFF. FOR LDO AT COMPRESSION RATIO 14.

Variation in brake thermal efficiencywith variation in load for different fuels presented in figure 4.10. The brake thermal efficiencyfor diesel was 15.13%, 22.41%, 21.51 and 23.53% for load 3, 5, 8 and 10 respectively at CR 16.5. This is the basis of calculation on which performance of LDO blends are compared. Brake thermal efficiencyis higher for pure LDO at C.R.16.5 and other blends than diesel. The decrease in brake thermal efficiencyfor pure LDO than that of diesel was 1.76%, 5.52% and 3.95%. for load 3, 5 and 8 respectively. It was observed that brake thermal efficiency decreases with increase in load for pure LDO than that of diesel. The percentage decrease in brake thermal efficiencyfor pure LDO was 12.41, 32.68 and 22.49 for load 3,5 and 8 respectively. The maximum decreased brake thermal efficiencyfor pure LDO was 32.68 at 5 kg load than that of diesel. Similar trends are observed for 10% , 20% and 30% LDO. The decreased brake thermal efficiencyfor pure LDO and its blends may be due to higher calorific value of pure diesel.Higher density of pure LDO and its blends lead to more discharge of fuel for same plunger displacement in the fuel injection pump may be another reason for lowerbrake thermal efficiency.

The decreased brake thermal efficiency at lower C.R. for pure LDO may be due to the incomplete combustion of fuel. Less combustion time and less compression temperature achieved may be the other reason for lower brake thermal efficiency at C.R. 14.and it is well known fact that engine gives better performance for higher compression ratio.

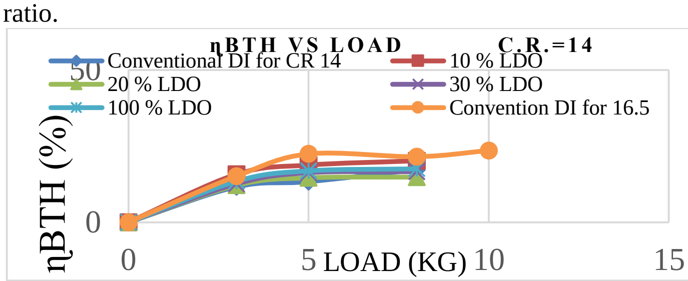


Fig :4.10EFFECT OF LOAD ON BRAKE THERMAL EFFECIENCY FOR LDO AT COMPRESSION RATIO 14.

4.1.10EFFECT OF LOAD ON B.S.F.C FOR RAW OIL AT COMPRESSION RATIO 14.

Variation in brake specific fuel consumption with variation in load for different fuels presented in figure 4.11.The specific fuel consumption for diesel was 0.56 ,0.38, 0.39 and 0.38 Kg/Kw-hr for load 3,5,8 and 10 respectivelyat C.R.16.5.This is the basis of calculation on which performance of LDO blends are compared.The specific fuel consumption is higher for pure LDO at C.R.14 and other blends than diesel at C.R.16.5. The increase in specific fuel consumption for pure LDO than that of diesel was 0.09, 0.08 and 0.15 Kg/Kw-hr for load 3, 5 and 8 respectively. It was observed that specific fuel consumption increases with increase in load for pure LDO than that of diesel. The percentage increase in specific fuel consumption for pure LDO was 13.85 ,17.39 and 38.46 for load 3,5 and 8 respectively. The maximum increased BSFC for pure LDO was 38.46 at 8 kg load than that of diesel. Similar trends are observed for 10% , 20% and 30% LDO. The increased specific fuel consumption for pure LDO and its blends may be due to higher calorific value of pure diesel .Higher density of pure LDO and its blends lead to more discharge of fuel for same plunger displacement in the fuel injection pump may be another reason for higher specific fuel consumption.

The increased BSFC at lower C.R. for pure LDO may be due to the incomplete combustion of fuel. Less combustion time and less compression temperature achieved may be the other reason for higher specific fuel consumption at C.R. 14.and it is well known fact that engine gives better performance for higher compression ratio.

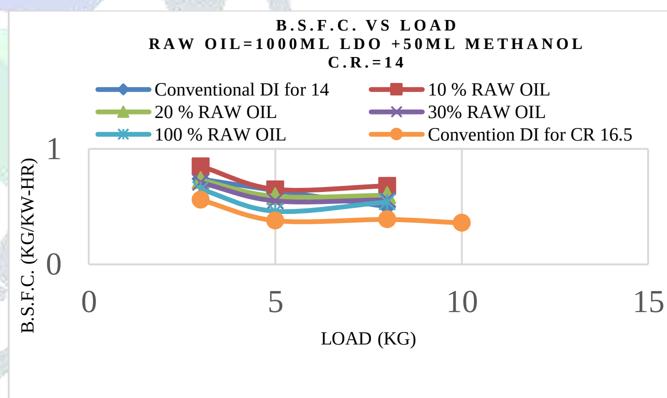


Fig: 4.11EFFECT OF LOAD ON B.S.F.C FOR RAW OIL AT CR 14

4.1.11 EFFECT OF LOAD ON BRAKE THERMAL EFFECIENCY FOR RAW OIL AT COMPRESSION RATIO 14.

Variation in brake thermal efficiency with variation in load for different fuels presented in figure 4.10. The brake thermal efficiency for diesel was 15.13%, 22.41%, 21.51 and 23.53% for load 3, 5, 8 and 10 respectively at CR 16.5. This is the basis of calculation on which performance of LDO blends are compared. Brake thermal efficiency is higher for pure LDO at C.R.14 and other blends than diesel C.R.16.5. The decrease in brake thermal efficiency for pure LDO than that of diesel was 2.63%, 5.81% and 9.37% for load 3, 5 and 8 respectively. It was observed that brake thermal efficiency decreases with increase in load for pure LDO than that of diesel. The percentage decrease in brake thermal efficiency for pure LDO was 21.10, 35.00 and 77.18 for load 3, 5 and 8 respectively. The maximum decreased brake thermal efficiency for pure LDO was 77.18 at 8 kg load than that of diesel. Similar trends are observed for 10%, 20% and 30% LDO. The decreased brake thermal efficiency for pure LDO and its blends may be due to higher calorific value of pure diesel. Higher density of pure LDO and its blends lead to more discharge of fuel for same plunger displacement in the fuel injection pump may be another reason for lower brake thermal efficiency.

The decreased brake thermal efficiency at lower C.R. for pure LDO may be due to the incomplete combustion of fuel. Less combustion time and less compression temperature achieved may be the other reason for lower brake thermal efficiency at C.R. 14.and it is well known fact that engine gives better performance for higher compression ratio.

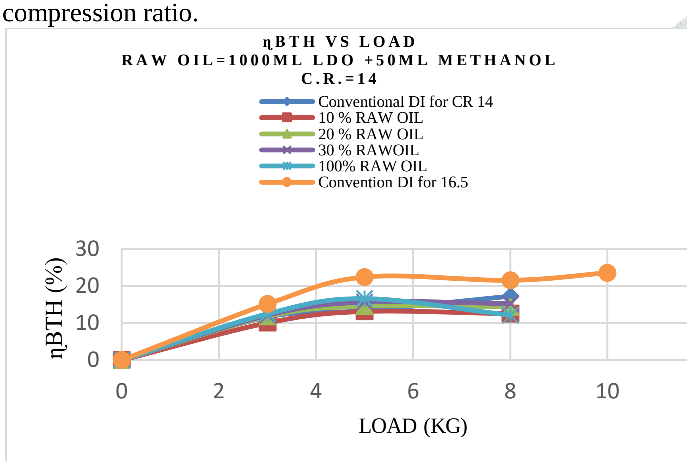


Fig: 4.12EFFECT OF LOAD ON BRAKE THERMAL EFF. FOR RAW OIL AT COMPRESSION RATIO 14.

4.1.12EFFECT OF LOAD ON B.S.F.CFOR B1 OIL AT COMPRESSION RATIO 14.

Variation in brake specific fuel consumption with variation in load for different fuels presented in figure 4.13.The specific fuel consumption for diesel was 0.56, 0.38, 0.39 and 0.38 Kg/Kw-hr for load 3,5,8 and 10 respectively. This is the basis of calculation on which performance of LDO blends are compared. The specific fuel consumption is higher for pure LDO at C.R.14 and other blends than diesel C.R.16.5. The increase in specific fuel consumption for pure LDO than that of diesel were 0.02, 0.04 and 0.15Kg/Kw-hr for load 3, 5 and 8 respectively. It was observed that specific fuel consumption increases with increase in load for pure LDO than that of diesel. The percentage increase in specific fuel consumption for pure LDO was 3.4, 10.53 and 38.46 for load 3,5 and 8 respectively. The maximum increased BSFC for pure LDO was 38.46 at 8 kg load than that of diesel. Similar trends are observed for 10%, 20% and 30% LDO. The increased specific fuel consumption for pure LDO and its blends may be due to higher calorific value of pure diesel .Higher density of pure LDO and its blends lead to more discharge of fuel for same plunger displacement in the fuel injection pump may be another reason for higher specific fuel consumption.

The increased BSFC at lower C.R. for pure LDO may be due to the incomplete combustion of fuel. Less combustion time and less compression temperature achieved may be the other reason for higher specific fuel consumption at C.R. 14.and it is well known fact that engine gives better performance for higher compression ratio.

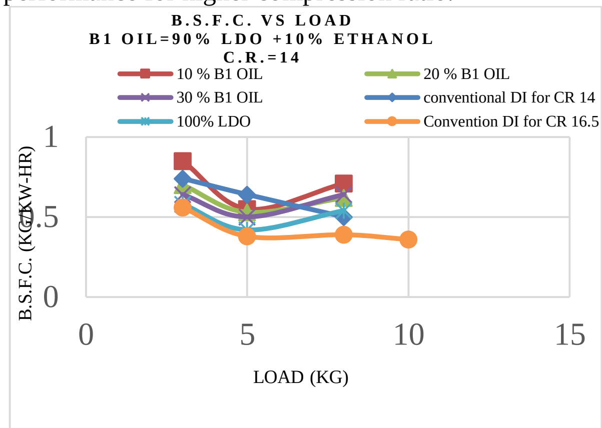


Fig: 4.13EFFECT OF LOAD ON B.S.F.CFOR B1 OIL AT COMPRESSION RATIO 14.

CONCLUSION

When the results found of this study following conclusion we get

1. Is Light diesel oil is one such promising fuel for Direct Injection Spark Ignition engines, Compare characteristics with diesel.
2. LDO as a fuel can be very efficient, cheap and easily available as it is made from the waste engine oil. India is the fastest growing country when we relate with the use of motor vehicles, it will not a big problem of making LDO from waste engine oil by extending extraction facilities and adopting suitable technology.
3. A comparision of physical and fuel properties of LDO with those of diesel fuel indicates that the LDO are quite similar in nature to diesel fuel. The higher flash point of LDO made safe storage and handling of these oil.
4. Experimental results will shows that diesel engine shows the performance of LDO and its blends with diesel.
5. When compare to C.I. Engine , LDO and its blend performance

FUTURE SCOPE:

According to the investigation reported in this topic, the following recommendation could be offered for the future study and research

1. DISI engine is used in multi cylinder engine for 4 wheelers uses fuel as gasoline which gives higher performance rate at comparatively lower fuel consumption which needs a very complex functioning of ECU. If ECU functioning make simpler so that it can be used in two wheelers also at minimum cost.
2. The results of the short duration test performance of an engine on LDO were found very much encouraging, study on endurance test should be carried out using neat LDO and its blends with diesel.
3. The environmental impact of LDO should be studied.
4. Engine modification in engine system should be tried to suit other biofuels.

5. Economics of biodiesel is very important issue and it should be carried out.
6. Storing and handling of biodiesel is equally important and it needs further research .
7. Further study for engine deposits, engine performance and crank oil dilution should be carried out.

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