Energy Conversion and Management 123 (2016) 209–217

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

Effect of the Miller cycle on the performance of turbocharged hydrogen internal combustion engines

Qing-he Luo, Bai-gang Sun *

School of Mechanical Engineering, Beijing Institute of Technology, Beijing 100081, China

ARTICLE INFO

Article history:
Received 11 January 2016
Received in revised form 16 May 2016
Accepted 12 June 2016
Available online 20 June 2016

Keywords:
Miller cycle
Hydrogen internal combustion engine
Numerical calculation

ABSTRACT

Hydrogen is a promising energy carrier, and the port fuel injection (PFI) is a fuel-flexible, durable, and relatively cheap method of energy conversion. However, the contradiction of increasing the power density and controlling NOx emissions limits the wide application of PFI hydrogen internal combustion engines. To address this issue, two typical thermodynamic cycles—the Miller and Otto cycles—are studied based on the calculation model proposed in this study. The thermodynamic cycle analyses of the two cycles are compared and results show that the thermal efficiency of the Miller cycle (η_{Miller}) is higher than η_{Otto}, when the multiplied result of the inlet pressure and Miller cycle coefficient ($\delta_M\gamma_M$) is larger than that of the Otto cycle (i.e., the value of the inlet pressure ratio multiplied by the Miller cycle coefficient is larger than the value of the inlet pressure ratio of the Otto cycle). The results also show that the intake valve closure (IVC) of the Miller cycle is limited by the inlet pressure and valve lift. The two factors show the boundaries of the Miller cycle in increasing the power density of the turbocharged PFI hydrogen engine. The ways of lean burn + Otto cycle (LO), stoichiometric equivalence ratio burn + EGR + Otto cycle (SEO) and Miller cycle in turbocharged hydrogen engine are compared, the results show that the Miller cycle has the highest power density and the lowest BSFC among the three methods at an engine speed of 2800 rpm and NOx emissions below 100 ppm. The brake power of the Miller cycle increases by 37.7% higher than that of the LO and 26.3% higher than that of SEO, when γ_M is 0.7. The BSFC of the Miller cycle decreases by 16% lower than that of the LO and 22% lower than that of SEO. However, the advantage of the Miller cycle decreases with an increase in engine speed. These findings can be used as guidelines in developing turbocharged PFI hydrogen engines with the Miller cycle and indicate the boundaries for the development of new hydrogen engines.

1. Introduction

Given the seriousness of the energy problem and the lack of stringent emission regulations, the search for a new type of clean renewable energies has become the most important topic in the energy sector. As an energy carrier, hydrogen is a promising means of achieving clean combustion given that its major combustion product is H_2O when NOx emissions are well controlled [1]. Hydrogen can generally be used in two ways: through direct combustion and through hydrogen-fueled cells. However, hydrogen fuel cells are limited by technical bottleneck and high price, such as proton exchange membrane, carbon plate, and the high price of the metals for catalysis. The direct combustion of hydrogen via hydrogen-fueled internal combustion engines (H_2ICE) can easily and cheaply be achieved without substantial modifications to traditional internal combustion engines [2–4].

Two fuel supply methods can be implemented for H_2ICE: the port fuel injection hydrogen internal combustion engine (PFI-H_2ICE) and direct injection hydrogen internal combustion engine (DI-H_2ICE). DI-H_2ICE is considered an effective means of increasing power density and limiting NOx emissions [5]. The direct injection strategy can have a higher in-cylinder trapped air mass at the start of fuel injection after IVC. However, this method is difficult to apply because of the limited lubrication and cooling of the injector and the large cycle variation. The PFI strategy does not result in such problems, and most of the parts of the gasoline engines can be used in PFI-H_2ICE, which makes the PFI-H_2ICE system cheaper and more reliable than the DI-H2ICE system [6,7].

However, the contradiction of the power density and controlling NOx emissions limits the PFI-H_2ICE, because hydrogen is the gas fuel that takes up the cylinder volume, leading to a decrease in the volume coefficient [8]. To solve this contradiction, the

* Corresponding author.
E-mail address: sunbg@bit.edu.cn (B.-g. Sun).

http://dx.doi.org/10.1016/j.enconman.2016.06.039
0196-8904/

supercharged PFI-H_2ICE has been studied by many researchers. Verhelst et al. [9] results showed that maintaining the supercharging stoichiometric mixtures coupled with exhaust gas recirculation (EGR) is a better method of controlling NOx emissions than the method using lean mixtures. Ghent University developed a supercharging H_2ICE (GM 7.4 L V8) [10], which can increase maximum torque output by 60% compared to naturally aspirated H_2ICE. White et al. [11] presented a review of H_2ICEs on boosted H_2ICEs and showed that such engines are a compelling alternative to fuel cells when used in a series hybrid vehicle featuring very low NOx emissions without any post-treatment at low equivalence ratios. Berckmüller et al. [12] worked on a supercharged stoichiometric equivalence ratio H_2ICE and showed that the peak power output of a four-stroke single cylinder engine is about one third higher than that of a naturally aspirated gasoline engine. Natkin et al. [13] reported work on a supercharged 2.3 L PFI engine with a standard (air-to-air) intercooler and a fixed equivalence ratio of 2. Their results showed that the torque decreases by 28% compared to the base (naturally aspirated) gasoline engine. In the recently introduced Ford E-450 H_2ICE vehicles, a 6.8 L supercharged PFI engine was used at a low equivalence ratio (not specified), eliminating the need for post-treatment [14]. From the above, supercharging can increase the power density under acceptable NOx limits. However, supercharging consumes the useful work of the output, and the EGR cooler system is difficult to achieve because of the combustion products, including a huge amount of water vapor. The combustion products lead to a very high export temperature in the EGR cooler, along with an increase in the inlet temperature.

In the 1940s, Miller proposed a different Otto cycle with unequal compression and expansion stroke called the Miller cycle. The most interesting feature of the Miller cycle is the shorter effective compression stroke and longer expansion stroke, which can increase the thermal efficiency [15]. Li et al. [16] reported the effects of early intake valve closure (EIVC) and later intake valve closure (LIVC) on the fuel economy of a boosted DI gasoline production engine reformed with a geometric CR of 12.0 are experimentally compared at low and high loads. They found that the LIVC could improve the brake specific fuel consumption (BSFC) up to 4.7% than the CR of 9.3 at high load, but not ELVC. At the low load operation, combined with CR12.0, LIVC and EIVC improve the fuel economy by 6.8% and 7.4%, respectively, compared to the production engine. Hou [17] comparison of performances of air standard Atkinson and Otto cycles with heat transfer considerations including combustion constants, compression ratio and intake air temperature. They found that an Atkinson cycle (one kind of Miller cycle) has a greater work output and a higher thermal efficiency than the Otto cycle at the same operating condition. On the other hand, the technology of PFI-H_2ICEs has been very mature [18] and the way of PFI can ensure the uniformity of mixture to decrease the cycle variations. Thus, this study proposes using the Miller cycle in PFI-HICEs with turbocharger to solve the contradiction of the power density and controlling NOx emissions as the limitation of PFI-H_2ICE. To study the effect of the Miller cycle, a calculation model is developed based on experimental data on a turbocharged hydrogen engine.

2. Materials and method

2.1. Test equipment and process

A correctly set-up engine and test cell with an appropriate data-acquisition system were built to acquire precise and reliable information. Numerous tests were conducted on a 2.3 L four-cylinder hydrogen engine with turbocharger.

The test cell included a CW250 eddy current dynamometer, which was used for energy absorption and engine speed regulation. The dynamometer had a capacity of 250 kW and a maximum rated speed of 8000 rpm and was attached to an external blower, which volume flow rate is 500 m^3/h. The engine speed could be controlled at the desired level, and the torque was varied. The speed, engine oil temperature, coolant temperature, and intake air temperature were recorded automatically from the dynamometer control console. The cylinder pressure was measured using a Kislter 6117B pressure sensor, and the crank angle position was specified by the crank angle encoder, which type is Kislter 2613B (which can measure 0–20,000 r/min and the accuracy can reach to 0.1 °CA). The cylinder pressure and the corresponding crank angle were captured through a high-speed data acquisition system (Kibox combustion analyzer). The output from these measurements was the diagrams of pressure versus crank angle. The inlet air flow was measured by the Hot-film Air Mass Flow Meter, which type is ToCeiL20N. The hydrogen mass flow was measured by the Coriolis Mass Flow Meter, which type is CMF025 made by EMERSON. The specifications were listed in Table 1.

Before conducting the test, the engine was warmed up to ensure that it reached the operating temperatures and that it stabilized. Tests were conducted after running the engine until it reached a steady state at an oil temperature of 90 °C and cooling water temperature of 80 °C. The data was then recorded after running the engine with hydrogen fuel. All the tests were run at the MBT to obtain comparable data. The MBT tests were done through modifying the ignition timing at different engine speed and load. For example, to get the MBT at 2800 r/min and the wide open throttle, the engine speed should be kept at 2800 r/min and the throttle was wide open, then modifying the hydrogen mass flow to keep the equivalence ratio at constant (such as 0.55). After these steps, the ignition timing was modified internal of 1 °CA or bigger values through the electronic system. Record the test data until finding the MBT. Repeat above steps to get the other test data at different speeds, equivalence ratios and throttle open angles.

All measurements of physical quantities have some degree of uncertainty, owing to various sources. Therefore, uncertainty analysis is desirable, to confirm the precision of the tests. The uncertainty for the experimental results is determined according to the principle of root-mean square method, to get the magnitude of the error given by Gaussian distribution, as follows:

$$\Delta R = \left[\left(\frac{\partial R}{\partial x_1} \Delta x_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} \Delta x_2 \right)^2 + \cdots + \left(\frac{\partial R}{\partial x_n} \Delta x_n \right)^2 \right]^{1/2} \tag{1}$$

Table 1
Average uncertainty of the main measured parameters.

Variable	Device	Accuracy	Parameters	2800 rpm, WOT[a]	Error	(%)
Engine speed	FC2000	±1 r/min	Brake power	28.1668	0.0574	0.20
Crank angle	Kislter 2613B	±0.02°	ITE[b]	32%	0.07%	0.23
In-cylinder pressure	Kislter 6117B	±0.4% AR[a]	ISFC[b]	93.7450	0.2127	0.23
Air mass flow rate	ToCeiL20N	±1%FS[a]	Volumetric efficiency	83.27%	0.77%	0.92
Hydrogen mass flow rate	CMF025	±0.1%FS[a]	Equivalence ratio	0.5170	0.0026	0.51

[a] FS: Full scale, AR: All range, WOT: wide open throttle.
[b] ITE: Indicated Thermal Efficiency, ISFC: Indicated Specific Fuel Consumption.