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Improving the Performance of Two Stroke Spark Ignition Engine by Direct Electronic CNG Injection

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Abstract

Two stroke spark ignition engines have high exhaust emissions and low brake thermal efficiency due to the short circuiting losses and incomplete combustion, which occur during idling and at part load operating conditions. To eliminate the short circuiting losses, direct injection has been developed. Electronic CNG injection system was developed for better fuel economy and reduced emissions. The fuel and time maps were generated for the various operating conditions of the engine using an electronic system. For the mapping, the visualization tool was used to estimate the fuel injection time and delivery quantity for required running conditions of the engine. Experiments were carried out at the constant speed of 3500 rpm with a compression ratio of 12:1. The performance and emission characteristics of direct CNG injection system and carbureted engine are described. The above studies indicate the improvement in brake thermal efficiency from 15.2% to 24.3%. This is mainly due to significant reduction in short circuit loss of fresh charge and precise control of air fuel ratio. The pollution levels of HC and CO were reduced by 79.3% and 94.5% respectively compared to a conventional carbureted engine.

Keywords: Two-stroke spark ignition engine; direct injection; CNG; microcontroller;

1. Introduction

In developed and developing countries considerable emphasis is being laid on the minimization of pollutants from internal combustion engines. A two-stroke cycle engine produces a considerable amount of pollutants when gasoline is used as a fuel due to short-circuiting. These pollutants, which include unburnt hydrocarbons and carbon monoxide, which are harmful to beings. There is a strong need to develop a kind of new technology which could minimize pollution from these engines. Direct fuel injection has been demonstrated to significantly reduce unburned hydrocarbon emissions by timing the injection of fuel in such way as to prevent the escape of unburned fuel from the exhaust port during the scavenging process.

The increased use of petroleum fuels by automobiles has not only caused fuel scarcities, price hikes, higher import bills, and economic imbalance but also causes health hazards due to its toxic emissions. Conventional fuels used in automobiles emit toxic pollutants, which cause asthma, chronic cough, skin degradation, breathlessness, eye and throat problems, and even cancer.

In recent years, environmental improvement (CO_2, NO_x and Ozone reduction) and energy issues have become more and more important in worldwide concerns. Natural gas is a good alternative fuel to improve these problems because of its abundant availability and clean burning characteristics.

1.1. The objectives of present study are:

To compare the performance of a carbureted and injected engine at constant speed.

Direct injection system was developed which eliminates short circuiting losses completely and injection timing was optimized for the best engine performance and lower emissions.

In a lean burn engine, air fuel ratio is extremely critical. Operation near the lean mixture limit is necessary to obtain the lowest possible emission and the best fuel economy. However, near the lean limit, a slight error in air-fuel ratio can drive the engine to misfire. This condition causes drastic increase in hydrocarbon emission; engine roughness and poor throttle response [2-4]. A reliable electronic gaseous fuel injection system was designed and built in order to control the engine and also for the evaluation of control strategies. The electronic control unit is used to estimate the pulse width of the signal that would actuate the fuel injector and the start of fuel injection. The experiments were carried out on the engine using state-of-art instrumentation.

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2. Fuel Induction Techniques

The performance characteristics of an engine and the concentration level of the exhaust emissions depend, to a large extent, on the combustion pattern. It directly depends on fuel system, which provides an appropriate mixture of fuel and air to the engine at the appropriate point in the cycle. The fuel air mixture must be in right proportion as per the condition of the speed and load on the engine. The overall engine behavior depends upon the fuel induction mechanism. Introduction of a CNG kit to the existing gasoline engine hardware does not involve any substantial modifications except inducting the mixture into the intake manifold. However, in spite of the excellent characteristics and various advantages of CNG as a fuel in vehicles, it has certain problems, when used in vehicles, like backfiring during suction, knocking at higher compression ratio with advanced spark timing; these problems are due to inappropriate technology used for the formation of the mixture.

In consideration of the inherent constraints in the design of carburetor, the engine manufacturers and automobile industries now are switching over to fuel injection system. The mode of fuel injection from an injector plays a critical role in determining the performance characteristics of an engine. The lean burn for the engine operation can be easily achieved with this technique. Keeping in view the requirement of the CNG fuel, an electronic direct CNG injection system was designed and developed in the present experimental work.

3. New Direct CNG Injection System

The short-circuiting losses of the two-stroke engine can be eliminated by directly injecting the fuel into the cylinder after the closure of the exhaust port. This requires the development of an electronically controlled direct fuel injection system fitted with suitable modification to the engine.

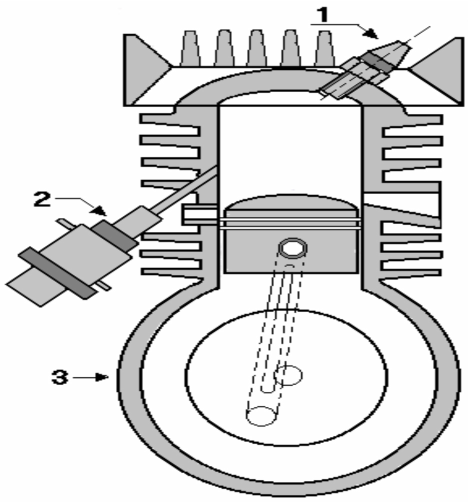
The Figure 1 shows the cylinder wall injection, with an injection nozzle installed in the cylinder wall. The injection nozzle was tilted by 40^0 from the horizontal and injects the fuel upward, different from the method of injecting the fuel at a right angle to the cylinder axis as employed by Vieillendent [5], Blair [6], etc. The spray would be concentrated on the upper position of the combustion chamber near the spark plug. The location of the nozzle on the cylinder was determined from the pressure crank angle diagram corresponding to an in-cylinder pressure of 2 bar attained after the closure of the exhaust port. Corresponding to this crank angle a hole is drilled in the cylinder bore at an inclination of 40^0 from horizontal. A water-cooled adaptor was designed for cooling the injector to prevent excess heating of the injector.

4. Experimental Test Setup

A 98 cc, two-stroke spark ignition engine was used in this study. Table 1 gives the engine specification of the engine.

Table 1 : Specifications of the engine

Make	*Yamaha*
Bore	*50 mm*
Stroke	*50 mm*
Max. Power	*8 kW @ 7500 rpm*
Displacement	*98 cc*
Ignition timing	*30^0 BTDC*



1. Spark plug,
2. Electronic CNG injector with water-cooled adaptor,
3. Crankcase.

Figure 1: Schematic diagram of direct cylinder wall injection system

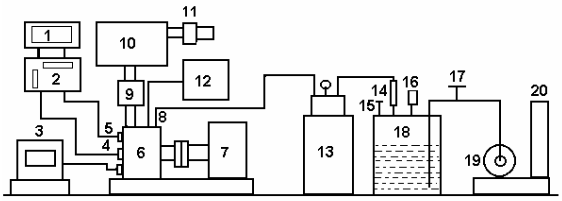
The Figure 2 shows the schematic diagram of experimental setup and the engine instrumentation. The fuel system consists of high-pressure storage cylinders with a filling pressure of about 22 MPa, a regulator to reduce the line pressure to 200 kPa and a high-pressure line for connecting the cylinder to the regulator. An injection system controller was used to control the pulse width of the injector. The engine was connected to brake dynamometer for loading purposes. Fuel consumption is measured using weighing machine and rotameter. Air consumption is measured by an air flow meter.

A pressure transducer in conjunction with a charge amplifier was used to measure the cylinder pressure. The transducer was mounted in the cylinder head. Signals of crankshaft angle were derived from a shaft encoder rigidly attached to the engine crankshaft. The top dead center (TDC) signal of the encoder was checked with the engine TDC, under dynamic conditions. The encoder provides the necessary signals for the data acquisition system to collect cylinder pressure at every degree during engine cycle. The exhaust emissions of HC and CO are measured with an exhaust gas analyzer. The performance testing of the engine is carried out at constant speed.

5. Results

5.1. Performance of Direct CNG Injection

The performance of the direct injection system was tested at 3500 rpm. The injection timing was initially set such that the start of injection takes place immediately after the closure of exhaust port (250^0 ATDC). The crank position sensor sends a signal corresponding to the closure of the exhaust port and then the injection is immediately started. The injection timing was optimized for the best engine performance and emissions. Any further advancement of the injection timing above the optimum injection angle results in poor performance and higher emissions as short-circuiting losses predominate. The performance of direct injection system with the optimized injection advance angle is compared with carbureted engine at a compression ratio of 12:1.



1. Computer *2. Signal analyzer*
3. Exhaust gas analyzer *4. TDC pickup*
5. Pressure pickup *6. Engine*
7. Dynamometer *8. Gas injector*
9. Air filter *10. Surge tank*
11. Air flow meter *12. Fuel injection system*
13. Stabilizing tank *14. Gas rotameter*
15. Safety valve *16. Pressure gauge*
17. Expansion valve *18. Flame trap*
19. CNG Cylinder *20. Weighing machine*

Figure 2: Schematic diagram of experimental setup

5.2. Brake Thermal Efficiency

The Figure 3 shows the variation of brake thermal efficiency with brake mean effective pressure (BMEP) at 3500 rpm. The maximum brake thermal efficiency of the direct injection mode (DI) is 24.3% at BMEP of 3.5 bar compared to carbureted engine mode (CE) is 15.2% at BMEP of 2.8 bar. This is due to reduction in short-circuiting losses and increase in air–fuel ratio. This indicates that the engine can operate in leaner air-fuel ratios without loss of power. This is achieved because of the precise timing and metering of the fuel by the microcontroller fuel injection system.

Figure 4 shows the variation of brake thermal efficiency with equivalence ratio. The maximum brake thermal efficiency of direct injection mode is 24.3% at an equivalence ratio of 0.88 where as in carbureted mode is 15.2% at an equivalence ratio of 0.99. Brake thermal efficiency increases from lean to rich and starts decreasing at engine rich mixtures. For the same equivalence ratio, the carbureted engine gives lesser brake thermal efficiency compared to the injected engine. This is due to incomplete combustion of the charge due to mixture limit inside the combustion chamber at a given compression ratio. Hence, the amount of fuel charge to give the mechanical power gets reduced and thus reduces the brake thermal efficiency.

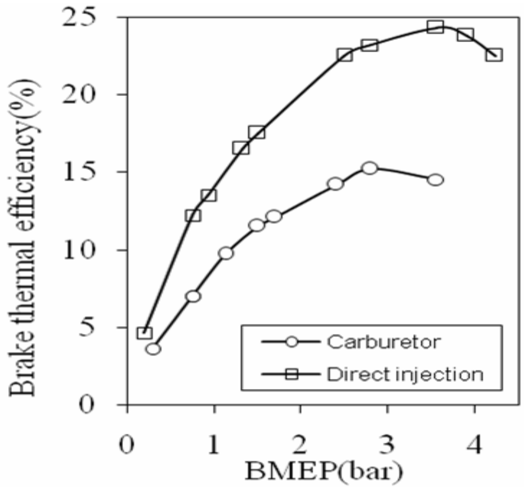


Figure 3: Variation of break thermal efficiency with BMEP at 3500 rpm

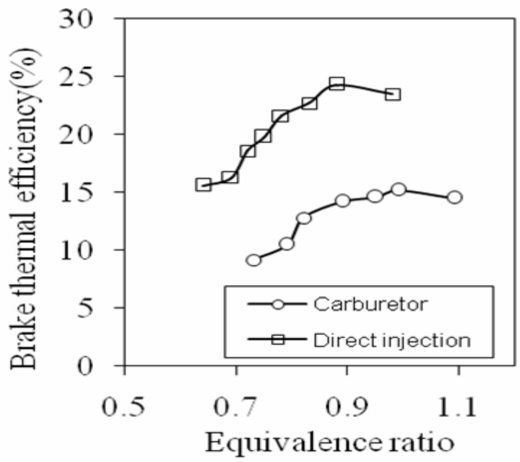


Figure 4: Variation of break thermal efficiency with equivalence ratio at 3500 rpm

5.3. Carbon Monoxide Emissions

The Figure 5 shows the variation of CO with equivalence ratio at 3500 rpm. Carbon monoxide being the product of incomplete combustion, therefore it is totally dependant on the air fuel ratio. Owing to the gaseous nature of compressed natural gas, it easily mixes with air because of diffusivity at high pressure. CO emissions with lean mixture are reduced because of CO getting converted into CO_2 with surplus amount of oxygen. In injection mode, significant lower concentration of CO is observed over entire range of operation.

The maximum reduction of CO emissions in direct injection system compared to carbureted engine is 94.5% at an equivalence ratio of 1.2. This reduction is due to leaner air fuel ratio mixtures and elimination of short circuiting losses.

5.4. Hydrocarbon Emissions

The Figure 6 shows the variation of HC emissions with equivalence ratio at 3500 rpm. The main source of hydrocarbons is due to the composition and patchy combustion occurring due to uneven mixture formation.

The maximum reduction of HC emissions in direct injection system compared to carbureted engine is 79.3% at an equivalence ratio of 0.63. This reduction is due to elimination of short-circuiting losses and precise control of air fuel ratio.

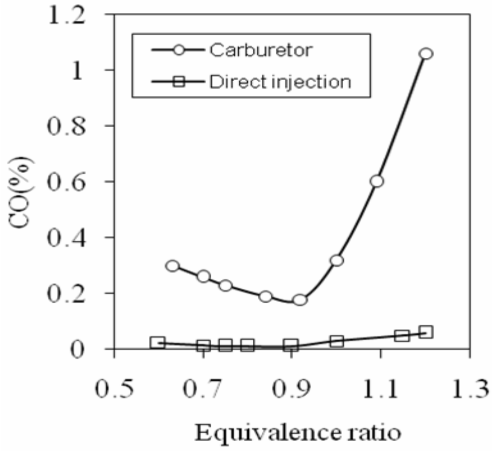


Figure 5: Variation of CO emissions with equivalence ratio at 3500 rpm

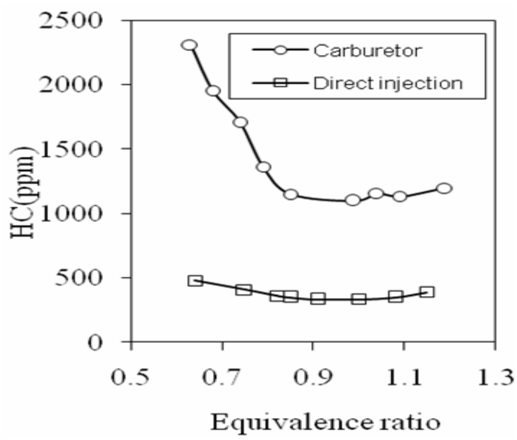


Figure 6: Variation of HC emissions with equivalence ratio at 3500 rpm

5.5. Coefficient of Variation of Peak Pressure

Figure 7 shows the variation of C.O.V. of peak pressure with BMEP for direct injection system and carbureted engine. The C.O.V of peak pressure of the direct injection system is lower than the carbureted engine at tested speed and loads. The reduction in C.O.V. peak pressure of the direct injection system compared to carbureted engine at minimum loads(BMEP of 0.75 bar) is 81.25%, whereas at maximum loads(BMEP of 4 bar), the reduction of C.O.V of peak pressure is 83.8%.

5.6. Coefficient of Variation of Indicated Mean Effective Pressure

Figure 8 shows the variation of C.O.V. of IMEP with BMEP for direct injection system and carbureted engine. The C.O.V of IMEP of the direct injection system is lower than the carbureted engine at tested speed and loads. The reduction in C.O.V. of IMEP of the direct injection system compared to carbureted engine at minimum loads (BMEP of 0.75 bar) is 73.9%, whereas at maximum loads (BMEP of 4 bar), the reduction of C.O.V of IMEP is 71.4%. At light loads, the C.O.V of IMEP of engine combustion increases rapidly. This reveals that the phenomena of mixture misfiring at light loads. The comparison between the C.O.V of IMEP for carburetor type engine and direct injection engine indicates that the fuel injected CNG engine is much more stable engine due to the better control of air fuel ratio.

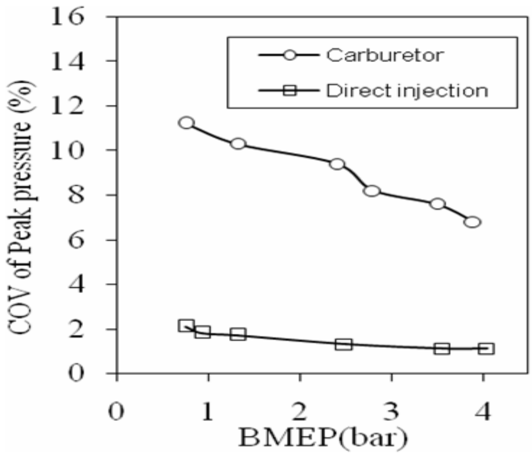


Figure 7: Variation of COV of peak pressure with BMEP at 3500 rpm

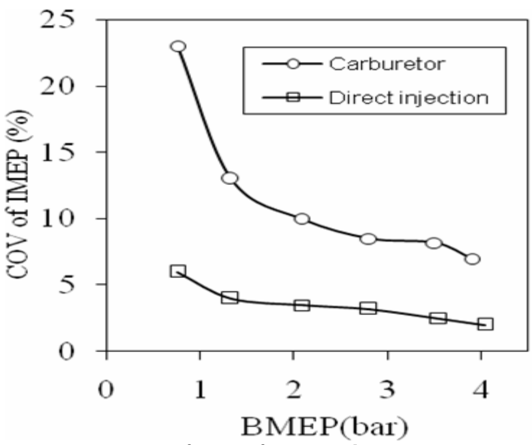


Figure 8: Variation of COV of IMEP with BMEP at 3500 rpm

5.7. Rate of Heat Release

The rate of heat release of the direct injection system is higher than the carbureted engine. The maximum increase in the rate of heat release compared to the carbureted engine as shown in Figure 9.

The increase in rate of heat release of the direct injection engine compared to carbureted engine is 38.1% at 3500 rpm. The increase in rate of heat release indicate that the combustion in the direct injection is faster than the carbureted engine due to the combustion of the relatively lean air fuel mixtures and reduction of short-circuiting loss of fresh charge through the exhaust port.

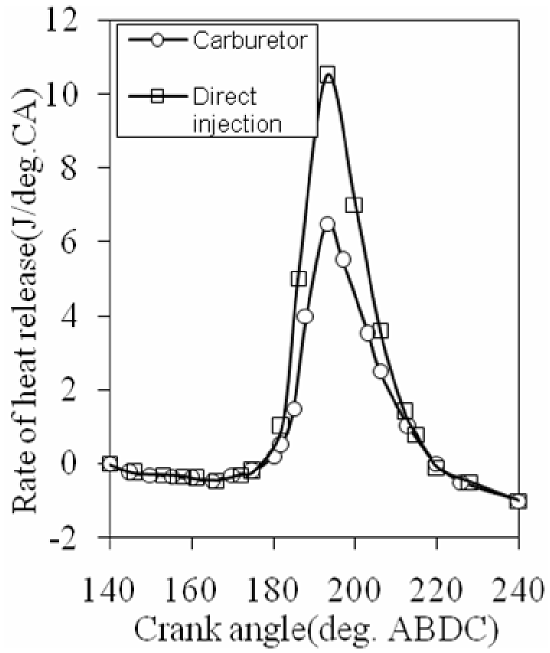


Figure 9: Variation of heat release with crank angle at 3500 rpm

6. Effect of Fuel Injection Timing on Engine Load

Fuel injection timing has a strong influence on the mixing process. In homogeneity in the cylinder charge creates limitations in the optimization of natural gas engines. It has been demonstrated, that poor mixture distribution increases the level of cycle-to-cycle combustion variability [7-8]. Mixture formation in a direct injected gasoline fueled engine is largely dependent on the atomization and evaporation of the fuel. While this complexity is not present in gaseous-fueled engines since the mixing process is far from trivial. Due to the lower momentum of injected fuel, the degree of mixing in the region of the jet is lower in the gaseous case than in the liquid case. For this reason, it is important to utilize the timing of the fuel injection event to optimize the mixing process [9].

In the present direct CNG injection system, the injection timing was optimized for the best engine performance and low emissions. Any further advancement of the injection timing above the optimum injection angle results in poor performance and higher emissions as short-circuiting losses predominate. The software of the control system was modified so that the injection timing and injection duration could be varied from TDC to any crank angle.

The Figure 10 shows that the performance of the direct injection with the injection advance angle of 250^0, 237^0 and, 233^0 after top dead center (ATDC). The maximum brake thermal efficiency with 250^0 advance angle is 20% at BMEP of 3.36 bar. When injection is advanced gradually, the performance of the engine was found to improve. At 237^0 injection advance angle, the maximum brake thermal efficiency is 24.3% at BMEP of 3.55 bar. This is due to increase in the time available for mixture formation. Any further increase in the injection advance angle of 233^0 results in reduction in maximum brake thermal efficiency is 22.1% at BMEP of 3.45 bar. This is due to the fact that the exhaust gases may carry a small fraction of injected fuel while scavenging.

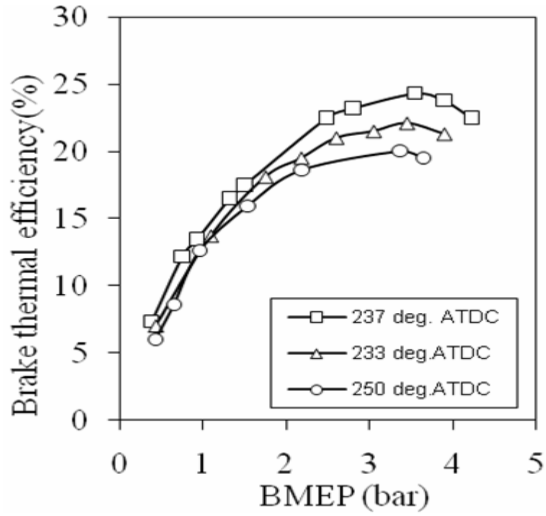


Figure 10: Variation of break thermal efficiency with BMEP at 3500 rpm

Variation of CO and HC emissions with equivalence ratio for different injection advance angle of 250^0, 237^0, and 233^0 ATDC are shown in Figures 11 and 12.

The CO and HC emissions are higher for 250^0 advance angle due to the reduction in mixture formation time and CO and HC emissions are higher for injection advance angle of 233^0 due to short-circuiting. CO and HC emissions are minimum with injection advance angle of 237^0. Hence, the optimum injection advance angle is 237^0 at 3500 rpm.

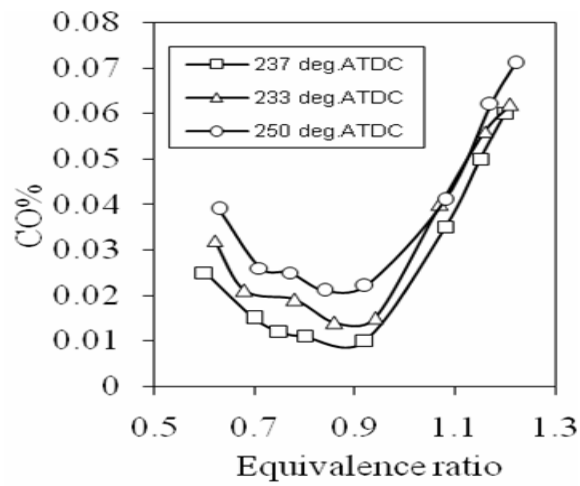


Figure 11: Variation of CO emissions with equivalence ratio at 3500 rpm

7. Conclusions

The following are the important conclusions based on the experimental analysis of the electronic CNG injected two-stroke spark ignition engine.

1. The maximum brake thermal efficiency of the direct injection engine is 9.1% more than the carbureted engine at 3500 rpm.

2. There is 79.3% reduction in the unburnt hydrocarbon with electronic fuel injection at 3500 rpm.
3. The CO emission is 94.5% less in the injected engine compared to the carbureted engine at 3500 rpm.
4. The maximum value of brake thermal efficiency of 24.3% is achieved at compression ratio of 12 with a spark timing of 30 ^{0} before top dead center.
5. With the on-line injection timing control the injection advance angle is optimized to 237^0 after top dead center at 3500 rpm.

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