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PERFORMANCE AND EMISSION CHARACTERISTICS OF A VARIABLE COMPRESSION RATIO (VCR) SI ENGINE

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ABSTRACT: The compression ratio displays a dominant role in the performance of reciprocating I.C engine. All the methods to increase the power output bring along with them a host of various other problems. For instance, increasing engine speed imposes dynamic load factors and increased wear thereby reducing reliability and life. High turbo-charging results in very high peak pressures and also higher thermal loads. One method of solving high-pressure problem encountered when the specific output is increased is to reduce the compression ratio at full load but at the same time keep it sufficiently high for good starting and part load condition. Thus a fixed compression ratio engine cannot meet the requirements of high specific output and hence felt is the need for a variable compression ratio engine. The effect of compression ratio on brake thermal efficiency, CO & NOX are analyzed using Greaves MK20 SI Engine (2.28kW). The engine has a fixed compression ratio of 4.8, but the cylinder head has been modified to operate at compression ratios 3.6 to 7.4 in this project work. This engine has been used to investigate the effect of different fuels operating at three compression ratios of 4.0, 4.8 and 6.0.

KEYWORDS: Greaves MK20, Compression ratio, Fuel Efficiency, Emission Characteristics, Thermal Efficiency.

1 INTRODUCTION

The concept of a variable compression ratio engine has been the goal of designers since the inception of internal combustion engines. Over the past several decades, numerous designs for varying the compression ratio have been proposed. Most of them were either impractical or too complicated to evaluate on an engine though a few of them were tried experimentally and adapted in limited production. The reason for exploring this technology is used to meet the different situation on road use and maximize the fuel economy. For example, at low engine speed, the speed of car usually low and air intake is inefficient. The engine have to increase its compression ratio so that the power output is higher due to high pressure produced from combustion process, also air and fuel used will be less compare to low compression ratio to get more power. At high engine speed and vehicle speed, the engine has to work harder to get more power. But the combustion chamber will get hotter at the same time, the fuel may burn itself and cause knocking while the compression ratio is too high at the combustion process, so that engine gets hurt. In order to prevent this, lower compression ratio is required to lower the temperature of combustion chamber. To satisfy both conditions, we have to use variable compression ratio engine. The VCR and small engine displacement are used to achieve high energy-conversion efficiency at low power levels. High efficiency at low power levels is important for achieving high vehicle mileage because automobile engines operate at low power levels most of the time. The VCR is also used to prevent knock at high power levels (with a low compression ratio) and to improve efficiency at low power levels (with a high compression ratio). The spark-ignition port-fuel-injected VCR engine operates efficiently and cleanly on gasoline and a range of alternative fuels. All else being equal, an internal combustion engine with a high compression ratio makes more efficient use of the energy in the fuel, and also produces more power, than one with a lower compression ratio. However, at high engine loads, that high compression causes knock, ping and other forms of power-robbing untimed ignition. The advantages of variable compression ratio are well documented and understood in the automotive industry. Yet there are no such engines on the market today. The reason for this is that most VCR concepts do not provide a reliable sealing between piston and cylinder liner. In order to contain the high pressure of an internal combustion, the entire sealing surface of the cylinder liners must continuously be swept. Otherwise the sealing will not be properly lubricated, resulting in abnormal wear, and the ring might be sticking to the liner, resulting in engine breakdown.

2 DESIGN OF VARIABLE COMPRESSION RATIO ENGINE

2.1 SELECTION OF ENGINE

- A side valve engine is selected as it is easy to vary the compression ratio
- If the normal compression ratio is less, the clearance volume will be large and it will have space for modifications like introduction of an auxiliary chamber.
- Service and other facilities for the purchased engine should be available immediately.
- If the engine operates with two fuels, the effect of fuels can also be investigated.

With all the above in mind, the Greaves MK-20 engine with the following specifications is chosen.

- MAKE = Greaves Engine (MK-20)
- TYPE = Side valve
- POWER = 2.2 kW
- SPEED = 3000 rpm
- BORE = 68 mm
- STROKE = 50 mm
- DISPLACEMENT = 192 cc
- COMPRESSION RATIO = 4.8
- FUELS PROPOSED = Petrol, Kerosene

2.2 DESIGN OF NEW CYLINDER HEAD FOR VARIABLE OPERATION OF COMPRESSION RATIO

The swept volume of the engine is kept constant at 192cc and the Compression ratio is varied by varying the clearance volume by the introduction of an auxiliary chamber. As this will only reduce the compression ratio, the new cylinder head is redesigned to have a lower clearance volume of 30cc and then the clearance volume is increased by adding extra volume in the auxiliary chamber. The plunger is raised by means of a square thread 3mm pitch with a nut fixed and the lift can be measured by rotating the wheel provided on the square thread. Table 1 shows the relationship between the rotation of the wheel, lift of the plunger, the clearance volume and the compression ratio. The compression ration can be varied from 7.4 at the lowest position of the plunger to 3.6 at the highest position.

Table 1. Relationship Between Rotation Of Wheels and Lift Of The Plunger

CR	3.6	4.0	4.4	4.8	5.2	5.6	6.0	7.4
Clearance Volume	73.846	64	56-47	50.53	45.71	41.739	38.4	30
Additional value added	43.846	34	26.47	20.53	15.71	11.739	8.4	0
Lift of the plunger(mm)	22.33	17.32	13.48	10.45	7.99	5.98	4.279	0
Rotation of the wheel (degrees)	2679.6	2078.28	1618.17	1253.88	953.7	727.63	513	0

Calculation of lift and angle of rotation of the wheel for a compression ratio of 4.8 is shown below:

Clearance at C.R 7.4 = Vs/r-1 = 192/6.4 = 30 cc.

Clearance at CR 4.8 = Vs/M = 192/3.8= 50.5 cc.

Change in clearance volume = 50.5 - 30 = 20.5 cc

Diameter of plunger = 50 mm.

Area of plunger = $\pi/4\ (5.0^2) = 19.64\ cm^2$.

Lift = 20.5/19.64 =1.043 cm.

Pitch = 3 mm

Rotations required = 1.0437;   0.3 revolutions = 3.48 revolutions

Rotations required in degree = 3.47 * 360 = 1253 degrees.

2.3 DESIGN OF CYLINDER HEAD THICKNESS

Ultimate tensile strength of IP131	=	173 N/mm^2
Yield Strength of IP131	=	86.5 N/mm^2
Maximum Pressure in the cylinder	=	25 Bar
Area of the cylinder head	=	6600 mm^2
Force acting on the cylinder head	=	16.5 kN

2.4 DESIGN OF FINS

The fins are designed for the maximum power output of 2.2 kW. From the fuel consumption test, the following measurements are made.

Time for 10 cc flow of fuel = 20 seconds

Exhaust temperature = 500 °C

Room temperature = 30 °C

Air Fuel Ratio = 10: 1

Surface temperature of the cylinder head ts = 710 °C

The dimensions of the Cylinder fins on the annular surface of the cylinder are:

Type of cylinder fins = annular

Number of fins = 7

Inner radius = 34 mm

Length of fins = 23.5 mm

Thickness = 3 mm

From the data book,

CoefficientoftherrnalconductivityforlP131 (LM13) = 204W/mK.

Convective heat transfer coefficient at a velocity of 5m/s = 100 W/m^2K

1. Heat Input

Qi = mf * Cv =1.584 * 43500 / 3600= 19.14 kJ/s

2. Brake output = 2.2 KW = 2.2 kJ/s

3. Heat Lost through annular fins

From the heat and mass transfer data book (C.P.Kothandaraman, 2012) and

Heat Transfer (J.P.Holman 2003, 3rd Reprint),

Efficiency of fins = 90%

Heat transfer through seven fins Q = η * h * As * (tex – tr) * 7

= 0.9 * 100 * 14608 * (10 - 6) * (1000 - 30) * 7

= 8.89 kJ/ s

4. Heat Carried away by exhaust gases:

Q3 = mex * C pex * (tex - tr)

= {1.584 * 11 * 1.05 * (500 - 40)} / (3600)

= 2.34 kJ/S

5. Heat lost due to Radiation

Assumed to be 20% of heat input,

Q4 = Q1 * 0.2

= 19.14 * 0.2

= 3.83 kJ/s.

6. Heat loss through cylinder head fins

Q5 = Q1 - (Q2 + Q3 + Q4)

= 19.14 - (2.2 + 2.34 + 3.83 + 8.89)

= 1.88 kJ/s.

There are seven rectangular annular fins of 7 mm thickness of length 25 mm and breadth 35 mm.

Therefore perimeter P = 2 * (35 + 7) = 84 mm.

Cross sectional area = 35 * 7 = 245 mm^2

m = {(h'p) / (k *A)}(1/2)

= {(100 * 84 * 10-3) / (205 * 2.45 * 10 - 6)}(1/2)

= 12. 93

Total Heat transfer	= 7 * (hpkA) 1/2 * tanh ml * (ts - tr)
Therefore Qtotal	= 1.4 kJ/s.
The required heat transfer	= 1.88 kJ/s.

This implies that more fins have to be provided. Due to the constraints arising from the provision for bolts, spark plugs and the piezo electric pick up this could not be done. Hence it is suggested that the length of the fins can be increased from 25 to 35 mm to meet the requirement of cooling. But this was not possible too. So the forced cooling technique to increase heat transfer has been adopted using a blower.

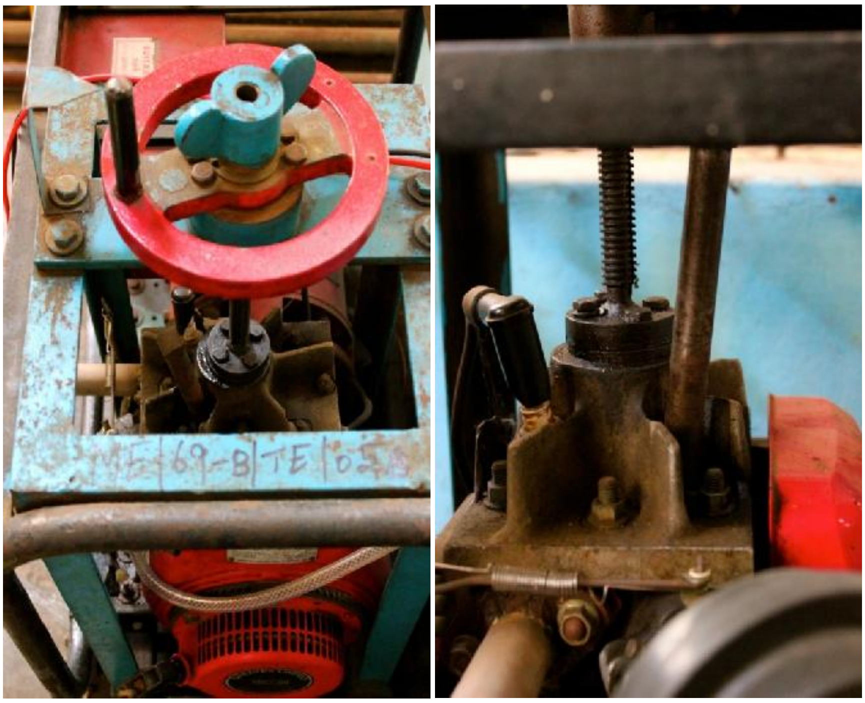

Fig. 1. Lead Screw Mechanism

Fig. 2. Exhaust Setup for Gas-Pickup

3 EXPERIMENTAL INVESTIGATIONS

The tests are conducted with four fuels:

- Petrol
- Kerosene.
- 5% Ethanol blend with Petrol
- 10% Ethanol blend with Petrol

For each fuel, the test procedure is as follows:

- o The rise for the desired compression ratio is set.

- The fuels and the recommended jet are selected.
- Starting from no load, the engine is loaded in steps till full load is attained with the speed being maintained constant at 3000rpm.
- For each load the following measurements are made
- Time for 10cc of fuel consumption
- The voltmeter and ammeter readings
- The exhaust gas temperature
- Exhaust gas analysis using CRYPTON five gas analyzer shown in figure 2 & 3. Exhaust gas analyzer consists of a probe provided with a long polymer cable connected to the analyzer consist of a probe provided with a long polymer cable connected to the analyzer by means of a filter. Exhaust gas analyzer measures NOX and UBHC in ppm and CO, CO_2 and O_2 in percentage by volume. It also measures the a/f ratio and the equivalent ratio. Software is provided with the facility of recording data over a period of a time 1min to 3min the time interval between two readings can also be changed. The data can be stored and printed.
- The experiment is repeated for different compression ratios.
- The readings are tabulated in tables 2 to 13 for petrol, kerosene and gasohol fuels.

Fig. 3. Experimental Setup

Formula used for performing calculations:

- Brake power = V * I/ηg Watts
  where ηg is the efficiency of generator.
- T.F.O 10/tf * (ρf/1000) * 3600 Kg/hr.
  where tf stands for 10cc of fuel consumptions,
  ρf stands for density of fuel in gm/cc.
- Brake thermal efficiency
  ηBT = [(Bp*3600*100) / (T.F.C*CV*1000)]%
  Where CV is calorific value of fuel in kJ/Kg.

Table 2. Test Results For Petrol, C.R=4.0, N=3000rpm

S.No	V (volt)	I (A)	η_g (%)	BP (Watt)	T_f (S)	η_{BT} (%)	Emissions				
							NO_X ppm	UBHC ppm	CO %vol	CO_2 %vol	O_2 % vol
1	230	0	0	0	40	0	101	322	3.76	12	1.03
2	230	1	0.41	560.9	36	6.29	111	336	4.43	12.2	0.54
3	230	2	0.54	851.8	31	8.22	106	343	6.3	11.3	0.4
4	230	3	0.63	1095.2	26	8.87	98	580	9.12	9	0.35
5	230	4	0.68	1352.9	19	9.07	95	746	11.34	7.9	0.22
6	230	5	0.73	1575.3	17	9.37	88	821	12.02	7.1	0.16
7	230	6	0.78	1769.2	15	9.83	86	1154	12.57	6.4	0.14
8	230	7	0.81	1987.7	13	10.66	79	1340	13.24	5.8	0.11

Table 3. Test Results For Kerosene, C.R=4.0, N=3000rpm

S.No	V (volt)	I (A)	η_g (%)	BP (Watt)	T_f (S)	η_{BT} (%)	Emissions				
							NO_X ppm	UBHC ppm	CO %vol	CO_2 %vol	O_2 % vol
1	230	0	0	0	42	0	96	302	4.03	12.2	0.28
2	230	1	0.41	560.9	37	5.88	116	331	6.88	11	0.19
3	230	2	0.54	851.8	31	7.19	99	293	8.69	9.7	0.13
4	230	3	0.63	1095.2	27	7.97	97	387	9.31	9.2	0.09
5	230	4	0.68	1352.9	23	8.93	88	609	12.09	8.6	0.06
6	230	5	0.73	1575.3	21	9.23	85	811	14.22	7.8	0.05
7	230	6	0.78	1769.2	18	9.81	81	1266	15.48	6.4	0.03
8	230	7	0.81	1987.7	15	11.02	76	1464	16.33	5.7	0.02

Table 4. Test Results For 5% Ethanol Blend, C.R=4.0, N=3000rpm

S.No	V (volt)	I (A)	ηg (%)	BP (Watt)	Tf (S)	ηBT (%)	Emissions				
							NO_X ppm	UBHC ppm	CO %vol	CO_2 %vol	O_2 % vol
1	230	0	0	0	38	0	117	302	3.19	13.2	0.31
2	230	1	0.41	560.9	34	6.04	98	394	6.75	11.1	0.26
3	230	2	0.54	851.8	28	7.55	126	563	8.52	10.3	0.32
4	230	3	0.63	1095.2	23	7.97	119	877	9.75	9.3	0.21
5	230	4	0.68	1352.9	18	8.74	104	971	10.36	8.8	0.17
6	230	5	0.73	1575.3	16	8.96	112	1078	11.48	7.9	0.1
7	230	6	0.78	1769.2	14	10.04	119	1274	11.98	7.2	0.07
8	230	7	0.81	1987.7	11	11.12	125	1460	12.36	6.6	0.06

Table 5. Test Results For 10% Ethanol Blend, C.R=4.0, N=3000rpm

S.No	V (volt)	I (A)	η_g (%)	BP (Watt)	T_f (S)	η_{BT} (%)	Emissions				
							NO_X ppm	UBHC ppm	CO %vol	CO_2 %vol	O_2 % vol
1	230	0	0	0	36	0	142	339	5.1	13.2	0.31
2	230	1	0.41	560.9	30	5.42	129	423	7.12	11.1	0.23
3	230	2	0.54	851.8	26	7.13	116	590	8.33	10.3	0.36
4	230	3	0.63	1095.2	21	7.4	111	664	10.26	9.3	0.21
5	230	4	0.68	1352.9	17	8.39	104	791	11.2	8.1	0.12
6	230	5	0.73	1575.3	15	8.54	97	880	11.69	6.3	0.07
7	230	6	0.78	1769.2	12	9.68	88	1279	11.83	5.9	0.04
8	230	7	0.81	1987.7	9	10.86	83	1411	12.44	5.5	0.03

Table 6. Test Results For Petrol, C.R=4.8, N=3000rpm

S.No	V (volt)	I (A)	η_g (%)	BP (Watt)	T_f (S)	η_{BT} (%)	Emissions				
							NO_X ppm	UBHC ppm	CO %vol	CO_2 %vol	O_2 % vol
1	230	0	0	0	47	0	83	388	2.51	13.5	0.35
2	230	1	0.41	560.9	41	7.28	91	412	4.52	12.1	0.28
3	230	2	0.54	851.8	34	8.15	108	581	6.78	9.8	0.17
4	230	3	0.63	1095.2	28	9.71	116	707	8.1	10.2	0.09
5	230	4	0.68	1352.9	23	10.29	129	892	9.37	9.4	0.12
6	230	5	0.73	1575.3	18	10.8	132	1004	12.19	7.7	0.1
7	230	6	0.78	1769.2	11	11.06	121	1176	12.79	6.5	0.08
8	230	7	0.81	1987.7	7	11.39	110	1367	13.55	5.8	0.07

Table 7. Test Results For Kerosene, C.R=4.8, N=3000rpm

S.No	V (volt)	I (A)	η_g (%)	BP (Watt)	T_f (S)	η_{BT} (%)	Emissions				
							NO_X ppm	UBHC ppm	CO %vol	CO_2 %vol	O_2 % vol
1	230	0	0	0	48	0	93	298	1.47	14.2	0.66
2	230	1	0.41	560.9	42	7.34	113	464	5.57	10.8	0.19
3	230	2	0.54	851.8	34	9.02	122	561	7.46	11	0.15
4	230	3	0.63	1095.2	29	9.89	138	683	9.12	9.4	0.16
5	230	4	0.68	1352.9	20	10.55	119	740	11.42	8.1	0.12
6	230	5	0.73	1575.3	17	10.71	111	984	12.59	7.3	0.07
7	230	6	0.78	1769.2	13	10.93	103	1401	12.99	6.7	0.04
8	230	7	0.81	1987.7	10	11.21	96	1126	13.36	6.1	0.01

Table 8. Test Results For 5% Ethanol Blend, C.R=4.8, N=3000rpm

S.No	V (volt)	I (A)	η_g (%)	BP (Watt)	T_f (S)	η_{BT} (%)	Emissions				
							NO_X ppm	UBHC ppm	CO %vol	CO_2 %vol	O_2 % vol
1	230	0	0	0	44	0	95	391	3.51	13	0.72
2	230	1	0.41	560.9	38	6.2	111	489	6.69	10.5	0.53
3	230	2	0.54	851.8	30	7.44	120	601	7.6	10.4	0.24
4	230	3	0.63	1095.2	26	8.29	133	643	8.41	10	0.12
5	230	4	0.68	1352.9	22	9.32	141	802	12.53	7.4	0.09
6	230	5	0.73	1575.3	18	10.32	113	988	13.61	6.5	0.06
7	230	6	0.78	1769.2	15	10.87	104	1216	13.98	6.1	0.05
8	230	7	0.81	1987.7	11	11.29	99	1437	14.35	5.8	0.03

Table 9. Test Results For 10% Ethanol Blend, C.R=4.8, N=3000rpm

S.No	V (volt)	I (A)	η_g (%)	BP (Watt)	T_f (S)	η_{BT} (%)	Emissions				
							NO_X ppm	UBHC ppm	CO %vol	CO_2 %vol	O_2 % vol
1	230	0	0	0	39	0	107	473	4.63	12.2	0.62
2	230	1	0.41	560.9	32	5.78	115	553	6.2	10.4	0.4
3	230	2	0.54	851.8	27	7.4	124	691	7.97	10.3	0.22
4	230	3	0.63	1095.2	24	8.46	139	838	8.24	10.1	0.11
5	230	4	0.68	1352.9	20	8.87	143	960	11.96	7.9	0.09
6	230	5	0.73	1575.3	16	9.11	121	1009	12.06	8	0.06
7	230	6	0.78	1769.2	13	9.86	117	1157	12.45	7.6	0.03
8	230	7	0.81	1987.7	10	10.77	111	1381	12.92	7.1	0.02

Table 10. Test Results For Petrol, C.R=6.0, N=3000rpm

S.No	V (volt)	I (A)	η_g (%)	BP (Watt)	T_f (S)	η_{BT} (%)	Emissions				
							NO_X ppm	UBHC ppm	CO %vol	CO_2 %vol	O_2 % vol
1	230	0	0	0	40	0	126	472	5.3	12	0.27
2	230	1	0.41	560.9	30	5.51	90	321	6.75	11.1	0.26
3	230	2	0.54	851.8	25	6.97	107	603	8.33	9.2	0.3
4	230	3	0.63	1095.2	21	7.53	121	574	8.67	9.9	0.09
5	230	4	0.68	1352.9	18	7.97	113	882	9.99	9	0.11
6	230	5	0.73	1575.3	15	8.69	104	1076	11.51	8.1	0.12
7	230	6	0.78	1769.2	13	9.54	96	1242	12.03	7.9	0.1
8	230	7	0.81	1987.7	12	10.83	93	1459	12.66	7.3	0.08

Table 11. Test Results For Kerosene, C.R=6.0, N=3000rpm

S.No	V (volt)	I (A)	η_g (%)	BP (Watt)	T_f (S)	η_{BT} (%)	Emissions				
							NO_X ppm	UBHC ppm	CO %vol	CO_2 %vol	O_2 % vol
1	230	0	0	0	53	0	92	391	0.91	13.3	1.99
2	230	1	0.41	560.9	42	7.46	100	466	2.82	13	0.9
3	230	2	0.54	851.8	31	8.36	109	619	3.2	13.5	0.39
4	230	3	0.63	1095.2	25	8.67	124	681	7.79	10.2	0.25
5	230	4	0.68	1352.9	23	10.29	131	833	9.29	9.4	0.2
6	230	5	0.73	1575.3	19	10.64	122	1002	13.03	7.1	0.14
7	230	6	0.78	1769.2	17	10.97	101	1160	14.11	6.6	0.11
8	230	7	0.81	1987.7	15	11.27	90	1316	15.95	5.9	0.08

Table 12. Test Results For 5% Ethanol Blend, C.R=6.0, N=3000rpm

S.No	V (volt)	I (A)	η_g (%)	BP (Watt)	T_f (S)	η_{BT} (%)	Emissions				
							NO_X ppm	UBHC ppm	CO %vol	CO_2 %vol	O_2 % vol
1	230	0	0	0	48	0	99	370	3.01	12.2	1.26
2	230	1	0.41	560.9	44	7.18	103	436	5.9	10.9	0.86
3	230	2	0.54	851.8	38	9.42	109	570	7.16	10.1	0.39
4	230	3	0.63	1095.2	35	9.78	121	703	8.22	9.01	0.25
5	230	4	0.68	1352.9	27	10.04	125	791	11.1	8.45	0.24
6	230	5	0.73	1575.3	19	10.65	113	859	12.22	7.9	0.11
7	230	6	0.78	1769.2	17	10.97	107	1032	12.79	7.3	0.08
8	230	7	0.81	1987.7	14	11.23	99	1366	13.46	6.1	0.03

Table 13. Test Results For 10% Ethanol Blend, C.R=6.0, N=3000rpm

S.No	V (volt)	I (A)	η_g (%)	BP (Watt)	T_f (S)	η_{BT} (%)	Emissions				
							NO_X ppm	UBHC ppm	CO %vol	CO_2 %vol	O_2 % vol
1	230	0	0	0	48	0	103	422	1.19	13.8	1.14
2	230	1	0.41	560.9	35	7.46	112	583	2.06	13.6	0.8
3	230	2	0.54	851.8	30	8.36	119	704	7.32	10.7	0.24
4	230	3	0.63	1095.2	24	8.67	127	870	8.32	9.9	0.15
5	230	4	0.68	1352.9	22	9.54	133	1086	9.09	9.4	0.17
6	230	5	0.73	1575.3	16	10.29	125	1209	11.36	8.7	0.15
7	230	6	0.78	1769.2	15	10.64	113	1361	12.94	8.1	0.11
8	230	7	0.81	1987.7	12	11.07	103	1517	13.81	7.3	0.06

4 RESULTS AND DISCUSSION

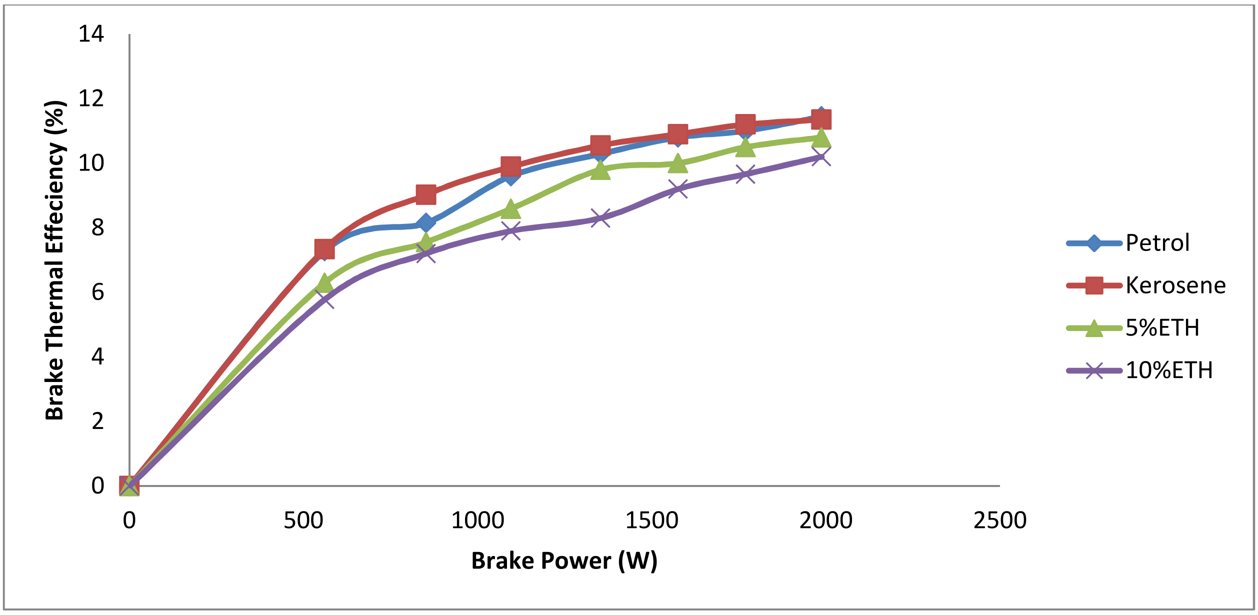

Fig. 4. Brake Thermal Efficiency vs. Brake Power at CR = 4.8

- Figure 4 shows the variation in brake thermal efficiency with load for various fuels at compression ratio of 4.8.
- It is observed from the figure, the brake thermal efficiency is highest for gasoline for all loads.
- It is expected because gasoline has the highest calorific value. Kerosene fuel shows better results compared to gasohol (gasoline-Ethanol) blends, because of lower latent heat of vaporization.

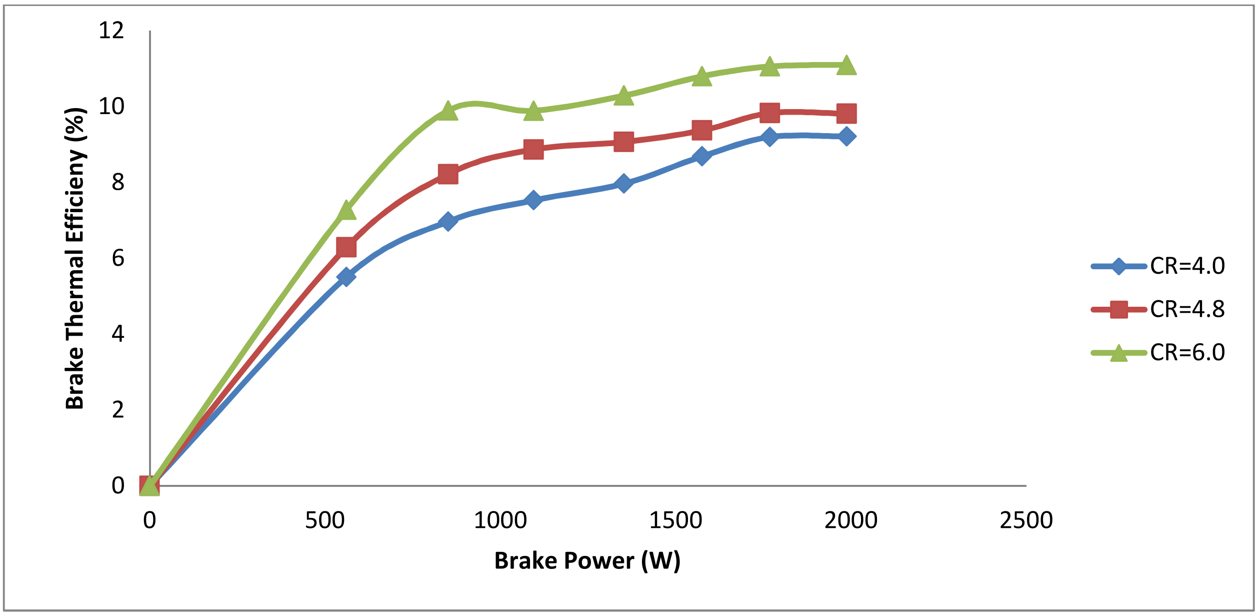

Fig. 5. Brake Thermal Efficiency vs. Brake Power for Petrol

- Figure 5 shows variation of brake thermal efficiency for petrol at different compression ratios.
- It is seen that brake thermal efficiency is high at high compression ratios at all loads.
- This is due the fact that the temperature is high before the beginning of combustion which results in better vaporization and mixing.

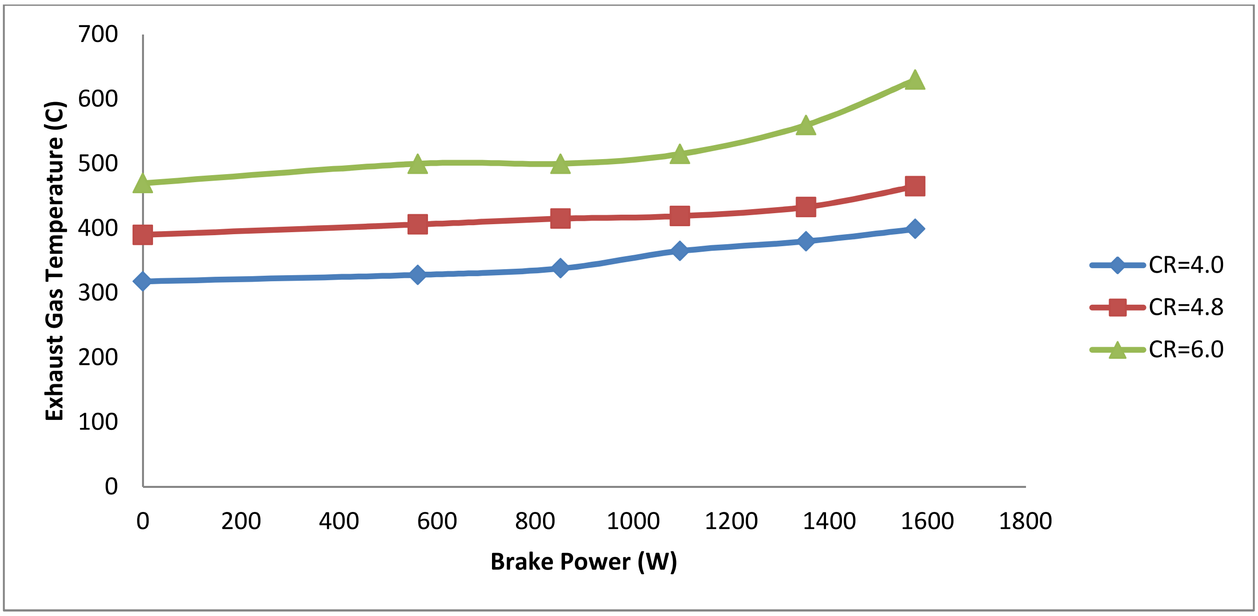

Fig. 6. Exhaust Gas Temperature vs. Brake Power for Petrol

- Figure 6 shows the variation of exhaust gas temperature with brake power at various compression ratios.
- The exhaust gas temperature increases with load and also with compression ratio.
- It is seen that the exhaust temperature is very high (500-600°C) at higher compression ratio (6.0) and hence heat carried away by the exhaust gases is high.
- This is one of the reasons for the lower brake thermal efficiency for this engine.

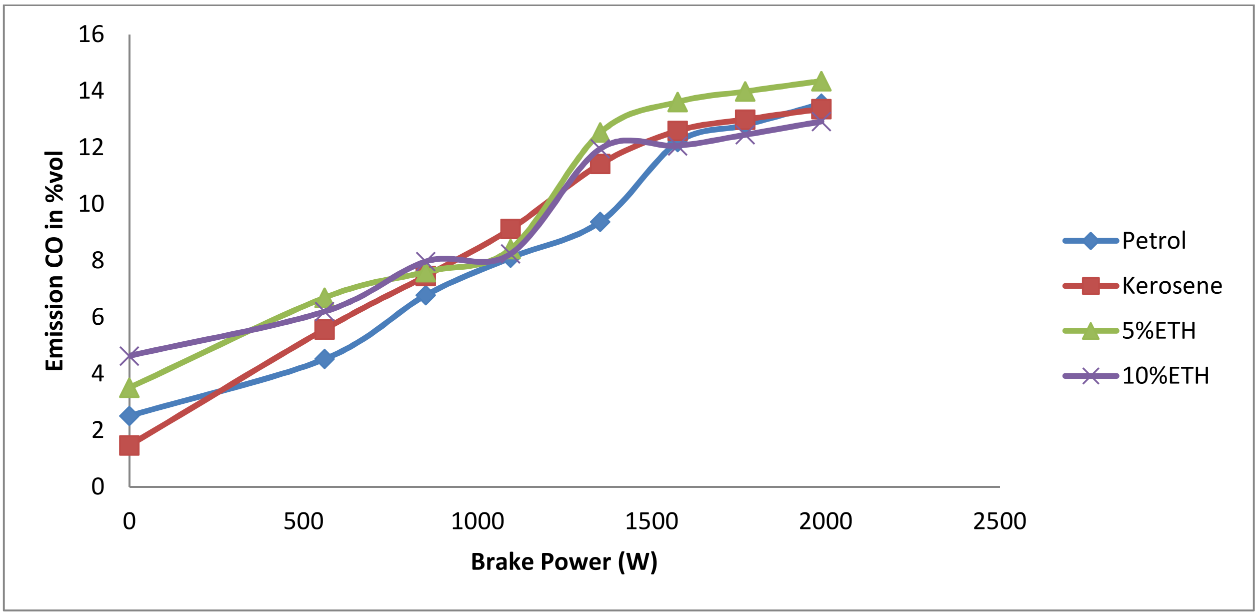

Fig. 7. Emission Characteristics of CO at CR = 4.8

- Figure 7 shows the variation of CO for different fuels at compression ratio 4.8.
- It is observed from the above graph, the level of CO increases with increase in loads for all fuels but no definite trend could be seen between different fuels.
- As shown in the graph, the CO emission gradually increases as brake power increases, for petrol.
- Whereas for ethanol blend, the CO emissions increases at a higher rate compared to other fuels.

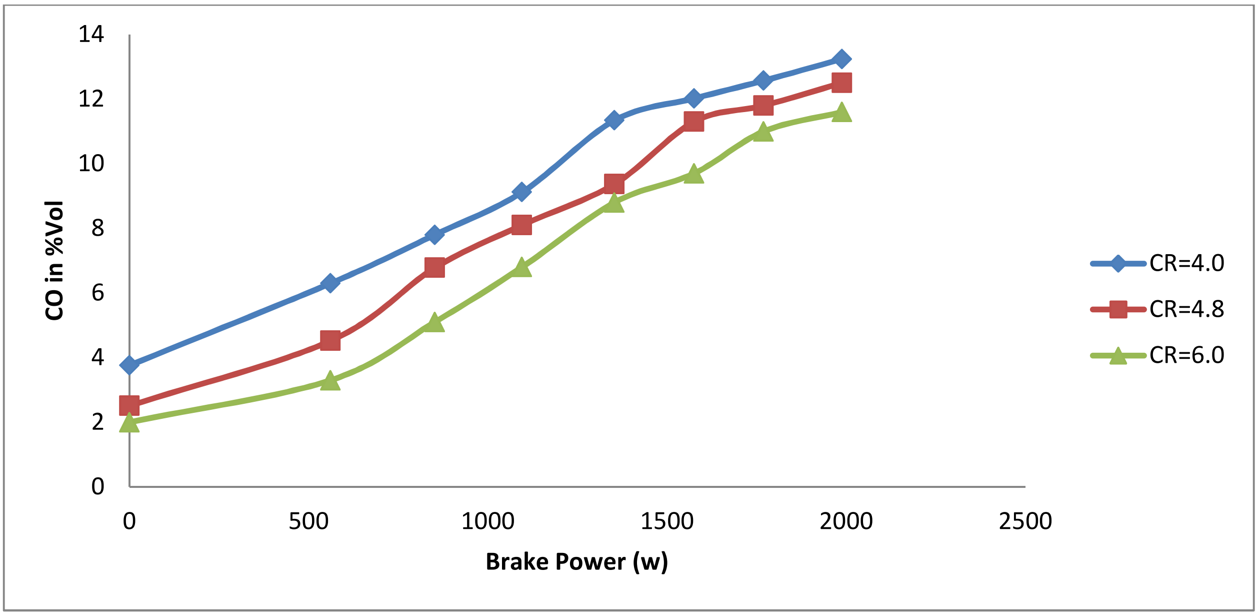

Fig. 8. Emission Characteristics of CO at various CR for Petrol

- Figures 8 show the variation of CO at different compression ratios for petrol.
- It is seen that the level of CO increases with load. As the compression ratio increases, the level of CO decreases, which shows that the combustion is better at higher compression ratios.

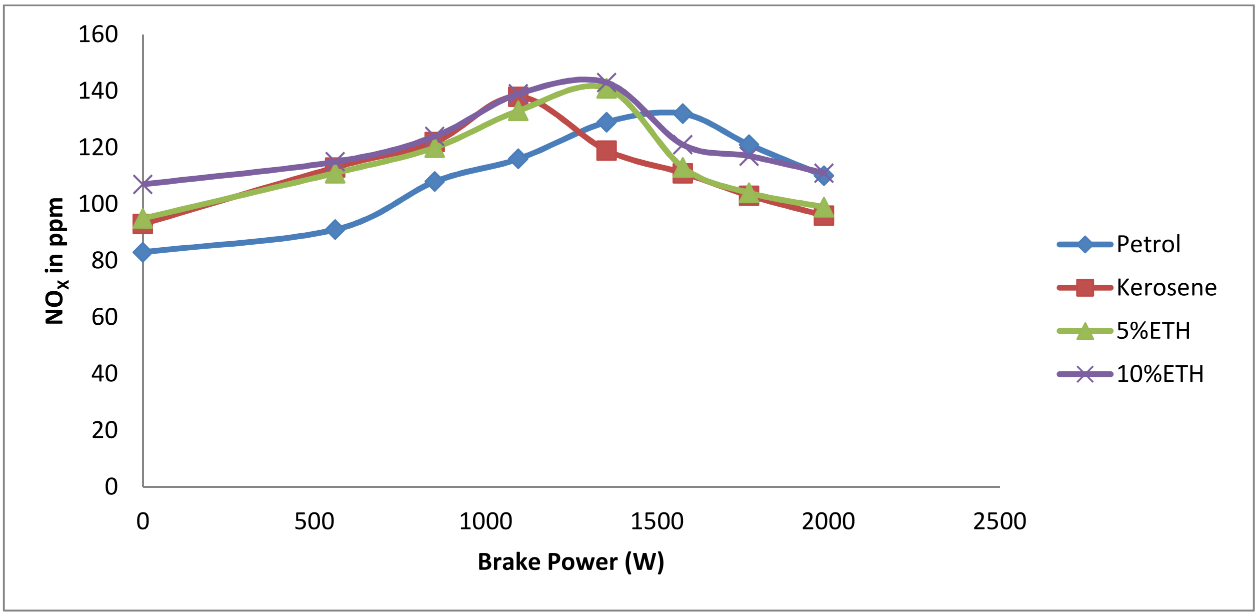

Fig. 9. Emission Characteristics of NO_X at CR = 4.8

- The figure 9 shows the NOx increases with load, reaches peak at nearly 50% full load (900-1000 W).
- After reaching the peak value (50% load) it gradually decreases commonly for all fuels.

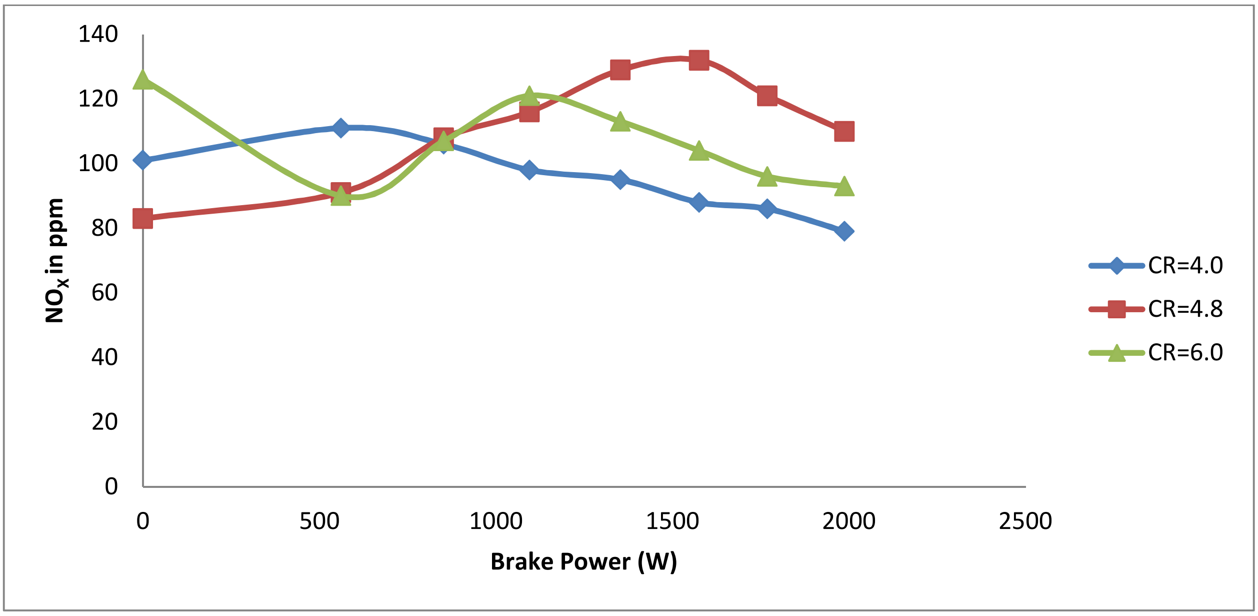

Fig. 10. Emission Characteristics of NO_X at various CR for Petrol

- Figure 10 shows the variations of NOx at with various compression ratios for petrol.
- It is observed that NOx increases as the load increases, reaches a maximum value and then decreases with further increase in load.
- At any load, NOx is high for the higher compression ratio.
- This is natural as high temperatures are expected with high compression ratios.

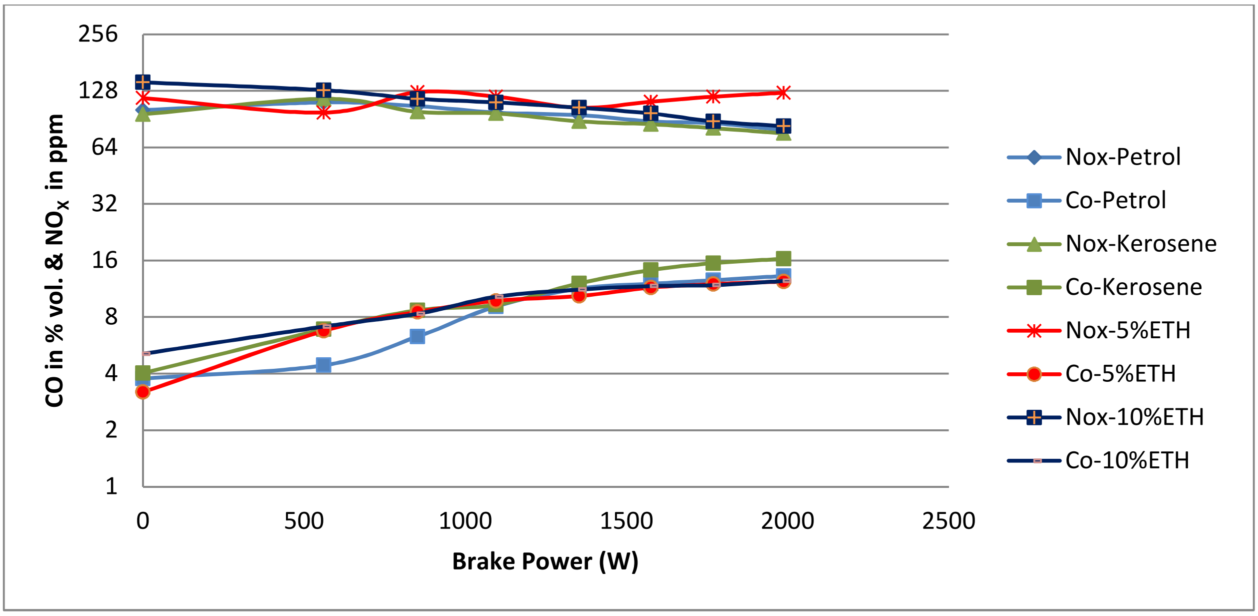

Fig. 11. Comparison of CO with NO_X at CR = 4.0

- Figure 11 shows the variation of NOx and CO with load for compression ratio 4.0 for various fuels used.
- The trend is same as in the case of compression ratio 4.0 except the peak NOx is higher in the case of compression ratios of 4.8 and 6.0.

5 Conclusion

- A variable compression ratio SI engine is fabricated with the compression ratio capable of being varied from 3.6 to 7.4.
- The tests were conducted using four fuels namely Petrol,
- Kerosene, 5% Ethanol blend, 10% Ethanol blend for compression ratios varying from 4.0 to 6.0.
- From the experiments conducted on the VCR SI engine fabricated, it is observed that the performance of the engine matches with the original engine.
- The maximum brake thermal efficiency of the engine is found to be abnormally low (around 11%).
- For Petrol, the compression ratio of 4.8 is found to be optimum from the point of view of emissions and efficiency.
- For Kerosene also, the optimum compression ratio is found to be 4.8.
- For Gasohol also, the optimum compression ratio is found to be 4.8

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