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Effect of Fuel Injection Pressure on Performance of Single Cylinder Diesel Engine at Different Intake Manifold Inclinations

M.L.S Deva Kumar, S.Drakshayani, K.Vijaya Kumar Reddy

Abstract: - In present days an automobile engine has to satisfy the strict environmental constraints and fuel economy standards in addition to meeting the competitiveness of the world market. Today the automobile engines have synthesized the basic knowledge of many disciplines like thermodynamics, fluid flow, combustion, chemical kinetics and heat transfer. Now-a-days internal combustion engines play an important role in automobile field. There are various factors that influence the performance of engine such as compression ratio, atomization of fuel, fuel injection pressure, and quality of fuel, combustion rate, air fuel ratio, intake temperature and pressure and also based on piston design, inlet manifold, and combustion chamber designs etc. Growing demand on reduction of internal combustion engine fuel consumption with increase of its performance new designs and optimization of existing ones are introduced. Air motion in CI engine influences the atomization and distribution of fuel injected in the combustion chamber. Fuel injection pressure plays an important role in better atomization of injected fuel allows for a more complete burn and helps to reduce pollution. In present work a single cylinder 5HP diesel engine is used to investigate the performance characteristics. The main objective of this work is to study the effect of the fuel injection pressure on performance and pollution of the single cylinder diesel engine at different intake manifold inclinations. From experiment it is found that engine at 60^0 manifold inclinations at 180 bars has given efficient performance and less pollution.

I. INTRODUCTION

Diesel engine plays a dominant role in the field of power, propulsion and energy. The diesel engine is a type of internal combustion engine ,more specifically it is a compression ignition engine ,in which the fuel ignited solely by the high temperature created by compression of the air-fuel mixture. The engine operates using the diesel cycle. The engine is more efficient than the petrol engine, since the spark –ignition engine consumes more fuel than the compression –ignition engine. The fuel injection system is the most vital component in the working of CI engine. The engine performance, power output, economy etc is greatly dependent on the effectiveness of the fuel injection system. The injection system has to perform the important duty of initiating and controlling the combustion process. When the fuel is injected in to the combustion chamber towards the end of compression stroke, it is atomized into very fine droplets. These droplets vaporize due to heat transfer from the compressed air and from an air-fuel mixture. Due to continued heat transfer from hot air to the fuel, the temperature reaches a value higher than its self- ignition temperature. This causes the fuel to ignite spontaneously initiating the combustion process. Quick and complete combustion is ensured by a well designed fuel injector. By atomizing the fuel into very fine droplets, it increases the surface area of the fuel droplets resulting in better mixing and subsequent combustion. Atomization is done by forcing the fuel through a small orifice under high pressure. The injector assembly consists of.

1. A needle valve
2. A compression spring
3. A nozzle
4. An injector body

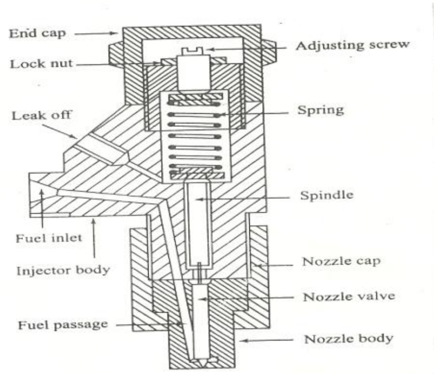

Fig 1. Fuel Injector

When the fuel is supplied by the injection pump it exerts sufficient force against the spring to lift the nozzle valve, fuel is sprayed into the combustion chamber in a finely atomized particles. After fuel from the delivery pump gets exhausted; the spring pressure pushes the nozzle-valve back on its seat. For proper lubrication between nozzle valve and its guide a small quantity of fuel is allowed to leak through the clearance between them and then drained back to fuel tank through leak of connection. In present diesel engines, fuel injection systems have designed to obtain higher injection pressure. So, it is aimed to decrease the exhaust emissions by increasing efficiency of diesel engines. When fuel injection pressure is low, fuel particle diameters will enlarge and ignition delay period during the combustion will increase .This situation leads to increased pressure and the engine performance will decrease. When injection pressure increases fuel particle diameters will become small. Since the mixing of fuel and air becomes better during ignition period. Engine performance will increase

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if injection pressure is too high, ignition delay period becomes shorter. Possibilities of homogeneous mixing decreases and combustion efficiency fall down. The fuel injection system in a direct injection diesel engine is to achieve a high degree of atomization in order to enable sufficient evaporation in a very short time and to achieve sufficient spray penetration in order to utilize the full air charge. The fuel injection system must be able to meter the desired amount of fuel, depending on engine speed and load, and to inject that fuel at the correct time and with the desired rate. Further depending on the particular combustion chamber, appropriate spray shape and structure must be produced. usually, a supply pump draws the fuel from the fuel tank and carries its through a filter to the high-pressure injection pump, dependent on the area of application and engine size, pressures between 150 and 200 bar is generated. The high pressure injection pump carries the fuel through high-pressure pipes to the injection nozzles in the cylinder head. Excess fuel is transported back into the fuel tank. The functionality of the so-called unit pump system is practically identical to that of the unit injector system. The details of the diesel engine design vary significantly over the engine performance and size range. In particular, different combustion chamber geometries and fuel `injection characteristics are required to deal effectively with major diesel engine design problem achieving sufficiently rapid fuel-air mixing rates to complete the fuel burning process in the time available. According to Heywood (1988) and Ganesan (1999), a wide variety of inlet port geometries, cylinder head and piston shapes, and fuel-injection patterns are used to accomplish this over the diesel size range. The engine ratings usually indicate the highest power at which manufacturer expect their products to give satisfactory of power, economy, reliability and durability under service conditions. Maximum torque and the speed at which it is achieved, is usually given also by Heywood (1988). The importance of the diesel engine performance parameters are geometrical properties, the term of efficiency and other related engine performance parameters. The engine efficiencies are indicated thermal efficiency, brake thermal efficiency, mechanical efficiency, volumetric efficiency and relative efficiency (Ganesan, 1999). The other related engine performance parameters are mean effective pressure, mean piston speed, specific power output, specific fuel consumption, intake valve mach index, fuel-air or air-fuel ratio and calorific value of the fuel (Heywood, 1988; Ganesan, 1999; Semin et al., 2007). According to Heywood (1988) in the diesel engine geometries design written that diesel engine compression ratio is maximum cylinder volume or the displaced volume or swept and clearance volume divided by minimum cylinder volume. And the power delivered by the diesel engine and absorbed by the dynamometer is the product of torque and angular speed. The engine efficiencies, every its efficiencies defined by Ganesan (1999).

Effect of injection pressure on diesel engine performance

Diesel engine has gained the name and fame in serving the society in many ways. Its main attractions are ruggedness in construction, simplicity in operation and ease of maintenance. But due to the shortage of fossil fuel, we may not be able to avail its services for long time .Hence efforts are being made all over the world, to bring out non-conventional fuels for use in diesel engines. The performance and emission characteristics of diesel engines depends on various factors like fuel quantity injected, fuel injection timing, fuel injection pressure, shape of combustion chamber, position and size of injection nozzle hole, fuel spray pattern, air swirl etc. The fuel injection system in a direct injection diesel engine is to achieve a high degree of atomization for better penetration of fuel in order to utilize the full air charge and to promote the evaporation in a very short time and to achieve higher combustion efficiency. The fuel injection pressure in a standard diesel engine is in the range of 2000 to 1700 rpm depending on the engine size and type of combustion system employed(John B Heywood, 1988). The fuel penetration distance become longer and the mixture formation of the fuel and air was improved when the combustion duration became shorter as the injection pressure became higher (Seang-wock Lee et al., 2002). When fuel injection pressure is low, fuel particle diameters will enlarge and ignition delay period during the combustion will increase. This situation leads to inefficient combustion in the engine and causes the increase in NOx, CO emissions. When the injection pressure is increased fuel particle diameters will become small. The mixing of fuel and air becomes better during ignition delay period which causes low smoke level and CO emission. But, if the injection pressure is too high ignition delay become shorter. So, possibilities of homogeneous mixing decrease and combustion efficiency falls down. Therefore, smoke is formed at exhaust of engine (Rosli Abu Bakar et al., 2008; Venkanna et al., 2009). The diesel engine performance and fuel consumption have been measured at fixed engine speed-variation loads by changing the fuel injection pressure. In the investigation it is to be studied that effects of injection pressure in fixed engine speed-variation engine loads in the fuel injection pressures are setting at 160,180&200 bar.

II. LITERATURE SURVEY

The details of the diesel engine design vary significantly over the engine performance and size range. In particular, different combustion chamber geometries and fuel injection characteristics are required to deal effectively with major diesel engine design problem achieving sufficiently rapid fuel-air mixing rates to complete the fuel burning process in the time available. According to Heywood (1988) and Ganesan (1999), a wide variety of inlet port geometries, cylinder head and piston shapes, and fuel-injection patterns are used to

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III. EXPERIMENTAL SET UP

Inlet manifold plays an important role in performance of an engine. Air flows into the cylinder through the intake port via intake manifold in the form of a jet with maximum velocity at the exit of the intake valve. In the present work inlet manifolds are manufactured with different angle inclinations using mild steel. The manifolds are manufactured with angles 30^0,60^0,90^0 and normal. Also in most of the cases, it is seen that the jet of air after leaving the intake port impinges onto the cylinder wall and diverts back causing the formation of small and large scale vertices within the cylinder. Arrangement of intake manifolds in to different angles as shown in below figures.

- Fig.1 shows the manifold inclination with 90^0 it is observed that there is a jet flow at the exit of the intake valve, vortex formation and flow reversals. The reversal and vortex formation may be due to that cylinder wall hitting of the air jet. Turbulence is created due to the helical grooves provided inside the manifold.
- From fig.2 inclination 30^0, it is similar under 0^0 conditions but from this manifold i9nclination impinging position on the cylinder wall is moving down from lower.
- From fig.3 inclination 60^0 it is like as previous manifolds inclination. It is observed that the flow pattern is irregular , it may due to the impinging of jet to the cylinder wall is far from the exit i.e. towards the open end of the cylinder causing disturbance of the flow with in the cylinder.
- From fig.4 it is normal manifold

Fig: 2 Intake Manifold at 90^0 Inclinations

Fig:3 Intake Manifold at 60^0 Inclinations

Fig:4 Intake Manifold at 30^0 Inclination

Fig: 5 Normal Intake Manifold

IV. VARIATION OF THE INJECTION PRESSURE

The fuel injection system in a direct injection diesel engine is to achieve a high degree of atomization in order to enable sufficient evaporation in a very short time and to achieve sufficient spray penetration in order to utilize the full air charge. The fuel injection system must be able

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to meter the desired amount of fuel, depending on engine speed and load and to inject that fuel at the correct time and with the desired rate. Further on, depending on the particular combustion chamber, the appropriate spray shape and structure must be produced. A supply pump draws the fuel from the fuel tank and carries it through a filter to the high-pressure injector. During this phase of the project, injection pressure is varied. The injection pressure of this high speed diesel engine is approximately 180 bar. The injection pressure of the injector can be varied by tightening or loosening the screw of the injector as shown in the figure1. The injector pressure can be determined by a fuel injector pressure tester as shown in the figure2 given below.

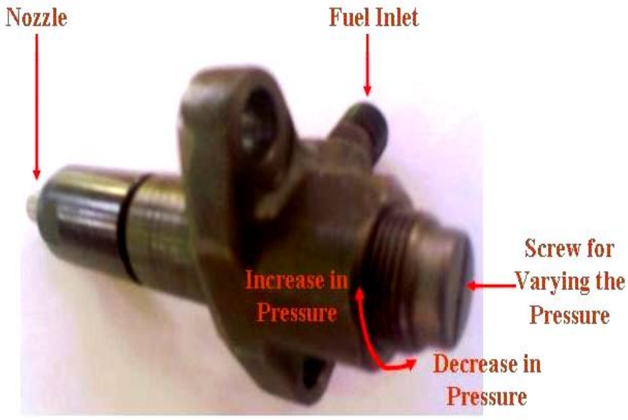

Fig.6 Fuel Injector

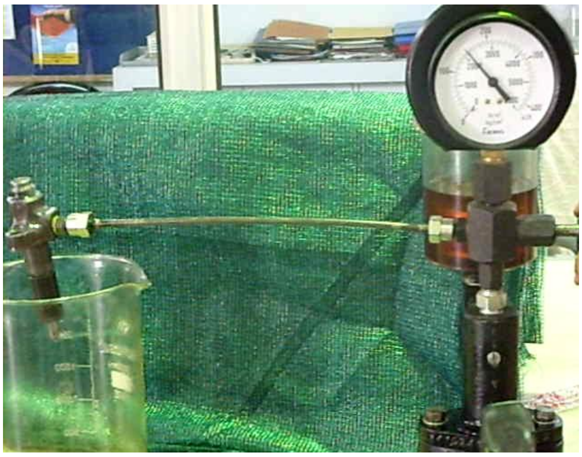

Fig.7. Pressure Tester Gauge

V. ENGINE PROCESS

In this project we had conducted experiment on 4-stroke engine. The engine utilizes 4-stroke namely suction, compression, power and exhaust stroke for completion of cycle. And it consists of 2 valves, inlet valve and exhaust valve. Inlet valve is used for the inflow of the air fuel mixture or pure air. Exhaust valve is used for removal of exhaust burnt gases. At the start of the working cycle the piston is at TDC. As the piston begins its outward stroke the inlet valve opens and a mixture of fuel and air are metered proportionally flow in. the air or air fuel mixture thus trapped between the piston and cylinder head is now compressed, until the piston reaches the TDC at the end of compression stroke. Just before the end of compression stroke of the piston a spray of oil injected into the cylinder. The thermal energy released makes the compressed gas expand rapidly and drives the piston outwards. The resulting stroke is called power stroke. As the piston completes the power stroke and returns TDC. The exhaust valve opens and burnt exhaust gases in cylinder are driven out.

VI. EXPERIMENTAL SETUP

A single cylinder 4-stroke water-cooled diesel engine having 5 hp as rated power at 1500 rpm was used for the research work. The engine was coupled to an electrical dynamometer for loading it. A photo sensor along with a digital rpm indicator was used to measure the speed of the engine. The fuel flow rate was measured on volumetric basis using were used for measuring the engine and exhaust gas temperatures. The experimental set-up of the engine is shown in the figure.

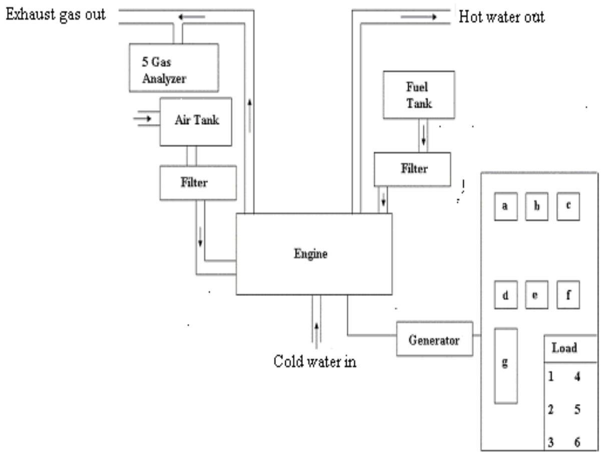

Fig.8. Experimental Setup

VII. ENGINE SPECIFICATIONS

Engine: kirloskar diesel engine with D.c generator
Coefficient of discharge: 0.65
Injection timing : 28^0 BDC
Compression ratio : 16.5:1
Orifice diameter : 20mm
Maximum H.P : 5 H.P
Stroke : 110mm
Bore : 80mm
Type : water cooled
Maximum load of engine: 15.252 amps

Preliminaries:-

- check the fuel level in the tank
- check lubricating oil level with the help of oil stick.
- open the three way cock so that the water flows continuously.
- check the water level and top up the tank if the water is not up to the level in the water rheostat.

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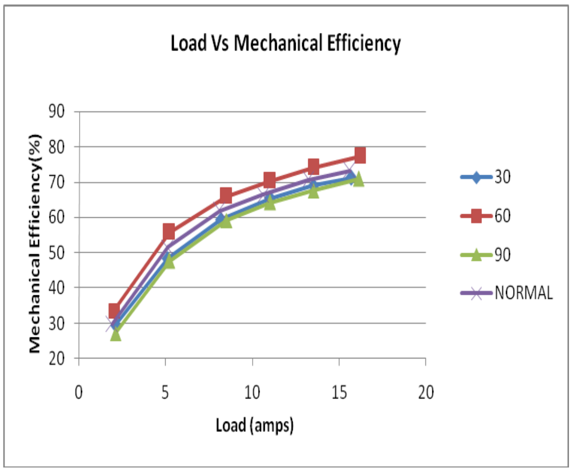
VIII. PROCEDURE OF EXPERIMENT

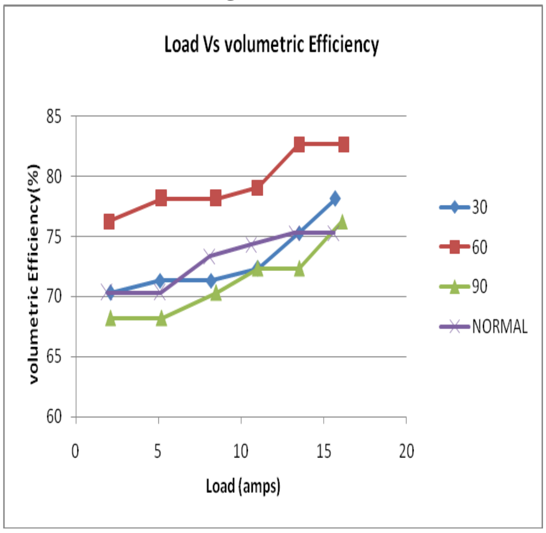
1. After carefully going through the preliminaries the decompression lever is pressed on so that there will not be any air trapping in between the cylinder head and piston.
2. Then the engine is started by rotating the crank by means of hand crank lever by throwing of the decompression lever at sufficient speed.
3. The engines is allowed and adjusted to pick the speed and run at rated speed, smoothly for few seconds at rated speed check speed by using the tachometer.
4. Record the time taken for 5cc fuel consumption at no load and the manometer reading on the panel board.
5. Then the engine is loaded electrically and takes time 5cc of fuel consumption and record the manometer reading and note pollution values from the pollution setup.
6. Repeat the experiment at different loads at different fuel injection pressures.160, 180 and 200 bars.
7. Again replace manifold with another inclination repeat above procedure.

GRAPHS

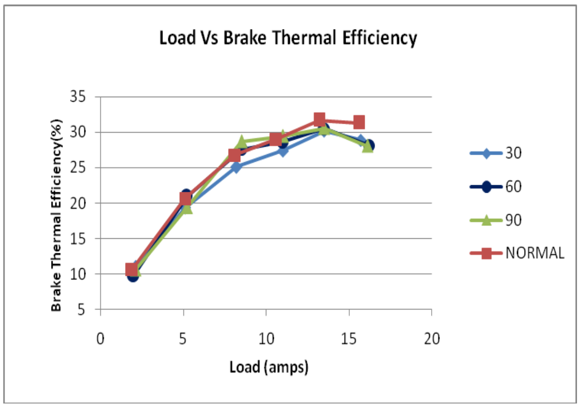
GRAPHS AT 160 BAR



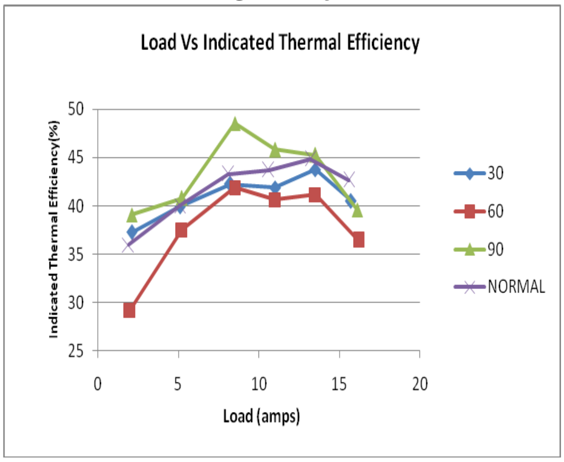
GRAPH 1



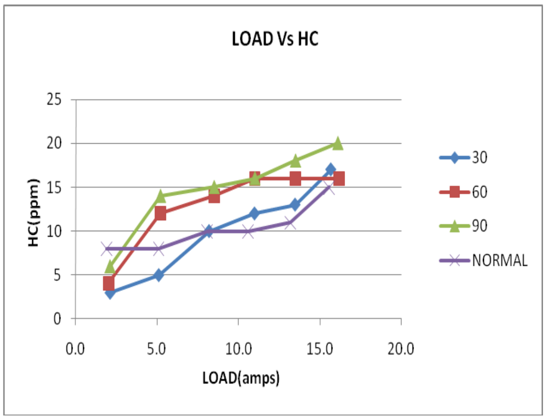
GRAPH 2



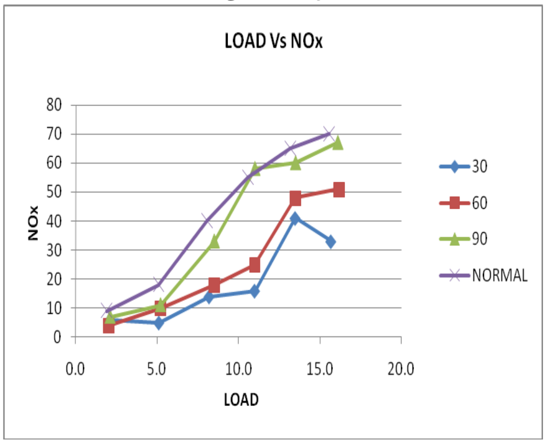
GRAPH 3



GRAPH 4



GRAPH 5



GRAPH 6

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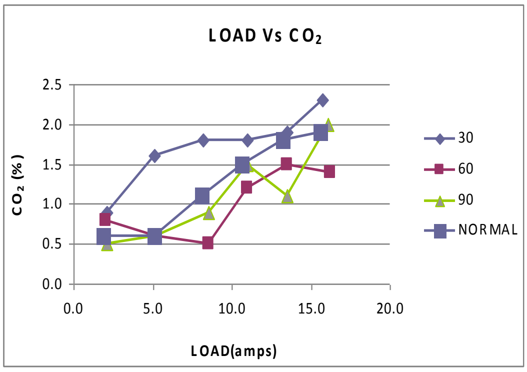
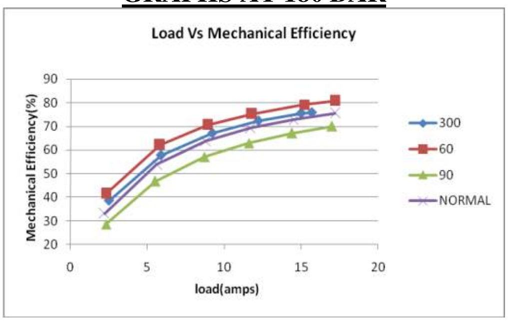
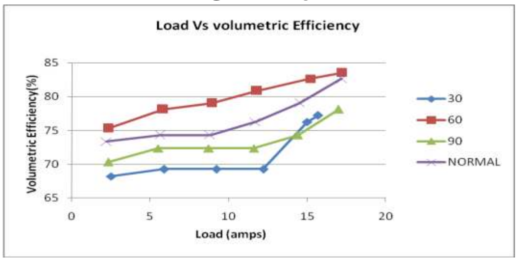


GRAPH 7

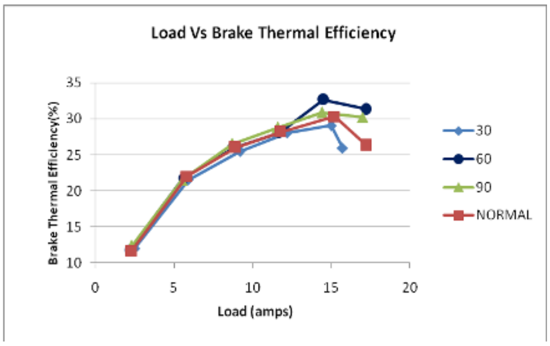
GRAPHS AT 180 BAR



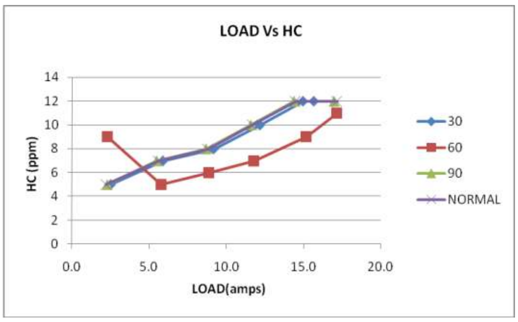
GRAPH 8



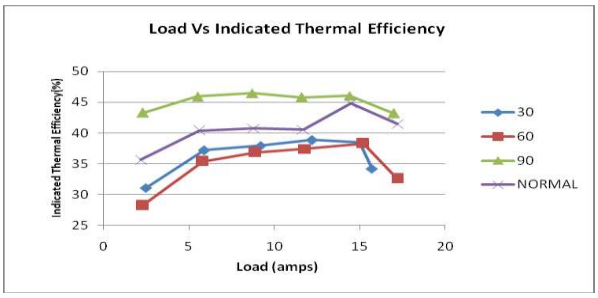
GRAPH 9



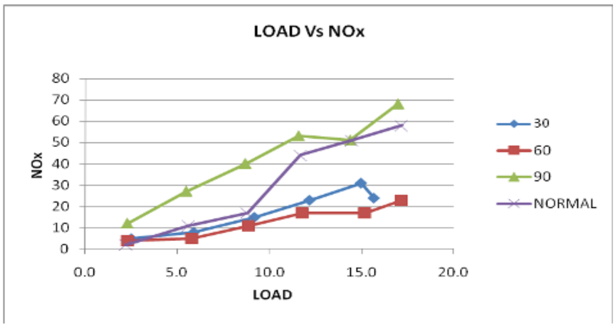
GRAPH 10



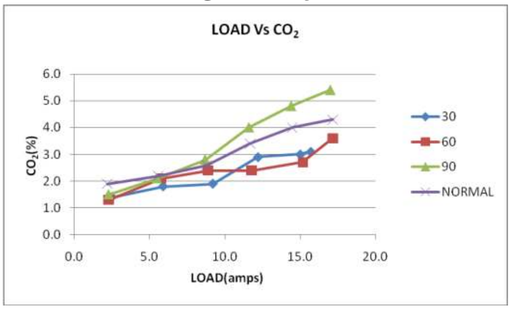
GRAPH 11



GRAPH 12

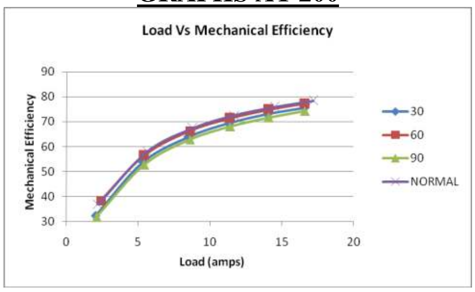


GRAPH 13

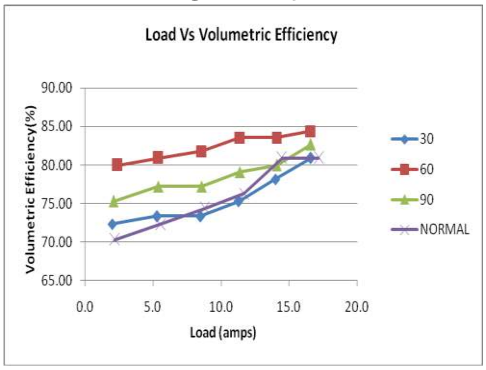


GRAPH 14

GRAPHS AT 200



GRAPH 15



GRAPH 16

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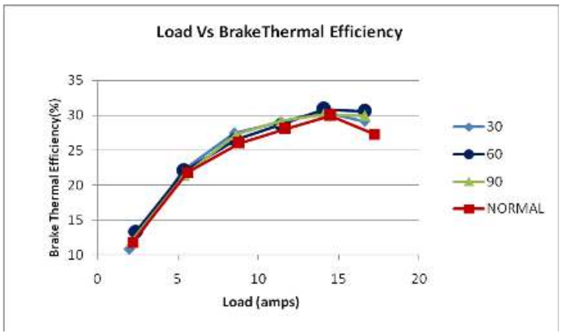
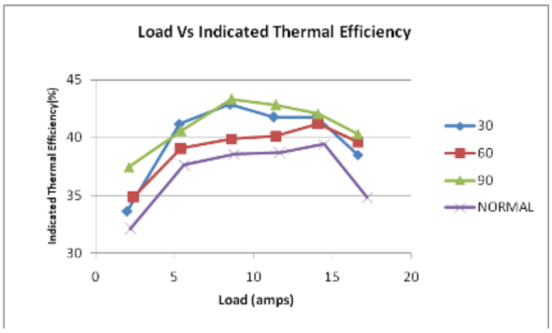
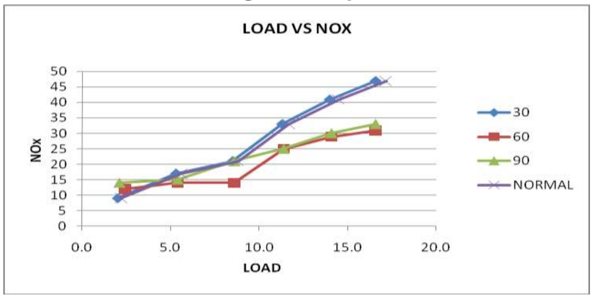


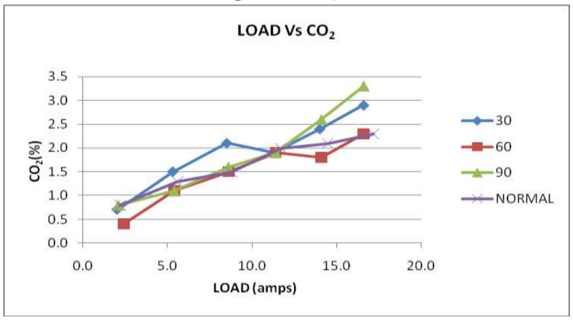
GRAPH 17



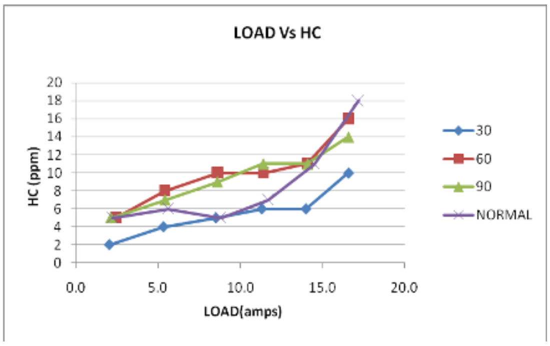
GRAPH 18



GRAPH 19



GRAPH 20 & GRAPH 21



Formulae used

1. Break horse power (BHP) = (V*I)/ (Eff of gen* 1000)

Where V= voltage in volts

I = load on the engine in amps

Efficiency of D.C generator =0.9

2. Total fuel consumption (TFC)=Measured fuel (5cc)*sp gravity*3600 Time taken in Seconds*1000

3. Friction power from the graph in between TFC and BP by WILLIAN'S LINE METHOD:- In this method total fuel consumption (TFC) vs. Brake power at constant speed is plotted and graph is extrapolated back to zero fuel consumption the point where the graph cuts the brake power axis is an indication of the friction power of the engine at that speed. This negative work represents the combined loss due to mechanical friction, pumping and blow by. This test is applicable to compression ignition engines.

4. Indicated power (IP) = BP + FP

Where BP = brake power

FP = friction power

5. Actual discharge (Q_{act}) = C_d * A * $(2gH)^{1/2}$

Where C_d = coefficient of discharge = 0.65

A = Area of orifice section = $\pi d^2/4$

d = diameter of orifice = 0.02M

H=manometer reading in meters in air = $(10*h_w)$/density of air

h_w = water column in meters.

Density of air (ρ_a) = atmospheric pressure. (R*T)

$\rho_a = \rho_a * 10/ (29.27*(273+T)$

6. Theoretical discharge (Q_{the}) = ($\pi/4 * D^2 * L*N/2)/60$

Where D = diameter of the cylinder in meters = 0.08m

L = stroke length = 0.11m

N = speed in R.P.M = 1500 rpm

7. Mechanical efficiency (η_{mech}) = BP/IP

8. Volumetric efficiency (η_{vol}) = Q_{act}/Q_{the}

9. Brake thermal efficiency (η_{bthe}) = BP/ (TFC* CV)

10. Indicated thermal efficiency (η_{ithe}) = IP/ (TFC * CV)

11. Air fuel ratio (A/F) = (W/TFC)*3600

Where weight of air drawn (W) = density of air *Q_{act}

TFC = Total fuel consumption

Sample calculations:-

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1. Break horse power (BHP) = (V*I)/ (Eff of gen* 1000)

= (5.4*269)/(0.9*1000)

= 1.488 kW

Where V= voltage in volts

I = load on the engine in amps

Efficiency of D.C generator =0.9

2. Total fuel consumption (TFC) = $\frac{\text{Measured fuel (5cc)*sp gravity*3600}}{\text{Time taken in Seconds*1000}}$

= $\frac{5*0.836*3600}{}$

= 0.579 kg/hr.

3. Friction power from the graph in between TFC and BP by willian's line method

FHP = 1.15 kW

4. Indicated power (IP) = BP + FP

= 1.488 + 1.15

= 2.638 kW

Where BP = brake power = 1.488 kW

FP = friction power = 1.15kW

5. Actual discharge (Q_{act}) = C_d * A * $(2gH)^{1/2}$

= $(0.65*3.14*10^4)*(2*9.81*38.27)^{1/2}$

= $5.592*10^{-3}$ m^3/sec

Where C_d = coefficient of discharge = 0.65

A = Area of orifice section = $\pi d^2/4$

= $(\pi*0.02^2)/4 = 3.14*10^{-4}$

d = diameter of orifice = 0.02M

H = manometer reading in meters in air = $(10*h_w)$/density of air = (10*4.5)/1.176

= 39.27

h_w = water column in meters.

Density of air (ρ_a) = atmospheric pressure. (R*T)

$\rho_a = \rho_a * 10/ (29.27*(273+T)$

= 1.176

6. Theoretical discharge (Q_{the}) = $(\pi/4 * D^2 * L*N/2)/60$

= (π/4 *0.082*0.11*1500/2)60

= $6.9115 *10^{-3}$ m^3/sec

Where D = diameter of the cylinder in meters

= 0.08m

L = stroke length = 0.11m

N = speed in R.P.M = 1500 rpm

7. Mechanical efficiency (η_{mech}) = BP/IP

= 1.488/2.638

= 56.41%

8. Volumetric efficiency (η_{vol}) = Q_{act}/Q_{the}

= $(5.592*10^{-3})/(6.9115*10^{-3})$

= 80.88 %

9. Brake thermal efficiency (η_{bthe}) = BP/ (TFC* CV)

= 1.488/((0.579/3600)*42000)

= 22.04 %

10. Indicated thermal efficiency (η_{ithe}) = IP/ (TFC * CV)

= 2.638/((0.579/3600)*42000)

= 39.07 %

11. Air fuel ratio (A/F) = (W/TFC)*3600

= $((6.599*10^{-3})/0.579)*3600$

= 40.80

Where weight of air drawn (W) = density of air *Q_{act}

= $1.173* 5.592*10^{-3}$

= $6.559*10^{-3}$

Maximum load:-

Maximum power = $\frac{\text{voltage * Max load}}{\text{Generator efficiency*1000}}$

3.7285 = $\frac{\text{220* Max load}}{0.9*1000}$

Maximum load = $\frac{3.7285*0.9*1000}{220}$ = 15.252amps

Where Maximum Power = 3.7285 kW

Generator Voltage = 220 volts

Generator Efficiency = 90%

IX. RESULTS

- From the graphs we can observer that we had better mechanical efficiency at 60^0 manifold inclination at 160 bar,60^0 manifold inclination at 180 bar and normal manifold inclination at 200 bar . among them we had best mechanical efficiency of inlet Manifold with inclination of 60^0 at 180 bar .
- From the graphs we can observer that we had better volumertic efficiency of intake manifold at 60^0 manifold inclination at 160 bar,60^0 manifold inclination at 180 bar and 60^0 manifold inclination at 200 bar . among them we had best volumetric

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efficiency of inlet Manifold with inclination of 60^0 at 200 bar .

- From the graphs we can observer that we had better brake thermal efficiency of intake manifold at normal manifold inclination at 160 bar,60^0 manifold inclination at 180 bar and 60^0 manifold inclination at 200 bar . Among them we had best brake thermal efficiency of inlet Manifold with inclination of 60^0 at 180 bar .
- From the graphs we can observer that we had better indicated thermal efficiency at normal intake manifold inclination at 160 bar,60^0 intake manifold inclination at 180 bar and 60^0 intake manifold inclination at 200 bar . Among them we had best indicated thermal efficiency of inlet Manifold with inclination of 90^0 at 180 bar .
- From the graphs we can observer that we had less HC emissions at normal intake manifold inclination at 160 bar,60^0 intake manifold inclination at 180 bar and 30^0 intake manifold inclination at 200 bar . Among them we had very less emissions at intake manifold with inclination of 30^0 at 200 bar .
- From the graphs we can observer that we had less NO_x emissions at 30^0 intake manifold inclination at 160 bar,60^0 intake manifold inclination at 180 bar, 60^0 intake manifold inclination at 200 bar and 90^0intake manifold inclnation at 200 bar. . Among them we had very less emissions at intake manifold with inclination of 60^0 at 180 bar .
- From the graphs we can observer that we had less CO_2 emissions at 60^0 intake manifold inclination at 160 bar,30^0 intake manifold inclination at 180 bar , 60^0 intake manifold inclination at 200 bar ,normal manifold inclnation at 200 bar. Among them we had very less emissions at intake manifold with inclination of 60^0 at 160 bar .

X. CONCLUSION

- In cylinder flow structure is greatly influenced by the intake manifold inclination.
- It is found that at 60^0 intakes manifold inclination, at 180bar gives the maximum brake thermal efficiency.
- This work improves both performance and fuel economy.
- By varying the manifold inclination we get better performance than normal one.
- By increasing fuel injection pressure, pollution levels reduce due to complete combustion of fuel.
- Emissions are reduced at 200 bar with different manifold inclinations compared to other pressures.
- Finally, it is concluded that the information obtained in this investigation is very much useful in reduction of pollution and increasing the performance of the engine by varying the manifold inclination of the modern I.C engines.

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