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Full Paper

Effect of high-pressure fuel injection on two-stroke spark-ignition engine performance

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Abstract: The present paper investigates the effects of direct fuel injection on a two-stroke spark-ignited engine at moderately high-pressure. A conventional carburetted engine was modified to facilitate gasoline injection from the cylinder head. Experiments were carried out in both the conventional carburettor mode and modified gasoline direct injection mode. The measurements show that gasoline direct injection results are superior to the carburettor mode. The brake mean effective pressure exhibits significant improvement of about 20.1% as compared to that of the conventional carburetted engine. Also, the brake thermal efficiency is 16.3% higher than that of the conventional engine. This enhancement in thermal efficiency may be attributed to the fine atomisation and vapourisation of the fuel and the reduced losses due to fuel short-circuiting. The elimination of fuel short-circuiting losses results in a 17% reduction in brake specific fuel consumption. The gasoline direct injection mode shows 27.5% and 88.5% reduction in carbon monoxide and unburned hydrocarbon emission respectively as compared to the conventional carburettor mode.

Keywords: direct fuel injection, fuel short-circuiting, spark ignition, two-stroke engine

INTRODUCTION

The use of the two-stroke spark-ignited (2S-SI) engines is fairly extensive in several applications. The appeal of these engines as power sources for automotive applications is in the design, low cost and high power-density. Compared with a four-stroke engine, a 2S-SI engine has the advantages of double firing frequency without valve mechanism or associate driving parts. However, these carburetted 2S-SI engines have been reported to emit high levels of hydrocarbons (HC) and carbon monoxide (CO). Therefore, these engines have low fuel economy and high emissions [1, 2]. The operating mechanism of the 2S-SI engine is such that the transfer port and exhaust port open simultaneously, resulting in the fuel-air mixture or fuel going out directly through the exhaust port. This loss of fuel-air mixture or fuel is called charge short-circuiting. It is the leading cause of the high level of emissions and low fuel economy during the scavenging phenomena [3-5]. Incomplete scavenging results in gas residues in the combustion chamber. The scavenging process is necessary for the internal combustion engine, especially the 2S-SI engine, as it affects the quality of the trapped charge, power output, fuel economy and emissions in such engines [6-8].

A survey of the literature on 2S-SI engines reveals that high fuel consumption and pollutant emissions are the results of charge short-circuiting and incomplete scavenging phenomena. Scavenging is a gas exchange process during combustion in the operation of the engine. The loss of fuel or fuel-air mixture through the exhaust port opening is known as short-circuiting during the scavenging process. Many researchers have attempted to improve engine performance by eliminating fuel short-circuiting during the scavenging process. Yamagishi et al. [9] injected gasoline through the cylinder head using a mechanical injector. Vieilledent [10] performed electronically controlled injection through the cylinder barrel with the injector being located on the opposite side of the exhaust port. Douglas and Blair [11] carried out injection from a swirl chamber through the cylinder barrel and obtained a significant reduction in unburned HC emissions and fuel consumption. Advanced scavenging techniques and direct injection (DI) can significantly improve the performance of 2S-SI engines. Direct in-cylinder fuel injection after the closure of the exhaust port eliminates fuel short-circuiting, resulting in lower fuel consumption and pollutant emissions [12-14]. Moreover, the performance parameters of the 2S-SI engine can be further enhanced by charge stratification [15-18], advanced combustion concept such as controlled auto-ignition/ homogeneous charge compression ignition [19-21], and partially premixed combustion [22, 23].

To completely understand the fresh fuel or charge short-circuiting during the scavenging process in the 2S-SI engine, three-dimensional computational fluid dynamics simulation techniques have been employed to predict the flow behaviour inside the combustor. Mallikarjuna et al. [24] reported that the fuel trapping efficiency in the combustion chamber is affected by the port configurations in the 2S-SI engine. Pradeep et al. [25] experimented with manifold injection (MI) and DI systems on a 2S-SI engine running on gaseous liquefied petroleum gas at 25% and 100% throttle conditions. They reported that the DI system was more efficient than the MI system in all operating conditions. The DI system significantly increased brake thermal efficiency (BTE) and reduced HC emission in both conditions due to the reduction in fuel short-circuiting. A 12.5% and 40% increase in BTE and a 93% and 88% decrease in HC were obtained at 25% and 100% throttle respectively. However, CO emission was higher due to rich fuel mixture while nitrogen oxides (NO_x) emission was lower due to charge stratification in the DI engine.

Kumarappa and Prabhukumar [26] developed an electronic DI system to eliminate fuel short-circuiting. A compressed natural gas fuel injector was installed in the cylinder wall and tilted 40° from the cylinder axis, and fuel was injected upward in direction. They reported that scavenging efficiency and thermal efficiency increased by about 95% and 9.1% respectively. Also, 79.3% decrease in HC and 94.5% decrease in CO were obtained. Loganathan and Ramesh [27] compared semi-DI and MI on the 2S-SI engine using 8-bar fuel injection pressure. They noticed that semi-DI is superior when compared to MI. The thermal efficiency in semi-DI mode significantly increased from 23% to 28% and HC emission decreased from 3000 ppm to 728 ppm. It was also reported that NO_x emission increased due to better combustion. Darzi et al. [28] analysed the performance of

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low-pressure DI to improve efficiency and reduce HC emission and compared it with prior port-injection results. They confirmed that trapping efficiency is a function of engine speed, which shows a significant effect on engine overall efficiency. Ausserer et al. [29] studied an energy balance audit on a small two-stroke engine. The brake power was estimated at 13% of the total inward chemical energy of the fuel while the short-circuiting loss was 40-60% and incomplete combustion was around 20%, making it the second largest loss pathway.

The focus of the majority of the research reported so far has been on the low-pressure DI technique [30, 31], which is relatively easy to implement and is a cost-effective system. The time taken for mixture formation is a significant issue in the low-pressure fuel-injected two-stroke SI engine. High-pressure DI techniques can resolve this issue. Spray characteristics and fuel atomisation are improved with the use of high-pressure DI technology. The injection timing is the most critical factor in the working of the 2S-SI engine. The low-pressure DI technique requires far less complexity to maintain fuel pressure.

On the other hand, the high-pressure DI technology system requires heavier components to maintain high fuel rail pressure. The working of the engine is complex even though the size and simplicity of the engine have been maintained. Recently, Dubey and Ramesh [32] performed their experiment on a 200-cc two-stroke engine with different fuel injection pressures (100, 130 and 150 bar) and used a continent-made injector. They noticed that the two-stroke gasoline direct injection (GDI) is superior when compared to the two-stroke MI. They also observed that the fuel trapping efficiency for the two-stroke GDI at 130-bar is nearly 11.3% higher than that of the two-stroke MI. The two-stroke GDI has 40.4% saving in fuel and 40.6% increase in BTE with a reduction of 90% HC and 80% CO emissions at the same brake power compared to a two-stroke MI mode. However, there is a paucity of literature on the development of high-pressure DI technology [33].

In the present paper the performance of the modified engine with high-pressure DI is compared with the conventional two-stroke engine with a carburettor. Simple and cost-effective modifications are incorporated and the engine is tested to verify the improvement in the operation and performance of the 2S-SI engine. Experiments are performed on a 150-cc two-stroke engine mounted with a single-hole pintle injector while maintaining the uniqueness of the two-stroke engine. The injector is mounted on the cylinder head at 30° angle from the cylinder axis and fuel is injected opposite to the main transfer port supplying air from the crankcase. Fuel is timed at 101° crank angle before top dead centre which is at 5° angle after closure of the transfer ports and just after closure of the exhaust port. Therefore, the fuel injection is timed after the exhaust port is shut by the piston such that it helps in saving fresh fuel from getting short-circuited through the exhaust port. The configuration provides sufficient time for charge preparation in the combustion chamber. The performance in terms of operation is smooth and the output is reasonably good. The scope for more ideal solution is under investigation to further make the operation compatible and more competitive. The novel approach introduced in this research indicates that this compact engine has good potential and may prevent it from obsolescence.

MATERIALS AND METHODS

Experimental Set-up

A conventional carburetted 2S-SI engine was modified and used for the experimental investigations. Detailed specifications of the engine are provided in Table 1. The conventional 2S-SI engine cylinder head was adapted to accommodate a single-hole outwardly open pintle injector

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to facilitate high-pressure fuel injection. An exhaust gas analyser (CDS-250, AVL DITEST, Anstalt für Verbrennungskraftmaschinen List, Austria) was used to measure unburned HC and CO emissions. Efforts were made to reduce errors in experimentation, yet some errors were likely to creep in depending upon the accuracy of the instruments used for various measurements. Uncertainty analysis was done by using the method proposed by Moffat [34]. These uncertainties are presented in Table 2.

Table 1. Engine specifications

Engine make, model	Bajaj, chetak
No. of cylinder	01
Bore	57 mm
Stroke	57 mm
Compression ratio	7.4:1
Swept volume	150-cc
Type of cooling	Air
No. of exhaust port	01
No. of transfer port	03

Table 2. Uncertainty values

Parameter	**Uncertainty (%)**
Speed	±0.6
Load	±2
Brake mean effective pressure	±2
Brake thermal efficiency	±5.5
Brake specific fuel consumption	±5.5
HC emission	±1.6
CO emission	±0.45

Engine Modification

In the combustion chamber pressurised gasoline fuel was injected in the cylinder directly. Fins were cut on the cylinder head on the opposite side of the spark plug location to inject gasoline fuel in DI mode. A hole was drilled in the cylinder head and an adapter was fitted for holding the pintle injector. A Bosch-made high-pressure fuel pump was used to supply gasoline fuel to the pintle injector. Figure 1 shows a view of the modified cylinder head with pintle injector and a spark plug. Figures 2 and 3 show an experimental set-up and schematic diagram of the modified test rig of a two-stroke GDI engine respectively.

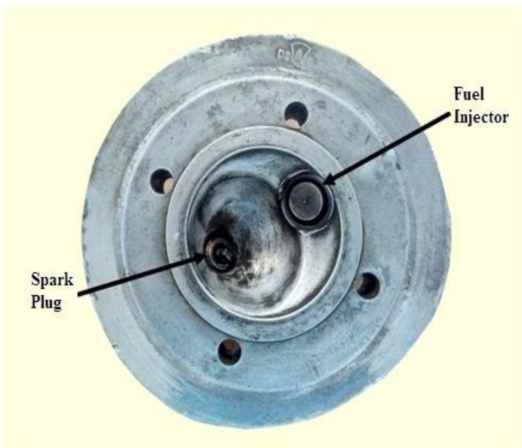

Figure 1. Modified cylinder head with pintle injector and spark plug

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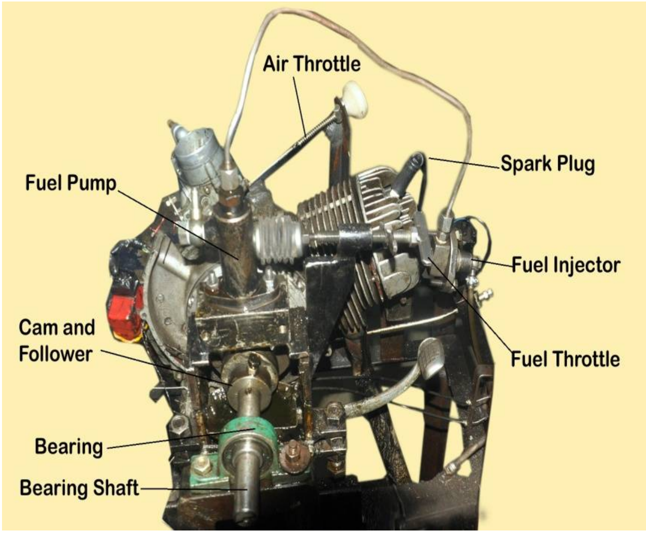

Figure 2. Modified two-stroke GDI test rig

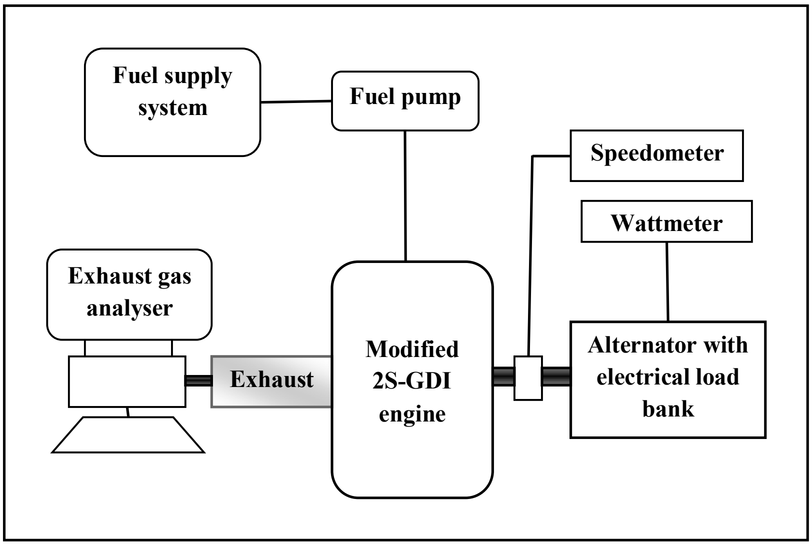

Figure 3. Schematic diagram of modified test rig of two-stroke GDI engine

Experimental Procedure

The experiments were performed, first, in the carburettor mode and then in the modified in-cylinder GDI mode at 80% throttle conditions over a varied range of speed of 1500 to 4000 rpm. The tests on the conventional carburettor mode were conducted as per a normal test routine with variable mixture strength. The modified in-cylinder injection engine was operated at different injection pressures. The optimal fuel injection pressure for the experiments was 80 bar based on the initial test. It was noticed that the engine operation became erratic above 80-bar injection pressure. The performance evaluation was done based on observations of the alternator with an electrical load bank, the fuel measurement system and emissions from the exhaust. All engine tests for direct fuel injection mode were performed for the fuel pump rack position corresponding to the stoichiometric mixture ratio. Finally, the data were collected for a wide range of speed at 80% throttle opening to observe the performance and emission characteristics of the engine under the in-cylinder GDI mode.

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RESULTS AND DISCUSSION

Brake Thermal Efficiency

Figure 4 shows the effect of engine load on BTE using the engine in GDI mode and carburettor fuel system (CFS) mode. BTE is higher in GDI mode than in CFS mode as seen in Figure 4. The fuel injection is timed at 101° crank angle before top dead centre after the closing of exhaust port, which improves the fuel trapping efficiency in the combustion chamber. This also eliminates short-circuiting of the fresh charge. Increased fuel trapping efficiency results in improved BTE at all engine loads in GDI mode. Improvement in BTE could be due to a remarkable enhancement in the scavenging efficiency. In GDI mode the BTE is 16.2% and 23% corresponding to the brake mean effective pressure (BMEP) of 1.65 and 5.78 bar respectively. The maximum improvement in BTE is 16.3% corresponding to BMEP of 4.12 bar.

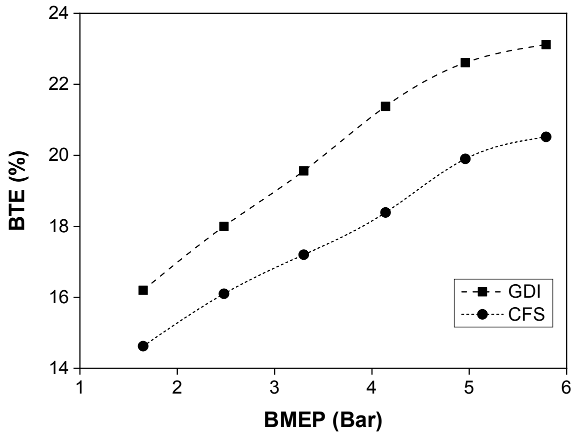

Figure 4. Effect of engine load on BTE

Brake Mean Effective Pressure (BMEP)

Figure 5 shows the effect of engine speed on BMEP when the engine is operated in GDI mode and CFS mode. BMEP is higher in GDI mode as compared to that in CFS mode because of better spray atomisation and vaporisation of charge at high injection pressure, which leads to improved combustion. The combustion performance can influence BMEP. In GDI mode the pressurised crankcase air removes residual gases from the combustion chamber, resulting in increased scavenging efficiency. Higher fuel trapping is obtained in the GDI mode due to reduction in charge short-circuiting. In GDI mode the BMEP varies in the range of 3.96 bar at 1500 rpm to 4.43 bar at 4000 rpm with a maximum BMEP of approximately 5 bar at 3000 rpm. The maximum enhancement of BMEP is around 20.1% in GDI mode as compared to that in CFS mode.

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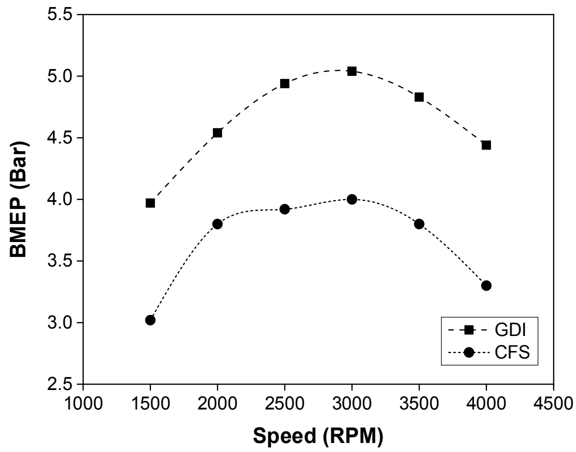

Figure 5. Effect of engine speed on BMEP

Brake Specific Fuel Consumption

Figure 6 shows the effect of engine speed on brake specific fuel consumption (BSFC) when the engine is operated in GDI mode and CFS mode. In GDI mode fresh air pressurised in the crankcase is solely used for the scavenging process and fuel is injected at 101° crank angle before top dead centre which is 5° angle after the closure of transfer ports and just after the closure of exhaust port. Consequently, this reduces fuel short-circuiting in the engine. However, only a small amount of air is short-circuited during the scavenging. Figure 6 clearly shows a drastic reduction in BSFC in GDI mode as compared to that in CFS mode. In GDI mode BSFC is 293 gm/kW-hr at 4000 rpm. The maximum decrement in BSFC is 17% in GDI mode as compared to that in CFS mode.

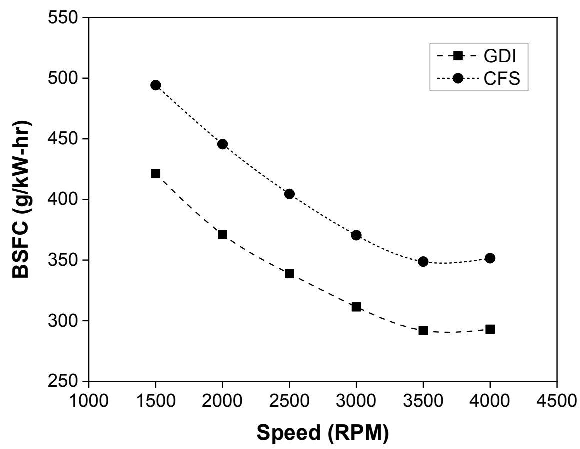

Figure 6. Effect of engine speed on BSFC

Hydrocarbon Emission

Figure 7 shows the effect of engine load on HC emission using the engine in both GDI mode and CFS mode. The fuel loss due to charge short-circuiting and partial combustion leads to unburned HC. This is the major problem encountered when using conventional carburetted engines. Figure 7 clearly shows that HC emission significantly decreases in GDI mode. Fuel short-circuiting is also reduced during the scavenging process. The mixture formation process is initiated by the fuel injection after the closing of exhaust port. In GDI mode the reduction in HC emission lies between 73% at 1.65 bar and 88.5% at 4.14 bar as compared to that in CFS mode. This occurrence may be

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mainly attributed to the reduced fresh fuel short-circuiting through the exhaust port while pressurised air is transferred via the transfer ports from the crankcase.

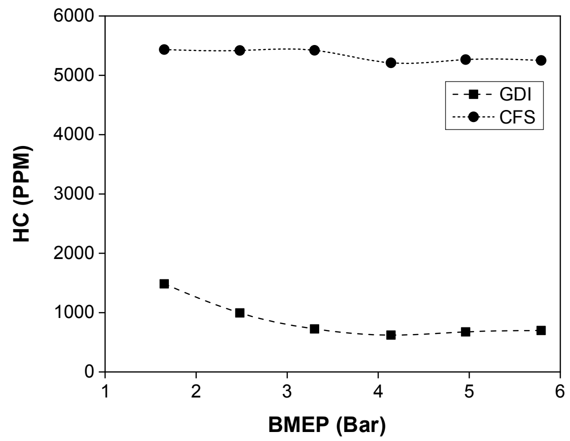

Figure 7. Effect of engine load on HC emission

Carbon Monoxide Emission

Figure 8 shows the effect of engine load on CO emission using the engine in both GDI mode and CFS mode. CO is an indicator of incomplete combustion and mainly depends on the quality of charge, fuel trapping efficiency and combustion. Figure 8 clearly shows that CO emission is higher in CFS mode than in GDI mode. Improved spray atomisation and vaporisation provides good mixture homogeneity, which in turn improves combustion and reduces CO emission. In GDI mode the maximum reduction of CO is 27.5% and the minimum reduction is approximately18.8% as compared to that in CFS mode.

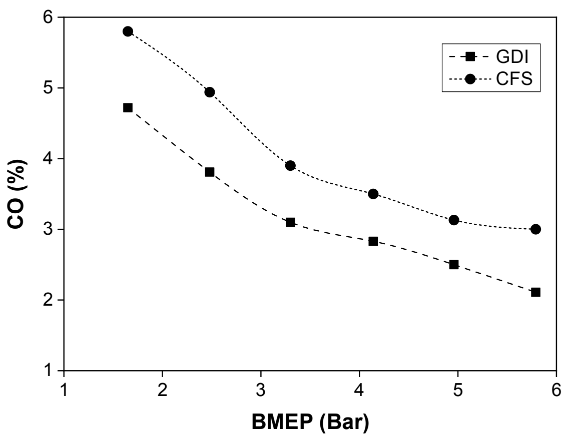

Figure 8. Effect of engine load on CO emission

CONCLUSIONS

The modified two-stroke engine with high-pressure GDI mode shows significant improvement in performance and emission characteristics over that of the CFS mode. The fuel injection timing is crucial for reducing fuel short-circuiting in GDI mode, causing significant reduction in HC emission and boost to fuel economy. Both the spray formation and quality of charge improve as a result of improvement in scavenging with the GDI mode. Thus, the combustion is also enhanced resulting in higher BMEP and BTE with greater reduction in BSFC and HC and CO emissions compared to the carburetted mode.

These findings clearly establish the importance of modifications carried out on the conventional carburetted engine, which permits direct fuel injection to the combustion chamber and is beneficial for the optimisation of the two-stroke SI engine.

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