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EVALUATION OF EMISSION AND EFFICIENCY OF SINGLE CYLINDER FOUR – STROKE DIESEL ENGINE USING EXHAUST GAS RECIRCULATION PROCESS

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ABSTRACT

In recent days, internal combustion (IC) engines are playing an important role in the field of automobiles due to their advantages. Still, the fuel consumption and thermal efficiency are the factors defining the performance of the engine. In an IC engine, homogeneous charge refers to the complete mixture of fuel (petrol or diesel) and air entering the chamber. In an ideal world, this would imply complete dispersion of the atomized fuel by air. This work utilizes the four-stroke diesel engine for the experimental purpose to evaluate the efficiency of the engine and also the emission level. The efficiency of the engine is enhanced by using exhaust gas recirculation (EGR) process. The improvement of engine efficiency is performed by minimizing the throttling losses, reducing the heat rejection, and decreasing the molecular separation with the help of EGR process. The experimental setup includes a heat exchanger at the engine's manifold, and when the heat exchanger's fresh charge and outlet pipes come into contact, heat transfer can occur. The experimental results proved that the EGR process is enhancing the brake thermal efficiency and reduced the fuel consumption ratio. Moreover, the mechanical efficiency of the engine also increases and reduces the emission of carbon monoxide (CO) from the engine.

***Keywords**:* IC engine; Heat exchanger; Exhaust gas recirculation; Pre-heater; Thermal efficiency; Exhaust gas; Fresh charge; Fuel consumption

I. INTRODUCTION

One of the most interesting challenges in the automotive industry is to reduce emissions and increase the performance of engines by compromising between both of two. EGR (exhaust gas recirculation) is not a new technique but has a lot of scope to reduce the emission and performance improvements [1].

The exhaust residual gases and effective release energy are known as sensitive variables in ignition engines, and they have an impact on engine performance. Through the exhaust valve port, some internal exhaust residual gases are returned to the cylinder This leftover gas will combine with the fresh air–fuel combination in the intake, affecting combustion stability, charge mass, flame speed, and hazardous product emissions. A proven approach for lowering NOx emissions and PM emissions from compression ignition engines exhaust gas recirculation was used in spark ignition engines to enhance thermal efficiency and minimize NOx emissions.. The HCNG fuel combination is an excellent choice for reducing carbon emissions and NOx emissions in internal combustion engines. The engine performance (brake thermal efficiency, brake specific fuel consumption, in-cylinder pressure, heat released rate, mass fraction burned) and emissions (NOx, CO, HC, and CH4 emissions) of hydrogen enriched compressed natural gas in spark ignition at various engine speeds, manifold absolute pressures, and ignition timings under lean burn conditions have been optimized . The quantity of charge that can be inducted into the cylinder determines the power output of an internal combustion engine. Basic design and performance parameters of internal combustion engines' are Mean Piston Speed, Specific Power Output, Specific Fuel Consumption, Air-Fuel Ratio, and Fuel Calorific Value [2-3]. For reducing NOX emissions in internal combustion engines EGR is a promising technique but we have to compromise with bsfc (basic specific fuel consumption)and particulate emission. It is more significant in the case of Diesel engines [4,5,7].

EGR influences the performance under different conditions such as stoichiometric ratio ignition delay, speed of piston and mean effective pressure. The excess warmth is removed to the environment through fumes gases and motor cooling frameworks, resulting in entropy rise and genuine ecological contamination. As a result, the main thermal losses that occur in the use of waste warmth in useful work are required in the engine in the form of losses through exhaust gases and incomplete fuel combustion. The major goal of this article is to decrease heat losses and enhance the thermal efficiency of the engine. To interchange heat from the exhaust gas with input fresh charge gas by employing a copper tube with a convective mode of heat transfer, which resulted in alterations in the IC engine's performance [6].

Lakshmipathi et.al [8] in their research, evaluated the performance and emission characteristics of a four stroke single cylinder direct injection diesel engine with different dilution EGR and concluded work with decrease level of NOx emission but compromise with HC and CO emission with increase of other thermal efficiency under different load condition.

Jaffar Hussain et al. [9] carried out his work with the same condition as carried out by Lakshmipathi et.al [8] but for a three cylinder DI compressed ignition engine by varying load the effect with the varying EGR(0-25%) and estimated the effect on emission level and basic specific fuel consumption and other efficiencies and found that 15% EGR rate is optimum value for reduction in NOx emission substantially without compromising engine performance in terms of thermal efficiency, bsfc and emissions. At low load conditions ,EGR reduces NOx without decreasing the performance and emissions. At high load conditions, an increased rate of EGR reduces NOx to a great extent but deteriorates the performance and emissions.

The performance and characteristics of diesel engine blended with Plastic Pyrolysis Oil with EGR condition is also carried out by M Paul Daniel et.al [10] and .K. Venkateswarlu et. al. [13] with appropriate blending mixture can lead to a significant increase in efficiency and significant decrease in emission level. Another blending mixture of Calophyllum inophyllum methyl ester fuelled with EGR CI engines with different injection timing was carried out by B. Ashok et al.[11] and different emission and performance parameters were observed. The study done by Sumit Roy et al. [12] explores the potential of artificial neural networks to predict the performance and exhaust emissions of an existing single cylinder four-stroke CRDI engine under varying EGR strategies. Effect of EGR on advanced diesel combustion with alternate fuels were carried out by J. Thangaraja, C. Kannan [14] and concluded the remark that for future the usage of EGR is a necessary strategy for implementing advanced combustion concepts and to control emissions particularly NOx from alternate fuelled diesel engines.

II. EXPERIMENTAL SETUP:

A single cylinder, four stroke diesel engine was used for this experiment as shown in Fig [2]. The engine specifications in details are presented in Table 1. The engine is connected to the counter flow heat exchanger with a throttle valve. The exhaust gas recirculation unit (EGR) which plays a major role in reducing the NOx emission and fuel consumption. The flow rates are calculated and governed by the throttle valve. In this setup as shown in Fig [2], we attached the heat exchanger at the manifold of the engine and in the heat exchanger both the fresh charge and outlet pipes are being in contact then the heat transfer can occur. We use copper material for the tubes because only copper has a low cost and the best heat transfer coefficient as compared to the other materials. For thermally stable and robust heat exchangers, copper has several desirable properties. Copper is, first and foremost, a very good heat conductor. The experimental setup of Exhaust Gas Recirculation (EGR) on diesel engine evaluates the efficiency of the emission level and engine performance.

Table-1 Engine Specifications

Table-1 Engine Specifications	
Parameters	**Specifications**
Made	Kirloskar
Bore diameter	80mm
Stroke	110mm

Rated speed	1500 rpm
Rated power	5 hp / 3.7 kW
Ratio of compression	16.5:1
Fuel	diesel
Piston offset	0.02
Connecting rod length	235
Density of diesel	0.827 gm/ml
Brake drum diameter	0.3 m
Rope diameter	0.015 m
Calorific value of diesel	45,350 Kj/kg
Swept volume	553 cc
Clearance volume	36.87 cc

III. EXPERIMENTAL PROCEDURE:

The engine was started and the heat is transmitted to the exhaust gas recirculation (EGR) unit as shown in Fig [1]. The engine speed of 1500 rpm as constant and the speed changes with different loads. Also, the fuel consumption time and temperature are observed in Table [1,2]. The observations are done in two different criteria such as without heat exchanger and with heat exchanger with the exhaust gas recirculation (EGR) unit. The engine speed and the loads are changed and observed. Exhaust back pressures are measured by manometer in two different conditions such as along with heat exchanger and without heat exchanger. Observations are noted in Table[1] and [2], we calculate the brake power, indicative power, mass of fuel consumption, brake mean effective power and indicative mean effective pressure using analytical method. The thermal efficiencies are also calculated for brake, indicative and mechanical using the same analytical method.

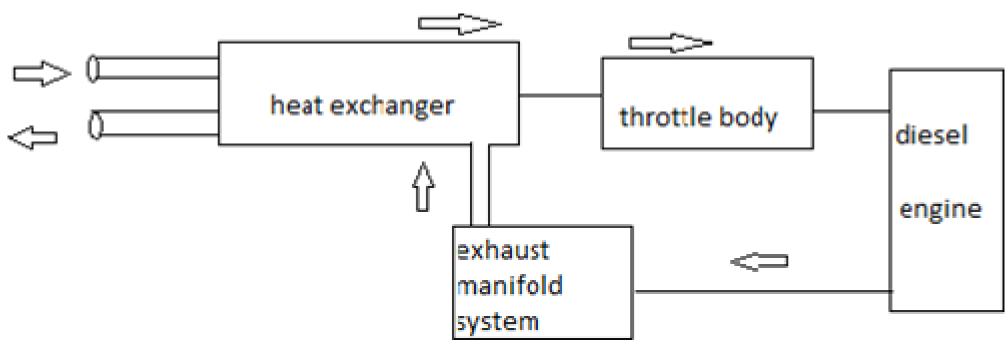

Fig 1: Basic block diagram of the experiment setup

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Fig 2: four stroke single cylinder diesel engine

Analytical Calculation of the performance parameter:

The performance parameters of the diesel engine such as Brake power, Indicative power, mass of fuel consumption, Brake mean effective pressure, indicative mean effective pressure, brake thermal efficiency, indicative thermal efficiency and mechanical efficiency were calculated using the following derived equations are given below.

Brake Power, BP = $$\frac{2\pi N(S1+S2)(\frac{D+d}{2})}{60000} \quad \text{……… (1)}$$

Where N represents the revolution per minute of the crankshaft, $S1$, $S2$ are the net load acting on the brake drums and D, d is the diameter of the brake drums. Brake power is the power produced at the crankshaft.

Indicative Power (IP) = BP + FP ……… (2)

Where, BP represents the brake power and FP denotes the frictional power. By adding the brake power and frictional power we obtain the indicative power which is the total power generated within the cylinder in one complete cycle.

Mass of fuel consumption, $$mf = \frac{(X \times Sg \times 3600)}{(1000 \times t)} \quad \text{……… (3)}$$

Where the X represents the Manometer reading, with Sg denotes specific gravity of fuel and the t is the time duration.

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$$\textbf{Brake mean effective pressure, } B_{mep} = \frac{(BP \times 1000) \times 120}{10^5 \times (\frac{\pi d^2}{4}) \times L \times N} \quad \text{......... (4)}$$

Where BP is the Brake power, N represents the revolution per minute of the crankshaft and L denotes the stroke length. Brake mean effective pressure reveals the potential of the engine to bring out the highest brake power depends upon the engine displacement volume.

Indicative mean effective pressure, ***I***

$$\textbf{Indicative mean effective pressure, } I_{mep} = \frac{(IP \times 1000) \times 120}{(10^5 \times (\frac{\pi d^2}{4}) \times L \times N)} \quad \text{......... (5)}$$

Where IP is the Indicative power with the revolution per minute/speed of the crankshaft represented by N and L denotes the stroke length.

$$\textbf{Mechanical efficiency, } \eta_{mech} = \frac{BP}{IP} \quad \text{......... (6)}$$

Where, BP stands for brake power and IP denotes the Indicated power. Mechanical efficiency is the ratio of brake power to Indicated power which is represented in the above equation.

$$\textbf{Indicative thermal efficiency, } \eta_{ith} = \frac{(IP \times 3600)}{(mf \times CV)} \quad \text{......... (7)}$$

Indicative thermal efficiency is calculated using indicated power, mass of fuel consumed and the calorific value of fuel. In this equation IP denotes Indicated power, stands for mass of fuel consumed and CVrepresents the calorific value of the fuel. The above equation is used to find the amount of heat changed to indicative power.

$$\textbf{Brake thermal efficiency, } \eta_{bth} = \frac{(BP \times 3600)}{(mf \times C_V)} \quad \text{......... (8)}$$

Brake thermal efficiency is denoted as ηbth, and the above equation is the ratio of brake power to the power produced by the combustion of fuel is calculated using brake power, mass of fuel consumed and the calorific value of fuel. In this equation BP denotes Brake power, mf stands for mass of fuel consumed and CV represents the calorific value of the fuel. The observed readings such as the speed, load and the fuel consumption are mentioned in the table 1 and table 2 with two different criteria such as with heat exchanger and without heat exchanger as given below

Table: 1 Observation table without heat exchanger

S.NO	SPEED(N)	S1(kg)	S2(kg)	MANOMETER READING(X)	TIME OF FUEL CONSUME	Co level (ppm)

1	1540	0	0	6	39	310
2	1538	1	2	6	38	450
3	1534	2	4	6	36	510
4	1532	4	5	6	31	560
5	1527	5	5	6	25	610

Table: 2 Observation table with heat exchanger

S.no	(N) Rpm	S1 (kg)	S2 (kg)	MANOMETER READING(X)	TIME OF FUEL CONSUME	Co level (ppm)	T Exit ˚C	T In ˚C
1	1560	0	0	6	43	425	154	46
2	1554	1	2	6	42	433	154	48
3	1542	2	5	6	40	450	155	49
4	1538	3	6	6	36	470	159	50
5	1535	4	7	6	33	560	162	52

IV. RESULTS AND DISCUSSION:

Using the analytical calculation, we obtain two results in two different parameters such as with heat exchanger and without heat exchanger. The parameter such as Brake power is improved by using heat exchanger with the increased net load. The efficiencies such as brake thermal efficiencies, indicative thermal efficiency and mechanical efficiency are also improved in using heat exchanger increases with respect to the net load increases. Figure 3 to Figure 9 shows the plotted points of the graphical representation of comparing two factors of using and without using heat exchanger with various parameters such as brake power, indicative power, mass of fuel consumption, brake mean effective pressure, indicative mean effective pressure, mechanical efficiency, indicative thermal efficiency and the brake thermal efficiency with respect to the corresponding mechanical loads and it is shown below. The results clearly reveal the improvement of the efficiencies using the heat exchanger by the benefit of exhaust gas recirculation unit.

The analytical calculation is performed in terms of various input weights like 0, 3, 6, 9, and 10 kg by two different criteria such as without heat exchanger and with heat exchanger. The performance parameters such as brake power, indicative power, mass of fuel consumption, brake mean effective pressure, indicative mean effective pressure, mechanical efficiency, brake thermal efficiency and indicative thermal efficiency are observed and noted in two criteria and given below.

Table: 3 Result table without heat exchanger

OBJECTIVE	S=0 kg	S=3 kg	S=6 kg	S=9 kg	S=10 kg
BP kw	0	0.3	0.6	0.9	1.0
IP kw	1.6	1.9	2.2	2.5	2.6
ŋ	0%	15%	27%	35%	38%
Bmep	0	0.24	0.49	0.75	0.8
Imep	1.31	1.56	1.83	2.06	2.15
ŋ B thermal	0%	5.4%	10.2%	13.2%	11.9%
ŋ I thermal	29.3%	34.4%	37.8%	36.1%	30.0%

Table: 4 Result table with heat exchanger

OBJECTIVE	S=0 kg	S=3 kg	S=7 kg	S=9 kg	S=11 kg
BP kw	0	0.30	0.71	0.97	1.10
IP kw	1.6	1.9	2.31	2.57	2.70
ŋ	0%	15%	30%	37%	40%
Bmep	0	0.24	0.58	0.79	0.91
Imep	1.29	1.54	1.89	2.10	2.22
ŋ B thermal	0%	5.9%	13.3%	16.5%	17.4%
ŋ I thermal	32%	37%	43%	44%	42%

The above given table 3 and 4 shows the obtained values from the analytical calculations. The attained outcomes for the performance parameters are discussing in following,

Brake power (BP)

The power produced in the crankshaft increases with the effect increasing loads. It is observed from the result Table [3] and Table [4] that from the five-weight variation we notice that the improvement of the brake power. In the net weight, 0, 3, 6, 9 and 10 kg we obtain the result of 0, 0.3, 0.6, 0.9, 1 kw respectively for the case of without using heat exchanger. While using the heat exchanger the net weight used as same as 0, 3, 6, 9 and 10 kg we acquire the brake power as 0, 0.3, 0.71, 0.97, 1.10 kw. Now we know that the brake power increases in the case of using the heat exchanger prominently and specifically when the net weights increase to 6 kg and above.

Indicative Power (IP)

The indicative power produced within the cylinder show a rise in the changes of the net weights such as 0, 3, 6, 9 and 10 kg gives the output of 1.6, 1.9, 2.2, 2.5, 2.6 kW in the case of not using heat exchanger are shown in Table [3]. In the case of using heat exchanger for same net weights such as 0, 3, 6, 9 and 10 kg. We notice the Table [4] that the first two results are the same as the previous case but for 6, 9 and 10 kg we obtain a mild change of increase in result such as 2.31, 2.57, 2.7 kw respectively. It is noted that with using the heat exchanger and increased load weight gives a valid output.

Mechanical efficiency ($\eta mech$)

The mechanical efficiency is calculated by the brake power and the indicated power and the results are obtained in the Table [3] and Table [4] in varying the net weights. While comparing the two table results, Table [3] and Table [4] clearly shows the increase in the efficiency of the increasing net weights. For Table [3], the net weights increased as 0, 3, 6, 9 and 10 kg the result efficiencies obtained are 0%, 15%, 27%, 35% and 38% respectively while not using the heat exchanger. Using the heat exchanger, the result changes to the improvement of efficiencies for the same net weights such as 0, 3, 6, 9 and 10 kg, the output gained are 0%, 15%, 30%, 37%, 40% respectively.

Brake mean effective pressure ($Bmep$)

The potential of the engine by means of torque calculated by the brake power and it is viewed from the Tables in terms of brake mean effective pressure, from two different conditions of without using heat exchanger in Table [3] and using heat exchanger in Table [4]. Table [3] and Table [4] consists of the same net weights such as 0, 3, 6, 9 and 10 kg produces different pressure as 0, 0.24, 0.49, 0.75 and 0.8 bar in Table [3] and 0, 0.24, 0.58, 0.79 and 0.91 bar pressure in Table [4] respectively. The result of both the tables varies from the net weights 6, 9 and 10 kg in an increasing manner.

Indicative mean effective pressure ($Imep$)

Indicative mean effective pressure is calculated by means of effective pressure of the indicative power and the result values are displayed in two tables. Table [3] defines the result value as 1.31, 1.56, 1.83, 2.06 and 2.15 bar pressures for the following net weights 0, 3, 6, 9 and 10 kg while not using the heat exchanger. In Table [4], while the heat exchanger is used, it yields the result of 1.29, 1.54, 1.89, 2.10 and 2.22 bar pressure for the same net weights used in Table [3]. While comparing the two table results, we find the lower bar values in the set up using heat exchanger than the set up without using heat exchanger for the net weights 0 and 3 kg. For the net weights 6, 9 and 10 kg we attain higher indicative mean effective pressure than without using heat exchanger.

Brake thermal efficiency (ηbth)

In table [3], for the following net weights 0, 3, 6, 9 and 10 kg in a certain condition of not using heat exchanger gives the percentage of 0%, 5.4%, 10.2%, 13.2% and 11.9%. We find the result fluctuations of percentage in increase in net weights. From the results, we obtain increase in percentage and reached maximum percentage at net weight 9 kg and decrease in percentage at 10 kg. In Table [4], the same net weights were used; we achieved a gradual increase in the result percentage as 0%, 5.9%, 13.3%, 16.5% and 17.4%. Comparing both the tables we found that using heat exchanger results in the standard increase in efficiency with respect to increase in the net weights.

Indicative thermal efficiency (ηith)

The percentage efficiency is calculated by the indicated power with the combustion fuel power along with the mass of fuel consumed with calorific values. In Table [3] and Table [4], we obtain the following result percentages as 29.3%, 34.4%, 37.8%, 36.1% and 30.0% without using heat exchanger and 32%, 37%, 43%, 44% and 42% respectively for the same net weights such as 0, 3, 6, 9 and 10 kg. From Table [3], we found that the increase in percentage with increase in net weight and decreases after 6 kg and when the net weight increases to 9 and 10 kg,

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the percentages decreases. From Table [4], we attain the result percentages as the increase in the percentage with increase in the net weights and at 10 kg net weight, the percentage reduced sharply as found.

The above parameters clearly speak out the comparison of two different criteria such as without using heat exchanger and using heat exchanger. The results give a clear cut idea of the increasing efficiency of the above parameters. The results obtained from both the cases describe the method of increasing power, pressure and efficiencies. We found that the method of using heat exchanger gives high yield in output and performance.

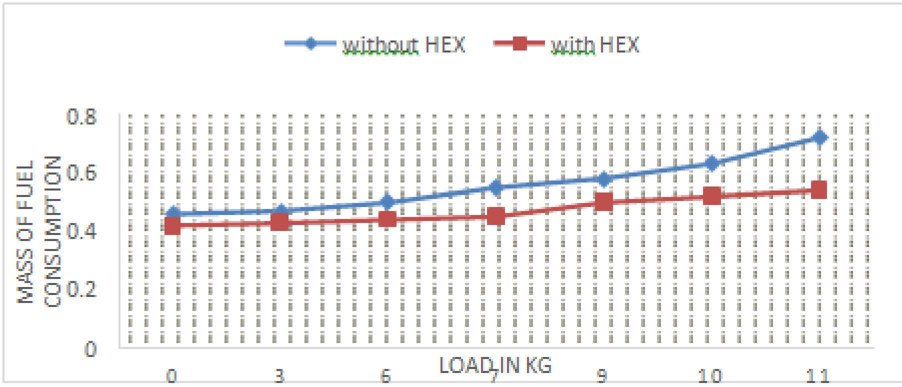

Figure 3: Mechanical loads Vs Mass of fuel consumption (mf)

Figure 3 represent the Mechanical loads in kg vs mass of fuel consumption. There is a significant change has been occurred in comparison with heat exchanger and without a heat exchanger, those changes occur mainly due to several reasons the main probable reason is mass of Consumption of fuel, which implies the efficiency of transforming the chemical energy content of fuel into useful work.

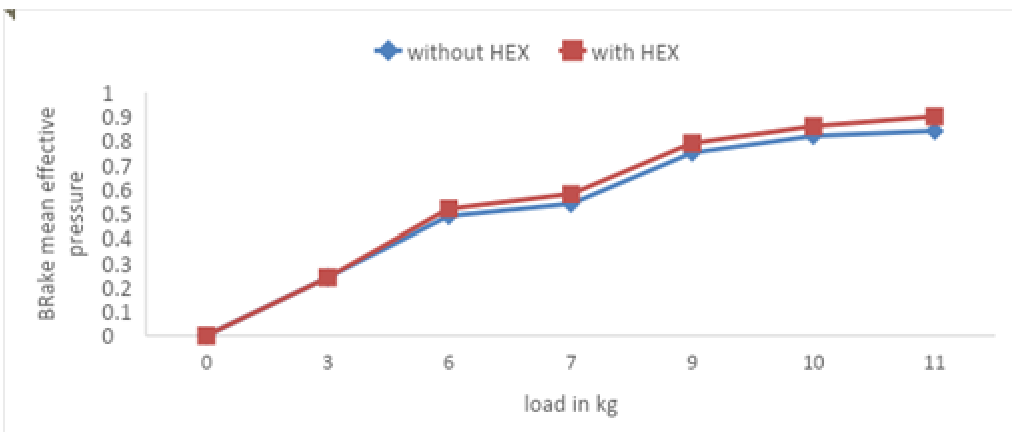

Figure 4: Mechanical loads vs Break mean effective pressure

Figure 4: represents Mechanical loads in kg vs break mean effective pressure which leads to moderate changes as compared to the standard engine arrangement. This thing happens due to the applied load on the engine at different temperature levels of the fresh charge, which impacts the combustion process in the engine, gives to the moderate changes as compared to the standard arrangement of the engine.

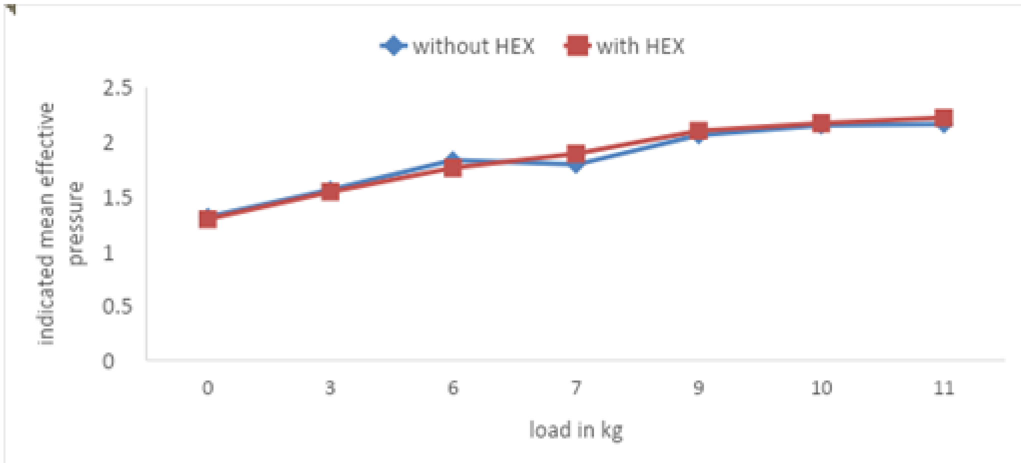

Figure 5: Mechanical loads vs Indicate mean effective pressure

Figure 5 represents the Mechanical loads vs indicative meant effective pressure at a different level of the temperature to the inlet air. The moderate changes have been occurred due to the changes in brake power because the indicative power is dependent on the broken power and frictional power which is directly proportional to the brake power. Based on that indicative power and speed of the engine concerning the load, we can get the indicated mean effective pressure.

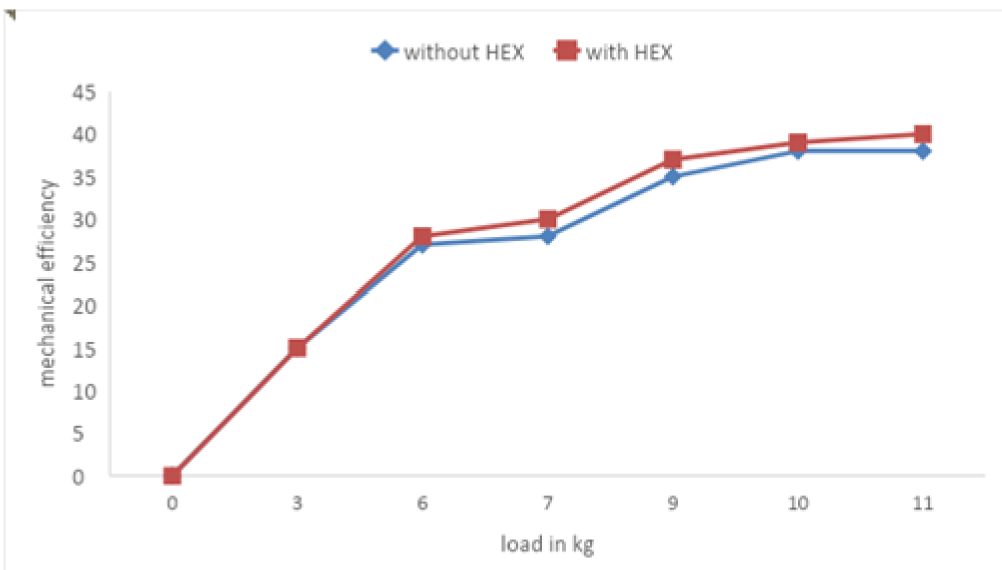

Figure 6: Mechanical loads vs mechanical efficiency

Figure 6: represents Mechanical loads in kg vs mechanical efficiency. The mechanical efficiency is directly proportional to the broken power and inversely proportional to the indicative power. At a particular load, the efficiency can start decreasing due to several reasons like increasing the wear and tear, the mass of fuel consumption is more, and inside air temperature can be increased.

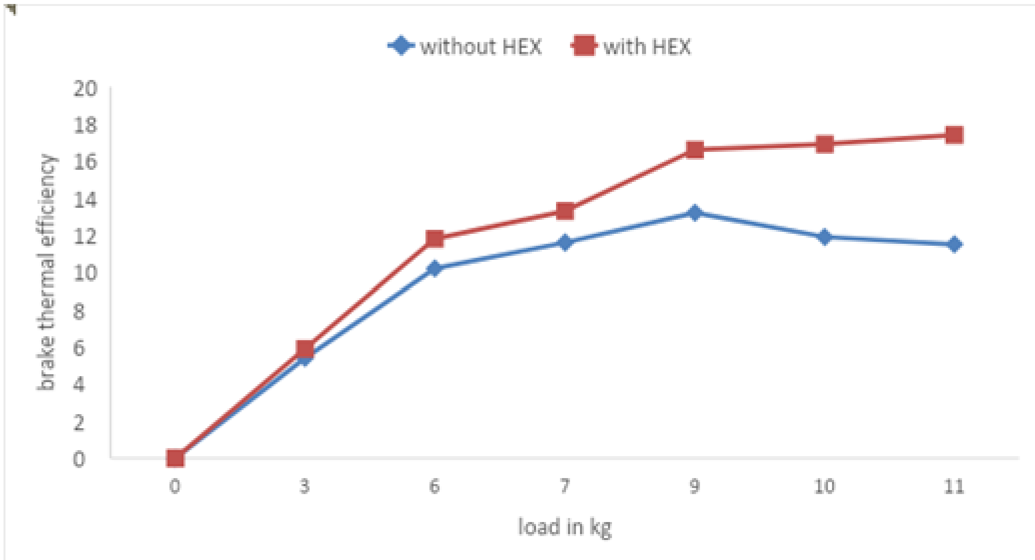

Figure 7: Mechanical loads vs break thermal efficiency

Figure 7 represents Mechanical loads in kg vs brake thermal efficiency. There are significant changes that have been occurring in the brake thermal efficiency with the comparison of the engine with heat exchanger and to the standard engine arrangement. These changes are occurring by influencing the major factors like brake power and mass of fuel consumption.

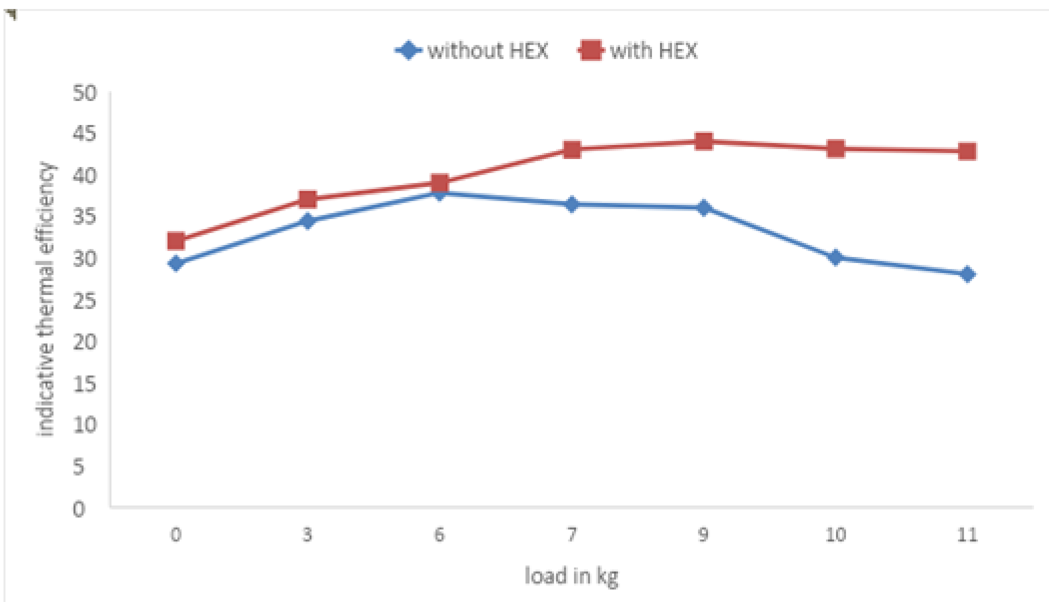

Figure 8: Mechanical loads vs indicative thermal efficiency

Figure 8 represents Mechanical loads in kg vs indicative thermal efficiency of the engine. These major significant changes are also obtained due to the factors influencing indicative power and mass of fuel consumption, which these are indicating that the thermal performance of the engine has been raised.

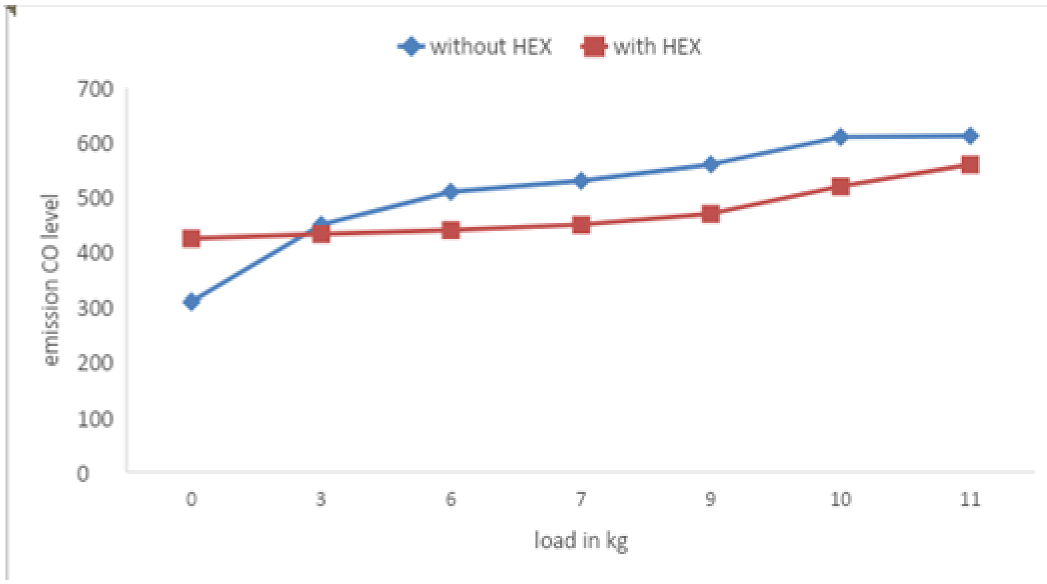

Figure 9: Mechanical loads vs Emission (CO)

Figure 9 represents Mechanical loads vs emission levels. Mechanical load plays predominant role in emission and it results in major difference in the emission level. While preheating the charge, the temperature level of the air will raise which leads to complete combustion of the fuel so finally, we get less pollute gas from exhaust and the emission is reduced.

From the graphs and results from the tables, we conclude that the thermal efficiency has been increased and the mass of fuel consumption is reduced and the emission levels also reduce.

V. CONCLUSION:

This work was conducted with a Kirloskar diesel engine with the exhaust gas recirculation (EGR) unit set up. Net loads and engine speed with 1500 rpm were taken as the input parameters and the exhaust back pressures were observed in manometer. Carbon monoxide content in the fumes gas somewhat diminishes with increment in admission air temperature. Subsequently, the investigation shows that we are getting more preferences by utilizing the preheated air and the focal points picked up are more with increment in inlet air temperature. By this paper, we gain effective aspects like utilizing waste heat reduces the number of greenhouse gases. This will reduce the ignition delay and starts of the engine even in the cold conditions also. Pre heating leads to a homogeneous mixture in the combustion chamber. The inlet air temperature increases and indicated thermal efficiency also increase to 8%. Fuel consumption reduces to 0.18 % and brake thermal efficiency increases to 5%. Finally, the mechanical efficiency has been raised to 2% concerning load. Thus, the presented methodology has showed that the using of heat exchanger gives a prominent output than without using heat exchanger with the reduced mass of fuel consumption and emission. This paper proves that it yields higher performance and efficiencies.

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