AEC001

Effects of Compression Ratio on the Performance of EFI Gasoline Engine Fueled with E85-Gasohol

Praphon Chooprasert* and Boonrit Prasartkaew

Combustion and Solar Energy Research Laboratory (CASE Lab.), Department of Mechanical Engineering,
Faculty of Engineering, Rajamangala University of Technology Thanyaburi, Klong 6, Patumthani, 12110, Thailand
*Corresponding Author: e-mail: praphon.tmec@gmail.com, Tel.: +66 804801040

Abstract

Nowadays, energy crisis and global warming become two major issues which substantially affect on all lives on the earth. They must be mitigated with the efficient means as soon as possible. One of the best ways to simultaneously address these serious problems is the renewable/alternative energy utilization via a high efficiency system. It can be said that these problems are attributed to the fossil fuel combustion, mainly in the internal combustion engines. Almost all internal combustion engines are reciprocation piston engine. From the theoretical analysis shown that, the compression ratio is proportion to the thermal efficiency and it is limited by the auto ignition, owing to the properties of fuel. This research aims to study on the performance of the variable-compression-ratio EFI petrol engine run on gasohol E85. Regarding its size in the near future transportation sector and hence due to its popularity, 1,500 cc engine has been selected for present experimental investigations. The comparison study was carried out to show how the performance obtained by comparing with the other fuels. The study results show that increasing of compression ratio has resulted in the increase of engine performance.

Keywords*:* Spark ignition engine, Engine performance test, Gasohol E85.

1. Introduction

Over the last few decades, the most major issues affecting on the human being are the energy crisis and environment problems. To mitigate or tackle these serious issues, renewable energy with the environmental friendly systems should be encouraged. Regarding the energy utilization, however, not only the energy conservation should be implemented but the energy efficiency should be also taken into account [1].

Almost all energy used in Thailand is imported from many countries. In 2011, more than 60% of all commercial primary energy requirements were imported. Among this primary energy, more than 80% of fuel used in Thailand was also imported and the usage trend was continuously increased. To address this issue, exploration of alternative fuel and renewable energy based fuel should be intently promoted. If this measure can be widely opted, the aforesaid issues can be possibly mitigated.

Ethanol has been used as a fuel for engines since 19th century. It is known as the most suited renewable, bio-based and eco-friendly fuel for spark-ignition (SI) engines. The most attractive properties of ethanol are that it can be produced from renewable energy sources such as sugar, cane, cassava, many types of waste biomass materials, corn and barley. In addition, it has positive influence on engine performance and reduces exhaust emissions owing to higher evaporation heat, octane number and flammability temperature [2].

Gasohol is one of the renewable energy based fuel which is mostly used for gasoline engines. Currently, there are three types of gasohol are being sold in the fuel stations, E10 (90% gasoline blended with 10% ethanol), E20 (80% gasoline blended with 20% ethanol) and E85 (15% gasoline blended with 85% ethanol). Each type of gasohol can be utilized efficiently via appropriate internal combustion engine which should be designed or modified for it.

The Thermodynamics theory demonstrates that the thermal efficiency of Otto cycle (theoretical cycle for gasoline engine) is direct proportion to the compression ratio (CR) [3]. However, the compression ratio of all conventional gasoline engines is limited by the uncontrolled knocking or pre-ignition phenomena. This behavior substantially affects on the engine performance. To overcome this problem, the high octane number fuel may be required.

Regarding the fuel octane number or anti-knock property of fuel, the literatures reveal that all types of gasohol have higher octane number, compared to gasoline fuel [2,3], as shown in Table 1. Therefore, using gasohol as fuel, the knocking phenomena will be reduced and the compression ratio may be increased. To know what is the compression ratio which is appropriate for E85-gasohol fuel, this paper presents the experimental study on the performance of the variable-compression-ratio EFI petrol engine.

Table 1. Properties of gasohol fuels [3]

Fuel Blend	Vol. % Ethanol	LHV KJ/g	Density g/cc	Wt.% Water	RON
ASTM	D5501	D240	D4052	E203	D2699
E0*	0	43.397	.7426	0.013	90.8
EEE#	0	42.890	.742	0	96.8
E10	10.46	41.47	.7449	.1289	95.6
E20	21	39.53	.7512	.2373	99.7
E50	49.7	34.38	.7666	.4947	104
E85	82.2	29.2	.7854	.7653	106
E85C	82.8	29.16	.7856	.8179	(-)
E100 x	96.6	26.7	.794	0.9	(-)
* 91 RON (87 (R+M)/2) Test fuel M52642 -Gage products					
# EPA TIER II EEE – Haltermann Products					
x Ethanol feed stock includes 2% denaturant					

There are many researches were carried out to study on the influence of ethanol blending to the gasoline fuel. He et.al. [5] investigated on the effect of ethanol blended gasoline fuels on emissions and catalyst conversion efficiencies in a spark ignition engine with an electronic fuel injection (EFI) system. Their results show that ethanol can decrease engine-out regulated emissions. The fuel containing 30% ethanol by volume can drastically reduce engine-out total hydrocarbon emissions, CO and NOx emissions, but unburned ethanol and acetaldehyde emissions increase. Moreover, the blended fuels can decrease brake specific energy consumption.

Bayraktar [6] investigated experimentally and theoretically on the effects of ethanol addition to gasoline on an SI engine performance and exhaust emissions. Experimental studies were carried out with the blends containing 1.5, 3, 4.5, 6, 7.5, 9, 10.5 and 12 vol% ethanol, whilst the numerical study was performed up to 21 vol% ethanol. His results obtained shown that among the various blends, the blend of 7.5% ethanol was the most suitable one from the engine performance and CO emissions points of view.

The effects of ethanol gasoline blended fuel on cold-start emissions of an SI engine were studied by Chen et.al. [7]. In their study, the content of E5, E10, E20, and E30 were used. Their results showed that HC and CO emissions decreased significantly with more ethanol than 20% added. However, for E40 the engine idling became unstable because the air-fuel mixture was too lean. Therefore, the ethanol content in gasoline for best cold-start emissions was determined to be at least 20% but no greater than 30%.

Tutak [8] experimentally investigated the potential of E85 fuelling in a diesel engine as a dual-fuelling system, E85 is introduced into the intake manifold using a port-fuel injector while diesel is injected directly into the cylinder. His study was performed using a three-cylinder a direct injection diesel engine. His objective aims to determine the operating parameters of the engine powered on E85 bio-ethanol fuel in dual fuel system. His results showed that with E85 fuel participation, NOx and soot emissions were reduced, whereas CO and HC emissions increased considerably. It was found that E85 participation in a combustible mixture reduced the excess air factor for the engine and this led to increased emissions of CO and HC, but decreased emissions of nitrogen oxides and soot.

Turkoz et.al. [9] investigated the best ignition timing in an SI engine using an E85 ethanol blend by altering the timing angle with respect to gasoline use regarding the output performance parameters such as power and efficiency. Their experiments were carried out on a 4-stroke, 4-cylinder spark ignition engine and the fuel orifice holes of the carburetor were suitably enlarged for the E85 ethanol blend. They reported that increasing the delay in ignition timing caused poorer combustion and hence more HC emissions and fuel consumption.

Yucesu et.al. [10] studied on the effects of ethanol–gasoline blends and compression ratio on engine performance and exhaust emissions in a single cylinder, four stroke, with variable compression ratio and spark ignition engine.

Koc et.al [11] also experimentally investigated on the effects of gasoline–ethanol blends (E50 and E85) on engine performance and pollutant emissions in a single cylinder four-stroke spark-ignition engine at two compression ratios (10:1 and 11:1). Their results showed that ethanol addition to gasoline increase the engine torque, power and fuel consumption and reduce carbon monoxide (CO), nitrogen oxides (NOx) and hydrocarbon (HC) emissions. It was also found that ethanol–gasoline blends allow increasing compression ratio (CR) without knock occurrence.

Costa and Sodré [12] investigated the influence of compression ratio on the performance of a spark ignition engine fuelled by a blend of 78% gasoline-22% ethanol (E22) or hydrous ethanol (E100) in a 1.0-L, eight-valve, four cylinder, production engine. Three compression ratios were investigated: 10:1, 11:1

and 12:1. Their results showed that higher compression ratios improved engine performance for both fuels throughout all the speed range investigated, with major effects being observed when hydrous ethanol was used.

Balki and Sayin [13] also investigated experimentally on the effect of compression ratio (CR) on a spark ignition using pure ethanol, methanol and unleaded gasoline. In the experiments, an engine having a CR of 8.5:1, having a single cylinder and air-cooled was used. These tests were conducted on four different CRs of 8.0:1, 8.5:1, 9.0:1 and 9.5:1. The test results obtained shown that pure ethanol and methanol provided a lower exhaust emission compared to gasoline's emissions at all CRs. Furthermore, with an increasing CR, the cylinder gas pressure generally increased and heat release rate rose earlier than those values in unleaded gasoline.

From aforesaid literature reviews, it can be summarized that there are some major research gaps: the good fit between commercial fuel (E85) and commercial engine (with size that widely used) with higher compression ratio (all above are lower than 12), should be taken into account. This study aims at addressing these regarding research gaps.

2. Engine performance calculation

2.1 Torque

Torque of engine was calculated as [4,5]:

$$T = FR \tag{1}$$

where,

T = Torque (N.m)

F = Tangent force on dynamometer rotor (N)

R = Radius of dynamometer rotor (m)

2.2 Brake power

Brake power of engine was calculated as [4,5]:

$$W_B = 2\pi Tn / 60 \tag{2}$$

where, W_B = Brake power (kW)

T = Torque of engine (N.m)

n = Speed of engine (rpm)

2.3 Brake specific fuel consumption

Brake specific fuel consumption (bsfc) is quality or mass of fuel consumed per a unit of brake power [14,15], can be determined as:

$$bsfc = \frac{m_F}{W_B} \times 3600 \tag{3}$$

where,

$bsfc$ = Brake specific fuel consumption (kg/kWh)

m_F = fuel mass consumption rate (kg/s)

2.4 Brake thermal efficiency

Brake thermal efficiency was calculated as [4,5]:

$$eff = \frac{W_B}{LHV \times m_F} \times 100\% \tag{4}$$

Where,

eff = Brake thermal efficiency (%)

LHV = Low heating value of fuel (kJ/kg)

3. Experimental Setup

The test engine used in this study is an EFI water cooled gasoline engine (NISSAN, QG 15DE), four stroke, four cylinders. The technical specifications of the test engine are given in Table 2. It was modified for tested on laboratory based variable compression ratio with different loads and compression ratios of the engine.

Table 2. Specifications of the test engine

Description	Specification
Engine	Nissan QG-15 DE
Cylinder	Inline 4 Cylinder
Bore x Stroke	73.6 mm x 88.0 mm
Size	1,497 CC.
Compression ratio	9.9 : 1

A 120 kW hydraulic brake dynamometer was employed for measuring engine torque and power. The fuel consumption is assessed using load-cell weighting and monitoring via digital monitor. Almost all required data was recorded via a built-in data logger. The schematic layout of the experimental setup of the engine and its attachments are shown in Fig. 1. The engine was coupled to a dynamometer. The inlet side of the engine consists of anti pulsating drum and air temperature measuring device. The actual photo for the test engine rig and its control panel are shown in Fig. 2 and 3, respectively.

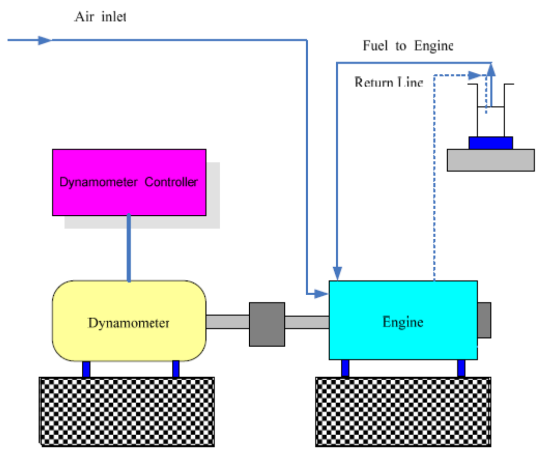

Fig. 1 Schematic diagram of the experimental setup

Fig. 2 The actual photo for the test engine rig

The engine speed was measured manually using a tachometer, DIGICON, model: DT 235T, as shown in Fig. 4

Fig. 3 Control and monitoring panel

Fig. 4 Measurement of engine speed using tachometer

3.2 Experimental Procedure

After the engine was installed on the test rig, before starting the tests some pre-commissions have been taken such as checking cooling water supply to the dynamometer, engine lubricating oil level, and the control panel power supply. First, the engine is started under no load condition for a few minutes so that the speed stabilizes at the rated value. It is fueled with E85gasohol. These test subjected to engine performance test. In this current study, the engine was run at three different compression ratios (9.9, 13.5 and 16), where the ratio of 9.9 is a standard value.

First, the commercial E85 gasohol (with the properties shown in Table 1.) was used for the standard engine with the compression ratio of 9.9. To vary the compression ratio the cylinder head of the test engine, as shown in Fig. 5, was removed and grinded via CNC machine. In this study, the compression ratio was measured using oil filling to know the current clearance volume of the cylinder head. From this measurement, it was found that when the thickness of cylinder head was cut out for 1 mm, the compression ratio was increased to be 13.5. With this compression ratio, the engine was reinstalled and tested as same as the first time. With the same procedure, the compression ratio of the tested engine was increased to be 16. With this compression ratio, the engine was tested under the same operating conditions to assess its performance.

The test subjected to engine performance indices, e.g.: torque, brake power, specific fuel consumption and thermal efficiency of engine. These parameters were measured, via dynamometer, at different engine speed of 3000, 3500, 4000, 4500 and 5000 rpm. Actually, only torque and fuel consumption at each engine speed were recorded and the other parameters were calculated using these test data. The engine tests were done for the compression ratio of 9.9, 13.5 and 16, respectively, with the same manner.

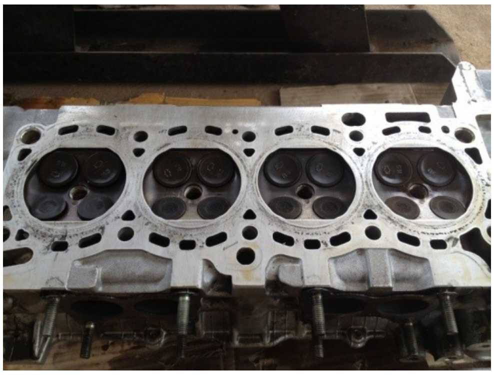

Fig. 5 Cylinder head of the test engine

The fuel consumption is measured using 200 cc. burette and stop watch. A damping tank and an orifice plate along with manometer are used to measure air flow rate.

4. Experimental Results and Discussions

Figure 6 shows the measured torques as a function of engine speed of the proposed engine which was tested at different compression ratio. The engine performance tests show that the maximum torque at the compression ratio of 9.9, 13.5 and16 are 59.5 N.m, 71.1 N.m and83.3 N.m, respectively. The results demonstrated that when the increment of compression ratio can improve the torque of engine at all engine speed.

The effect of compression ratio on the engine brake power is shown in Fig.7. The study results show that, when the compression ratio was increased from 9.9 to 16, the maximum brake power of the proposed engine was improved with the trend varied in the range of28 to 40 kW.

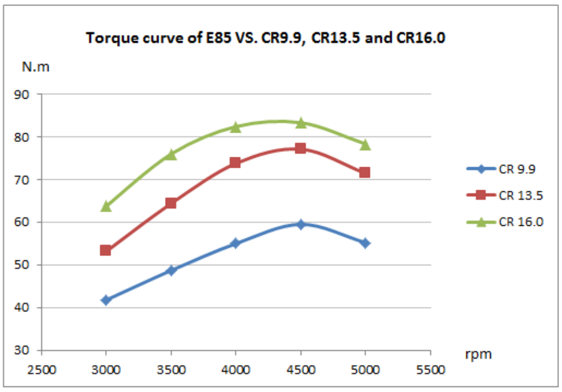

Fig. 6 Measured torques as a function of engine speed of the proposed engine which was tested at different compression ratio

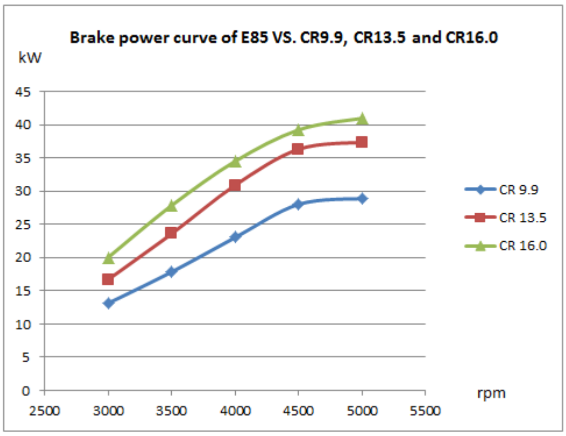

Fig. 7 Effect of compression ratio on the engine brake power

Figure 8 shows the influence of compression ratio on the brake specific fuel consumption at different engine speed. The obtained minimum of brake specific fuel consumption at the compression ratio of 9.9, 13.5 and16 are0.295 kg/kWh, 0.230 kg/kWhand0.227 kg/kWh, respectively.

Figure 9 shows the effect of compression ratio on the brake thermal efficiency of the proposed engine at any engine speed. It can be observed that when the compression ratio was increased and this engine was fueled with E85-gasohol, the brake thermal efficiency of the proposed can be increased. However, the test results show that the maximum compression ratio for this engine was about 16 with the maximum efficiency of about 54% was obtained.

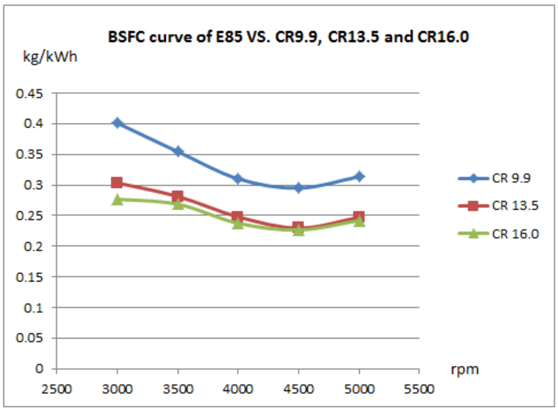

Fig. 8 Influence of compression ratio on the brake specific fuel consumption at different engine speed

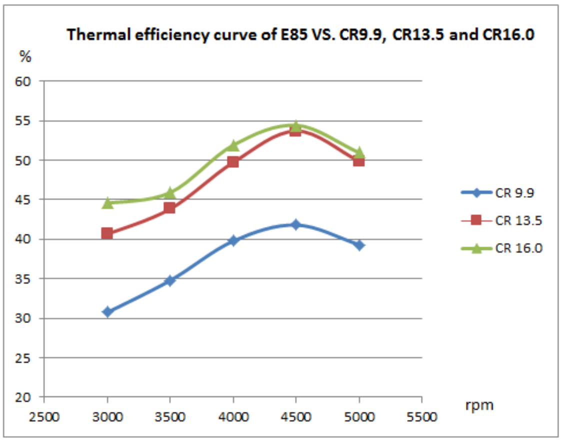

Fig. 9 Effect of compression ratio on the brake thermal efficiency of the proposed engine at any engine speed

5. Conclusion

The effect of compression ratio on performance parameters of the proposed engine fueled with E85-gasoholwas evaluated. The experiments are performed using combination of different preset of compression ratios of 9.9, 13.5, and 16 varying engine speed from 3,000 rpm to 5,000rpm in steps of 500rpm. The performance parameters investigated are torque, brake thermal efficiency and brake specific fuel consumption.

The experiment results demonstrate that, when this engine was fueled with E85-gasohol, the compression ratio of 16 outperforms the others with the maximum of torque, power and efficiency at 83.3 N.m, 39.25 kW and 54.4%, respectively. Compared to the standard compression ratio, these performance parameters can be increased for28.6%, 7.4%, and 23.2%, respectively. The brake specific fuel consumption was also decreased about 30.2%, compared with the standard compression ratio.

It can be also concluded that increment of compression ratio, together with higher octane number fuel like E85-gasohol, all of important performance parameters can be improved.

6. Acknowledgement

The author is greatly indebted to The Rajamangala University of Technology Thunyaburi for providing the financial support for his research and publications. The author also expresses their sincere thank to Rajamangala University of Technology Isan for supporting the experimental laboratory for this research.

7. References

[1] Pongsing, C. and Prasartkaew, B. (2014). Experimental Investigation on the Performance of a Porous-Medium Combustor for Solid Feedstock, *1st International Conference on Engineering Science and Innovative Technology (ESIT 2014)*, Krabi, Thailand.

[2] Koc, M., Sekmen, Y., Topgul, T. and Yucesu, H. S. (2009) . The effects of ethanol–unleaded gasoline blends on engine performance and exhaust emissions in a spark-ignition engine, *Renewable Energy*, 34, pp. 2101 – 2106.

[3] Cegel, Y. A. and Boles, M. A. (1994), “Thermodynamics an Engineering Approach,”

[4] Brinkman, N.D. (1981), “Ethanol Fuel a Single-Cylinder Engine Study of Efficiency and Exhaust Emission, paper presented in the International Congress and Exposition, USA.

[5] He, B.Q., Wang, J.X., Hao, J.M., Yan, X.G. and Xiao, J.H. (2003). A study on emission characteristics of an EFI engine with ethanol blended gasoline fuels, *Atmospheric Environment*, 37, pp. 949 – 957.

[6] Bayraktar, H. (2005). Data bank: Experimental and theoretical investigation of using gasoline–ethanol blends in spark-ignition engines, *Renewable Energy*, 30, pp. 1733 – 1747.

[7] Chen, R.H., Chiang, L.B., Chen, C.N. and Lin, T.H. (2011). Cold-start emissions of an SI engine using ethanol gasoline blended fuel, *Applied Thermal Engineering*, 31, pp. 1463 – 1467.

[8] Tutak, W. (2014). Bioethanol E85 as a fuel for dual fuel diesel engine, Energy Conversion and Management, 86, pp. 39 – 48.

[9] Turkoz, N., Erkus, B., Karamangil, M. I., Surmen, A. and Arslanoglu, N. (2014). Experimental investigation of the effect of E85 on engine performance and emissions under various ignition timings, *Fuel*, 115, pp. 826 – 832.

[10] Yucesu, H. S., Topgul, T., Cinar, C. and Okur, M. (2006). Effect of ethanol–gasoline blends on engine performance and exhaust emissions in different compression ratios, *Applied Thermal Engineering*, 26, pp. 2272 – 2278.

[11] Koc, M., Sekmen, Y., Topgul, T. and Yucesu, H. S. (2009) . The effects of ethanol–unleaded gasoline blends on engine performance and exhaust emissions in a spark-ignition engine, *Renewable Energy*, 34, pp. 2101 – 2106.

[12] Costa, R. C., Sodré, J. R. (2011). Compression ratio effects on an ethanol/gasoline fuelled engine performance, *Applied Thermal Engineering*, 31, pp. 278 - 283.

[13] Balki, M. K., Sayin, C. (2014). The effect of compression ratio on the performance, emissions and combustion of an SI (spark ignition) engine fueled with pure ethanol, methanol and unleaded gasoline, *Energy*, 71, pp. 194 – 201.

[14] Willard. W. P. (1997) "Engineering Fundamental of the Internal Combustion Engine,"

[15] Heywood, J.B. (1988). "Internal Combustion Engine Fundaments," Mc Graw Hill Inc., Newyork.