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Performance Enhancement of Two-Stroke Spark-Ignition Engine Using High-Pressure Fuel Injection

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Keywords：Direct-Injection, Scavenging, Short Circuiting, Two-Stroke.

ABSTRACT

Two-stroke spark-ignited engines mainly suffer from the low fuel economy and high emissions. However, these compact engines do have adequate potential and provide sufficient leverage from salvaging them from going into extinction if the present methodology is adopted. This research investigates the effect of direct in-cylinder fuel injection on a two-stroke spark-ignited engine with moderate high-pressure direct injection. A conventional carburetted engine was modified for injecting gasoline from a cylinder head. The performance of the in-cylinder injection engine was superior compared to the carburetted engine. There was a significant improvement of about 20.1% in the Brake Mean Effective Pressure. The Brake Thermal Efficiency was 16.3% higher than the carburetted engine. The enhancement was due to the fine atomization and vaporization of the charge and reduced loss due to mixture short-circuiting leading to improved combustion efficiency. There was a 17% reduction in the Brake Specific Fuel Consumption as a result of the elimination of short-circuiting losses. A comparative analysis between the carburetted engine and the direct injection system showed a 27.5% reduction in the Carbon Monoxide and an 88.5% reduction in the unburned Hydrocarbon emissions due to the absence of the short-circuiting of charge.

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INTRODUCTION

Global warming and fuel shortages are two of the major problems today. Euro-VI & Bharat-VI regulations have new exacting standards regarding exhaust emission, including HC, CO, CO_2, and NO_x, that demand advanced IC engines with highly sophisticated combustion processes. Automobile vehicles contribute significantly to air pollution in urban areas, and in fact, are the prime contributors to environmental degradation. Worldwide researchers and automobile makers have been challenged by stringent emission standards to develop eco-friendly and better fuel-efficient engines.

The two-stroke spark-ignited engines have many advantages as compared to the four-stroke spark-ignited engines, due to double firing frequency, the absence of valve mechanism and associate driving parts. However, the carburettor two-stroke spark-ignited engines emit unacceptable high levels of HC and CO emissions (Pradeep et al. 2014, Dalla et al. 2015, and López et al. 2017). The low fuel economy and high emissions have led to abandoning the two-stroke spark-ignited engines in the commercial sector. The two-stroke spark-ignited engines have a major limitation of charge short-circuiting during the simultaneous opening of the transfer and the exhaust port. Charge short-circuiting is the main reason for the high level of emissions and low fuel economy. Also, the presence of gas residues in the combustion chamber is the result of incomplete scavenging (Bakshi et al., 2007, Bertsch et al., 2013, Pradeep et al., 2015, and Borghi et al., 2017).

The short-circuiting during the scavenging process in two-stroke spark-ignited engines is very complicated and requires a thorough understanding. To better understand these phenomena, the computational fluid dynamics (CFD) techniques have been applied to predict the flow behaviour inside the combustion chamber. Balduzzi et al. (2015) studied the influence of the injection pressure on in-cylinder flow and combustion by using 3D-CFD analysis. They concluded that the increase in injection pressure resulted in improvement in mixture formation and engine performance. Mallikarjuna et al. (2011) reported that the trapping efficiency is directly affected by the port configurations in the crankcase scavenged two-stroke spark-ignited engine. Mattarelli et al. (2013) compared a four-stroke diesel engine with a two-stroke gasoline direct injection engine using various operating parameters. They found the two-stroke gasoline direct injection engine was more fuel-efficient as compared to a diesel engine with the same power output. Addepalli et al. (2016) compared experimental results with the CFD results. They reported that the tumble ratio and turbulent kinetic energy increased by about 20% and 25%, respectively, when the engine speed was increased from 1000 rpm to 1500 rpm. As the compression ratio increased from 7 to 8, the tumble ratio and turbulent kinetic energy decreased by about 26% and 13%, respectively. Fukang et al. (2015) investigated the scavenging process in the two-stroke engine using 3D-CFD in the opposed-piston case for three different piston shapes. They concluded that a flat piston with a uniform scavenging chamber could provide a higher swirl ratio and lower tumble ratio as opposed to a pit piston non-uniform scavenging chamber which can provide higher tumble ratio, higher turbulent kinetic energy and turbulent intensity. The piston providing lower tumble ratio and higher swirl ratio can result into improvement in the scavenging process.

Most of the reported researches have so far focused on low-pressure direct injection (LPDI) technique (Padala et al. 2008, Loganathan et al. 2009, Krimplstätter et al. 2014, Romani et al. 2015, and Darzi et al. 2018), which is comparatively easy to implement and cost-effective system. Mixture formation time is an important issue in the low-pressure fuel-injected two-stroke spark-ignited engine, which can be resolved by high-pressure direct injection (HPDI) technique. HPDI improves fuel atomization and spray characteristics. The location of injector and injection timing are the most critical things in working of two-stroke SI engine. The LPDI system has less complexity in maintaining fuel pressure, while the HPDI system requires more heavy components to maintain the high fuel rail pressure. Dubey et al. (2016) performed experimental investigation with different injection pressures (100, 130 and 150 bar). They observed that the two-stroke gasoline direct injection is superior as compared to manifold injection. They further found that fuel trapping efficiency for two-stroke gasoline direct injection at 130 bar is nearly 11.3% is higher than that of two-stroke manifold injection mode. Two-stroke gasoline direct injection has 40.4% and 40.6% better fuel economy in terms of brake thermal efficiency (BTE) with reduction of 90% HC and 80% CO emission, at the same brake power as compared to two-stroke manifold injection mode. However, very limited literature is available on the development of HPDI (Gentili et al. 2004, Schmidt et al. 2004, Schmidt & Winkler et al. 2004, Bertsch et al. 2013, and Dube et al. 2016).

The novel methodology presented in this paper demonstrates that this compact engine does have adequate potential and can be salvaged from going into extinction. This paper attempts to investigate the influence of direct high-pressure fuel injection on two-stroke spark-ignited engine performance while maintaining the engine's quality. These objectives have been achieved by modifying an existing two-stroke spark-ignited engine. A single hole pintle type injector has been installed on the cylinder head inclined at an angle of 30° from the cylinder axis. Additional experiments were then done on a modified HPDI system for fuel injection pressure ranging from 50 to 80 bar and 5° after the exhaust port closure; this resulted in prevention of fuel getting short-circuited through the exhaust, leading to fresh fuel savings. Experiments were also performed to determine engine performance and emission characteristics at speeds from 1500 rpm to 4000 rpm.

EXPERIMENTAL SETUP

A 150 cm^3 carburetted two-stroke spark-ignited engine was modified and used for the experimental investigations. This engine was widely used in two/three wheelers utilized for public transportation in India. In the modified test-rig, fins were cut on the cylinder head on the opposite side of the spark plug location to inject gasoline fuel in direct injection mode. A hole was drilled in it, and an adapter was fitted for holding the injector. The conventional engine cylinder head was adapted to accommodate a single hole pintle type injector to facilitate high-pressure fuel injection. A Bosch high-pressure fuel pump was used to supply gasoline fuel to the pintle injector. In the combustion chamber, highly pressurized gasoline fuel was injected into the cylinder directly. Figure 1 shows a view of the modified cylinder head with the fuel injector and the spark plug. A schematic diagram of the modified test-rig of the two-stroke spark-ignited engine is shown in Figure 2. An AVL (CDS-250) exhaust gas analyzer was used to measure unburned HC and CO emissions. Detailed specifications of the engine are provided in Table 1, and properties of gasoline are shown in Table 2. Despite all efforts made to reduce errors in experimentation, the accuracy of the instruments used for various measurements left room for a margin of error. Uncertainty analysis was done using the method proposed by Moffat (1988). The typical values of uncertainties are shown in Table 3.

Table 1. Engine specifications

Type	Air-Cooled, Two-Stroke SI Engine
No. of cylinder	1
Bore	57mm
Stroke	57mm
Compression Ratio	7.4:1
Swept Volume	150 CC
Ignition timing	18° BTDC
Inlet Port Opening (IPO)	114° BTDC
Inlet Port Closing (IPC)	100° ATDC
Transfer Port Opening (TPO)	156°ATDC
Transfer Port Closing (TPC)	112° BTDC
Exhaust Port Opening (EPO)	110°ATDC
Exhaust Port Closing (EPC)	96° BTDC

Table 2. Properties of gasoline fuel

Lower Heating Value	44 MJ/Kg
Typical Stoichiometric A/F Ratio	14.7:1
Motor Octane Number (MON)	80-90
Density	735 Kg/m3
Auto-Ignition Temperature	260 °C

Table 3. Uncertainty values

Parameters	Uncertainties (%)
Speed	±0.6
Load	±2
Brake Power	±2
Brake Specific Fuel Consumption	±5.5
Brake Thermal Efficiency	±5.5
HC Emission	±1.6
CO Emission	±0.45

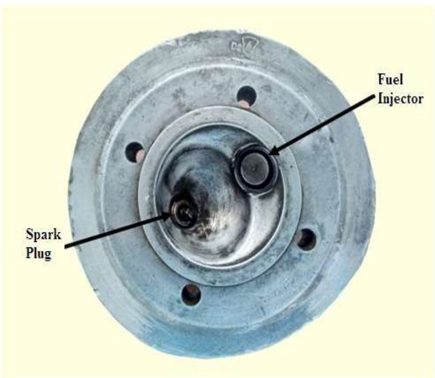

Fig. 1. View of the modified cylinder head with fuel injector and spark plug

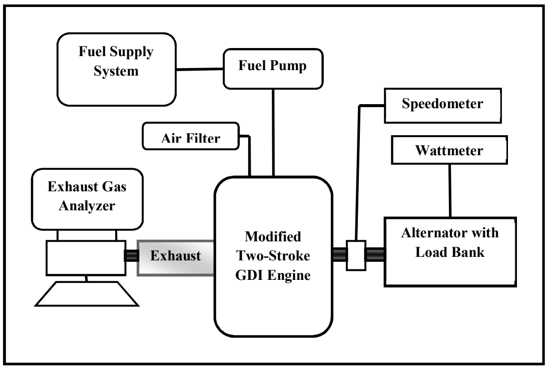

Fig. 2. Schematic diagram of the modified test rig of two-stroke SI Engine

TEST PROCEDURE

The experiments were conducted in the carburettor mode first, and then in the modified in-cylinder injection mode at different throttle conditions from 20%-80% over a varied range of speed of 1500 rpm to 4000 rpm. The tests on the conventional carburettor mode were conducted as per a normal test routine with variable mixture strength. The fuel injection pressure for the tests was fixed at 80 bar based on preliminary tests which were conducted at different injection pressures. It was observed that beyond 80 bar injection pressure, the engine operation became erratic. Thus, an injection pressure of 80 bar was taken as the optimum. The performance evaluation was done based on observations of the electrical alternator, the fuel measurement system and emissions from the exhaust. All the engine tests for the direct fuel injection mode were performed for a fuel pump rack position corresponding to the stoichiometric mixture ratio. Finally, data were collected for a wide range of speeds to observe the performance and emission characteristics of the engine under the gasoline in-cylinder injection mode.

RESULTS AND DISCUSSIONS

The air induction system has been modified by eliminating the carburettor fitted on the crankcase with a direct fuel injection system fitted on the cylinder head. The removal of carburettor provides higher coefficient of discharge in the absence of venturi, which improves the volumetric efficiency of the engine. Higher volumetric efficiency provides extra air induction in the crankcase, which correspondingly increases the quantity of fuel injected in the combustion chamber to get the required mixture ratio during the process. Better fuel atomization due to fuel injection improves mixture homogeneity and fuel vaporization, which affects combustion quality in comparison to the conventional engine.

Effect of Engine Load on BTE

Figure 3 shows the effect of the engine load on BTE using the engine in GDI and CFS modes. It is evident from Figure 3 that the maximum BTE was higher in the GDI mode as compared to the CFS mode. The fuel injection was timed after the closing of exhaust port, which improved the fuel trapping efficiency in the combustion chamber resulting in the elimination of short-circuiting of fresh charge. The higher fuel trapping efficiency resulted in improved BTE at all engine loads. This rise in BTE was achieved due to a significant improvement in the scavenging process. In the GDI mode, the BTE was 16.22% and 23%, corresponding to BMEP of 1.65 bar and 5.78 bar, respectively. The maximum improvement in the BTE was 16.3%, corresponding to a BMEP of 4.12 bar as compared to the CFS mode.

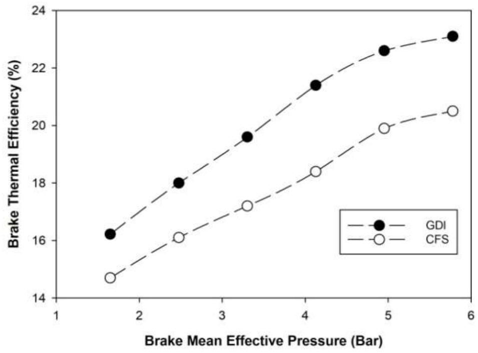

Fig. 3. Effect of engine load on BTE using the engine in GDI and CFS modes

Effect of Speed on BMEP

Figure 4 shows the effect of speed on the BMEP using the engine in GDI and CFS modes. The combustion performance influences BMEP. The improvement in BMEP is attributed to the modified high-pressure fuel injection directly in the combustion chamber after the closure of the exhaust port. This eliminates fuel loss through the exhaust port, which is normal in any conventional CFS engines due to charge short-circuiting. However, air transferred through the transfer port from the crankcase to the main combustion chamber continues to get short-circuited through the exhaust port. This loss of air is partly compensated due to improved volumetric efficiency on account of the larger induction port in the crankcase in the absence of conventional carburettor.

Better spray atomization, mixture homogeneity and fuel vaporization of charge at high injection pressure of the modified engine help in higher gas pressure on the piston crown due to more efficient heat release during combustion in the chamber. Thus, the BMEP in the form of engine mechanical energy transferred from the efficiently burning charge is enhanced as evident from the figure. In the GDI Mode, the BMEP is 3.96 bar to 4.43 bar corresponding to a range of speed of 1500-4000 rpm, respectively, and the maximum enhancement in the BMEP is 20.1% as compared to the CFS mode.

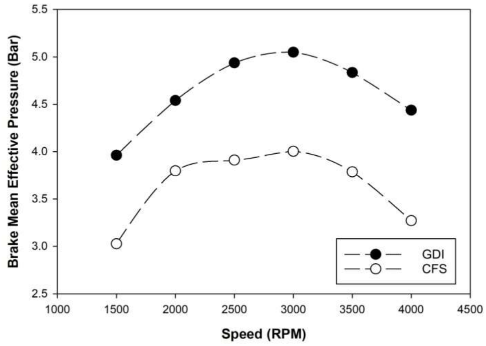

Fig. 4. Effect of speed on BMEP using the engine in GDI and CFS modes

Effect of Speed on BSFC

Figure 5 shows the effect of speed on BSFC using the engine in GDI and CFS modes. In the GDI mode, only fresh air was used for scavenging, and fuel injection was done after the closure of exhaust port, which reduced the short-circuiting phenomenon in the engine. However, only a small amount of air is short-circuited during the scavenging process. Figure 5 clearly shows that there was a significant reduction in BSFC in the GDI mode as compared to the CFS mode. In the GDI mode, the obtained BSFC was 293 gm/kW-hr at a speed of 4000 rpm. The maximum reduction in the BSFC was 17 % as compared to the CFS mode at that speed.

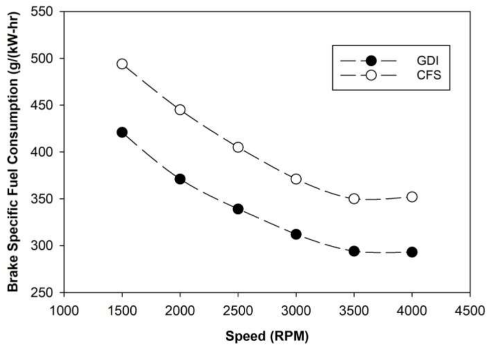

Fig. 5. Effect of Speed on BSFC using the engine in GDI and CFS modes

Effect of Engine Load on HC Emissions

Figure 6 shows the effect of the engine load on HC emissions using the engine in both the GDI and CFS modes. The unburned HC is the product of loss of fuel due to charge short-circuiting and incomplete combustion. These are major problems that occur while using the conventional engine with a carburettor. Figure 6 shows that the unburned HC emissions drastically reduced in the GDI mode. The fuel short-circuiting phenomenon reduced during the scavenging process. The mixture formation process was done by fuel injection after the closure of the exhaust port. In the GDI mode, the maximum reduction in HC emissions was 88.5%, and the minimum was 73% as compared to the CFS mode. The maximum reduction in unburned HC emission was 88.5% at 4.12 bar as compared to the CFS mode. This occurrence is mainly attributed to the reduced fuel short-circuiting through the exhaust port while the charge is transferred through the transfer ports from the crankcase.

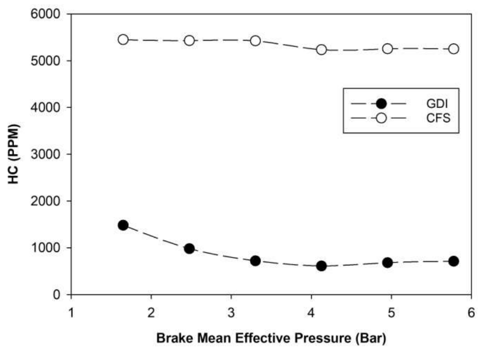

Fig. 6. Effect of engine load on HC emissions using the engine in GDI and CFS modes

Effect of Engine Load on CO Emissions

Figure 7 shows the effect of engine load on CO emissions using the engine in both the GDI and CFS modes. The CO emissions were higher in the CFS mode as compared to the GDI mode. CO is an indicator of incomplete combustion. CO mainly depends upon the quality of charge, fuel trapping efficiency and charge combustion. Improved spray atomization and vaporization provided good mixture homogeneity, which was responsible for improved combustion and reduction in CO emissions. In the GDI mode, the maximum reduction of CO was 27.5%, and the minimum was 18.8% as compared to the CFS mode.

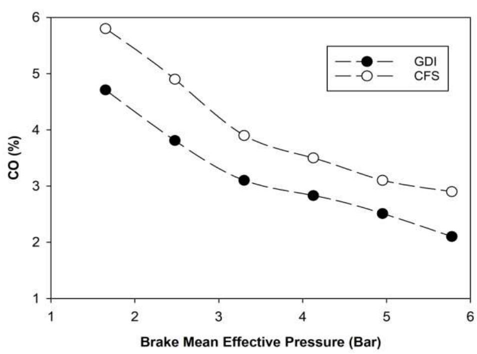

Fig. 7. Effect of engine load on CO emissions using the engine in GDI and CFS modes

CONCLUSIONS

Experiments carried out above show that high-pressure direct injection in the modified two-stroke SI engine has shown tremendous advantages over the conventional carburetted engine because of the curtailment in the short-circuiting. Two key factors in reducing the short-circuiting phenomena in the two-stroke engine are fuel injection timing and location. The HC emissions also drastically reduced, and the fuel economy of the engine improved significantly. The amelioration in the scavenging phenomena showed better spray formation and quality charge. As a result, the combustion phenomenon improved, leading to higher power-output and brake thermal efficiency. Compared to the carburetted engine, the in-cylinder direct injection system showed a significant upgrade BTE and BMEP and greater reductions in HC, CO emissions and BSFC. The engine performance of the modified high-pressure GDI resulted in notable improvements as listed below:

1. Maximum BTE with the GDI mode was 16.3% better than the CFS mode.
2. Improvement in BMEP with GDI mode was 20.1% more than CFS mode.
3. Minimum BSFC in GDI mode was 17% lower than CFS mode.
4. HC emission reduced by 88.5% in GDI mode as compared to CFS mode.
5. CO emission was 27.5% lower with GDI mode as compared to CFS mode.

The above outcomes convulsively establish the superiority of the GDI mode over the CFS mode and highlight the importance of redesigning the conventional carburetted engine with fuel injection placed directly inside the combustion chamber.

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NOMENCLATURE:

ATDC After Top Dead Center

BP Brake Power

BSFC Brake Specific Fuel Consumption

BTDC Before Top Dead Center

BTE Brake Thermal Efficiency

CFS Conventional Fuel System (Carburetted Mode)

CO Carbon Monoxide

CO_2 Carbon Dioxide

GDI Gasoline Direct Injection

HC Hydrocarbon

IC Internal Combustion

SIE Spark Ignition Engine

^{o}Degree

高壓噴油增强二衝程火花點火發動機性能之研究

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摘 要

二衝程火花點火發動機主要存在燃油燃燒率低、排量高的問題。但是，如採用現行方法，這些小型發動機則可具備足够的電壓並提供充足的能量使之免於熄火。本文研究了缸內直接噴油對中高壓直噴式二衝程火花點火發動機的影響。對傳統的化油器發動機進行了改進，使之可從氣缸蓋噴油。與化油器發動機相比，缸內噴油發動機之性能更優。其平均有效制動壓力顯著提高了約20.1%。其發動機有效熱效率比化油器發動機高了16.3%。因强化了燃油霧化和電荷氣化程度並減少了因混合短路而造成的損耗，從而提高了燃燒效率。由於消除了短路損耗，從而使得燃油消耗率降低了17%。化油器發動機和直噴式系統之間的對比分析表明，因不存在電荷短路，從而使得一氧化碳排放量減少了27.5%，未未燃碳氫化合物排放量減少了88.5%。