EXPERIMENTAL STUDY OF WASTE HEAT RECOVERY TECHNIQUE TO INCREASE EFFICIENCY AND TO DECREASE HAZARDOUS EMISSIONS IN CI ENGINE

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Abstract:

The quest for higher efficiencies has spurred the innovation of energy efficient technologies such as waste heat recovery. The recovery and utilization of waste heat not only conserves fuel, usually fossil fuel, but also reduces the amount of waste heat and greenhouse gases dumped to the environment. CI engines exchange approximately 30-40% of heat generated in the process of fuel combustion into useful mechanical work with the current available technology. The remaining heat is emitted to the environment through the exhaust gases (35%) and the engine cooling systems (31%). Experiment show that the boiling point of fossil fuels can be reduced by 50% by preheating. Large part of waste heat exhaust gases can be recovered by vaporizing the fuel through a small heat exchanger. This technical paper shows the experimental results for significant reduction of engine fuel consumption and hazardous emissions could be attained by recovering of exhaust heat by using self made Heat Exchanger. One of the most important issues is to develop an efficient heat exchanger which provides optimal recovery of heat from exhaust gases.

Key words: Emissions, Efficiency, Fuel consumption, Heat Exchanger, Waste Heat Recovery.

I: INTRODUCTION

Energy is an underlying driver of economic growth and social development. Human consumption of energy in the form of fossil fuels, primarily in developed countries, is altering the Earth's climate and has been a matter of great concern. While evidence suggests a need for both demand reduction through energy efficient engine and less harmful emissions.

The internal combustion engines have already become an indispensable and integral part of our present day life style, particularly in the transportation and agricultural sectors unfortunately the survival of these engines has, of late, been threatened due to the problems of fuel crisis and environmental pollution. Therefore, to sustain the present growth rate of civilization, a nonreplicable, clean fuel must be expeditiously sought.

Internal combustion engine convert about 25% to 35% of the chemical energy contained in the fuel into the mechanical energy. About 35% of the heat generated is lost to the cooling medium and the remaining heat is dissipated through the exhaust gases and other radiation losses. During the process of combustion the cylinder gas temperature reaches quite a higher value. A considerable amount of heat is transferred into walls of the combustion chamber. Therefore it is necessary to provide proper cooling specially to walls of the combustion chamber. Inside surface temperature of the cylinder walls will be maintained in ranges, which provides correct clearance between the parts and which promotes vaporization of fuel. After the combustion, heat is transferred to the walls of the combustion chamber, which is continuously removed (almost 30 to 35% of the total heat) by employing a cooling system [1].

While recent improvements in diesel engine design and calibration have greatly reduced both NOx and smoke emissions [2]–[4], there are still many alternatives being researched to improve the engine-out emission further. In recent years, a lot of attention has been focused on air pollution caused by automotive engines. Diesel engines have been particularly targeted for their production of oxides of nitrogen (NOx) and smoke emissions. NOx is formed at high rates when temperatures are high, whereas smoke is formed in fuel rich regions within the combustion chamber [5], [6]. Hence, it is essential to keep the peak cylinder temperature low in order to minimize NOx emission and also to allow for better fuel–air mixing thereby, reducing the smoke emission.

Homogeneous charge compression ignition (HCCI) is a combustion concept that constitutes a valid approach to achieve high efficiencies and low nitrogen oxides and particulate emissions in comparison with traditional compression ignition (CI) direct injection (DI) engines [7]. Although HCCI combustion was demonstrated about 20 years ago, only the recent advances made in airflow, fuel and exhaust gas recirculation (EGR) electronic control have made it feasible. HCCI has been successfully applied both to spark ignition (SI) and compression ignition (CI) engines, and proved to be fuel flexible since it has been achieved with gaseous fuels such as propane or natural gas, as well as liquid fuels like traditional gasoline or diesel fuels. The HCCI process operates on the principle of having a lean, premixed, homogeneous charge that reacts and burns volumetrically throughout the cylinder as it is compressed by the piston [8].

From the previous research works [9]-[14] it is noticed that early injection, late injection and port fuel injection systems like Air assisted port injection with DI system (PCCI-DI) were used. It is well known that EGR [15]-[18] is a useful way to vary the cylinder gas temperature and the ignition timing could be delayed. In addition to that high levels of EGR and reduced compression ratio were demonstrated for simultaneous and substantial reduction of NOx and smoke emissions. In the above detailed methods the mixture was partially homogeneous. Hence the present system was developed to prepare a homogeneous mixture.

The present work deals with the study of performance, combustion and emission characteristics of diesel vapour combustion process in a DI diesel engine with mixture formation in heat exchanger- accumulator mechanism. In this investigation a stationary four stroke, single cylinder, direct injection diesel engine was modified to operate in dual mode- without diesel vapor mixture and with diesel vapor mixture formation of fuel. A heat exchanger-accumulator mechanism was used to vaporize the diesel fuel and catalytic cracking. It was mounted in the intake system to prepare the homogenous diesel vapour-air mixture. The experiments were conducted with diesel vapour induction with different injection timings. Experimental results obtained are compared with the base line readings. The results show that through this approach simultaneous and substantial reduction of NOx, fuel consumption and smoke emission can be achieved. Also it increases the efficiency of the engine.

II: EXPERIMENTAL WORK

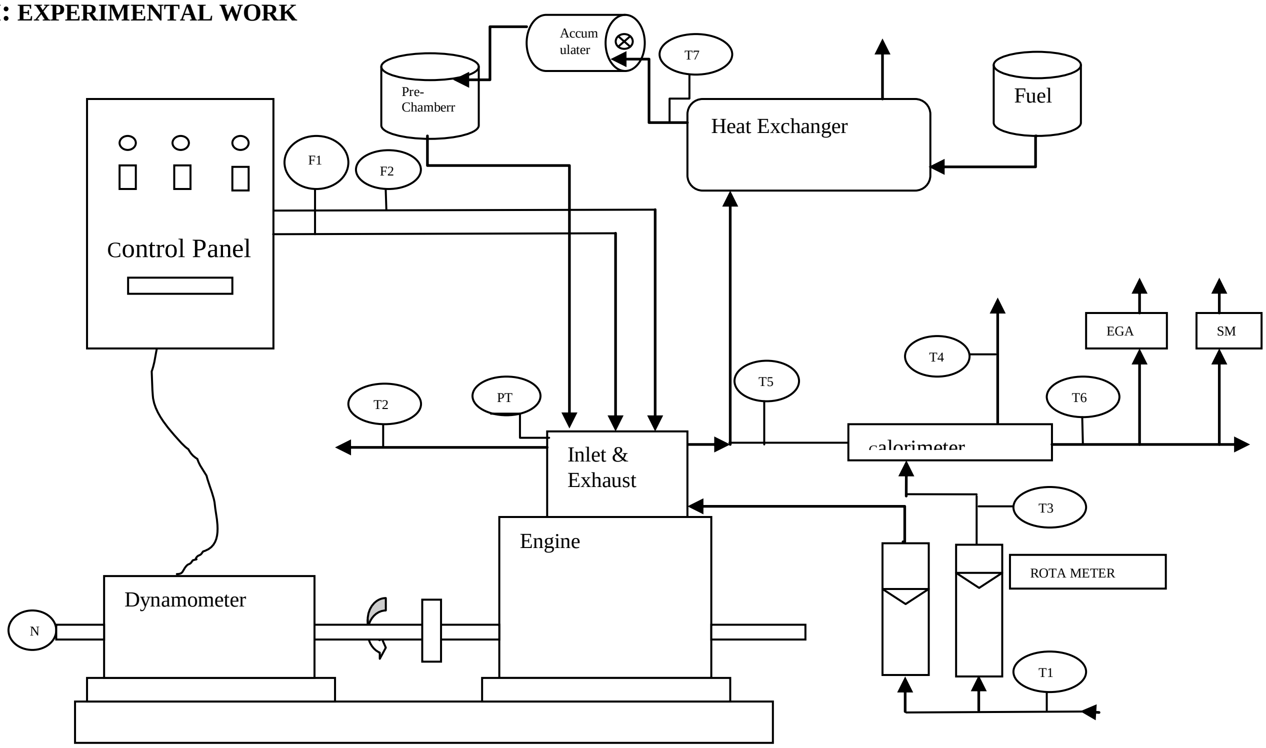


Figure 1: Schematic of the experimental set-up

T1, T3	Inlet Water Temperature	F1	Fuel Flow DP (Differential Pressure) unit
T2	Outlet Engine Jacket Water Temperature	F2	Air Intake DP unit
T4	Outlet Calorimeter Water Temperature	PT	Pressure Transducer
T5	Exhaust Gas Temperature before Calorimeter	N	RPM Decoder
T6	Exhaust Gas Temperature after Calorimeter	EGA	Exhaust Gas Analyzer (5 gas)
T7	Vaporised fuel Temperature after HE	SM	Smoke meter

Experiments were conducted on a modified single-cylinder, water-cooled, direct injection diesel engine developing 5.2 kW at 1500 rpm. Test rig is provided with necessary equipment and instruments for combustion pressure and crank angle measurements with accuracy. The schematic diagram of the experimental set-up is shown in Figure 1 and Table 1 shows the test rig specifications.

Engine	4-stroke,single cylinder, constant speed, water cooled Diesel Engine
BHP	5.2 kW @ 1500 rpm
Bore x Stroke	87.5 x 110 mm
Compression Ratio	17.5 : 1
Connecting rod	234 mm
Dynamometer type	Eddy current with loading unit
Load Measurement	Strain gauge load cell
Water flow	Rota meter
Fuel and air	Differential pressure unit
Speed measurement	Rotary Encoder

Table 1:- Engine Test Rig Specification

Time taken for fuel consumption was measured with the help of a digital stopwatch with rotameter. Exhaust emission from the engine was measured with the help of gas analyzer, was used to measure the NOx (ppm), CO_2 (%) and UBHC (ppm) emissions in the exhaust. Smoke intensity was measured with the help of a Smoke meter. Fig. 2 shows the schematic of diesel fuel vaporizing heat exchanger.

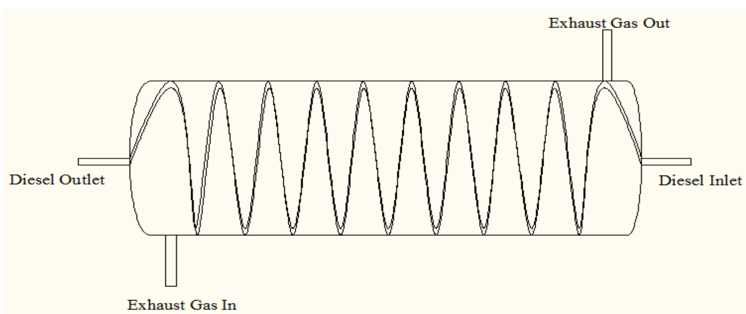


Figure 2: Schematic of Heat Exchanger

The Table 2 represents Specifications of the Heat exchanger.

Sr. No.	Particular	Dimension in mm
1	Diameter	420
2	Length	280
3	Thickness of the cylinder	2
4	Diameter of the copper pipe used	12
5	Length of the copper pipe	2000

Table 2:- Specification of Heat Exchanger

A liquid-gas accumulator is totally covered by the exhaust gases flow and inside the accumulator using catalysts. This is low temperature activating and material of the accumulator is copper which also acting as a catalysts. Both the end of accumulator have one way flapper check valve. One end is connected to the pre-chamber and another end is connected to the heat exchanger. The accumulator has 20mm inner shell diameter and 50mm of outer shell diameter. The accumulator was mounted in the intake manifold system to supply the diesel fuel in vapour form in the intake manifold and it was mixed with the air. The Figure 3 is the real picture of heat exchanger.

Figure 3: Real Picture of Heat Exchanger

III: PERFORMANCE CHARACTERISTICS

1) <u>Brake Thermal Efficiency (BTE):</u>

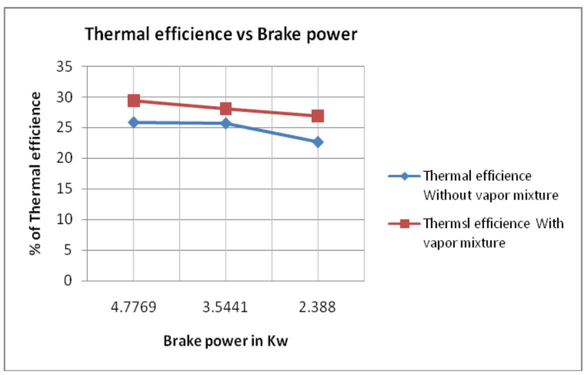


Figure 4: BTE variation of diesel and blend of diesel vapor mixture at 27^0 crank angle IT

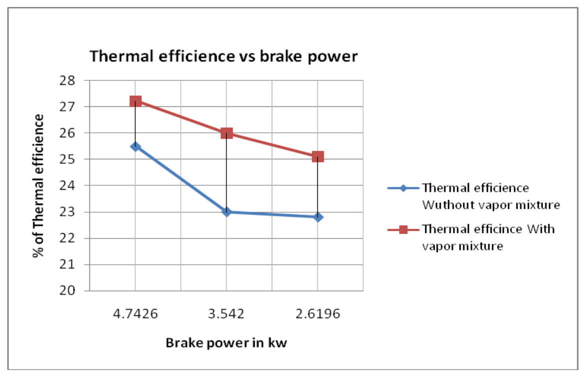


Figure 5: BTE variation of diesel and blend of diesel vapor mixture at 30^0 crank angle IT

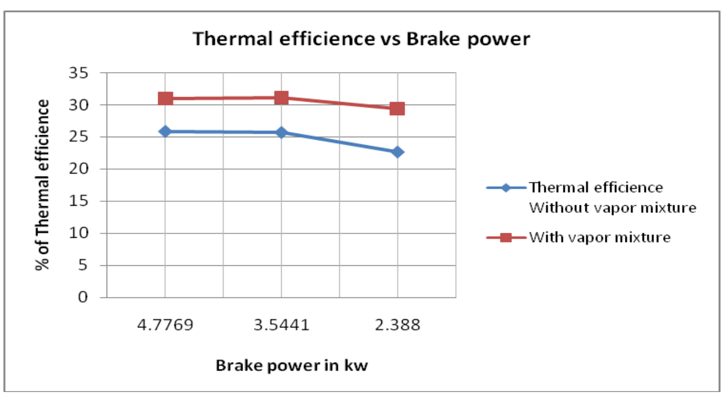


Figure 6: BTE variation of diesel and blend of diesel vapor mixture at 27^0 crank angle IT with DPS

Figure 4 to Figure 6 shows plotted graphs for the variations of Brake Thermal Efficiency with loads at different Injection Timing (IT) for without diesel vapour mixture and with diesel vapour mixture condition at constant flow rate of EGR (4%). It shown in Figure 4 that for 27^0 Injection Timing with respectively 100%, 75% and 50% loads the percentile increment of the brake thermal efficiency for 100% load is 15%, for 75% load is 13% and 50% load is 10% and for 30^0 Injection Timing shown in Figure 5, is 13.6%, 12.12% and 10.9% respectively with loads 100%, 75% and 50%. The third observation of the Brake Thermal Efficiency on 27^0 but the mixing of diesel vapour mixture fuel with air after pre-chamber Direct Port Supply

(DPS), thus percentile increment of the brake thermal efficiency with loads is 19.8%, 12.17%, and 14% respectively shown in Figure 6.

The trend of the graph for all injection timings remains same. The percentage increment of the brake thermal efficiency for 27^0 IT with DPS is more because of mixing of the charge potential is better than to other thus perfect combustion inside the chamber and reduce physical delay for combustion, thus increasing the peak pressure and less fuel consumption.

2) <u>Brake Specific Fuel Consumption (BSFC):</u>

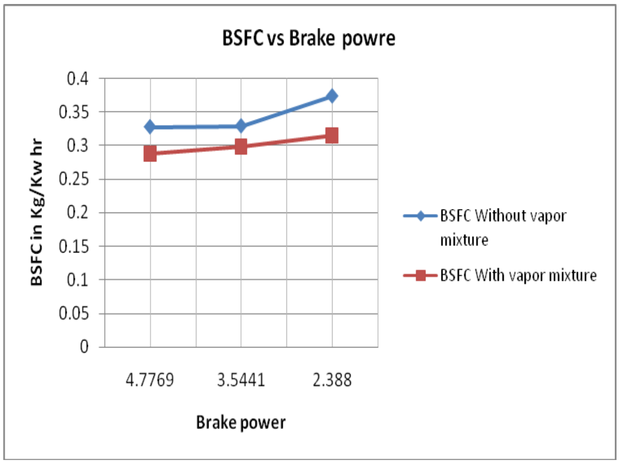


Figure 7: BSFC variation of diesel and blend of diesel vapor mixture at 27^0 crank angle IT

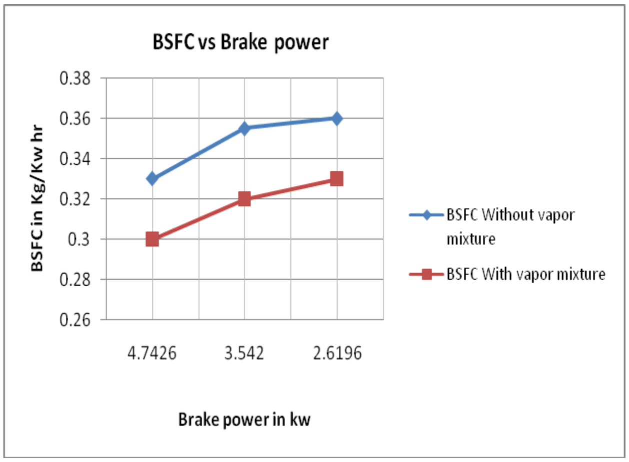


Figure 8: BSFC variation of diesel and blend of diesel vapor mixture at 30^0 crank angle IT

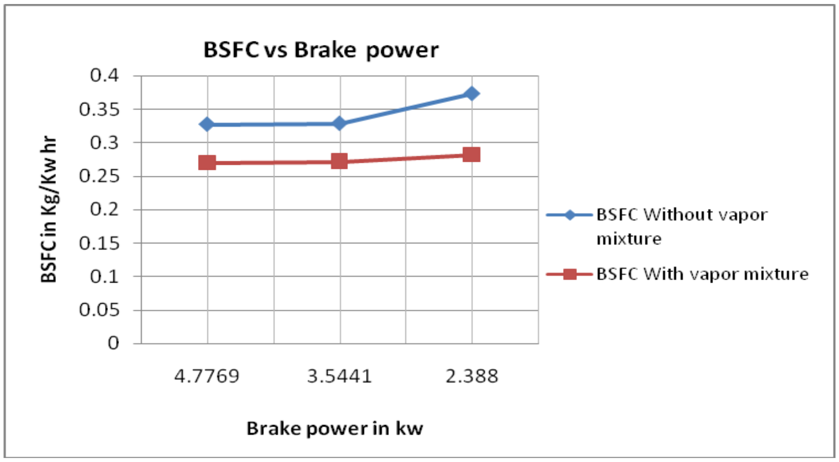
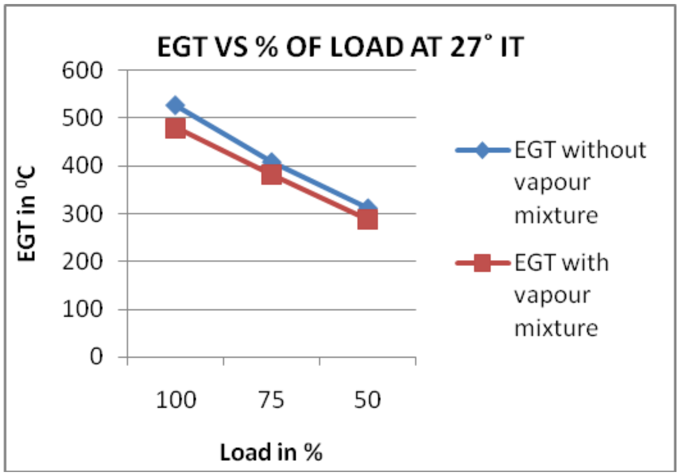
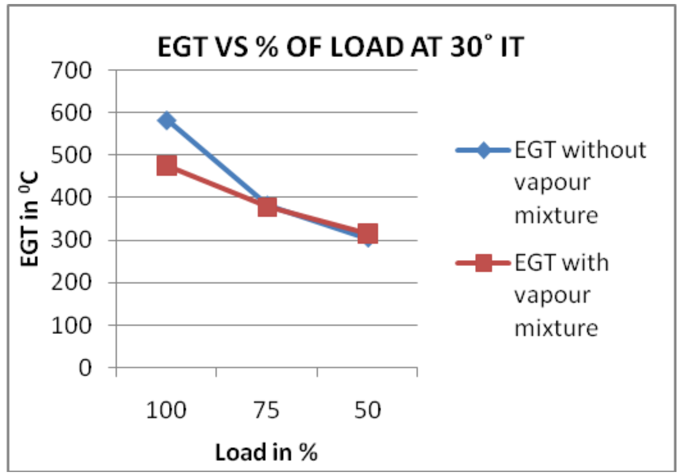


Figure 9: BSFC variation of diesel and blend of diesel vapor mixture at 27^0 crank angle IT with DPS

Observations provided the variation of brake specific fuel consumption as a function of load for diesel vapour mixture at different injection timing. The total bake specific fuel consumption estimated from the brake power output of the engine and it measures mass flow rate of the fuel. The total brake specific fuel consumption affected by mixing of the diesel vapour mixture as shown in Figure 7, 8 and 9 respectively. Percentage of load under dual fuel operation, the brake specific fuel consumption is comparable with normal diesel fuel operation at different injection timing. It is observed that as the % of load increases BSFC decreases by 5.44% for 27^0IT, 15% for 30^0 IT and 14% for 27^0 IT with DPS at 50% to 100% load. According to the trend of graph decrement of the brake specific fuel consumption is 30^0 and 27^0 with DPS, thus 30^0 is optimum condition for BSFC.

3) Exhaust Gas Temperature:





vapor mixture at 27^0 crank angle IT

vapor mixture at 30^0 crank angle IT

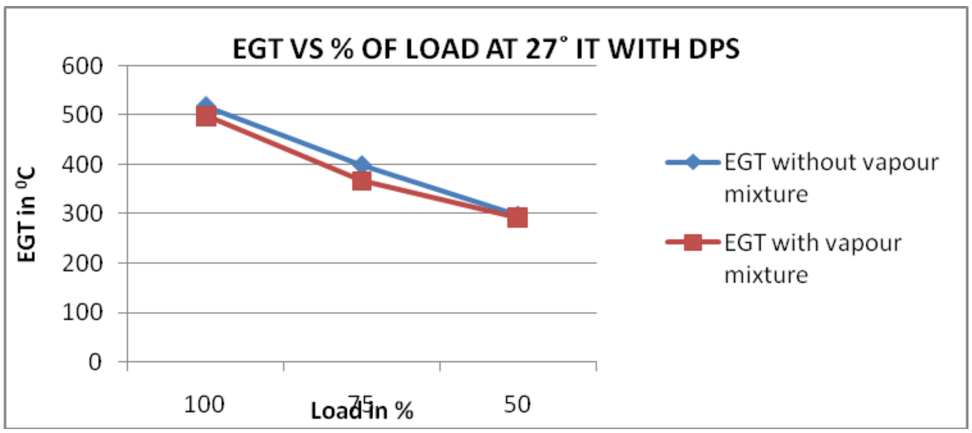


Figure 12: EGT variation of diesel and blend of diesel vapor mixture at 27^0 crank angle IT with DPS

It was found that exhaust gas temperature increase with the load. The effects of the exhaust gas temperature are shown in Figure 10, 11 and 12. From those charts it was also found that exhaust temperature decreases with diesel vapour mixture and EGR for all loads at different injection timing. Because of low temperature nitrogen oxide reduces and heat loss of the engine decreases.

Iv: emission characteristics

1) NO_X Emission:

The variation of NOx with load at different injection Timing for the diesel and diesel vapour mixture by varying load is shown in Figure 13, 14 and 15 respectively. It is observed that the NOx emission for standard diesel fuel has increases from 290ppm to 899ppm in 27^0 injection timing with load vary from 50% to 100% and for diesel vapour mixture NOx is increasing from 230ppm to 455ppm. It reduces 50% in comparison to standard diesel fuel. Same for 30^0 IT it is 679ppm to 1790 ppm for normal diesel and for diesel vapor mixture it is 594.25ppm to 880ppm. For $27^0$with DPS it is 483.75 to 894 for standard fuel and for diesel vapor mixture it is 305ppm to 635ppm respectively with 50% to 100% loads. NOx reduces by reducing the combustion temperature and reaching homogeneous charge of diesel vapor fuel and air inside the chamber and also affected by the 4.0% mixing of the EGR along with vapor mixture.

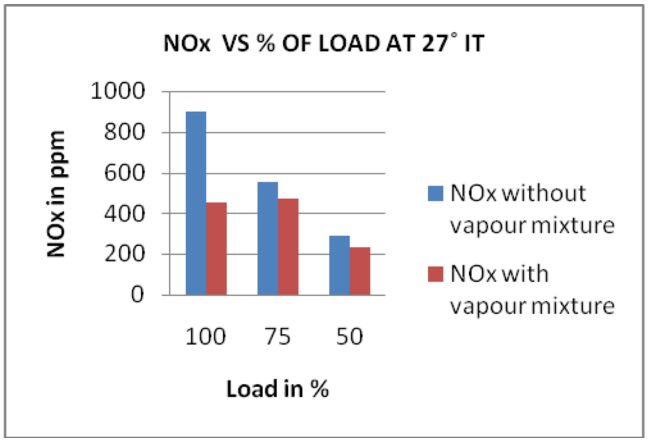


Figure 13: NOx variation of diesel and blend of diesel vapor mixture at 27^0 crank angle IT

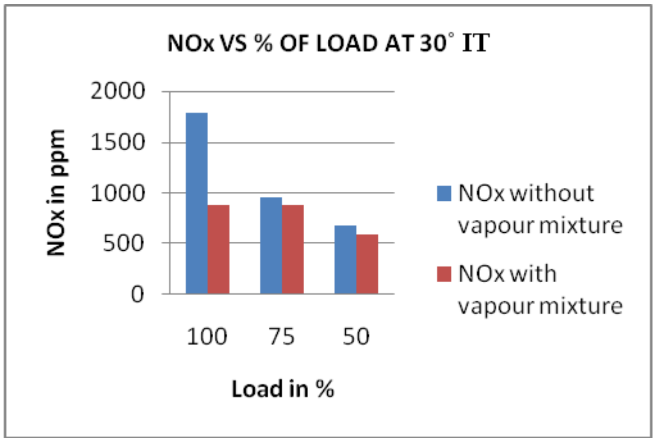


Figure 14: NOx variation of diesel and blend of diesel vapor mixture at 30^0 crank angle IT

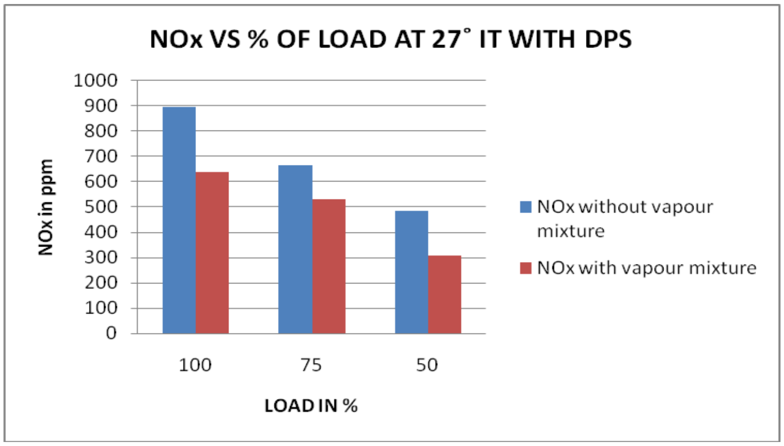


Figure 15: NOx variation of diesel and blend of diesel vapor mixture at 27^0 crank angle IT with DPS

2) <u>Carbon Monoxide (CO) Emission:</u>

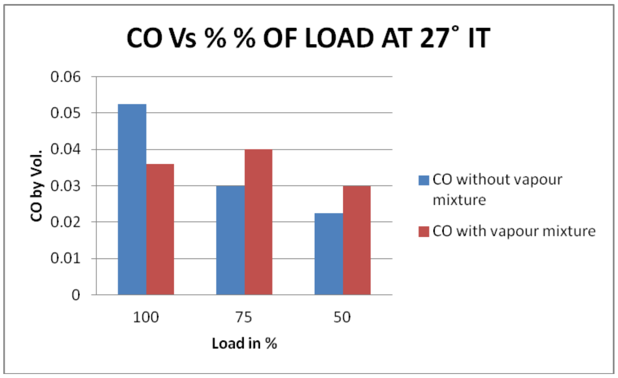


Figure 16: CO variation of diesel and blend of diesel vapor mixture at 27^0 crank angle IT

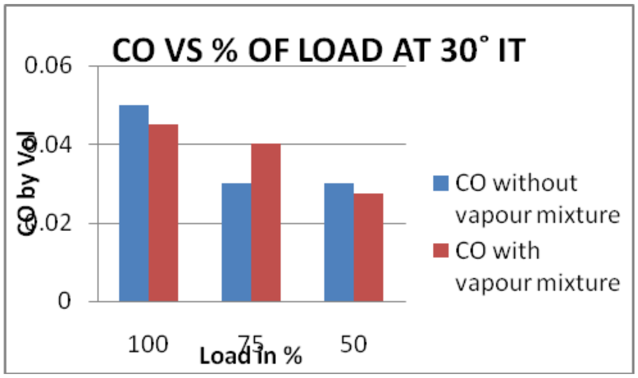


Figure 17: CO variation of diesel and blend of diesel vapor mixture at 30^0 crank angle IT

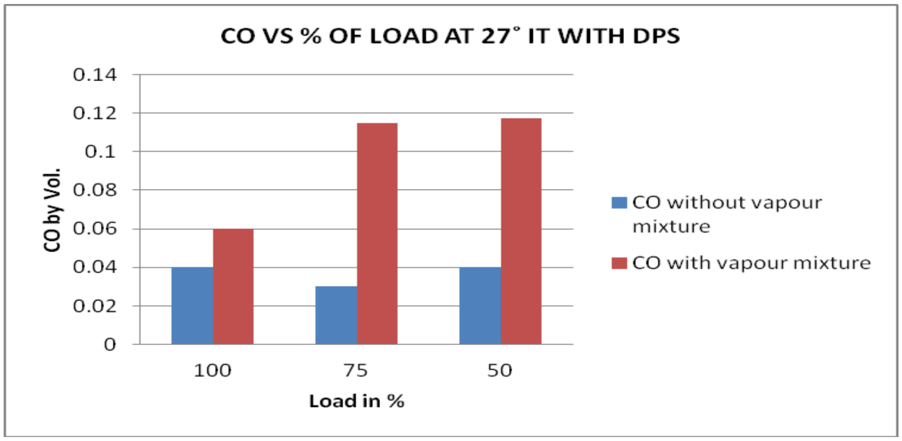


Figure 18: CO variation of diesel and blend of diesel vapor mixture at 27^0 crank angle IT with DPS

The CO emission with load at different injection timing is shown in Fig. 16, 17 and 18. It increases from 50% to 100% load for diesel fuel. It is observed that CO emissions are lower for diesel vapour mixture at 27^0 and 30^0 but its increasing in 27^0 with DPS. The CO emission reduces 20% at full load in 27^0 and 30^0 but it increases 30% in 27^0 DPS. This may be because of reducing the UHC according to increasing load and by mixing of the 4% of EGR along with diesel vapor mixture. It is also affected by lean mixture in combustion.

3) Un-burnt Hydro Carbon Emissions

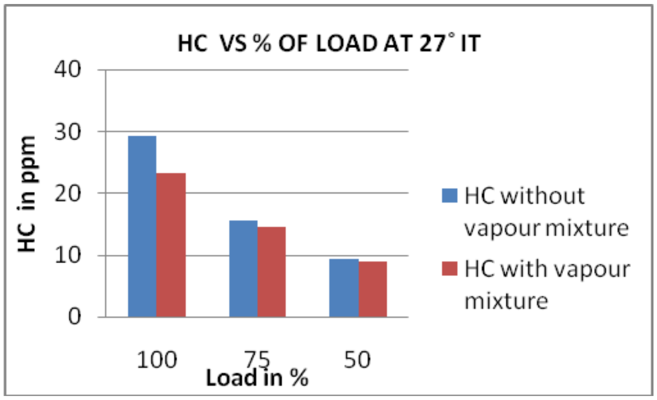


Figure 19: HC variation of diesel and blend of diesel vapor mixture at 27^0 crank angle IT

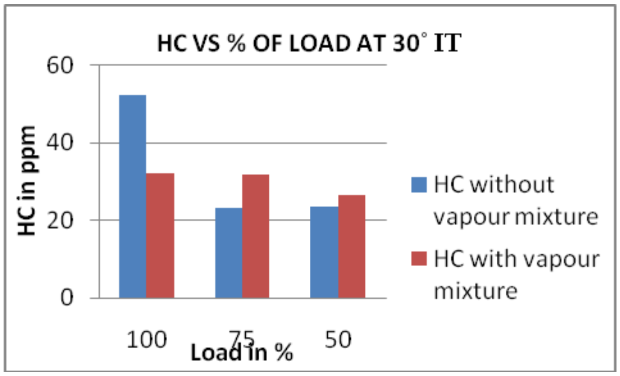


Figure 20: HC variation of diesel and blend of diesel vapor mixture at 30^0 crank angle IT

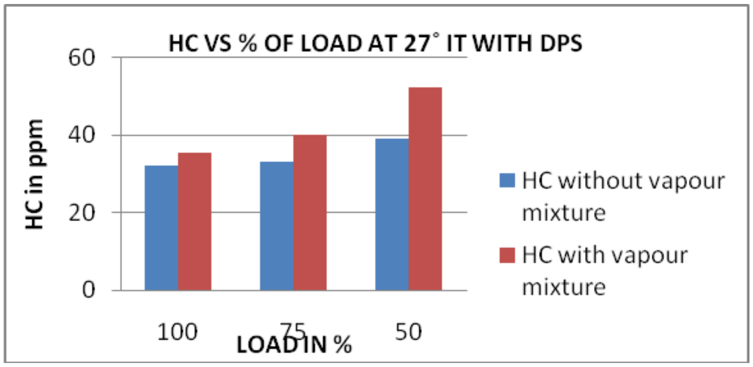


Figure 21: HC variation of diesel and blend of diesel vapor mixture at 27^0 crank angle IT with DPS

The variation of the HC emission with load at different injection timing as shown is Figure 19, 20 and 21. For standard diesel fuel HC emission at different injection timing 27^0,30^0 and 27^0 DPS is 9ppm to 29.25ppm, 23.5ppm to 52.5ppm and 39.2ppm to 32.5ppm respectively, as for the diesel vapour mixture it reduced in 27^0 and 30^0 injection timing but for 27^0 with DPS its increase with 50% load condition. This may because of mixing of EGR and constant flow and low temperature of the combustion chamber.

V: conclusion

The large amount waste heat is through exhaust which has high temperature is utilized to vaporizing the diesel fuel and mixed with intake air thus mixture getting homogeneity. This homogenous charge in compression ignition engine increases the combustion response and reducing the physical delay period so getting increment in combustion pressure develops more power.

1. Vaporizing of fuel by exhaust gas heat that means specific heat of the fuel is getting increases so less fuel consumption and perfect combustion carried out because the fuel gets ignited at multipoint, and thus no flame propagation in combustion chamber which causes less emission.
2. Brake thermal efficiency in the case of diesel vapor mixture, percentile increment of engine thermal efficiency is 15% at full load 27^0 injection timing, 13% at 30^0IT and 19.8% for 27^0DPS. In case of 27^0 DPS thermal efficiency is high with compared to other cases, it may because of perfect mixing of homogeneous charge and air fuel ratio.
3. Brake specific fuel consumption (BSFC):
   It is observed that as the % of load increases BSFC decreases by 5.44% for 20^0, 15% for 30^0 and 14% for 27^0 with DPS for 50% load to 100% load range. According to the graph trend it shows the decrement of brake specific fuel consumption for 30^0 and 27^0 with DPS. Thus 30^0 IT is optimum condition for BSFC.
4. NOx Emissions was reduced with diesel vapor mixture. It is because of reduced combustion temperature. It has reduced up to 50% at full load due to mixing 2.2% of EGR. At $27^0$with DPS the NOx is 894ppm to 635ppm and at 27^0 and 30^0 it is 899ppm to 455ppm, 1790pmm to 880ppm respectively.
5. Carbon monoxide (CO) is lower for diesel vapour mixture at 27^0 and 30^0 but its increases in 27^0 with DPS. The CO emission reduces about 20% at full load in 27^0 and 30^0 but it increases 30% in 27^0 DPS. This is because of insufficient oxygen inside the combustion chamber.
6. Here the opacity increases from 9% for 30^0, 10.34% for 27^0 and 13% for 27^0DPS injection timing. This is because EGR reduces availability of oxygen for combustion of fuel which results in incomplete combustion and increases formation PM. The smoke opacity for diesel vapour mixture is higher than diesel fuel.
7. The exhaust gas temperature is reduced because of reducing the NOx emission and cylinder pressure is increases 2 to 4 bar. Thus work output is increases and this results increased thermal efficiency.
8. Energy captured by this modification is 308.349KJ/hr from 20-30% exhaust gases.

Thus the advantage of diesel vapors combustion with the recovering the waste exhaust gas heat was utilized to get the benefits of low emissions and saving in fuel consumption through high efficiency at different load conditions.

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