INVESTIGATION OF GAS EXCHANGE PARAMETERS OF SMALL TWO STROKE ENGINE CORRELATED WITH HYDROGEN FUEL SUPPLEMENT

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Due to the lack of complete dedicated suction and exhaust strokes and obviate the need of valvetrain assembly in two stroke engines, the bottom surface of the piston and the crankcase is used as a scavenging pump leads to that about one fifth of the fresh charge is short-circuited to the exhaust results in very high hydrocarbon emissions and poor fuel consumption. It is suggested that a combination of gasoline and hydrogen could be a superior solution combining the advantages of both engine itself and hydrogen doping. An accurate amount of hydrogen that has high flame propagation speed blended with gasoline could provide faster combustion of fuel mixture over the few crank angle degrees in the short gas exchange process reducing short-circuited charge with little modifications to the engine system. In this paper hydrogen-gasoline blend as an alternative fuel in a 63.3 cc, air cooled single cylinder, crankcase scavenged two stroke engine is tested. The unseen decommissioning of small two stroke engine in near future, the attractiveness of studying alternative fuel specially hydrogen and absence of investigation of the effect of hydrogen fuel supplement on gas exchange parameters molded this paper.

Keywords: Two-stroke engine, Hydrogen, Engine Performance.

Received 21. 05. 2024, Accepted 11. 06. 2024

1. Introduction

There are millions of two-stroke engines worldwide, powering motorcycles, mopeds, three-wheelers and drones. Small two stroke engine could be characterized as carburetted, simple ported, single cylinder and air-cooled, there are millions of them around the world in many applications, and it is not expected in near future that they will be abandoned, so it is necessary to search for solutions to mitigate their shortcomings at low cost. The consideration of efficiency and pollutants norms in the ICE manufacture is a question that makes quite logical in the field of decreasing the consumption of hydrocarbons. For that four-stroke engines have been prevalent and two stroke ones were omitted despite of the simplicity, higher power to weight ratio and lower manufacturing costs. On the other hand, limited crude oil reserves and increasing concern about pollution norms has pushed to research in alternative fuel as CNG, LPG, HCNG, H_2 etc. The main characteristic of two-stroke engines is the doubled number of power strokes per cycle compared to 4-strokes engines, which increases the power output per engine swept volume. However, the charging losses are unavoidable due to the scavenging of exhaust gases from the engine cylinder is executed by fresh charge [1]. A waste lubrication system is used in simple ported two-stroke engines, the oil is mixed with the fuel which either short-circuited or trapped in the cylinder deepening the deficiencies of two stroke engine.

The following is a summary of the studies made in two-stroke engines. A combination of gasoline and hydrogen could be a significant fuel, combining the advantages of both fuels. By accurately blending the high flame propagation speed of hydrogen with gasoline, fast combustion of the fuel mixture can be achieved. An investigation was conducted into the performance and emissions of a single-cylinder, two-stroke carburetted engine. The engine was operated in a dual fuel mode, with gasoline used as the pilot fuel and hydrogen used as the inducted fuel. The results of the investigation showed an improvement in the brake thermal efficiency of the engine. Additionally, the emission levels of CO and HC were noticeably reduced. The best performance was achieved at constant volume flow rate of 0.4 L/min of hydrogen with gasoline and constant speed and different loads [2].

The design of a simple ported direct-injected pure hydrogen engine, based on a 9.9 HP carburetted crankcase scavenged gasoline engine, aimed to achieve higher efficiency and lower emissions. The carburetor fuel system was replaced with a commercially available hydrogen injector to eliminate charge short-circuiting and a new lubrication oil injection pump was added. This resulted in the best point of gross indicated thermal efficiency increase from 39 % of baseline hardware to 42.4 % for the new prototype. The NOx emissions were also reduced by 18 %, with a 3.3 % increase in fuel consumption. Late fuel injection was found to be the main factor in improving thermal efficiency, but on the expense of increased NOx emissions [3].

A conventional two-stroke gasoline engine for motorcycles was built to run on a mixture of gasoline and hydrogen produced from hydrochloric acid and aluminium. The study demonstrated that hydrogen could enhance brake thermal efficiency and decrease HC and CO emissions. The average brake thermal efficiency was 23 % higher compared to a gasoline-only engine, . The total hydrocarbon production for different loading conditions of the gasoline/hydrogen blend was 34 % lower than when using pure gasoline [4].

A gasoline direct injection retrofit kit for small two-stroke engines was developed based on Orbital direct injection system. New fuel, air injectors, cylinder head, piston-style air compressor to supply pressurized air to the new system and a throttle body to complete the kit replacing the carburetor, fuel and oil pumps and an engine control unit were installed. Compared to the carbureted system, the retrofitted system displayed an 88 % reduction in unburned hydrocarbon emissions and a 72 % reduction in carbon monoxide emissions. Due to the near elimination of short-circuiting losses as well as more complete combustion, the retrofit system also showed a 32 % increase in fuel economy while the direct fuel injection technique is not convenient on a cost and weight basis [5].

A two stroke CNG engine performance was examined when it was equipped with scavenging ports for intermittent low-pressure fuel injection. The test engine was a two-cylinder, 398 cm^3, two-stroke cycle spark ignition engine. Gaseous fuel injectors were attached to the engine block, and CNG was injected into the scavenging passage through a fuel injection pipe. The fuel injection pressure was set at 0.255 MPa, and the fuel was injected intermittently during the scavenging process. The BSFC was reduced by 25 %, and the relative air to fuel ratio (λ) was extended from 1.2 to 1.46. The peak of the NOx emission shifts to the leaner side and the maximum achieved reduction of HC emission was about 47 % [6].

An investigation was performed using petrol/ethanol mixtures (5 % and 10 % v/v) as an alternative fuel for an unmodified 100 cc two-stroke petrol engine. The goal was to determine if a petrol/ethanol mixture is a suitable fuel for a conventional two-stroke engine without any modifications. Performance characteristics have shown that brake-specific fuel consumption remains constant [7]. The gas exchange parameters of a two-stroke gasoline engine, including delivery and scavenging ratios, trapping, charging and scavenging efficiencies were tested as its fuel was mixed with ethanol supplement. Additionally, the amounts of emitted pollutants from this engine were measured at various engine speeds and loads. In the experiments, ethanol was combined with gasoline in different percentages. The results showed that in most cases when alcoholic fuel was used, scavenging efficiency and delivery ratio increased due to the rapid evaporation of ethanol. However, fuel converging efficiency and brake-specific fuel consumption decreased. The most outstanding result of using an ethanol additive is a significant reduction in CO emissions, at 25 % full throttle and 4500 rpm, CO decreased approximately by 71 % when 15 % ethanol added to gasoline (on mass basis). For other engine speeds and throttle opening positions, CO decreased averagely by 35 % [8].

Most of the previous studies on operating small two-stroke engines with alternative fuel have studied the effect of alternative fuel on engine performance and emissions. However, they have overlooked investigating to what extent alternative fuel affects short-circuiting phenomenon. Vast investigations directly studied the positive effect of running hydrogen fuel supplement on performance and emissions of two stroke engines, neglecting firstly to find the effect on gas exchange parameters, delivery ratio, trapping, charging, scavenging efficiencies and scavenging ratio then correlate them to engine performance characteristics, which is the goal of this work.

2. Gas exchange parameters

Following the definitions used in this paper in the field of gas exchange process characteristic of the two-stroke engine:

2.1. Delivery ratio (DR)

It compares the real delivered fresh mixture (M_d) to the engine, which can be easily measured by the air mass flowmeter at engine intake to the mass of gas that would flow to the crankcase of volume equal to the cylinder capacity in normal conditions (M_o) [9]. Usually, the increase in crankcase delivery ratio causes an increase in the engine torque.

$$DR = M_d / M_o \qquad (1)$$

2.2. Trapping efficiency (η_{tr})

The fresh mixture delivered is split into two parts the short-circuited air, which leaves through the exhaust port, and the retained fresh mixture, which is trapped in the cylinder and participate in the next combustion. Trapping Efficiency indicates the ability of the cylinder to retain the fresh charge. It is defined as the ratio of the mass of charge retained in the cylinder to the mass of fresh mixture delivered to the engine, it can be measured using the Watson method by measuring the oxygen concentration in the exhaust gas under the following assumptions:

1) No oxygen exists in the exhaust gas based on a rich mixture running.

2) The mixture blown out the exhaust port mixes evenly with the exhaust gas in the exhaust system, consider oxygen concentration in ambient air $[O_2]_{amb}$ is 21 %, measurement of O_2 in the exhaust gases $[O2]_{exh}$, directly gives the engine's trapping efficiency, [10].

$$\eta_{tr} = M_{tr} / M_d = 1 - [O2]_{exh} / [O2]_{amb} \qquad (2)$$

2.3. Charging efficiency (η_{ch})

It indicates the efficiency of filling the cylinder with fresh air by comparing the mass fresh charge retained to reference mass and calculated from trapping efficiency and delivery ratio as follows:

$$\eta_{ch} = M_{tr} / M_o = DR \times \eta_{tr} \qquad (3)$$

2.4. Scavenging efficiency (η_{sc})

It indicates to what extent the burnt residuals have been replaced with fresh charge. It compares the fresh charge retained to the perfectly calculated mass of cylinder charge M_{cyl} at the ideal scavenging process (which is occupied by retained fresh charge and the residual products from the past cycle).

M_{cyl} is calculated based on cylinder density ρ_{cyl} deduced from gas laws assuming ideal gas issue at volume V_{cyl} at inlet temperature T_i and exhaust pressure P_{exh} just after exhaust port close, R universal gas constant [11].

$$\eta_{sc} = M_{tr} / M_{cyl} = \frac{M_{tr}}{M_{tr}+M_{res}} = \frac{M_{tr}}{\frac{r}{r-1}\times V_D \times \frac{P_{exh}}{R\times T_i}} \qquad (4)$$

2.5. Scavenging ratio (R_{sc})

It is defined as the mass of fresh charge delivered to the calculated mass of cylinder charge in the ideal scavenging process.

$$Rsc = \frac{M_d}{M_{cyl}} \qquad (5)$$

So scavenging efficiency can be easily calculated as follows [11].

$$\eta_{sc} = R_{sc} \times \eta_{tr} \qquad (6)$$

3. Methodology

Experiments were conducted on a test rig using a gasoline crankcase scavenged air-cooled single-cylinder two-stroke engine. The engine was fueled with a mixture of gasoline and hydrogen.

3.1. Engine and measuring equipment

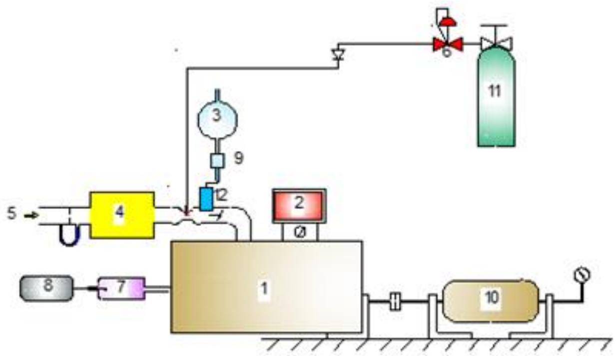
The engine used is a wide commercially small single-cylinder, air-cooled, two-stroke gasoline engine. It has been modified with a hydrogen/air mixture preparation at the intake duct to ensure premixed fuel induction. Table 1 contains information on the engine specs.

Table 1 Engine specifications.

Engine type	2 Stroke
Model	1e48f
Cylinder Bore	48 mm
Stroke	35 mm
Connecting rod length	100 mm
Compression Ratio	6
Geometric compression ratio of the crankcase	1.4
Beginning of the inlet port open	60° BTDC
Induction	Carburetor/ reed valve
Rated Output	650 W @ 3000 rpm
Engine Displacement	63.3 cm^3
Lubrication	Fuel oil premix 50:1

During testing, the engine was run at a constant speed of 3000 rpm with a variable throttle opening ranging from 25 % to 100 % of the engine's full throttle opening. The following is a summary of the measuring equipment used:

- The temperature of the engine cylinder head is accurately monitored by a resistance thermometer-type PT100 temperature sensor which is connected to an Omron 5EN digital screen. This is to anticipate potential engine overheating due to the high combustion temperature of the hydrogen fuel additive.
- The induction air is measured using pressure differential device, an orifice plate inserted in a cylindrical conduit with a 0.5D/D pressure tapping configuration installed on the inlet duct of a 6 L surge tank at the engine air intake, air pressure difference is displayed on a U-tube manometer, air mass flow rate is calculated at discharge coefficient of 0.6 and room temperature.
- Hydrogen and gasoline consumptions are measured using a calibrated high-precision Mettler Toledo SR32001 balance.
- Graduated cylinder is used as a gasoline fuel tank, for more perception of engine fuel consumption.
- A DC dynamometer equipped with RPM gauge is used for testing the engine, with the load applied by the dynamometer being controlled by a resistive load bank.
- The hydrogen tank is connected to a pressure regulator to ensure a constant pressure hydrogen fuel line to the engine, regardless of its pressure.
- The AGS 888 is a calibrated nondispersive infrared (*NDIR*) exhaust gas analyzer that is used to measure the concentration of oxygen, *CO*, CO_2, *HCs*, and *NOx* in exhaust gases. Figure 1 shows a schematic drawing of the test rig and measuring equipment used in this work.



1 Engine	7 Tailpipe
2 Temperature display	8 AGS888 Gas Analyzer
3 Fuel tank	9 Fuel measuring device
4 Induced air measuring	10 Dynamometer
5 Intake air	11 Hydrogen tank
6 Hydrogen pressure regulator	12 Carburetor

Fig. 1 A schematic drawing of the Test Rig and the measuring equipment.

3.2. Hydrogen concentration setting

To achieve this, the throttle adjustment screw in the pressure regulator was turned either clockwise or counterclockwise for several trials to maintain the concentration within a narrow range. After many trials to ensure engine stable running on gasoline hydrogen blend, setting was adjusted to provide the engine hydrogen flow of about 1.12 L/min, or approximately 0.0017 g/s. The variation of gasoline flow rates and hydrogen mix percentage with throttle opening is shown in Table 2.

Table 2 Variation of gasoline quantity and hydrogen enrichment ratio with throttle opening.

Throttle opening	25 %	50 %	75 %	100 %
Gasoline (g/sec)	0.09	0.133	0.155	0.163
Hydrogen (% m/m)	1.89	1.28	1.10	1.04

3.3. Fuel delivery method

Figure 2 shows a simple suitable technique for gas carburetion such as approved in [4], which is used for the delivery of hydrogen fuel. The shining merit of the system is no need for an expensive complicated *EFI* system which conflicts with the requirement of a low cost of a small two-stroke engine. Another merit is the easy conversion of the original two-stroke gasoline engine to operate with a gasoline/hydrogen blend. The drawback of this technique is the large volume occupied by the hydrogen in the crankcase before pushed to combustion chamber via transfer ports, which affects the delivery ratio and engine power. The objective was to optimize the hydrogen concentration to achieve engine stable operation at minimum negative effects.

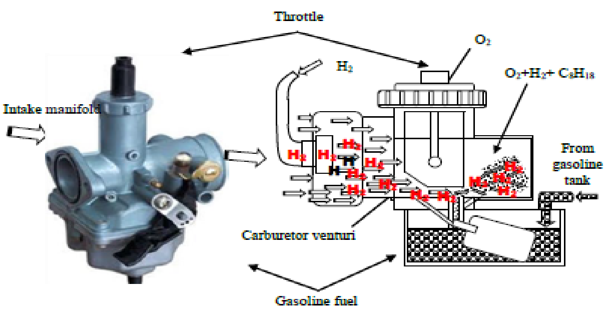


Fig. 2 A schematic drawing of the hydrogen delivery system.

3.4. Test procedure

Engine is started with gasoline and warmed until reaching stable operation then throttle lever is gradually raised to required opening against applied dynamometer load to keep engine speed at 3000 rpm, power, speed, induced air, fuel consumption, emissions (oxygen concentration) are recorded and repeated with hydrogen enriched gasoline running.

4. Results and discussions

In the present work, parameters of the gas exchange process of a 63.3 cc small crankcase scavenged two-stroke engine are correlated to hydrogen fuel additive to attain a deeper understanding of its impact on engine performance characteristics. experiments are performed at engine rated speed 3000 rpm in two different phases of operations, gasoline, and hydrogen-enriched gasoline.

4.1. Effect of Hydrogen Addition on the fuel mixture strength

Hydrogen has a wide flammability range in air varying from 4 to 75 % mole percentage, resulting in it being combusted in ICE at a lower limit of equivalence ratio 0.1 compared to 0.7 of gasoline. The stoichiometric AFR for the combustion of hydrogen with air is about 34:1 on a mass basis compared while it is 14.7:1 for gasoline, pursuing leaning fuel mixture and targeting higher thermal efficiency as the lower limit of hydrogen combustion AFR reaches about 180:1 while about 20:1 for gasoline. This is the most significant advantage of using hydrogen to run engines with leaner mixtures. Table 3 shows the properties of hydrogen and gasoline.

Table 3 Properties of hydrogen and gasoline [12].

Parameter	Hydrogen	Gasoline
Equivalence ratio ignition lower limit in *NTP* air	0.1	0.70
Mass lower heating value(kJ/kg)	119930	46000
Density of gas at *NTP* (kg/m^3)	0.089	4.4*
The stoichiometric air-to-fuel ratio on a mass basis	34	14.7
Volumetric fraction of fuel in air, φ = 1 at *NTP*	0.290	0.018
Laminar burning speed (cm/sec)	265-325	37-43

* Gasoline vapor density.

Relative air to fuel ratio (λ)

It is generally defined as actual air/fuel ratio to stoichiometric air/fuel ratio. For fuel-lean (excess air) mixtures, $\lambda > 1$, and for fuel-rich (air deficient) mixtures, $\lambda < 1$. Figure 3 illustrates (λ) versus throttle opening. The original manufacturer setting in the test engine was rich to ensure stable operation, especially at low loads, and to guarantee maximum laminar burning velocity occurs with slightly richer than stoichiometric mixtures, and where the temperature of the burned gases produced by combustion is a maximum. Figure 3 shows that the addition of hydrogen at a rate of 1.12 L/min permits easy leaning of air-fuel mixture taking advantage of hydrogen's high flame speed with sensible stable engine operation, more mixture leaning not permitted respecting Watson method constraints.

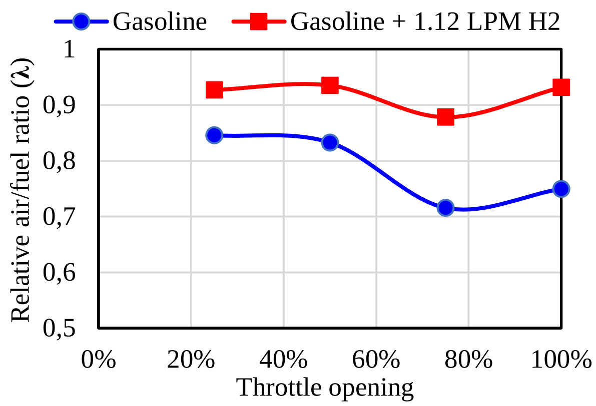


Fig. 3 Relative air/fuel ratio *(λ)* vs. throttle opening at 3000 rpm and *MBT.*

4.2. Effect of hydrogen addition on the engine gas exchange parameters

Calculation of occupied volume percentage by hydrogen gas for a stoichiometric mixture (% *Vcyl*)

A mole of oxygen is needed for every two moles of hydrogen.

$$2H_2 + 1O_2 \rightarrow 2H_2O$$

Including nitrogen in the air 1 mole of oxygen (O_2) is with 3.762 moles of nitrogen (N_2)

Number of moles of air = Moles of O_2 + moles of N_2

= 1 + 3.762= 4.762 (mole)air

AFR on volume basis = (moles) air/ (moles) fuel = 4.762 / 2 = 2.4:1

% *Vcyl* = Volume (moles) of H_2 / total volume

= Volume H_2 / (volume air + volume of H_2)

= 2 / (4.762 + 2) = 29 %

About one-third of the combustion chamber in case of stoichiometric combustion would be consumed by hydrogen. The hydrogen feeding rate was kept very low at 1.12 L/min H_2 to reduce the effect on the delivery ratio. As shown in Figure 4, the delivery ratio is perfectly related to the consumed air (Throttle Opening Percentage). The impact of the hydrogen gaseous state slightly negatively affects the gasoline +1.12 L/min H_2 operation.

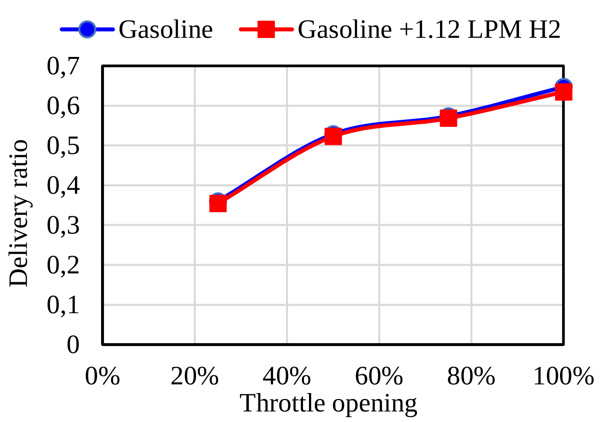


Fig. 4 Delivery ratio vs. throttle opening at 3000 rpm and *MBT.*

Hydrogen physicochemical properties

The ability of hydrogen to disperse into the air easily more than gasoline fuel due to its high diffusivity which permits a more uniform mixture of fuel and air, higher flame speed of hydrogen at stoichiometric ratios is nearly an order of magnitude higher than that of gasoline which bring hydrogen combustion closer to ideal thermodynamic cycle.

Figure 5 illustrates the trapping efficiency obtained under the above-mentioned oxygen measuring conditions in section 2.2 is plotted against load percentage, the increased amount of fresh charge retained directly due to the higher flame propagation of hydrogen that rapidly completed combustion of the fresh mixture before the exhaust valve opens reducing short-circuited charge although a leaner set of excess air ratio, best-achieved improvement of trapping efficiency about 17 % at full load.

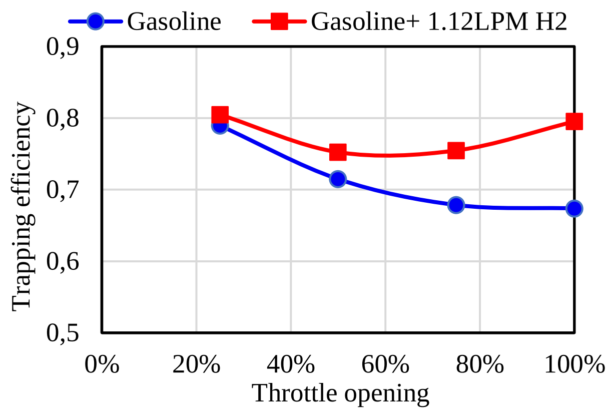


Fig. 5 Trapping efficiency vs. throttle opening percentage at 3000 rpm and MBT

Figure 6 demonstrates the enhancement of Charging Efficiency with hydrogen added, despite of drop of the Delivery ratio the more retained fresh mixture improves the filling of the cylinder with fresh air, best improvement of about 16 %.

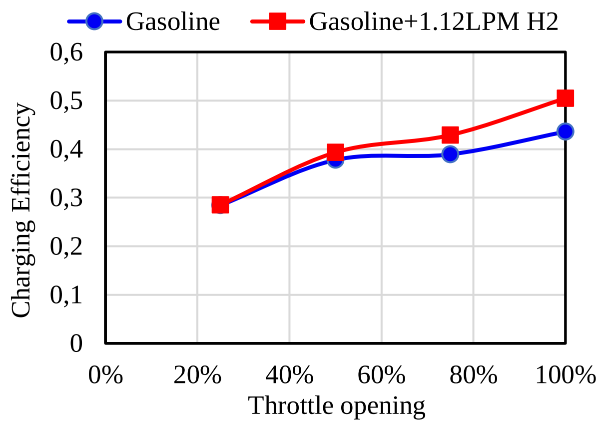


Fig. 6 Charging efficiency vs. throttle opening at 3000 rpm and MBT.

Figure 7 illustrates the slight negative effect of adding hydrogen on the Scavenging ratio, the low density of hydrogen gas 90 grams per cubic meter, causes occupation of the intake system with gaseous fuel reducing induced air.

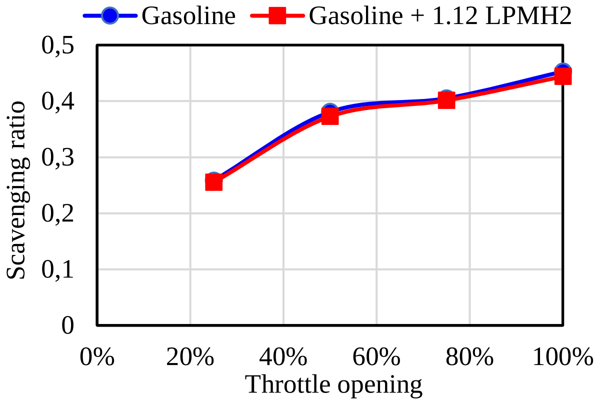


Fig. 7 Scavenging ratio vs. throttle opening at 3000 rpm and *MBT*.

The scavenging efficiency is equal to the scavenging ratio in the ideal scavenging process where η_t is unity, figure 8 shows the scavenging efficiency versus load percentage. A significant enhancement was achieved in replacing residual gases with the fresh mixture by adding hydrogen that has higher diffusivity forming a uniform mixture with air and higher flame speed rapidly consumes unburned hydrocarbons most importantly about 16 % at full load.

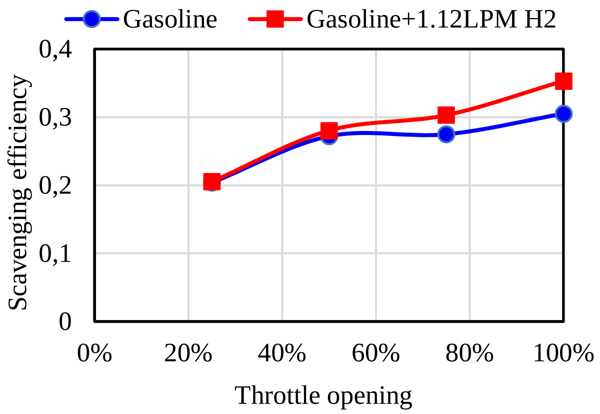


Fig. 8 Scavenging efficiency vs. throttle opening at 3000 rpm and *MBT*.

4.3. Effect of Hydrogen Addition on the engine performance

An ideal Otto cycle engine's theoretical thermal efficiency η_{th} is determined by the engine's compression ratio r and the mixture's specific-heat ratio γ, and can be calculated by the following equation.

$$\eta_{th} = 1 - \frac{1}{r^{\gamma - 1}}$$

Because hydrogen has a greater specific-heat ratio (γ = 1.4) than gasoline (γ = 1.1), it can slightly contribute the improvement of engine thermal efficiency. Also receding of fresh charge short-circuiting as trapping, charging and scavenging efficiencies improved **sec 4.2.** positively affected thermal efficiency. At 100 % throttle opening, due to high heating value of hydrogen relative to gasoline (about three times on mass basis) enrichment of the gasoline by 0.1 gm/min of hydrogen saves about 22 % of gasoline fuel consumption provides more lean mixture with little sacrifice of power while maintaining stable operation. Brake thermal efficiency (*BTE*) is inversely proportional to the specific fuel consumption as illustrated in figures 9 and 10. All above mentioned factors lined up together, lead to enhance the brake thermal efficiency (BTE) and reduce brake-specific fuel consumption (BSFC) over the whole load range The best results are an improvement of thermal efficiency by 18 % and a reduction of BSFC by 20 % respectively at full load. Assigning the relative impact of each factor separately could be a future work.

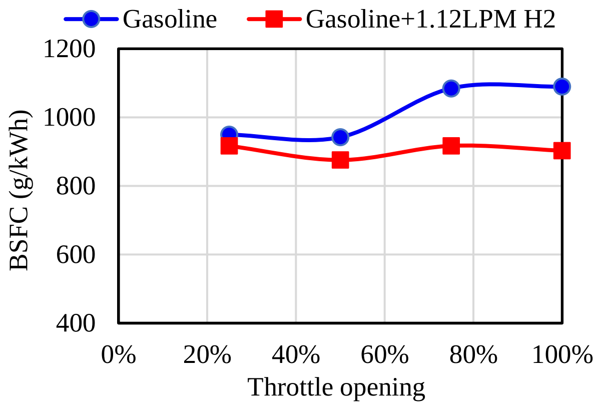


Fig. 9 Engine brake specific fuel consumption vs. throttle opening at 3000 rpm and *MBT*.

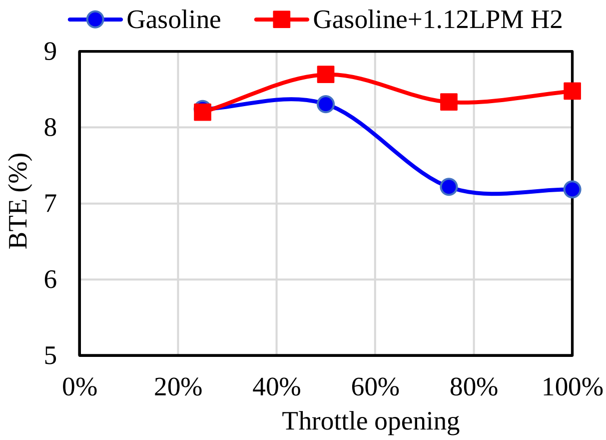


Fig. 10 Engine Brake Thermal Efficiency vs. Throttle Opening Percentage at 3000 rpm and *MBT*.

5. Conclusions

The following are the conclusions from the results obtained after experiments performed on 63.3 ccm two-stroke engines, fueled with gasoline and constant supply rate of 1.12 L/min H_2 plus gasoline at constant speed and varied throttle opening percentage.

1. The effect of the hydrogen gaseous state negatively affects the delivery ratio and scavenging ratio, but hydrogen's high flammability limit permits slight mixture leaning with stable operation under the constraint of Watson method rich operation.
2. Enriching gasoline with 1.12 L/min H_2 shows an improved ability of the engine to minimize the short-circuiting phenomenon, filling the cylinder with fresh air, and replacing burnt residuals with a fresh charge which is characterized by Trapping, Charging, and Scavenging Efficiencies respectively.
3. Improved gas exchange efficiencies positively reflected on brake specific fuel consumption and brake thermal efficiency compared to pure gasoline running.
4. Due to the high burning rate of hydrogen, hydrogen-enriched fuel shows a less advanced angle for *MBT*.
5. Excellent results can be easily obtained if more mixture leaning is permitted to take advantage of the wide flammability limit of hydrogen.

Acknowledgments

The paper was completely supported by Ain shams university.

List of Abbreviations

AFR *Air-fuel ratio*
ICE Internal combustion engines
BTE Brake thermal efficiency
BSFC Brake specific fuel consumption
$BSNO_x$ Brake specific production of NO_x emissions
DR Delivery ratio
EFI Electronic Fuel Injection
L/min Liter Per Minute
MBT Minimum advance for best torque
NTP Denotes normal temperature (293 K) and normal pressure (1 atm)
NO_X Nitrogen Oxides
WOT Wide Open Throttle
LHV Lower Heating Value

Nomenclature

M_o Mass of gas that would flow to the crankcase of volume equal to the cylinder capacity in normal conditions (g)
Md Mass of gas delivered to the crankcase (g)
Mtr Mass of fresh charge trapped at the end of the gas exchange process.
Mres Mass of residual products from the past cycle.
M_{cyl} Calculated mass of fresh charge trapped at the end of the ideal scavenging process.
r Compression ratio.
γ Specific heat ratio.
φ Equivalence ratio.
V_D Engine displacement volume.
T_i Inlet temperature of the mass of the cylinder charge
P_{exh} Exhaust pressure of the cylinder charge.
λ Relative air/fuel ratio (actual AFR to stoichiometric AFR).

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