Journal for Educators, Teachers and Trainers
The PROFESIOLab electronic, peer-reviewed, open-access Magazine

ISSN 1989-9572

DOI: 10.47750/jett.2022.13.06.078

Investigating the Impact of EGR on Performance and Emissions in a CI Engine Using Diesel and Jatropha Oil Biodiesel Blends

[1]*Dr. SRINIVASULU PULLURU,*
[2]*K. SATHEESH KUMAR,*
[3]*KRISHNA MURTHY UPPULA,*
[4]*MANDALA VENKATESH,*
[5]*DAMERA SRILATHA*

Journal for Educators, Teachers and Trainers, Vol.13 (6)

https://jett.labosfor.com/

Date of Reception: 20 Oct 2022

Date of Revision: 18 Nov 2022

Date of Acceptance: 12 December 2022

***Dr. SRINIVASULU PULLURU, K. SATHEESH KUMAR, KRISHNA MURTHY UPPULA,MANDALA VENKATESH, DAMERA SRILATHA* (2022). Investigating the Impact of EGR on Performance and Emissions in a CI Engine Using Diesel and Jatropha Oil Biodiesel Blends. Journal for Educators, Teachers and Trainers, Vol.13(6).791-811.**

Journal for Educators, Teachers and Trainers
The LabOSfor electronic, peer-reviewed, open-access Magazine

Journal for Educators, Teachers and Trainers
The PROFESIOLab electronic, peer-reviewed, open-access Magazine

Journal for Educators, Teachers and Trainers, Vol. 13(6)

ISSN1989 –9572

https://jett.labosfor.com/

Investigating the Impact of EGR on Performance and Emissions in a CI Engine Using Diesel and Jatropha Oil Biodiesel Blends

[1]Dr. SRINIVASULU PULLURU,[2]K. SATHEESH KUMAR,[3]KRISHNA MURTHY UPPULA, [4]MANDALA VENKATESH ,[5]DAMERA SRILATHA

[1]Professor, [23]Assistant Professor, [4,5]B. Tech Student
Department Of Mechanical Engineering
Vaagdevi College of Engineering, Warangal, Telangana.

ABSTRACT

This study systematically investigates the effects of Exhaust Gas Recirculation (EGR) on the performance and emissions of a compression ignition (CI) engine fueled with diesel and various blends of Jatropha oil biodiesel. EGR is employed to reduce nitrogen oxide (NOx) emissions and improve combustion efficiency. The research involves experimental analysis under different EGR rates and engine operating conditions. Performance metrics such as brake power, torque, and thermal efficiency are evaluated alongside emissions data, including NOx, carbon monoxide (CO), and unburned hydrocarbons (UHC). The results demonstrate that optimal EGR levels can enhance the performance of CI engines while significantly reducing harmful emissions, particularly when using Jatropha oil biodiesel blends. This study contributes to the ongoing efforts in promoting sustainable biofuels and optimizing CI engine performance, highlighting the potential of EGR as an effective strategy for improving the environmental footprint of biodiesel-powered engines.

Key Words: Diesel, sustainability, Alternate fuels, emissions, exhaust gas recirculation etc.

INTRODUCTION

Diesel may use a variety of vegetable oils, including as sunflower oil, rapeseed oil, linseed oil, and peanut oil. Using vegetable oil has many advantages, including improved agriculture, regional development, sustainability, and lower greenhouse gas emissions.Vegetable oils' chemical makeup helps to reduce the emissions of undesirable substances during combustion. Vegetable oil fuels have engine performance and exhaust gas emission levels for short-term operation only, but with prolonged use, they cause carbon deposit buildup and piston ring sticking, claim Murayama et al.On land and at sea, diesel engines have long been the most affordable form of commercial transportation. However, lowering emissions from diesel engines is a big problem, which is driving attempts to create special in-cylinder platforms that work with treatment equipment and, to the maximum degree possible, look for alternative fuels.India's demand for diesel is around "6" times higher than that of the rest of the world, hence searching for alternatives to diesel is a natural decision. The greatest alternatives to diesel are esters made from vegetable oils as they don't alter the engine. The chemical makeup of vegetable oils aids in lowering the emissions of undesirable substances during combustion.

According to Murayama et al., vegetable oil fuels resulted in acceptable exhaust gas emission levels and engine performance for short-term usage only, but after prolonged use, they caused carbon fund accumulation and piston ring sticking. Along with offering workable answers to these issues, they recommended raising the fuel's temperature to 2200 degrees Celsius, converting 25% diesel fuel into vegetable oil, and combining 15% eth oils.While needing little engine modification and fuel processing, the mixing approach boosts the efficiency of vegetable oil fuel. Due to increased fuel use and pollution from industrial and vehicle emissions, biodiesel has become a competitive alternative to petro-diesel. Natural gas, coal, and petrochemical resources provide most of the energy needed. It will soon run out at the present pace of use. Diesel engines may run on a variety of vegetable oils, such as sunflower oil, rapeseed oil, linseed oil, and peanut oil. Using vegetable oil has many advantages, including improved agriculture, regional development, sustainability, and lower greenhouse gas emissions.

BIOFUELS

A biofuel is a fuel made from living things, usually plants or components produced from plants. Because of its ability to lessen the many environmental stressors brought on by the usage of fossil fuels, liquid biofuels are being investigated more and more as fossil fuel substitutes for diesel and petrol. Biodiesel may be used as a diesel fuel alternative, either alone or in combination with diesel oil, due to its comparable fuel qualities (Pasias et al., 2206). It performs better than diesel made from petroleum in the following ways: It is biodegradable, carbon neutral, and sustainable. quicker, less harmful, with a greater flash point, and with less sulphur. The main reason cucurbitapepo is planted is for its leaves and fruits, which are widely eaten as vegetables in Nigeria. These days, seed oil has no useful product. The oil from cucurbitapepo seeds seems to be a good alternative for making biodiesel. When mixed with honey or olive oil, the seed oil may be used to season a salad. Oil is seldom, if ever, used in cooking since boiling eliminates its important fatty acid (Schinas et al., 2209).

BIO-DIESEL

One kind of alternative fuel that is specifically designed for diesel engines is biodiesel, which is derived from vegetable or animal fat.

Biodiesel is an environmentally beneficial, clean-burning, organic fuel that is nontoxic. Any compression ignition engine (diesel engine) may utilise it.

Since it is not petroleum-based, fossil fuels like coal or oil are not used in its production. Vegetable oils and fats are among the renewable materials used to make this eco-friendly fuel. These oils and fats undergo transesterification to provide a fuel that may be used straight in any diesel engine without any changes or mixed in any ratio with regular petroleum-based diesel.

Transformative change:It is the process of changing an ester's organic gang R″ into an alcohol's organic collective R′. Acidic or basic catalysts are often used to catalyse these reactions. The process may also be carried out by lipases and other enzymes, sometimes known as biocatalysts.The reaction of a triglyceride (fat or oil) with an alcohol to produce esters and glycerol is known as enzymatic hydrolysis. Three long-chain fatty acids joined by a glycerin molecule form a triglyceride. The characteristics of fat are determined by the kind of fatty acids that are connected to glycerin. The qualities of biodiesel may be affected by the kind of fatty acids utilised. A catalyst, usually a strong alkaline like sodium hydroxide, is present when the triglyceride and alcohol react during the transesterification process. The single-ester, often referred to as crude glycerol and biodiesel, is created when the alcohol and fatty acid combine.

RENEWABLE RESOURCES & NON-RENEWABLE SOURCES

EGR – EXHAUST GAS RECIRCULATION – WORKING

The operation of this process is straightforward, but the implications are profound. After the exhaust manifold and before the catalytic converter, there is an EGR-valve. This valve opens, allowing some of the exhaust gases to be redirected to the intake manifold and mixed with fresh air. Once mixed, the supply air in the fresh air is reduced or the temperature of the fresh air is slightly raised. Because flue gases have already been burned, they are now inert gases, with no free oxygen present. As a result, the nitrogen in the air is unable to react with the excess oxygen, and NOx (Nitrogen Oxides) establishment is reduced.This is one of the most dangerous gases in a tail pipe and must be controlled according to emission laws. Because these gases are static in character, they prevent ignition from reaching high temperatures, where all of the dangerous toxic gases are formed. To even further reduce the temperature of the burning, an EGR cooler is often attached to the modules before mixing it with fresh air.

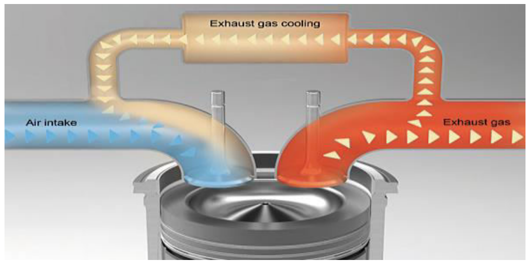

Fig 1: Exhaust gas recirculation representation

EGR – EXHAUST GAS RECIRCULATION – PRINCIPLE AND NEED

Exhaust Gas Recirculation, as the name implies, is the redirection of exhaust gases after the combustion process and their use to achieve various goals in the vehicle's engine. This could include anything including controlling the temperature inside the combustion chamber to improving a motor's fuel mileage and everywhere in between. As we all know, a quality of the air inside the combustion chamber, the ignition process, and the pressure and temperature inside the engine determine all of the performance, fuel, and emission parameters and output of a car; adjusting these could result in a slew of advantages in all of these areas. That is the reason why so much research has gone into these aspects and new technologies keep coming up as the emission and performance requirements keep on increasing. Governments are always bringing in tougher emission regulations to keep the environmental pollution in check and the customers are looking for more powerful or more fuel-efficient cars at the same time.

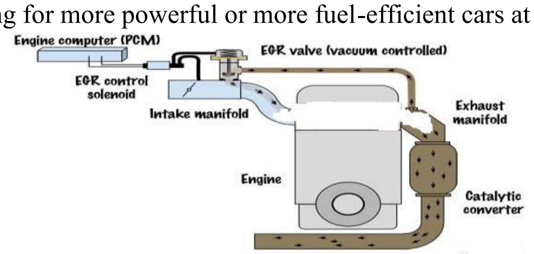

Fig 2: Exhaust gas recirculation principle

EGR – EXHAUST GAS RECIRCULATION – ADVANTAGES

Now, as mentioned before, there are a ton of benefits to incorporating this technology into vehicles. A lot of work has been done on this technology and it is a perfectly reliable technology. Many modern cars make use of this technique along with advanced components like the EGR cooler and sensors to take even more control of the exhaust gases. The main challenge in this is to keep a track ofAs previously stated, there are numerous advantages to incorporating this technology into vehicles. This technology was put through extensive testing and is completely reliable. Many modern cars use this technique in conjunction with advanced components such as the EGR cooler and sensors to gain even more control over the exhaust gases.exhaust gas mass and recirculate according to the requirements. Once this is accomplished, EGR offers numerous benefits such as reduced fuel consumption, control of temperature and pressure inside the combustion chamber, reduction of harmful and toxic pollutants, and much more. Under low load, the ignition lag of diesel engines is contained. There aren't many drawbacks to this technique, but ignition knock may be an issue in the case of gasoline engines. Because the addition of flue gases raises the temperature slightly. Another consideration could be the addition of components and the maintenance of ti. As a result, caution is required. Apart from that, this technology only has advantages.

LITERATURE REVIEW

The investigation of exhaust gas recirculation (EGR) in compression ignition (CI) engines has garnered significant attention due to its potential to reduce harmful emissions and enhance engine performance. EGR operates by recirculating a portion of the engine's exhaust back into the combustion chamber, which lowers the combustion temperature and dilutes the fuel-air mixture. This mechanism has been extensively studied in various contexts, particularly in relation to diesel engines and alternative fuels.

1. EGR in Compression Ignition Engines

Research indicates that the implementation of EGR can effectively reduce nitrogen oxide (NOx) emissions, which are a major contributor to air pollution and smog formation. According to Kato et al. (2016), the introduction of EGR results in lower peak combustion temperatures, thereby minimizing NOx formation. Furthermore, studies by Zhang et al. (2017) demonstrated that optimizing EGR rates can improve brake thermal efficiency while simultaneously reducing particulate matter emissions.

2. Biodiesel as an Alternative Fuel

Biodiesel has emerged as a viable alternative to traditional fossil fuels due to its renewable nature and potential for lower emissions. Jatropha oil, derived from the seeds of the Jatropha curcas plant, has gained attention for its favorable characteristics, including high lipid content and suitability for cultivation in arid regions. Research by Sinha et al. (2018) highlighted the benefits of Jatropha biodiesel, which include reduced carbon emissions and improved biodegradability compared to petroleum-based diesel.

3. Blends of Diesel and Jatropha Biodiesel

The performance and emission characteristics of CI engines running on blends of diesel and Jatropha biodiesel have been explored in several studies. Kumar et al. (2019) investigated the effects of varying biodiesel percentages on engine performance, finding that blends of up to 20% Jatropha biodiesel resulted in comparable power output and torque to pure diesel while significantly reducing CO and UHC emissions. However, they noted that higher biodiesel content could lead to increased fuel viscosity, which may affect fuel atomization and combustion efficiency.

4. Combined Effects of EGR and Biodiesel Blends

The synergistic effects of EGR and biodiesel blends have also been the subject of recent research. According to Ali et al. (2020), combining EGR with Jatropha biodiesel blends leads to significant improvements in emissions profiles. Their findings indicated that optimal EGR rates not only reduced NOx emissions but also enhanced the thermal efficiency of the engine. Moreover, the study revealed that EGR effectively mitigated the increase in NOx emissions typically associated with high biodiesel blends.

Property	Jatropha oil	Jatropha biodiesel	Diesel
Density(15℃, kgm^{-3})	940	880	850
Viscosity (mm^2s^{-1})	24.5	4.8	2.6
Flash point (℃)	225	135	68
Pour point (℃)	4	2	-20
Water content (%)	1.4	0.025	0.02
Ash content (%)	0.8	0.012	0.01
Carbon residue (%)	1.0	0.20	0.17
Acid value ($mgKOHg^{-1}$)	28.0	0.40	-
Calorific value ($MJkg^{-1}$)	38.65	39.23	42

Fig 4: Properties of Diesel and Biodiesel

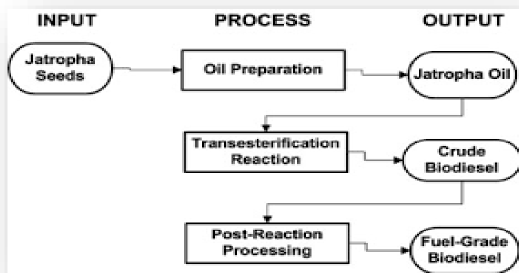

Fig 5: Jatropha oil Biodiesel processing

Fig 6: Diesel and Jatropha oil Biodiesel

Direct injection (DI) diesel engines are generally more efficient than indirect intravenous (IDI) systems. Because of the undivided combustion chamber and no loss at the throat, the direct injection system of a diesel engine (DI) consumes less fuel than a divided-chamber combustion system (in other words, an oblique injection system) (IDI). As a result, the DI system is the primary stream for automobile diesel engines. In contrast, the DI system is subject to a severe barter between NOx and PM in exhaust emission and also produces more white or blue smoke than the IDI system, which requires the reduction of the these harmful substances from the standpoint of global environmental protection.

THEORY INTERNAL COMBUSTION ENGINES

- Heat Engines: A heat engine is a device that transform a fuel's chemical energy into thermal energy and then uses that energy to produce mechanical work. There are 2 types of heat engines.

1. External combustion engines;
2. internal combustion engines.

The product of burning is directly the motive liquid in an internal combustion engine. Internal-combustion engines include petrol, gas, and diesel engines, Rotary engine engines, and open cycle gas turbines. Internal combustion engines are also used in jet engines and rockets.

First Stroke: Charge effect during the piston's outward stroke.

2nd Stroke: Charge compaction during the piston's inward stroke.

3rd Stroke: Ignition of the air-fuel mixture during an inward dead centre, followed by plunger widening on the next outward stroke.

4th Stroke: Exhaust during the piston's next inward stroke.

The Diesel Engine (1892) :-

The term "diesel engine" refers to compression-ignition oil engines, either two or four stroke, with air less fuel injection. Rudolf Diesel (1858-1913), a German electrician born in Paris, is credited with creating the knowledge of compression ignition. In 1892, he proposed compressing air alone until it reached a high enough temperature to ignite the fuel that would be injected at the end of the compression stroke. In his first experiments, he attempts to inject coal dust into a cylinder containing highly compressed air.

IC ENGINES CLASSIFICATIONS

1. Otto Cycle Engines or Spark Ignition Engines
2. Diesel Cycle Engines or Compression Ignition Engines.
3. Four Stroke Engines (One power stroke in two revolution of crankshaft)
4. Two Stroke Engines. (One power stroke in one revolution of crankshaft)

Description	S.I. Engines	C.I. Engines
1. Basic Cycle	Based on Otto Cycle	Based on Diesel Cycle
2. Fuel	Petrol, gasoline, High self ignition temp desirable.	Diesel oil, low ignition temp. desirable.
3.Introduction of fuel	Carburetor is used to mix fuel & air in proper proportion in suction stroke.	Fuel pump is used to inject fuel through injector at the end of compression stroke.
4. Ignition	Ignites with the help of spark plug.	Ignition due to high temp. caused by high compression of air & fuel.
5. Compression ration	6 to 10.5	14 to 22
6. Speed	High RPM	Lower RPM.
7. Weight	Lighter	Heavier
8. Starting	Low cranking effort	High cranking effort
9. Noise	Less	More

4.2 COMPARISION OF S.I. & C.I. ENGINES

APPLICATION OF S.I. & C.I. ENGINES.

S.I. Engines :-

Small 2 stroke petrol engines are used when the cost of the prime mover is the most important consideration. Previously, moped.

. **C.I. engines: S.I.** engines with four strokes are used in automobiles and mobile generator sets.

Two-stroke C.I. engines are used in powering vehicles where very high power diesel engines are used.

For all HEMMs, a 4 C.I. engine is used.

FIRING ORDER

Every engine cylinder must fire once in every cycle. This requires for a 4 stroke 4 cylinder engine, the ignition system must fire spark plug for every 180 deg. Of crank rotation. For a 6 cylinder engine the time available is still less.

Following are the firing order of muliti cylinder engines

Sl.no.	Engines Cylindres	Firing Order
1.	3 Cylinder Engine	1-3-2
2.	4 Cylinder Engine	1-3-4-2
3.	6 Cylinder Engine	1-5-3-6-2-4
4.	8 Cylinder V shape Engine	1-8-4-3-6-5-7-2
5.	12 Cylinder V shape Engine	1-4-9-8-5-2-11-10-3-6-7-12

FUEL INJECTION

There are two types of injection system. They are

1. Air injection: A cam shaft motivated fuel pump quantifies and pumps fuel to the fuekl valve. The fuel valve is opened by a mechanical linkage controlled by a crank shaft, which also controls the injection timing.

2. Solid Injection: This is the injection of fuel into the chamber of combustion without first atomizing it. Every solid injection system must include a pump and an atomizing unit (Injector). Individual pump and injector systems, common rail systems, and distributor systems are the three types of solid injectors.

ENGINE PERFORMANCE

Engine performance is indicated by the term efficiency.

Various type of efficiencies are

Indicated thermal efficiency :- It is a ratio of energy in the indicated horse power to the fuel energy.

Mechanical efficiency ;- It is ration of brake horse power to the indicated horse power.

Brake thermal efficiency :- It is the ration of energy in the brake horse power to the fuel energy.

Volumetric efficiency :- Volumetric efficiency is defined as the ration of air actually induced at ambient conditions to the swept volume of the engine.

Specific fuel consumption :- It is a ratio of fuel consumption per hour to the horse power.

Indicated Horse power :- is the power produced inside the cylinder.

Brake Horse Power :- is the power available at the crankshaft.

SUPER CHARGING :-

Supercharging is a method of increasing inlet air density. This is accomplished by using a stress device known as a super charger to supply air at a pressure higher than the compression at which the engine naturally aspirates the ambient air.

TURBOCHARGING:-

Supercharging is a method of increasing inlet air density. This is accomplished by using a stress device known as a super charger to supply air at a pressure higher than the compression at which the engine naturally aspirates the ambient air.

Turbochargers are centrifugal compressors that are powered by exhaust gas turbines. They are now widely used for supercharging nearly all types of 2 and 4 stroke motors. By utilising the engine's exhaust energy, it tries to recover an important part of the energy that would otherwise be wasted, and thus the turbo engine will not draw on engine power.

EXPERIMENTAL SETUP

A single cylinder, four stroke diesel engine is connected to an eddy current type dynamometer for loading. It is outfitted with the necessary instruments for measuring combustion pressure and crank angle. For strain crank angle-PV graphs, these messages are linked up to the computer via the engine indicator.

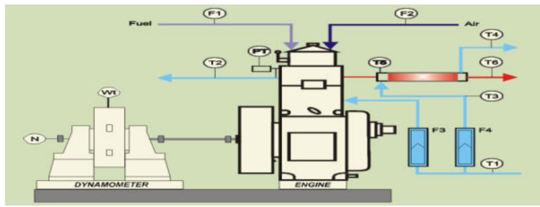

Fig 7: AVL smoke meter and Experimental setup & Fig 8: Engine Experimental setup and connection to computer

PT	Pressure Transducer	F2	Air flow	T2	Jacket water outlet temperature
N	Rotary encoder	F3	Jacket water flow	T3	Calorimeter water inlet temperature = T1
Wt	Weight	F4	Calorimeter water flow	T4	Calorimeter water outlet temperature
F1	Fuel flow	T1	Jacket water inlet temperature	T5	Exhaust gas to calorimeter tempera
				T6	Exhaust gas from calorimeter temperature

Fig 9: Engine Experimental setup Line Diagram

SPECIFICATIONS

Product	Engine test setup 1 cylinder, 4 stroke, Diesel (Computerized)
Product code	224
Engine	Make Kirloskar, Model TV1, Type 1 cylinder, 4 stroke Diesel, water cooled, power 5.2 kW at 1500 rpm, stroke 110 mm, bore 87.5 mm. 661 cc, CR 17.5
Dynamometer	Type eddy current, water cooled
Propeller shaft	With universal joints
Air box	M S fabricated with orifice meter and manometer
Fuel tank	Capacity 15 lit with glass fuel metering column
Calorimeter	Type Pipe in pipe
Piezo sensor	Range 5000 PSI, with low noise cable
Crank angle sensor	Resolution 1 Deg, Speed 5500 RPM with TDC pulse.
Data acquisition device	NI USB-6210, 16-bit, 250kS/s.
Piezo powering unit	Model AX-409.
Temperature sensor	Type RTD, PT100 and Thermocouple, Type K
Temperature transmitter	Type two wire, Input RTD PT100, Range 0–100 DegC, I/P Thermocouple, Range 0–1220DegC, O/P 4–20mA
Load indicator	Digital, Range 0-50 Kg, Supply 230VAC
Load sensor	Load cell, type strain gauge, range 0-50 Kg
Fuel flow transmitter	DP transmitter, Range 0-500 mm WC
Air flow transmitter	Pressure transmitter, Range (-) 250 mm WC
Software	"Enginesoft" Engine performance analysis software
Rotameter	Engine cooling 40-400 LPH; Calorimeter 25-250 LPH
Pump	Type Monoblock
Overall dimensions	W 2200 x D 2500 x H 1500 mm
Optional	Computerized Diesel injection pressure measurement

KIRLOSKAR ENGINE TV1 SPECIFICATIONS

Type: Single cylinder, four stroke vertical water cooled diesel engine

Rated power -5.2kw

Rated speed – 1500rpm

Bore Dia(D) – 87.5mm

Stroke (L) – 110mm

Compression ratio – 17.5: 1

C.V. of fuel for diesel – 42,000kj/kg

Density of diesel – 830kg/m^3

Eddy Current Dynamometer

Make: Techno Mech

Model: TMEC-10

KW = (Nm x RPM)/9549305

RPM: 1500-1600rpm

Dynamometer arm length –(Rm)-185mm

AVL DI GAS 444 N (five gas analyzer)

Measurement Data	Resolution
CO – 0-10% Vol	0.0001% Vol
HC – 0-22000ppm Vol	1ppm/ 10ppm
CO_2 – 0-20 % Vol	0.1 % Vol
O_2 – 0-25 % Vol	0.01 % Vol
NO_x – 0-6000ppm Vol	1ppm Vol

AVL 437C SMOKE METER

Measurement Data	Resolution
Opacity – 0-100%	0.1 %
Absorption(K Value)	0-99-99m^{-1} 0.01 m^{-1}

EXPERIMENTAL ANALYSIS: RESULTS OBTAINED

In this analysis, the engine runs at a continual injection pressure of 240 bar, an injection timing of 23°btdc, and a constant load (50%) with variable exhaust gas flow rates of (5%, 15%, and 25%).

BLENDS USED

PD100 is a pure diesel.

JOBD15 is a 100% volume mishmash of 15% Jatropha oil biodiesel and 85% diesel.

JOBD30 is a 100% volume mixture of 30% Jatropha oil biodiesel and 70% diesel.

PERFORMANCE, COMBUSTION AND EMISSION RESULTS

Brake specific fuel consumption:

BSFC	PD100	JOBD15	JOBD30
5	0.2468	0.25023	0.25898
15	0.24926	0.25189	0.2592
25	0.25125	0.2587	0.2625

Table 1: bsfc values with respect to exhaust gas flow rates of 5%, 15% and 25% for fuels PD100, JOBD15 and JOBD30

Brake thermal efficiency:

The amount of torque available at the crankshaft of the engine is referred to as brake power. It is also given by the product of the torque available at the crankshaft and the crankshaft's angular speed.

The brake thermal efficiency is a type of engine thermal efficiency that is defined as the ratio of brake power at the engine crankshaft to power generated by fuel combustion.

BTE	PD100	JOBD15	JOBD30
5	36.9178	35.8952	35.2104
15	35.9854	35.0242	34.2654
25	34.8982	34.0123	33.2127

Table 2: brake thermal efficiency values with respect to exhaust gas flow rates of 5%, 15% and 25% for fuels PD100, JOBD15 and JOBD30

Exhaust gas temperature:

The exhaust system releases engine exhaust methane into the environment. Mufflers and emission aftertreatment devices are among the specialised components of the exhaust system. A number of exhaust gas properties must be known by the designer of the engine bay and/or exhaust system components.

EGT(°c)	PD100	JOBD15	JOBD30
5	197.625	187.9542	183.5656
15	201.354	197.898	191.0502
25	203.7878	199.0206	192.5566

Table 3: Exhaust gas temperature with respect to exhaust gas flow rates of 5%, 15% and 25% for fuels PD100, JOBD15 and JOBD30

Unburnt Hydrocarbons HC ppm:

Diesel combustion is heterogeneous in nature as opposed to spark-ignited engines, where the combustible mixture is primarily homogeneous. Diesel fuel is injected into a cylinder that is filled with hot compressed air. Emissions produced by burning this heterogeneous air/fuel mixture are determined by the conditions present not only during combustion, but also during expansion and, especially, prior to the exhaust valve opening. Emmisions creation is heavily influenced by mixture preparation during the ignition delay, fuel ignition quality, residence time at different combustion temperatures, expansion duration, and general engine design features. In essence, the concentration of various emission species in exhaust is a result of their formation and reduction in the exhaust system.Incomplete combustion products produced early in the combustion chamber may be oxidised later during the expansion stroke.

The blending of hydrocarbon emissions with oxidising gases, as well as the high combustion chamber temperature and sufficient residence time for the oxidation process, allow for more complete combustion.

HC	PD100	JOBD15	JOBD30
5	59	53	49
15	62	57	55
25	67	62	59

Table 4: HC emissions with respect to exhaust gas flow rates of 5%, 15% and 25% for fuels PD100, JOBD15 and JOBD30

5.4 Oxides of nitrogen:

It's about nitrogen oxides. Purists would argue that it only relates to nitric oxide (NO) and nitrogen oxides (NO2), but most people include nitrous oxide (N2O) in this definition as well. Other variants exist, but their concentrations in the atmosphere are insufficient.

Nox	PD100	JOBD15	JOBD30
5	414	421	435
15	398	402	407
25	365	387	398

Table 5: NOx emissions with respect to exhaust gas flow rates of 5%, 15% and 25% for fuels PD100, JOBD15 and JOBD30

Carbon monoxide:

Like other internal combustion engines, the diesel uses the energy in the fuel into mechanical power. Diesel fuel is a hydrocarbon mixture that, in an ideal combustion process, produces only carbon dioxide (CO2) and water vapour (In fact, diesel exhaust gases are primarily composed of CO2, H2O, and the remainder of the engine charge air). Other sources can contribute to pollutant emissions from internal combustion engines, usually in low concentrations but occasionally containing highly toxic material. Metals and other compounds from engine wear, as well as compounds emitted by emission control catalysts, are examples of these additional emission levels (via catalyst attrition or volatilization of solid compounds at high exhaust temperatures). Catalysts can also aid in the creation of new species that are not normally present in engine exhaust.

CO	PD100	JOBD15	JOBD30
5	0.27	0.265	0.262
15	0.274	0.273	0.269
25	0.282	0.276	0.272

Table 6: CO emissions with respect to exhaust gas flow rates of 5%, 15% and 25% for fuels PD100, JOBD15 and JOBD30

Combustion duration:

Diesel engine combustion is extremely complex, and its detailed mechanisms have been unknown until the 1990s. Despite the availability of modern tools such as high speed photography used in "transparent" engines, computational power of modern computers, and the many mathematical models designed to mimic burning in diesel engines, its complexity seemed to defy researchers' tries to unlock its many secrets for decades.In the 1990s, the implementation of light image processing to the diesel burning process was critical to greatly increasing understanding of this process. Diesel combustion is distinguished by a low overall A/F ratio. Peak torque environments frequently have the lowest average A/F ratio. To avoid excessive smoke formation, the A/F ratio at peak torque is typically kept above 25:1, well above the methyl ester (chemically correct) equivalence ratio of approximately 14.4:1. At idle, the A/F ratio in turboshaft engines can exceed 160:1..

CD	PD100	JB15D85	JB20D80
5	19	19	18
15	18	18	17
25	18	17	17

Table 7 : Combustion duration with respect to exhaust gas flow rates of 5%, 15% and 25% for fuels PD100, JOBD15 and JOBD30

As a result, excess air in the cylinder after the fuel has burned continues to mix with ignition and hitherto burned gases throughout the combustion and expansion processes. Excess air and combustion products are overwhelmed when the exhaust valve is opened, explaining the oxidising nature of diesel exhaust. Although combustion occurs when vaporised fuel mixes with air, forming a locally rich but combustible mixture and reaching the proper ignition temperature, the overall A/F ratio is low. In other words, the majority of the air injected into the cylinder of a diesel engine is compressed and heated but never burns.The o2 in the excess air aids in the fermentation of gaseous hydrocarbons and carbon monoxide, minimise their concentrations in the exhaust gas to minimal levels..

Rate of Pressure rise:

In general, as engine speed increases, the pressure rise rate (dP/dq) decreases for both the diesel and the dual fuel turbocharger cases.

Crank Angle	ROPR AT 5 % EGR PD100	ROPR AT 15 % EGR PD100	ROPR AT 25 % EGR PD100	ROPR AT 5 % EGR JOBD15	ROPR AT 15 % EGR JOBD15	ROPR AT 25 % EGR JOBD15	ROPR AT 5 % EGR JOBD30	ROPR AT 15 % EGR JOBD30	ROPR AT 25 % EGR JOBD30
-10	1.1251	0.9863	0.9452	0.9670	0.9562	0.9238	0.9655	0.9496	0.9223
-9	0.9978	0.9602	0.9117	0.9652	0.9459	0.9220	0.9548	0.9346	0.9116
-8	0.9654	0.9190	0.9222	0.9499	0.9178	0.9067	0.9365	0.9088	0.8933
-7	0.8832	0.9102	0.8566	0.9118	0.8802	0.8686	0.8982	0.8602	0.8550
-6	0.8545	0.8516	0.8113	0.8567	0.8504	0.8134	0.8465	0.8059	0.8032
-5	0.8142	0.8020	0.7622	0.8050	0.7981	0.7618	0.7978	0.7444	0.7546
-4	0.7985	0.7514	0.7341	0.7833	0.7502	0.7401	0.7748	0.7385	0.7316
-3	0.8954	0.8242	0.7689	0.8251	0.8198	0.7819	0.8168	0.8185	0.7736
-2	0.9902	0.9712	0.9154	0.9833	1.0366	0.9401	0.9898	1.0406	0.9466
-1	1.4526	1.3413	1.2604	1.3209	1.4258	1.2777	1.3591	1.4498	1.3159
0	1.8978	1.8565	1.7806	1.8517	2.0000	1.8085	1.9178	2.0189	1.8746
1	2.3654	2.2345	2.1425	2.4612	2.6123	2.4180	2.5307	2.6178	2.4875
2	2.6426	2.6245	2.3202	2.9194	3.0246	2.8762	2.9725	3.0169	2.9293
3	2.9996	2.8042	2.7562	3.0268	3.0578	2.9836	3.3542	3.0477	3.3110
4	2.7565	2.7325	2.5602	2.7625	2.7342	2.7193	2.8063	2.7231	2.7631
5	2.5412	2.2162	2.2178	2.2743	2.2150	2.2311	2.3272	2.2039	2.2840
6	2.1896	1.6673	1.7898	1.7297	1.6661	1.7900	1.7880	1.6550	1.7900
7	1.3452	1.2141	1.1499	1.2122	1.1622	1.1690	1.2778	1.1511	1.2346
8	0.7985	0.7854	0.6954	0.7500	0.7184	0.7068	0.8229	0.7073	0.7797
9	0.4502	0.3489	0.2990	0.3578	0.3356	0.3145	0.4236	0.3245	0.3804
10	0.0223	0.0222	-0.0209	0.0313	0.0053	-0.0119	0.0700	-0.0058	0.0268

Table 8: Rate of pressure rise Vs Crank angle with respect to exhaust gas flow rates of 5%, 15% and 25% for fuels PD100, JOBD15 and JOBD30

The rate of pressure rise with relation to crank angle for fuel oil at rated load. Among the test fuels, diesel has the fastest rate of pressure rise. It is also observed that as JOBD in the fuel increases, the maximum pressure rate decreases.

Heat release rate:

The data logger provided thrust force and crank angle signals for the specified engine load. For 100 cycles, the information was saved in a computer-based digital data acquisition system. Because no spatial variations were taken into account, the model is said to be zero-dimensional. The net heat release rate (NHRR) was calculated using the data obtained for the combustion cycle and the first law of thermodynamics by taking into account the average value of pressure and crank angle data.

Crank Angle	HRR AT 5% EGR PD100	HRR AT 15 % EGR PD100	HRR AT 25% EGR PD100	HRR AT 5% EGR JOBD15	HRR AT 15 % EGR JOBD15	HRR AT 25% EGR JOBD15	HRR AT 5% EGR JOBD30	HRR AT 15 % EGR JOBD30	HRR AT 25 % EGR JOBD30
-10	-2.614	-2.240	-2.614	-2.883	-2.316	-2.883	-2.184	-2.684	-2.184
-9	-0.253	-2.080	-0.253	-2.633	-2.036	-2.633	-1.904	-2.554	-1.904
-8	-2.270	-2.020	-2.270	-2.373	-1.826	-2.373	-1.694	-2.434	-1.694
-7	-1.780	-1.850	-1.780	-1.833	-1.436	-1.833	-1.304	-2.064	-1.304
-6	-0.790	-0.650	-0.790	-0.823	-0.696	-0.823	-0.564	-1.144	-0.564
-5	0.950	0.780	0.950	0.777	0.604	0.777	0.736	0.626	0.736
-4	3.560	3.320	3.560	3.517	3.154	3.517	3.286	3.846	3.286
-3	8.680	7.640	8.680	8.507	8.124	8.507	8.256	9.356	8.256
-2	16.540	14.980	16.540	16.127	15.884	16.127	16.016	17.246	16.016
-1	23.980	23.020	23.980	24.867	24.884	24.867	25.016	26.006	25.016
0	32.120	30.250	32.120	31.647	32.074	31.647	32.206	32.726	32.206
1	34.780	33.160	34.780	34.117	34.994	34.117	35.126	34.996	35.126
2	32.060	32.010	32.060	32.527	33.764	32.527	33.896	32.976	33.896
3	29.460	27.850	29.460	29.437	30.664	29.437	30.796	29.456	30.796
4	27.020	26.860	27.020	26.967	27.934	26.967	28.066	26.776	28.066
5	25.140	24.930	25.140	24.927	25.834	24.927	25.966	24.796	25.966
6	22.980	23.580	22.980	23.287	24.284	23.287	24.416	23.206	24.416
7	22.140	22.660	22.140	22.337	23.314	22.337	23.446	22.456	23.446
8	21.540	21.810	21.540	21.547	22.484	21.547	22.616	21.766	22.616
9	21.020	21.270	21.020	21.117	21.774	21.117	21.906	21.346	21.906
10	20.980	20.670	20.980	21.027	21.211	21.027	21.356	20.976	21.356

Table 9 : Heat release rate Vs Crank angle with respect to exhaust gas flow rates of 5%, 15% and 25% for fuels PD100, JOBD15 and JOBD30

The in-cylinder pressure reflects the combustion process, which includes piston work on gas, heat transfer to combustion chamber walls, and mass flow in and out of divot regions in between piston, piston rings, and cylinder liner. The combustion process spreads throughout the combustion chamber, and each of these processes must be linked to the cylinder pressure.

RESULTS (GRAPHS) AND CONCLUSIONS

ENGINE OPERATING AT 240 BAR PRESSURE AND 23°bTDC AND AT A CONSTANT LOAD (50 %) WITH VARIABLE EGR VALVE OPENINGS(5%, 15% AND 25%):

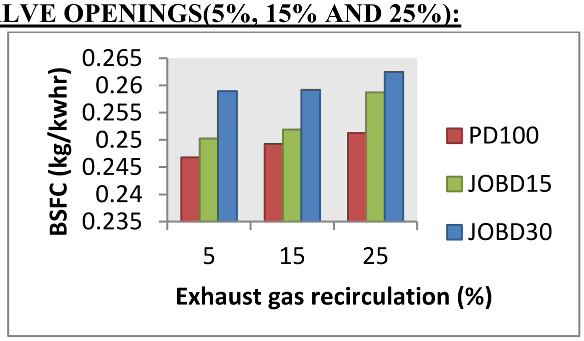

Fig 7: Brake specific fuel consumption with respect to variable Exhaust gas flow rates

Brake thermal efficiency- The comparison of brake thermal efficiency of various fuels PD100, JOBD15, and JOBD30 at 240bar injection pressure and 23°bTDC of injection timing at rated engine speed 1500 rpm at various loads is shown in the figure. The thermal efficiency of the brakes increases as the load for all fuels. The thermal efficiency of Jatropha-diesel blends decreases as the percentage of Jatropha oil in the blend increases. The decrease in thermal efficiency as the ration of Jatropha-diesel blends intensifies is due to the earlier start of combustion than for diesel, that increases compression work. Because the engine operates with constant injection advance, the shorter ignition delay of JOBD causes combustion to begin much before TDC. This increases compression work as well as heat loss, lowering the engine's efficiency.

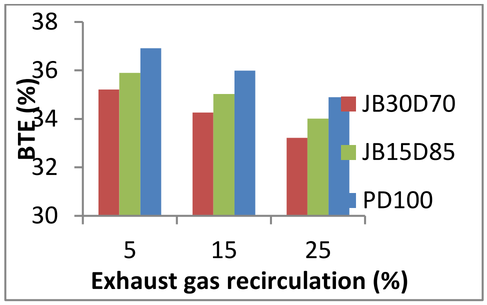

Fig 8: Brake thermal efficiency with respect to variable Exhaust gas flow rates

Ignition delay- In a diesel engine, ignition delay is the time between the start of injection and the start of combustion. Figure depicts the variation of fuel injection with pile for gas and jatropha biofuels blends. In a diesel engine, ignition delay is the time between the Figure depicts the variation of ignition delay with load for diesel, JOBD, and its mixes. It has been discovered that the ignition delay of JOBD and its blends is less than that of diesel at all loads, and that the ignition delay decreases with an increase in the amount of JOBD in the blend at all loads.

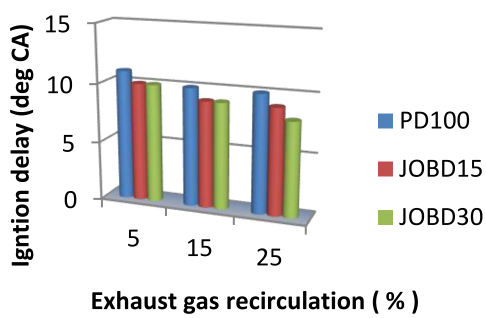

Fig 9: Ignition delay with respect to

VARIABLE EXHAUST GAS FLOW RATES

Combustion duration - Figure depicts the differences of combustion with load for all test fuels. Because of the increased fuel mass injected, the duration of combustion increases with an increase in load for all fuels.

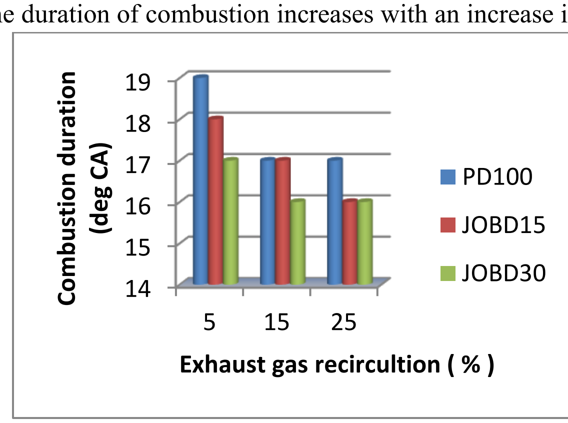

Fig 10: Combustion duration with respect to variable Exhaust gas flow rates

Unburned hydrocarbon (UBHC) emissions- The graph depicts the variation of UBHC emissions with load for test fuels. As the pressure increases, there is an obvious increase in HC emissions for all energy sources. This trend could be explained by the presence of fuel-rich mixtures at higher loads It has been discovered that increasing the proportion of JOBD in the blend reduces UBHC emissions. This demonstrates that the presence of oxygen in JOBD, as well as the higher combustion temperature, promote hydrocarbon oxidation.

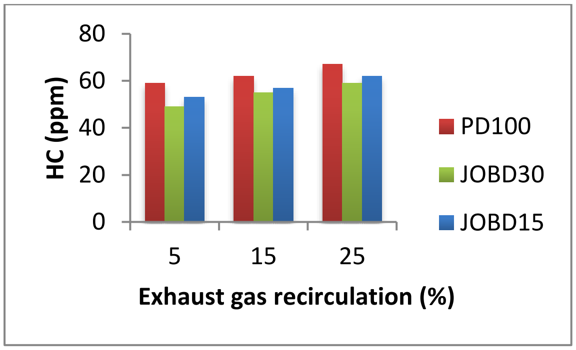

Fig 11: Hydrocarbons with respect to variable Exhaust gas flow rates

Figure illustrates the thermal efficiency of hydrocarbon (HC) with brake power for varied EGR percentages. Diesel has higher HC emissions than biodiesel without and with EGR up to 50% of engine load. At 50% engine load, the HC emission for 5%, 15%, and 25% biodiesel increases in comparison to diesel. At rated load, EGR HC emissions are cut by 5% and 15%, respectively.

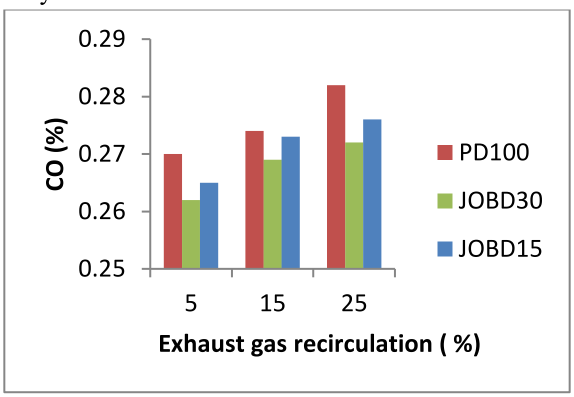

Fig 12: CO emissions with respect to variable Exhaust gas flow rates

The figure depicts the change in percentage Co and hc with respect to brake power. All curves with EGR have nearly the same amount of CO emission and a typical value of 0.03% by volume up to 50% of engine load. However, CO emissions will increase at full load. The CO emission rises as the percentage of EGR with bioethanol increases. At rated load, 25% EGR is found to be 0.77% by volume, 1.57% by volume, and 2.54% by amount for 5% and 15%, respectively.

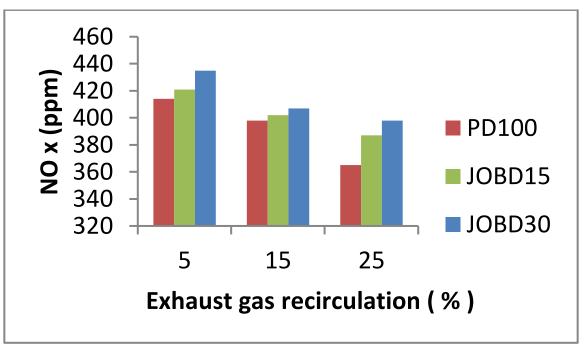

Fig 13: NOx emissions with respect to variable Exhaust gas flow rates

It is also observed that as the sum of JOBD in the blend increases, so do the NOx emissions. This increase could be attributed to the fact that JOBD is an oxygenated fuel, which results in better combustion and thus higher combustion. temperature is attained. This higher temperature promotes NOx formation.

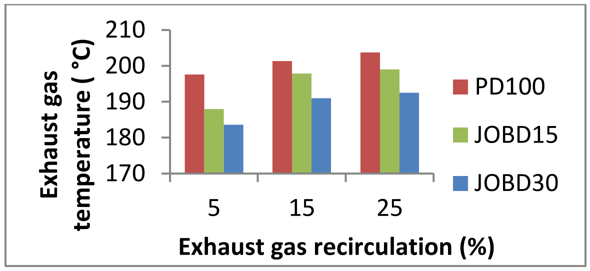

Fig 14: Exhaust gas temperature with respect to variable Exhaust gas flow rates

For all loads, the exhaust gas temperature improves as the percentage of JOBD in the test fuel increases. This could be due to the JOBD's high oxygen content, which improves combustion and thus raises exhaust gas temperatures.

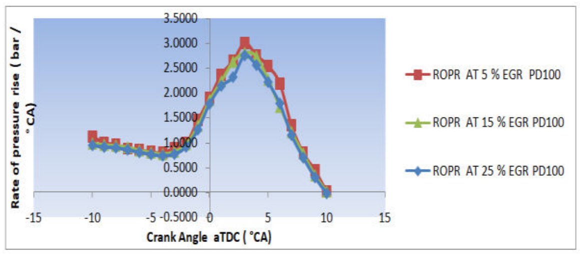

Fig 15: Rate of pressure rise Vs Crank angle at variable Exhaust gas flow rates for PD100 fuel

Among the test fuels, diesel has the fastest rate of pressure rise. It is also observed that as JOBD in the gas increases, the maximum pressure rate decreases. This is because the combustion duration varies with the percentage of bio - fuels in the fuel.

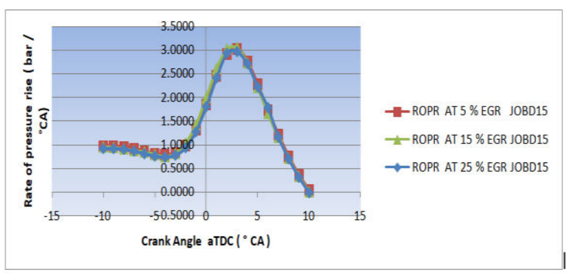

Fig 16: Rate of pressure rise Vs Crank angle at variable Exhaust gas flow rates for JOBD15 fuel

When tried to compare to JOBD and its mixes, the which was before phase of diesel is very intense, resulting in a high pressure rise as more fuel is accumulated during the delay period.

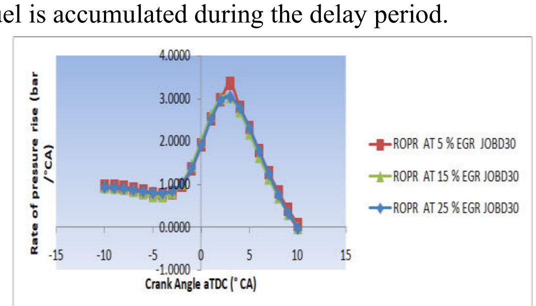

Fig 17: Rate of pressure rise Vs Crank angle at variable Exhaust gas flow rates for JOBD30 fuel

As a result, the quantity of fuel gathered during the ignition delay is inversely proportional to the amount of JOBD in the fuel.

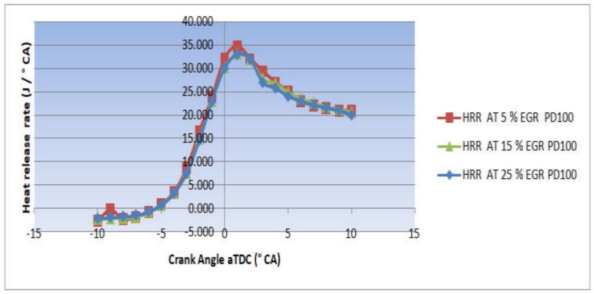

Fig 18: Heat release rate Vs Crank angle at variable Exhaust gas flow rates for PD100 fuel

It is observed that the value of maximum heat release rate decreases with the increase of JOBD in the fuel.

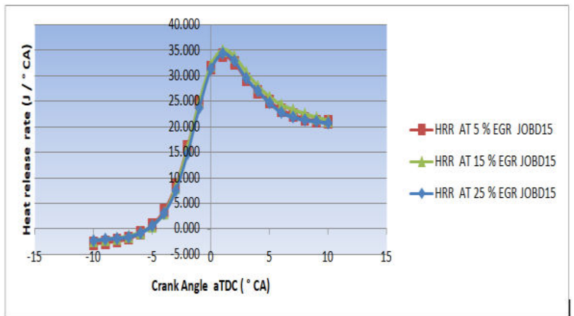

Fig 19: Heat release rate Vs Crank angle at variable Exhaust gas flow rates for JOBD15 fuel

Heat release rate is a variable that varies with the state of the combustion process and the rise in stress inside the cylinder, and high temperature release rates are observed here as a result of hot gases through the exhaust gas recirculation.

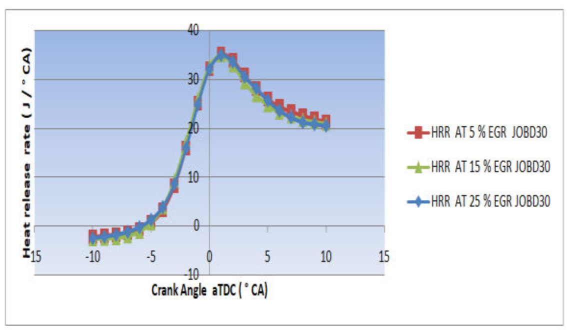

Fig 20: Heat release rate Vs Crank angle at variable Exhaust gas flow rates for JOBD30 fuel

According to the same figure, the maximum heat release rate for 5% and 15% biodiesel fuels is around 3-4° CA bTDC and 5° CAbTDC for those other fuel sources.

SAMPLE CALCULATIONS

Maximum load calculation

Maximum load (W) = $\frac{BP\times60\times1000}{2\pi NRm}$ = ------ N = ----kgf

BP = Rated power in kw

N = Rated speed = 1500 rpm

Rm = Radius of the dynamometer arm length in m

1. Brake power (BP) = $\frac{2\pi N(W\times9.81)Rm}{60\times1000}$ = ----kw

N = speed of the engine in rpm

W = Applied load in 'N'

Rm = Radius of the dynamometer arm length in m

2. Indicated Power (IP) = $\frac{IMEP \times LAN/2 \times 10^5}{60 \times 1000}$ = ----kw
IMEP = Indicated mean effective pressure in bar
L = Stroke length in m
D = Cylinder diameter in m
A = Cylinder area in m²
Where, A = $\frac{\pi \times D^2}{4}$ = ---m²
3. Total fuel consumption (TFC) = (q x Density of the fuel)/t = -----kg
Where, q = volume of fuel consumed = 10 x 10^{-6} m³
t = time taken for 10g of fuel consumption in sec
4. Specific fuel consumption (SFC) = TFC/BP = kg/kwh
Where, TFC = Total fuel consumption
BP = Brake power in kw
5. Mechanical efficiency (η_m) = (BP / IP) x 100 = ----%
6. Brake thermal efficiency (η_{bt}) = BP / (TFC x Cv) x 100 = ---%

CONCLUSIONS

The performance, emission, and spark ignition of a single cylinder four-stroke diesel engine powered by diesel PD100 and blends of jatropha oil biodiesel JOBD15 and JOBD30 at an injection pressure of 240bar and constant injection timing of 23°bTDC at engine rated speed of 1500rpm are examined for varying exhaust gas flows of 5%, 15%, and 25%. The main conclusions are as follows:

1. It was discovered that the ignition timing for Jatropha and its blends JOBD15 and JOBD30 was lower than that of diesel oil PD100 due to the natural oxygen content.
2. Diesel PD100 has a faster rate of pressure increase than blends of biodiesel made from jatropha oil (JOBD15 and JOBD30).
3. As the proportion of jatropha oil biodiesel in the fuel rises, the rate of pressure rise falls.
4. The braking thermal efficiency of jatropha biodiesel and its blends is somewhat lower than that of diesel PD100 due to the early combustion start of seed oil biodiesel and its blends JOBD15 and JOBD30, which increases compression work.
5. Jatropha oil biodiesels JOBD15 and JOBD30 typically emit more NOx than diesel, but produce lower amounts of HC, CO, and soot density emissions than diesel D100. It is feasible to draw the conclusion that diesel may be replaced by jatropha methyl ester and its derivatives. Blends of biodiesel made from jatropha oil have good combustion and performance characteristics and produce less pollutants than petroleum-based diesel, with the exception of NOx.
6. Increasing ORC Plant Efficiency Through Combustion Process PRAKASH, V. V., & KUMAR, B. S.
7. According to the results, the JOBD30 blend uses about the same amount of fuel as the traditional diesel PD100 while having improved mechanical and thermal brake efficiency by load fluctuation.
8. After 50% load, a sharp rise in in-cylinder gas pressure is seen.
9. Compared to diesel, biodiesel has a greater specific fuel consumption due to its higher CV.
10. Because full combustion boosts the temperature of the combustion chamber, the exhaust gas temperature of biodiesel is greater than that of diesel.
10. Because of incomplete combustion caused by dilution in the combustion chamber, the amount of smoke emitted by biodiesel rises as the proportion of EGR increases. The quantity of smoke generated rises in tandem with the EGR percentage.
11. Jatropha biodiesel emits less CO and HC than diesel. This could be as a result of the higher oxygen content of biodiesel. CO and HC emissions rise as the proportion of EGR increases. An increase in engine load causes a continual rise in NOx emissions, which may be caused by a rise in cylinder temperature.

12. Because of incomplete combustion and a lower cylinder temperature, it is seen that the generation of NOx decreases as the percentage EGR increases.
13. Compared to clean diesel, NOx emissions with 5%, 15%, and 25% reductions are 25%, 48%, and 61% lower, respectively.
14. The Brake Thermal Efficiency decreases, while NOx generation is much decreased and is determined to be lowest at 15% EGR.
15. It is necessary to take into account the most effective utilisation of exhaust heat energy for NOx reduction.

SCOPE FOR FUTURE WORK

1. By switching the biodiesels and adjusting the injection pressures and infusion timings, diesel engines' performance, combustion, and emission characteristics may be evaluated in order to achieve better performance.
2. In the future, ternary fuels, such as adding low reactivity fuels (alcohols) to diesel and biodiesel, may be used to assess an engine's performance, combustion, and emission characteristics.

BIBLIOGRAPHY

1 PRAKASH, VODNALA VEDA, and BOMMANA SHRAVN KUMAR. "IMPROVING NANO MATERIAL COTING OF GAS TURBINE BLADES MODEL ANALYSIS.".

2 Mathew BC, Thangaraja J, Sivaramakrishna A. Combustion, performance and emission characteristics of blends of methyl esters and modified methyl esters of karanja and waste cooking oil on a turbocharged CRDI engine. Clean TechnolEnviron Policy 2019;21:1791-807. https://doi.org/10.1007/s10098- 019-01750-9.

3 3 C. Patel, K. Chandra, J. Hwang, R. Agarwal, N. Gupta, C. Bae, T. Gupta, and A. Agarwal. Compression ignition engine performance, combustion and emission characteristics, and trace metals in waste cooking oil particulates. Fifty or so years derived from Jatropha and Karanja oils Fuel 2019;236:1366-76. doi:10.1016/j.fuel.2018.08.137.

4 (4) Pramanik, K. (2203). The properties of Jatrophacurcas oil and diesel fuel blends, as well as one's use in compression ignition engines. 28:239-248. Renewable Energy.

5 F.K. Forson, E.D. Oduro, and E. Hammond-Donkoh (2204). Jatropha oil blends' performance in a diesel engine. 29:1135-1145 (Renewable Energy).Palaniswamy, E., Manjula, P., Manoharan, N. (2206). Effects of Pre-Heating of JatrophaCurcas Oil on Performance and Emission Characteristics of C.I. Engines. Proceedings of the International Conference on Resource Utilisation and Intelligent Systems, New Delhi:Allied Publishers Private Limited, 315–319.

6 Reddy, J.N., Ramesh, A. (2206). Parametric studies for improving the performance of a Jatropha oil- fuelled compression ignition engine. Renewable Energy 31:1994–2016.

7 PRAKASH, V. V., & KUMAR, B. S. IMPROVING ORC PLANT EFFICIENCY BY COMBUSTION PROCESS.

8 Prakash, V. V. Use OfNdtApproches For Radiator Fabrication And Reliability For Nano Fluids-A Study.

9 Prakash, Vodnala Veda, and S. Chakradhara Goud. "A Schematic Design and an NDT Approach for a Radiator Tubes Using Nano fluids.".

10 M.A. Rahman, A.M. Ruhul, M.A. Aziz and R. Ahmed. 2017. Experimental exploration of hydrogen enrichment in a dual fuel CI engine with exhaust gas recirculation, Int. J. Hydrogen Energy, 42(8), 5400-5409. https://doi.org/10.1016/j.ijhydene.2016.11.109.