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High load performance and combustion analysis of a four-valve direct injection gasoline engine running in the two-stroke cycle

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HIGHLIGHTS

- The two-stroke cycle was achieved in a four-valve highly boosted gasoline engine.
- High charging efficiencies were realised by means of valve timing optimisation.
- The engine achieved 2.4 MPa equivalent IMEP with 7 MPa in-cylinder pressure.
- Scavenging inefficiencies and poor air–fuel mixing limited the high speed operation.

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ABSTRACT

With the introduction of CO_2 emissions legislation or fuel economy standards in Europe and many countries, significant effort is being made to improve spark ignition gasoline engines because of their dominant market share in passenger cars and potential for better fuel economy. Amongst several approaches, the engine downsizing technology has been adopted by the automotive companies as one of the most effective methods to reduce fuel consumption of gasoline engines. However, aggressive engine downsizing is constrained by excessive thermal and mechanical loads as well as knocking combustion and low speed pre-ignition (also known as super-knock). In order to overcome such difficulties, a gasoline direct injection single cylinder engine was modified to run under the two-stroke cycle by operating the intake and exhaust valves around bottom dead centre (BDC) at every crankshaft revolution. The combustion products were scavenged by means of a reversed tumble flow of compressed air during the positive valve overlap period at BDC. The engine output was determined by the charging and trapping efficiencies, which were directly influenced by the intake and exhaust valve timings and boost pressures. In this research a valve timing optimisation study was performed using a fully flexible valve train unit, where the intake and exhaust valve timings were advanced and retarded independently at several speeds and loads. A supercharger was used to vary the load by increasing the intake pressure. The effects of valve timing and boost pressure in this two-stroke poppet valve engine were investigated by a detailed analysis of the gas exchange process and combustion heat release. Gaseous and smoke emissions were measured and analysed. The results confirmed that the two-stroke cycle operation enabled the indicated mean effective pressure to reach 1.2 MPa (equivalent to 2.4 MPa in a four-stroke cycle) with an in-cylinder pressure below 7 MPa at an engine speed as low as 800 rpm. The engine operation was limited by scavenging inefficiencies and short time available for proper air–fuel mixing at high speeds using the current fuel injector. The large amounts of hot residual gas trapped induced controlled auto-ignition combustion at high speeds, and thus the abrupt heat release limited higher loads.

Abbreviations: ATDC, after top dead centre; BSFC, brake specific fuel consumption; CA, crank angle; CAI, controlled auto-ignition; CO, carbon monoxide; COV_{IMEP}, covariance of the indicated mean effective pressure; $dP/d\theta$, rate of pressure rise; ECR, effective compression ratio; EER, effective expansion ratio; EGR, exhaust gas recycling; EVC, exhaust valve closing; EVO, exhaust valve opening; GDI, gasoline direct injection; UHC, unburned hydrocarbon; IMEP, indicated mean effective pressure; ISCO, indicated specific carbon monoxide; ISFC, indicated specific fuel consumption; ISUHC, indicated specific unburned hydrocarbon; ISNOx, indicated specific oxides of nitrogen; IVC, intake valve closing; IVO, intake valve opening; KLS, knock limited spark advance; LHV, lower heating value; MBT, minimum spark advance for maximum break torque; NOx, oxides of nitrogen; rpm, revolutions per minute; SI, spark ignition.

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1. Introduction

Two-stroke engines are well known for their superior power density and reduced weight compared to equivalent four-stroke units and are employed to power handheld tools to large marine engines [1,2]. Their use for high performance purposes is widely spread for motorbikes, snowmobiles and outboard vehicles, with claimed power densities above 220 kW/L [3]. However, these advantages, mainly related to crank-case scavenged two-stroke engines, are often offset by drawbacks regarding gaseous emissions, thermal efficiency and engine components durability [4].

On the subject of emissions, the fuel short-circuiting in mixture scavenged two-stroke engines results in significant unburned hydrocarbon (UHC) emissions. The lubricant added to the fuel has much less effect on emissions from crank-case scavenged two-stroke engines according to [3], as modern units use proportions as low as 1% of oil in the fuel. Regarding the thermal efficiency, conventional two-stroke engines usually lose expansion work in favour of enhanced scavenging through early exhaust port opening. This procedure uses the exhaust blow-down phase to reduce the levels of residual gas trapped prior to the intake process, ensuring higher degrees of charge purity [5]. Lastly, the reduced components durability (piston, rings and liner) of ported two-stroke engines can be attributed to uneven thermal loads and reduced lubricant oil film when UHC emissions is a concern [6]. It is important to keep in mind that all these disadvantages are related to cross-scavenged and loop-scavenged two-stroke engines with intake and exhaust ports, where the crank-case is employed as a pump for the air or air/fuel mixture and therefore lubricant oil needs to be added to the air stream. Such problems can be avoided by the uniflow two-stroke engine concept, in which externally compressed air is supplied through ports at bottom dead centre (BDC) and the exhaust gas is forced out through conventional poppet valves in the cylinder head. Greater charging efficiencies can be achieved with such designs [1], but production complexity and packaging restrictions have limited its application to large marine diesel engines so far, though some attempts have been made to adopt such an engine design for vehicular applications [7].

In the beginning of 1990 a new concept of two-stroke operation was proposed as a possible solution to overcome the problems related to conventional ported two-stroke engines. Based on the design of modern four-stroke engines, the two-stroke scavenging process was achieved through the overlap period of overhead intake and exhaust valves around BDC at every engine revolution [2,8]. Because of the use of poppet valves higher power outputs could be achieved with the same engine durability as four-stroke engines. The high levels of UHC emissions due to fuel short-circuiting had been addressed by direct fuel injection and air-assisted fuel injection [5]. The lubricant oil consumption, characteristic of crank-case scavenged engines, had been eliminated by using wet sump and external scavenge pump, mostly roots blower superchargers. When applying this concept to Diesel engines, 40% higher torque and reduced combustion noise compared to an equivalent four-stroke model was demonstrated by Toyota [6].

A reported problem of gasoline direct injection (GDI) two-stroke poppet valve engines was the insufficient mixing between fuel and air in the cylinder, mainly attributed to the shorter time available and the relatively lower injection pressures used at the time [8]. This poor air–fuel mixing results in stratified charging and incomplete combustion, with large emissions of carbon monoxide (CO), UHC and soot [9]. The boosted air supplier, essential for this type of two-stroke engine, has been also reported as responsible for hindering fuel consumption due to isentropic and mechanical inefficiencies [10]. Nevertheless, significant advances have been made in high pressure fuel direct injection systems and high efficiency boosting devices (superchargers, turbochargers and e-boosters). In addition, flexible variable valve actuation systems, particularly fast variable cam devices, have been developed to enhance air management for production engines [11]. Such technological improvements have prompted renewed interest in developing two-stroke poppet valve gasoline engines [12,13] and diesel engines [10,14], considering their potential for engine downsizing with lower in-cylinder pressures and less structural stresses than downsized four-stroke engines [15,16]. Furthermore, the two-stroke poppet valve engine shares nearly all components from the contemporary four-stroke engine architecture and hence can be produced from the same manufacturing process.

The majority of studies with two-stroke poppet valve engines has taken advantage of the inherent residual gas trapped at reduced charging efficiencies to achieve high in-cylinder thermal conditions, and hence induce auto-ignition combustion in both spark-ignition [12,13,17] and compression-ignition engines [18,19]. In the auto-ignition combustion high thermal efficiency and near zero oxides of nitrogen (NOx) emissions can be achieved across the part load engine operation range. However, the low charging efficiency, usually 70% at most, results in large amounts of hot residual gas trapped and increased combustion noise at high loads. The violent combustion in this case limits the engine load to values of indicated mean effective pressure (IMEP) from 0.5 to 0.7 MPa using commercial gasoline.

Considering this, the present study aims at investigating the two-stroke poppet valve engine behaviour under charging efficiencies above 90% at low speeds, showing the full potential of this type of engine for aggressive downsizing and downspeeding compared to four-stroke engines. Such level of charging efficiency could be only achieved with high values of boost pressure and a valve timing optimisation study, where the intake and exhaust valves were operated independently aiming at reducing the levels of residual gas trapped to a minimum. The effect of the scavenging process on the engine performance was evaluated at several speeds and loads, whilst measurements of gaseous and smoke emissions, as well as a heat release analysis, were performed to study the air–fuel mixing and combustion process. High values of IMEP with low in-cylinder pressures are expected in this two-stroke direct injection gasoline poppet valve engine.

2. Experiments

2.1. Experimental facilities

All the experiments were conducted on a single cylinder research engine mounted on a transient test bed. The engine was equipped with an electro-hydraulic fully variable valve train unit capable of independent control over the timings and lifts of each of the four valves, enabling both two-stroke and four-stroke cycles operation [20]. The valve control unit operated under closed loop control over the oil pressure, oil temperature and valve timing/lift, ensuring precise valve operation up to 3000 rpm in the two-stroke cycle. A Ricardo rCube engine control unit was used to manage the throttle position, injection pulse width, spark timing and valve parameters. An AC dynamometer enabled both motored and fired operations whilst an external cooling system provided fully automated control over the engine oil and coolant temperatures. Commercial gasoline (95 RON) was directly injected into the combustion chamber through a Denso solenoid double-slit type injector [21]. The instantaneous fuel mass flow rate was measured