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Evaluation of Lean Operation Limit on Performance Features of Stratified Charge Gasoline Direct Injection (GDI) Engine

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Abstract **– The major parameters that affect spark ignition engine performance, efficiency and emissions are compression ratio, injection timing and mixture composition. To prevent the knock, the compression ratio of gasoline engine cannot be increased beyond 12:1. For the same engine volume increasing the volumetric efficiency raises fuel economy and engine power output. An extensive research work was focused on air fuel ratio and injection timing of a gasoline direct injection engine to evaluate its influence on lean operating limit and performance features of the engine on stratified charge mode operation. A modified engine and a customized fuel injection module were developed for the proposed task. A test matrix of variable spark timing from 14 to 20 degree bTDC, air fuel ratio beyond stoichiometric and variable injection timing from 60 to 140 bTDC at compression stroke was implemented. The experimental results showed that the maximum brake torque timing, air fuel ratio and injection timing were implied significant influence on lean limit operation, fuel economy and exhaust emission reductions. The test report revealed that the developed engine attained lean limit at air fuel ratio 16.7 without affecting speed fluctuations. A module of air fuel ratio 16.7, SOI 100 degree with MBT 18 degree bTDC was identified as a most suitable operating variable for stratified charge gasoline direct injection operation. Reduction of fuel consumption, HC, CO and NOx was observed 15.62%, 83.33%, 14.28%, and 31.08% respectively with reference to the SOI 100 degree bTDC at AFR 16.7 compared with baseline PFI engine operating AFR 14.7.**

Keywords **– Air Fuel Ratio, AFR, Exhaust Emission, GDI Engine, Gasoline Direct Injection, Lean Limit Extension, Maximum Brake Torque Timing, Injection Timing, MBT Operating Variables, Spark Advance, SOI.**

I. Introduction

The parameters that have the greatest influence on engine efficiency are compression ratio and air/fuel ratio. The effect of raising compression ratio is to increase the power output and to reduce the fuel consumption. In gasoline engines, the compression ratio observed is about 9:1to 10:1[1]. To prevent the knock, the compression ratio cannot be increased more. The maximum efficiency or minimum specific fuel consumption occurs with a mixture is weaker than stoichiometric [2]. The port fuel injection engines (PFI) are designed to work at stoichiometric air/fuel ratio, because of that it is impossible to see more improvement in the fuel economy. For the same engine volume, the increasing volumetric efficiency also raises the engine power output. The use of turbocharger is the best choices for increasing volumetric efficiency however, PFI turbocharged engines are still limited to certain operating points of engine [3, 4].

Downsizing the engine is seen as a major way of improving fuel consumption and reducing greenhouse emissions of spark ignited engines. As a contradictory, in the same weight and size, significant decreases in CO_2 emissions, more power and higher break mean effective pressure can be obtained with the aid of advanced fuel injection technology. GDI engines have showed great potential to meet the contradictory targets of lower fuel consumption as well as high torque and power out. The use of GDI engine with turbocharger provides also high engine knock resistance especially at high load and low engine speed [5].

At full load, as the GDI engine operate with throttle, only a small reduction of fuel consumption can be obtained to the PFI engine. At full load GDI engine operates with homogeneous charge and stoichiometric or slightly rich mixture, this engine gives a better power output. In homogeneous operation, fuel starts injecting into cylinder at intake stroke at full loads [6].

GDI engine operate with lean mixture and unthrottled at part loads, this operation provides significantly improvements in fuel economy [7]. At compression stroke, since air is given the cylinders without throttle for stratified charge mode, pumping losses of the GDI engine is minimum at part loads [8]. The improvements in thermal efficiency have been obtained as a result of reduced pumping losses, higher compression ratios and further extension of the lean operating limit under stratified combustion conditions at low engine loads. In the DI gasoline engines, fuel consumption can be decreased by up to 20%, and a 10% power output improvement can be achieved over traditional PFI engines [9].

The design aim of gasoline direct injection engine is to reduce fuel consumption while meeting the requirements of stringent emission regulations and the good drive performance. Gasoline direct injection (GDI) technology proved to be a valuable solution to the aim because the GDI engine adopts lean bum system and has less pump losses at part loads. New control strategies for mixture-preparation and combustion of GDI engines have been proposed and developed in the past decade. However, most of the literatures are focused on GDI engine with high end analysis and little information is available for the effect of injection timing and mixture composition on lean limit operation of engine. Mixture composition effects are usually discussed in terms of air fuel ratio or fuel air ratio.

In order to evaluate the GDI engine performance, a customized test system was developed for the proposed task. The effect of injection timing at stratified charge mode and air fuel on GDI engine performance was analyzed.

II. Design of GDI Engine

The fuel injection system mainly comprises three parts: fuel supply system, electronic control unit (ECU) and injector as shown in Fig. 1. The fuel supply system provides a constant pressure resource for the injector. The ECU controls the injection quantity and injection timing of the injector by special programs according to calculation and analysis of analog and digital inputs of various sensors.

A. Fuel supply system

The fuel supply system is a customized device of GDI system contained by low pressure fuel pump, high pressure pump, pressure regulator and condenser etc. The low pressure fuel pump supplies fuel to the high pressure pump at 5 bar. The high pressure pump pressurizes fuel at 80 bar. The pressurized fuel is then supplied to the injector through a condenser for dissipation of heat absorbed during the pumping process. The fuel rail stores fuel and prevents the pressure from fluctuation caused by nozzle opening and closing. According to the requirements of the test, the specifications of fuel injection system to each part were selected carefully.

B. Electronic control unit

The current ECU uses a compactRIO microcontroller as the main chip and extends other peripheral circuits according to the requirements of electronic control system. The ECU collects signals such as engine speed, TDC, throttle position and air flow through sensors. These signals are processed and then used to control engine operation. Meanwhile, the ECU transmits several relative signals to PC for monitoring.

C. Gasoline direct injector

Usually, inward opening swirl injector is the best choice for the engine with small displacement related to the optimum angle of fuel atomization and spray penetration [1].The minimum injection quantity of the selected injector is 2.0 mg/pulse at 100 bar with static flow rate 10 g/sec. The closing and opening response time of injector is 0.25 msec. The specification of the injector can suit the custom built GDI system development.

Limited by the structure of the cylinder head, the fuel injector is side-mounted by an attachment at an included angle of 36 degree with the horizontal plane between the intake and exhaust valves. The included angle of spray and injector axis is 40 degree.

III. Engine Modification

A single cylinder four stroke 5 HP diesel engine was modified to operate a spark ignited direct injection (GDI) engine for the proposed task. The preset compression ratio (CR) 17:1 of the diesel engine was modified to CR 9:1 by increasing the clearance volume in the engine head. Diesel injection system was removed from the engine and in the place of diesel pump a dummy flange was mounted to stop the oil spillage from the engine crankcase. Engine head was sectioned and examined to identify the location for mounting fuel injector, spark plug and combustion pressure sensor as the space is limited by the structure of the cylinder head for GDI development. The engine head was drilled with two holes of size M14 for mounting gasoline injector and M10 size for combustion pressure sensor. A spark plug was fitted in the place of diesel injector and a high energy ignition coil was mounted on the cylinder body. A GDI injector was side-mounted on the cylinder head with an attachment. The spark plug and injector were connected with NI 9474 digital output module for ignition and injection timing of the engine. Other peripherals such as sensors, crank angle encoder, were also connected with the developed ECU system. The modified engine was interfaced with the ECU as shown in fig.1.

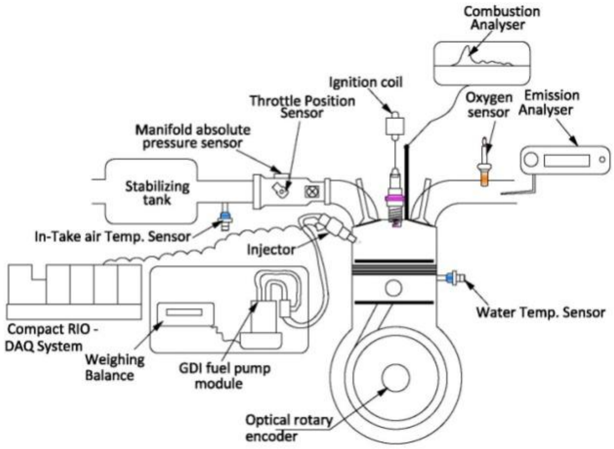

Fig.1. GDI Engine experimental setup

Table I: Specification of GDI Engine

Engine type	Single cylinder, Water cooled
Bore x Stroke	80 x110 mm
Displacement	484 CC
Compression ratio	9:1
Injection angle	Through 145˚.2' bTDC
Combustion chamber	Hemisphere open type
Piston	Flat - bowl piston
Ignition type	Coil on plug
Injector	Inward swirl
Nominal power/speed	3.7 kW/1500 ± 100 rpm
Max. Torque	23.55 Nm

IV. Results and Discussions

A. Setting maximum brake toque (MBT) timing

Variations of spark timing relative to top centre affected the pressure development in the SI engine cylinder. If combustion starts too early in the cycle, the work transfer from the piston to the gases in the cylinder at the end of the compression stroke is too large; if combustion starts too late, the peak cylinder pressure is reduced and the expansion stroke work transfer from the gas to the piston decreases. There exists a particular spark timing which gives maximum engine torque at fixed speed and mixture composition and flow rate. The optimum timing which gives the maximum brake torque called maximum brake torque, or MBT timing - occurs when the magnitudes of these two opposing trends just offset each other. Timing which is advanced or retarded from this optimum gives lower torque [1]

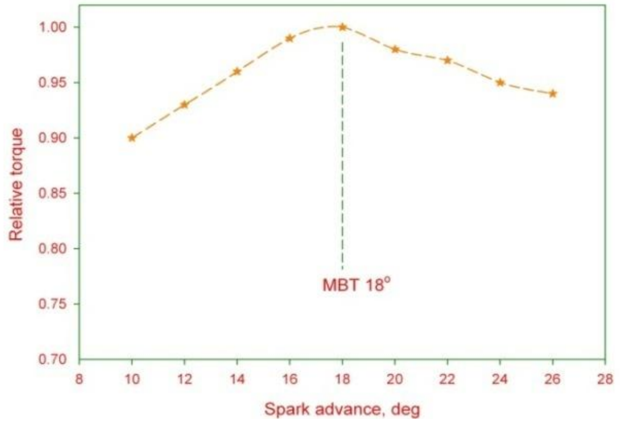

Fig.2. Relative torque Vs spark advance

Tests were conducted at different spark advance angle from 12°to 26° bTDC with an increment of 2°at each of start of injection angle (SOI). The tests report revealed that the engine produced maximum brake torque at specific spark advance. As such spark advance angle 18° bTDC for SOI 100° is shown in Fig 2.

It was observed that a maximum 10% loss of torque was measured against the reduction of 9.8% of average speed before reaching MBT at each of SOI. A minimum 1% loss of torque was recorded against the reduction 1% of speed when the spark angle approached nearer the region of MBT at each of SOI. Maximum torque produced when the engine was operated at 16°, 18° and 20° spark advance for SOI 60, 100, and 140 degree respectively. The engine started gaining speed as spark advances and attained the observed torque 23.55 N – m at MBT with specific air fuel ratio at wide open throttle. The observed torque begin to declined when the engine was operated beyond MBT at each of SOI. It was clearly evident from the trial that the maximum power condition of engine strongly responded to specific engine operating points only.

B. Effect of air fuel ratio on engine performance

Stratified-charge mode is used for light-load running conditions, at constant or low speeds. The fuel has to be injected shortly before the ignition, so that the small amount of air-fuel mixture is optimally placed near the spark plug. This technique enables the usage of ultra lean mixtures with very high air-fuel ratio, impossible with traditional carburetors or even port fuel injection.

Because of early intake and late intake off point of air, the fuel induction period of GDI engine is set in range of 30° to 180° after bottom dead centre. The modifications restrict the fuel injection period up to 145°.2' bTDC crank angle for stratified charge operation. The table II shows engine operating points at specified AFR.

Table II: GDI Engine Operating Point

Air fuel ratio 14.7,15.7, 16.7, 18, 19			
SOI	60° CA	100° CA	140° CA
MBT	16° bTDC	18° bTDC	20° bTDC

The use of electronic control fuel injection system makes the engine possible to burn lean mixture. Fig.3 exhibits the performance comparison of the GDI engine at different air-fuel ratios under speed condition of 1500 rpm. In the course of inspection, it was found that the engine speed became unstable when air-fuel ratio was reached leaner than 17. At AFR 17, the coefficient of speed fluctuation is exceeding 10%, hence the A/F 16.7 was chosen as lean burn limit of the test. The fig.4 show the fuel consumption decreases as the mixture gets leaner. Compared with that of A/F 14.7, the fuel consumption descends 16% from 0.136 kg/h to 0.114 kg/h. The reason why the lean bum limit is not very high can be explained as follows: when the air-fuel ratio reaches a certain value, flammable mixture forms hardly near the spark plug because of the weak air motion at low speed condition. This leads to the low flame propagation velocity and insufficient mixture combustion.

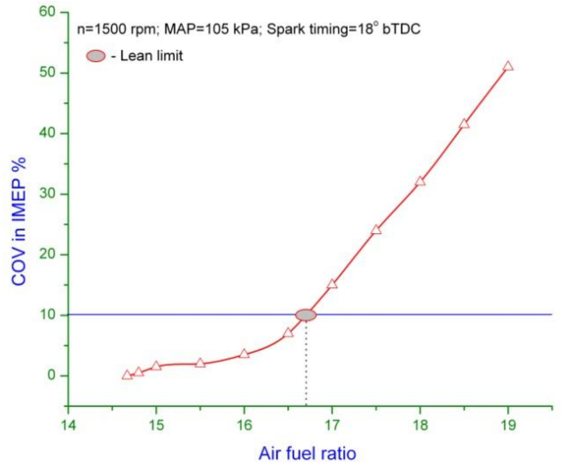

Fig.3. COV in IMEP Vs Air fuel ratio

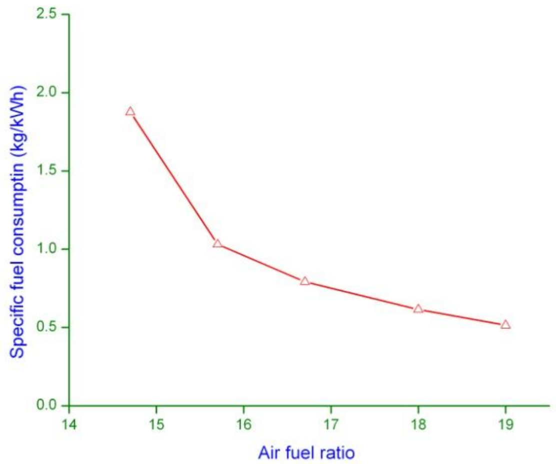

Fig.4. Specific fuel consumption Vs air fuel ratio

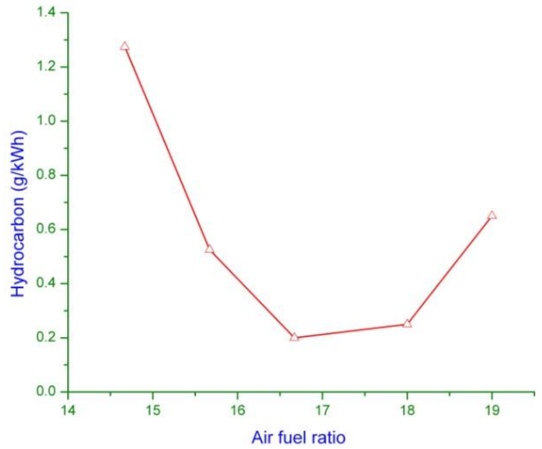

Fig.5. Hydrocarbon Vs air fuel ratio

The HC emission descends gradually as the air-fuel ratio increases from A/F=14.7 and reaches the minimum at about A/F=16.7 as shown in fig.5. At this time the HC emission has reduced nearly by 20% from that of stoichiometric operation due to better spray vaporization and combustion process. The HC starts to rise when the air-fuel ratio is further enlarged. The contributing factors for the rise of HC emission are the extended quench area on the cylinder wall, decreased burning speed and unstable combustion.

The CO emission mainly relies on the excess air ratio. Fig.6 show the CO has a reduction of 89% at A/F 16.7 as compared to that of stoichiometric air-fuel ratio and then holds on the line.

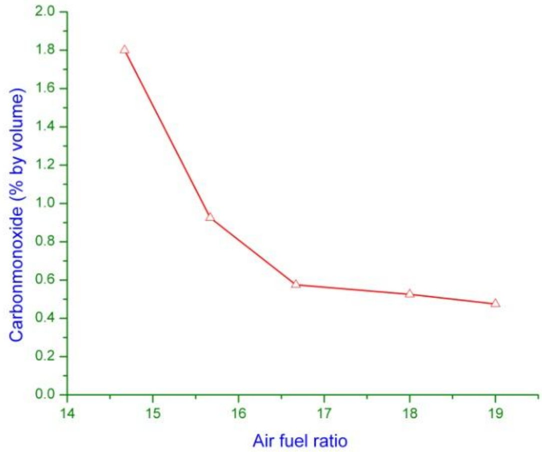

Fig.6. CO Vs air fuel ratio

The flame temperature, oxygen – enriched flame frontier and duration time of high temperature play an important part in the formation of NOx [1]. Because of the low combustion temperature caused by the latent heat of vaporization, the GDI engine provides a lower NOx emission than that obtained with the port fuel injection (PFI) engine operation. The dilution effect of air will reduce the combustion temperature further as the air fuel ratio increases. The NOx concentration at AFR 19 almost decreases 77% compared with that of AFR 16.7 as shown in fig.7. But NOx concentration registered its peak at AFR 16.7 as the air fuel mixture undergoes lean combustion. Despite high NOx emission, operating the engine at this AFR was inevitable as NOx concentration could be possibly treated by after treatment device. The sequence of this effect could be witnessed in low HC and CO emission generation.

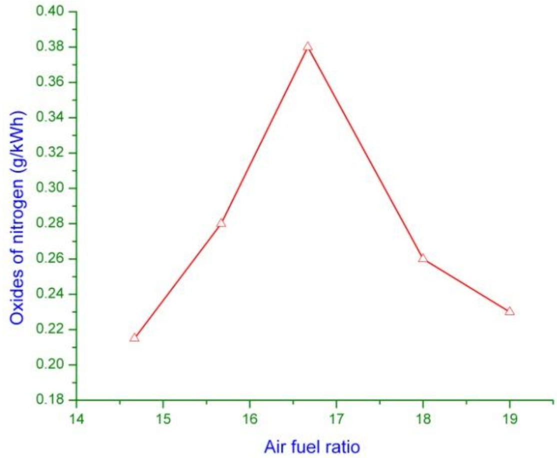

Fig.7. Oxides of nitrogen Vs air fuel ratio

C. Effect of injection timing on engine performance

Owing to compromised features of AFR 16.7, engine experiments were performed at that AFR with specified SOI and MBT as shown in table 2.As illustrated in Fig.8 advancing the start of injection angle at the specified air fuel ratio, the specific fuel consumption descends gradually and achieves the minimum at S0I 100° bTDC. When SOI is 60° bTDC, the distance between the piston crown and the injector is relatively short, and the piston moves up rapidly. The fuel has not enough time to mix with air before impingement, so thicker fuel film occurs on the piston crown, this will lead to the poor combustion.

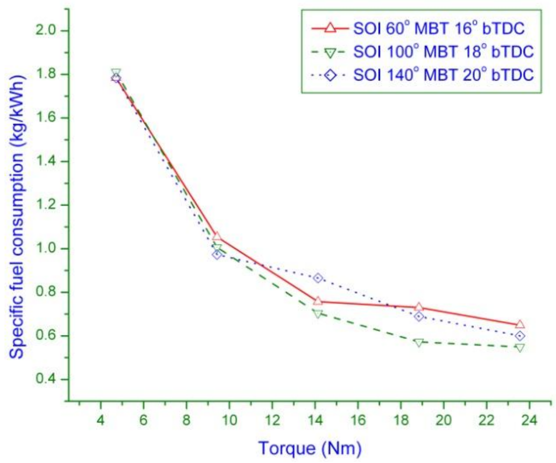

Fig.8. Specific fuel consumption Vs SOI

On the contrary, when SOI is 100°bTDC, injection begins in the middle of compression stroke, the piston crown is comparatively far from the injector, and the piston moves up with a rapid velocity, therefore the spray impingement is reduced. This will improve the mixing and combustion. When SOI is 140° bTDC, injection begin shortly after the intake valve closes, hence longer time is available for spray droplet vaporization, so the charge

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cooling effect becomes strong and volume efficiency increases. However injection performed at SOI 140° and beyond this limit makes air fuel mixture too lean for stratified combustion.

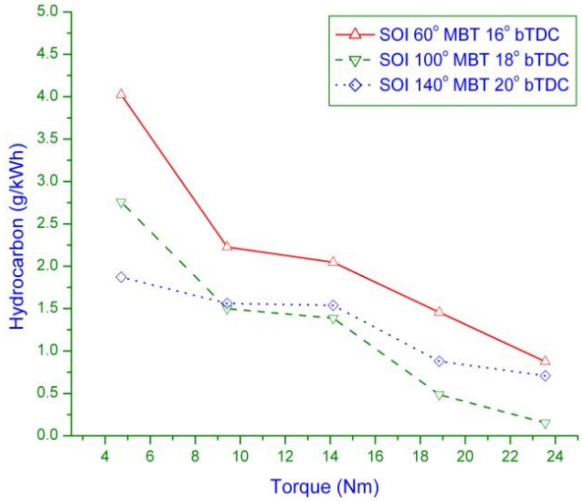

Fig.9. Hydrocarbon Vs Torque

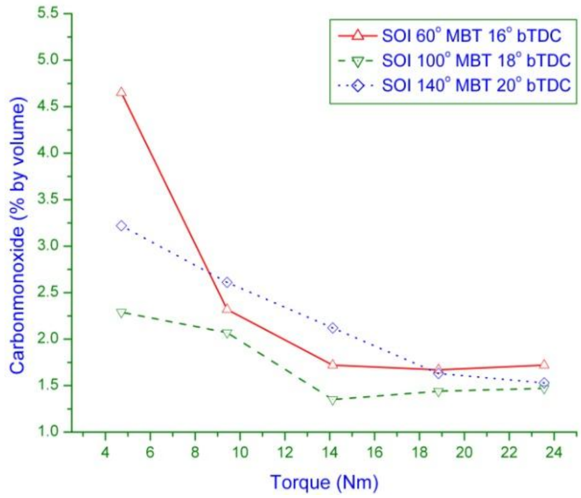

Fig.10. Carbon monoxide Vs Torque

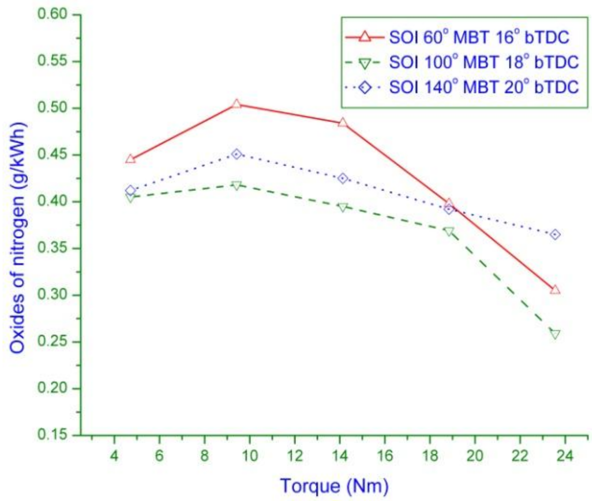

Fig.11. Oxides of nitrogen Vs Torque

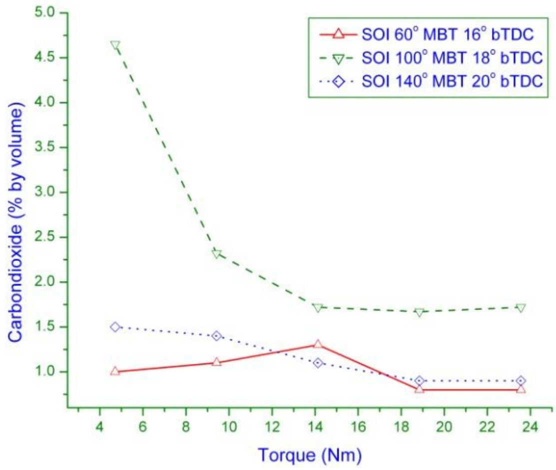

Fig.12. Carbon dioxide Vs Torque

The emission parameter can be explained by referring the figure 9, 10, 11 and 12. With the increase of the start of injection angle, the HC emission descends gradually and reaches the lowest at SOI 100° bTDC as shown in figure 9. The reasons are similar with those of fuel consumption. It was observed from figure 10, that the injection timing has large effect on CO emission as SOI 100° bTDC provides comparable CO concentration with other SOI. At the same air-fuel ratio, the NOx emission increases up to part load and declines continuously as the start of injection angle increases. These can be attributed to that the smaller SOI angle produces more increased mass of air in the cylinder resulting from the decrease of the charge temperature due to the latent heat of vaporization of the fuel, thus the flame temperature will be relatively higher which helps the production of NOx emission. According to the figure 11, AFR 16.7 with SOI 100° bTDC are appropriate for the present test condition. The sequential action also reflected in figure 12 as SOI 100° bTDC registered optimum carbon monoxide emission because of better combustion characteristics. Compared with those of the other injection timing, the fuel consumption, HC, CO and NOx has reduced by 20.45%, 33.33%, 38.63% and 19.38 for part load operation and 15.62%, 83.33%, 14.28 % and 31.08 % for full load operation respectively with reference to the operating variables of AFR 16.7, SOI 100° with spark advance of 18° bTDC.

D. Effect of air fuel ratio on cylinder pressure

In cylinder combustion related data are important parameters to reveal air fuel ratio effect on engine performance if not more as much as cyclic variation. The cylinder pressure was measured for individual cycle at each crank angle by a piezo electric pressure pickup. The combustion event must be properly located relative to top - centre to obtain the maximum power or torque. The combined duration of the flame development and propagation process is typically between 30 and 90 crank angle degrees. Combustion starts before the end of the compression stroke, continues through the early part of the expansion stroke and ends after the point in the cycle at which the peak cylinder pressure occurs [1].

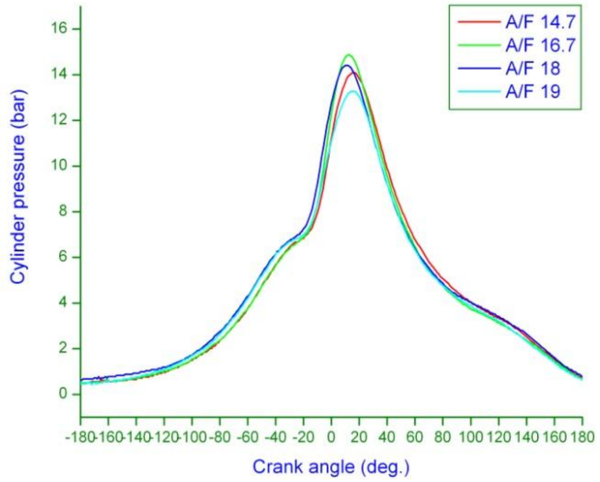

Fig.13. Cylinder pressure Vs crank angle

As the start of the combustion process is progressively advanced before TDC by MBT timings, the compression stroke work transfer increased. Similarly, if the end of the combustion process is progressively delayed by retarding the spark timing, the peak cylinder pressure occurred later in the expansion stroke and is reduced in magnitude. These changes reduce the expansion stroke work transfer from the cylinder gases to the piston.

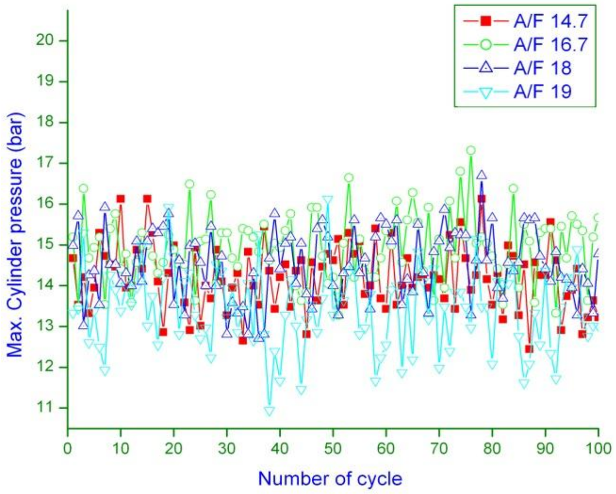

Fig.14. Maximum cylinder pressure Vs number of cycle

In internal combustion engines, the cylinder pressure profile variation is caused by uncertainty in charge inlet and its imperfect mix during combustion, moreover, ignition time when combustion starts is also important to cylinder pressure [1, 7]. Even though the MBT was fixed to all tests, the variation in air fuel ratio and its subsequent mixing has caused different in cylinder pressure profile at each operating variable. It is clearly evident from figure 13, 14 the operating variable combination AFR 16.7, SOI 100° bTDC with MBT 18° showed better combustion pressure profile. Being AFR 16.7 was identified as lean limit air fuel ratio and this operating module exhibits significant difference in cylinder pressures profiles and huge difference at TDC.

V. Conclusion

When the engine operates in the lean burn combustion mode, the fuel economy and emissions are two prime aspects mainly influenced by the air-fuel ratio. Fuel consumption and emissions can be improved further more by retarding the injection timing appropriately, but its importance is less than extending the lean burn limit.

The experimental results showed that air fuel ratio, injection timing and MBT influences on lean limit operation and played significant role for improving fuel economy and exhaust emissions. The conclusive report revealed that the developed engine attained lean limit at AFR 16.7, SOI 100° with MBT 18° bTDC. The same operating variable can also be referred for better fuel economy and exhaust emissions achievement. Compared with those of the baseline PFI engine, the fuel consumption, HC, CO and NOx has reduced by 15.62%, 83.33%, 14.28 %, and 31.08 % respectively with reference to lean limit operation on stratified charge mode.

Appendix

AFR – air fuel ratio
bTDC – before top dead centre
BDC – bottom dead center
CA – crank angle
CO – carbon monoxide
CR – compression ratio
GDI – gasoline direct injection
HC – hydrocarbon
MBT – maximum brake torque
NOx – oxides of nitrogen
PC – personal computer
SI – spark ignition
SOI – start of injection
TDC – top dead center

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