



# Methanol Fuel Utilization In Dual Fuel Compression Ignition (Ci) Engine

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**Abstract:** The experimentation was carried out on a setup consists of single cylinder port injection dual fuel, four stroke Common rail direct injection (CRDI), Variable Compression Ratio (VCR) and open Electronic Control Unit (ECU) engine connected to eddy current dynamometer. The diesel fuel is injected directly into the combustion chamber as a pilot fuel with common rail fuel injection system and methanol fuel is injected into the intake manifold by using separate fuel injector as main fuel. The aim of this project is to perform the experimentation for Methanol (CH<sub>3</sub>OH) utilization in Dual fuel compression ignition and investigation of the Combustion characteristics and performance analysis and emission analysis of the engine running at constant 1500 rpm and 100% load. Five methanol replacement ratios (MRR) such as 0% (pure diesel), 15 %, 25%, 35%, and 45% were conducted. From study it is observed that, premixed diesel combustion was retarded as well as improved, diffusion diesel combustion was increased and progressed, ignition delay was longer, combustion duration was lowered, methanol is the most beneficial in terms of extending ignition delay and lowering combustion duration for the specific methanol replacement ratio. Net heat release rate (HRR) and NO<sub>x</sub> increases consistently for increasing MRR at 100% load. The BTE decreases consistently with increasing the MRR. Lower CO and smoke opacity emission was reported with high MRR. HC and CO<sub>2</sub> emissions were higher for high MRR.

**Index Terms** - Compression Ignition Engine, Dual fuel, port injection, Methanol, Pilot fuel.

## 1. Introduction

Now a day increases the demand of fossil fuel and resources of the fossil fuel decreases in the world as well as more hard and fast emission restrictions make look towards for alternative effective fuel due to this bio fuel become a very famous topic for researcher. At current situation in the world 80% fossil fuel used as primary energy source and out of which 58% used only by the transportation sector [7].

Bio fuel is a fuel which is produced through immediate process from biomass rather than fossil fuel formation from very slow geological process. Only biomass and bio fuels are non-conventional energy sources that can directly replace fossil fuels. Bio fuels are the renewable fuels obtained from the biological feedstock such as bio ethanol which is equivalent to gasoline or biodiesel which is equivalent to diesel. Bio fuels produced from the domestic, agricultural, industrial, commercial waste and plants (energy trees) [1]. Bio fuels are grouped in two classes first is primary bio fuels and another is secondary bio fuels. Primary bio fuels which are obtained from wood burning, also from cellulosic plants and animal dry waste. The secondary bio fuels are also divided in three generations; the first, second and third generations are found in various plant as well as animal materials. The first-generation bio fuel which is known as ethanol which is produced from the food contents crops which are plentiful in starch and biodiesel produced by wasted fats of the animal similarly grease cooking. Another second-

generation bio fuels are the bio ethanol which is obtained from cellulosic foodless biomass as well biodiesel obtained from seed of more oil contents plant as like *Jatropha* and soybean. The microbes, microalgae, and cyan bacteria are used to produce third generation bio fuels which are the most challenging work to meet the demands of the all-world energy [2].

Diesel engine is also known as compression ignition engine (CI Engine) has wide application in the automobile, agricultural, industrial and transportation sectors due to its suitable properties such as reliability, low cost, high thermal efficiency, no need of spark plug to start the ignition of the fuel, no require throttle control to supply air to engine, load control is possible by varying fuel injection, high density of fuel gives more energy per gallon, no pumping losses, heterogeneous combustion in diesel engine provide lean air fuel ratio etc. Emission of harmful gases is the major challenge in diesel engine due to this researcher try to find alternate fuel [1].

Diesel fuel in CI Engines is useful due to ignition take place without the spark. Because of the higher efficiency of CI Engines compare to SI Engines makes them desirable to use in transportation even though increased prices. Combustion of diesel fuel in CI Engine emits some harmful gases such as, (HC) hydrocarbons, (PM) particulate matter, (NO<sub>x</sub>) nitrogen oxides, and (CO) carbon monoxide which affect the environmental pollution. Out of these emissions black smoke and (NO<sub>x</sub>) oxides of nitrogen are harmful and disgusting. Worldwide research is going on to reduce the (NO<sub>x</sub>) oxides of nitrogen and smoke exhaust emissions from Diesel (CI) Engine. Reduction in both emissions concurrently is become difficult [2].

### 1.1 Research objectives

The aim of this project is to perform the experimentation for Methanol (CH<sub>3</sub>OH) utilization in Dual fuel port injection Diesel Engine or compression ignition (CI Engine) and investigation of the Combustion characteristics, performance analysis and emission analysis of the engine.

The main objectives of this study are-

- 1) To perform the experimentation for combustion characteristics of the dual fuel engine.
- 2) To carried out experimentation for performance analysis of the dual fuel engine.
- 3) To perform the experimentation for emission analysis of the dual fuel engine.

## 2. Literature Review

[1] Zhanming Chen , Jingjing He, Hao Chen , Limin Geng and Peng Zhang(29 June 2021) studied the comparative analysis of methanol-diesel, ethanol-diesel and diesel n-butanol fuel combustion and emissions experimentation is carried out by using dual fuel common rail engines. They also studied effect of the energy substitution ratio (ESR) on the combustion and performance characteristics of dual fuel engine. They carried out experimentation at constant speed 1600rpm, brake mean effective pressure 0.921 Mpa and medium load. They used five ESR 0% (pure diesel), 10%, 20%, 30% and 40% respectively for experimentation. The results of the experimentation are shows that premixed combustion of the diesel retarded and implemented, diffusion combustion of diesel was advanced and accelerated, ignition delay is extended and combustion duration reduced. Especially methanol is the beneficial than that of the other fuels for extending ignition delay and reducing combustion duration. BTE is continuously increasing from 41.6% to 42.3% and 42.5% respect to the continuous increases in proportion from 0 to 40% for diesel methanol and diesel ethanol. But BTE decreases from 41.6% to 41.1% for diesel and n- butanol. For high value of the ESR NO<sub>x</sub> emissions reduced but increased for diesel n-butanol. Particle concentration also increases with respect to the high proportion of the alcoholic fuel.

[2] Arkadiusz Jamrozika , Wojciech Tutaka , Michał Pyrca , Michał Grucaa , Marek Kočiškob (14 February 2018) studied co- combustion of the diesel fuel with oxygenated alcohols in CI dual fuel Engine. They used single cylinder CI dual fuel Engine for experimentation. They studied the analysis of combustion of ethanol, methanol, 1-butanol and 2-propanol with compare to diesel fuel and their effect on the performance, combustion parameters and emissions of the exhaust gases of the engine. They used percentage of alcohol fuels 15%, 30%, 45%, 55% and 70% energy supplied to the engine out of the total energy supplied. After the experimentation they found results as compared to the pure diesel fuel and the results are positive with stability of the research engine, mean indicated pressure and thermal efficiency. They also found that there is increase in indicated mean effective pressure by 22%, increase of indicated thermal efficiency by 13% and reduction of coefficient of variation 52%. In terms of indicated pressure and high efficiency methanol is found to be more efficient than other alcoholic fuel with the values IMEP=0.86MPa and ITE=35.3% at DM55. Increase in percentage of alcohol above 55% which leads to increase NO<sub>x</sub>, CO and CO<sub>2</sub> emissions from the engine. After results they concluded that the utilization of alcoholic fuel has main disadvantage that

NOX emission increases and to avoid these things author given suggestion that preference must be given to use of precise alcohol fuel injection system, the research engine should also be equipped in a modern common-rail system that allows for control of the injection characteristics, injection pressure and rate of combustion in the engine cylinder.

[1] Yabin Donga , Ossi Kaarioa, Ghulam Hassana , Olli Rantaa , Martti Larmia , Bengt Johanssonb (23 April 2020) studeid high pressure direct fuel (DI) method of injection of methanol and diesel as a pilot fuel , it is the non premixed dual fuel engine method. For experimentation they prepared model of cylinder head, it contains two injectors one is methanol injector which is located centrally and other is diesel pilot injector which located at off- centered position. The main intention is to inject methanol at TDC as like standard diesel engine. Both fuels methanol and diesel injected direct simultaneously. Single-cylinder heavy-duty research engine is used for experimentation at 1500rpm engine speed constantly at 16.5 compression ratio. The value of indicated mean effective pressure changed in between 4.2 to 13.8 bar and methanol substitution ratio varies from 45 to 95%. Finally they conclude that use of this method using methanol as main fuel gives strong ignition characteristics, stable combustion, and low CO and HC emissions. The results shows that increasing both load and methanol substitution ratio can increase the thermal efficiency and the stability of combustion together with decreased CO and HC emissions.

[2] Hongyuan WeiHongyuan Wei, Chunde Yao , Wang Pan, Guopeng Han, Zhancheng Dou, Taoyang Wu, Meijuan Liu, Bin Wang, Jian Gao, Chao Chen, Junjie Shi (9 October 2016) studied the effect of pilot injection of the diesel fuel on the combustion parameters and the emission parameters by using experimentation methods. They used six cylinder engine with turbocharged intercooling and common rail injection system experimental setup for experimentation. They try to optimize combustion process at high methanol substitution ratio and reduce the emissions at low load conditions. The experimental results shows that combustion stability and fuel economy improved by using pilot injection system at high methanol substitution ratio. It can also reduce regulated emissions CO, THC except NOx, and unregulated emissions tested in this study except CO2 on M0 and M10 mode and toluene on M50 mode when compared with single injection cases. Increase pilot injection quantity and advance pilot injection timing both cause an increase in incylinder temperature and pressure before main combustion. The variations of pilot injection quantity and timing have less effect on regulated and unregulated gaseous emissions except NOx on M0 and M10 mode than higher MSR mode. With the rise of pilot quantity and the advance of pilot injection timing, HC and CO emissions decrease gradually but NOx emissions ascend; almost all unregulated emissions tested in this study on M30 mode reduce gradually; but when the MSR continues to rise to 50%, increasing emissions in unburned methanol, 1,3-butadiene and benzene are observed.

### 3. Methods

1. The experimentation conducted on the Single cylinder four stroke diesel CRDI engine.
2. The engine is operated at 50% load, 100% maximum load and constant rotational speed of 1500 rpm.
3. Methanol is injected in the intake manifold at an angle 210<sup>0</sup>CA bTDC.
4. Methanol and air homogeneous mixture inducted in the cylinder.
5. This homogeneous mixture is compressed in the cylinder.
6. Diesel is injected at the end of compression stroke as pilot fuel at 7<sup>0</sup>CA bTDC.
7. Slightly reduce the utilization of diesel percentage and increase the methanol percentage adjusting methanol injection timing 2ms, 3ms, 4ms and 5ms. By using open ECU.
8. Fuel mass calculation, Injection Control (Timing), Fuel Pressure Control, Engine Synchronization (Identification of TDCs), Engine Speed Control these facilities are available in the software.
9. The measurement stand allowed for e.g. recording of the variable in-cylinder pressure in the research engine every 1 deg CA.
10. Measurement of other values such as rotational speed, air consumption, diesel fuel consumption, alcohol fuel consumption, air temperature, exhaust gas temperature and ambient pressure and temperature.

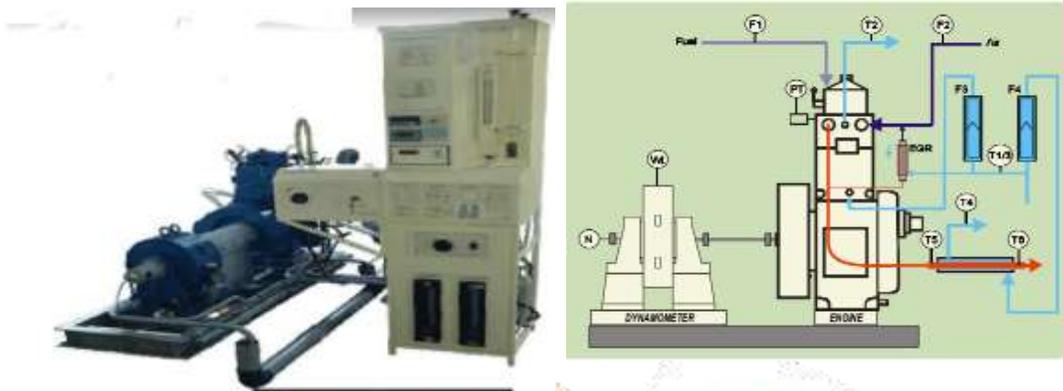
The exhaust gas analyzers recorded concentration of components in engine exhaust gas, such as NOx, HC, CO and CO2 and smoke meter was used to measure the smoke emission from the engine. Methanol replacement ratio (MRR) was calculated by using formula -

$$\text{MRR} = \frac{(\text{mM}) * (\text{LHVM})}{(\text{mD} * \text{LHVD} + \text{mM} * \text{LHVM})} * 100\%$$

Following five cases are considered for experimentation as shown in below table no 5

**Table.3.1 Cases considered for experimentation**

SR. NO.	CASE	Methanol %	Diesel %
1	Case 1	0 %	100 %
2	Case 2	15 %	85 %
3	Case 3	25 %	75 %
4	Case 4	35 %	65 %
5	Case 5	45 %	55 %



**Fig.3.1 Research Engine Setup and Layout Table 3.2 Research engine**

specifications

Sr. No.	Descriptions	Specifications
1	Make	Kirloskar
2	Engine capacity (CC)	661
3	Number of cylinders	1
4	Application	Automotive (Constant-speed)
5	Number of Strokes	4
6	Compression Ratio	CR range 12-18
7	Bore (mm)	87.5
8	Stroke Length (mm)	110
9	Ignition	Compression Ignition
10	Max. Power @ RPM	3.5KW @ 1500 RPM
12	Cooling System	Water Cooled
13	Number of Valves	2
14	Connecting Rod Length (Centre to Centre) (mm)	234
15	Engine Oil Specification, Capacity	SAE 15W40, 3.25 Ltrs

**Engine Details:**

ICEngine set up under test is Research Diesel having power 3.50 kW @ 1500 rpm which is 1 Cylinder, Four stroke , Constant Speed, Water Cooled, Diesel Engine, with Cylinder Bore 87.50(mm), Stroke Length 110.00(mm), Connecting Rod length 234.00(mm), Compression Ratio 18.00, Swept volume 661.45 (cc)

**Combustion Parameters:**

Specific Gas Const (kJ/kgK): 1.00, Air Density (kg/m<sup>3</sup>) : 1.17, Adiabatic Index : 1.41, Polytrophic Index : 1.09, Number Of Cycles : 25, Cylinder Pressure Reference : 6, Smoothing 2, TDC Reference : 0

**Performance Parameters:**

Orifice Diameter (mm): 20.00, Orifice Coeff. Of Discharge: 0.60, Dynamometer Arm Length (mm): 185, Fuel Pipe dia (mm): 12.40, Ambient Temp. (Deg C): 27, Pulses Per revolution: 360, Fuel Type: Diesel, Fuel Density (Kg/m<sup>3</sup>): 830, Calorific Value of Fuel (kj/kg): 42000.

4. Results and Discussion

4.1 Combustion Characteristics of the Dual Fuel Engine.

4.1.1 Net Heat release rate (HRR) analysis.

Heat release rate is the concept gives detail understanding of combustion model. It is defined as the rate at which the chemical energy of the fuel is released by the combustion process. Two phases present in the HRR diagram, first phase which is known as premixed phase and second is diffusion combustion phase. Increase in methanol percentage in dual fuel engine increases the premixed combustion intensity at 100% loading condition. Due to high latent heat of vaporization of methanol fuel prolonging the ignition delay period as well as delayed in start of combustion at high MRR.

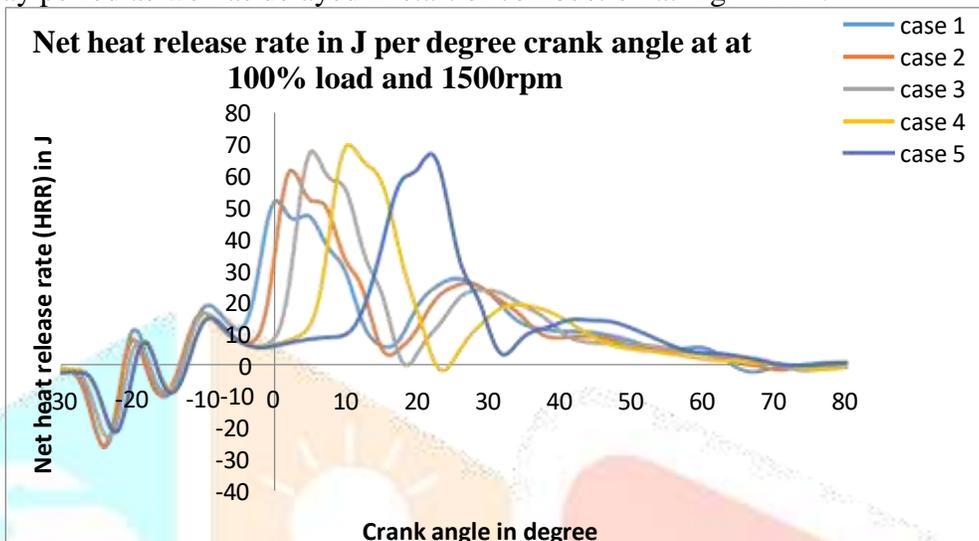


Fig.4.1.1. Net heat release rate in J per degree crank angle at at 100% load and 1500rpm

Table. 4.1.1 Values of Net heat release rate in J per degree crank angle at at 100% load and 1500rpm

	case 1	case 2	case 3	case 4	case 5
NHRMax in J/deg CA	56.3	61.12	67.31	69.18	66.75
Angle	360	362	365	370	382

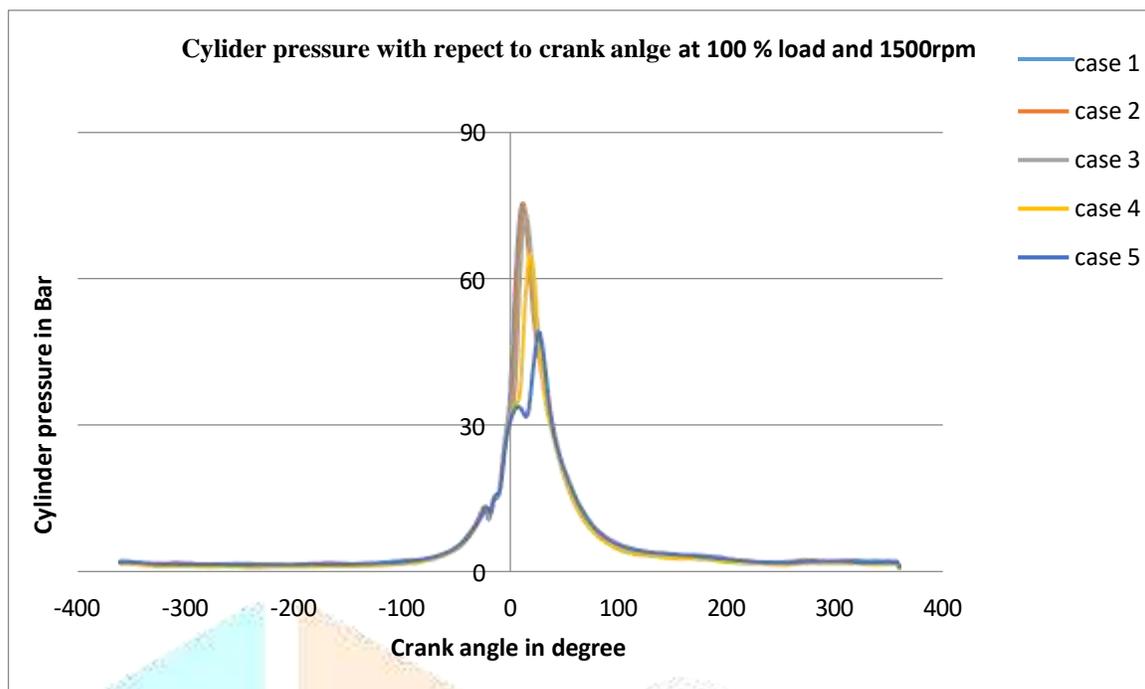
From the engine data logger, in-cylinder pressure and crank angle signals were received for the specified load and recorded on a high-speed computer-based digital data collecting system. A total of 25 cycles of data were recorded. Using the average value of the pressure crank angle data and the first law of thermodynamics, the net heat release rate was computed after collecting data for 25 cycles. Key input parameters for the modeling of NOx emissions are also provided by the HRR diagram. The first law of thermodynamics is used to model the rate of heat release. In equation (1) the simplified model is given.

$$dQ/d\theta = 1/(\gamma-1) * (\gamma P dV/d\theta + V dp/d\theta).....(1)$$

Where dQ/dθ is rate of heat release (kJ/deg), P is the in cylinder gas pressure, V is in-cylinder pressure, and γ is the ratio of specific heats in the equation (1). Cp/ Cv = γ = 1.34 for diesel.

4.1.2 Cylinder pressure (CP)

To acquire quantitative information on the progress of combustion, cylinder pressure versus crank angle data collected throughout the compression and expansion strokes of the engine's running cycle can be employed. As MRR increases cylinder pressure decreases at 100% loading conditions. This is occurs due to high latent heat of vaporization maximum amount of the heat is absorbed by the fuel and cylinder temperature as well as pressure reduces Piezo sensor Make PCB USA, Combustion: Range 350Bar with low noise cable are used to measure in cylinder pressure in this experimental setup. Cylinder pressure maximum peak values mentioned in below table and table for corresponding cases 50 % as well as 100% loading conditions respectively.



**Fig.4.1.2 Cylinder pressure with respect to crank angle at 100 % load and 1500rpm**

**Table 4.1.2 Values of Cylinder pressure with respect to crank angle at 100 % load and 1500rpm**

	case 1	case 2	case 3	case 4	case 5
CPMax in Bar	75.36	74.82	73.36	64.97	49
Angle in degree	372	372	374	379	386
IMEP in Bar	9.5	9.41	9.35	8.93	8.93
IP in Bar	8.01	7.91	7.87	7.53	7.5

**4.1.2 Cumulative Heat Release (CHRR)**

An essential factor to assess the effectiveness of the combustion process is the cumulative heat release. The combustion cylinder's cumulative heat release ( $Q_c$ ) is calculated by using an equation (2).

$$Q_c = \int dQ = \int \frac{P dV + V dP}{\gamma - 1} \text{ ----- (2)}$$

Following figure 12 shows cumulative heat release rate with respect to crank angle at 1500rpm and 100% loading conditions respectively. CHR decreases at the early stage of combustion due to lower heating value of methanol as compare to diesel fuel.

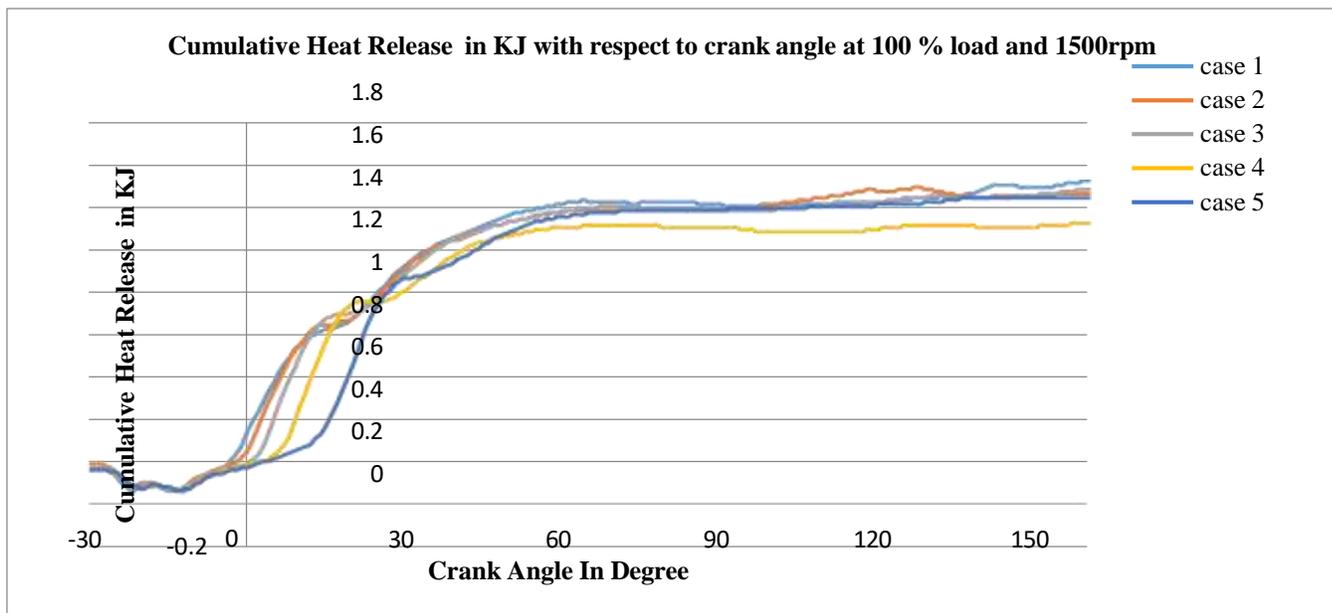


Figure 4.1.3. Cumulative Heat Release in KJ with respect to crank angle at 100 % load and 1500rp

Table 4.1.3 Values of Cumulative Heat Release in KJ with respect to crank angle at 100% load and 1500rpm.

	case 1	case 2	case 3	case 4	case 5
CHRMax in KJ	1.53	1.49	1.48	1.33	1.44
Angle in deg	528	488	520	524	500

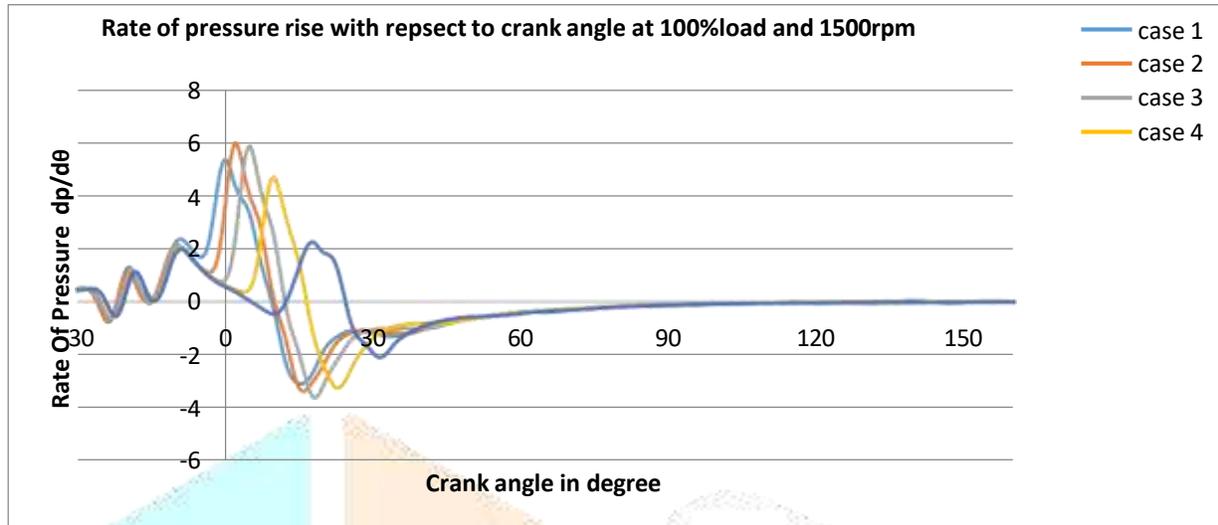
At 100% load CHR decreases at early stage of combustion and increases at later stage of combustion. CHR decreases at the early stage of combustion due to lower heating value of methanol as compare to diesel fuel. CHR Increases due to rise in fuel quantity inside the cylinder due to high load and engine speed. Total CHR values decreases at 100% load due to methanol fuel quantity increases inside the cylinder.

#### 4.1.4 Rate of Pressure (RPR)

A parameter that provides details on the rate of force transfer owing to in-cylinder combustion pressure applied by burning and expanding gases against the engine's mechanical linkages and directly affects structural safety is the rate of pressure rise. The rate of pressure rise reaches its maxima during premixed combustion phase due to rapid combustion and very fast premixed heat release. After attaining the maxima, it reduces in the expansion stroke due to mixing controlled combustion, where the combustion is relatively slower in addition to increase in combustion chamber volume due to movement of piston in expansion stroke. RPR is calculated by using following equation (3).

$$RPR \left( \frac{\text{bar}}{^{\circ}\text{CA}} \right) = \frac{dP}{d\theta} = \frac{(P2 - P1)}{(\theta2 - \theta1)} \dots\dots\dots (3)$$

The traces in Figure no 14 shows the rate of pressure rise diagram for different five cases of methanol diesel ratios at 100% load and 1500rpm engine speed. From graph it is clear that rate of pressure rise increase at premixed combustion phase and decreases at diffusion combustion phase at 100% load. The intensity of RPR increases with increase in MRR as compare to pure diesel traces at 100 % load.



**Fig.4.1.4 Rate of pressure rise with respect to crank angle at 100%load and 1500rpm**

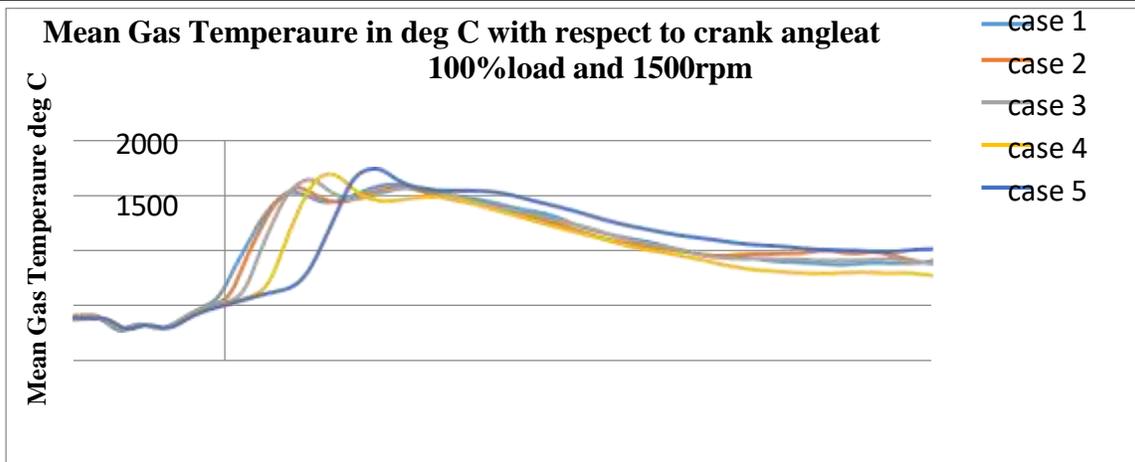
	case 1	case 2	case 3	case 4	case 5
RPRMax in bar/°CA	5.37	5.98	5.88	4.69	2.24
Angle	360	362	365	370	378

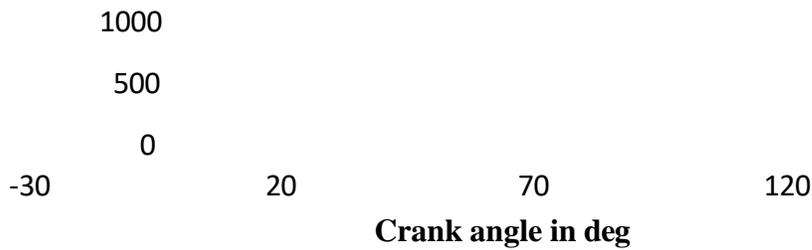
**Table.4.1.4.Values of Rate of pressure rise with respect to crank angle at 100%load and 1500rpm**

The intensity of rate of pressure rise increases upto case 3 and then consistently decreases due to lower heating value of methanol and maximum fuel is consumed for high loading conditions. From these results it is clear that RPR is reduces at high MRR.

**4.1.5 Mean Gas Temperature (MGT)**

Figure 4.1.5.1 shows the comparative study of effect of different MRR on mean gas temperature of the dual fuel engine fueled with methanol and diesel. Mean gas temperature drops at the starting of combustion for higher MRR due high latent heat of vaporization of methanol fuel as compare to diesel. Also at the end of compression stroke mean gas temperature increases for higher MRR. This was happen due to inbuilt oxygen contain in the methanol fuel. Due to high oxygen contain combustion of the fuel accelerate at the end of compression stroke.





**Fig.4.1.5. Mean Gas Temperature in deg C with respect to crank angle at 100% load and 1500rpm**

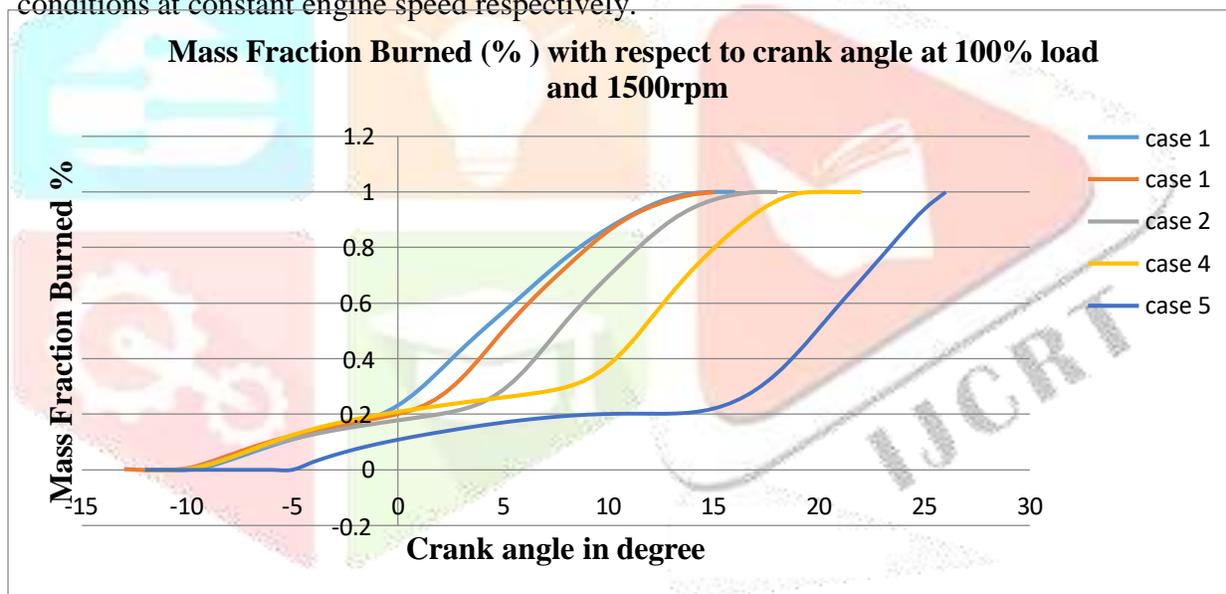
**Table.4.1.5. Values of Mean Gas Temperature in deg C with respect to crank angle at 100% load and 1500rpm**

	case 1	case 2	case 3	case 4	case 5
<b>MGTMax in deg C</b>	1595.93	1566.24	1638.88	1689.23	1737.9
<b>Angle</b>	393	395	377	381	390

As the MRR increased, a more homogeneous fuel-air combination was created, which resulted in a faster burning rate during the diffusion combustion process. It can be noted that mean gas temperature is increases at 100% load.

**4.1.6 Mass Fraction Burned (MFB)**

The mass fraction burned traces are shown in figure 18 at 100% load dual fuel engine loading conditions at constant engine speed respectively.



**Fig.4.1.6. Mass Fraction Burned (%) with respect to crank angle at 100% load and 1500rpm**

**Table.4.1.6. Values of SOC, EOC MFB5% MFB50% MFB90% at 100% load and 1500rpm**

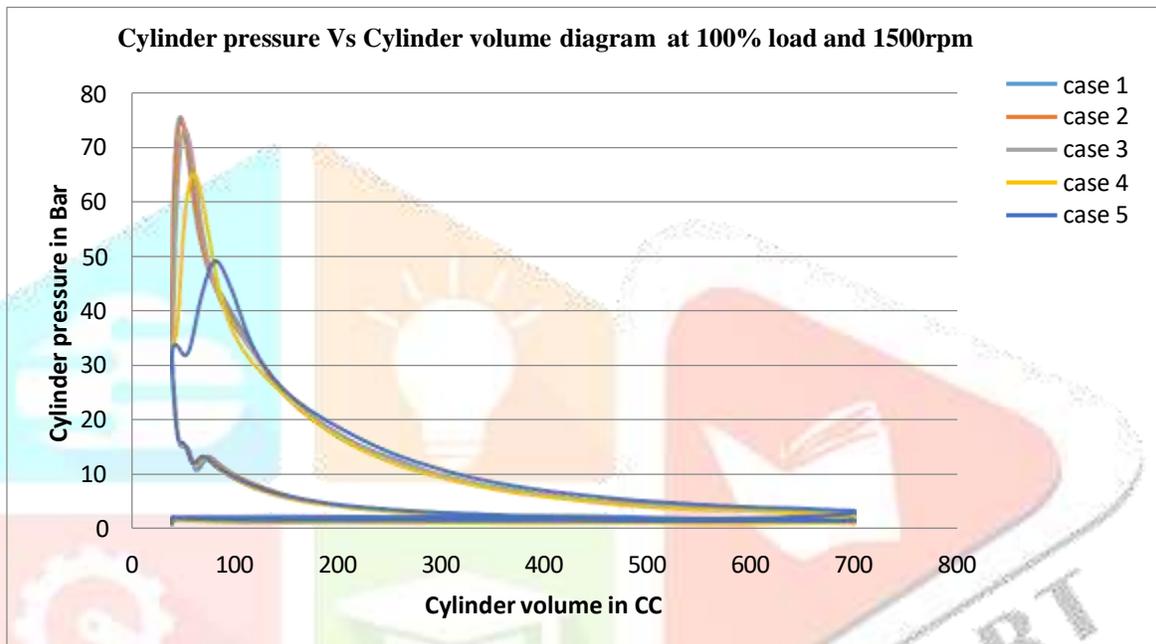
	case 1	case 2	case 3	case 4	case 5
SOC	-13	-14	-13	-13	-5
EOC	16	16	18	22	26
MFB5%	-8.48	-9.19	-8.83	-8.91	-4.08
MFB10%	-6.51	-7.27	-6.7	-7.3	-1.33
MFB50%	2.89	3.83	6.49	10.08	19.06

MFB90%	9.44	9.3	11.72	14.43	23.87
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**4.1.7 Cylinder pressure**

**4.1.7 Cylinder pressure Vs Cylinder volume (P-V Diagram)**

A heat engine is the internal combustion engine. The fluctuation of pressure and volume inside the engine's cylinders is the basis of its operating principle. A pressure-volume diagram, or P-V diagram, which depicts the variation in cylinder pressure as a function of volume over the course of an engine cycle, is a defining feature of all heat engines. Figure.20 shows the p-v diagram for dual fuel engine fueled with methanol/diesel for different MRR. The area of the diagram decreases with respect to increase of methanol percentage. The reduction in the area of the p-v diagram reduces the indicated power of the engine.



**Fig.4.1.7 Cylinder pressure Vs Cylinder volume diagram at 100% load and 1500rpm**

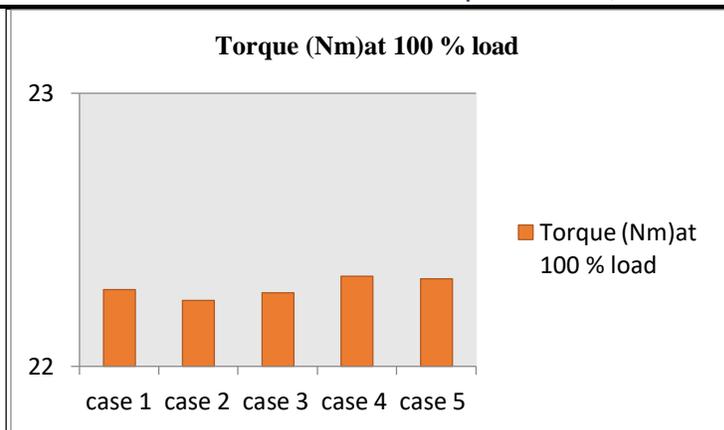
**Table 4.1.7 Values of Cylinder pressure Vs Cylinder volume diagram at 100% load and 1500rpm**

	case 1	case 2	case 3	case 4	case 5
Volume Max in cc	700.36	700.36	700.36	700.36	700.36
Angle	180	180	180	180	180
CPMax in bar	75.36	74.82	73.36	64.97	49
Angle	372	372	374	379	386

**4.2 Performance analysis**

**4.2.1 Engine Torque (T)**

Eddy current dynamometer: It consists of a stator on which are fitted a number of electromagnets and a rotor disc and coupled to the output shaft of the engine. When rotor rotates eddy currents are produced in the stator due to magnetic flux set up by the passage of field current in the electromagnets. These eddy currents oppose the rotor motion, thus loading the engine. These eddy currents are dissipated in producing heat so that this type of dynamometer needs cooling arrangement. A moment arm measures the torque. Regulating the current in electromagnets controls the load. The torque produced at 100% load the



**Table 4.2.1 Values of Torque in Nm at 50% and 100% load**

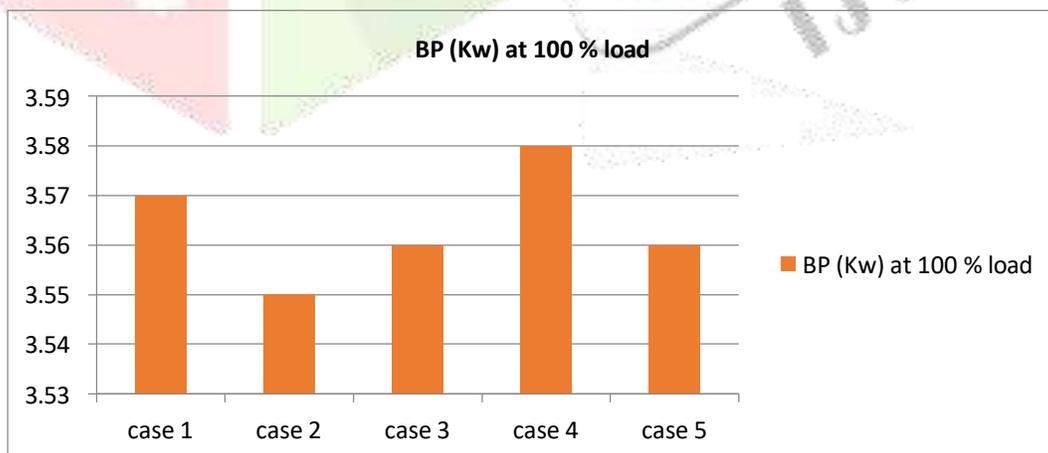
case	Torque (Nm) at 100 % load
case 1	22.28
case 2	22.24
case 3	22.27
case 4	22.33
case 5	22.32

**4.2.2 Brake power (BP)**

The common way of engine rating is brake power which is measured at engine flywheel. This is the actual mechanical power output of the engine. The term “brake” comes from the use of a brake in a dynamometer test to load the engine. Brake power is measured in KW by using below formula.

$$Brake\ power\ in\ (KW) = \frac{2\pi NT}{60 * 1000}$$

Where N- Engine speed and T- Torque measured by using Eddy current dynamometer. The below figure shows the brake power output of the dual fuel port injection diesel engine at 100% loading condition and with constant engine speed 1500 rpm.



**Fig.4.2.2 Brake power in KW at 100% load**

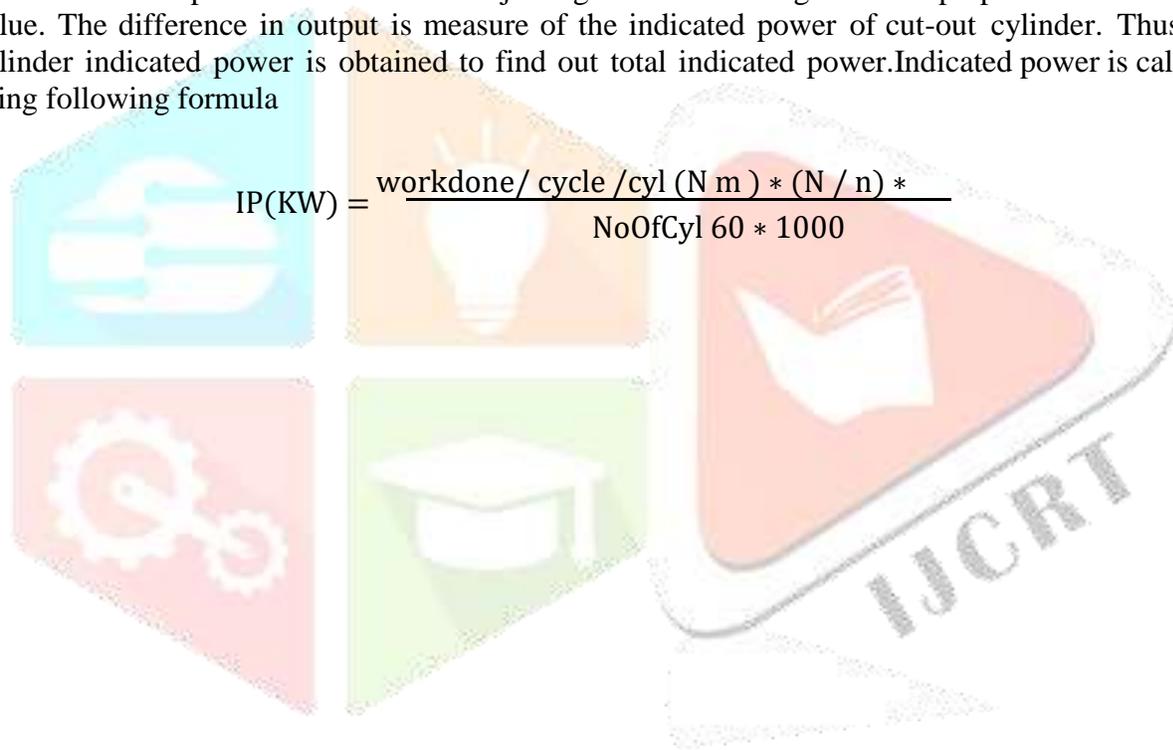
**Table 4.2.2 Values for Brake power in KW at 100% load**

case	BP (Kw) at 100 % load
case 1	3.57
case 2	3.55
case 3	3.56
case 4	3.58
case 5	3.56

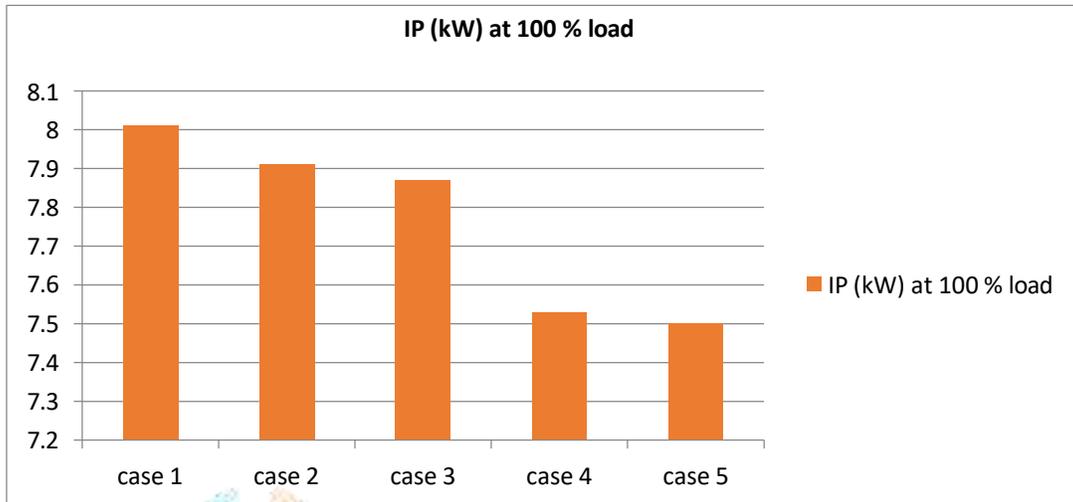
**4.2.3 Indicated Power (IP)**

The indicated power is defined as the total power developed by combustion of fuel in the combustion chamber is called Indicated Power (IP). Indicated power is the total power of engine which is graphically represented by area of cycle on P-V chart. As the actual P-V diagram of an engine is drawn by an indicator mechanism, this power is called indicated power. Morse test: It is applicable to multi-cylinder engines. The engine is run at desired speed and output is noted. Then combustion in one of the cylinders is stopped by cutting off the fuel supply. Under this condition other cylinders “motor” this cylinder. The output is measured after adjusting load on the engine to keep speed constant at original value. The difference in output is measure of the indicated power of cut-out cylinder. Thus for each cylinder indicated power is obtained to find out total indicated power. Indicated power is calculated by using following formula

$$IP(KW) = \frac{\text{workdone/ cycle /cyl (N m ) * (N / n) *}{\text{NoOfCyl 60 * 1000}}$$



Where,  $\text{workdone/ cycle / cyl(Nm)} = \text{Area of PVdiagram} \times \text{Xscalefactor} \times \text{Yscalefactor} \times 100000$ , N-engine speed and n= 2 for four stroke.



**Fig.4.2.- Indicated power in KW at 100% load Table 4.2.3 Values of Indicated power in KW at 100% load**

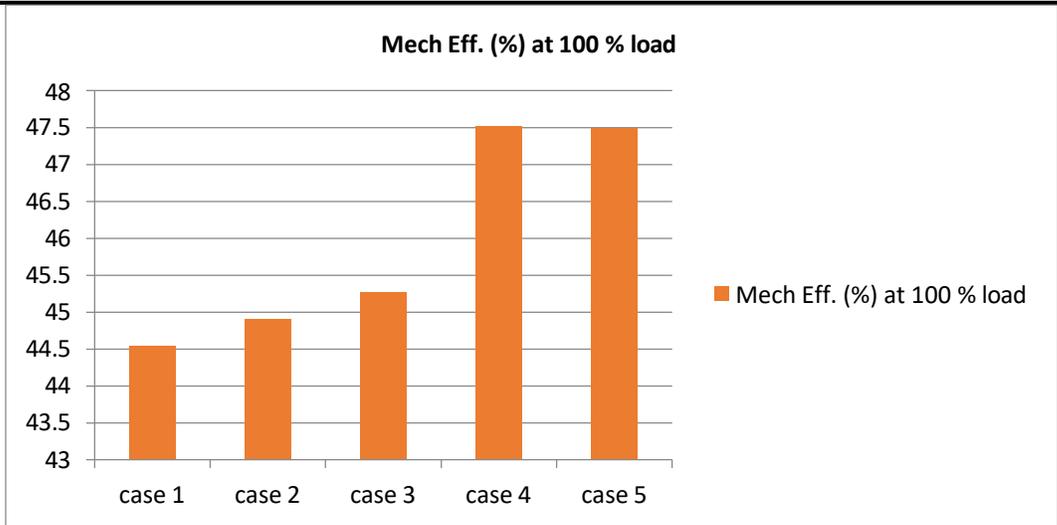
case	IP (kW) at 100 % load
case 1	8.01
case 2	7.91
case 3	7.87
case 4	7.53
case 5	7.5

**4.2.4 Mechanical Efficiency ( $\eta_{\text{mech}}$ )**

It is the ratio of brake power to the indicated power of an IC Engine. The mechanical efficiency is calculated by using formula.

$$\text{Mechanical Efficiency } (\eta_{\text{mech}}) \text{ in } \% = \frac{\text{Brake power}}{\text{Indicated power}} \times 100$$

Figure shows the mechanical efficiency of the dual fuel port injection engine at 100% loading condition with constant engine speed. As the load on the engine increases the mechanical efficiency of the engine increases.



**Fig.4.2.4 Mechanical efficiency at 100% loading conditions**

**Table. 4.2.4 Values of Mechanical efficiency at 100% loading conditions**

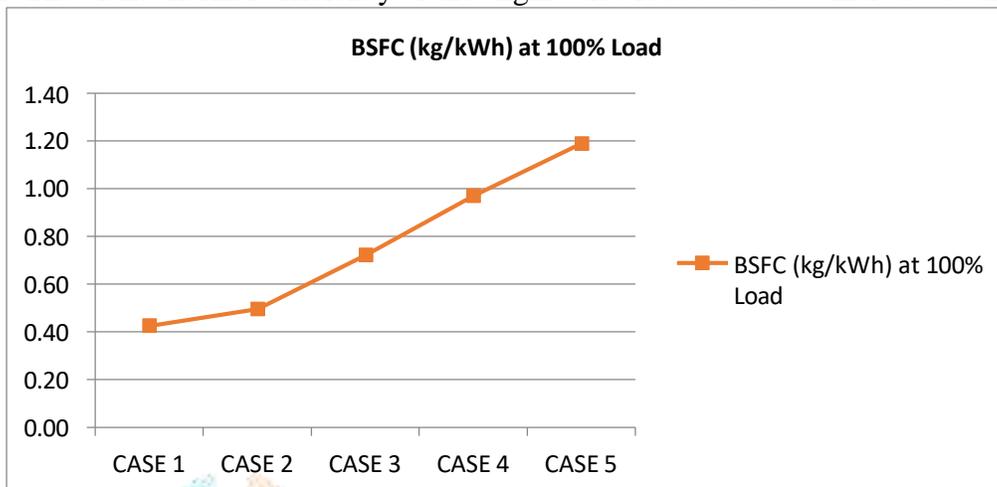
case	Mech Eff. (%) at 100 % load
case 1	44.55
case 2	44.9
case 3	45.27
case 4	47.51
case 5	47.49

**4.2.5 Brake specific fuel consumption (BSFC)**

The brake specific fuel consumption is also known as fuel conversion efficiency of the engine. It is defined as the ration of fuel flow in Kg/hr to the brake power in KW of the engine. It is calculated by using following formula.

$$BSFC \left( \frac{K}{g} \right) = \frac{\text{Fuel flow In kg /hr}}{\text{Brake Power (KW)}}$$

Following figure shows the BSFC increases at both loading conditions with increase in methanol percentage in dual fuel engine. BSFC increases due to lower heating value of the methanol as compare to diesel fuel. The brake thermal efficiency of the engine also reduced due to increased values of BSFC.



**Fig.4.2.5 Brake specific fuel consumption (  $\frac{\text{Kg}}{\text{KWHr}}$  ) at 100% load**

**Table. 4.2.5 Values of Brake specific fuel consumption (  $\frac{\text{Kg}}{\text{KWHr}}$  ) at 100% load**

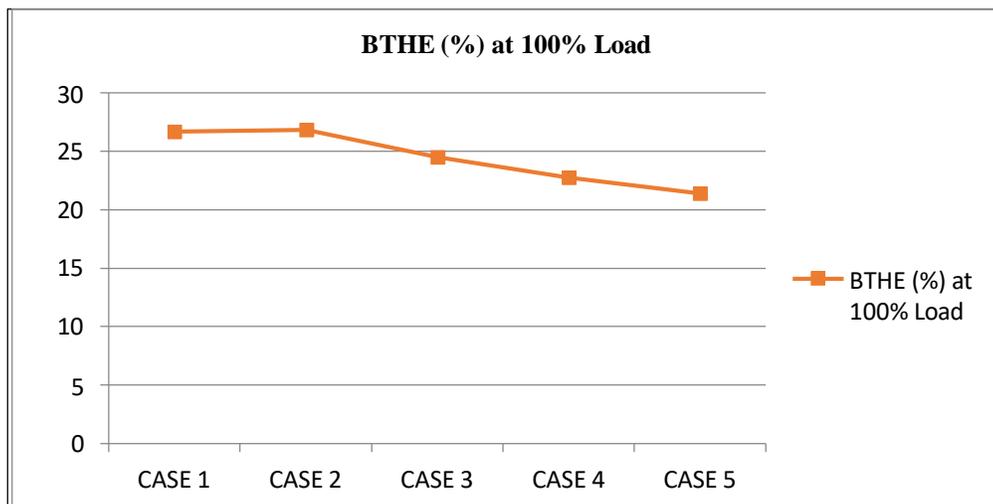
Case	BSFC (kg/kWh) at 100% Load
Case 1	0.43
Case 2	0.50
Case 3	0.72
Case 4	0.97
Case 5	1.19

**4.2.6 Brake thermal efficiency( $\eta_{bth}$ )**

A measure of overall efficiency of the engine is given by the brake thermal efficiency. Brake thermal efficiency is the ratio of energy in the brake power to the fuel energy.

$$\eta_{bth}(\%) = \frac{\text{BrakePower}}{\text{FuelEnergy}} = \frac{\text{BrakePower(KW)} * 3600}{\text{FuelFlow (Hr)} * \text{CalorificValue(Kg)}}$$

$$BTE = \frac{3.6 \times 10^3 * P_e}{m_{DLHVD} + m_{MLHVM}}$$



**Fig.4.2.6 Brake thermal efficiency in (%) at 100% load**  
**Table.4.2.6 Values of Brake thermal efficiency in (%) at 100% load**

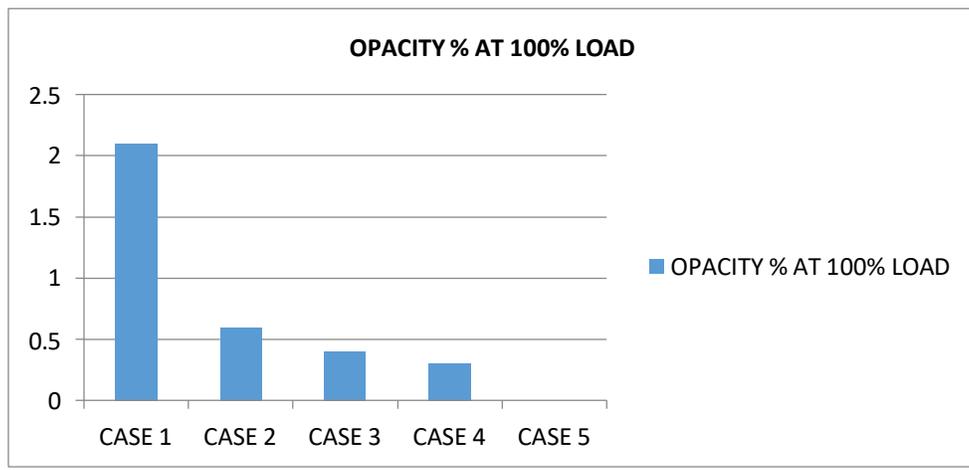
Case	BTHE (%) at 100% load
CASE 1	27
CASE 2	27
CASE 3	25
CASE 4	23
CASE 5	21

### 4.3 Exhaust gas Emissions

The preparation of the mixture during the ignition delay, the quality of the fuel used for ignition, the amount of time spent at various combustion temperatures, the duration of expansion, and general engine design elements all have a significant impact in the generation of emissions. The combustion in diesel engine is heterogeneous in nature due to this temperature increases at the particular point and which leads to formation of harmful gases. The origin of HC and NO in direct-injected diesel engines both the premixed and diffusion (mixing regulated) combustion stages produced different species. Additionally, this nitrogen can interact with oxygen to form nitrogen oxides (NO<sub>x</sub>), which are harmful pollutants.

#### 4.3.1 Smoke

It is well known that soot is the primary element contributing to the generation of smoke, which is a significant source of particulate matter. Smoke is produced when there is a severe lack of oxygen in an area that is fuel-rich at high temperatures and pressures. An increase in the air-fuel ratio reduces the lack of oxygen in the combustion chamber. Long-chain molecules in the air shortage environment undergo thermal breaking, which is the primary mechanism by which smoke is formed. A significant portion of the charge burns in the premixed combustion phase during the methanol-diesel dual-fuel operation, which decreases the smoke emission greatly.



**Fig.4.3.1 Smoke opacity in (%) at 100 load.**

**Table.4.3.1 Values of Smoke opacity in (%) at 100 load.**

Case	Opacity % at 100% load
Case 1	2.1
Case 2	0.6
Case 3	0.4
Case 4	0.3
Case 5	0

**5. Conclusions**

From all over studies it is clear that lower alcoholic fuel methanol is usable for CI Engines. This fuel definitely reduces the future demand of the fossil fuels. Also reduces some harmful emissions will help to maintain the environment clean. Utilization of this fuel in some fraction with diesel fuel also promotes to develop agricultural sectors. Now a day’s utilization of this fuel becomes a need. Most of the alcoholic fuels are manufactured by using renewable energy sources so it will easily available with low cost as compare to fossil fuel.

1. Efficiency of the engine at high load will improve but there is opposite remark at low load condition of the engine.
2. From all over study it is clear that (BSFC) brake specific fuel consumption increases and for most of the study (BTE) Brake Thermal Efficiency decreases.
3. Methanol is the oxygenated fuel which results in reduction of smoke emissions from the engine exhausts.
4. With addition of Methanol to diesel in Dual fuel CI engine Exhaust gas emission (EGT) CO emission decreases as MRR increases.
5. From study it is conclude that cylinder pressure decreases as MRR increases for both loading conditions.
6. NOx emission decreases as MRR increases at medium loading condition and increases at high loading conditions.
7. Heat releases rate (HRR) increases at high load and decreases at medium loading conditions as MRR goes on increasing. Also ignition delay was longer and combustion duration was shorter for higher MRR.
8. CO2, HC, emissions low at medium load and increases at high load.
9. Mean gas temperature increases with increases methanol percentage in the dual fuel engine.

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