

# EXPERIMENTAL INVESTIGATION OF EFFECTS OF PISTON CROWN GEOMETRY ON PERFORMANCE OF BIOGAS FUELED 4 STROKE SI ENGINE

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**Abstract:** *The gradual reduction of the petroleum reserves and increase of pollutant emissions have demanded the development of alternative and renewable energy sources which can help to reduce the current oil dependency in the field of Internal combustion engine(ICE). Utilization of natural gas in I.C. engine is an option to overcome some problems and various renewable energy sources are also available to overcome this energy crisis. Biogas is a well established renewable and environment friendly fuel for rural energy needs. The main problem that all researchers and manufacturers are facing now is the low power output of Biogasengine due to low density, low volumetric efficiency, low flame speed and absence of fuel evaporation. Even Biogas engine require compact and turbulent combustion chamber for effective combustion of the charge inside the combustion chamber. This fact focuses towards the need for development of a Biogasengine technology which can offer same or better power output with lower specific fuel consumption and emissions with respect to its counterparts i.e. gasoline fueled engines within same volumetric capacity. So, for the development of the Biogas engine technology experiments were performed on a small 100-cc single-cylinder, four stroke spark-ignition engine which was operating on biogas as well as on petrol. The SI engine is converted to Biogas and is experimentally investigated at various combinations of crown geometries to estimate the optimal piston that could develop a compact and turbulent combustion chamber. The study concludes that proper selection of piston geometry reduces the de-rating of biogas fueled SI engine.*

**Keywords:** *Biogas, piston crown geometry, performance, IC engine.*

## Nomenclature

P = brake power developed by the engine, kW

I = current produced, A

V = voltage produced, V

BSGC=brake specific gas consumption, kg/kWh.

M=mass of fuel consumed, kg/h.

$\eta$ =Brake thermal efficiency

LHV =lower heating value of fuel, MJ/kg.

SI= spark ignition

PIV= particle image velocimetry.

CFD= computational fluid dynamics

## I. INTRODUCTION :

The world faces twin problems of energy crisis and environmental degradation. Currently petroleum fuels are also faced with challenges due its shortage. Fossil fuels contribute more than 80% of the world total energy [1]. Use of fossil fuels produced more emission of greenhouse gases. [2-3].

Unlike many other energy industries, transportation has not experienced significant fuel substitution, with petroleum based fuels accounting for over 95% of energy used. Recently, alternative transportation fuels have been receiving increasing attention as likely solutions to the problems of urban air quality, global warming and strategically important excessive dependence on imported oil. The search for a vehicle fuel that will produce minimum emissions and maximize fuel efficiency and economy has become the important aspect of cleaner vehicle fuel development. In this context, Biogas is a viable safe and clean fuel and can be converted to both S.I and C.I mode.

Biogas is produced by anaerobic digestion of various organic substances such as kitchen wastes, agricultural wastes, municipal solid wastes and sewage, cow dung etc., which offers low cost and low emissions than any other secondary fuels. [4] India is largest cattle breeding country; there is abundance of raw material for producing biogas.

Biogas is also used in Internal Combustion (IC) engine, because of its better mixing ability with air and clean burning nature. It can be supplemented to liquefied petroleum gas (LPG) and compressed natural gas (CNG), if it is used in compressed form in cylinders.

It is also used in Internal Combustion (IC) engine and used in both petrol as well as diesel engine. The reduction in concentration of CO<sub>2</sub> i.e. increasing methane content in the biogas will improve performance and reduces emissions of hydrocarbons and leads to higher faster combustion for the engine.

Biogas has an octane rating of 130, which is considerably higher than 93 octane for petrol; subsequently, Biogas vehicle is more energy efficient. Higher octane rating allows higher compression ratios and improved thermal efficiency.

Biogas being lighter than air, disperses easily into the atmosphere and does not form a sufficiently rich mixture for combustion to take place. The main problem that all researchers and manufacturers are facing now is the low power output of Biogas engine due to low density, low volumetric efficiency, low flame speed and absence of fuel evaporation. Due to low flame velocity combustion period is long which affect the performance of the engine. Proper shape of combustion chamber and piston plays an important role in generating maximum turbulence and increasing the desire rapid combustion and solved the problem of low burning velocity.

Huang et al.[5] Experimented in-cylinder flows of a motored four-stroke engine with flat-crown and slightly concave-crown pistons conclude that the shape of the piston and combustion chamber plays an important role in generating maximum turbulence and increasing the desire rapid combustion. Two pistons of different crown shapes (flat-crown and slightly concave-crown pistons) are studied. The results show that the flat-crown piston induces higher bulk-averaged tumble ratio and turbulence intensity than the slightly concave-crown piston and the engine with the flat-crown piston also presents larger torque and power outputs and lower hydrocarbon emission than that with the slightly concave-crown piston.

Rajesh C Iyer et al.[6] Investigate the influence of ignition voltage, Higher compression ratio and piston crown geometry on the performance of compressed natural gas engines. The authors found a major achievement in both performances as well as emissions on this 100 cc category of the Indian market. The performances improved by 20 – 30 % under certain cases with a considerable reduction in emissions. The 3 geometries (A,B,C) were identified based on local availability in Indian market and the piston selection was made such that the basic structure of the combustion chamber is not affected. The increase in compression ratio for Piston A from 6.2:1 to 7.1:1 as against the base configuration 8.2:1 has yielded better results.

Danaiah et al.[7] investigate performance evaluation of lean combustion technology in diesel engine. Enhancement of lean combustion of homogeneous mixtures can be achieved by using higher ignition energy. Providing high compression ratios. Creating high swirl in the combustion chamber and also suggest other techniques used in lean combustion are turbulence, squish, combustion chamber geometry, ignition timing, influence of ignition system, catalytically activated combustion, lean combustion using alternative fuels and its additives and EGR (exhaust gas recirculation).

Teresa et al.[8] developed an analytical methodology that uses empirical based models and CFD simulations to efficiently evaluate design alternatives in the conversion of a diesel engine to either CNG dedicated or dual fuel engines. A database of different combustion chambers that can be obtained from the original piston and optimized the combustion chamber by CFD study.

Antony et al.[9] analyzed efficient piston configuration for effective air motion by CFD with four piston configurations and compared the result with PIV. The result conclude that piston position plays a predominant role in the air pattern inside the cylinder and flat piston is more suitable because it creates more turbulence.

Johansson et al. [10] Study the effects of piston geometry on combustion for spark ignition engine.

The literature survey reveals that the shape of the piston crown geometry and combustion chamber geometry affects combustion parameters and selection of proper shape of the combustion chamber and piston plays an important role in generating maximum turbulence intensity, torque and power output. Proper selection of piston geometry improves combustion and reduces engine emissions. Any change in piston crown geometry results into change in combustion chamber configuration and larger cleavages on the piston crown increased the turbulence levels and which improve the combustion and improve performance of the Biogas fueled SI engine.

## II. EXPERIMENTAL SETUP

The main components of the engine test rig is SI engine, Biogas fuel, air intake and fuel mixing system, dynamometer, air consumption measurement system, engine speed measurement, load board, voltmeter, ammeter. The selected fuel produced from biomethanation of cow dung. It contains 50-60 % of methane and 30-40% carbon dioxide and rest of other gases as  $H_2$ ,  $N_2$  and  $H_2S$ . A single cylinder 100CC 4 stroke SI engine will be used for evaluation. The engine will be converted into a gas engine with extensive changes in the basic engine design such as removal of carburetor and mounting of a mixing device to maintain a constant air-fuel mixture, a governing mechanism and a starting motor. The pistons having different crown geometries been replaced one by one during evaluation. The selection of the pistons had been made such that, they can not affect the structure of the combustion chamber, stroke length, no changes should be made with the connecting rod assembly and the gudgeon pins fit easily in the small end of the crankshaft.

Three pistons geometries (A,B,C) had been identified as shown in figure 2,3,4.

The air mixture and gas vaporizer circuit were installed before fuel supply system and pressure of gas got maintained with pressure regulator. Electrical resistance load board which consists of bulbs of different watt were placed in series and used for load measurement. The loading capacity of load board will be 0–5.0 kW. The gas will be stored in the compressor having capacity of  $12 \text{ kg/m}^3$ . In present setup, the gas consumption for biogas will be measured by using Weightment technique in which the tank of biogas will be placed on the weighting scale and the gas consumption will be noted during the partial duration of time. 7-function analyzer was used to measure engine speed which was attached at spark plug. A multimeter was used for measurement of voltage and current. It can measure both voltage as well as current generated during the operation. It was connected to load board for measurement purpose. The schematic diagram of the experimental setup is shown in figure 1.

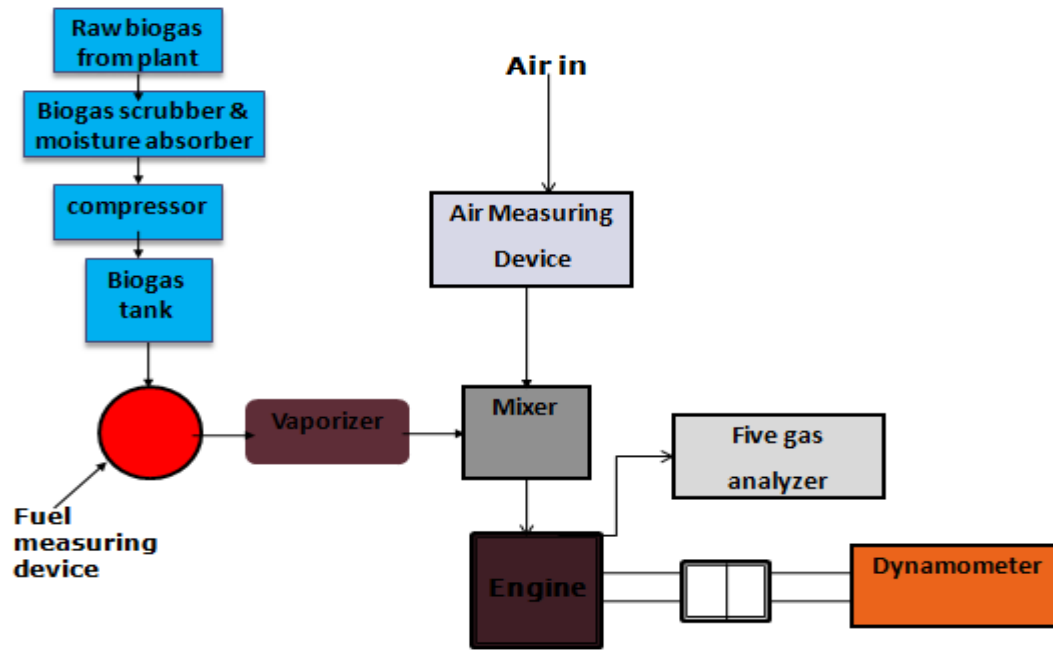


Fig 1, Experimental setup



Fig. 2Piston-A



Fig 3,Piston-B

Fig 4,Piston-C



Fig.5 Engine Head with pistons

Type	Four stroke, air cooled, single cylinder ,OHC engine
Fuel	Biogas
Number of cylinders	One
Bore	50 mm
Stroke	49.5 mm
Compression ratio	8.2:1
Connecting rod length	90 mm
Rated power	5.5 KW @8000 rpm
Maximum Torque	7.95Nm @5000 rpm
Displacement volume	97.30cc
Idle speed	1400 $\pm$ 100

Fig.6 Engine specification

### III. EXPERIMENTAL PROCEDURE



Fig.7 Experimental test rig.

Experiments were performed on a single cylinder 100cc, 4-S S.I engine to run on Biogas. It was a stationary test rig setup in which different piston crown geometries were investigated for evaluation of performance and emission and compared with petrol engine. The selection of piston crown geometries was such that they have not affected the basic structure of combustion chamber. By experimentations best configuration among all piston crown geometries had been chosen which had given best performance. All performance and emission tests had been performed at different speed and different load. 7-function analyzer was used to measure the engine speed in rpm. The test parameters of the experimentation were

(i) Brake power (kW) (ii) Engine speed (rev/min), (iii) Brake specific gas consumption (kg/kWh), and (iv) Brake thermal efficiency (%).

Brake power:

$$P = \frac{V \times I}{1000}$$

Brake specific gas consumption:

The brake specific gas consumption will be calculated by using equation,

$$BSGC = \frac{M}{P}$$

Brake thermal efficiency:

The brake thermal efficiency of the engine will be determined by using equation,

$$\eta = \frac{P \times 3600}{LHV \times M} \times 100$$

**IV. RESULTS AND DISCUSSIONS**

Initially the engine was tested on petrol using the identified 3 piston geometries to check for performance parameters, fuel consumption and power. The results were noted and based on the tabulations, graphs were plotted accordingly to compare the performance parameters of all the pistons. The graphs are plotted as below.

(1) Load

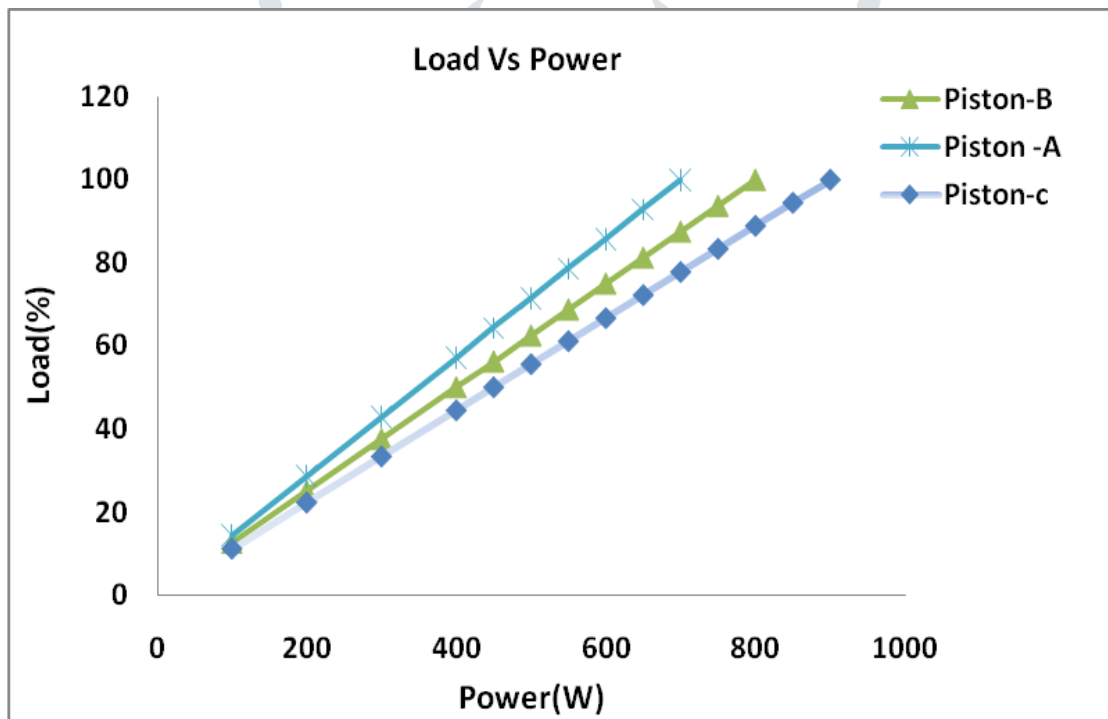


Fig. 8, Load Vs Power curve for Pistons A ,B, C on Petrol

Based on tabulation and graph we can see that for piston-A, Piston-B and Piston-c, as load increases the power will increase up to 700W, 800W and 900W respectively. Piston-A shows better performance of power as load increases gradually but it can produce power up to 700W only. Piston-B can produce 800W power. As load increases, it can produce less power compared to piston-A. Piston-c can produce 900W as a maximum power but as load increases, it can produce less power compared to other two pistons.

(2) Specific fuel consumption.

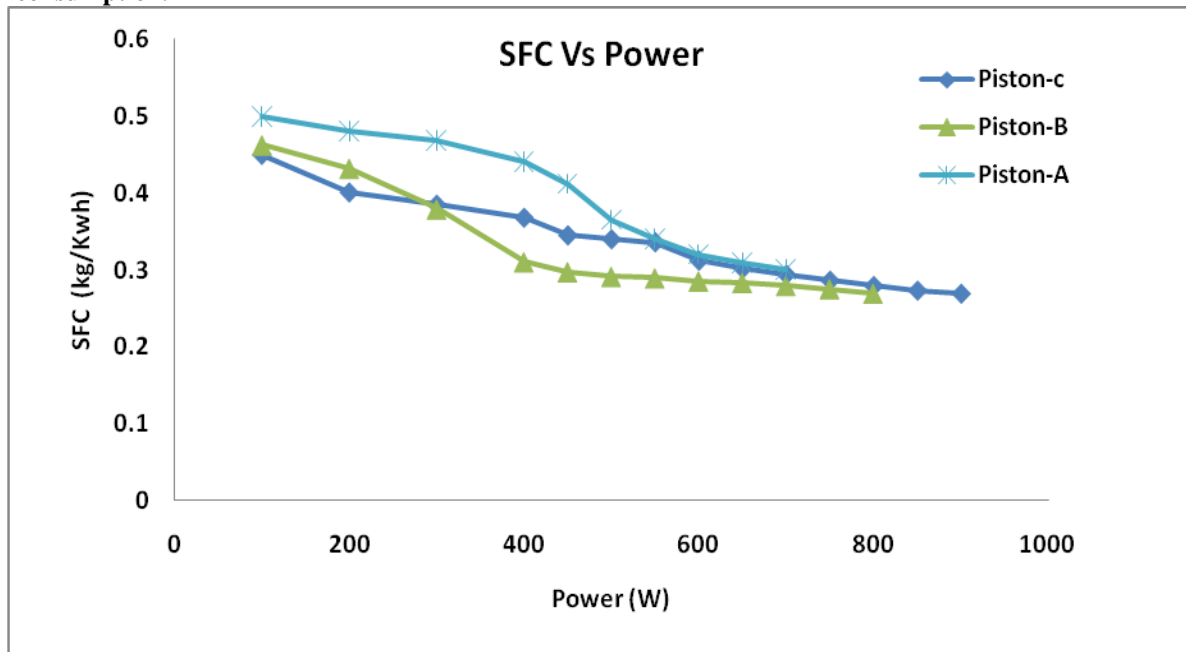


Fig.9, BSFC Vs Power curve for Pistons A , B, C on Petrol

As load increases the fuel consumption will decrease relatively for all piston configurations. The fuel consumption for engine having piston-A was from **0.5kg/kwh to 0.3kg/kwh**. The fuel consumption for engine having piston-B was from **0.46kg/kwh to 0.26 kg/kwh**. The fuel consumption for engine having piston-C was from **0.45kg/kwh to 0.25 kg/kwh**. Engine having piston C consumed little more fuel per unit power than other two piston geometries.

(3) Thermal Efficiency

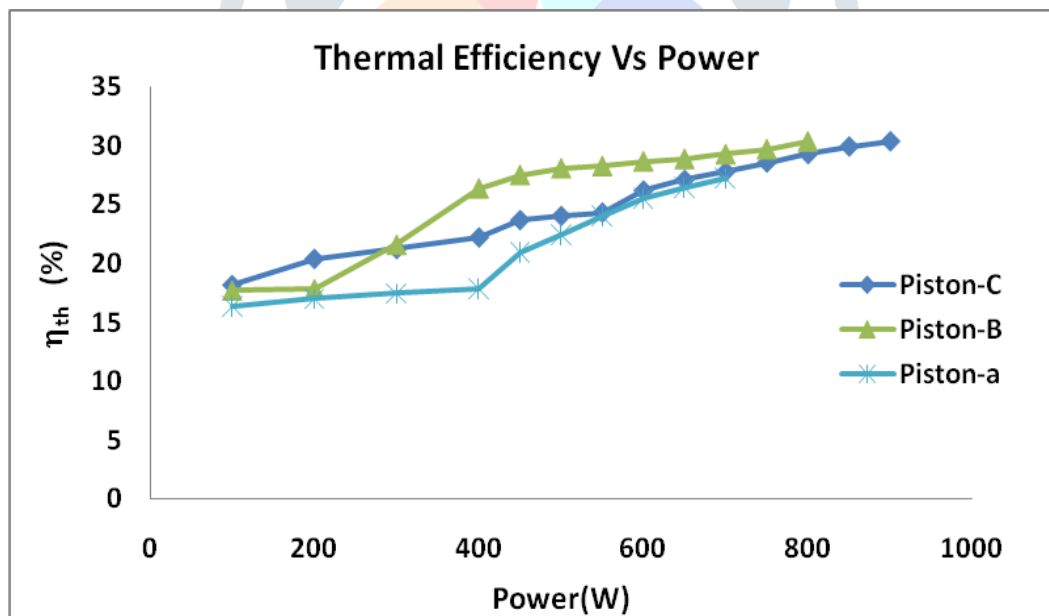


Fig.10, ηth Vs Power curve for Pistons A,B,C on Biogas

Thermal efficiency increases as load increases for all piston geometries. ηth for piston-A ranges from **16% to 27%** for minimum to maximum load. ηth for piston-B ranges from **17% to 30%** for minimum to maximum load. ηth for piston-C ranges from **18% to 30%** for minimum to maximum load. Engine configuration having piston-C showed highest thermal efficiency compared to other piston configurations

(4) Mass of fuel consumed

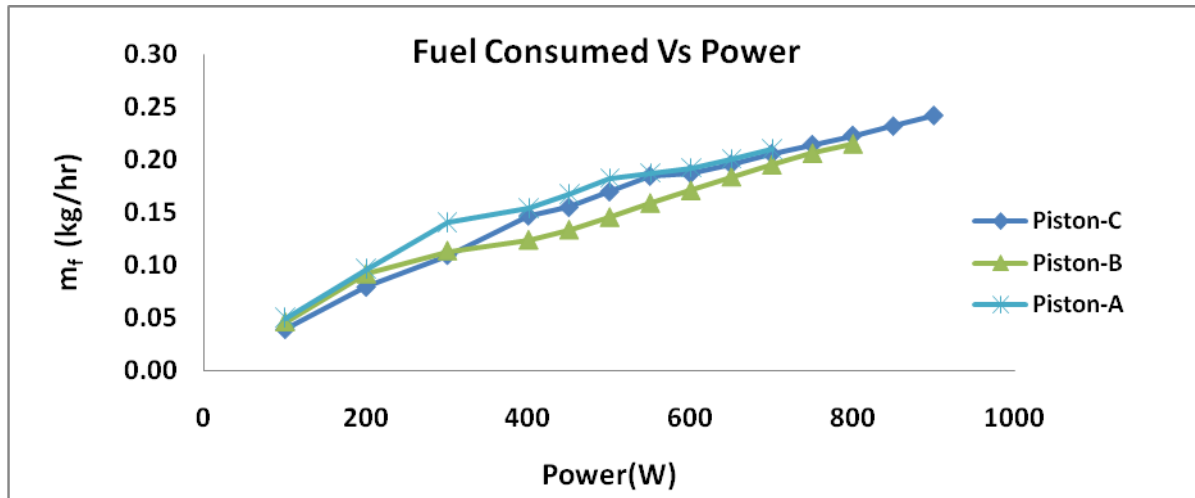


Fig.11. Fuel Consumption Vs Power curve for Pistons A, B, C on Petrol

As load increases the mass of fuel per unit power will increase accordingly for all piston configurations. An engine configuration having piston-A consumed fuel from **0.05kg/kw to 0.21kg/kw** for minimum to maximum load values. An engine configuration having piston-B consumed fuel from **0.05kg/kw to 0.22kg/kw** for minimum to maximum load values. An engine configuration having piston-C consumed fuel from **0.05kg/kw to 0.24kg/kw** for minimum to maximum load values. Thus all piston configurations have consumed almost the same amount of fuel but developed different amounts of power. An engine configuration having piston-C has produced higher power with the same amount of fuel consumption compared to other configurations.

Behaviors of an engine operated with biogas as a fuel have been plotted based on measured parameters as following.

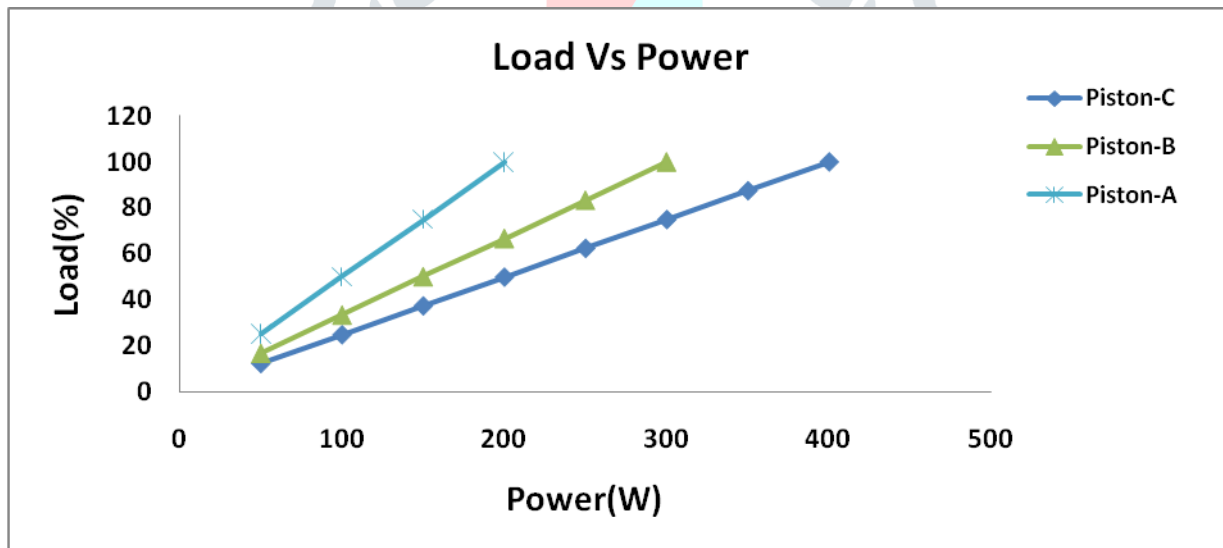


Fig.12, Load Vs Power curve for Pistons A,B, C on Biogas

Power increases gradually as load increases. The engine configuration having piston-A developed **200W** power at 100% loading. The engine configuration having piston-B developed **300W** power at 100% loading. The engine configuration having piston-C developed **400W** power at 100% loading.

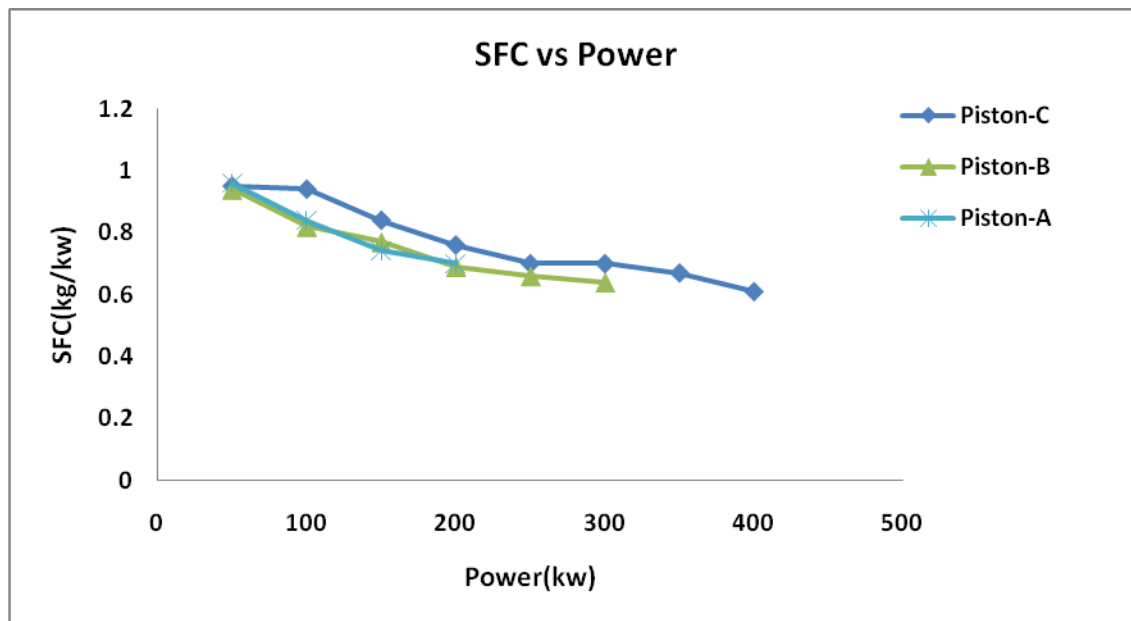


Fig. 13, BSFCVs Power curve for Pistons A, B, C on Biogas

Brake specific fuel consumption decreases with increase in load. Brake specific fuel consumption for engine configuration having piston-A was ranged as **0.96 kg/kwh to 0.7** from minimum to maximum loading. Brake specific fuel consumption for engine configuration having piston-B was ranged as **0.94 kg/kwh to 0.64 kg/kwh** from minimum to maximum loading. Brake specific fuel consumption for engine configuration having piston-C was ranged as **0.95 kg/kwh to 0.61kg/kwh** from minimum to maximum loading.

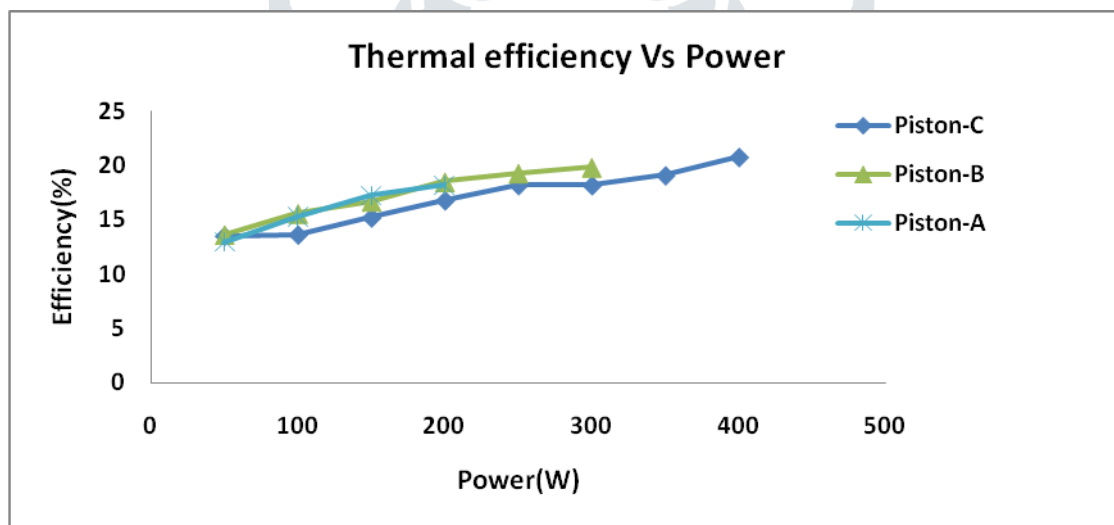


Fig.14 $\eta_{th}$  Vs Power curve for Pistons A, B, C on Biogas

Thermal efficiency increases with increase in load. Engine configuration having piston-A had efficiency ranging from **12.9 to 17.1** from minimum to maximum load. Engine configuration having piston-B had efficiency ranging from **13.6 to 19.8** from minimum maximum load. Engine configuration having piston-C had efficiency ranging from **13.5 to 20.7** from minimum to maximum load. Engine configuration having piston-C had higher efficiency compared to other two configurations.



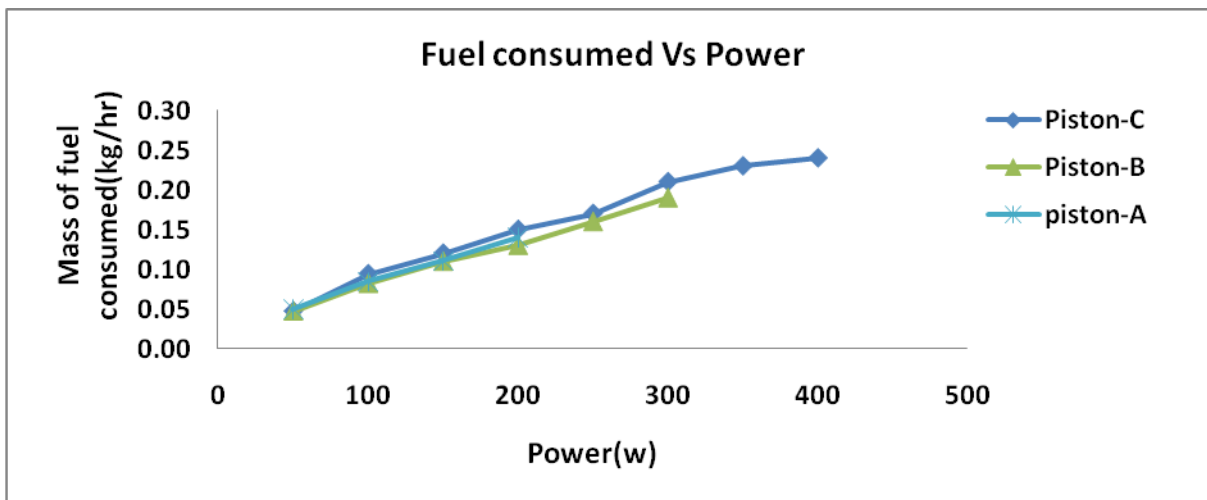


Fig.15, Fuel Consumption Vs Power curve for Pistons A,B,C on Biogas.

The mass consumption of fuel will increase with load. Mass consumption of gas for engine configuration having piston-A was **0.05 kg/kw to 0.14 kg/kw** for minimum to maximum value. Mass consumption of gas for engine configuration having piston-B was **0.05 kg/kw to 0.19 kg/kw** for minimum to maximum value. Mass consumption of gas for engine configuration having piston-C was **0.05 kg/kw to 0.24 kg/kw** for minimum to maximum value. Engine configuration having piston-C consumed more gas per unit power though it has developed highest power compared to other two engine configurations.

## V. CONCLUSION

The experiment was performed on a single cylinder 100cc, four stroke SI engine with biogas as a fuel and compared with petrol. Three types of piston crown geometries having different crown shape were identified and performance was evaluated for each piston.

From the experiments it can be concluded following points.

- Engine having Piston-C developed 400W for biogas and 900W for petrol. Thus it has reduced deration of engine to about 55% compared to other piston configurations as 66% for piston-B and 78% for piston-A.
- Engine having piston-c has shown highest brake thermal efficiency as 20.7% compared to other two pistons as 19.8% for piston-B and 18.01% for piston-A.
- Brake specific fuel consumption was less for engine having piston-C for all loading conditions compared to other two engine configurations.

Thus engine configuration having piston-c showed highest performance among three identified pistons.

## Future scope

The future scope of the present research work includes the research parameters like,

- The piston of engine having highest performance (Piston-c) will be chosen. The compression ratio of that piston can be augmented by slightly machining the surface of the piston. Performance of engine having modified piston can be evaluated and compared with earlier engine configuration.
- The value of ignition voltage can be increased by increasing the winding of ignition coil of an engine. The influence of increased ignition voltage can be evaluated and compared with earlier configurations.

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