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Parametric Computational Investigation on Single Cylinder SI Engine Fitted to Generator Using dual fuel under Constant Speed and Varying Compression Ratio

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ABSTRACT

In this study, the effects of LPG–Gasoline (10%, 20%, 30% LPG and 100% Gasoline) dual fuels on the performance of a single cylinder 4 stroke spark ignition (SI) engine were investigated. In the theoretical study, the Single-cylinder, four-stroke, Single-point injection system SI engine fitted to a Generator was used. For this purpose, simulations were carried out using AVL Boost Software, without catalytic convertor under constant engine speed (3600 rpm) and varying load conditions. The variations in Performance brake power, Torque, brake specific fuel consumption, and exhaust gasses were examined at varying compression ratios for dual fuel single cylinder SI engine. Variable compression ratio (VCR) technology has long been recognized as a method for improving the fuel economy of SI engines. The results obtained from the use of LPG–Gasoline dual fuel were compared to those of gasoline fuel. The results indicated that when LPG–Gasoline dual fuel were used, the brake specific fuel consumption increased and engine performance parameters such as torque and power increases with increasing LPG amount in the blended dual fuel. Positive results were obtained at all LPG usage levels in terms of exhaust emissions. Best results were achieved at using 100% Gasoline for exhaust emissions.

Introduction

Alternative fuels are derived from resources other than petroleum. When these fuels are used in internal combustion engines, they produce less air pollution compared to gasoline and most of them are more economically beneficial compared to oil. Last but not least, they are renewable. The most common fuels that are used as alternative fuels are natural gas, propane, ethanol, methanol and hydrogen. Lots of work have been written on engine operating with these fuels individually; but a small number of publications has compared some of these fuels together in the same engine [1–2]. 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. In recent years, in order to reduce the environmental damage of motor vehicles and meet the stringent emission regulations, clean alternative fuels such as liquefied petroleum gas (LPG), natural gas (NG), and Hydrogen (H) have been applied in motor vehicles [3–4]. LPG is well known as a clean alternative fuel for vehicles because it contains less carbon molecules than gasoline or diesel. Its higher ratio of carbon (C) to Hydrogen (H) reduces the amount of carbon dioxide (CO_2) and other non-regulated emissions, such as formaldehyde and acetaldehydes. LPG also has other many advantages such as high Octane number, high combustion value, little carbon accumulation, easy storage, and low cost. Searches for alternative energy sources in automotive industry have brought forward the use of LPG in vehicles as fuel. Nowadays, LPG is widely used as fuel in cars in developed Countries (Italy, Netherlands, France, Belgium, Japan, and U.S.A). LPG fuel is preferred as a clean alternative fuel for internal combustion engines due to easy availability and storage; low cost, high octane number, high combustion efficiency, and low exhaust emissions with respect to other fuels [5].In an attempt to decrease air pollution and obtain fuel economy, several LPG fuel supply system, which range from gas vaporization with an open-loop control system to liquid injection with a closed-loop control system have been developed and applied in various internal combustion engines. Most commercially available LPG engines adopt a mixer type system, which supplies gas fuel into the intake air upstream of the throttle body with a vaporizer [3]. However, conventional mixer systems have problems for meeting stringent low emission regulations because of the difficulty in air–fuel ratio (A/F) control precisely [6]. The gaseous sequential injection (GSI) system which is a LPG gas phase port injection system that was considered as one of the next generation fuel supply systems for internal combustion engines has reduced the problems of A/F control in light and middle duty vehicles [7].

This paper investigates for a four stroke spark ignition engine (The Briggs and Stranton variable compression ratio engine) the influence of varying compression ratio on the Brake

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power, Brake Torque, Brake specific fuel consumption, and emissions were examined The theoretical performance characteristics for the engine, obtained from derived equations, were also presented.

Nomenclature

CFD computational fluid dynamics
$\dot{m}$ mass flow rate (kg/s)
m mass (kg)
BSFC brake specific fuel consumption (kg/kw-hr)
NOX nitrogen Oxide
LPG liquefied petroleum gas
CO carbon monoxide
RON research octane number
OHV over head valve.

Simulation Setup

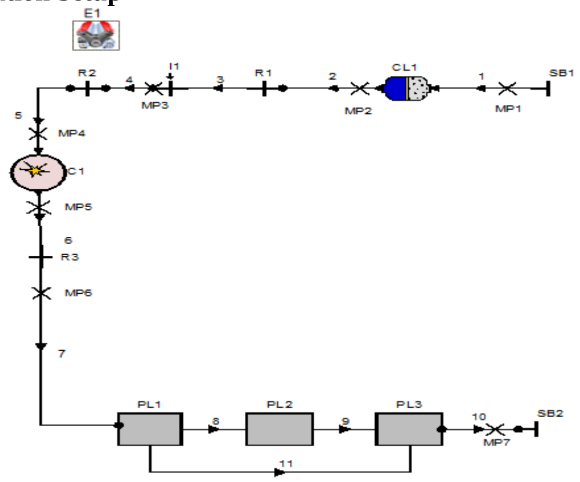

Fig A. Layout of Gasoline engine model

The 1-Dimentional engine simulation model is developed by using the software AVL BOOST and has been employed to study the engine performance working on LPG-gasoline dual fuel blends. The engine model used in this simulation was performed on a four stroke, Single cylinder spark ignition engine without catalytic convertor and port fuel injection. The gasoline engine model was calibrated by AVL and its layout is shown in Fig. A with engine specification shown in Table 1.The pre-processing step of AVL Boost enable the user to model a 1-Dimensinal engine test bench setup using the predefined elements provided in the software toolbox. The various elements are joined by the desired connectors to establish the complete engine model using pipelines.

In Fig. A, E1 represents the engine while C1, represent the single cylinder of the engine. MP1 to MP7 represent the measuring points. PL1 PL2 and PL3 represent the plenum. SB1 and SB2 are for the system boundary. The flow pipes are numbered 1 to 10. CL1 represents the cleaner. R1 to R3 represent flow restrictions.

Mathematical Model [8]

Mathematical models for spark–ignition engines can be divided into two main groups' thermodynamic and dimensional models. Thermodynamic models can in turn be classified in two sup groups_ single and multi zone models_ whereas dimensional models can be divided into one and multi dimensional models. In single –zone models the cylinder charge is assumed to be uniform in pressure, temperature, and composition. These models can be used as diagnostic (heat release analysis) or predictive tools. Because of their simplicity, single zone models, can account for mass flows into and out of crevices. However they ignore the flame propagation and combustion chamber geometry.

Single zone models can also be used as predictive tools if the mass burning rate is specified. The mass burning rate depends on the combustion duration, ignition angle, engine geometry, equivalence ratio, residual mass etc.Therefore tuning may be required to predict the pressure diagrams in different engines or in the same engine operating under different conditions.

Multi zone models attempt to resolve the combustion phenomena in a more physical manner than do single-zone models. The combustion chamber is generally divided into burnt and un burnt regions; sometimes thermal boundary layers in the burnt and un burnt gases are also considered. The cylinder charge is frequently assumed to be composed of ideal gases (frozen in the un burnt-gas region and in chemical equilibrium in the burnt-gas region),and the first law of thermodynamics, equation of state, and conservation of mass and volume are applied to the burnt and unburnt gases.

Single – Zone Models:

In single zone models the pressure, temperature, and composition of the cylinder charge are assumed to be uniform. These models define the state of the cylinder charge in terms of average properties, do not distinguish between burnt and unburnt gases, and assume that the cylinder charge is homogenous.Multi zone models permit a more accurate treatment of the thermodynamic properties of the cylinder mixture; the burnt and un burnt gases are considered as separate thermodynamic systems that are uniform in composition and state. However the geometry of each zone must be tracked in order to calculate the heat transfer and composition of the burnt and un burnt gases are considered as separate thermodynamic systems that are uniform in composition and state.

Combustion in single-zone models can be considered as a heat addition process, and the chamber charge is regarded as a simple fluid. The first law of thermodynamics applied to an open system can be written as

$$\frac{d_{(me)}}{d_{\theta}} = -p\frac{dV}{d\theta} - \frac{dQ}{d\theta} + \sum \dot{m}i\, hi \qquad e = e^0 + \int_{T_0}^{T} C_v \; dT \;\ldots\ldots\ldots(1)$$

Where p,T, and m are the pressure, temperature, and mass of the cylinder charge, respectively is the mixture specific internal energy; Cv is the specific heat at constant volume; V is the combustion chamber volume; dQ/dθ represents the heat losses; hi is the specific enthalpy of the gases flowing into the cylinder with a mass flow rate equal to $\dot{m}$i;T0 is a reference temperature;e^0 is the internal energy of formation at the reference temperature T0; and θ is the crank shaft angle.

In the absence of injection and flows into crevices, dm/dθ = 0.In premixed charge engines, there are flows into and out of crevices (i.e..volumes between the piston, cylinder wall, and piston rings and the spark plug threads).The crevices can be modeled as a single volume at the cylinder pressure or as a series of volumes, connected by restrictions to simulate the piston ring- cylinder wall region and blow by.

Equation (1) can be written as

$$\frac{dQ_{CH}}{d\theta} = mC_v\frac{dT}{d\theta} + p\frac{dV}{d\theta} + \frac{dQ}{d\theta} + (h-e)\frac{dm_{CR}}{d\theta} \ldots\ldots\ldots\ldots\ldots\ldots\ldots(2)$$

Where *dQCH* /dθ represents the heat released by combustion,*dmCR*/dθ represents the mass flow rate into crevices, and h is the specific enthalpy.

Conservation of mass applied to the combustion chamber yields

$$\frac{dm}{d\theta} = -\frac{dm_{CR}}{d\theta} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3)$$

When the cylinder pressure is high,$\frac{dm_{CR}}{d\theta} > 0$ and the value of h corresponds to that in the combustion chamber:

h = e+PV/m………………………………………………..... (4)

However, during the expansion stroke, $dmCR/d\theta < 0$ and the value of h is that of the gases contained in the crevices. If the crevice volume and temperature are assumed constant and the crevice pressure is equal to that of the cylinder charge, the mass flow rate into the crevices can be written as

$$\frac{dm_{CR}}{d\theta} = V_{CR}\frac{dp/d\theta}{RTw} \quad \text{…………………………………} \quad (5)$$

Where the crevice temperature was set equal to the temperature of the cylinder wall Tw and V_{CR} is the crevice volume. Equation (5) can be substituted into equation (2) and (3) to obtain an equation for the heat released by combustion once the heat transfer losses dQ/dθ are specified. Heat transfer correlations once the heat transfer losses dQ/dθ are specified.

Materials and methods

The test engine

The Briggs & Stratton variable compression ratio engine with direct current electric dynamometer is a four stroke air cooled OHV single cylinder petrol engine. It has a 69.09mm cylinder bore and 61.19mm stroke, displacement 206 cc and magnetron electronic ignition system. Variation of the compression ratio was achieved by raising the cylinder head up in order to decrease the compression ratio and by lowering it down to increase the compression ratio.

Results and Discussion:

Brake Power

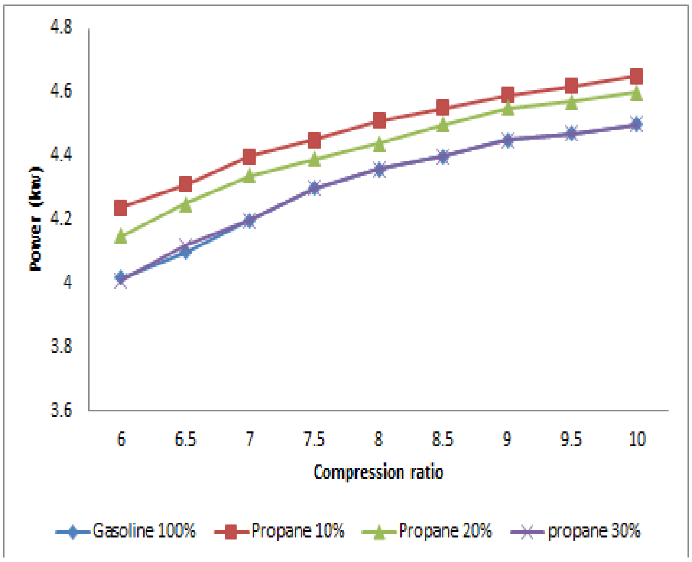

Fig 1. Shows the graph between Power and Compression ratio

The Fig.1 is graph between Power and Compression ratio which shows that Power increases with increase in compression ratio in case of Gasoline 100%.Power increases for other dual fuel also using gasoline and LPG in various proportions i.e. 10%,20%,and 30% LPG blends in Gasoline. This increase in Power with increase in Compression ratio increases thermal efficiency of engine also. At higher compression ratio 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.

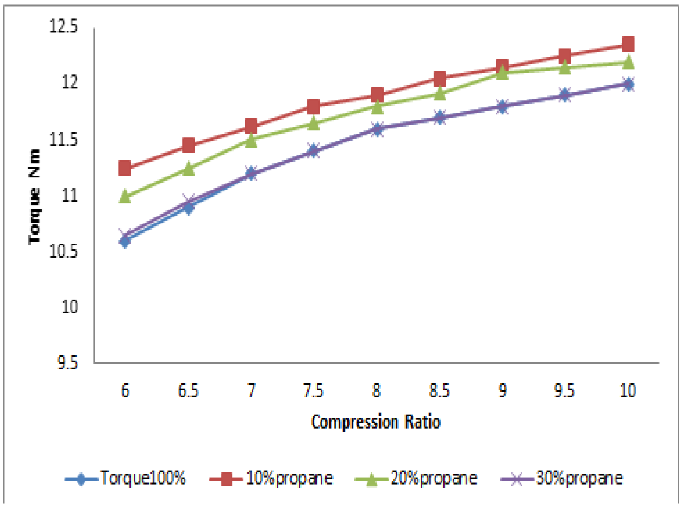

Fig 2. Shows the graph between Torque and Compression ratio

In Fig. 2 increase in compression ratio induces greater turning effect on the cylinder crank. That means that the engine is getting more push on the piston, and hence more torque is generated. The torque gain due to compression ratio Increase can be given as the ratio of a new compression ratio (new rc) to the old compression ratio (old rc) given by Torque gain/loss = (new rc /old rc).From the fig 2, it is clearly shown that with increase in compression ratio the Torque is increased for the dual fuel at all percentage of LPG i,e (10%,20%,30%) respectively.

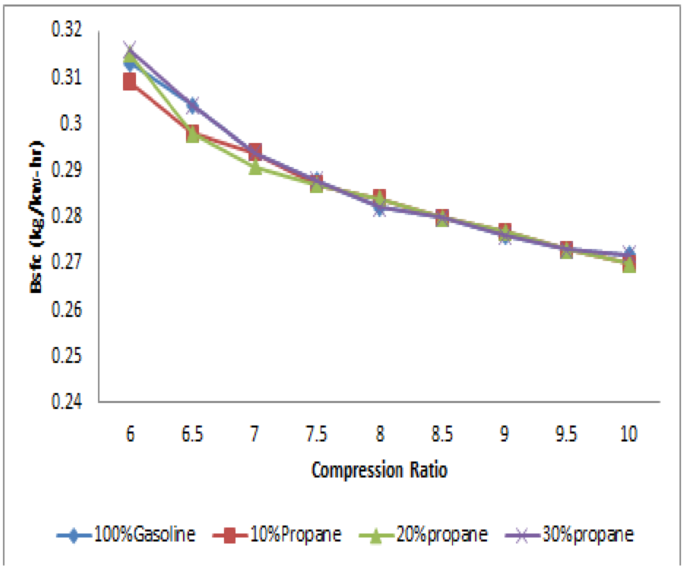

Fig 3. Shows the graph between Bsfc and Compression ratio

Fig. 3 shows the effect of compression ratio on Bsfc. As it can be seen from the figure that the Bsfc decreases with increase in compression ratio. This is due to the reason that LPG has less brake specific fuel consumption than gasoline. This phenomenon occur because of the dissimilar properties of the fuel itself, where LPG has a higher heating value and higher stoichiometric air-fuel ratio compared to Gasoline, so high specified amount of heat can be released with less amount of fuel. Figure shows increases in HC emissions with CR increases across all percentage of dual fuel of LPG and gasoline. This is due to varying surface-to-volume ratios and the increased importance of crevice volume effects at high CR. These increasing HC effects are difficult to avoid due to more of the premixed charge being forced into the crevices as the CR increases.

Table 1. Engine Specifications

Criteria	Description
Make	Briggs & Stratton
Type	Air Cooled 4stroke OHV Gasoline
Displacement	206 cc
Compression ratio	9
Bore & Stroke	65.09 × 61.91
Maximum HP	6.5
Cooling System	Air Cooled
Ignition System	Magnetron Electronic

Table 2. [9] Fuel Properties

Fuel Type	RON	Formula	Molecular Weight	Density (kg/m3)	Heat of Vaporization 298K (Kj/Kg)	Lower Heating Value (MJ/Kg)	Stoichiometric air/fuel Ratio
Gasoline	95.8	C8H18	106.22	750	305	44	14.6
Methane	120	CH4	16.04	720	305	50	17.23
Methanol	106	CH4O	32.04	792	1103	20	6.47
Ethanol	107	C2H6O	46.07	785	840	26.9	9.00
Propane	112	C3H8	44.10	545	426	46.4	15.67
Hydrogen	106	H2	2.015	90	426	120	34.3

Emissions:

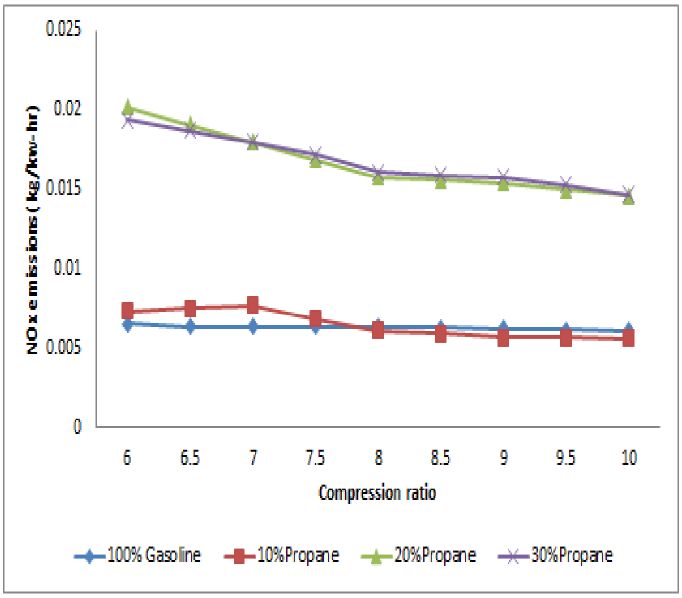

Fig 4. Shows the graph between NOx and Compression ratio

The Fig 4.Shows that NOx concentrations approached one another for the hydrocarbon fuels, when the engine worked at 100% Gasoline the NOx level is considerably less as Compared with using dual fuel LPG blending with Gasoline. This can be reached by referring to fig. (4), which represents the relation between compression ratio CR, and maximum NOx concentrations for all fuels used in this work. NOx concentration reduced in the rich side with CR increased because of lack of oxygen needed for reactions, the high increase in burning temperatures, low NOx formation and dissociation equilibrium, all can be considered additional reasons of low NOx concentrations, where dissociation reactions freeze in the expansion stroke. Lowering the air-to-fuel ratio in rich burn engines limits oxygen availability in the cylinder, thus decreasing NOx emissions. LPG has high heating value on mass basis, its flame speed a little more than gasoline, for these reasons the LPG is the second higher NOx levels and gasoline the third one.

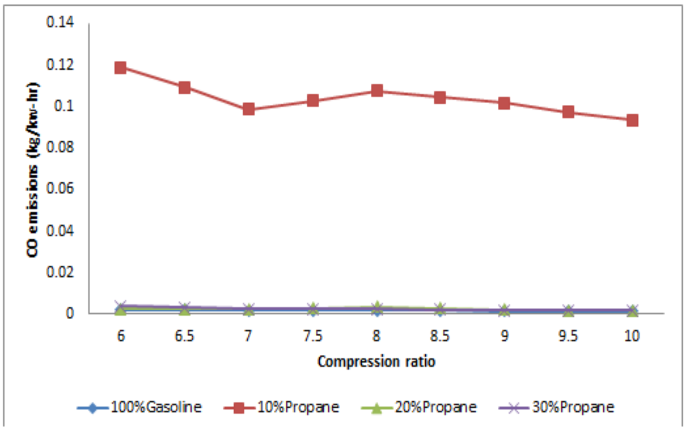

Fig 5. Shows the graph between CO emissions and Compression ratio

The Fig5. Shows the effect of compression ratio on CO emissions when using various blends of LPG in Gasoline as dual fuel in SI engine. The fig clearly shows that the CO emissions are less when using as dual fuel in SI engine. This is due to the reason that LPG contains less carbon than Gasoline.LPG powered vehicles produce 50 percent less carbon monoxide emissions per kilometer. Therefore emission is much reduced by the use of LPG. As the compression ratio increases, the level of CO decreases, which shows that the combustion is better at higher compression ratios.

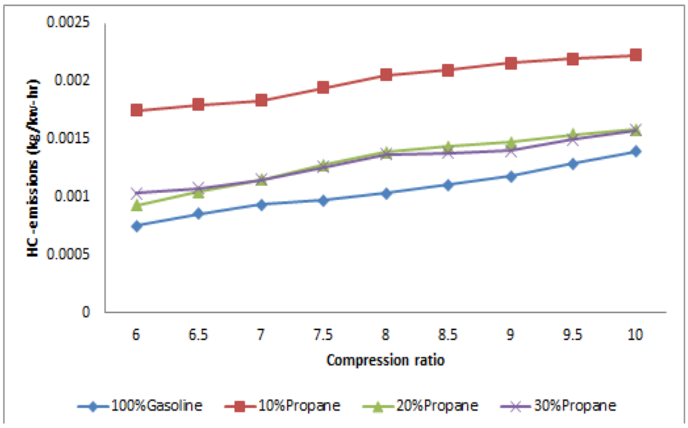

Fig 6. Shows the graph between HC emissions and Compression ratio

In Fig 6. Hydro carbon emissions will be different for each gasoline blend, depending on the original fuel components. Combustion chamber geometry and engine operating parameters also influence the HC component spectrum. When hydro carbon emissions get into atmosphere, they act as irritants and odorants; some are carcinogenic. All components except CH_4 react with atmospheric gases to form photo chemical smog. In fig 6 it is clearly shown that HC emissions are increasing with increase in compression ratio.

Conclusion

Simulations were carried out under constant engine speed and different compression ratios. The variations in Brake power, Torque, BSFC, and exhaust gasses were examined. Results obtained in this study were outlined below.

- A variable compression ratio SI engine is made as global parameter in AVL boost software with the compression ratio capable of being varied from 6 to 12 for SI engine.
- The Simulation were conducted using two fuels namely Gasoline and LPG as dual fuel.
- The Power was shown a increase depending on the LPG usage. The brake power increased considerably with the use of 30% LPG. As for the 10%, 20% LPG usage, power increased in proportion to LPG usage level. BSFC decreases with the increase in LPG usage level and the minimum BSFC value was obtained at 30% LPG usage.
- Positive results in terms of engine performance were only achieved at all percentage of LPG mixture ratio. With the use of mixture containing 10%, 20%, and 30% LPG, BSFC decreased with increase in compression ratio.

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