EFFECT OF USING DIFFERENT ETHANOL-GASOLINE FUEL BLENDS AT DIFFERENT COMPRESSION RATIOS ON SI ENGINE

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ABSTRACT

Using different ethanol-gasoline fuel blends, a VARICOMP engine was used to study the effect of varying the compression ratio on SI engine performance. The performance tests were carried out using different percentages of ethanol in gasoline fuel, up to 40%, under variable compression ratio conditions. The results show that the engine indicated power improves with ethanol addition. The maximum improvement occurs at 10% ethanol-90% gasoline fuel blend.

INTRODUCTION

The effect of ethanol on gasoline as a fuel additive has been studied through the years in spark ignition engines. It has been found that 10% ethanol in gasoline as a fuel additive improves the engine power by 5% [1]. Addition of ethanol to lead-free gasoline has resulted in an increase of fuel research octane number by 5 units for each 10% ethanol addition [1]. It has been further reported that the exhaust emission of carbon monoxide was reduced considerably (by about 30%).

Gasoline and diesel fuel additives that help reducing exhaust gas pollutants have been also investigated. Exhaust emissions were significantly lowered in the range of 45 to 93% for a fleet of 50, 1986-1987 model year cars designed for unleaded gasoline over a corresponding fleet of leaded fueled cars of 1980 model year average [2]. A synthetic fuel (made up of soybean oil or animal fat and hydrolyzed to glycerol) was tailored to match petroleum diesel fuel and blended with ethanol [3]. Such a mixture of the synthetic fuel and ethanol has revealed its superiority to petroleum diesel fuel; better vehicle efficiency, better cetane quality, lower combustion noise, better cold start characteristics and better exhaust odor and emissions.

Fuel injection has become increasingly used in cars throughout the world because of its higher performance and its easier control of exhaust emissions and improved fuel economy [4]. The use of alcohols and ethers as gasoline blending components instead of lead additives has become increasingly a target to many oil refineries and fuel blenders all over the world. This is done in an attempt to evaluate alcohol supplements in blend with gasoline in order to define their scope and ensure market satisfaction, Palmer [1] ran a wide range of vehicle performance tests on oxygenated fuel blends generated by the British Petroleum company. Savage et al [5], reported their results of using multi-point port injection alcohol fumigation of a four-stroke cycle turbocharged diesel engine in which the fumigation injection cycle was varied. Such different fumigation cycles lead to significant differences in the engines pressure-volume history and alcohol energy replacement tolerance. The engine was fumigated with both industrial grade ethanol and methanol and complete performance and emission data (excluding aldehydes) were measured.

McCall et al [6], concluded that an automobile could not be operated solely on dissociated or steam reformed methanol over the entire required power range and that the use of reformed methanol, compared to liquid methanol, may result in a small improvement in thermal efficiency although dissociated methanol is a better fuel than steam reformed methanol for use in spark ignition engine. Also, they found that the use of dissociated or steam reformed methanol may result in lower exhaust emissions compared to liquid methanol. In a comparison study, Kaneko et al [7] found that the maximum methanol and propane engine output is higher by approximately 20% and 8% respectively than the gasoline engine. They also concluded that the brake thermal efficiency of the methanol and propane engine is better than the base gasoline engine. It was found that the engine exhaust emission levels of methanol fuel are lower than gasoline, especially for NO_x and HC.

Hamdan et al [8] had studied the effect of ethanol addition on the performance of diesel and gasoline engines. A TD43 engine was used, which can be converted from gasoline to diesel version. The performance tests were carried out using different fuel blends of ethanol-gasoline and ethanol-diesel. The maximum percentage of ethanol used was 15%. They found that ethanol-gasoline blend has a higher effect on the engine performance than an ethanol-diesel blend. The best performance was achieved when 5% ethanol-gasoline blend was used, with thermal efficiency increase of 4% to 21%. However, all their tests were carried out under part load conditions.

The objective of the present work is to investigate the effect of varying the compression ratio on the engine performance working with different ethanol-gasoline fuel blends. It is also aimed to study the effect of changing ethanol percentage in an ethanol-gasoline fuel blend on engine performance.

EXPERIMENTAL SET-UP

The experiments were carried out using variable compression ratio engine (VARICOMP, model no. SA 306/013), manufactured by the Prodit company of Italy. The engine was designed to operate according to Otto or Diesel cycle. The compression ratio can be varied by shifting the mechanism of the cylinder head through a worm and wheel gear.

The engine speed was measured using an electronic RPM meter. The torque was measured using a load cell and strain gauge. A traditional flow meter was used to measure the fuel consumption. The air flow rate was measured using traditional air flow meter, consisting of calibrated nozzle, damping chamber and differential manometer.

The engine was provided with cooling unit, specially designed to keep the water inlet temperature at a pre-selected value. The engine load was varied using a hydraulic dynamometer. The engine and dynamometer were coupled by means of rubber joint so as to balance eventual residual misalignment between axes and to damp out any vibrations that may occur during transmission.

An oscilloscope, Topward 7000 series, was used to record the indicated pressure-volume diagram. The signal input of the oscilloscope was taken from a pressure transducer (P-3061), mounted on the cylinder head. The horizontal axis input signal comes from inductive proximity with linear output and cam mounted on the driving shaft. The technical data of the engine and pressure sensors are included in Appendix (A).

TEST PROCEDURE

Different fuel blends of pure ethanol (C_2H_5OH) and pure gasoline (64% Rheniformate, 11% low straight run nephtha and 25% Pentane) [9] were prepared and kept in a sealed glass container. The fuel blends were prepared with 0.0% ethanol up to 40% ethanol with an increment of 10%. Although no separation has been noticed after 72 hours, a fresh sample of the fuel blend to be tested was always prepared just before starting the exiperment. By so doing, one ensures that the fuel mixture is homogenous and the ethanol is not given a chance to react with water vapour.

In all experiments, the following procedure was carried out:

1- Preparing the fuel blend sample and filling the fuel tank.
2- Adjusting the engine to the required compression ratio.
3- Running the engine with fully-opened throttle (WOT) and maximum resisting torque applied to the engine dynamometer until we reach the minimum possible speed.
4- Varying the engine load (speed) by closing the hydraulic brake gradually till the required engine speed and torque obtained.
5- Recording the experiment raw data and taking photographs for the engine P-V diagram, upon reaching steady state conditions.

The measured quantities are:

- The fuel consumption; by recording the time needed to consume 20 cubic centimeter of fuel.
- The air flow rate; using the nozzle-water manometer arrangement.
- The driving torque
- The water inlet temperature
- The water exit temperature
- The engine speed

Table 1. Experimental Results.

Measured data					Calculated results								
CR	RPM	T N.m	t sec	Δp mm H_2O x10^3	m_f kg/s x10^4	m_a kg/s x10^3	BkW	IKW	η_m	η_{thb}	η_{thi}	bsfc kg/kWhr	isfc kg/kWhr
Pure Gasoline, L.C.V=44000 kJ/kg, ρ = 728.4 kg/m^3, Octan No. 87 (ref.[9])													
8	2196	14.8	30.7	67	4.6	6.6	3.4	4.7	71.4	17.3	22.3	0.49	0.36
10	2405	15.0	26.4	70	5.5	6.7	3.8	5.1	65.4	16.0	21.3	0.52	0.39
10	2810	15.1	24.9	82	5.7	7.33	4.4	5.8	78.4	17.7	22.5	0.47	0.38
10	3400	12.0	22.6	90	6.2	7.68	4.3	5.7	75.1	15.8	20.8	0.52	0.39
10 % Ethanol & 90 % Gasoline, L.C.V.= 42290 kJ/kg, ρ = 734.8 kg/m^3, Octan No. 91.8 .													
8	2141	14.5	17.4	75	8.4	7.02	3.25	6.46	50.3	9.1	18.1	0.93	0.47
10	2150	14.5	17.5	75	8.4	7.02	3.26	6.29	50.4	9.2	17.7	0.89	0.48
12	2121	14.5	17.3	70	8.5	6.78	3.22	6.2	61.9	9.2	17.2	0.95	0.49
20 % Ethanol & 80 % Gasoline, L.C.V.= 40580 kJ/kg, ρ = 741.3 kg/m^3, Octan No. 966.3													
8	2147	14.5	16.9	75	8.8	7.02	3.26	5.66	64.4	9.1	15.9	0.97	0.56
10	2143	14.5	17.0	70	8.7	7.4	3.28	5.86	65.4	9.2	16.6	0.95	0.53
12	2170	14.5	17.2	60	8.6	7.05	3.30	5.73	67.6	9.5	16.4	0.92	0.54
30 % Ethanol & 70 % Gasoline, L.C.V.= 38870 kJ/kg, ρ = 747.8 kg/m^3, Octan No. 99.3 .													
8	2175	14.4	16.9	70	8.7	7.61	3.3	5.5	60.0	9.8	16.3	0.95	0.57
10	2186	14.4	16.9	70	8.7	7.61	33.3	5.65	58.4	9.8	16.7	0.95	0.60
12	2194	14.6	17.1	70	8.8	7.5	3.33	5.7	58.4	9.7	16.7	0.95	0.56
13	2167	14.5	16.9	70	8.8	7.5	3.29	5.5	59.6	9.6	16.5	0.98	0.58
40 % Ethanol & 60 % Gasoline, L.C.V.= 37000 kJ/kg, ρ = 750.9 kg/m^3.													
8	2152	14.5	18.0	65	8.3	6.78	3.26	5.65	57.2	11.7	18.3	0.84	0.53
12	2144	14.5	18.8	69	8.1	6.53	3.25	5.9	55.2	10.8	19.5	0.90	0.49
13	2154	14.5	18.4	70	8.0	6.27	3.27	5.06	64.7	10.9	17.0	0.89	0.57
10	2174	14.5	18.6	75	7.6	6.53	3.3	5.4	62.7	11.7	19.2	0.83	0.53
10	2495	14.5	18.5	80	8.1	7.25	3.8	6.0	61.9	12.6	20.3	0.77	0.49
10	3000	14.5	18.0	90	8.3	7.69	4.5	6.3	74.5	14.6	20.4	0.67	0.37

RESULTS AND DISCUSSION

Sample of the data and results of experimental runs carried out in the present work are tabulated in Table (1) and presented in Figures (1) to (7).

Figure (1) shows the effect of adding ethanol to gasoline fuel on the P-V diagram (1-a, 1-b, 1-c & 1-d) and pressure-crank angle(Φ) diagram (1-e). The P-Φ diagram was obtained from the transformation from the P-V photographs as explained in appendix (B). It can be noticed from this figure that the 10% ethanol-gasoline fuel blend increases the maximum pressure over that of pure unleaded gasoline. However, the same figure shows that increasing ethanol percentage above 10% results in a decrease of the maximum pressure to a value even lower than that of pure unleaded gasoline. This may be explained as; the addition of ethanol to gasoline has two effects on the fuel blend properties; the first is an increase of the Octane number (see Table 1) since ethanol latent heat of evaporation is much greater than that of gasoline and accordingly, ethanol addition helps delaying the chain reactions of end gas [10]. The second is a decrease in the heating value. It is to be noticed that both effects have opposite roles on engine performance. The first effect is likely dominating up to ethanol percetage of 10%, after which the second effect starts to take role.

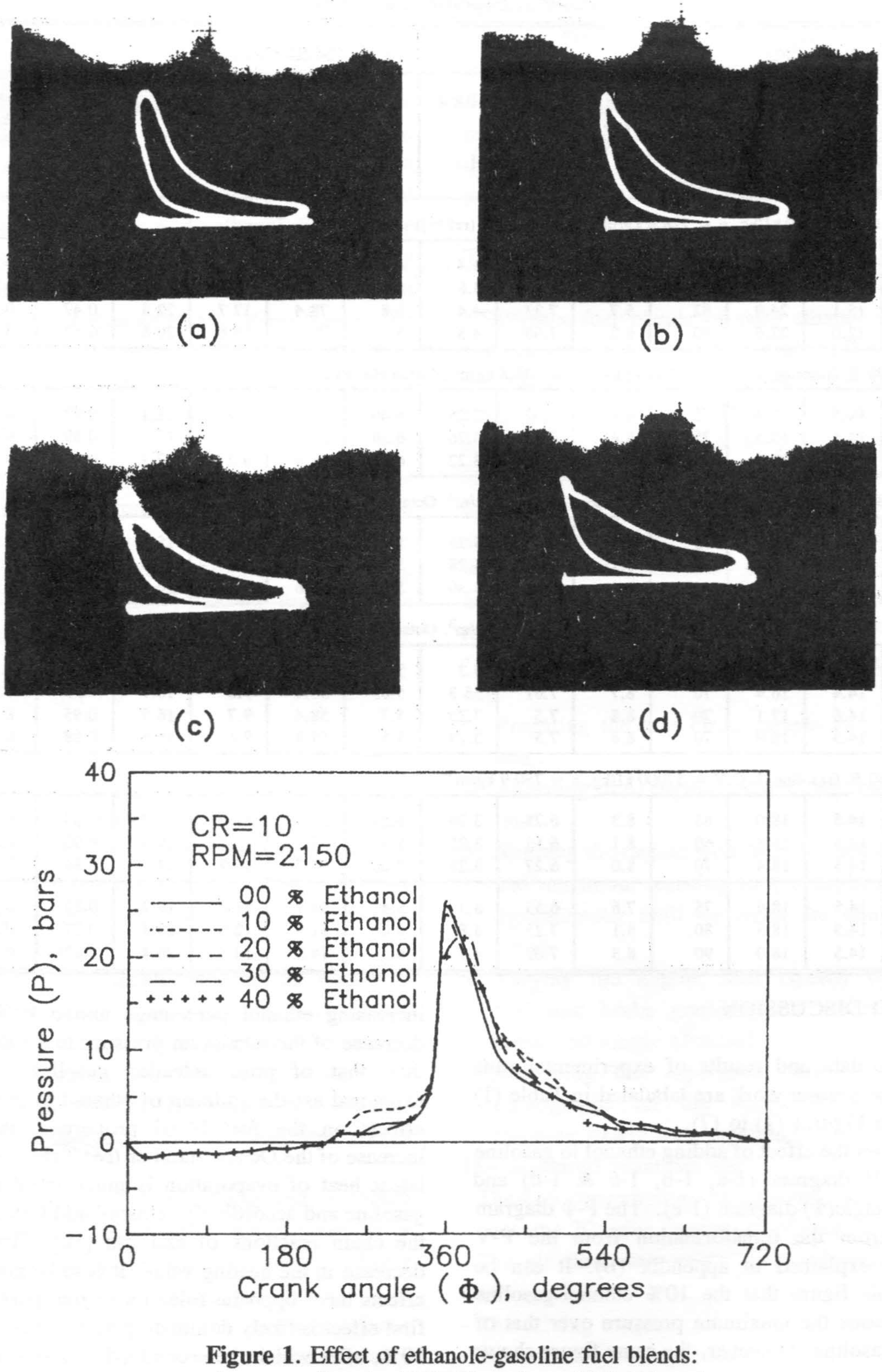

Figure 1. Effect of ethanole-gasoline fuel blends:
a- Pure gasoline b- 10% Ethanol; c- 20% Ethanol;
d- 30% Ethanol e- P-Φ for all blends used.

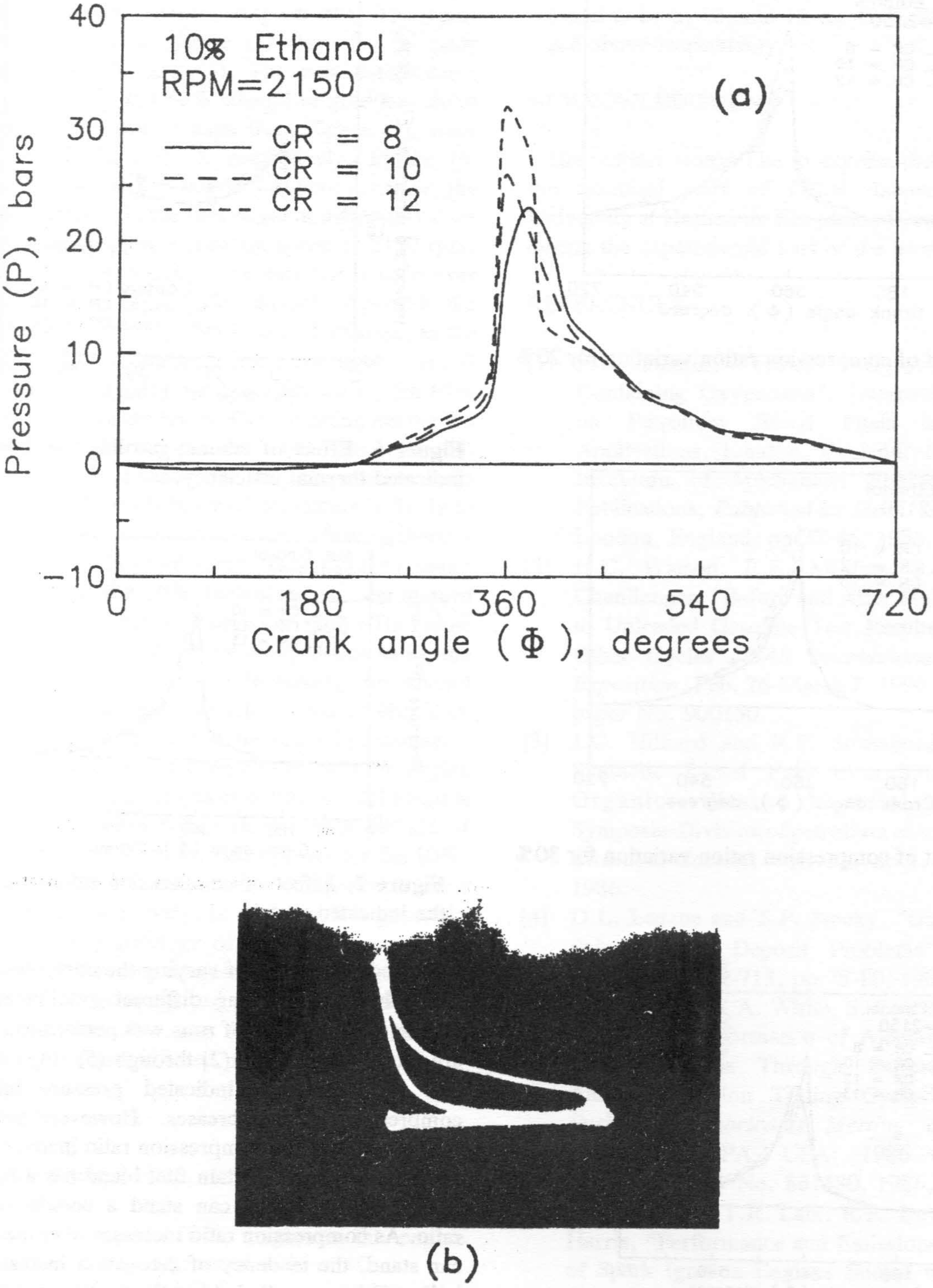

Figure 2. Effect of compression ratio varation
a- 10% Ethanol; b- P-V photo for CR = 12, 10% ethanol.

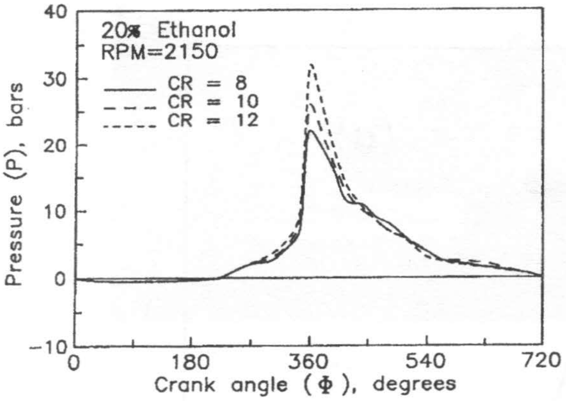

Figure 3. Effect of compression ration variation for 20% ethonal.

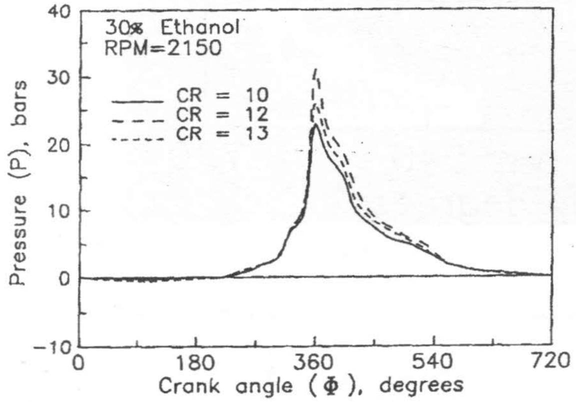

Figure 4. Effect of compression ration variation for 30% ethonal.

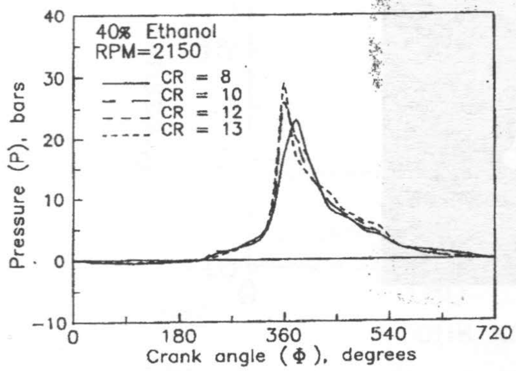

Figure 5. Effect of compression ration variation for 40% ethonal.

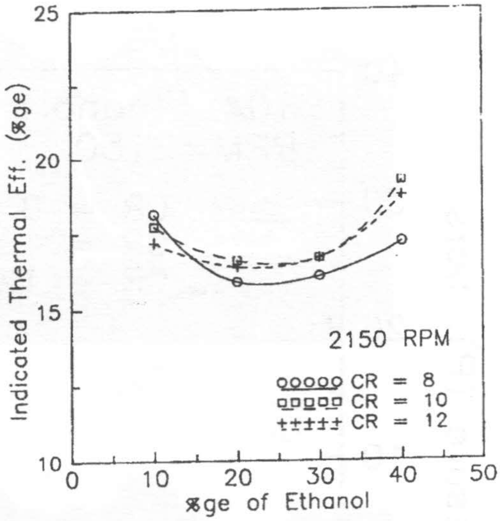

Figure 6. Effect of ethonal-gasoline fuel blends on the indicated thermal efficiency.

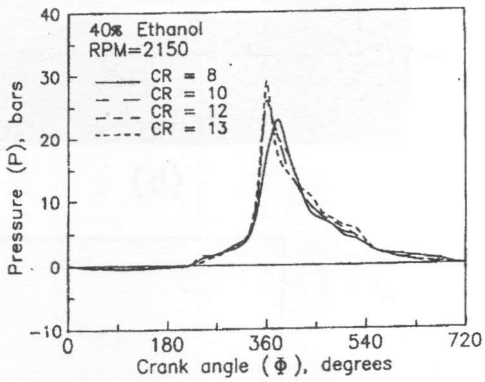

Figure 7. Effect of compression ration variation on the indicated power.

To study the effect of varying the compression ratio on the P-Φ diagram using different gasoline-ethanol fuel blends, different sets of runs was performed. The results are plotted in Figures (2) through (5). Figure (2) shows that the maximum indicated pressure increases as compression ratio increases. However, the indicated power decreases as compression ratio increases. This can be explained by: a certain fuel blend has a fixed Octane rating which in turn can stand a certain compression ratio. As compression ratio increases over the limit a fuel can stand, the tendency of detonation increases. This is indicated by a small shaking line in the photograph taken from oscilloscope and shown in Figure (2-b). From figure (2) with the assist of Table (1), it can be concluded that the best compression ratio for a 10% ethanol fuel blend is 8.

Figures (3), (4) and (5) show the effect of varying the compression ratio on the P-Φ diagram for fuel blends of 20%, 30% and 40% ethanol respectively. The same observations made from Figure (2) may also be made from Figures (3), (4) and (5). The best compression ratios for 20%, 30% and 40% ethanol to give maximum indicated power, as can be seen from Figure (7), were found to be 10, 12 and 12 respectively. Figure (6) represents the indicated thermal efficiency versus the percentage of ethanol in the fuel blend at different values of compression ratios for a constant speed of 2150 rpm. The figure shows, increasing the compression ratio over 8 for fuel blend above 20% ethanol improves the indicated thermal efficiency, while at 10% ethanol, as the compression ratio increases, less indicated thermal efficiency is obtained. This can be explained as; the 10% ethanol fuel blend has the lowest Octane rating among the fuel blends tested in the present work (see table 1). Therefore, for 10% ethanol, as the compression ratio increases over 8, the probability of detonation is likely to occur, thus resulting in a decrease in indicated thermal efficiency. For the other fuel blends, their Octane number is higher than that for 10% fuel blend, which in turn perform better at higher compression ratio. The same figure also shows that at the same compression ratio, the effeiciency curves decrease with increasing the ethanol percentage till it reaches a mimimum value, and then starts to increase with increasing the ethanol percentage.

The effect of changing the compression ratio on engine indicated power at different gasoline-ethanol fuel blend is given in figure (7). The figure shows, with the aid of table (1), that the maximum indicated power for the 10%, 20%, 30%, and 40% ethanol occurs at compression ratios 8, 10, 12, and 12 respectively. In addition, it is generally noticed that as the percentage of ethanol increases, the maximum indicated power decreases. This may be due to the fact that as percentage of ethanol increases, the heating value of fuel decreases (see table 1), which in turn results in a lower indicated power.

CONCLUSIONS

Based on the present study, the following conclusions are drawn:

- Ethanol addition, up to 10%, improves the engine indicated power at a compression ratio of 10.
- When the ethanol percentage increases over 20% as the compression ratio increases, the indicated power increases.
- For each fuel blend, there is an optimum compression ratio which gives a maximum indicated power. In the present study, these optimum compression ratios were found to be 8, 10, and 12 for 10%, 20%, 30% ethanol and above respectively.

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APPENDIX (A)

(1) Engine Specifications

Bore	92 mm
Stroke	85 mm
No. of Cylinders	1
Stroke Volume	565 cc
Compression Ratio	continuosly varied from 5:1 to 22:1
Maximum Power	8 kW at 3000 RPM

Forced Circulation Water Cooling
Forced Lubrication by Gear Pump
Overhead Camshaft
Flat-Top Combustion Chamber (Heron Type)
Parallel Valves Actuated through Valve Rockers
Fixed spark advance: 10 degrees before TDC

(2) Pressure Sensors

Piezoquartz Type P-3061 Transducer

Electric Input:	10-32 Volt DC, unregulated, 1.2 mA operating current
Electric Output:	0-5 Volt DC, 5 mA maximum
Output Impedance:	less than 1 ohm
Combined static error:	less than ±0.5% of full range output.

(3) Temperature Sensor

Three digital thermometers:	range of -50.5°C to +1999.9°C.
Semi-conductor probe	
Instrument resolution :	0.1°C
Power supply	220 Volt, 50 Hz

APPENDIX (B)

This appendix describes the technique of obtaining the pressure and crank angle values from an Oscilloscope photograph for the P-V diagram.

Figure (B) is a sample P-V diagram. First, the stroke volume (V_s) was determined, from which the stroke can readily be calculated. Such stroke was divided into a suitable number (N) of equal intervals. The piston displacement at any interval, say x is related to the crank angle Φ by [11]:

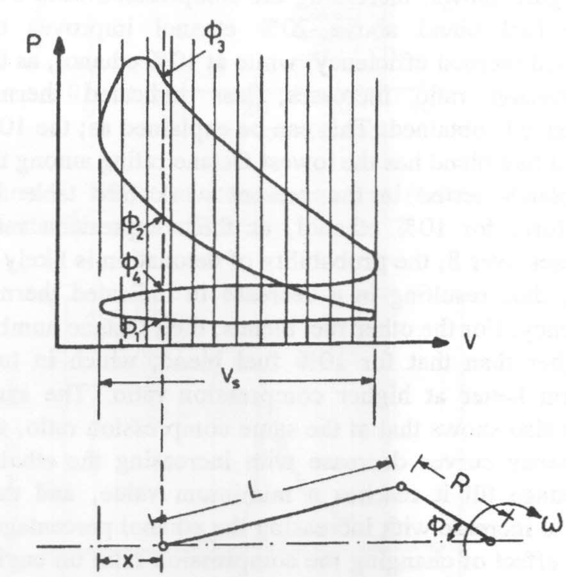

Figure B. The transformation from the P-V to the P-ϕ diagram.

$$X=(R+L)-\left[R\cos\phi+L\sqrt{1-\left(\frac{R\sin\phi}{L}\right)^2}\right] \quad \text{(B1)}$$

where R, L and Φ are the crank raduis, connecting rod length and crank angle respectively. It is to be noted that at each piston displacement x, there are four values for the crank angle ; one for the intake stroke (Φ1) which can be calculated directly from equation (B1). The other three values (Φ2, Φ3 and Φ4) represent the crank angles in the compression, expansion and exhaust strokes respectively, and are related to Φ1 as follows:

$$\begin{aligned}\Phi 2 &= 360 - \Phi 1\\ \Phi 3 &= 360 + \Phi 1\\ \Phi 4 &= 720 - \Phi 1\end{aligned} \quad \text{(B2)}$$

The corresponding pressures at each crank angle can be measured off the oscilloscope photograph (y-axis) taking into consideration the scale factor. The area under the P-V diagram, which represents the indicated power, was measured using planometer.