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# An Experimental Investigation on NOx Emission Reduction and Performance Evaluation of CI Engine using EGR System

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**ABSTRACT**

Internal combustion engines are established as the main power source for the automobile vehicles. At present emission norms became strict for any IC engine. The main pollutants are HC, CO, NOx, Particulate Matter, Soot, etc. out of which NOx is of the most harmful component. It is possible to limit the negative effect of the NOx on the environment by using exhaust gas recirculation (EGR).

The earlier research work on the EGR system by several researchers revealed that NOx emission from tail pipe of homogeneous charged engine is reduced. Because, EGR system lowers the oxygen concentration inside the combustion chamber. It is also found that EGR system increases intake charge temperature, decreases peak cylinder temperature and decreases the air fuel ratio.

The aim of the proposed project work is to investigate experimentally the effect of rate of EGR on $NO_X$ emission reduction and performance at different loads. A single cylinder, direct injection, compression ignition engine has been selected for investigation, arrangement for EGR system and loading have been made by the project investigation. The research work has been carried out at IC Engines Laboratory, Mechanical Engineering Department, JNTUA College of engineering (Autonomous), Anantapuramu. The experiments obtained have been presented in the project and an analysis of it has been carried out. The results obtained have been presented in the project work.

**Keywords:** SingleCylinder I/C Engine,Direct Injection ,Compression Ignition,Exhaust Gas Recirculation(EGR)

## 1. Introduction:

An engine is a device, which transforms one form of energy into another form. While transforming energy from one form to another, the efficiency of conversion plays an important role. Normally, most of the engines convert thermal energy into mechanical work and therefore they are called 'heat engines'. Heat engine is a device that transforms the chemical energy of a fuel into thermal energy and utilizes this thermal energy to perform useful work. Thus, thermal energy is converted to mechanical energy in a heat engine.

Fuel economy of engines is greatly improved from the past and probably continued to be improved, increased in number of automobiles alone dictate that there will be a great demand for fuel in the near future. Alternative fuel technology, availability, and use must and will become more common in thecomingdecades. However, the use of an alternative fuel decreases the break thermal efficiency of engines, due to lower calorific values compared to normal diesel fuel. Because of the high cost of the petroleum products, some developing countries are trying to use alternative fuels for their vehicles.

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### 1.1Classification of heat engines

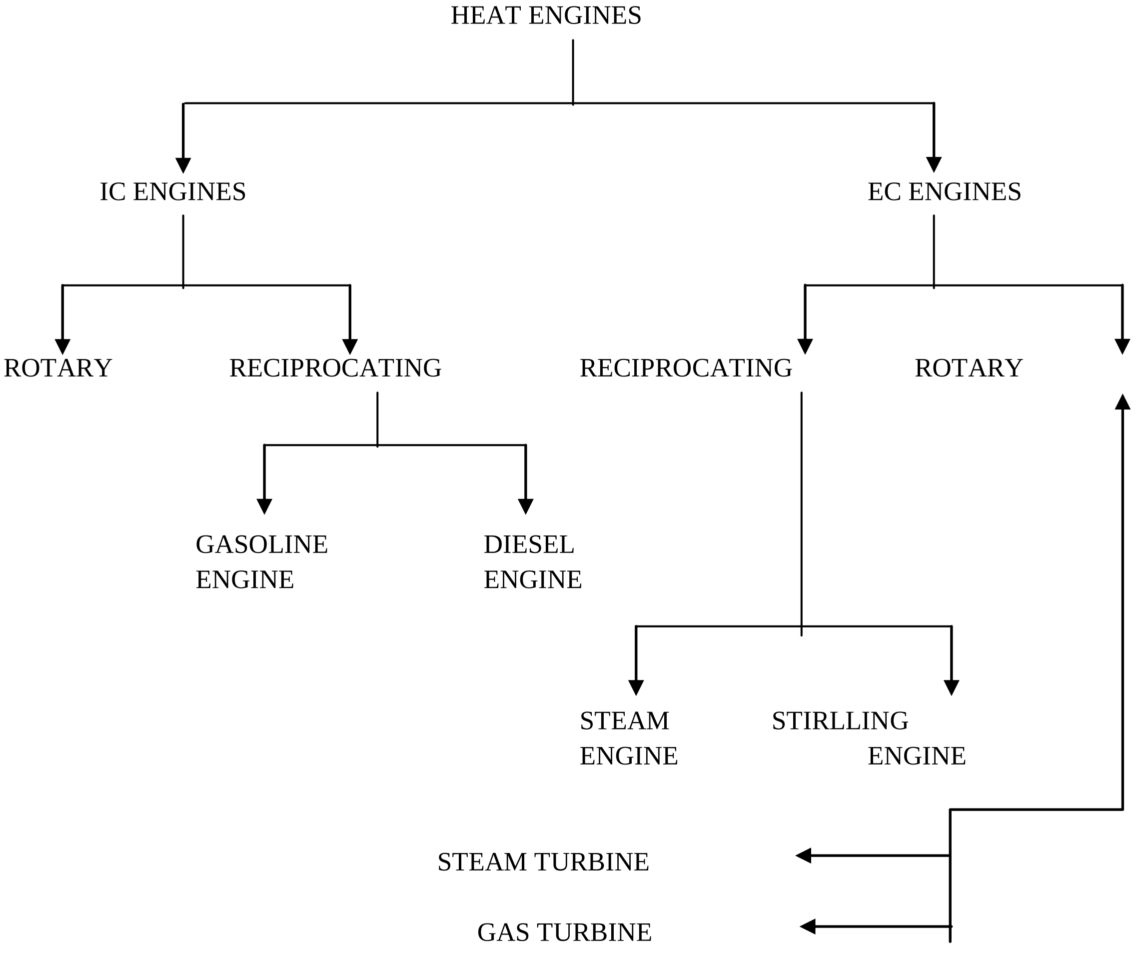



**Fig 1.0 classifications of heat engines**

**Basic Engine Terminology:**A number of basic terms are used to describe and compare the I/C engines.
Top dead Centre (T.D.C)/Bottom Dead Center (B.D.C)/Clerance Volume(VC) .

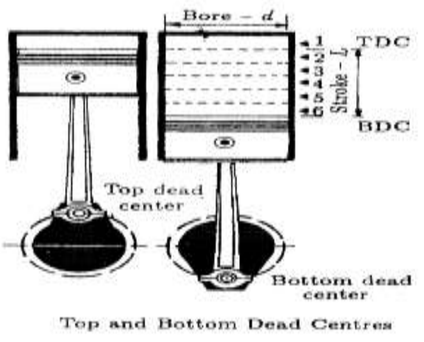



**Figure 1.1 Top and Dead Centre**

Piston displacement/ Engine Capacity /Compression rati0 / Mean effective pressure / Power(P) ./ Indicated power (I.P). Brake Power (B.P) Engine Torque(T)
Working of Four Stroke Compression Ignition Engine:
Suction stroke,Compression stroke,Expansion stroke,Exhaust stroke.

**Performance Parameters:**
Indicated Thermal efficiency . Brake thermal efficiency, Mechanical efficiency,Volumetric efficiency, Mean effective pressure, Air fuel ratio, Calorific value.

**Thermo physical properties:**

Density,Viscosity, Flash point,Fire point.

**Exhaust Gas Recirculation System**

Diesel engines are predominantly used to drive tractors, heavy Lorries and trucks. Owing to their low fuel consumption, they have become increasingly attractive for smaller Lorries and passenger cars also. But higher NOx emissions from diesel engine remain a major problem in the pollution aspect.

For reducing vehicular emissions, baseline technologies are being used which include direct injection, turbo-charging, air-to-air inter-cooling, combustion optimization with and without swirl support, multi-valve cylinder head, advanced high pressure injection system i.e. split injection or rate shaping, electronic management system, lube oil consumption control etc. However, technologies like exhaust gas recirculation (EGR), soot traps and exhaust gas after-treatment are essential to cater to the challenges posed by increasingly stringent environmental emission legislations.

**Mechanism of $NO_X$ Formation:**

A major hurdle in understanding the mechanism of formation and controlling its emission is that combustion is highly heterogeneous and transient in diesel engines. While NO and $NO_2$ are lumped together as NOx, there are some distinctive differences between these two pollutants. NO is a colorless and odourless gas, while $NO_2$ is a reddish- brown gas with pungent odour. Both gases are considered toxic, but NO2 has a level of toxicity 5 times greater than that of NO. Although $NO_2$ is largely formed from oxidation of NO, attention has been given on how NO can be controlled before and after combustion (Levendis *et al* 1994).

NO is formed during the post flame combustion process in a high temperature region. The most widely accepted mechanism was suggested by Zeldovich (Heywood 1988). The principal source of NO formation is the oxidation of the nitrogen present in atmospheric air. The nitric oxide formation chain reactions are initiated by atomic oxygen, which forms from the dissociation of oxygen molecules at the high temperatures reached during the combustion process. The principal reactions governing the formation of NO from molecular nitrogen are,

$$N2 + O \rightarrow NO + N,$$
$$N + O2 \rightarrow NO + O,$$
$$N + OH \rightarrow NO + H.$$
$$NO + HO_2 \rightarrow NO_2 + OH.$$

Subsequently, conversion of this $NO_2$ to NO occurs via

$$NO_2 + O \rightarrow NO + O_2,$$

Unless the $NO_2$ formed in the flame is quenched by mixing with cooler fluid. This explanation is consistent with the highest $NO_2$ /NO ratio occurring at high load in diesels, when cooler regions which could quench the conversion back to NO are widespread (Heywood 1988). The local atomic oxygen concentration depends on molecular oxygen concentration as well as local temperatures. Formation of NOx is almost absent at temperatures below 2000 K. Hence any technique, that can keep the instantaneous local temperature in the combustion chamber below 2000 K, will be able to reduce NOx formation.

**Composition of Air:**The air around us is a mixture of gases, mainly nitrogen and oxygen, but containing much smaller amounts of water vapour, argon, and carbon dioxide, and very small amounts of other gases. Air also contains suspended dust, spores, and bacteria.

**EGR Technique for $NO_X$ Reduction**:EGR is a useful technique for reducing NOx formation in the combustion chamber. Exhaust consists of $CO_2$, $N_2$ and water vapours mainly. When a part of this exhaust gas is re-circulated to the cylinder, it acts as diluent to the combusting mixture. This also reduces the $O_2$ concentration in the combustion chamber. The specific heat of the EGR is much higher than fresh air; hence EGR increases the heat capacity (specific heat) of the intake charge, thus decreasing the temperature rise for the same heat release in the combustion chamber,

$$\%\mathrm{EGR} = \frac{\text{Volume of EGR}}{\text{total intake charge to the cylinder}} \times 100$$

1. **The percentage of EGR for 2LPM:**

Volume of EGR = 2lt/min

= $3.33\times10^{-4}$ $m^3$/s

$$Total\ intake\ charge\ into\ the\ cylinder\ = \frac{\text{mass of fuel}}{\text{density of air}} \times air\ fuel\ ratio, m^3/sec$$

$$mass\ of\ fuel\ cosumpption\ = \frac{\text{heat supplied}}{\text{calorific value of fuel}} kg/sec$$

$$mass\ of\ fuel\ cosumpption\ = \frac{8.715}{43000} kg/sec$$

$= 2.026\times10^{-4}$kg/sec

Total intake charge into the cylinder $= \frac{2.026\times10^{-4}}{1.164}\times15, m^3$/sec

$=2.088\times10^{-3}$, m³/sec

$$\%\ EGR = \frac{3.33\times10^{-4}}{2.088\times10^{-3}} \times 100$$

=15.92%

The percentage of EGR for 2LPM is 16%

2.Subquently % of EGR for 4LPM is 32%

3. Subquently % of EGR for 6LPM is 48%

Another way to define the EGR ratio is by the use of $CO_2$ concentration (Baert *et al* 1999),

$$\text{EGR Ratio} = \frac{[CO2]intake - [CO2]ambient}{[CO2]exhaust - [CO2]ambient}$$

Three popular explanations for the effect of EGR on NOx reduction are increased ignition delay, increased heat capacity and dilution of the intake charge with inert gases. The ignition delay hypothesis asserts that because EGR causes an increase in ignition delay, it has the same effect as retarding the injection timing. The heat capacity hypothesis states that the addition of the inert exhaust gas into the intake increases the heat capacity(specificheat) of the non- reacting matter present during the combustion. The increased heat capacity has the effect of lowering the peak combustion temperature. According to the dilution theory, the effect of EGR on NOx is caused by increasing amounts of inert gases in the mixture, which reduces the adiabatic flame temperature (Pierpont *et al* 1995).

At high loads, it is difficult to employ EGR due to deterioration in diffusion combustion and this may result in an excessive increase in smoke and particulate emissions. At low loads, unburnt hydrocarbons contained in the EGR would possibly re-burn in the mixture, leading to lower unburnt fuel in the exhaust and thus improved brake thermal efficiency. Apart from this, hot EGR would raise the intake charge temperature, thereby influencing combustion and exhaust emissions.

With the use of EGR, there is a trade-off between reduction in NOx and increase in soot, CO and unburnt hydrocarbons. A large number of studies have been conducted to investigate this. It is indicated that for more than 50% EGR, particulate emissions increase significantly, and therefore use of a particulate trap is recommended. The change in oxygen concentration causes change in the structure of the flame and hence changes the duration of combustion. It is suggested that flame temperature reduction is the most important factor influencing NO formation. Implementation of EGR in diesel engines has problems like (a) increased soot emission, (b) introduction of particulate matter into the engine cylinders. When the engine components come into contact with high velocity soot particulates, particulate abrasion may occur. Sulphuric acid and condensed water in EGR also cause corrosion. Some studies have detected damage on the cylinder walls due to the reduction in the oil's lubrication capacity, which is hampered due to the mixing of soot carried with the particulate laden recirculated exhaust gas. This necessitates the use of an efficient particular trap (Mehta *et al* 1994).Studies have shown that EGR coupled with a high collection-efficiency particulate trap, controls smoke, unburnt hydrocarbon and NOx emissions simultaneously. The particulate trap, however, needs to be regenerated since its pores get clogged by the trapped soot particles. Clogged soot traps increase backpressure to the engine exhaust, thus affecting engine performance also. These traps need to be regenerated from time to time using thermal or electrostatic regeneration techniques. Other methods of reducing the particulate emission from diesel engines include multiple injections, supercharging and higher fuel injection pressure etc. The highest attention is currently being paid to two self-regenerating systems: fuel additive-supported regeneration by using cerium or iron-based additives, and a continuous regeneration trap (CRT) using sulphur-free diesel fuel. During the last 20 years, plenty of research work has been done on EGR and its effects on the engine performance in terms of fuel efficiency, volumetric efficiency, power generated etc.

**Classification of EGR Systems**:Various EGR systems have been classified on the basis of EGR temperature, configuration and pressure.

**Classification based on temperature**

(i) Hot EGR: Exhaust gas is recirculated without being cooled, resulting in increased intake charge temperature.

(ii) Fully cooled EGR: Exhaust gas is fully cooled before mixing with fresh in take air using a water-cooled heat exchanger. In this case, the moisture present in the exhaust gas may condense and the resulting water droplets may cause undesirable effects inside the engine cylinder.

(iii) Partly cooled EGR: To avoid water condensation, the temperature of the exhaust gas is kept just above its dew point temperature.

**Classification based on configuration**:(i) Long route system (LR): In an LR system the pressure drop across the air intake and the stagnation pressure in the exhaust gas stream make the EGR possible. The exhaust gas velocity creates a small stagnation pressure, which in combination with low pressure after the intake air, gives rise to a pressure difference to accomplish EGR across the entire torque/speed envelop of the engine.

(ii) Short route system (SR): These systems differed mainly in the method used to set up a positive pressure difference across the EGR circuit.

Another way of controlling the EGR-rate is to use variable nozzle turbine (VNT). Most of the VNT systems have single entrance, which reduce the efficiency of the system by exhaust pulse separation. Cooled EGR should be supplied effectively. Lundquist and others used a variable venturi, in which EGR-injector was allowed to move axially, thus varying the critical area was used (Lundquist et al 2000).

**Classification based on pressure:**Two different routes for EGR, namely low-pressure and high-pressure route systems may be used (Kohketsu et al 1997).

(i) Low pressure route system: The passage for EGR is provided from downstream of the turbine to the upstream side of the compressor.It is found that by using the low pressure route method, EGR is possible up to a high load region, with significant reduction in NOx.However, some problems occur, which influence durability,prohibitionary high compressor outlet temperature and intercooler clogging.

(ii) High pressure route system: The EGR is passed from upstream of the turbine to down- stream of the compressor.In the high pressure route EGR method, although EGR is possible in the high load regions, the excess air ratio decreases and fuel consumption increases remarkably.

Exhaust gas recirculation is an effective means to reduce $NO_x$-emissions of Diesel engines. Unfortunately too high EGR rates increase the emissions of hydrocarbons, carbon monoxide and soot. So EGR has to be controlled precisely. The paper describes an EGR-system for Diesel engines which senses the air-flow and the fuel-flow of the engine and limits the EGR-rate so that the air-fuel-ratio is lean enough for a clean combustion. Furthermore the paper shows the emission results which have been obtained with the described EGR-system on different Diesel passenger cars.

Exhaust Gas Recirculation (EGR) is a method to control Nitrogen Oxide (NOx) emissions from automobile exhaust. In this method, small amount of the exhaust gas is recirculated into the combustion chamber through the air intake system. The exhaust gas is mixed with charge air just before entering intake manifold. Appropriate mixing of exhaust gas with charge air is necessary to ensure nearly equal amount of exhaust gas flow into all the cylinders. Present research efforts by various automobile manufacturers rely on commercial CFD simulation tools to identify and resolve flow and thermal issues occurring in various vehicle systems. The focus of this paper is to optimize the EGR system to obtain uniform distribution of recirculated exhaust gas to every cylinder of a six cylinder heavy duty diesel engine. CFD simulations were done for this engine using a commercial CFD solver. Mixing and transport of the two fluids was modeled using species transport approach. Appropriate geometric modifications were proposed based on simulation results.

"**Effect of Exhaust Gas Recirculation on Exhaust Emissions from Diesel Engines Fuelled with Biodiesel**," by Kawano, D., Ishii, H., Goto, Y., Noda, A. et al., SAE Technical Paper 2007-24-0128, 2007, doi:10.4271/2007-24-0128.

Biodiesel was applied to conventional diesel engines without modification for biodiesel, NOx emission was increased by the change in fuel characteristics. It is necessary to introduce some strategies into diesel engines fuelled with biodiesel for lower NOx emission than conventional diesel fuel case. The purpose of this study is to reveal that exhaust gas recirculation (EGR) is one of the solutions for the reduction of NOx emission and meeting the future emission regulations when using biodiesel.Neat Rapeseed oil methyl ester (RME) as a biodiesel (B100) was applied to diesel engines equipped with high-pressure-loop (HPL) EGR system and low-pressure-loop (LPL) EGR system. Cooled HPL EGR was increased during steady-state operations and JE05 transient mode tests. An increase in HPL EGR rate drastically reduced NOx emission,and did not increase PM emission, because soot formation was suppressed by theoxygen in RME. It could be confirmed that an increase in EGR rate made it possible to achieve low emission meeting 2009 regulation in Japan.

Engine-out emissions measurements were made to develop a comparison between HPL and LPL EGR in steady-state operation test. LPL EGR improved CO and NOx emissions, compared with the case of HPL EGR. An increase in the rate of LPL EGR caused simultaneous reduction of NOx and smoke emissions.

This study investigated the effect of exhaust gas recirculation (EGR) on combustion and odorous emissions in a direct injection (DI) diesel engine. EGR up to 60% was examined in the engine start-up conditions as well as once the engine was stabilized. In emissions, exhaust odor, irritation, formaldehyde (HCHO) and total hydrocarbon (THC) are compared without or with different EGR rates. After engine warm-up, reduced exhaust odor and eye irritation are obtained at 30% and 60% EGR rates than no EGR in the outdoor assessment environment, although HCHO emission at 60% EGR rate is 50% higher than at 30% EGR rate or non EGR. Early in the engine start-up time, 60% EGR rate shows very high emissions of HCHO and THC. Combustion analysis is performed by taking cylinder pressures during combustion and analyzing heat release rates. Cylinder pressures at 1, 2, 4, 8, 16 and 32 minutes after engine start for different EGR rates are taken. There is no change in ignition delays between non EGR and 30% EGR rate. The ignition delay with 60% EGR rate early in the engine start-up time is 1-1.5°CA longer than non EGR or 30% EGR rate and main combustion takes place after top dead center (TDC). 60% EGR rate also showed lower combustion temperature. Longer ignition delay with lower combustion temperature and occurrence of main combustion after TDC are thought to be responsible for higher HCHO and THC emissions at 60% EGR rate.

"**Buick's 1972 Exhaust Gas Recirculation System**," by Thompson, A., SAE Technical Paper 720519, 1972, doi: 10.4271/720519.In 1972 models for sale in California, Buick first employed programmed-metered exhaust gas dilution of the engine intake charge as the major means of reducing oxides of nitrogen ($NO_x$) emission levels. With more stringent $NO_x$ emission standards applicable nationally to 1973 light duty vehicles, it is probable that similar systems will be more widely used. The major considerations in the decision to use this means of reducing $NO_x$ emissions, the design details of the total system, and the field experience to date is discussed. The author concludes with some thought on the limit of usefulness of exhaust gas recirculation for $NO_x$ reduction, and further refinements needed to approach that limit.

**Pollution Due To Emissions:**

Global warming/Acid rain /Smog/Odors

**Engine Emissions:**Engine emissions can be classified into two categories.

Exhaust emissions and Non-exhaust emissions.

**Exhaust Emissions**

Unburnt hydrocarbons,(HC)/Oxides of carbon,(CO and $CO_2$)/Oxides of nitrogen,(NO and $NO_2$)Oxides of sulphur/,($SO_2$ and $SO_3$)/Particulates/Soot and smoke

**Diesel Engine Emissions:**They contain the products of combustion including:

Carbon (soot)/Nitrogen /Water /Carbon monoxide (CO) /Carbon dioxide ($CO_2$)/Nitrogen oxides (NOx) Hydrocarbons (HC)/Particulate matter

**Nitrogen Oxides (NOx)**

Nitrogen is the main constituent of the air that we breathe. When it is exposed to high pressures and temperatures it combines with oxygen in the air to form nitrous oxides. The nitrous oxides then combine with low level ozone to form smog. Because of the way a diesel engine works, with an excess of air inside the engine (rather than “just enough” as in a petrol engine, which is what causes CO emissions), nitrous oxides are more likely to be formed. However tests of actual cars reveal that whilst emissions of Nox are higher in a new diesel than a new petrol car, that by 50,000 miles or so they are the same, and after that the petrol engine produces more than the diesel. Therefore over the life cycle of the car, petrol and diesel engine emissions of nitrous oxides are similar. Emissions of nitrous oxides can be effectively reduced in both petrol and diesel cars by use of exhaust gas recirculation (EGR). EGR reduces the combustion temperature to below the point where nitrogen effectively burns.

**EXPERIMENTAL SET UP:**

The experimental set up consists of engine, an alternator, and top load system, fuel tank, exhaust gas recirculation arrangement, exhaust gas measuring instrument and manometer.

**Fig 1.2 Experimental Set up**

**Engine specifications:**

| | | |
|---|---|---|
| Engine | : | NEW KISSAN ENGINE, 4 stroke-stationary. |
| Type | : | water-cooled |
| Injection | : | direct injection (DI) |

Maximum speed : 1500

Number of Cylinder : One

Bore : 85 mm

Stroke : 110 mm

Compression Ratio : 16.5:1

Maximum HP : 5 HP

Injection timing : $25^0$ before TDC

Injection pressure : 200 bar

**Exhaust Gas Recirculation Arrangement:**

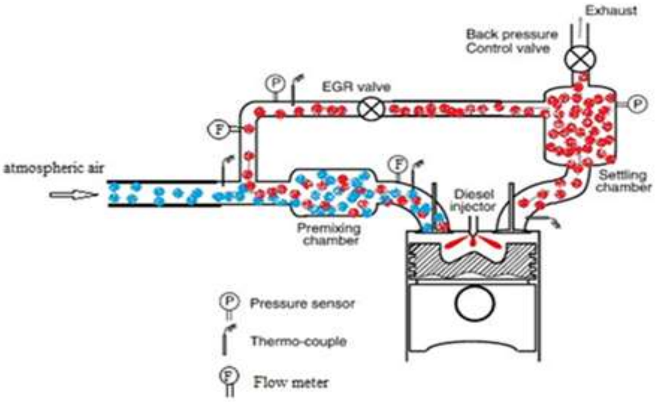



**Figure 5.2 line diagram of exhaust gas recirculation arrangement**

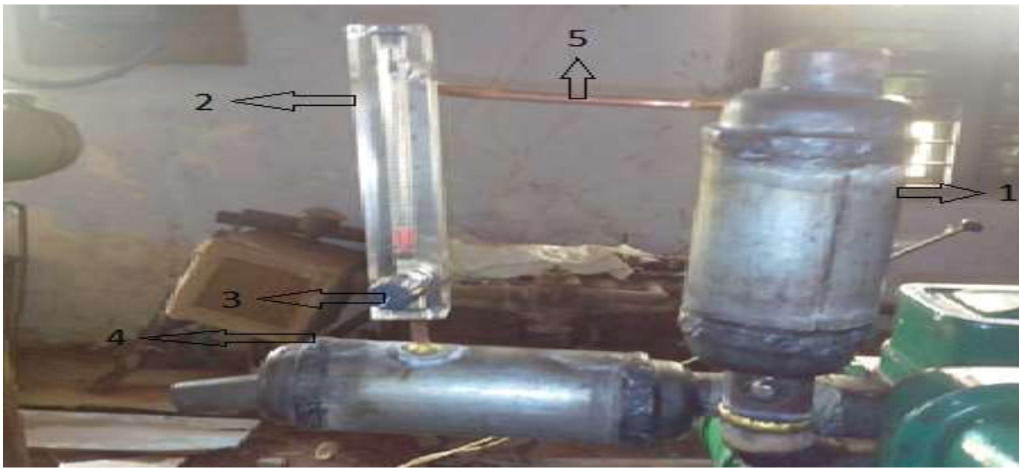



**Fig No 1.3 Line Diagram of Exhaust gas Recirculator**

**Various parts of exhaust gas recirculation arrangement :**
Pre-mixing chamber/rGas flow meter/Exhaust gas regulator/Settling chamber/Copper tube.

**Pre-mixing chamber:** The material selected to prepare the pre-mixing chamber is galvanized iron. The length and diameter of pre-mixing chamber are 20cm & 8.5cm. In this chamber, the air coming from the atmosphere and the exhaust gases are mixing together and sent into the engine intake manifold. This chamber consists of a small provision at bottom end to measure intake charge temperature.

**Gas flow meter:**This flow meter consists of glass body, tube, needle, and regulator. The main function of this gas flow meter is to measure the gas flow rate in terms of liters per minute. The range of gas flow meter is 0 to 10%. Accuracy of this gas flow meter is ±0.25%.

**Exhaust gas regulator:**The main function of this regulator is to control the flow of exhaust gases. It regulates the flow of exhaust gases from settling chamber to the pre mixing chamber.

**Settling chamber:**The material selected to prepare the pre-mixing chamber is galvanized iron. The length and diameter of pre-mixing chamber are 25cm & 8.5cm. In this chamber a part of the exhaust gases re-circulated and remaining are sent to the atmosphere. This chamber consists of a small provision at one end to measure the exhaust gas temperature.
**Copper tube:**Copper tube is connected between the gas flow meter, premixing, and settling chambers. Copper tube is a connection pipe between flow meter, pre mixing chamber and settling chambers. The main aim of selection of copper tube is it reduces the exhaust gas temperatures and easy to adjust in any direction. The diameter of copper tube is 8mm.

**Performance evaluation:**

Measurement of brake power**:**The power developed by the engine is measured by using electrical dynamometer. An alternator is run by using the power developed by the engine. The total power is obtained by adding dynamometer power to the product of voltage and current.

Measurement of fuel**:**The fuel flow is measured by volume through a burette tube which is fixed between fuel tank and fuel pump. A T-joint prepared and one side of it is connected to the fuel measuring tube. The remaining two sides of the joints are connected to the fuel tank and the fuel pump respectively. Fuel flow is measured by nothing the time taken for 10cc of fuel consumption by stop watch.

Measurement of air flow**:**Air flow is measured by using a fan type velocity flow meter. A fan is provided at one end. Place this at opening of intake manifold gives the velocity of air flow into the cylinder. This flow meter is electronically controlled one. It displays the readings directly on instrument display board.

Load on engine**:**Initially fuel tank is filled with right amount of required fuel. The instruments such as NOx meter and CO/HC analyzer are connected to the exhaust pipe. The engine is started and allowed to run for 20 minutes to attain steady state condition.

Measurement of emissions**:** Engine emissions are measured by using exhaust gas analyzer. In this experiment five gas analyzer is used to measure the exhaust emissions. The gases to be measured are CO, HC, NOx, O2, and $CO_2$.

Measurement of exhaust gas temperature**:**The temperature of exhaust gas is measured by using K-type thermometer. It gives the exhaust gas temperature directly. The normal range of this thermometer is -50 to1500°c.

**Experimental Procedure:**

Before starting the engine, premixing chamber is fitted to the intake manifold and settling chamber is fitted to the exhaust manifold. Finally, a gas flow meter is fitted in between pre mixing and settling chambers. As first said, diesel alone is allowed to run the engine for about 30 min, so that it gets warmed up and steady running conditions are attained. Before starting the engine, the lubricating oil level in the engine is checked and it is also ensured that all moving and rotating parts are lubricated.

**Sample calculations for 32% EGR at load 2500watts:**

1. Brake power $= \frac{\mathrm{VIcos}\emptyset}{\eta_{tran}\ \mathrm{x}\ \eta_{gen}\mathrm{x}\ 1000}$ KW

Where,

Voltage (V) =230 volts

Current (I) =7.5 amperes

Cos Ø=Power factor=1

Transmission Efficiency ($\eta_{tran}$) = 0.98

Generator Efficiency ($\eta_{gen}$) = 0.9

$$\text{Brake power} = \frac{230 \times 7.5 \times 1}{0.98\text{x}0.9\ \text{x}\ 1000}\ \text{KW}$$

B.P=1.956 KW

. T.F.C = $\frac{20\ \mathrm{x}\ 0.85\ \mathrm{x}\ 3600}{\mathrm{t\ x}\ 1000}$ Kg/hr.

Where,

Specific gravity of diesel= 0.85

Time taken for 10 c.c fuel (t) = 50seconds

$$= \frac{20\ \text{x}\ 0.85\ \text{x}\ 3600}{50\ \text{x}\ 1000}\ \text{Kg/hr.}$$

Total Fuel Consumption= 0.598Kg/h r

. Brake Specific Fuel Consumption, bsfc= $\frac{\mathrm{T.F.C}}{\mathrm{B.P}}$ Kg/KWh

$$=\frac{0.958}{1.956}\ \text{Kg/kWh}$$

Brake Specific Fuel Consumption, bsfc= 0.306 Kg/KWh

. Heat Input= T.F.C X C.V, KW

Calorific Value of Fuel = 43000 KJ/kg °k

T.F.C =0.598 KW

Heat Input= 0.598×43000 KW

Heat Input= 7.147 KW

. Frictional Power, F.P=1.62 KW (from graph by William's line method)

. Indicated Power =B.P + F.P, KW

Indicated Power =1.956+1.62 KW

Indicated Power= 3.576 KW

. Mechanical efficiency = $\frac{B.P}{I.P}$x 100 %

$=\frac{1.956}{3.576}$ x 100 %

Mechanical Efficiency= 54.69 %

. Brake thermal efficiency, $\eta_{b.th}$ = $\frac{B.P}{\text{Heat Input}}$ x 100 %

$=\frac{1.956}{7.147}$ x 100 %

Brake thermal efficiency = 27.37%

Indicated thermal efficiency, $\eta_{i.th}$= $\frac{I.P}{\text{Heat Input}}$ x 100 %

$=\frac{3.576}{7.147}$ x 100 %

Indicated thermal efficiency= 50.03%

Brake Mean Effective Pressure, bmep $=\frac{B.P \times 60}{L \times A \times n \times k}$ KN/m$^2$

Where,

L = length of the stroke, m

n = speed of the engine = N/2

A = Area of the cylinder, m$^2$

k = no. of cylinders

$$= \frac{1.956 \text{ x } 60}{0.110 \text{x } 5.675 \times 10^{-3} \text{x } 680 \text{x } 1} \text{KN/m}^2$$

Brake Mean Effective Pressure = 276.47 KN/m$^2$

Indicated Mean Effective Pressure, Imep $=\frac{I.P \times 60}{L \times A \times n \times k}$, KN/m$^2$

$$= \frac{3.576 \text{ x } 60}{0.110 \text{x } 5.675 \times 10^{-3} \text{x } 680 \text{x } 1} \text{KN/m}^2$$

Indicated Mean Effective Pressure, Imep= 505.47 KN/m$^2$

. Volumetric efficiency, $\eta_{vol}$ = $\frac{\text{actual volume flow rate of air}}{\text{the rate at which volume is displaced}}$ x 100 %

$$= \frac{\text{area of inlet pipex velocity of air}}{\begin{bmatrix}\text{are of}\\ \text{the cylinder}\end{bmatrix} \text{x} \begin{bmatrix}\text{length of}\\ \text{the stroke}\end{bmatrix} \text{x} \begin{bmatrix}\text{revolutions}\\ \text{per second}\end{bmatrix}} \text{ x } 100\%$$

$$= \frac{3.17 \times 0.024^2 \times 6.4}{3.17 \times 0.8^2 \times 0.11 \times 11.16} \text{ X } 100\%$$

Volumetric efficiency, $\eta_{vol}$ =77.38 %

**Table NO 1.0 Performance and Emission Characteristic results for I/C Engine**

| S .No | LOAD (watts) | VOLTAGE (volts) | CURRENT (amps) | TIME FOR 10cc OF FUEL (sec) | | | | VELOCITY OF AIR INTAKE(m/s) | | | | EXHAUST GAS TEMPERATURE(°C) | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | 0% | 16% | 32% | 48% | 0% | 16% | 32% | 48% | 0% | 16% | 32% | 48% |
| 1 | 0 | 230 | 0 | 87 | 87 | 87 | 86 | 4.3 | 4.3 | 4.2 | 4.3 | 157 | 145 | 158 | 142 |
| 2 | 500 | 230 | 2.0 | 83 | 77 | 78 | 76 | 4.7 | 4.9 | 4.5 | 4.6 | 190 | 172 | 188 | 172 |
| 3 | 1000 | 230 | 3.0 | 73 | 67 | 64 | 68 | 5.1 | 5.3 | 4.9 | 4.9 | 217 | 195 | 200 | 194 |
| 4 | 1500 | 230 | 5.0 | 65 | 62 | 55 | 61 | 5.4 | 5.6 | 5.3 | 5.1 | 221 | 214 | 215 | 214 |
| 5 | 2000 | 230 | 6.0 | 58 | 54 | 52 | 55 | 5.7 | 5.9 | 5.8 | 5.6 | 247 | 239 | 247 | 237 |
| 6 | 2500 | 230 | 7.5 | 51 | 49 | 50 | 52 | 6.1 | 6.3 | 6.4 | 6.5 | 258 | 252 | 252 | 257 |
| 7 | 3000 | 230 | 10 | 43 | 43 | 42 | 42 | 6.8 | 6.9 | 7.0 | 7.1 | 313 | 309 | 312 | 313 |
| 8 | 3500 | 230 | 11 | 41 | 41 | 40 | 37 | 7.4 | 7.4 | 7.4 | 7.3 | 335 | 323 | 330 | 328 |

**Performance and emission Characteristic Results for ZERO %EGR (Without EGR)**

**Table No 1.1 Performance and Emission Characteristic results for 0% EGR**

| S.No | B.P (KW) | TFC (Kg/hr) | BSFC (Kg/KWh) | F.P (KW) | I.P (KW) | $\eta_{mech}$ (%) | H.I (KW) | $\eta_{b.th}$ (%) | $\eta_{i.th}$ (%) | BMEP $(KN/m^2)$ | IMEP $(KN/m^2)$ | $\eta_{vol}$ (%) | $O_2$ %vol | CO %vol | HC ppm | $CO_2$ %vol | $NO_X$ ppm |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 0 | 0.344 | 0 | 1.60 | 1.600 | 0 | 4.107 | 0 | 45.30 | 0 | 263.04 | 51.99 | 23.24 | 0.039 | 17 | 0.7 | 1 |
| 2 | 0.522 | 0.360 | 0.691 | 1.60 | 2.122 | 24.58 | 4.305 | 12.11 | 49.28 | 73.72 | 299.90 | 56.82 | 22.10 | 0.04 | 19 | 0.9 | 2 |
| 3 | 0.782 | 0.410 | 0.524 | 1.60 | 2.382 | 32.84 | 4.895 | 15.98 | 48.67 | 110.59 | 336.76 | 61.66 | 21.92 | 0.046 | 21 | 1 | 7 |
| 4 | 1.304 | 0.460 | 0.353 | 1.60 | 2.904 | 44.90 | 5.497 | 23.72 | 52.82 | 184.31 | 410.48 | 65.29 | 21.56 | 0.061 | 25 | 1.1 | 9 |
| 5 | 1.565 | 0.516 | 0.330 | 1.60 | 3.165 | 49.44 | 6.161 | 25.40 | 51.37 | 221.17 | 447.34 | 68.91 | 21.06 | 0.088 | 32 | 1.3 | 12 |
| 6 | 1.956 | 0.587 | 0.300 | 1.60 | 3.556 | 55.00 | 7.006 | 27.91 | 50.75 | 276.47 | 502.64 | 73.75 | 20.48 | 0.136 | 39 | 1.5 | 15 |
| 7 | 2.608 | 0.696 | 0.267 | 1.60 | 4.208 | 61.97 | 8.310 | 31.38 | 50.63 | 368.62 | 594.80 | 82.21 | 19.86 | 0.194 | 45 | 1.7 | 18 |
| 8 | 2.868 | 0.730 | 0.254 | 1.60 | 4.468 | 64.19 | 8.715 | 32.91 | 51.27 | 405.48 | 631.66 | 89.47 | 19.08 | 0.310 | 58 | 2.5 | 27 |

**Performance and emission Characteristic Results for 16% EGR**

**Table No 1.2 Performance and Emissions Characteristic for 16% EGR**

| S.No | B.P (KW) | TFC (Kg/hr) | BSFC (Kg/KW.hr) | F.P (KW) | I.P (KW) | $\eta_{mech}$ (%) | H.I (KW) | $\eta_{b.th}$ (%) | $\eta_{i.th}$ (%) | BMEP $(KN/m^2)$ | IMEP $(KN/m^2)$ | $\eta_{vol}$ (%) | $O_2$ %vol | CO %vol | HC ppm | $CO_2$ %vol | $NO_X$ ppm |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 0 | 0.344 | 0 | 1.54 | 1.54 | 0 | 4.107 | 0 | 43.84 | 0 | 254.55 | 51.99 | 22.87 | 0.032 | 12 | 0.9 | 0 |
| 2 | 0.522 | 0.389 | 0.745 | 1.54 | 2.062 | 25.30 | 4.641 | 11.24 | 44.42 | 73.72 | 291.42 | 59.24 | 21.84 | 0.039 | 18 | 1.0 | 2 |
| 3 | 0.782 | 0.447 | 0.571 | 1.54 | 2.322 | 33.69 | 5.333 | 14.67 | 43.54 | 110.59 | 328.28 | 64.08 | 19.81 | 0.046 | 19 | 1.1 | 2 |
| 4 | 1.304 | 0.483 | 0.370 | 1.54 | 2.844 | 45.85 | 5.763 | 22.62 | 49.34 | 184.31 | 402.00 | 67.71 | 19.66 | 0.061 | 29 | 1.2 | 3 |

| | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 5 | 1.565 | 0.554 | 0.354 | 1.54 | 3.105 | 50.40 | 6.617 | 23.64 | 46.92 | 221.17 | 438.86 | 71.33 | 19.34 | 0.088 | 34 | 1.6 | 4 |
| 6 | 1.956 | 0.611 | 0.312 | 1.54 | 3.496 | 55.95 | 7.292 | 27.82 | 47.94 | 276.47 | 494.16 | 76.17 | 18.81 | 0.132 | 39 | 1.7 | 4 |
| 7 | 2.608 | 0.696 | 0.267 | 1.54 | 4.148 | 62.87 | 8.310 | 31.38 | 49.91 | 368.62 | 586.31 | 83.42 | 18.23 | 0.199 | 50 | 1.8 | 5 |
| 8 | 2.868 | 0.730 | 0.254 | 1.54 | 4.408 | 65.07 | 8.715 | 32.91 | 50.58 | 405.48 | 623.18 | 89.47 | 17.84 | 0.34 | 58 | 2.3 | 10 |

**Performance and emission Characteristics Results for 32% EGR**

**Table No 1.3 Performance and Emission Characteristic for 32% EGR**

| **S.No** | **B.P** (KW) | **TFC** (Kg/hr) | **BSFC** (Kg/KW.hr) | **F.P** (KW) | **I.P** (KW) | **ηmech** (%) | **H.I** (KW) | **ηb.th** (%) | **ηi.th** (%) | **BMEP** (KN/m²) | **IMEP** (KN/m²) | **ηvol** (%) | **$O_2$** %vol | **CO** %vol | **HC** ppm | **$CO_2$** %vol | **$NO_x$** ppm |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 0 | 0.344 | 0 | 1.62 | 1.62 | 0 | 4.107 | 0 | 45.79 | 0 | 265.86 | 50.78 | 22.78 | 0.036 | 16 | 1 | 1 |
| 2 | 0.522 | 0.384 | 0.735 | 1.62 | 2.142 | 24.35 | 4.581 | 11.38 | 46.75 | 73.72 | 302.72 | 54.41 | 21.58 | 0.042 | 22 | 1.1 | 3 |
| 3 | 0.782 | 0.467 | 0.598 | 1.62 | 2.402 | 32.56 | 5.583 | 14.01 | 43.03 | 110.59 | 339.59 | 59.24 | 19.76 | 0.056 | 22 | 1.4 | 2 |
| 4 | 1.304 | 0.544 | 0.417 | 1.62 | 2.924 | 44.59 | 6.497 | 22.87 | 45.00 | 184.31 | 413.31 | 64.08 | 19.67 | 0.082 | 27 | 1.6 | 2 |
| 5 | 1.565 | 0.575 | 0.368 | 1.62 | 3.185 | 49.13 | 6.872 | 25.77 | 46.34 | 221.17 | 450.17 | 70.12 | 19.26 | 0.096 | 32 | 1.7 | 1 |
| 6 | 1.956 | 0.598 | 0.306 | 1.62 | 3.576 | 54.69 | 7.147 | 27.87 | 50.03 | 276.47 | 505.47 | 77.38 | 18.68 | 0.148 | 38 | 1.8 | 1 |
| 7 | 2.608 | 0.712 | 0.273 | 1.62 | 4.228 | 61.68 | 8.508 | 30.65 | 49.69 | 368.62 | 597.62 | 84.63 | 18.04 | 0.206 | 49 | 2.2 | 3 |
| 8 | 2.868 | 0.748 | 0.261 | 1.62 | 4.488 | 63.91 | 8.933 | 32.11 | 50.24 | 405.48 | 634.48 | 89.47 | 17.82 | 0.348 | 60 | 2.4 | 5 |

| **S.No** | **B.P** (KW) | **TFC** (Kg/hr) | **BSFC** (Kg/KW.hr) | **F.P** (KW) | **I.P** (KW) | **ηmech** (%) | **H.I** (KW) | **ηb.th** (%) | **ηi.th** (%) | **BMEP** (KN/m²) | **IMEP** (KN/m²) | **ηvol** (%) | **$O_2$** %vol | **CO** %vol | **HC** ppm | **$CO_2$** %vol | **NOx** ppm |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 0 | 0.348 | 0 | 1.56 | 1.560 | 0 | 4.155 | 0 | 43.82 | 0 | 257.38 | 51.99 | 22.27 | 0.054 | 18 | 1.1 | 1 |
| 2 | 0.522 | 0.394 | 0.755 | 1.56 | 2.082 | 25.06 | 4.702 | 11.09 | 44.27 | 73.72 | 294.24 | 55.62 | 21.00 | 0.065 | 20 | 1.2 | 2 |
| 3 | 0.782 | 0.440 | 0.562 | 1.56 | 2.342 | 33.40 | 5.255 | 14.89 | 44.57 | 110.59 | 331.11 | 59.24 | 19.72 | 0.068 | 25 | 1.6 | 3 |
| 4 | 1.304 | 0.490 | 0.376 | 1.56 | 2.864 | 45.53 | 5.858 | 22.26 | 48.89 | 184.31 | 404.83 | 61.66 | 19.38 | 0.082 | 26 | 1.7 | 4 |

**Performance and emission test results for 48% EGR**

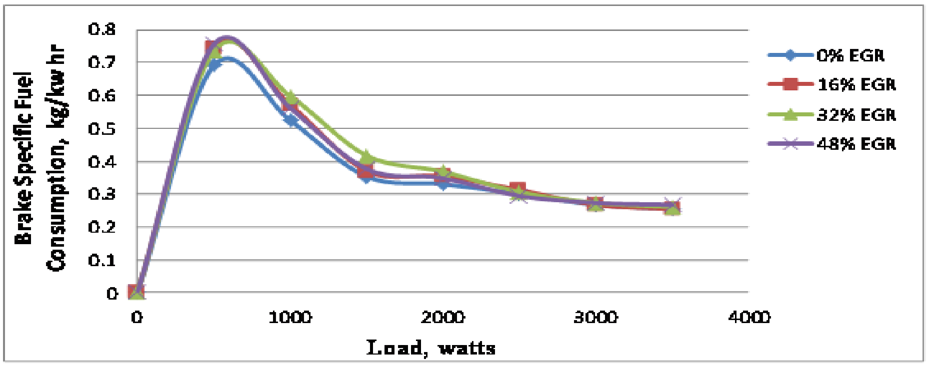
**Table No 1.4 Performance and emission characteristic for 48% EGR RESUL**

| | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 5 | 1.5 65 | 0.5 44 | 0.3 48 | 1.5 6 | 3.1 25 | 50. 07 | 6.4 97 | 24. 98 | 48. 09 | 221 .17 | 441 .69 | 67. 71 | 19. 14 | 0.1 09 | 35 | 1.8 | 3 |
| 6 | 1.9 56 | 0.5 75 | 0.2 94 | 1.5 6 | 3.5 16 | 55. 63 | 6.8 72 | 27. 46 | 51. 16 | 276 .47 | 496 .99 | 78. 59 | 18. 06 | 0.1 57 | 38 | 1.9 | 3 |
| 7 | 2.6 08 | 0.7 12 | 0.2 73 | 1.5 6 | 4.1 68 | 62. 57 | 8.5 08 | 30. 65 | 48. 99 | 368 .62 | 589 .14 | 85. 84 | 17. 64 | 0.2 16 | 50 | 2.2 | 4 |
| 8 | 2.8 68 | 0.7 67 | 0.2 67 | 1.5 6 | 4.4 28 | 64. 77 | 9.1 62 | 31. 31 | 48. 33 | 405 .48 | 626 .00 | 88. 26 | 16. 98 | 0.3 54 | 61 | 2.4 | 5 |

**RESULTS AND DISCUSSION**

Experiments were conducted when the engine was fuelled with normal diesel at different openings like 0%, 16%, 32%, and 48% of exhaust gas recirculation at different loads respectively.

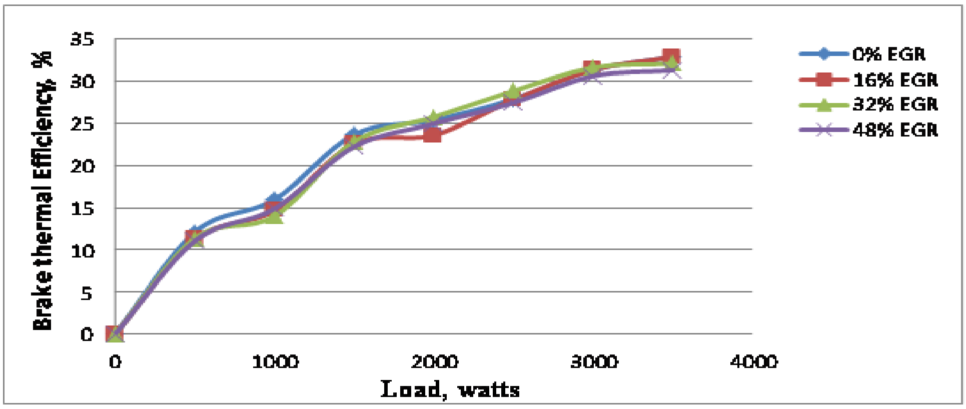
**Brake specific fuel consumption:**




**Graph 1.0 Load Vs Brake specific fuel consumption**

The result for the variations in the brake specific fuel consumption (BSFC) at different Exhaust Gas Recirculation openings with respect to load is shown in fig. For all the EGR openings the BSFC falls with increasing load. The maximum BSFC value at 0% EGR is 0.691Kg/KWh, at 16% EGR is 0.745 Kg/KWh, at 32% EGR is 0.735 Kg/KWh, and at 48% EGR is 0.75 Kg/KWh. But at full loads the brake specific fuel consumption (BSFC) is closer to the all EGR openings. The brake specific fuel consumption (BSFC) is remains unaffected at full load conditions.

**Brake thermal efficiency:**

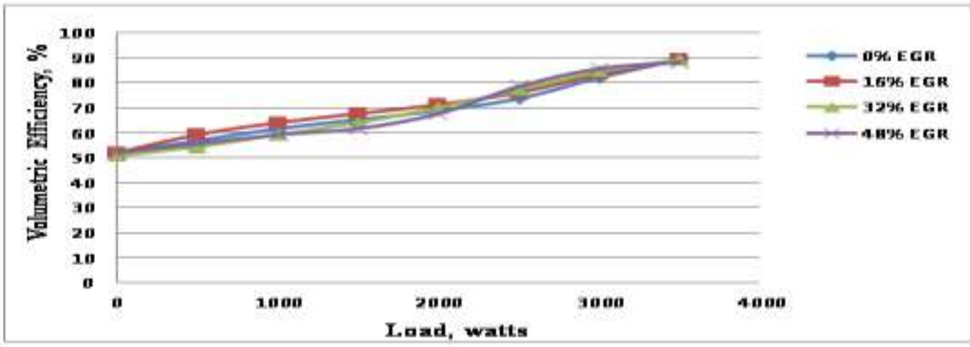



**Graph1.1 Load Vs Brake thermal efficiency**

The variation of brake thermal efficiency with respect to load is shown in fig.7.2. Brake thermal efficiency gives an idea of the output generated by the engine with respect to heat supplied in the form of fuel. For all EGR openings the brake thermal efficiency increases with load. The maximum brake thermal efficiency values at full load at 0% EGR is 32.91%, at 16% EGR is

32.91%, at 32% EGR is 32.11%, and at 48% EGR is 31.31%. The brake thermal efficiencies at 2000W and 2500W for 32% EGR are higher than the values of brake thermal efficiencies with 0%EGR. However at other loads the brake thermal efficiency for 0%EGR is higher in comparison with values at other EGR openings. This may be due the replacement of oxygen with exhaust gases inside the combustion chamber

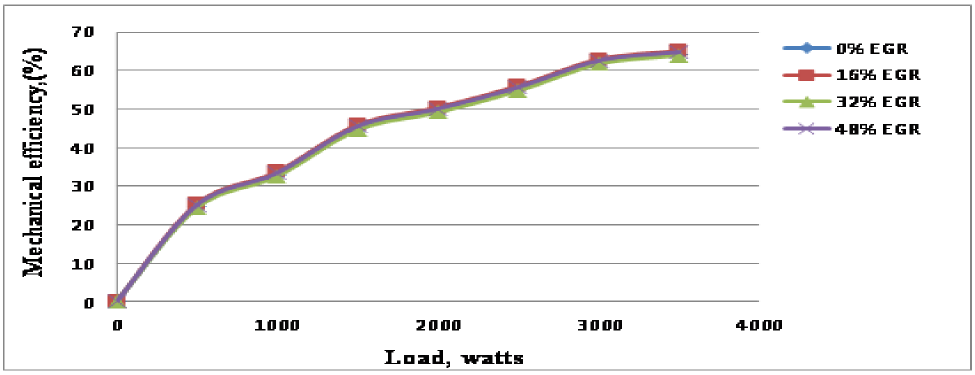
**Volumetric efficiency:** The fig.7.3 shows the variation of volumetric efficiency with load for various EGR openings. Volumetric efficiency is a measure of success with which the air supply, and thus the charge, is inducted in to the engine. It indicates the breathing capacity of the engine. From the figure it is evident that the volumetric efficiency is closer for with and without EGR openings at all loads.




**Graph 1.2Load Vs Volumetric efficiency**

The maximum volumetric efficiency at 0% is EGR 89.47%, at 16% EGR is 89.47%, at 32% EGR is 89.47% and at 48%EGR is 88.26%.

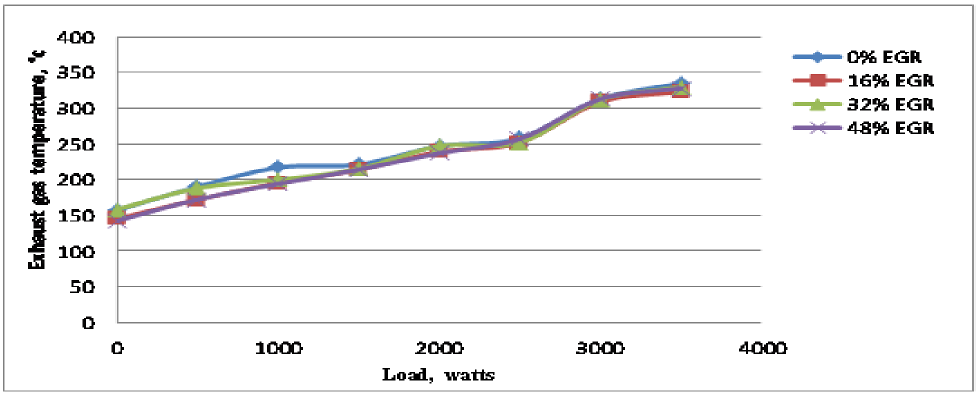
**Mechanical Efficiency:**




**Graph 1.3 Load Vs Mechanical efficiency**

The observed Mechanical efficiency at 0% EGR is 64.19%, at 16% EGR is 65.07%, at 32% EGR is 63.91%, and at 48% EGR is 64.77%. It is observed that the mechanical efficiencies for with EGR are increased than without EGR. The increasing trend of mechanical efficiency with load stands good even for the engine with EGR irrespective of its percentage.

**Exhaust gas Temperature:**




**Graph 1.4 Load Vs Exhaust gas temperature**

It is observed that the exhaust gas temperature increases with load because more fuel is burnt at higher loads to meet the power requirement. The exhaust gas temperatures are decreased when using EGR system. The maximum temperatures for without EGR is335°c and with EGR 323°c at full load condition and minimum exhaust gas temperatures for with and without EGR are, 157 °c and 142 °c

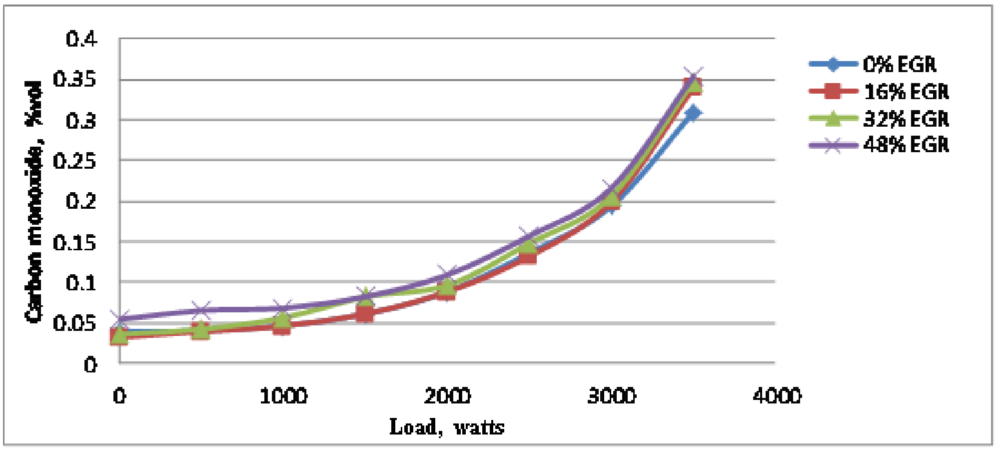
**HC Emission:**

The variation of hydro carbons respect to load shown in Fig.7.6. The HC emissions are minimum for with EGR at part loads when compared to without EGR i.e., at 0% EGR. The Hydrocarbon emissions for 32% &48% EGR are decreased at load 2000&2500W when compared to 0% EGR. Whereas, load increases the HC emissions are also increases for with EGR compared to without EGR

**Load Vs Hydrocarbons**

The maximum HC values at full load at 0% EGR is 58ppm, at 16% EGR is 58ppm, at 32%EGR is 60ppm, and at 48% EGR is 61ppm

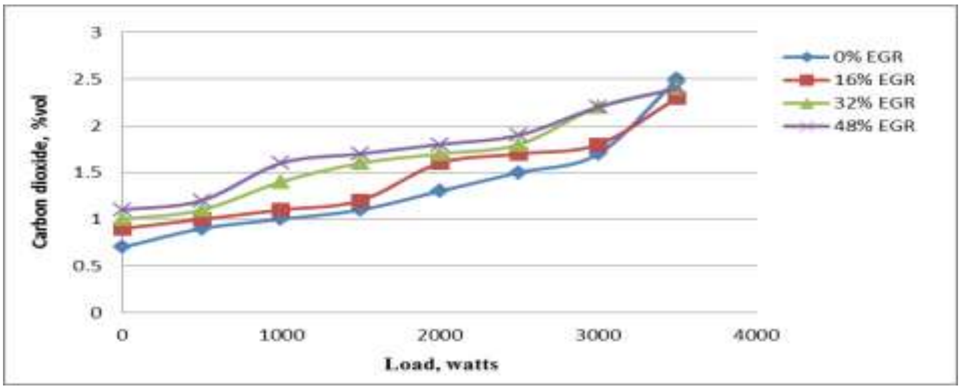
**CO Emission:**




**Graph 1.6 Load Vs Carbon monoxide**

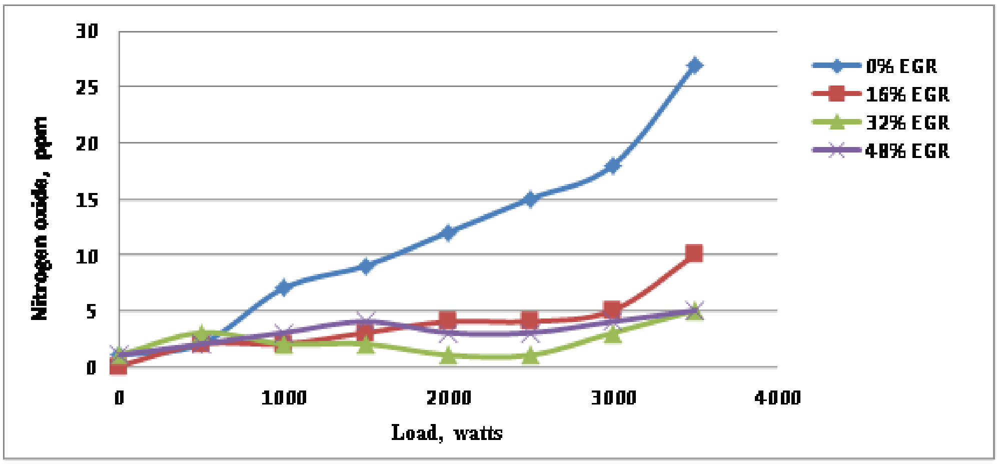
CO is predominantly formed due to the lack of oxygen. The fig. 7.7 shows that the CO emissions are increased with increase in load and EGR rates. The observed values at full load at 0% EGR is 0.31%vol, at16% EGR is 0.34%vol, at 32% EGR is 0.348%vol, and at 48% EGR is 0.354%vol.

**$CO_2$ Emission:**




**Graph 1.7 Load Vs Carbon dioxide**

The variation of carbon dioxide emission with respect to load is observed. The $CO_2$ are increased by increasing the load and EGR rates. At loads 2500W at 0% EGR is 1.5%vol, at 16% EGR is 1.7%vol, at 32% EGR is 1.8%vol, and at 48% EGR is 1.9%vol. From the observation the amount of $CO_2$ produced for with EGR system is increased when compared to without EGR.**$NO_X$ Emissions:**




**Graph 1.8 Load Vs Nitrogen oxide**

The variation of $NO_X$ emissions with respect to load at different EGR openings shown in fig.7.9 $NO_X$ emissions are more for normal diesel engine without EGR. It is observed that $NO_X$ emissions are drastically reduced by using the EGR system. The maximum $NO_X$ values at full load at 0% EGR is 27ppm, at 16% EGR is 10ppm, at 32% EGR is 5ppm, and at 48% EGR is 5ppm

**CONCLUSION:**

The conclusions based on the experimental results obtained while operating single cylinder water cooled diesel engine operated with, with and without EGR system.

1.Brake thermal efficiency is increased by 0.27% at 32% EGR, when compared to without EGR at load 3000W respectively.

2.Brake specific fuel consumption for all EGR openings at full load is remains unchanged when compared to without EGR.

3.The Mechanical efficiency is decreased at 32% EGR by 0.28%, when compared to without EGR at load 3500W.

4.Volumetric efficiency remains unchanged at 32% EGR when compared to without EGR at full load.

5.Exhaust gas temperature for all EGR openings is lower when compared to without EGR system.

6.Hydro carbon emission at 32% EGR decreases when compared to without EGR at load 2500W condition.

7.Carbon monoxide emissions at 32% EGR are increased by 0.037%, when compared to without EGR at full load.

8.Carbon dioxide emissions at 32% EGR are increased by 0.3%, when compared to without EGR at load 2500W.

9.$NO_x$ emissions are greatly decreased for all EGR openings, when compared with the without EGR.From the investigation, it is recommended that EGR system at 32% EGR at loads 2500W & 3000W for 4-stroke single cylinder diesel engine is preferable

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