NUMERICAL INVESTIGATION OF PERFORMANCE, COMBUSTION AND EMISSION OF VARIOUS BIOFUELS

Report submitted to National Institute of Technology Manipur for the Award of the degree of

Bachelor of Technology In Mechanical Engineering

by

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NATIONAL INSTITUTE OF TECHNOLOGY MANIPUR

CERTIFICATE

This is to certify that the Dissertation **Report** entitled**, "Numerical investigation of performance, combustion and emission of various biofuels"** submitted by **B.V.S SURYA PRASAD (15UME006) , Mr. BUDDHA DEV KUMAR (15UME008) & Mr. RAJ GAURAV (15UME012)** to National Institute of Technology Manipur, India, is a record of bonafide Project work carried out by them under my supervision and guidance and is worthy of consideration for the award of the degree of Bachelor of Technology in Mechanical Engineering of the Institute.

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We certify that

a. The work contained in this report is original and has been done by me under the guidance of my supervisor(s).

b. The work has not been submitted to any other Institute for any degree or diploma.

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LIST OF SYMBOLS & ABBREVIATIONS

Nomenclature

A_0, A_1, A_2	empirical factors
A/F	air/fuel
BMEP	brake mean effective pressure (bar)
BN	Bosch number
BSN	Bosch smoke number
BT	brake power (kW)
BTE	brake thermal efficiency (%)
b_m	depth of the spray forward front (m)
CA	crank angle (degrees)
CAE	crank angle encoder
CI	compression ignition
CN	cetane number of fuel
CPP	cylinder peak pressure (bar)
CR	compression ratio
CPT	cylinder peak temperature (K)
CSOBD	cotton seed oil biodiesel
DI	direct injection
ED	error deviation (%)
EGR	exhaust gas recirculation
EGT	exhaust gas temperature (K)
EEFO	ethyl ester fish oil
E_a	apparent activation energy for the auto ignition process (kJ/kmole)
FMEP	friction mean effective pressure (bar)
HSL	Hartridge Smoke Level
h_{wfr}	height of the NWF forward front
JME	jatropha methyl ester
K_T	evaporation constant
l	current distance between the injector's nozzle and the location of the EFM (m)
LCS	load cell sensor
LFR	liquid flow rate
LME	linseed methyl ester
L_c	cycle work done (kJ)
l_m	EFM's penetration distance (m)
MAOME	microalgae oil methyl esters
m	total mass (kg)
m_f	fuel mass per cycle (kg/h)
NOP	nozzle opening pressure (bar)
NWF	near wall flow
n	engine speed (rpm)
P	pressure (bar)
PD	pure diesel
PM	particular matter (g/kWh)
POU	percentage of uncertainty (%)
PSBD	palm stearin biodiesel
PTS	pressure transducer sensor
P_{max}	maximum cylinder pressure (bar)
P_b	brake power (kW)
q_c	cycle fuel mass (kg)
R	gas constant (J/(mol·k))
SE	standard error
SD	standard deviation
RME	rapeseed methyl ester
SFC	specific fuel consumption (g/kWh)
SME	soybean methyl ester
SOI	start of injection (degree)
SFC	specific fuel consumption (kg/kWh)
SFR	soot formation rate (1/deg)
SS	speed sensor
STP	spray tip penetration (mm)
V_p	mean piston velocity (m/s)
S_g	net generation rate of the ith species (kg/s)
T	temperature (K)
TDC	top dead centre
TIS	temperature indicator sensor
T_b	temperature in a burnt gas zone (K)
TME	tallow methyl ester
V	volume of cylinder (cm^3)
V_1	current velocity of the EFM (m/s)
V_0	initial velocity of the EFM at the nozzle of the injector (m/s)
V_m	fuel spray evaluation process in a medium speed diesel engine (m/s)
V_k	swept volume (cm^3)
V_i & V_c	cylinder volumes at injection timing and top dead centre (cm^3)
x	fraction of fuel burnt
X_0	fraction of burnt fuel during ignition delay
Y_i	mass fraction
$[N_2]_e$	equilibrium concentrations of an molecular nitrogen
$[NO]_e$	equilibrium concentrations of an oxide of nitrogen
$[O]_e$	equilibrium concentrations of molecular oxygen
$[O_2]_e$	equilibrium concentrations of atomic oxygen
r_{H_2O}	volume fraction of water vapor in a combustion chamber
Y_i^{cyl}	stoichiometric coefficients on the reactant side
Y_i^j	stoichiometric coefficients on the product side
α, β, λ	constants
α_1	air-fuel equivalence ratio
τ	time (second)
τ_k	travel time for the EFM to reach a distance l from the injector's nozzle
ρ	density (kg/m^3)
ν	specific volume (m^3/kg)
ϕ	crank angle (degree)
ω	angular crank velocity (rpm)
ε	compression ratio
ξ_b	cylinder air charge usage efficiency
σ_{ud}, σ_u	fuel fractions evaporated during ignition delay period and up
Ω_i	molar rate of production (mol/s)
$\frac{dt}{dx}$	heat release rate (J/deg)
VCR	variable compression ratio
VE	volumetric efficiency (%)

ABSTRACT

Alternative fuels have been a major concern in today's era due to its promising characteristics in internal combustion engines. The computational analysis is a proven method by many researchers in determining combustion, performance and emission characteristics of an internal combustion engine, offering accurate results at par with experimental results. An attempt is being made by the author(s) in this paper for performing numerical study of nine (9) different alternative biofuels and pure diesel. Comparison has been carried out with the results of pure diesel in a single cylinder, naturally aspirated, water cooled, direct injection diesel engine with different engine load (25%, 50%, 75% and 100%), constant compression ratio (CR17.5), higher nozzle opening pressure (220bar) and advance injection timing (23° CA before TDC). The study was performed after validating results of two experimental data using the proposed tool, which have shown that the numerical results were in good agreement with the experimental results. The numerical results of the different biofuels have depicted the possibility of using the fuels as alternative fuel for internal combustion engine.

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Chapter 01
INTRODUCTION-BIOFUELS

Biofuels are combustible fuels created from biomass; in other words, fuels created from recently living plant matter as opposed to ancient plant matter in hydrocarbons. The term biofuel is usually used to reference liquid fuels, such as ethanol and biodiesel that are used as replacements for transportation fuels like petroleum, diesel and jet fuel. Biofuels can also include solid fuels like wood pellets and biogas or syngas – however in this summary we will focus on liquid fuels.

There are two main types of biofuels – ethanol and biodiesel. The simplest way to distinguish between the two is to remember ethanol is an alcohol and biodiesel is an oil. Ethanol is an alcohol formed by fermentation and can be used as a replacement for, or additive to, gasoline whereas biodiesel is produced by extracting naturally occurring oils from plants and seeds in a process called trans-esterification. Biodiesel can be combusted in diesel engines.

Biofuels are grouped by categories - first generation, second generation, and third generation – based on the type of feedstock (the input material) used to produce them.

- First generation biofuels are produced from food crops. For ethanol, feedstocks include sugar cane, corn, maize, etc. For biodiesel, feedstocks are naturally occurring vegetable oils such as soybean and canola.
- Second generation biofuels are produced from cellulosic material such as wood, grasses, and inedible parts of plants. This material is more difficult to break down through fermentation and therefore requires pre-treatment before it can be processed.
- Third generation biofuels are produced using the lipid production from algae.

In addition, the term "Advanced Biofuels" is used to describe the relatively new technological field of biofuel production that uses waste such as garbage, animal fats, and spent cooking oil to produce liquid fuels.

Biofuels are not as energy dense as conventional transportation fuels. 1 gallon of biodiesel has 93% of the energy of 1 gallon of diesel and 1 gallon of ethanol (E85) has 73% of the energy of 1 gallon of gasoline.

Biofuels are currently the only viable replacement to hydrocarbon transportation fuels. Because biofuels can be used in existing combustion engines, minimal changes to infrastructure are required for their implementation. This is their most prominent advantage as concerns about the environmental impacts of fossil fuels continue to rise.

In regions that do not have hydrocarbon resources but do have suitable agricultural conditions, biofuels provide an alternative to foreign fuel imports. They also come from a wide variety of sources and therefore can be produced in many regions.

While there is some dispute over just how "renewable" biofuels are, it is generally accepted that the crops used to produce them can be replenished much faster than fossil fuels.

Concerns about biofuels are usually centered around the fact that they are an agricultural product. One key concern about biofuels is that crops grown for fuel production compete with other natural resources, particularly food and water. First generation biofuels use only edible crops which has led to biofuel crops displacing food sources in some regions. In many regions of the world, subsidies are provided for these crops which only amplifies these issues. In addition, increased agriculture of any form often comes with concerns of deforestation, water and fertilizer use, which all have their own respective environmental and climate impacts.

The increase in consumption is the reason for the increase in pollution and hence significant progress in the field of internal combustion engine research were made. Many researchers have used various biofuels as alternative and renewable source to determine the performance, emission and combustion characteristics of internal combustion engines. Pyrolysis is performed on waste plastics to obtain oil and the same is mixed with 5% and 10% with diethyl ether on single cylinder water cooled direct injection (DI) engine. The blended fuel (waste plastic and DEE) have improved the cetane rating of the fuel. It was seen that there were significant reduction in the smoke levels while using blended fuel as compared to pure waste plastic oil. The brake thermal efficiency (BTE) has increased and pollutants such as CO and NOx were decreased. Addition of oxygenates have improved theprocess of combustion while reducing emissions. Another study have investigated an engine fed with eight (8) different renewable diesel fuels which includes pure diesel, jet fuel, traditionally derived biodiesel (FAME), deoxygenated canola derived fatty acids (DCFA), DFCA with varying H2, continuous DCFA, deoxygenated lauric acid (DLA) and isomerize deoxygenated canola derived fatty acid alkanes. While diesel, jetfuel and FAME were used as benchmark fuels for the other new types of renewable fuels. The results have indicated lower mechanical efficiency but increased BTE of the renewable fuels. The combustion analysis indicated short ignition delays, low peak in cylinder pressures, reduced rate in increase in cylinder pressure and low heat release rate for the proposed renewable fuels. They have an added advantage of reduction in NOx, soot and greenhouse gas emission [1,2]. Waste tire pyrolysis oil (WTPO) blended with diesel fuel is proposed to be used as fuel for investigating the performance and emission characteristics of a four stroke, four cylinder, naturally aspirated, DI diesel engine. The results shows that WTPO blended with diesel can be used as a potential fuel since it has similar characteristics like torque and power output as compared to diesel, without much modification in the engine [3,52]. Various additives were also added to improve the biodiesel blends to reduce fuel consumption and NO emission from diesel engines, such as 2-ethylhexyl nitrate (EHN), di-tertiary-butyl peroxide (DTBP) and pentanol. The additives have improved the combustion pressure (0.11–0.53bar) and lower heat release rate (0.82–2.29J/°CA) [4]. Some studies on diesel engine using higher alcohol/diesel fuel blend shows that without any modification, a diesel engine can run properly on 30% 1-butanol/70% diesel fuel or 25% 1-pentanol/75% diesel fuel [5]. Use of diesel-aegle marmelos oil-diethyl ether blends on various blends has been performed. 60:30:10 (diesel: aegle marmelos: diethyl ether) have increased the BTE by 4.3%, NOx emission have been reduced by 3.9% at compression ratio (CR) 17.5 and full load conditions.

Heat engine is a machine which converts heat developed by burning fuel into useful work. It can also be seen as an equipment which generates thermal energy and transforms it into mechanical energy.

1. Classification of heat engines

1.1 Based on combustion of fuel:

a) External combustion engine
b) Internal combustion engine.

a) External combustion engine

In this engine steam is the working medium which is produced in a boiler, located outside the engine assembly and is fed into the cylinder to operate the piston to produce useful mechanical work.

b) Internal combustion engine

In this engine, the heat is generated by combustion of fuel inside the cylinder. Due to the addition of this heat to the air inside the cylinder the pressure of the air increases tremendously. The pressurized moves the piston which rotates the crank shaft and thus mechanical work is done.

1.2 Based on fuel used

1.2.1 Diesel engine – Diesel is used as fuel
1.2.2 Petrol engine – Gasoline is used as fuel
1.2.3 Gas engine – Propane, Butane or Methane gases are used

1.3 Based on ignition of fuel

a) Spark ignition engine – Air fuel mixture is drawn in to the engine cylinder. Spark plug is used to ignite the fuel. The spark plug produces a spark which ignites the mixture. Such a combustion is called constant volume combustion (CVC).

b) Compression ignition engine –In this engine, air is compressed in to the cylinder. Due to this compression, the temperature of the compressed air rises to about 700-900 C. At this stage diesel is sprayed in to the cylinder in the form of mist. Due to the high temperature, the fuel gets ignited. This type of combustion is called constant pressure combustion (CPC) because during the combustion, the pressure inside the cylinder is almost constant.

1.4 Based on working cycle

a) Four stroke cycle engine – When the cycle is completed in two revolutions of the crankshaft, it is called a four stroke cycle engine.

b) Two stroke cycle engine – When the cycle is completed in one revolution of the crankshaft, it is called a two stroke cycle engine.

2. Internal compression engine:

2.1 Construction of an I.C. engine:

Here the reciprocating motion of piston is converted into rotary motion of the crankshaft by means of a connecting rod. The piston in the cylinder is very close fit in the cylinder. A set of rings are inserted in the circumferential grooves of the piston to prevent leakage of gases from sides of the piston wall. Generally a cylinder is bored in the cylinder block and a gasket, made up of copper sheet or asbestos is inserted between the cylinder and the cylinder head to avoid leakage. The space for combustion is provided at the top of the cylinder head where the combustion takes place. The connecting rod connects the piston and the crankshaft. The end of the connecting rod which connects the piston is called small end. A pin called gudgeon pin or wrist pin is used for connecting the piston and the connecting rod at the small end. The other end of the connecting rod which connects the crank shaft is called big end. When piston moves up and down, the motion is transmitted to the crank shaft by means of connecting rod and the crank shaft makes rotary motion. The crankshaft rotates in main bearings which are fitted the crankcase. A flywheel is provided at one end of the crankshaft for smoothing the uneven torque produced by the engine. There is an oil sump at the bottom of the engine which contains lubricating oil for lubricating different parts of the engine.

2.2 Working principle of I.C. Engine/ four stroke cycle engine / two stroke cycle engine

A mixture of fuel with correct amount of air is exploded in an engine cylinder which is closed at one end. As a result of this explosion, heat is released resulting in increase of the pressure of the burning gases. This pressure forces a close fitting piston to move down the cylinder. The movement of piston is transmitted to a crankshaft by a connecting rod so that the crankshaft rotates and turns a flywheel connected to it. Power is driven from the rotating crank shaft to do mechanical work. The explosion has to be repeated continuously to obtain continuous rotation of the crankshaft. Before the explosion to take place, the used gases are expelled from the cylinder, fresh charge of fuel and air are admitted in to the cylinder and the piston moved back to its starting position. These sequences of events taking place in an engine are called the working cycle of the engine. The sequences of events taking place inside the engine are as follows

1. Admission of air or air-fuel mixture inside the engine cylinder (suction)
2. Compression of the air or air fuel mixture inside the engine (compression)
3. Injection of fuel in compressed air for ignition of the fuel or ignition of air-fuel mixture by an electric spark using a spark plug to produce thermal power inside the cylinder (power)

Removal of all the burnt gases from the cylinder to receive fresh charge (exhaust) Note: charge means admitting fresh air in to the cylinder in the case of compression ignition engines (diesel engines) or admitting a mixture of air and fuel in to the cylinder in the case of spark ignition engines which is as described in fig 1.1.

2.3. Two stroke cycle engine (petrol engine)

In two stroke cycle engines, the whole sequence of events i.e., suction, compression, power and exhaust are completed in two strokes of the piston i.e. One revolution of the crankshaft. There is no valve in this type of engine. Gas movement takes place through holes which are called ports in the cylinder. The crankcase of the engine is made air tight in which the crankshaft rotates.

2.3.1 Upward stroke of the piston (suction + compression)

When the piston moves upward it covers two of the ports, the exhaust port and transfer port, which are normally almost opposite to each other. This traps the charge of air- fuel mixture drawn already in to the cylinder. Further upward movement of the piston compresses the charge and also uncovers the suction port. Now fresh mixture is drawn through this port into the crankcase. Just before the end of this stroke, the mixture in the cylinder is ignited by a spark plug. Thus, during this stroke both suction and compression events are completed.

2.3.2. Downward stroke (power + exhaust)

Burning of the fuel rises the temperature and pressure of the gases which forces the piston to move down. When the piston moves down, it closes the suction port, trapping the fresh charge drawn into the crankcase during the previous upward stroke. Further downward movement of the piston uncovers first the exhaust port and then the transfer port. Now fresh charge in the crankcase moves in to the cylinder through the transfer port driving out the burnt gases through the exhaust port. Special shaped piston crown deflect the incoming mixture up around the cylinder so that it can help in driving out the exhaust gases. During the downward stroke of the piston power and exhaust events are completed.

2.3 Four stroke cycle engine (diesel/ petrol engine)

In four stroke cycle engines, the four events namely suction, compression, power and exhaust take place inside the engine cylinder. The four events are completed in four strokes of the piston (two revolutions of the crank shaft). This engine has got valves for controlling the inlet of charge and outlet of exhaust gases. The opening and closing of the valve is controlled by cams, fitted on camshaft. The camshaft is driven by crankshaft with the help of suitable gears or chains. The camshaft runs at half the speed of the crankshaft.

The sequence of events taking place in IC Engine are as follows:

1. Suction stroke 2. Compression stroke 3. Power stroke 4. Exhaust stroke

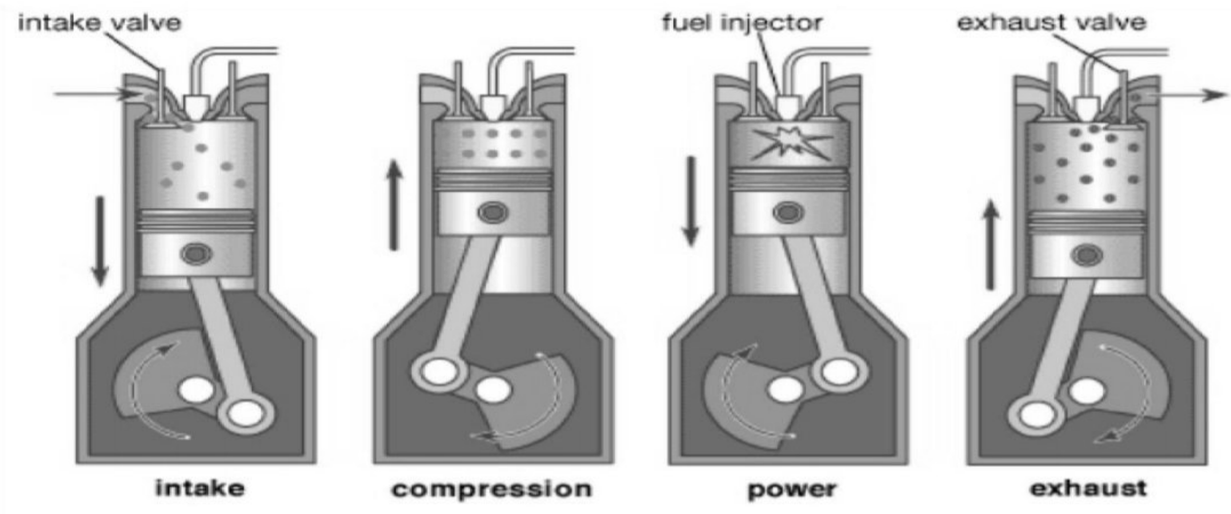
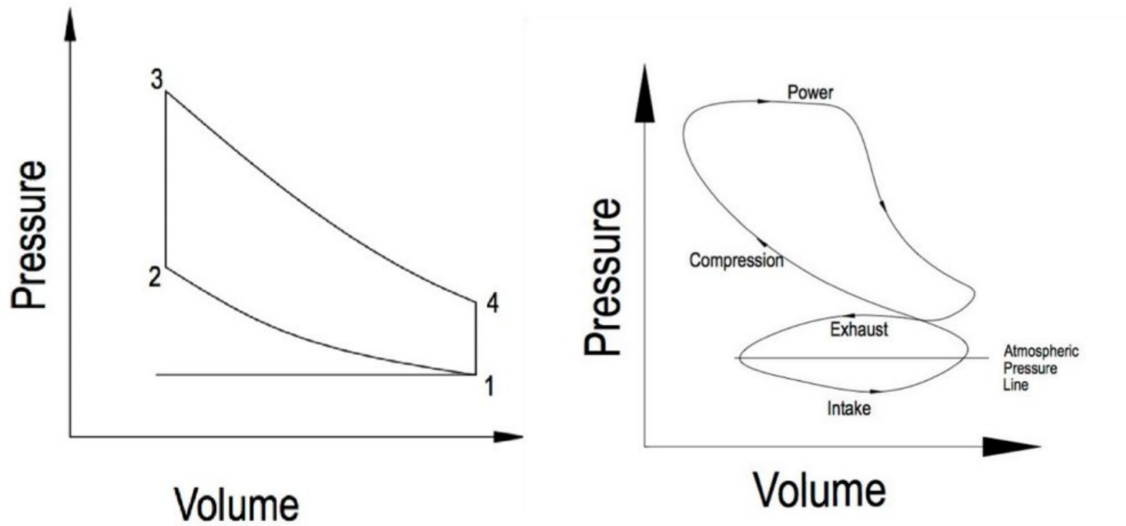


Fig.1.1 Four strokes of single cylinder CI engine

The plots of actual and real p-v cycle of single cylinder of a Four stroke engine is as given in the figure 1.2.



(i) (ii)

Fig. 1.2 P-V cycle of diesel engine (1) ideal (2) actual

2.4.1 Working principle of diesel engine

In diesel engines only air is drawn into the cylinder and the fuel is injected via injector. The engine has high compression ratio and hence the air in the cylinder attains very high temperature and pressure at the end of the compression stroke. The fuel is sprayed into the cylinder in atomized form using injectors at the end of the compression stroke.The fuel gets ignited, begins to burn and produce a lot of heat due to high temperature. Due to the heat produced expansion of gases takes place which moves the piston downward and rotates the crank shaft. The torque available at the rotating crank shaft is used to do any mechanical work.

2.4.2 Special features of diesel engine

1. High compression ratio ranging from 14:1 to 22:1.
2. The engine attains high pressure ranging from 30 to 45 kg/cm^2 and high temperature of about 500°C during the compression stroke.
3. The fuel is injected into the cylinder through injectors at a very high pressure ranging from 120 to 200 kg/cm^2 at the end of compression stroke.
4. Ignition of fuel takes place due to heat of compression only.
5. No external spark in diesel engine.
6. Diesel engine has comparatively better slogging or lugging ability i.e. it maintains a higher value of torque for a longer duration of time at a lower speed.

2.5 Spark ignition engines

Internal combustion engines are divided into spark ignition engines and compression ignition engines. Almost all automobiles today use spark ignition engines while trailers and some big trucks use compression ignition engines. The main difference between the two is the way in which the air to fuel mixture is ignited, and the design of the chamber which leads to certain power and efficiency characteristics.

The actions in the spark ignition engine can be divided into four parts. Each part consists of a stroke. One stroke is the movement of the piston from BDC to TDC or vice versa. The complete cycle of events inside the engine cylinder requires four piston strokes and they are suction, compression, power and exhaust. The crankshaft makes two complete revolutions to complete the four piston strokes. This makes the engine a four – stroke cycle engine.

Intake stroke

During the intake stroke of SI engine, the piston moves down and the intake valve opens. Air – fuel mixture flows through the intake port and into the cylinder. As the piston passes through BDC, the intake valve closes. This seals off the upper end of the cylinder.

Compression stroke

After the piston passes BDC, it starts moving up. Both the valves are closed. The upward moving piston compresses the air-fuel mixture between the top of piston and the cylinder head in the combustion chamber. The mixture is compressed 1/8 or less of its original volume. The amount of mixture that is compressed is compression ratio.

Power stroke

As the piston nears TDC, an electric spark jumps the gap at the spark plug at the end of the compression stroke .The heat from the spark ignites the compressed air-fuel mixture and then burns rapidly. These high temperatures cause very high pressure which pushes down the piston. The connecting rod carries this force to the crankshaft, which turn to move drive wheels.

Exhaust stroke

As the piston approaches BDC exhaust valve opens on the power stroke. After passing through BDC, the piston moves up again and the burned gases escape through the open exhaust port. As the piston nears TDC, the intake valve opens. When the piston passes through TDC and starts down again, the exhaust valve closes.

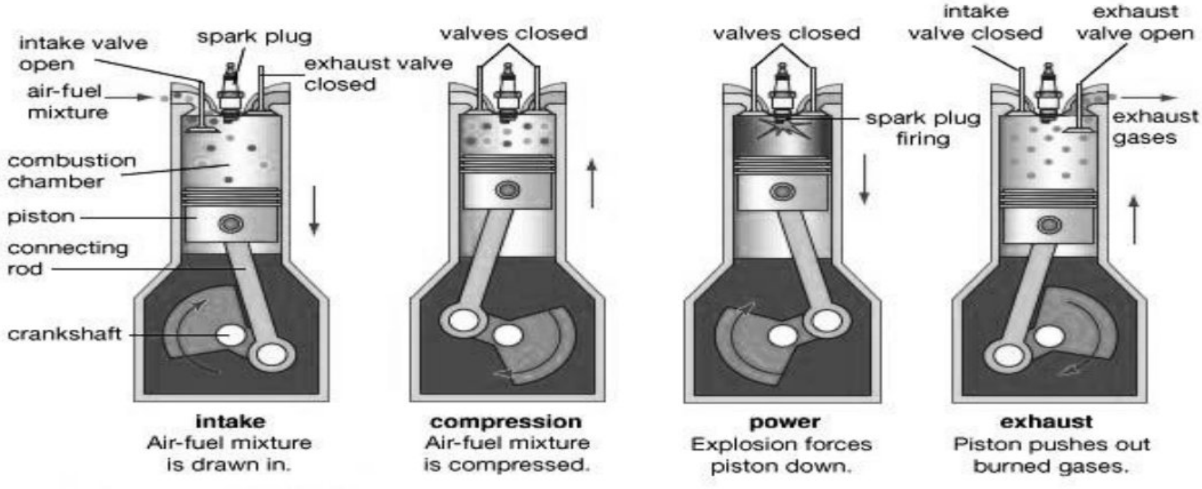


Fig. 1.3 Four strokes of single cylinder SI engine

2.6. Design for combustion chamber

An air-fuel mixture compressed to high pressure is used in SI engine. Due to being at high pressure, the air-fuel mixture must be stoichiometric to be chemically inert and to be able to ignite. The stoichiometric means that there is a particular ratio between the air and fuel. Therefore in order to ignite, the mixture must not contain too much of fuel (rich mixture) or must not contain too much of air (lean mixture) but must have an even amount of both for proper ignition. The most important aspects which are vital in a spark ignition engine are air-fuel ratio, chamber design and ignition systems used. Some basic criteria which must be kept in mind while designing a combustion chamber are:

- The distance travelled by the flame front must be minimized.
- The exhaust valve and the spark plug must be closed together
- There should be sufficient turbulence
- The end gas must be in a cool part of the combustion chamber

The design requires that the distance between the end gas and the spark plug must be close in order to have a rapidly progressing combustion. If combustion is sped up then, the engine speed is increased and therefore power output is higher, and the chain reactions that lead to knock are reduced. As stated in the second design criteria the exhaust valve, since it is very hot must be kept as far from the end gas in order to prevent knock or pre-ignition. There should be enough turbulence in order to promote rapid combustion, through mixing as suggested in third design criteria.

However too much turbulence, will lead to excessive heat transfer from the combustion chamber and too rapid combustion which causes a lot of noise. Squish areas or shrouded inlet valves generates turbulence in the combustion chambers. The fourth design criteria requires that the end gas be in a cool part of the combustion chamber. The cool part of the combustion chamber forms between the cylinder head and piston. Four common combustion chambers designs are:

- Wedge chamber head
- Hemispherical head
- Bowl in piston chamber head
- Bath-tub head

The wedge design is quite simple in design and gives good results. In the wedge chamber design the valve drive train is easy to install, but the inlet and exhaust manifold have to be on the same side of the cylinder head. The second type of combustion chamber is the hemispherical head and the advantage of this chamber is its angled valves which are used in high performance engines. It is an expensive design with twin overhead camshafts. This design allows for cross flow between inlet to exhaust, with cross flow occurring at the end of the exhaust stroke and at the beginning of the suction stroke while both valves are open. The third design is a cheaper and has good performance. The last combustion chamber design is a compact which might be expected to give economical performance.

The process by which the air to fuel mixture is prepared and put in the combustion chamber is through carburetors and fuel injectors. Spark plugs are part of all spark ignition engines. In order to start one of these engines a spark has to ignite a mixture into a flame. The way in which this spark is first initiated is through the car battery and a circuit directly which directly leads to the spark plug. The battery supplies the electric current to initiate a spark, the spark then ignites the air and fuel mixture. The type of fuel injectors used are multi-point and single-point injection. Carburetors divide into fixed and variable jet carburetors. The air and fuel mixture is analyzed as either a lean or rich mixture depending on the fuel content. A stoichiometric mixture is one in which there is a perfect ratio of air and fuel molecules. A lean mixture would be deficient in fuel where a rich one would be saturated with fuel. To achieve economic status and yet receive the maximum power the engine would have to use a lean mixture and a rich one at full throttle. When the throttle is fully opened (WOT) and a lean mixture is used the power output is economical because of the weak fuel. When the throttle is opened the combustion chamber needs the air to fuel mixture. Since, throttle is wide open, extra fuel is needed to compensate for the insufficient flow of fuel. To obtain maximum power a rich mixture is needed. For good fuel economy, all the fuel should be burnt and the "quench area where the flame is extinguished should be minimized."

2.7 Multi-fuel variable compression ratio engine

An MFVCR engine is an engine which can be run on multi-fuels by varying different compression ratios at each level. The special features of MVCR that is used in this project are:

- Extensive range of experiments
- Compression ratio variable from 5:1 to 10:1 for petrol
- Compression ratio variable from 14:1 to 20:1 for diesel
- Runs on both petrol and diesel fuels
- Variable spark timing from 0-70 deg
- Consists spark plug, ignition coil, diesel injector, diesel pump and carburetor

2.8 General description of MFVCR setup

The engine is mounted on sturdy base frame. The base frame is fabricated with mild steel "c" channel. The engine and the dynamometer are coupled using standard tyre coupling. The engine consists of a variable compression ratio head assembly. A standard air tank is fitted with a mass air flow sensor for measuring the actual volume of air drawn into the cylinder. The exhaust gas calorimeter provided within the rig enables one to conduct precise heat balance sheet. The thermocouple and necessary signal conditioner for the measurement of temperature at various points in the calorimeter are suitably provided. The panel is fabricated with suitable SWG CR sheet and as per IS standard; the front portion of the panel consists of computer, printer and UPS. The rear portion of the panel consists of all instrumentations and signal conditioner related components. Power and control wiring are suitably marked using farul for easy trouble shooting. A standard circuit drawing is normally pasted behind the panel door. The panel is finished with powder coating. The loading of the engine is controlled by the computer, hence precise loading is achieved. The test rig is provided with eddy current dynamometer.

2.9 Dynamometer

A dynamometer is a load device which is generally used for measuring the power output of an engine. Several kinds of dynamometers are common, some of them being referred to as "breaks" or "break dynamometers": dry friction break dynamometers, hydraulic or water break dynamometers and eddy current dynamometers.

Dry friction dynamometers are the oldest kind, and consist of some sort of mechanical breaking device, often a belt or frictional "shoe" which rubs a rotating hub or shaft. The hub or shaft is spun by the engine. Increasing tension in the belt, or force of the shoe against the hub increases the load on the engine.

Hydraulic dynamometers are basically hydraulic pumps where the impeller is spun by the engine. Load on the engine is varied by opening or closing a valve, which changes back pressure on the hydraulic pump.

2.10 Eddy current dynamometers

Eddy current dynamometers are electromagnetic load devices. The engine being tested spins a disk in the dynamometer. Electrical current passes through coils surrounding the disk, and induce a magnetic resistance to the motion of the disk. Varying the current varies the load on the engine. The dynamometer applies a resistance to the rotation of the engine. If the dynamometer is connected to the engine's output shaft it is referred to as an Engine Dynamometer. When the dynamometer is connected to the vehicles drive wheels it is called a Chassis Dynamometer. The force exerted on the dynamometer housing is resisted by a strain measuring device (for example a strain gage).

The software Diesel-RK is based on the first law of thermodynamics and is used for the calculation of combustion, engine performance and ecological analysis parameters. The Diesel-RK model analyses mixture formation and combustion in a diesel engine. It is also used as multi-parameter optimization characteristic technique. In this software, multi-zone of combustion model is applied in which the spray is divided into seven different zones, as shown in Fig. 5. The passes of spray evaluation are divided into three different steps: (1) initial formation of solid axial flow, (2) cumulative spray evaluation and (3) period of spray interaction with the combustion chamber wall & distribution of fuel on walls. The proposed computational software is used in analyzing combustion, performance and emission characteristics for pure diesel (PD) and four (4) biofuels: Jatropha methyl ester (JME), Ethyl ester fish oil (EEFO), Soybean methyl ester (SME) and Micro algae- oil methyl ester (MAOME) at various loading with constant compression ratio and speed.

The heat release rate in the cycle is divided into four steps and they have unique chemical and physical feature of speed of burning – (1) Ignition delay phase period, (2) premixed combustion phase (air and fuel vapor mixture of combustion during the ignition time duration), (3) mixing controlled combustion phase period and (4) late burning phase period. [33,36,39].

3.1. Governing equations

The following governing equations are implemented for the study. The fluid flow within the combustion chamber is of multi zone model [59].

3.1.1. Conservation of energy model

$$\frac{d(mu)}{dt} = -p\frac{dv}{dt} + \frac{dQ_{ht}}{dt} + \sum_{j} \dot{m}_J h_J$$

The above equation (1) is the conservation of energy. The left hand side denotes the rate of change of energy within the system. The first, second and the third term on the right hand side represents the rate of displacement work, heat transfer rate and enthalpy flux respectively.

2.1.2. Fractional mean effect pressure model

$$FMEP = a + bP_{max} + cV_p$$

In equation a, b and c are constants, P_{max} and V_p are cylinder peak pressure and mean piston velocity.

2.1.3. Specific fuel consumption model

$$SFC = \frac{\dot{m}_f}{P_b}$$

3.2 Heat release model

The fuel is sprayed into two stages: before jet impingement and after impingement

(a) Before impingement
   1. The dense conical core
   2. The dense forward front
   3. The dilute outer sleeve

(b) After impingement
   1. The axial conical core of the NWF
   2. The dense core of the NWF on the piston bowl surface
   3. The dense forward front of the NWF
   4. The dilute outer sleeve of the NWF

The following governing equations discussed here are taken into consideration in this model and used for calculation of heat release in the cycle, since the combustion of a fuel in an internal combustion engine occurs in different phases [33,36,39].

1. Ignition delay phase period :

   The calculation of auto ignition delay period by

$$\tau = 3.8 \times 10^{-6}(1 - 1.6 \times 10^{-4}.n)\sqrt{\frac{T}{p}}\exp\left(\frac{E_a}{8.312T} - \frac{70}{CN + 25}\right)$$

2. Premixed combustion

Premixed combustion period the heat release rate by:

$$\frac{dx}{d\tau} = \Phi_0 \times \left(A_0 \left({}^{m_f}\!\big/_{v_i} \right) \times \left(\sigma_{ud} - x_0 \right) \times \left(0.1 \times \sigma_{ud} + x_0 \right) \right) + \Phi_1 \times \left(\frac{d\sigma_u}{d\tau} \right)$$

3. Mixing controlled combustion phase period:

In mixing controlled combustion phase period the heat release rate can be given by:

$$\frac{dx}{d\tau} = \Phi_1 \times \left(\frac{d\sigma_u}{d\tau} \right) + \Phi_2 \times \left(A_2 \left({}^{m_f}\!\big/_{v_c} \right) \times \left(\sigma_u - x \right) \times \left(\alpha - x \right) \right)$$

4. Late burning phase period:

In this phase period the heat release rate is given by:

$$\frac{dx}{d\tau} = \Phi_3 A_3 K_T \left(1 - x\right)\left(\xi_b \alpha - x\right)$$

In these four phase period equation is given $\Phi = \Phi = \Phi = \Phi$ which function describe the completeness of fuel vapour combustion in this phase period.

5. Model of NO_x formation

The thermal NO is calculated using chain Zeldovich mechanism, which is given as below. The volume concentration of NO in combustion products is calculated using some equations discussed here. The equilibrium concentrations of eighteen species are calculated on every step and the overall system of equations contains fourteen equilibrium equations, three material balance equations and Dalton equation of partial pressure [39,41].

$$[O_2] \leftrightarrow [2O]$$

$$[N_2]+[O] \leftrightarrow [NO]+[N]$$

$$[N]+[O_2] \leftrightarrow [NO]+[O]$$

$$\frac{dE}{d\theta}=\frac{P \times A \times e^{\frac{B}{T_b}} D.E.\left\{1-\left(\frac{F}{G}\right)^2\right\}}{R.T_b.\left(1+\frac{C}{T_b}.e^{\frac{C}{T_b}}.\frac{F}{H}\right)}.\frac{1}{\omega}$$

6. Bosch smoke number

The relationship for exhaust gases of particulate matter emitted as an operation of a function designed with BSN by Alkidas and written as the mathematical form of the equation:

$$[PM]=Z_{PM}565\left(\ln\frac{10}{10-BN}\right)^{w}$$

CHAPTER 2

LITERATURE REVIEW

Various researcher have investigated the performance, exhaust emission and combustion parameter of engine used of different alternative fuel. The less pollutant fuel found by researcher for two and four stroke IC engine. These all things are presented in this literature review.

2.1 Review on CI Engine

Agarwal et al. [11] have made an investigation of the bio-fuel (alcohols and biodiesel) applications as fuel for internal combustion engines through a review. The review is focused on the production, characterization and current status of vegetable oil and biodiesel as well as the experimental research work carried out in various countries. After this review, it is found that the use of bio-fuels as IC engine fuels can play a vital role in helping the developed and developing countries to reduce the environmental impact of fossil fuels.

Heywood [12] has studied the current status and development potential of various types of internal combustion engines, expected changes in transportation fuels from natural petroleum, impact of alternative fuels etc. It has been suggested that dominant alternative fuels impact (even with massive and successful alternative fuels development programs) is probably 20 years or so beyond the year 2000. It has been also suggested to explore what engine options (heat engines or others) are potentially attractive on a 30-40 year time frame, when alternative energy sources to petroleum will begin to dominate.

Graboski et al. [13] have reviewed the status of fat and oil derived diesel fuels with respect to fuel properties, engine performance, and emissions and have concluded that these biodiesels can be suitable fuels for diesel engines, the lubricity of these fuels is superior to conventional diesel, emissions of PM can be reduced dramatically through the use of biodiesel in engines. Emissions of NOx increase significantly for both neat and blended fuels in both two and fourstroke engines.

Arbab et al. [14] have reviewed the fuel properties, engine performance and emission characteristics of commonly used different vegetable (jatropha, palm, coconut, cottonseed, sunflower, soybean and canola / rapeseed) based biodiesel derived from experimental results at different conditions performed worldwide. The potential guidelines to improve engine performance and emission characteristics have been discussed using different biodiesels and their blends as well. It has been concluded that single biodiesel cannot improve both engine performance and emission at a time, but blend of two or more biodiesels may be able to achieve this goal. In this respect, a blend of jatropha and coconut biodiesel has been suggested. This study provides a comparative baseline to make an easy comparison among the biodiesels in respect to fuel properties, engine performance and emission characteristics.

Shahir et al. [15] have reviewed renewable and sustainable energy sources for automobiles. The feasibility of biodiesel in automobiles has been evaluated with special emphasis on emission aspects. The few aspects of performance and durability have also been considered. It has been concluded that the use of biodiesel with little or no modification in engine leads to substantial reduction in exhaust emissions.

Guardiola et al. [16] presented study optimized heat released law in a diesel engine and maximised indicate efficiency rerated to different constraint like maximum cylinder pressure, maximum cylinder pressure derivate and NO_x emission. The applied model simple and provided combustion process and consist of energy balance model aimed to provided cylinder pressure temperature calculated high pressure loop for engine cycle at gas condition intake valve closed and heat release law. The gas pressure and temperature allow computing engine efficiency and exhausting NO_x emission. The result compared applied model and experimental data and shows that model result despite and able to reproduce the engine efficiency and exhaust NO_x emission. The after validation result to solve by dynamic programming (DP).

Usta [17] has experimentally investigated the performance and exhaust emission characteristics of a turbocharged indirect injection diesel engine fuelled with tobacco seed oil methyl ester at full and partial loads and have found that the addition of tobacco seed oil methyl ester to the diesel fuel reduces CO and SO_2 emissions while causing slightly higher NOx emissions. Meanwhile, it has been found that the power and the efficiency increase slightly with the addition of tobacco seed oil methyl ester.

Agarwal et al. [18] have investigated experimentally the performance (which include thermal efficiency, brake specific fuel consumption, brake specific energy consumption and exhaust gas temperature) and emissions characteristics (mass emissions of CO, HC, NO and smoke opacity) of a compression ignition engine fuelled with karanja oil and its blends (10%, 20%, 50% and 75%) with mineral diesel and the effect of temperature on the viscosity of karanja oil. A series of tests in a single cylinder agricultural diesel engine with and without preheating of karanja oil has been conducted and results obtained from the tests in each case have been compared with baseline data of mineral diesel and found that the karanja oil blends with diesel up to 50% (v/v) without preheating as well as with preheating would replace diesel for running the CI engine for lower emissions and improved performance.

2.2 Simulation work on CI Engine

Taghavifar et al. [19] presented numerical study addressing the specification engine and second law of thermodynamic of the compression ignition (CI) diesel engine used of hydrogen, dimethyl ether (DME) and diesel fueled at six different speed. 3-D numerical simulation first carried out and calculate availability of in- house code developed. Separately analysis of availability for chemical and thermo-mechanical in form of chemical and mechanical efficiency. The result obtained in cylinder pressure, mean effective pressure and temperature distribution in engine cylinder at all crank angle (CA) and engine speed. The engine performance is better at 2000rpm indicate power, mean effective pressure, chemical and thermo-mechanical availability respective fuel. The result shows that used of hydrogen highest temperature (2736K) and wall flux (29160W) and engine speed varies from 1500rpm to 4000rpm reduced by crank-angle revolution from 43.3% to 10.1% at 10-40 ˚CA after TDC (top-dead-centre).

Sharma et al. [20] used of orientation swirl ration in drooping peak pressure of a HCCI (homogenous charge compression ignition) engine by numerically analysed method under the various working parameter with used of bowl shape piston geometry of a single cylinder 1.6 L and used of ECFM-3D combustion model analysis by STAR-CD commercial software. This numerical study carried out the effect of compression ratio, equivalence ratio, and exhaust gas recirculation (EGR) with different swirl ration pressure reduction approached of combustion pressure. The result shows that swirl ration impacted effect reduction of peak pressure of combustion chamber of HCCI engine. The numerical result in achieving about 21% reduced peak pressure when used of a swirl ration 4 with 30% EGR and compared to a swirl ratio of 1 with 0% EGR.

Datta and Mandal [21] used of Diesel-RK model for numerical simulation a single cylinder, four stroke, direct injected (DI), naturally compression ignition (CI), naturally aspirated inline diesel engine at constant compression ratio (CR) 17.5 and speed 1500rpm with injection timing constant (23˚ bTDC). They observed that increase brake thermal efficiency and brake specific fuel consumption (BSFC) at addition of methanol and ethanol to diesel. The most important factor of exhaust pollutants reduction of CO_2 with use of alcohol blend with diesel and NO_x reduction with ethanol use with diesel and also observed reverse trend of particular matter (PM) and smoke emission.

Soni and Gupta [22] applied a two-stage strategy of emission reduction to a kirloskar single cylinder diesel locomotive (model TV1) to accomplish more stringent emission norms. First stage includes numerical simulation of methanol fuel at low load operating condition, the three different method was used in the second stage namely initial swirl ratio, EGR and water addition to further reduce the emission. The commercial CFD software AVL FIRE was used to perform numerical simulation. In addition performance parameter (BSFC and BTE) are analysed for diesel fuel, diesel-methanol fuel and effective method from three emission reduction methods. The study concluded with a result that water addition method is most effective method because it reduces the emissions effectively.

Soni and Gupta [23] numerically investigated work out a two stage strategy to achieve higher level of emission reduction to meet more stringent emission norms. In first stage is optimum blend from diesel-methanol blend in terms of emission reduction and next stage simulation has been performed by three different methods of emission reduction like variation of swirl ratio, EGR and adding water with diesel methanol blended. The k-zeta-f turbulence model using the numerical simulation is performed on a single cylinder, Di diesel engine having a hemispherical bowl shaped piston using CFD software AVL FIRE. The results found that water blend method tends to reduces NOx emission by 95% and soot by 14% with respect to emission of base fuel.

Chowdary et al. [24] used of combined effect of advance start of injection (SOI) and exhaust gas recirculation (EGR) by numerical analysis using commercial tool of CONVERGE CFD (computational fluid dynamic) an single cylinder CI engine at constant speed. Used of advanced start of injection from 11-14.5 degree crank angle at bTDC and EGR rate increase 0- 10 %. The result found that soot and NO_x emission decreased by 21% and 1.2 % respectively at advance SOI 3.5 and 10% of EGR.

Dhanasekaran et al. [25] used of eco-friendly fuel by recycling waste cooking oil (WCO) with diesel (D) and n-pentanol (P) to improve fuel-spray characteristics for single cylinder, four stroke ,direct injected (DI), diesel engine to experimentally analysed at three different blending volume basis (D50-WCO45-P5, D50-WCO40-P10 and D50-WCO30-P20). The analysis of combustion characteristics and comparison with diesel and D50-WCO50 (50% of diesel + 50% of WCO) not considering effect of exhaust gas recirculation. The results indicted when addition of n-heptanol shows that improved fuel properties all blending ration compared to D50-WCO50 but viscosity is slowdown up to 45%, cetane number and density are similar as a diesel. The addition of n-haptenol to D50-WCO50 increased brake specific fuel consumption (BSFC) all blending fuel territory. BSFC of the blend D50-WCO30-P20 was observed 8.8 % higher to diesel at high load condition without EGR and Brake thermal efficiency (BTE) higher comparable with diesel and smoke capacity reduced up to 13.6%.

Sharma and Murugan [26] experimentally investigated effect of nozzle opening pressure (NOP) an single cylinder four stroke directed injected compression ignition engine used of JMETPO blend (which contains 80% jatropha methyl ester and 20% tyre pyrolysis oil) volume basis. The optimum blend find five different NOP 210, 220, 230, 240 and 250 bar and found the cylinder peak pressure and maximum heat release rate higher at 220 bar compared to 200 bar at full load operating condition and brake thermal efficiency improved 5.12 %, smoke opacity, brake specific carbon monoxide and hydrocarbon emission were reduced by about 9.5%, 1.57% and 6.26% respectively at full load condition.

Datta and Mandal [27] numerically obtained performance of compression ignition (CI) engine used of different biodiesel-alcohol as a fuel. The simulation work obtained by Diesel-RK model on a single cylinder, four stroke, direct injected diesel engine. The effect of two alcohol namely ethanol and methanol addition with palm stearin biodiesel used of separately obtained and compared results. The result shown slightly more brake specific fuel consumption and brake thermal efficiency used of biodiesel-alcohol blends compared to biodiesel and also exhaust NOx emission is reduced. The heat release rate, ignition delay, particular matter (PM) and smoke emission is high with alcohol blended fuel.

CHAPTER 3

OBJECTIVE OF THE PROJECT

- To obtain the performance, combustion and emission parameters of pure diesel experimentally using MFVCR engine set up.

  In this experiment, the rig is used to determine the performance characteristics when operated with the supply of pure diesel at different load conditions. An engineering approach to analyze the characteristics of pure diesel was to be evaluated.
- To obtain the parameters of different bio fuels by Numerical simulation using Diesel RK software at various blend ratios (B0, B20, B40 & B100) at various loading conditions (25%, 50%, 75% & 100%).
- To analyse and compare the obtained parameters of various bio fuels at various blend ratios with the performance, combustion and emission parameters of pure diesel to get the optimum blend ratio to be used for Bio Diesels.

CHAPTER 4
METHODOLOGY

4.1. Finding the performance parameters of pure Diesel using experimental set up of MFVCR

In order to get the validation, numerical simulation and comparison of parameters of different bio fuels, we need to get the parameters of pure diesel at various loads experimentally.

The experimental set-up is located on the premises of NIT Manipur. Figure 1 shows the arrangement of the experiment, from which the analysis was made. The set up consists of: Main frame, Engine Assembly, Data Acquisition systems.

PROCEDURE

The VCR engine is set to a compression ratio 17.5. Diesel is filled onto the Diesel tanks before starting the experiment. The Rota meter checked to verify the flow of cooling water into the engine and calorimeter. The module is set to CI mode and the engine is started using key. Provisions are provided to make sure that the engine runs for 5-10 minutes before the data were taken. The load is provided by turning the knob on the dynamometer module on the main frame.

The engine is set to a compression ratio of 17.5 and the loads has been changed as 25%, 50%, 75% and 100% with a stable time gap or interval from load to load (say 15min each and the sensors senses each load change and the graphs were plotted on desktop between various parameters.

- Before starting the engine, we have to check the fuel condition, if not we have to refuel it. Always make sure that fuel should be contained more than half of the fuel tank.
- Check the water at least 40 to 60 lit/hr, check the fuel injector, check the pump
- Using torque wrench, fasten the heavy bolts by applying torque of (200 to 220 Nm) approx and set the compression ratio to CR17.5

- Now turn on the system and double click on the icon of MFVCR provided by legion brothers(as used for this set up) and click on the PFI mode i.e., port fuel injection mode and ready to record the average pressures, heat transfer rate and all other parameters that are needed/required for plotting the graphs
- Now start the engine by turning on the fuel supply (ON/OFF) button, and using rotating handle, rotate the shaft of the engine. Also make sure the engine is in

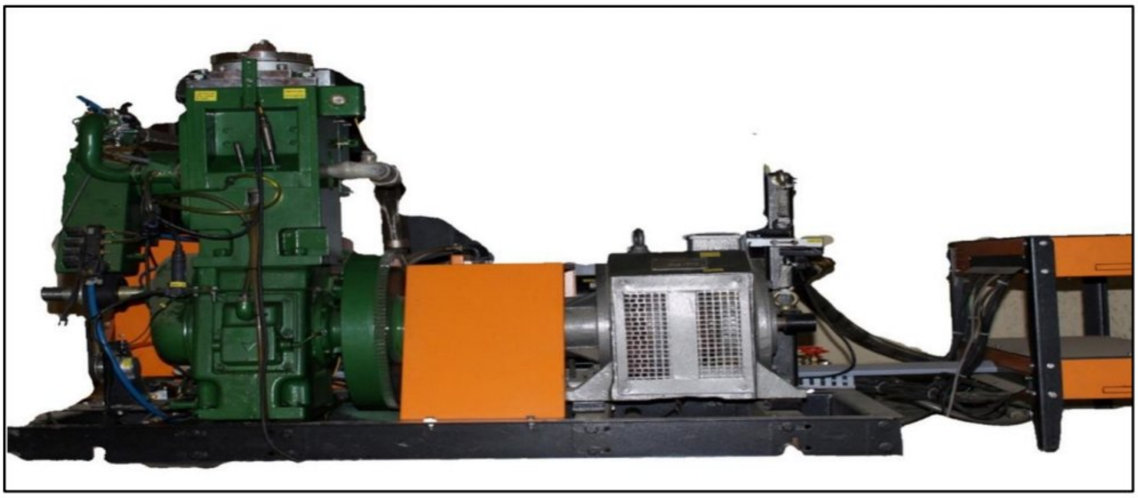


Fig. 4.1 Front view of MFVCR engine

decompression mode while starting and after rotating for some revolutions and then keep it down (decompression button)

- Run the engine atleast for 30mins in order to get stability. Once it gets stabilized, we start adding the loads or changing the loads in an order from 25% to 100%.
- Note the point that loading is done only then when the engine reaches the stable rpm of 1000 to 1200 rpm.
- Also maintain the stability always, while changing the loads to next from previous for atleast 15mins per change of the loads
- The type of loading is eddy current loading, as it provides greater accuracy and very precise in its loading
- Thus the experimentation results the graphs through GUI for the CR17.5 at various loads.

4.2 Validation of DIESEL RK Tool

To validate the results from the Diesel-RK model simulation tool proposed herewith, experimental results were compared, as shown in Fig.

The results are compared with the standard published results which are taken as reference from the work of **Gnanasekaran S, et al. (2016)**

Table 4.1**:** Comparison of input parameters

Parameter	Gnanasekaran S, et al. (2016)	Authors work
CR	17.5	17.5
FIP	180 bar	220 bar
Speed	1500 rpm	1500 rpm
Load	100%	100%
Injection timing	24.0° b TDC	23.0° b TDC
Cooling system	Air	water
Fuel	Diesel	diesel
Inlet valve open	5° before TDC	4.5° before TDC
Inlet valve closed	35° BDC	35.5° after BDC
Outlet valve open	35° before BDC	35.5° before BDC
Outlet valve closed	5° after TDC	4.5° after TDC

The above compared results are almost close which states that the experimental results are valid. Now we have to compare these results with numerically simulated results which are obtained using Diesel RK software.

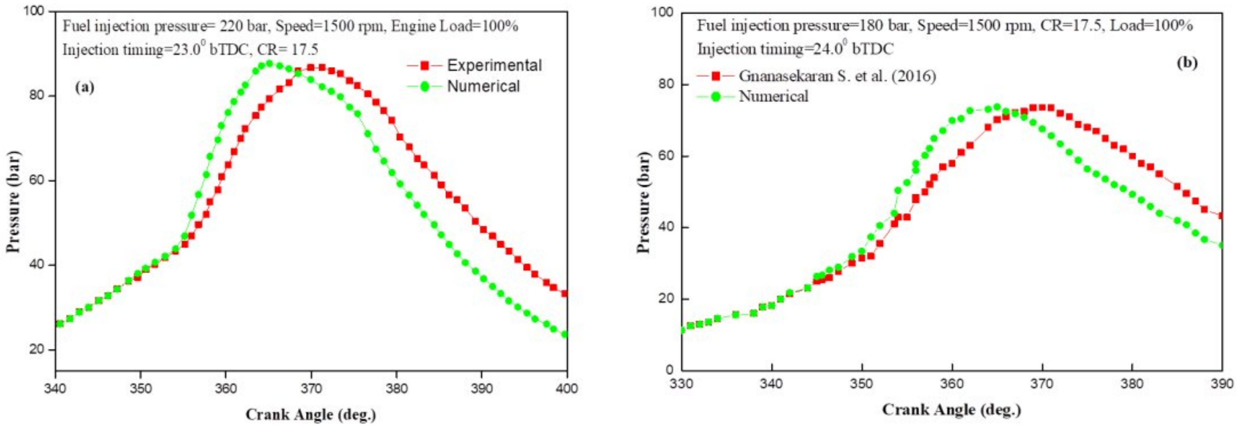


Fig.4.2 Validation of Diesel RK software.

4.3. Finding the fuel properties of taken bio fuels at different blend ratios numerically

The physical and chemical properties of various bio fuels are collected from the standard publications and are tabled as below.

Table 4.2: Physical and chemical properties of test fuels.

Properties	**PD**	**JME**	**SME**	**RME**	**PSBD**	**LME**	**CSOBD**	**EEFO**	**MAOME**
		[29]	**[36]**	**[36]**	**[39]**	**[42]**	**[43]**	**[44]**	**[45]**
Density (kg/m^3)	830	881	885	874	880	865	864	885	864
Viscosity at 40 °C	3.2	4.12	5.2	7.9	–	4.2	4.14	4.741	4.41
Cetane number	48	48.13	51.3	54.4	60	48	52	52.6	–
LHV (MJ/kg)	42.5	39.5	36.22	39.45	34.45	40.75	36.8	40.05	39.1
Flash point(°C)	56	165	–	244	–	161	–	114	115
Fire point (°C)	63	–	–	–	–	–	–	125	–
Oxygen (%)	0.4	9.6	10.81	10.9	11.5	11.72	10	–	–

Carbon (%)	87.0	–	77.31	77.0	77.0	78.14	–	–	–
Hydrogen (%)	12.6	–	11.88	12.1	11.5	9.98	–	–	–
Sulfur (%)	0	–	0.5	0.15	–	0.05	–	–	0

From the above fuel properties of the bio fuels some are selected and the properties of those bio fuels at various blend ratios are obtained using mathematical blend formulae and the parameters obtained are as follow.

Table 4.3: Physical and chemical properties of Soybean methyl ester at various blend ratio

Properties	**PD**	**SME20**	**SME30**	**SME40**	**SME100[36]**
Density (kg/m^3)	830	841	847	852	885
Viscosity at 40 °C	2.6	2.918	3.091	3.275	4.630
Cetane number	48	48.694	49.035	49.371	51.30
LHV (MJ/kg)	42.5	41.178	40.530	39.890	36.22
Flash point (°C)	50	64	71	78	120
Oxygen (%)	0.6	2.748	3.802	4.842	10.81
Carbon (%)	86.2	84.328	83.411	82.506	77.31
Hydrogen (%)	13.2	12.922	12.785	12.651	11.88

Table 4.4: Physical and chemical properties of microalgae oil methyl ester at various blend ratio

Properties	PD	B20	B30	B40	B100[45]
Density (kg/m^3)	830	836.172	839.225	842.256	860
Viscosity at 40 °C	2.6	3.037	3.283	3.549	5.66
Cetane number	48	48.841	49.254	49.661	52
LHV (MJ/kg)	42.5	42.260	42.142	42.026	41.360
Flash point (°C)	50	---	---	---	---
Oxygen (%)	0.6	2.477	3.357	4.306	2.520
Carbon (%)	86.2	233.145	305.186	376.297	784.400
Hydrogen (%)	13.2	12.955	12.836	12.718	12.040

Table 4.5: Physical and chemical properties of Ethyl ester Fish oil at various blend ratio

Properties	PD	FB20	FB30	FB40	FB100
Density (kg/m^3)	830	136.172	839.225	842.256	860
Viscosity at 40 °C	2.6	2.861	3.0023	8.149	4.2
Cetane number	48	48.968	49.442	49.911	52.6
LHV (MJ/kg)	42.5	41.984	41.731	41.482	40.05
Flash point (°C)	50	---	---	---	---
Oxygen (%)	0.6	2.767	3.830	4.879	10.9
Carbon (%)	86.2	84.263	83.314	82.377	77
Hydrogen (%)	13.2	12.968	12.854	12.742	12.1

Table 4.6: Physical and chemical properties of Waste coocking oil at various blend ratio

Properties	PD	WCO10	WCO20	WCO30	WCO40	WCO100
Density (kg/m^3)	830	834	838	842	846	871
Viscosity at 40 °C	2.6	2.8	3	3.2	3.4	4.6
Cetane number	48	50.1	50.2	50.3	50.4	51
LHV (MJ/kg)	42.5	---	----	---	---	---
Flash point (°C)	50	90.3	130.6	170.9	211.2	453
Oxygen (%)	0.6	1.62	2.64	3.66	4.68	10.8
Carbon (%)	86.2	85.29	84.38	83.47	82.56	77.1
Hydrogen (%)	13.2	13.09	12.98	12.87	12.76	12.1

4.4 Numerical simulation of obtained fuel properties at different blend ratios using Diesel RK Software.

The process of numerical simulation of properties of various bio fuels at various blend ratios and loads involves following:

- Run the Diesel RK software and enter the required data regarding the engine parameters, mode of running, fuel used etc.
- Select the specific bio fuel and give its fuel properties and put the load conditions at 25% and run the software.
- Now repeat the process for 50%, 75% and 100% load respectively.
- Repeat the process for all selected bio fuels at each blend ratio.
- Collect the results viz. Performance, combustion and emission parameters. Compare them with the parameters of pure diesel numerically and graphically.

CHAPTER 5

RESULTS AND DISCUSSIONS

The effect of physical and chemical composition of various alter- native fuels on the combustion, performance and emission of a C.I en- gine with single cylinder, four stroke, DI, water cooled, naturally aspirated, is discussed here.

5.1. Combustion analysis

The analysis of cylinder pressure is regarded as an important tool in identifying the CI engine combustion behavior, since pressure inside the engine directly affects the engine performance [18]. In analyzing combustion, parameters such as cylinder pressure, heat release rate, combustion cylinder temperature, ignition delay, and cylinder peak pressure, maximum rise of pressure rate and cylinder peak temperature were discussed.

5.1.1. Cylinder peak pressure

- The pressure of the cylinder gradually increases with increase in engine load. The greater the ignition delay, the higher the fuel consumption thereby resulting in high peak pressure of the combustion [50].

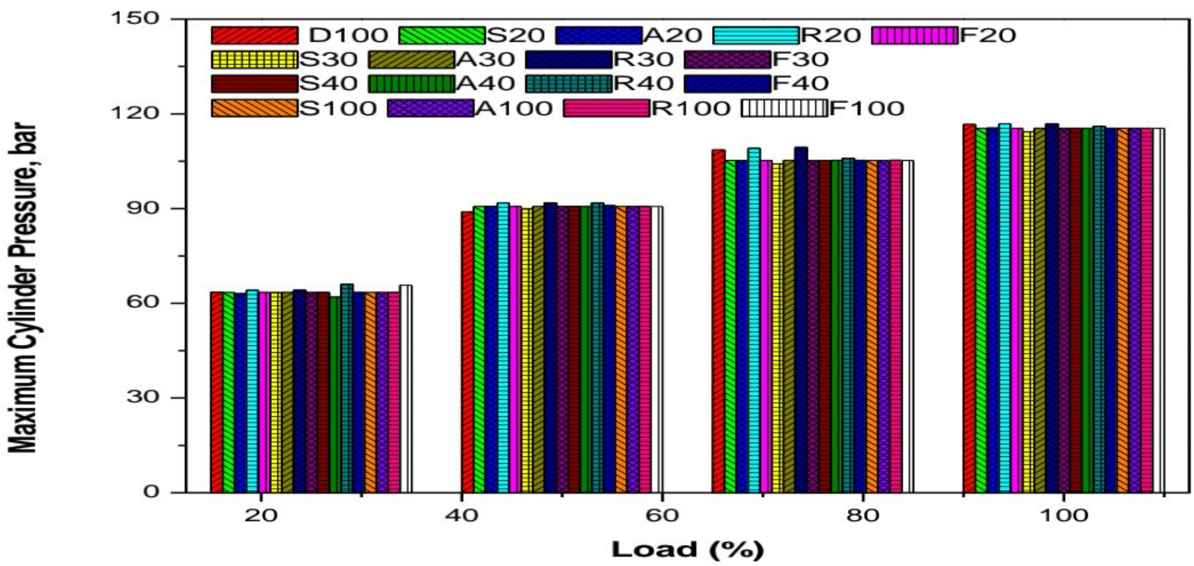


Fig. 5.1. Variation of maximum cylinder pressure versus load.

5.1.2 Cylinder peak temperature

- The cylinder peak temperature of the tested fuels increases with increase in load.

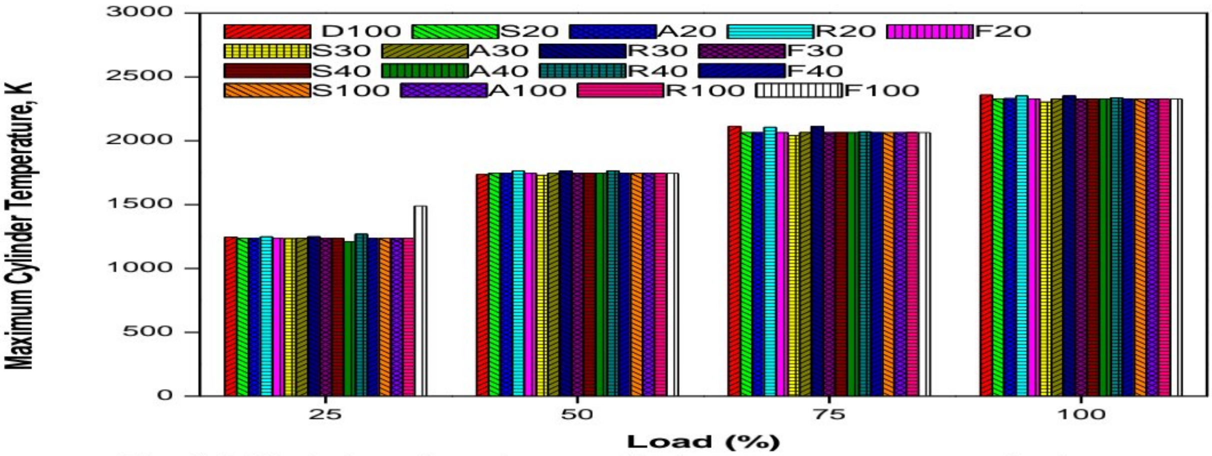


Fig. 5.2. Variation of maximum cylinder temperature versus load.

5.1.3 Maximum rate of pressure rise

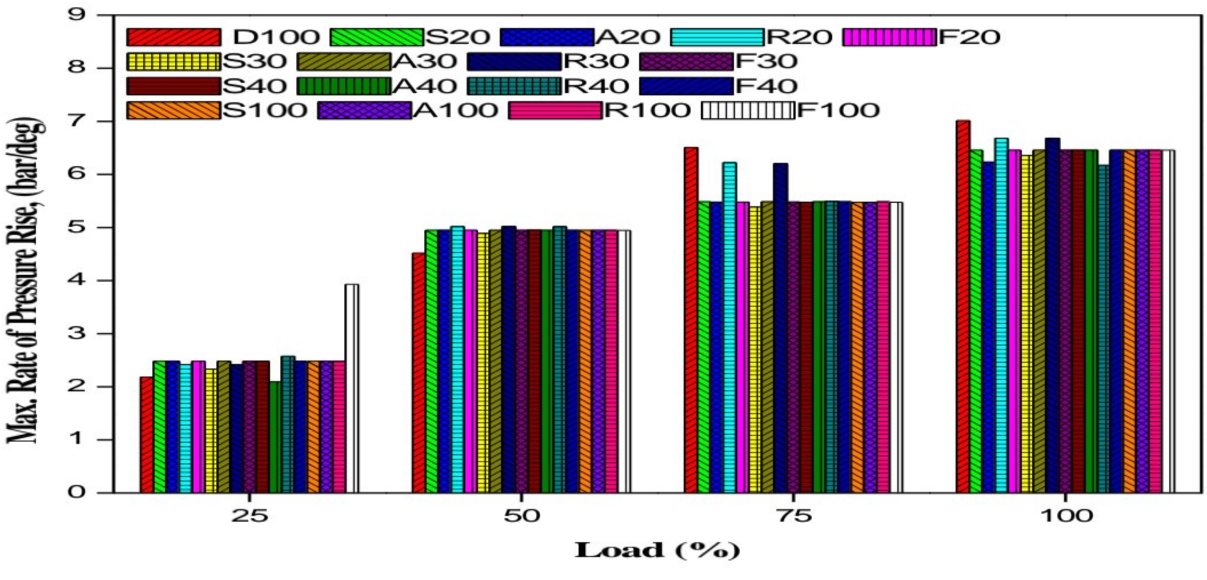


Fig. 5.3. Variation of maximum rate of pressure rise versus load.

- The maximum rate of pressure rise is an important parameter in determining the knocking tendency of an internal combustion engine [19].

- The variation in maximum rate of pressure rise with various loading is shown .The maximum rate of pressure rise increases with increase in engine load.
- The rate of pressure rise (maximum) was found to be highest for PD at maximum load.

5.2 Engine performance analysis

The parameters for performance analysis of the engine under the specified operating conditions are discussed herein. Parameters such as specific fuel consumption, brake thermal efficiency, volumetric efficiency and exhaust gas temperature is presented.

5.2.1 Indicated efficiency

- Indicated efficiency (IE) is the efficiency of the engine with respect to the power obtained inside the cylinder before transferring to the piston and cylinder. With increase in load of the engine, the IE decreases gradually.

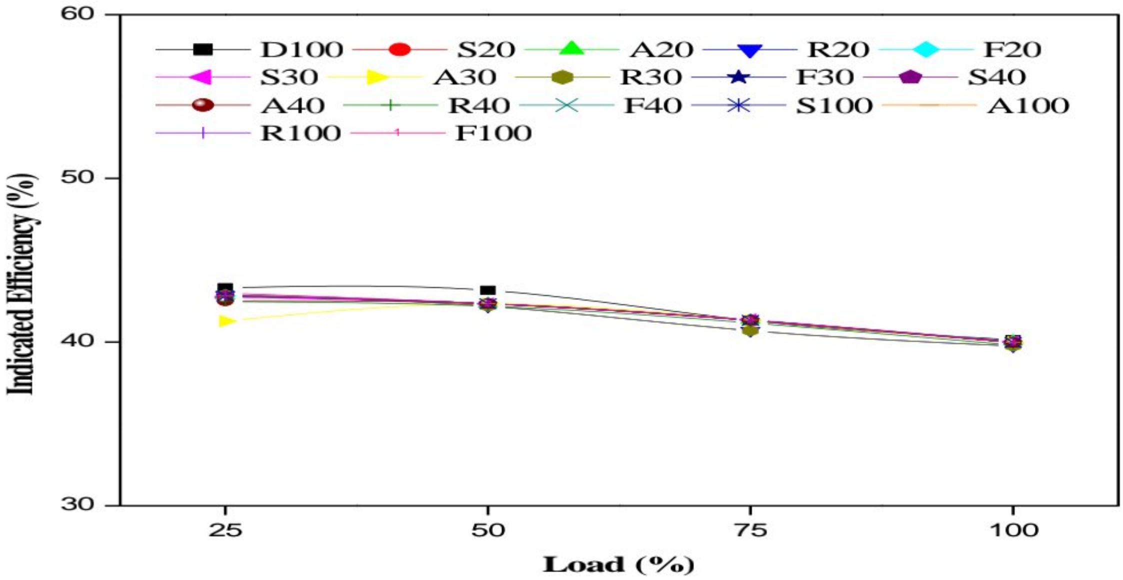


Fig. 5.4. Variation of Indicated efficiency versus load.

5.2.2 Volumetric efficiency

- The volumetric efficiency of an engine depends on inlet pressure and temperature. Since the inlet conditions are same for all fuels, there is steep decrease in volumetric efficiency of all fuels.

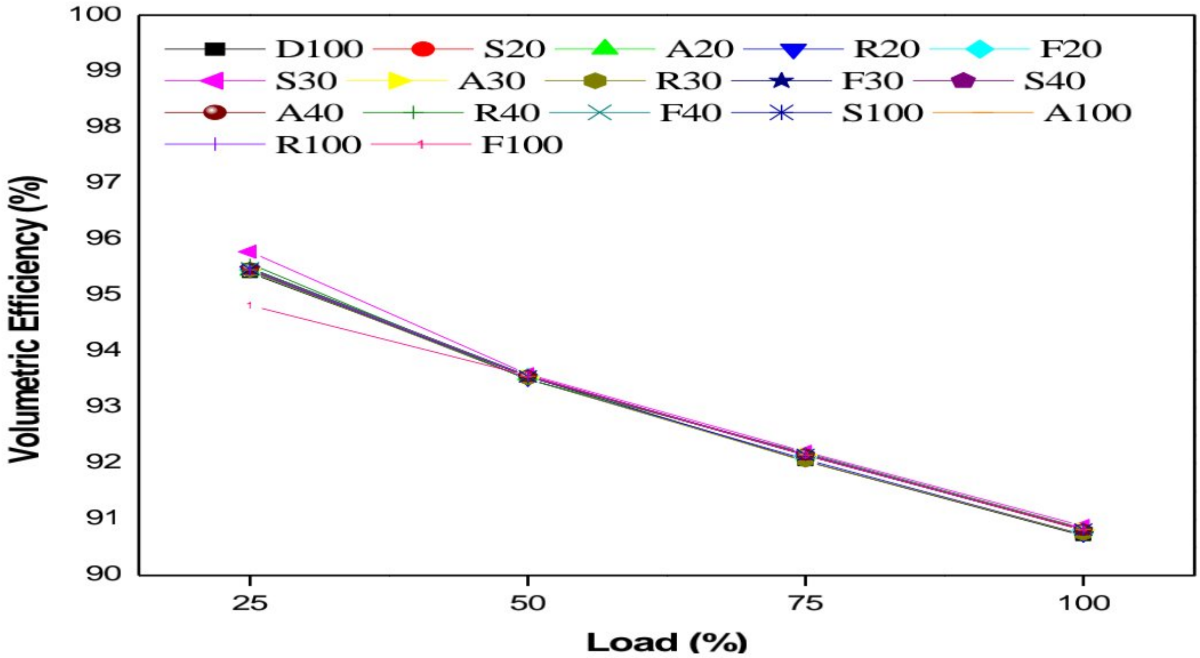


Fig. 5.5. Variation of Volumetric efficiency versus load.

5.2.3 Specific fuel consumption

- It can be observed that the SFC of the engine decreases with increase in loading of the engine.

- With increase in density and viscosity of biodiesels, there is increase in amount of injected fuel and hence SFC increases as compared to PD. But with increase in load, the SFC decreases due to reduction of engine speed and tends to stabilize for all biodiesels at higher load.

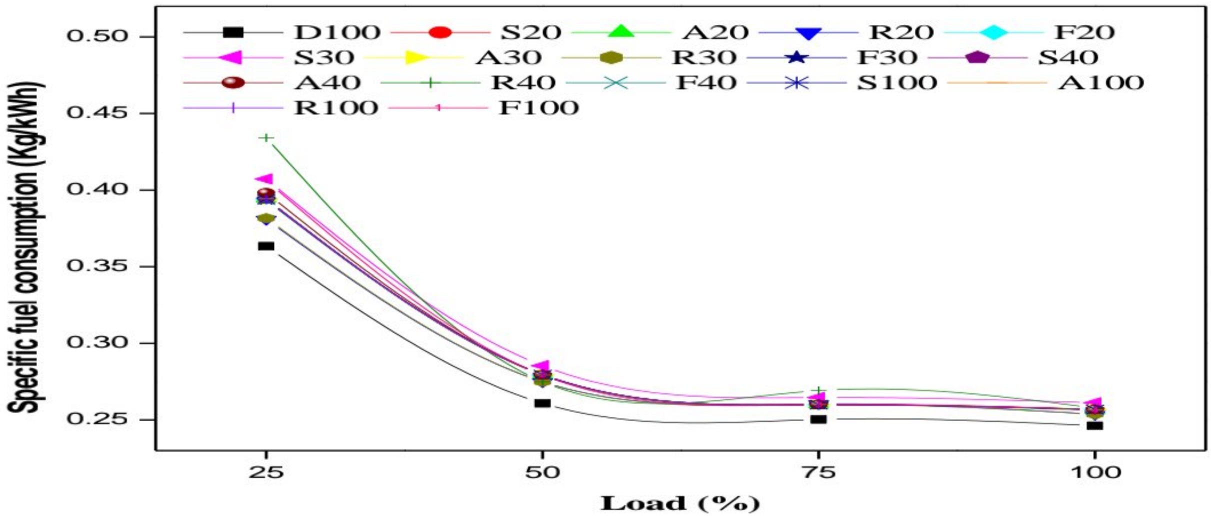


Fig. 5.6. Variation of Specific fuel consumption versus load.

5.2.4 Average Exhaust gas temperature (EGT)

- The exhaust gas temperature (EGT) is the temperature obtained at the end of expansion stroke. The EGT of an engine increases with increase in loading due to more fuel delivery inside the engine cylinder. The EGT also depends on the amount of oxygen present in the fuel and high cetane number in the fuel decrease the premixed duration time. This in turn continued the burning of the fuel till late combustion phase during the expansion stroke and hence more heat is released [43].

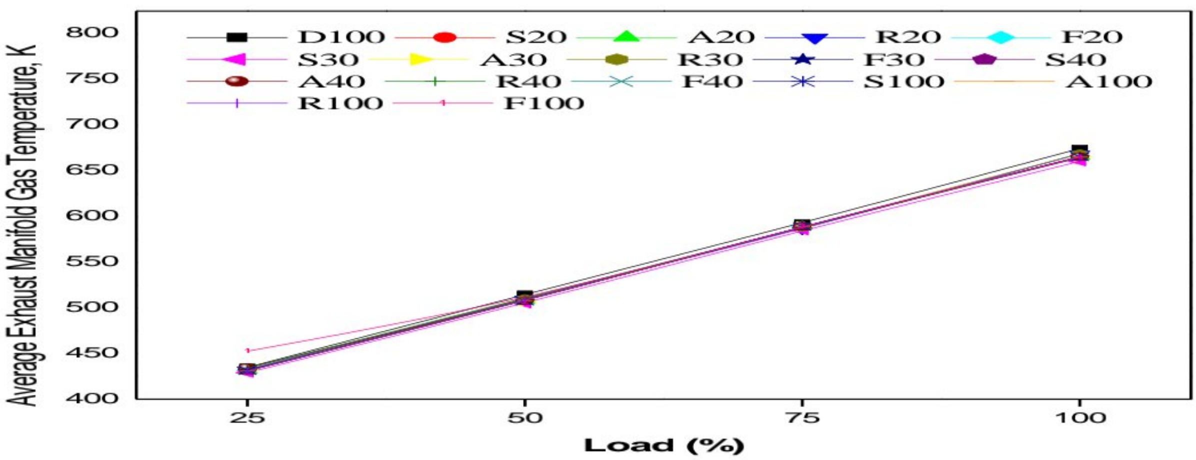


Fig. 5.7. Variation of Average Exhaust gas temperature versus load.

5.3. EMISSION ANALYSIS

The first exhaust gas emission was measured to control vehicular air pollution in 1964 by U.S.A. In the present era, more regulations and laws were amended for protection of environment. Euro 6 was kept as standard in most part of the globe [18]. The usage of petroleum products in an internal combustion engine produces many by-products in the exhaust of the engine, such as hydrocarbons, carbon monoxide, oxides of nitrogen, soot, oxides of sulfur, particulate matter, carbon-dioxide etc. [3,16]. Among the pollutants, carbon monoxide and oxides of nitrogen were more harmful. Efforts are being made for reduction of the emissions from an engine by various researchers.

5.3.1. Bosch smoke number

- The smoke decreases with high content of oxygen in the alternative fuel, contributing to complete combustion of the fuel even in rich zones [33]. The BSN of PD at full load was found to be 3.0272.

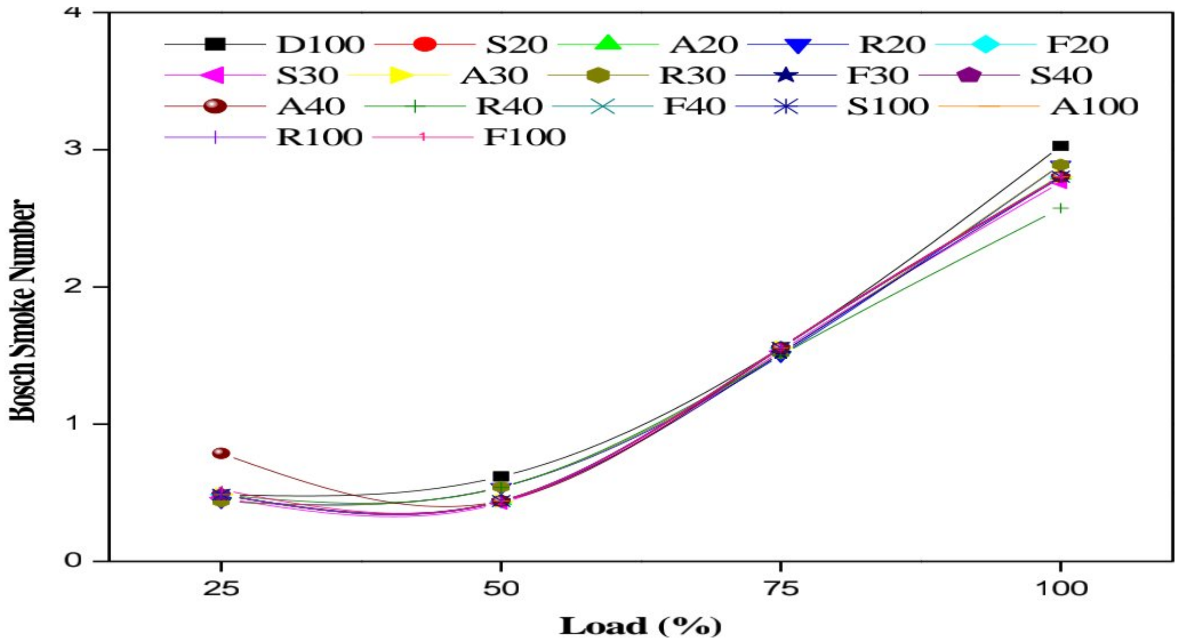


Fig. 5.8. Variation Bosch smoke number versus load.

5.3.2. Fraction of wet NOx in exhaust gas

- The NO_x emission is dependent on the temperature of engine cylinder, percentage of oxygen, time taken for reaction to take place during combustion, etc. Lower heat release rate due to low temperature of combustion chamber have led to reduction of NO_x emission [21,48,49]. The variations of NOx formation for different test fuels at various loading conditions were shown in Fig. The NO_x emission of all fuels increases with increase in load.

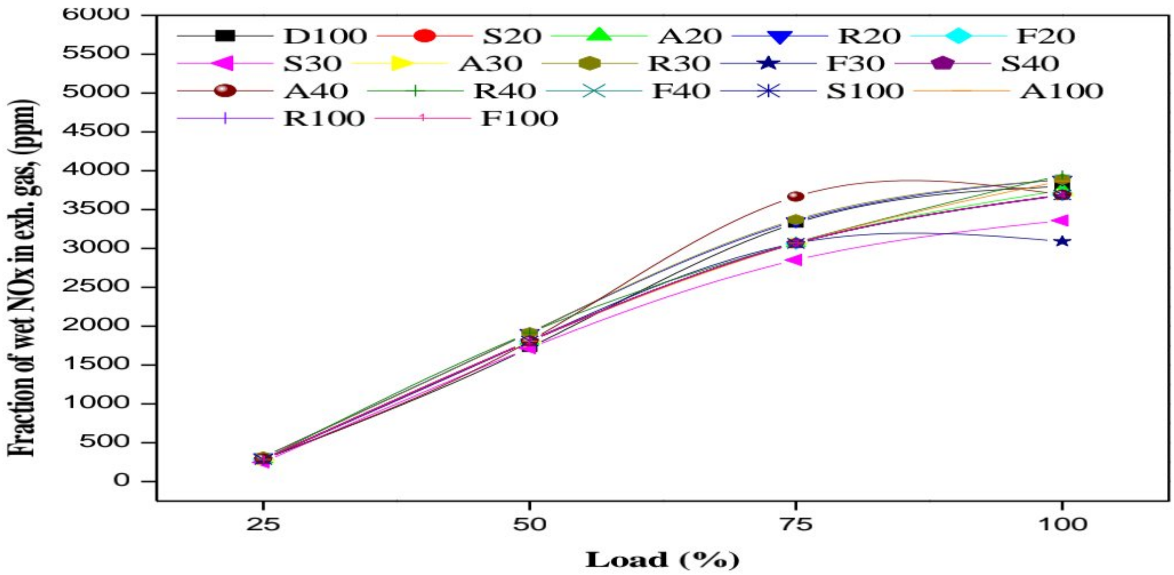


Fig. 5.9. Variation of Fraction of wet NOx in exhaust gas versus load.

5.3.3 Specific carbon dioxide emission

Fig. 5.10 shows the variation of specific carbon dioxide emission with different loading percentage for various test fuels at constant engine speed and CR. CO_2 emission form an engine shows the level of combustion rate in an engine. Increase in CO_2 emission shows complete combustion in the combustion chamber [39]. The rate of CO_2 emission decreases gradually with increase in engine load, leading to more fuel injection in the chamber. It can be observed that the CO_2 emission of PD is lower than all other tested fuels which can be explained by the hydrocarbon chain of PD.

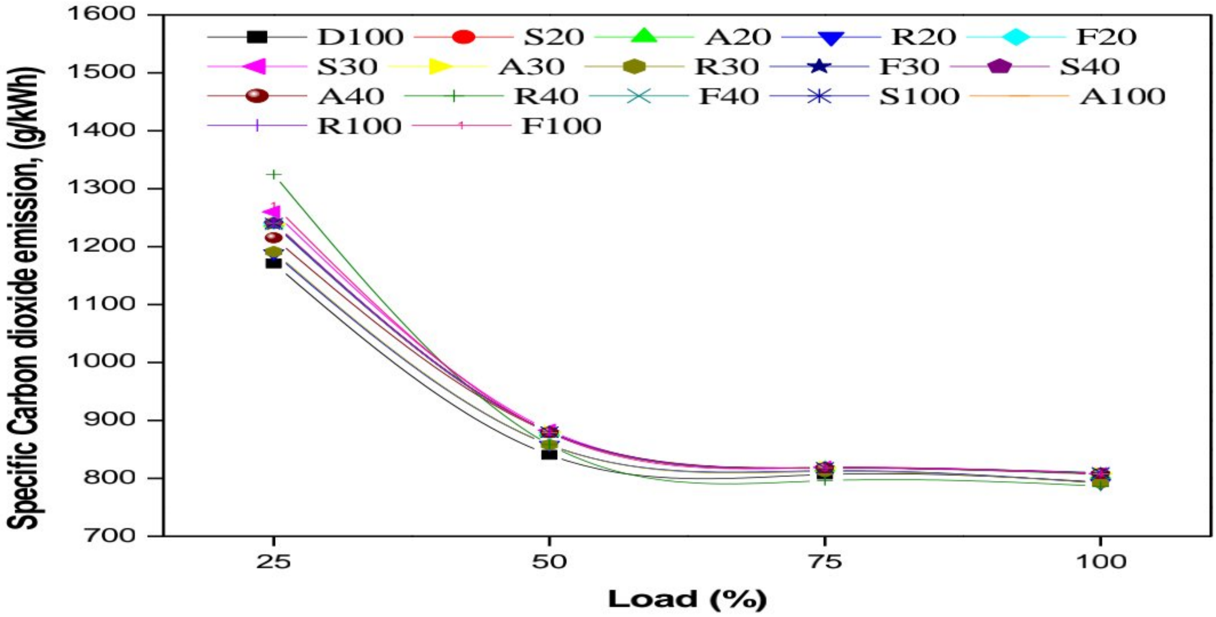


Fig. 5.10. Variation of Specific carbon dioxide emission versus load.

5.3.4. Specific SO2 emission

- The rate of Specific SO_2 emission decreases gradually with increase in engine load.The rate of Specific SO_2 emission of PD is almost 0.

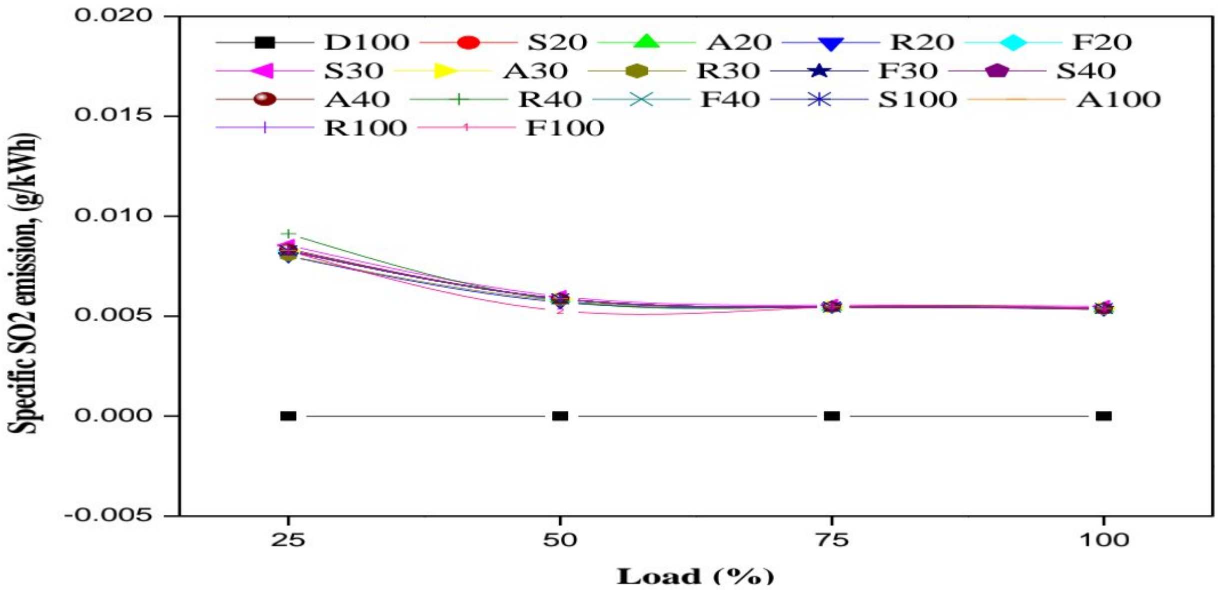


Fig.5.11 Variation of Specific SO2 emission versus load.

5.3.5 Summary of PM and NOx

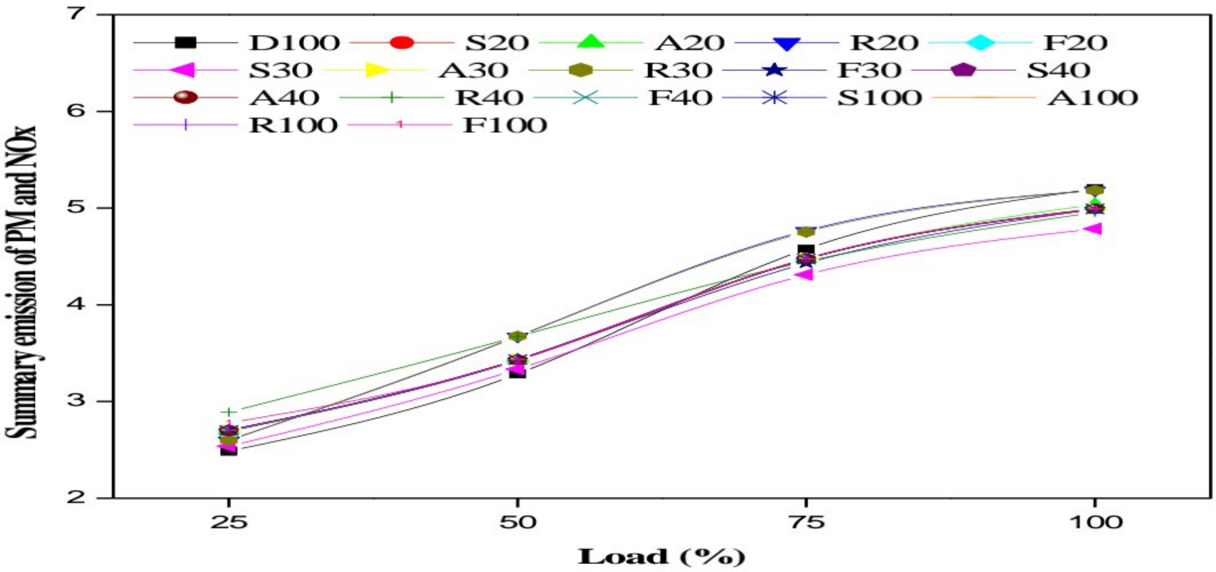


Fig.5.12 Variation of Summary of PM and NOx versus load.

CHAPTER 6
CONCLUSION

- In the combustion characteristics, the biodiesels have shown closeness in terms of cylinder pressure, temperature, cylinder peak pressure with PD. The biodiesels have lesser ignition delay, as compared with PD. The spray tip penetration of the fuels was close to PD, which shows that there is good utilization of inlet air and air-fuel mixing.

- Also in performance characteristics, the specific fuel consumption decrease for all fuels with increase in load of the engine. While the brake thermal efficiency increased for all tested fuels, the volumetric efficiency decreases with increase in load. The values of indicated and mechanical efficiency for all tested biofuels were close to PD.

- The soot and smoke formation was highest in PD and lesser for biodiesels. NO emission increased for all fuels during increment in engine load.

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