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An Experimental Investigation and Analysis for Engine Performance, Combustion and Emissions of Dual Fuel CI Engine

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ABSTRACT

The increasing concern regarding diesel vehicle emissions and the rising cost of liquid diesel, leads to the need for alternative fuel. The engines which use conventional pilot diesel fuel and primary gaseous fuels are referred to as 'dual fuel engines'. Dual fuel engine suffers from some deficiencies in combustion, performance and in emissions characteristics. Thus the selection of engine operating and design parameters plays a vital role in minimizing the divergences of 'dual fuel engine'. In this work, the combustion, performances, and exhaust gas emissions characteristics are investigated for LPG-Diesel dual fuel engine. A single cylinder, four stroke, water cooled, diesel engine coupled with eddy current type dynamometer is used as an engine test rig. Experiments were carried at constant speed with variations in compression ratios, loads and pilot diesel fuel quantities. Experimental results shows that the increasing the amount of pilot fuel quantity increases thermal efficiency at all load conditions. The BSFC decreases with increasing percentage of pilot fuel quantity at part load conditions. The unburned HC and CO emissions under dual-fuel mode are higher than that of the normal diesel mode at low load condition.

Keywords:- **Alternative fuel, Combustion, Conventional, Dual fuel engine, Emission, LPG, Performance**

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I. INTRODUCTION

The increasing concern regarding diesel vehicle emissions and the rising cost of liquid diesel, leads to the need for alternative fuel. Therefore, the option of using conventional diesel engines as 'dual fuel engines'; in which gaseous fuels as a supplement for the conventional diesel fuel can be used [1]. These improve the awareness of the effective use of present reserves and slowly switch over to the alternative fuels, which are environment friendly. The use of alternative gaseous fuels e.g. natural gas, liquefied petroleum gas (LPG), etc. is a promising approach for lowering the dependence on petroleum based liquid fuels and to reduce the emissions of CO_2 and other pollutants from diesel engine. LPG is a viable alternative gaseous fuel (also known as ''Auto gas'') which is a gas product of petroleum refining primarily consisting of propane, propylene, butane and other light hydrocarbons. It can be liquefied in a low pressure range of 0.7–0.8 Mpa at atmospheric temperature. So, storage and transportation of LPG is easier than other gaseous fuels. LPG has high calorific value compared to other gaseous fuels and also it has high octane number but a low cetane number. The high octane number of LPG makes it suitable for spark ignition engines. In contrast, the low cetane number of LPG makes it difficult to be used in large proportions in compression ignition engines, mainly due to high cyclic variation. Hence it can be used in the CI engine in the dual fuel mode only and in this mode it has been extensively studied. It leads to better performance, low particulate and smoke emissions. The engine, which uses conventional diesel fuel and LPG fuel, is referred to as 'LPG–Diesel dual fuel engine'. In this engine, LPG fuel is mixed with the air in the engine cylinders either through direct mixing in the intake manifolds with air or through injection directly into the cylinder. A dual fuel engine is basically a modified diesel engine in which a LPG fuel, called the primary fuel is inducted along with air. This fuel is the main source of energy input to the engine. The primary gaseous fuel is compressed with air, but does not auto ignite due to its high self-ignition temperature. A small amount of diesel, usually called the pilot, is injected as in a normal diesel engine near the end of compression of the primary fuel–air mixture. This pilot diesel fuel, auto ignites first and acts as a deliberate source of ignition for the combustion of the gaseous fuel–air mixture. The pilot diesel fuel, which is injected by the conventional diesel injection equipment normally, contributes only a small fraction of the engine power output.

Thus the combustion process in a dual fuel engine is complex as it combines the features of SI and CI engines. The dual fuel engines can also be reverted back to straight diesel operation easily. Dual fuel operation has advantages compared to diesel counterparts and spark ignition (SI) engines, theoretically higher thermal efficiency resulted from faster burning, less toxic emissions, high power density, strong ignition sources providing more reliable. By converting diesel engines to run on LPG we can significantly reduce the problem of diesel pollution while also improving emissions of greenhouse gases. Such conversions are however not a simple matter of changing the fuel, many technical problems present particularly with availability of specific fuel supply system, fuel injection control and engine optimization to ensure that the engine performance is maintained and the exhaust emissions are minimized. However, the dual fuel engine has some pitfalls such as the poor utilization of the LPG fuel at low and intermediate loads which results in poor engine performance (drop in engine efficiency), high HC, CO emissions and misfiring at higher gas inducted levels. Poor part load performance results from incomplete combustion of LPG. Due to this poor thermal efficiency high level of unburnt hydrocarbons in the exhaust is found.

The performance of a dual fuel engine at idling and low loads can be improved by optimizing some engine operating and design parameters, such as engine speed, load, pilot fuel quantity, injection timing, and intake manifold condition and intake gaseous fuel compositions. Variation of gas quality and composition affects on combustion characteristics. Combustion phenomenon depends on working as well as design parameters of engine like load, pilot fuel quantity, and compression ratio and injection pressure. So these parameters should be tune-up for particular gas and liquid fuel. The well selection of working parameters of engine plays a vital role in minimizing performance divergence of engine [2, 3].

Many researchers have carried out works on Gas- Diesel dual fuel engines. The review of the researcher's study gives us the knowledge of the use of different working and design parameters and their suitability to the engine performance, combustion and emission characteristics. Thus literature reviews given below gives us the new idea to find out the crucial parameters affecting the engine performance and to overcome the problems in present situation.

Selim, et al,[4] stated that, gaseous fuels like hydrogen, methane, CNG, LPG, LNG, Producer gas and Biogas etc. are to be good alternative fuels for passenger cars, truck transportation and stationary engines that can provide both good environmental effect and energy security.

Nirendra N. Mustafi, et al,[5] an experimental investigation was performed to investigate the influence of dual-fuel combustion on the performance and exhaust emissions of a DI diesel engine fuelled with natural gas (NG) and biogas (BG).

The engine was operated at a constant speed of 1750 rpm and at two different loads: low (~3 N m) and high (~28 N m), which were about 10% and 85% respectively of the rated torque output of the engine at 1800 rpm. About 27–30% higher maximum net heat release rates were obtained for NG and biogas fuelling respectively compared to diesel fuelling. The presence of higher CO_2 in the gaseous fuel caused a longer ignition delay, increase of BSFC, BSEC and UHC emission. Reduction of NOx by 9 to 12% for different D+BG fuelling condition with increasing CO_2 content in biogas.

R.G. Papagiannakis, at al, [6] conducted an experiment to examine the effect of load on the performance and pollutant emissions of a DI diesel engine. Measurements are taken at three engine speed 1500, 2000, and 2500 rpm and 40%, 60%, 80% of full load under both dual fuel operation and diesel alone. The results occurred are. Total heat release rate under dual-fuel operation is slightly higher compared to the one under normal diesel operation revealing late combustion of the gaseous fuel, at low engine loads, the combustion duration, even tends to become longer compared to normal diesel operation. At low engine loads, the total 'bsfc' for dual-fuel operation is considerably higher compared to the one under normal diesel operation. At low engine loads, the NOx concentration under dual-fuel operation is slightly lower compared to the one under normal diesel operation. CO and HC concentration under dual-fuel operation clearly decreases with the increase of engine load, while increase of load leads to increase of amount of gaseous fuel that forms no soot.

II. EXPERIMENTAL SETUP & METHODOLOGY

The experiments were performed at internal combustion engine laboratory at VPCOE, Baramati using a single cylinder, four stoke, water cooled VCR Kirloskar diesel engine test rig the specification of engine shown in TABLE

1. Fig.1 shows layout of experimental setup. Test runs were conducted with the Diesel engine test rig with various sets of straight diesel as well as dual fuel operation. For dual fuel operation, diesel as pilot fuel and LPG (Liquefied petroleum gas) were used. The experiments' were performed under following mode conditions the engine was operated at constant speed of 1500 rpm and at compression ratios of 15 to 18 with the step of 1, while the load varies from 0 to 100% of full load with the step of 25%. The pilot diesel fuel

quantities vary from 20% to 100% with step of 20%. The gas flow was adjusted according to pilot diesel fuel supply.

TABLE 1.
ENGINE SPECIFICATIONS

Parameters	Specifications
Constructor	Kirloskar
Engine type	1 Cylinder, 4 Strokes, Water Cooled, Modified VCR Diesel Engine.
Bore (mm)	87.5
Stroke (mm)	110
Compression ratio	12:1 to 18:1
Speed (RPM)	1500
Power output(kW)	4.2
Volumetric capacity (cc)	661

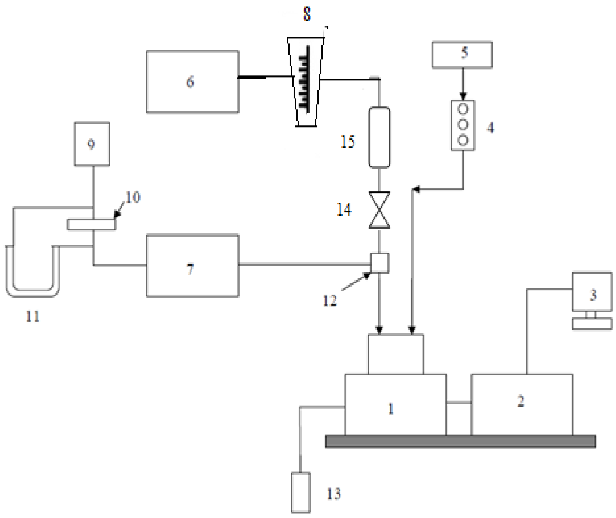


Fig.1 Layout of experimental setup

1. Test Engine, 2.Dynamometer, 3. Computer interface of the control panel, 4. Fuel consumption meter, 5. Fuel tank, 6. Gas tank, 7. Air surge tank, 8. LPG gas flow meter to measure flow rate of gas, 9. Air filter, 10. Orifice plate to calculate air flow rate, 11. Manometer to measure pressure difference across orifice, 12. Air gas mixer, 13. Exhaust gas analyzer, 14. Gas flow control valve , 15. Gas box.

TABLE2.
FUEL PROPERTIES

Property	Diesel	LPG
Chemical formula	$C_{12}H_{26}$	C_3H_8,C_4H_{18}
Density kg/m^3	830	2.155
Lover heating value KJ/kg	42500	46100
Stoichiometric air-fuel ratio kg/kg	14.5	15.5
Auto Ignition temperature ^{0}C	250	400

III. RESULT & DISCUSSION

This section presents engine combustion, performance and emissions characteristics that were studied experimentally through different operating modes. Five pilot fuel quantities of 20% to 100% with the step of 20% were chosen and the power percentages were set to be 0 to 100% with the step of 25%, for four compression ratios as 15, 16, 17 and 18. At each test condition, experiments were carried out with pure diesel and dual fuel operation modes. Comparison parameters for different operation modes were conducted.

A. Combustion characteristics

Combustion characteristics of diesel alone and diesel-LPG are shown in fig. 2,3,4. The cylinder pressure traces under dual fuel operation diverge from the respective values under normal diesel operation. The lower cylinder pressure observed under dual fuel operation during the compression stroke is the result of the higher specific heat capacity of the LPG-air mixture. Higher the combustion pressure, NOx emission will be higher.

Heat release diagram of diesel fuel operation (Fig.3) shows an apparent negative heat release prior to the main start of combustion. This is due to the cooling effect of injected liquid fuel. Total heat release rate under dual fuel operation is slightly higher compared to the normal diesel operation, revealing late combustion of the gaseous fuel.

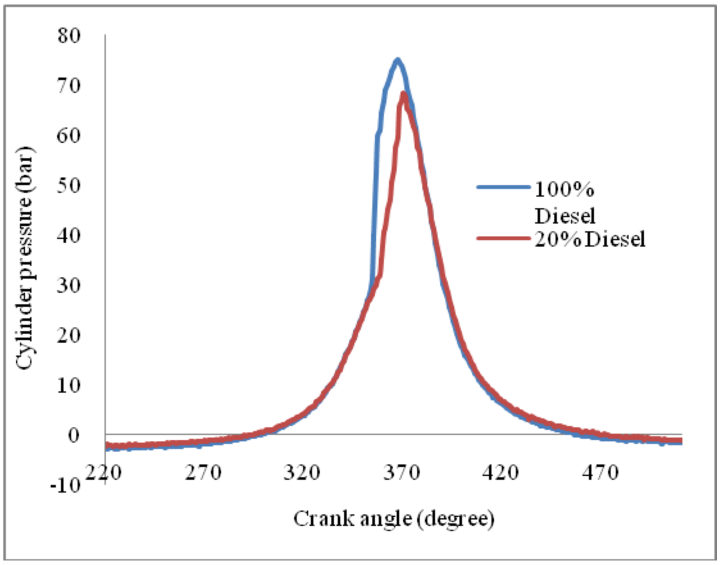


Fig.2 Pressure vs Crank angle of dual fuel engine at 18 CR and at full load.

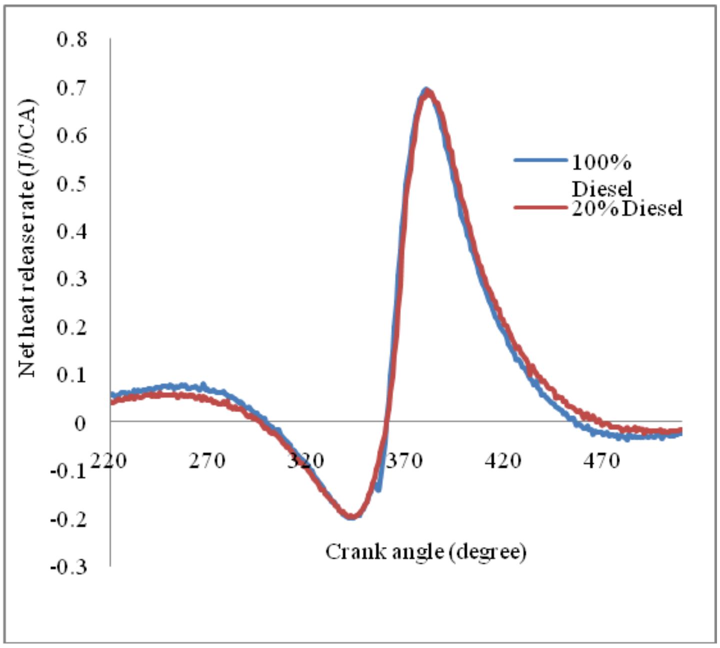


Fig.3 Net heat release rate vs Crank angle of dual fuel engine at 18 CR and at full load.

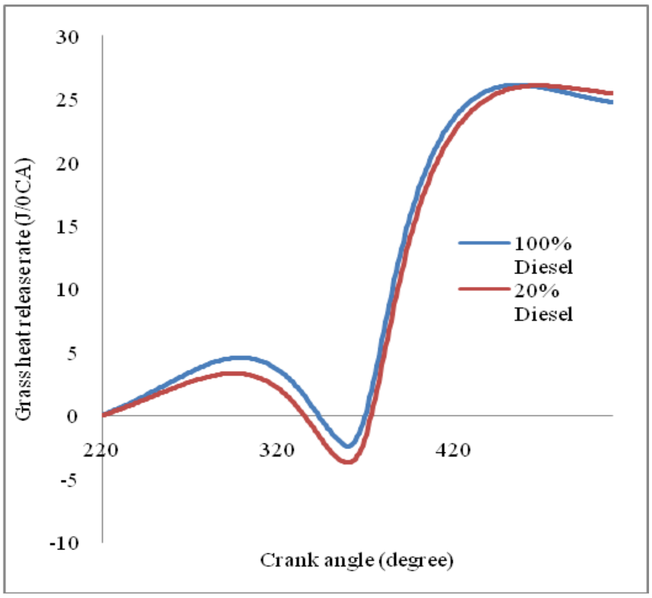
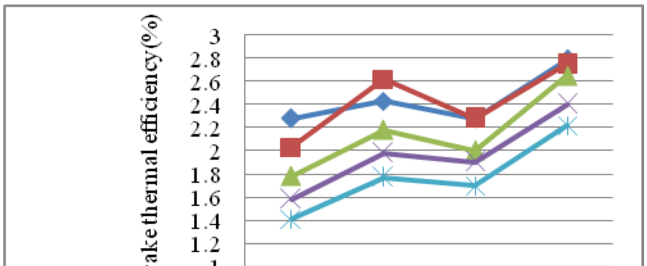


Fig.4 Gross heat release rate vs crank angle of dual fuel diesel engine at 18 CR and at full load.

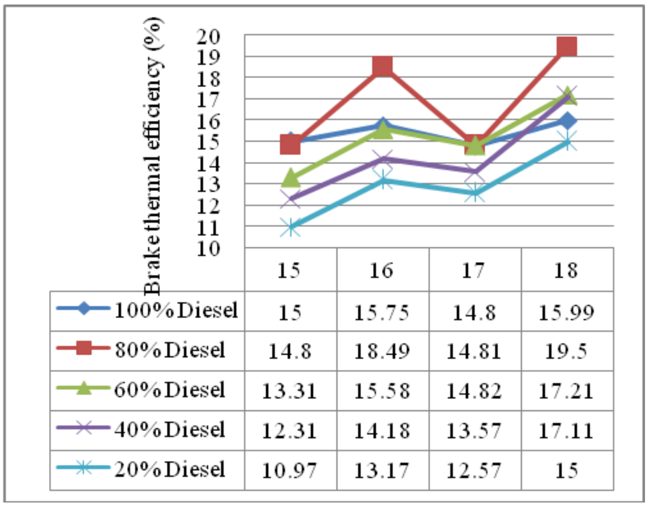
B. *Performance characteristics*

1) *Effect on Brake Thermal Efficiency*



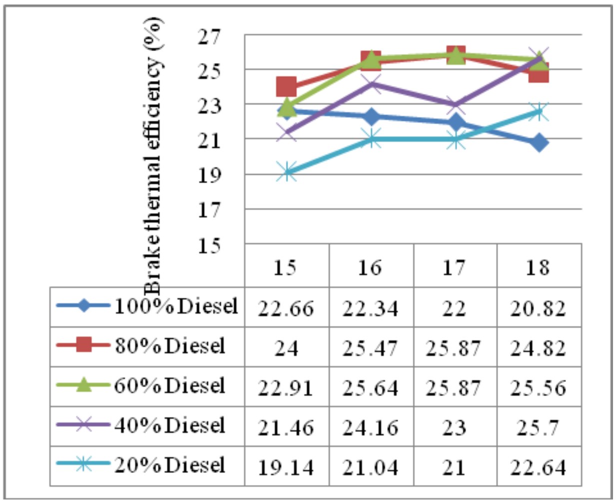
	15	16	17	18
100%Diesel	2.28	2.43	2.28	2.79
80%Diesel	2.03	2.62	2.28	2.75
60%Diesel	1.78	2.18	2	2.65
40%Diesel	1.58	1.98	1.9	2.4
20%Diesel	1.41	1.77	1.7	2.22

Fig.5 Effect of Compression ratio on brake thermal efficiency at 0 load



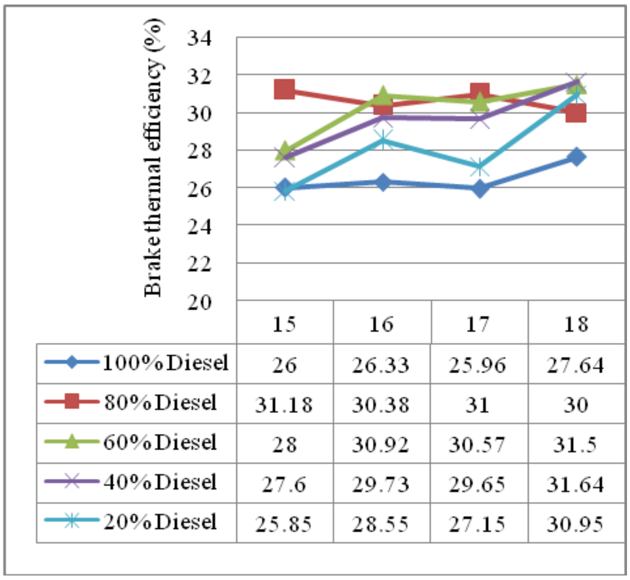
	15	16	17	18
100%Diesel	15	15.75	14.8	15.99
80%Diesel	14.8	18.49	14.81	19.5
60%Diesel	13.31	15.58	14.82	17.21
40%Diesel	12.31	14.18	13.57	17.11
20%Diesel	10.97	13.17	12.57	15

Fig.6 Effect of Compression ratio on brake thermal efficiency at 25% load



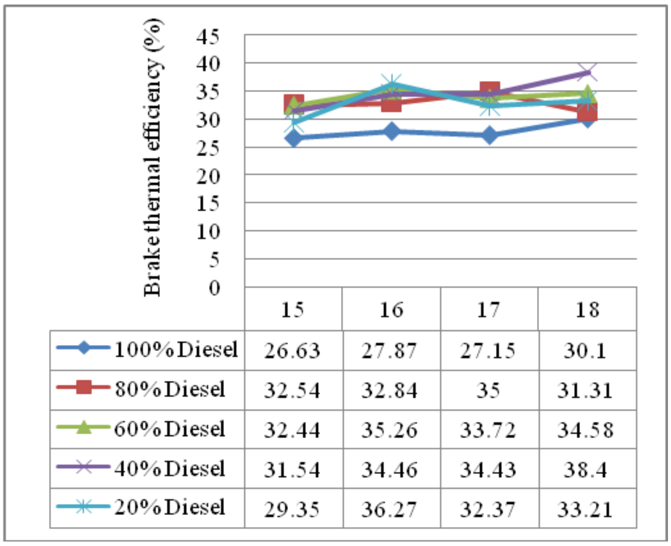
	15	16	17	18
100%Diesel	22.66	22.34	22	20.82
80%Diesel	24	25.47	25.87	24.82
60%Diesel	22.91	25.64	25.87	25.56
40%Diesel	21.46	24.16	23	25.7
20%Diesel	19.14	21.04	21	22.64

Fig.7 Effect of Compression ratio on brake thermal efficiency at 50% load



	15	16	17	18
100%Diesel	26	26.33	25.96	27.64
80%Diesel	31.18	30.38	31	30
60%Diesel	28	30.92	30.57	31.5
40%Diesel	27.6	29.73	29.65	31.64
20%Diesel	25.85	28.55	27.15	30.95

Fig.8 Effect of Compression ratio on brake thermal efficiency at 75% load



	15	16	17	18
100%Diesel	26.63	27.87	27.15	30.1
80%Diesel	32.54	32.84	35	31.31
60%Diesel	32.44	35.26	33.72	34.58
40%Diesel	31.54	34.46	34.43	38.4
20%Diesel	29.35	36.27	32.37	33.21

Fig.9 Effect of Compression ratio on brake thermal efficiency at 100% load

From the figures (5, 6, 7, 8, 9) it shows that as load increases brake thermal efficiency increases this is because at low and intermediate load conditions there is poor utilization of LPG takes place resulting incomplete combustion of LPG. At all loads as pilot fuel quantity increases, brake thermal efficiency increases as compared of diesel operation this is because with larger pilot fuel the combustion of the LPG fuel is better leading to higher mass fraction burnt and higher brake thermal efficiency in dual fuel mode as compared to the corresponding diesel operation. Higher brake thermal efficiency is due to better mixing of LPG with air which results in better combustion and also due to wider ignition limit and high burning velocity.

2) *Effect on Brake Specific Fuel Consumption:* It is observed that by increasing load BSFC decreases, as with increase in load cylinder pressure and temperature increases, which improves the combustion process resulting in decrease in BSFC. The BSFC increases with increasing percentage of LPG substitution at part loads may be due to incomplete combustion of the gaseous fuel, while at higher loads BSFC improves with the increase of LPG substitution.

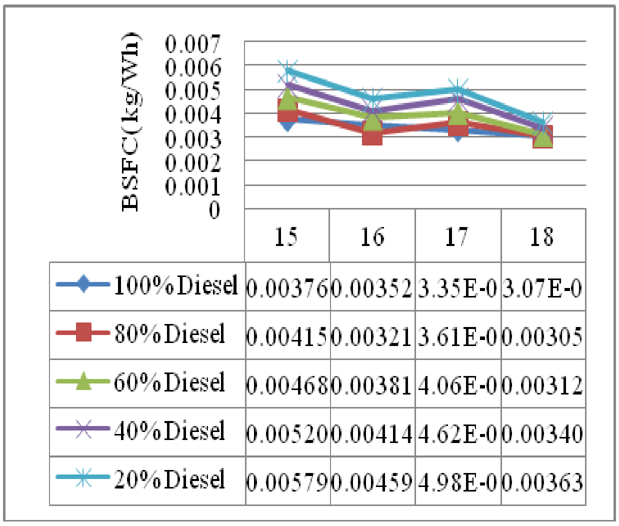


Fig.10 Effect of Compression ratio on BSFC at 0 load

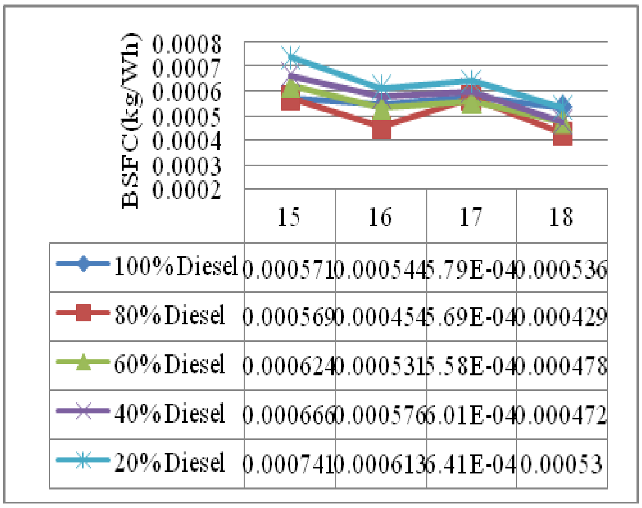


Fig.11 Effect of Compression ratio on BSFC at 25% load

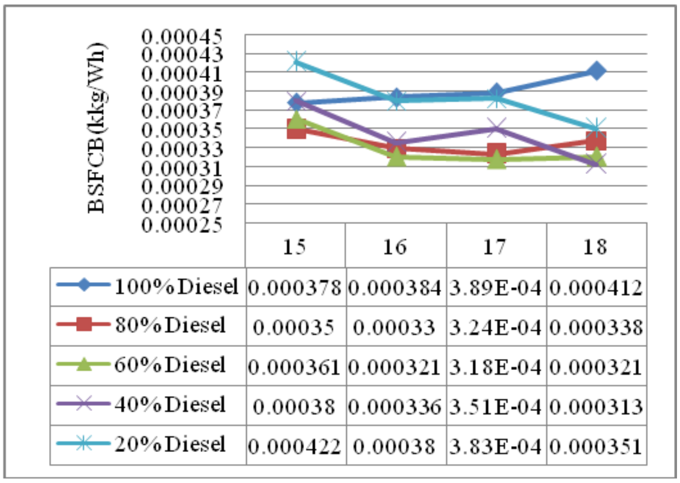


Fig.12 Effect of Compression ratio on BSFC at 50% load

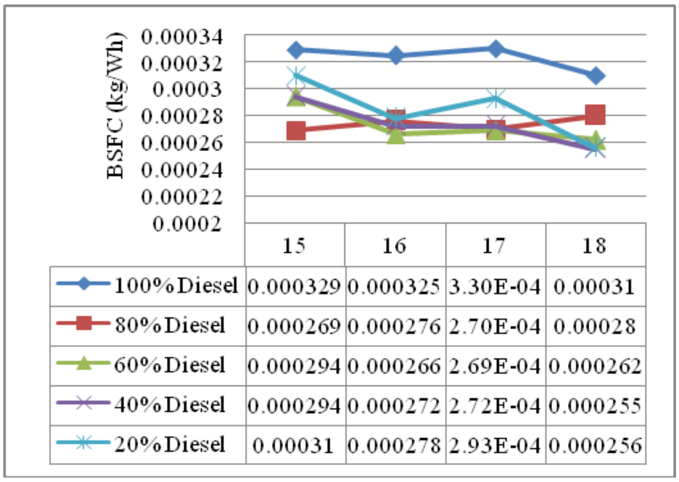


Fig.13 Effect of Compression ratio on BSFC at 75% load

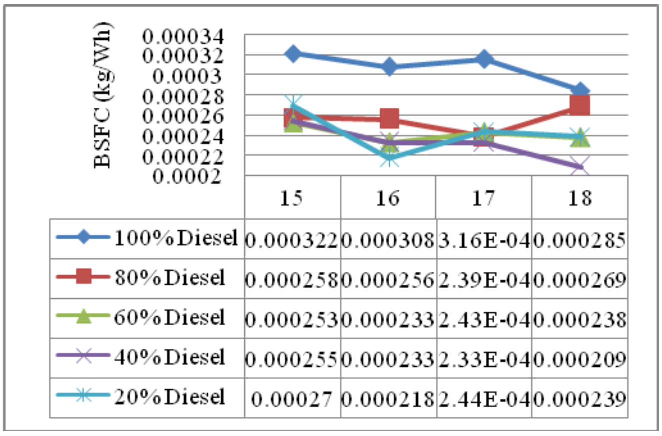


Fig.14 Effect of Compression ratio on BSFC at 100% load

3) Effect on Volumetric Efficiency: The volumetric efficiency of the engine has been found to be decrease with an increase in the LPG flow rate at all the loads. This is due to the fact that a part of the cylinder space is occupied by the LPG, providing reduced space available for the incoming air. The volumetric efficiency of the engine is found to be less when LPG is inducted with the main fuels. The reason for low volumetric efficiency is because of the high velocity of hydrogen tends to displace the air. The volumetric efficiency is calculated as the ratio of actual volume of air passed into the engine to the swept volume.

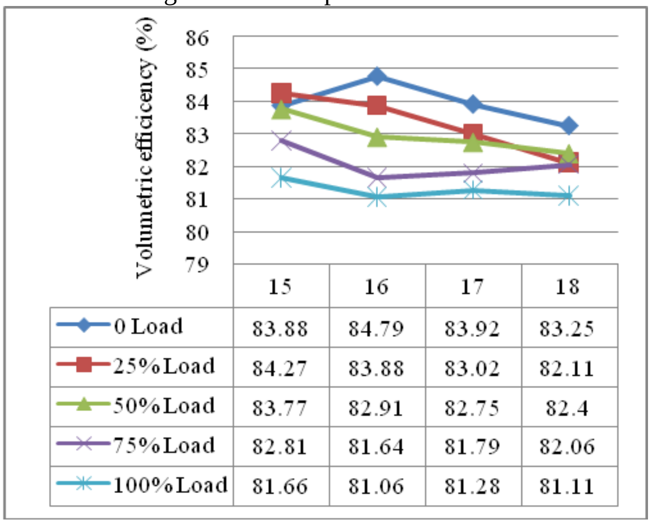
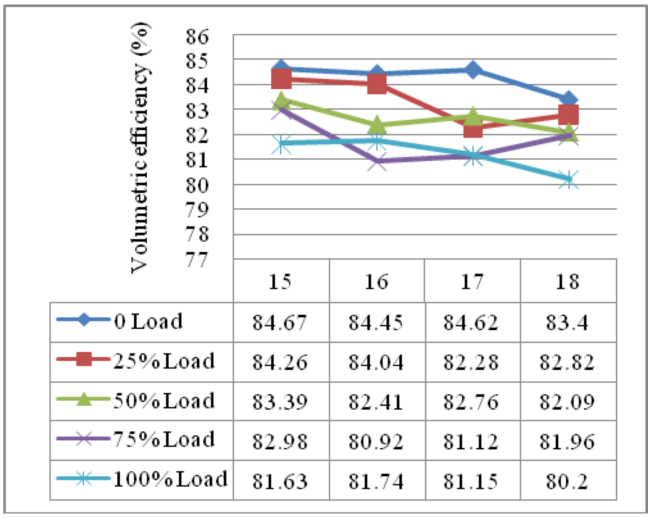
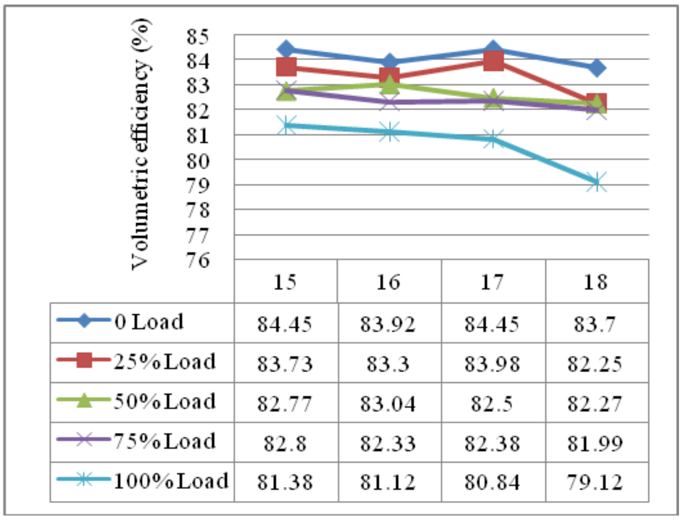


Fig. 15 Effect of Compression ratio on volumetric efficiency at 100% diesel.



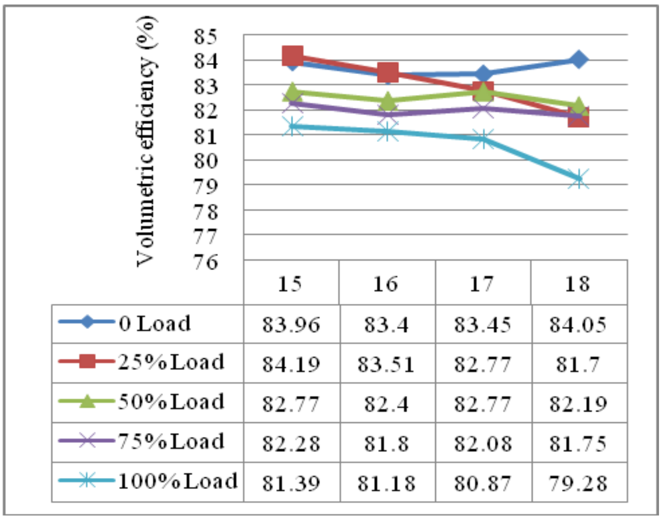
	15	16	17	18
0 Load	84.67	84.45	84.62	83.4
25%Load	84.26	84.04	82.28	82.82
50%Load	83.39	82.41	82.76	82.09
75%Load	82.98	80.92	81.12	81.96
100%Load	81.63	81.74	81.15	80.2

Fig. 16 Effect of Compression ratio on volumetric efficiency at 80% diesel



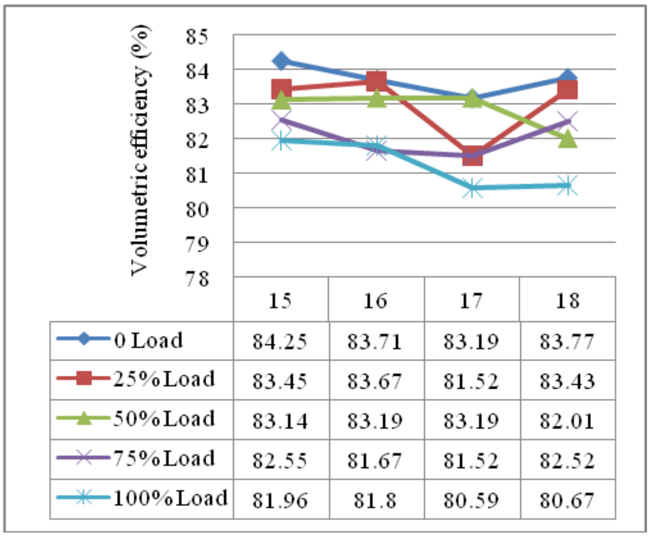
	15	16	17	18
0 Load	84.45	83.92	84.45	83.7
25%Load	83.73	83.3	83.98	82.25
50%Load	82.77	83.04	82.5	82.27
75%Load	82.8	82.33	82.38	81.99
100%Load	81.38	81.12	80.84	79.12

Fig.17 Effect of Compression ratio on volumetric efficiency at 60% diesel.



	15	16	17	18
0 Load	83.96	83.4	83.45	84.05
25%Load	84.19	83.51	82.77	81.7
50%Load	82.77	82.4	82.77	82.19
75%Load	82.28	81.8	82.08	81.75
100%Load	81.39	81.18	80.87	79.28

Fig. 18 Effect of Compression ratio on volumetric efficiency at 40% diesel.

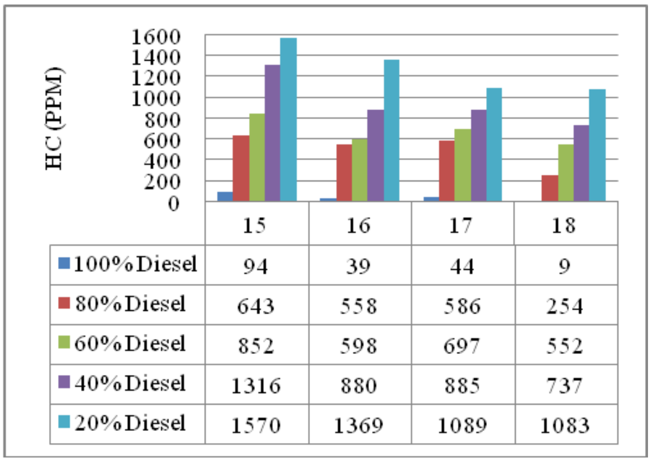


	15	16	17	18
0 Load	84.25	83.71	83.19	83.77
25%Load	83.45	83.67	81.52	83.43
50%Load	83.14	83.19	83.19	82.01
75%Load	82.55	81.67	81.52	82.52
100%Load	81.96	81.8	80.59	80.67

Fig.19 Effect of Compression ratio on volumetric efficiency at 20% diesel

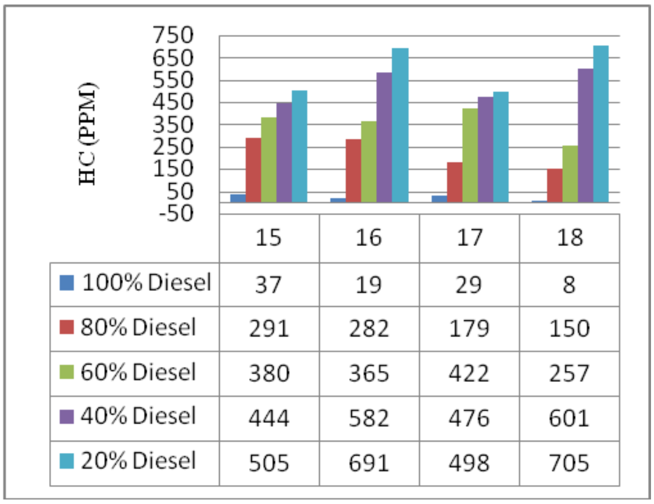
C. *EMISSION CHARACTERISTICS*

1) *Effect on HC emissions*



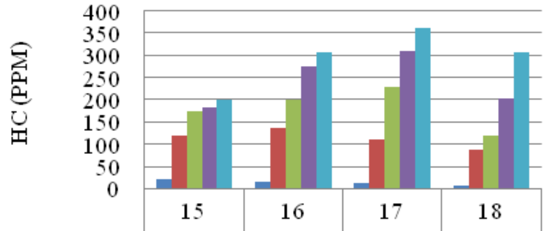
	15	16	17	18
100%Diesel	94	39	44	9
80%Diesel	643	558	586	254
60%Diesel	852	598	697	552
40%Diesel	1316	880	885	737
20%Diesel	1570	1369	1089	1083

Fig.20 Effect of Compression ratio on HC emission at 0% load.



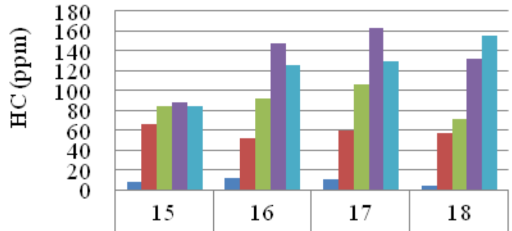
	15	16	17	18
100% Diesel	37	19	29	8
80% Diesel	291	282	179	150
60% Diesel	380	365	422	257
40% Diesel	444	582	476	601
20% Diesel	505	691	498	705

Fig.21 Effect of Compression ratio on HC emission at 25% load



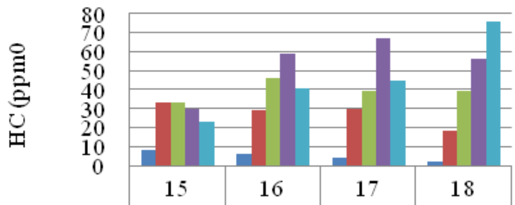
	15	16	17	18
100%Diesel	21	15	12	7
80%Diesel	118	135	110	88
60%Diesel	175	201	228	120
40%Diesel	184	276	310	202
20%Diesel	199	308	362	307

Fig. 22 Effect of Compression ratio on HC emission at 50% load.



	15	16	17	18
100%Diesel	8	12	11	5
80%Diesel	67	52	60	58
60%Diesel	84	92	106	71
40%Diesel	88	148	163	132
20%Diesel	85	126	130	156

Fig. 23 Effect of Compression ratio on HC emission at 75% load.

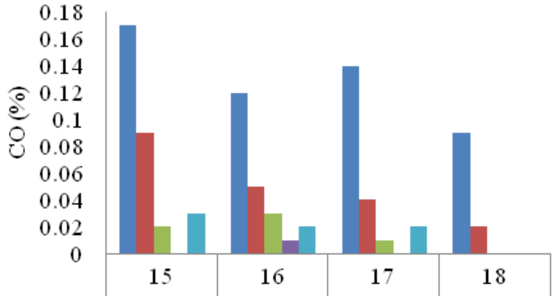


	15	16	17	18
100%Diesel	8	6	4	2
80%Diesel	33	29	30	18
60%Diesel	33	46	39	39
40%Diesel	30	59	67	56
20%Diesel	23	41	45	76

Fig.24 Effect of Compression ratio on HC emission at 100% load.

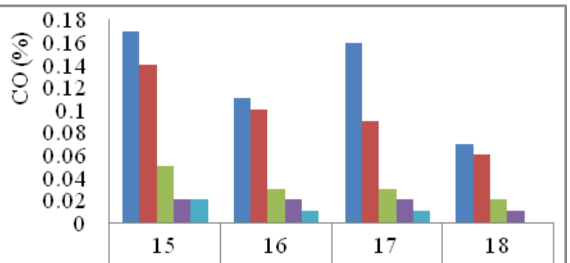
HC emissions consist of fuel that is completely unburned or partially burned. Typically, HC emissions are serious problems at light loads for diesel engines. HC emission arises when a part of the fuel inducted into the engine escapes combustion. During ignition delay period, fuel air mixtures becomes too rich to ignite and combust contribute to HC emissions. The comparison of HC emissions of all the pilot fuels in dual fuel mode was shown in Figures In dual fuel mode with increase in substitution of LPG, the HC emission increases. This may be due to reduction in fresh air with increase in LPG flow rate which results in incomplete combustion of the richer mixture. This is because of the flammability limit of the lean homogenous charge which leads to incomplete combustion or absence of combustion. With the increase of engine load, there is a sharp decrease of HC emissions under dual fuel operation. This is the result of the increase of burned gas temperature that helps oxidize efficiently the unburned hydrocarbons. Also as compression ratio increases HC emission decreases.

2) Effect on CO emission



	15	16	17	18
0 Load	0.17	0.12	0.14	0.09
25%Load	0.09	0.05	0.04	0.02
50%Load	0.02	0.03	0.01	0
75%Load	0	0.01	0	0
100%Load	0.03	0.02	0.02	0

Fig. 25 Effect of Compression ratio on CO emission at 100% Diesel.



	15	16	17	18
0 Load	0.17	0.11	0.16	0.07
25%Load	0.14	0.1	0.09	0.06
50%Load	0.05	0.03	0.03	0.02
75%Load	0.02	0.02	0.02	0.01
100%Load	0.02	0.01	0.01	0

Fig. 26 Effect of Compression ratio on CO emission at 80% Diesel.

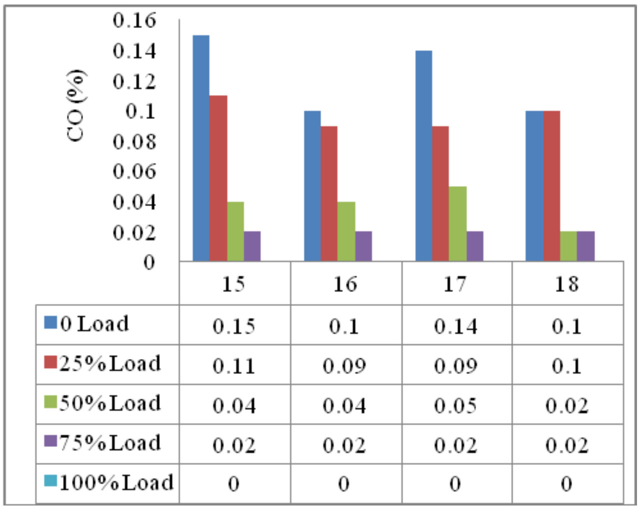


Fig. 27 Effect of Compression ratio on CO emission at 60% Diesel.

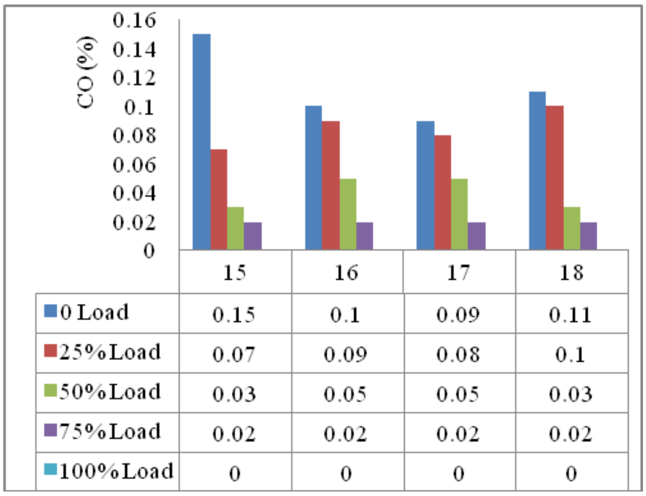


Fig. 28 Effect of Compression ratio on CO emission at 40% Diesel.

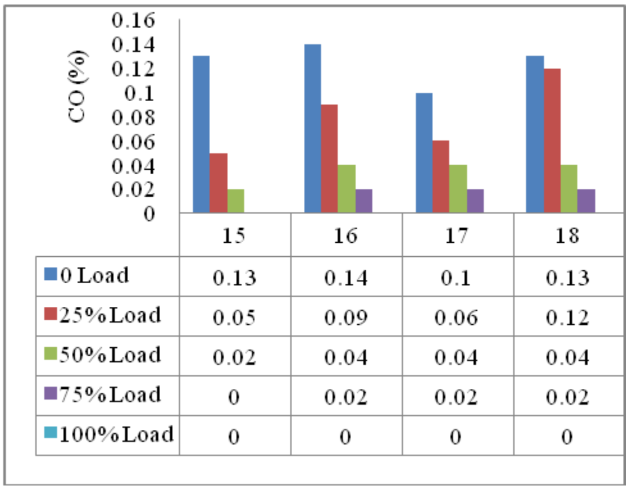


Fig. 29 Effect of Compression ratio on CO emission at 20% Diesel.

Carbon monoxide (CO) is formed due to deficiency of oxygen during combustion. Due to deficiency of oxygen the carbon present in fuel does not contribute fully in combustion process and coming out in the form of CO from the engine exhaust. As known, the rate of CO formation depends on the unburned gaseous fuel availability and the mixture temperature. As shown in Figures CO emissions are higher for dual fuel mode at low and moderate loads. However, CO concentration for dual fuel mode decreases with the increase of engine load as a result of the improvement of gaseous fuel utilization. On the other hand, for very high loads, because of locally very rich mixtures in conventional diesel, which result in bad combustion, the CO emissions are significantly higher. Less carbon monoxide emission indicates a proper combustion of the fuel.

3) Effect on CO_2 Emission

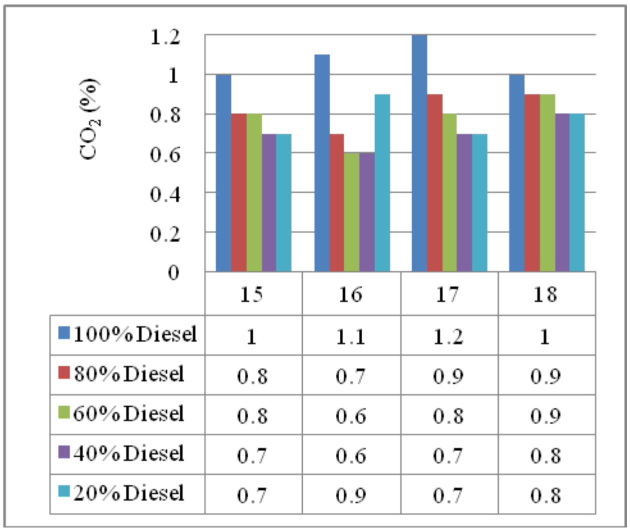


Fig.30 Effect of Compression ratio on CO_2 emission at 0 load.

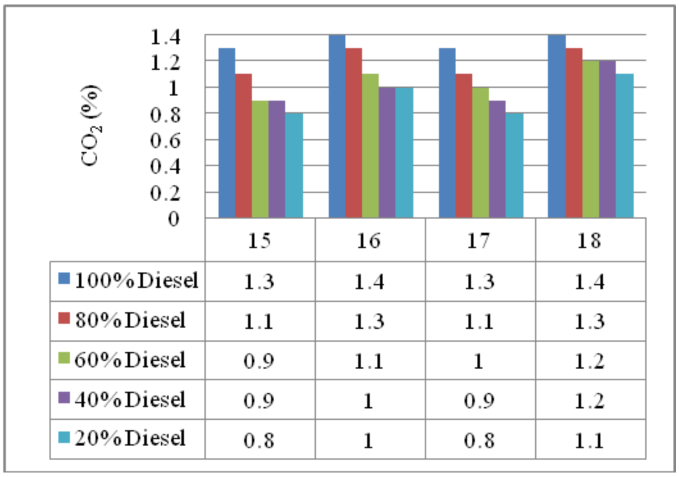
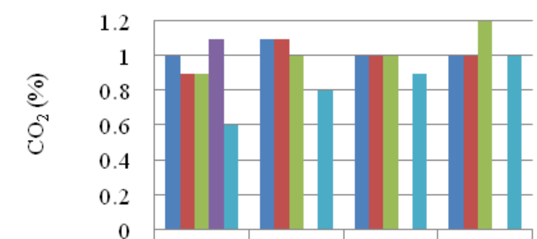
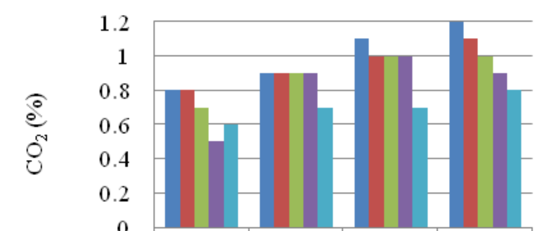


Fig.31 Effect of Compression ratio on CO_2 emission at 25% load.



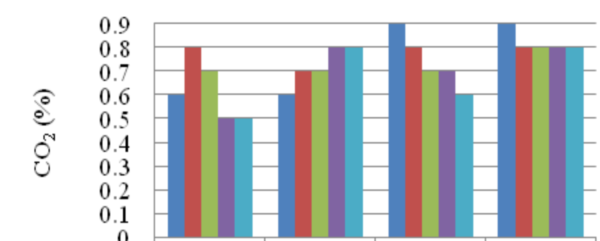
	15	16	17	18
100%Diesel	1	1.1	1	1
80%Diesel	0.9	1.1	1	1
60%Diesel	0.9	1	1	1.2
40%Diesel	1.1			
20%Diesel	0.6	0.8	0.9	1

Fig.32 Effect of Compression ratio on CO_2 emission at 50% load.



	15	16	17	18
100%Diesel	0.8	0.9	1.1	1.2
80%Diesel	0.8	0.9	1	1.1
60%Diesel	0.7	0.9	1	1
40%Diesel	0.5	0.9	1	0.9
20%Diesel	0.6	0.7	0.7	0.8

Fig. 33 Effect of Compression ratio on CO_2 emission at 75% load.



	15	16	17	18
100%Diesel	0.6	0.6	0.9	0.9
80%Diesel	0.8	0.7	0.8	0.8
60%Diesel	0.7	0.7	0.7	0.8
40%Diesel	0.5	0.8	0.7	0.8
20%Diesel	0.5	0.8	0.6	0.8

Fig.34 Effect of Compression ratio on CO_2 emission at 100% load.

Carbon dioxide (CO_2) is the most prominent human made Greenhouse gas. The present results confirm that the use of LPG in a dual fuel engine is an interesting technique to reduce this greenhouse gas especially at high loads. As due to high temperature achieved during combustion of LPG, the CO get oxidized and converted into CO_2. As pilot fuel quantity increases CO_2 emission also increases at all load conditions.

IV. CONCLUSION

Conclusion made from the above results are Dual fuel diesel engines suffer from problems such as high BSFC, poor brake thermal efficiency, and high HC and CO emissions at low load conditions. Increase in pilot fuel quantity results increase in cylinder pressure. Reduction of CO and HC emission is due to increase in amount of pilot fuel quantity. Increase in pilot fuel mass results higher CO_2 emission. An increase in compression ratio, pilot fuel quantity and load improves brake thermal efficiency and reduces BSFC, HC and CO emissions.

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