Effect of Fuel Octane Number on the Performance of Single Cylinder Four Stroke SI Engine

[1]Khin Mar Sein, [2]Htay Htay Win, [3]Nyein Aye San
[1] Ph.D Candidate, [2] Professor, [3] Associate Professor,
Department of Mechanical Engineering,
Mandalay Technology University, Mandalay, Myanmar
Email - [1] dawkhinmarseinsein@gmail.com, [2] htayhtayw@gmail.com, [3] nyeinayesanmdy@gmail.com

Abstract: *The effect of fuel octane number on the performance of spark ignition (SI) engine are investigated in this research. Heating value of fuel is directly proportional to octane number of fuel for SI engine. The research is conducted with three types of fuel (gasoline, 92 and 95).* According to the research, for constant compression ratio, constant engine speed and constant load, octane number 95 fuel has minimum specific fuel consumption of 0.343 kg/kWhr than octane number 92 and gasoline fuel. *The maximum brake thermal efficiency of 24% occur at octane number 95 fuel* for constant compression ratio, constant engine speed and constant load.

Key Words: octane number, *compression ratio, engine speed, brake thermal efficiency, brake specific fuel consumption*

1. INTRODUCTION:

Improving internal combustion engine efficiency is a prime concern today. A lot of engineering research has gone into the improvement of the brake thermal efficiency of the engines, so as to get more work from the same amount of fuel burnt. Fuel type is one of the factors effecting on SI engine performance. Different fuel has different octane number. Heating value of fuel is directly proportional to octane number of fuel. So for a given engine, higher octane number fuel give higher thermal efficiency and lower specific fuel consumption[3].

2. THEORTICAL ANALYSIS OF SI ENGINE:

In the working principle of SI engine, there are four processes such as admission process, compression process, combustion process and expansion process.

A. Admission Process

Pressure at the end of the admission process is

$$P_a = P_{in} - (\beta^2 + \xi_{is})\frac{V_{is}^2}{2} \times \rho_{in} \tag{1}$$

Where,

$\beta^2 + \xi_{is}$ = (2.5 ~ 4) and choose 4 [11]

Mean velocity of air during the suction process at the valve is

$$V_{is} = 2\pi\, NR\sqrt{1+\lambda_{rod}^2}\,\frac{\pi}{4}D^2 \times \frac{1}{a_{is}} \tag{2}$$

Where,

λ_{rod} = ratio between crank radius and connecting rod length, 3.4
R = crank radius, 0.02 m
a_{is} = area through the valve, m^2

Temperature at the end of the admission process is

$$T_a = \frac{T_{in} + \Delta T + r_{res} \times T_{res}}{(1 + r_{res})} \tag{3}$$

Coefficient of admission is

$$\eta_v = \frac{r_c}{r_c - 1} \times \frac{P_a}{P_{in}} \times \frac{T_{in}}{T_a(1 + r_{res})} \tag{4}$$

Coefficient of residual gases

$$r_{res} = \frac{T_0 + \Delta T}{T_{res}} \times \frac{P_{res}}{(r_c P_a - P_{res})} \tag{5}$$

Where,
ΔT =temperature increment due to charge heating, K

B. CompressionProcess

Pressure at the end of compression process is

$$P_{com} = P_a \times (r_c)^{n_1} \tag{6}$$

Temperature at the end of compression is,

$$T_{com} = T_a \times (r_c)^{n_1 - 1} \tag{7}$$

Where,
n_1= polytropic exponent for compression, 1.3 [11]

C. CombustionProcess

Pressure at the end of combustion process is

$$P_Z = \mu_{th} \frac{T_Z}{T_{com}} P_{com} \tag{8}$$

Where,
μ_{th}= theoretical coefficient of molar change

D. Expansion Process

Pressure at the end of expansion process is

$$P_e = \frac{P_z}{\delta^{n_2}} \tag{9}$$

Temperature at the end of expansion process is

$$T_e = \frac{T_z}{\delta^{n_2 - 1}} \tag{10}$$

The ratio of subsequence expansion is,

$$\delta = \frac{r_c}{\rho} \tag{11}$$

Mean indicated pressure for theoretical cycle is,

$$P_{i,d} = P_a \times \frac{r_c^{n_1}}{r_c - 1}\left[\lambda(\rho - 1) + \frac{\lambda\rho}{n_2 - 1}\left(1 - \frac{1}{\delta^{n_2 - 1}}\right) - \frac{1}{n_1 - 1}\left(1 - \frac{1}{r_c^{n_1 - 1}}\right)\right] \tag{12}$$

Mean indicated pressure for actual cycle is,

$$P_i = \phi_1 \times P_{i,d} \tag{13}$$

Where,
n_2 = polytropic exponent for combustion, 1.34
φ_1 = the coefficient of pressure reduction, 0.95
λ = ratio of pressure increase
ρ = pre-expansion pressure coefficient
Indicated specific fuel consumption is,

$$isfc = \frac{3600 \times \eta_V \times \rho_{in}}{P_i \times \alpha \times a_{th}} \tag{14}$$

Mean pressure for mechanical losses is,

$$P_{mech} = A + BV_p \tag{15}$$

Break mean effective pressure is,

$$P_b = P_i - P_{mech} \tag{16}$$

Mechanical efficiency is,

$$\eta_{mech} = \frac{P_b}{P_i} \tag{17}$$

Brake specific fuel consumption is,

$$bsfc = \frac{isfc}{\eta_{mech}} \tag{18}$$

Indicated thermal efficiency is,

$$\eta_{ith} = \frac{3600}{isfc * Q_{HV}} \tag{19}$$

Brake thermal efficiency is,

$$\eta_{bth} = \eta_{ith}\eta_{mech} \tag{20}$$

Indicated power is,

$$IP = \frac{P_i \times V_d \times N \times n}{30 \times \tau} \tag{21}$$

Brake power is,

$$BP = IP * \eta_{mech} \tag{22}$$

3. THEORETICAL RESULTS AND DISCUSSION:

Theoretical results of spark ignition engine at various compression ratio and three types of fuel are conducted at 3000 rpm engine speed and 700 W load.

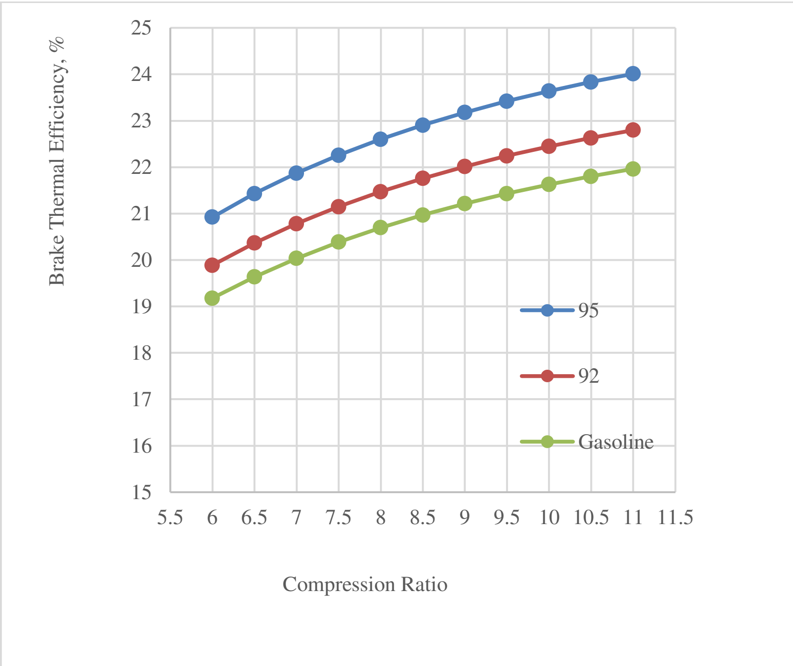

Figure 3.1. Brake Thermal Efficiency Versus Compression Ratio at 3000 rpm

The brake thermal efficiency of three different fuel are shown in Figure 3.1. This analysis is conducted by changing with compression ratio 6 to 11. According to this analysis, the brake thermal efficiency is directly proportional to the compression ratio. The maximum brake thermal efficiency is occurred at compression ratio 11 with 95 octane number fuel. The minimum fuel consumption of the engine is occurred by using 95 octane number fuels. By the increasing of the compression ratio from 6 to 11 with the increasement of 0.5, the fuel consumption is directly proportional to compression ratio. Higher octane number fuel occurs lower fuel consumption. These variations are shown in Figure 3.2.

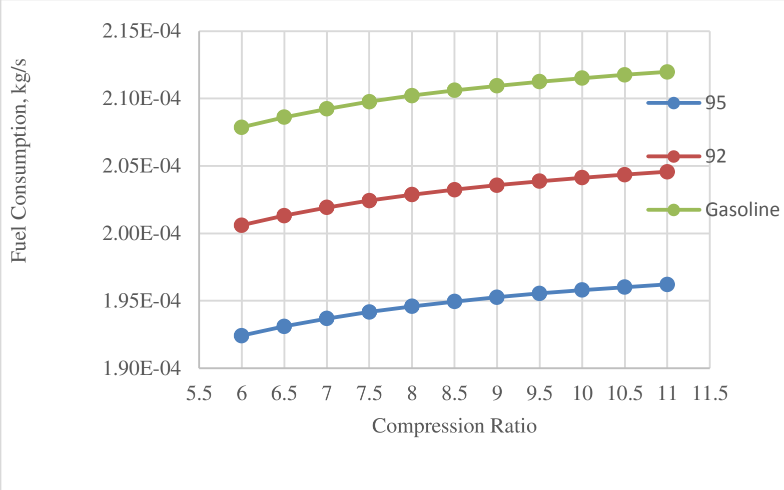

Figure 3.2. Fuel Consumption Versus Compression Ratio at 3000 rpm

The brake specific fuel consumption of three different fuel with the variation of compression ratio 6 to 11 are shown in Figure 3.3. The brake specific fuel consumption of 95 octane number is lower than other two fuel.

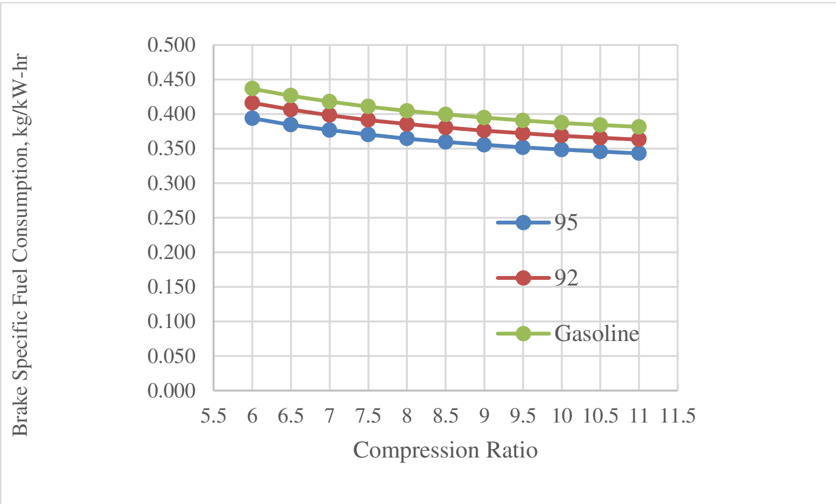

Figure 3.3. Brake Specific Fuel Consumption Versus Compression Ratio at 3000 rpm

According to the theoretical results, for constant engine speed and load, engine brake power and brake thermal efficiency are directly proportional to fuel octane number and compression ratio. But brake specific fuel consumption is inversely proportional to fuel octane number and compression ratio.

4. CONCLUSION:

Theoretically, higher octane number fuel can improve the brake thermal efficiency of the engine by producing more power output. For a given four- stroke spark ignition engine, higher octane number fuel must be used for higher efficiency and lower brake specific fuel consumption. From the theoretical result, for constant speed and load, octane number 95 fuel give maximum brake thermal efficiency, 24%, minimum brake specific fuel consumption ,0.343 kg/kWhr.

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