20th Australasian Fluid Mechanics Conference
Perth, Australia
5-8 December 2016

Investigation to Combustion and Emission Characteristics of the Dual Ethanol Injection Spark Ignition Engine

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Abstract

Ethanol fuel, as a bioproduct with greater octane number, combustion speed and latent heat of vaporization, has become a common choice as an additive and/or an alternative option to gasoline fuel in the spark ignition engines. In order to fully utilize ethanol fuel properties to improve engine performance, a new injection strategy, ethanol port injection plus ethanol direct injection (EDI), has been in development. Work reported in this paper aimed to investigate, experimentally, the effect of ethanol fuel and dual ethanol injection strategy on engine performance, combustion and emissions characteristics at two engine loads and optimized spark timing. The results of both engine loads, light and medium, demonstrated that the indicated mean effective pressure (IMEP) was significantly improved over all dual ethanol injection strategy compared to GPI. The maximum improvement was 3.3485% and 4.357% at light and medium engine loads respectively. The improvement was mainly due to the reduced combustion duration ($\theta_{10\text{-}90\%}$) which was reduced by 8.15CAD at light load and 4.28CAD at medium load compared to GPI. However, at higher EDI percentages, the over cooling effect and poor mixture quality adversely affected the combustion quality. The indicated specific nitric oxide emission was considerably reduced, at 100% of EDI, by up to 55.1% and 58.46% at light and medium loads respectively. Nevertheless, because of poor mixture quality and high wall wetting, the indicated specific hydrocarbon and the indicated specific carbon monoxide were raised with the increase of EDI percentage. Regarding the effect of spark timing, the dual ethanol injection strategy improved the IMEP significantly at the maximum IMEP spark timing.

Introduction

Global warming and climate change have become a strong driving force to issue more strengthen legislations against emissions from internal combustion engines. Focusing on the automotive industry, Euro 5 and 6 are clear instances that indicate the importance of the environmental protection issue. In order to reduce the pollutant emissions from vehicles, the specific fuel consumption of the vehicle engines need to be further reduced, and engine performance needs to be improved. Many new technologies have been developed to optimize the engine performance by controlling the amount of the consumed fuel on the real time engine conditions. Gasoline direct injection has been used in turbocharged spark ignition (SI) engines [1-3]. This technology aimed to reduce the engine size by enhancing the specific engine output power that could lead to moderate the specific fuel consumption and thus emission reduction.

Recently, more attention has been paid to the ethanol fuel in the automotive industry to use ethanol as an alternative fuel or enhancer material to the gasoline fuel. Ethanol as renewable fuel can be produced from a wide range of crops such as sugarcane [4]. Ethanol has a greater research octane number (RON), latent heat of vaporization and combustion speed compared to the unleaded gasoline fuel. Therefore, ethanol fuel has been adopted in the SI engines for vehicles. Extensive research has been conducted to investigate the effect of blended ethanol and gasoline fuels on the SI engine performance with port fuel injection systems [5-8]. Results showed that small percentage of ethanol fuel could slightly improve the engine performance. Higher ethanol percentages were commonly used in Brazil and United State mainly for reducing the consumption of hydrocarbon fuels. The effect of E85 on the engine performance, combustion and emission characteristics was investigated [9-12]. Dual fuel injection strategy has been applied in the naturally aspirated/turbocharged engines fuelled with gasoline only or blended gasoline plus ethanol [2, 13, 14]. Recently, a new technique of ethanol direct injection (EDI) plus gasoline port injection (GPI) has been in development, aiming to use ethanol fuel more efficiently in SI engines [15, 16]. However, using pure ethanol fuel to be injected directly into GPI engine requires two independent fuel feeding systems. Moreover, the high percentage of ethanol fuel directly injected into the combustion chamber could cause wall wetting and consequently deterioration in the combustion and emissions [15, 17].

In order to maximize the benefits of using ethanol fuel, dual ethanol injection strategy, EDI plus ethanol port injection (ethanol PI) was experimentally investigated. The reported work aimed to examine the effect of dual ethanol injection strategy on the SI engine performance, combustion and emissions characteristics. Experiments were conducted at a stoichiometric air to fuel ratio, two engine loads and optimized spark timing to the maximum indicated mean effective pressure (MBT) was identified. The results are compared with that at GPI only and 100% EDI.

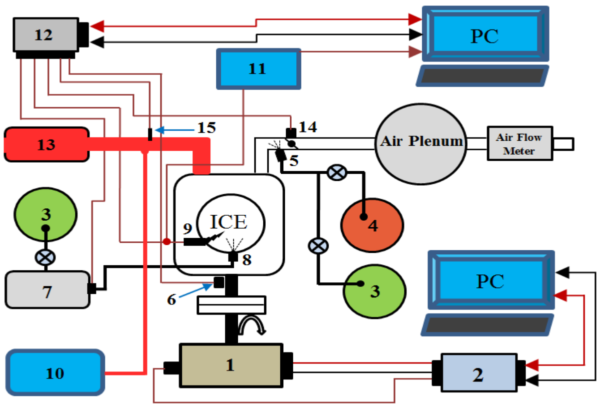
Experimental Setup and Methodology

I. Engine Test Rig

A single cylinder, naturally aspirated spark ignition engine was used to conduct experiments. Engine specifications were listed in table 1. This small air-cooled engine originally had a gasoline port injection only before it was modified to be equipped with a direct injection system. Figure 1 shows a schematic diagram of the engine testing rig. As shown in Figure 1, an electronic control unit (ECU, 12) developed by Hents Technology was used to adjust and control the engine operating parameters such as throttle position and mass of fuel in both fuel injection systems. In order to set the required engine speed and measure the engine torque, an eddy current dynamometer was coupled to the engine. A Kistler 6115B spark plug cylinder pressure transducer was used to record the in-cylinder pressure. MEXA-584L Horiba gas analyser was used to measure the exhaust gas emissions of CO, CO_2, HC, NO and lambda (λ). The intake air flow rate was measured using a ToCeiL20N thermal air-mass flow meter.

Engine Type	Single Cylinder, Four-Stroke
Stroke x Bore	58 mm x 74 mm
Displacement	249 cc
Compression Ratio	9.8:1
Intake Valve Open x Closed	22.2° bTDC x 53.8° aBDC
Exhaust Valve Open x Closed	54.6° bBDC x 19.3° aTDC
Ethanol Fuel System	Direct plus Port Injection
Gasoline Fuel System	Port Injection Only

Table 1: Engine Test Rig Specification.



1. Eddy Current Dynamometer 2. Dynamometer Controller 3. Ethanol Fuel Tank 4. Gasoline Fuel Tank 5. Low Pressure Injector 6. Crankshaft Encoder 7. High Pressure Ethanol Pump 8. High Pressure Ethanol Injector 9. Kistle 6115B Pressure Transducer 10. MEXA-584L Gas Analyser 11. Charge Amplifier 12. Electronic Control Unit 13. Exhaust Gas Catalyser 14. Throttle Position Sensor 15. Bosh Wide-band Lambda Sensor.

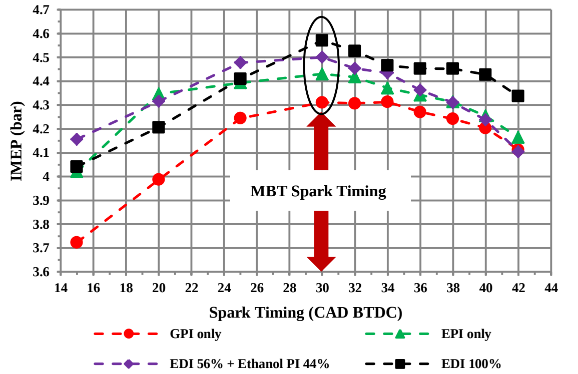
Figure 1. Schematic Diagram of Dual Fuel Injection SI Engine.

II. Experimental Procedure

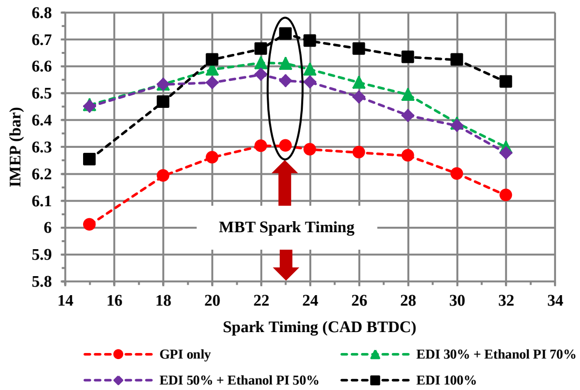
Gasoline port injection only (GPI) was set as a reference line for investigating the dual ethanol injection. The experimental work was conducted at a light engine load with 20% throttle opening and a medium engine load with 33% throttle opening. The volumetric ratio of the ethanol fuel in direct injection was changed from 0% as ethanol PI only to 100% EDI, but the total fuel heating energy was kept unchanged, ~540J at medium load and ~400J at light load. The direct injection timing was fixed at 300CAD BTDC and port injection timing at 410CAD BTDC. The pressure of the EDI and the fuel PI were fixed at 4 MPa and 0.25 MPa respectively. The air/fuel ratio set at about stoichiometric (λ=1). Pre-experiments were conducted to find the MBT spark timing to this experimental work at a fixed engine speed of 3500 RPM. The engine was started and warmed up to 200°C (cylinder head temperature), as the designated engine operating temperature, with GPI, and then the percentage of EDI gradually increased starting from ethanol PI only to 100% EDI. Five samples of data were taken at each tested engine operation condition and the sample average was used in the calculations and analyses. The in-cylinder pressure was recorded at a rate of 0.5 crank angle degree (CAD) intervals three times with 100 consecutive cycles each time. The ensemble average of the cylinder pressure data was used in calculations of IMEP and combustion duration.

III. Identification of MBT Spark Timing

Experiments for finding the MBT were conducted at 3500 RPM engine speed and two engine loads. Figure 2 shows the variation of IMEP with spark timing at a wide range of EDI percentages starting from 0% to 100% of EDI. As shown in figure 2, the spark timing was swept from 15CAD BTDC to 42 CAD BTDC at the light load and to 32 CAD BTDC at the medium load. The IMEP increases with the advanced spark timing from 15 CAD BTDC to around 30 CAD BTDC at light load and to around 23CAD BTDC at medium load. When the spark timing was further advanced, the value of IMEP was decreasing with the proceeded spark timing. As it was concluded, the best IMEP could be achieved when the majority of the combustion takes place near TDC [18]. The improvement of IMEP with advancing the spark timing could be attributed to the combustion quality enhancement and the right phase, near TDC, at which the largest portion of combustion [19]. On the other hand, the reduction in the IMEP with further advance of spark timing could be attributed to the mixture quality deterioration due to the time shortage to the fuel to be homogeneously mixed with air. Furthermore, the negative work due to the early spark timing could adversely effect on the IMEP value. Based on the results shown in Figure 2, the MBT spark timing was set to be 30CAD BTDC for the light load and 23CAD BTDC for the medium load.



a) Light Engine Load (20% throttle opening).



a) Medium Engine Load (33% throttle opening).

Figure 2. Variation of IMEP with Spark Timing.

Results and Discussion

I. Engine Performance and Combustion Characteristics

Experimental results of dual ethanol injection strategy will be compared with that in GPI only and 100% EDI conditions to examine the effect of dual ethanol injection on engine performance. The results of engine performance and the combustion characteristics will be presented and discussed in this section. Figure 3 shows the IMEP of dual ethanol injection compared with that of GPI only. The engine IMEP was significantly improved over all the range of dual ethanol injection percentages. However, at both engine loads, the effect of EDI strategy on the IMEP is essentially negligible. At light engine load, the maximum increase of IMEP was 3.485% at the EDI ratio of 46%. At medium engine load, the maximum increase of IMEP was 4.357% at the EDI ratio of 66%. The engine performance was improved when the ethanol injected to the intake port, and this improvement was steady until the EDI 100% percentage reached. This independent behaviour of the IMEP, at both engine loads, from the amount of the volumetric percentage of the EDI could be attributed to the constant air/fuel mixture properties (ethanol fuel only + air) to the ethanol dual injection strategy tests. This demonstrates that the ethanol properties could play a significant role in the IMEP enhancement rather than the EDI. Greater combustion speed and lower heat losses to the combustion chamber walls thanks to lower burnt gas temperature could play a significant role in enhancing the IMEP when a dual ethanol injection strategy was used [6].

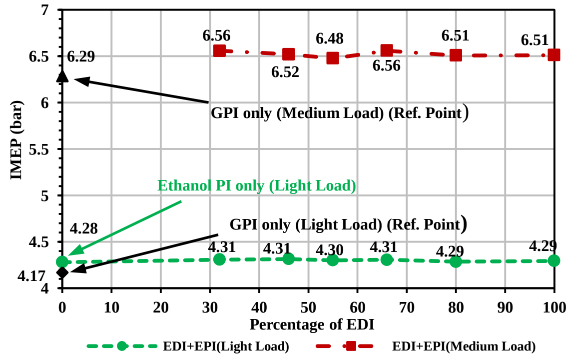


Figure 3. Variation of the IMEP with the EDI parentages.

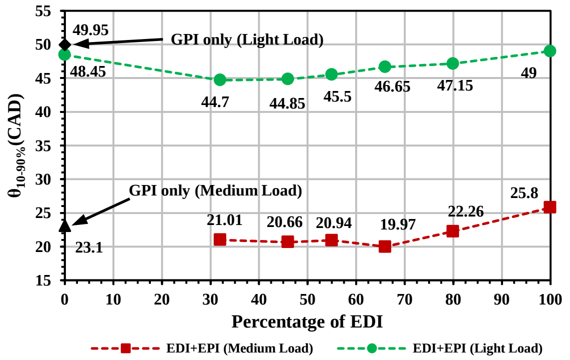


Figure 4. Variation of combustion duration with EDI percentages.

Figure 4 shows the combustion duration ($\theta_{10-90\%}$) of dual ethanol injection strategy compared with that in GPI only. The $\theta_{10-90\%}$ is defined by the crank angle degree from the crank angle at which 10% of the fuel is burnt to the crank angle at which 90% of the fuel burnt [18]. At light engine load, figure 4 shows that $\theta_{10-90\%}$ decreases directly after the ethanol port injection started and the combustion duration continue decreases with the percentage of ethanol directly injected reaching the minimum value at 32% of EDI. In medium engine load condition, the $\theta_{10-90\%}$ decreases slightly in the range of EDI from 32% to 66%. The shortest $\theta_{10-90\%}$ occurred at EDI 32% in light load conditions and at EDI 66% in medium load conditions. This could be attributed to two main reasons. Firstly, the ethanol's combustion speed is faster than the gasoline's one which could be combined with the oxygen content of ethanol resulted in a combustion quality improvement. Secondly, injecting a right portion of ethanol fuel directly into the combustion chamber probably enhance the homogeneity of the air-fuel mixture and thus reduce the combustion duration. However, the $\theta_{10-90\%}$ becomes to increase with the increase of EDI ratio when the EDI exceeds 66% at light load and 80% at medium load. This could be attributed to two main reasons. Firstly, at high EDI percentages, fuel impingement might become significant and lead to wall wetting which could adversely affect the mixture quality. Secondly, the over cooling effect might reduce the combustion temperature to be too low and thus reduce the combustion speed leading to a longer combustion duration.

II. Emission Characteristics

Figure 5 shows the variation of the indicated specific nitric oxide ($ISNO_X$) at the light and medium engine loads with dual ethanol injection strategy. $ISNO_X$ emission is highly related to the combustion temperature [20]. As shown in figure 5, at medium load, the $ISNO_X$ was significantly decreased directly after the ethanol dual injection strategy started at EDI 32%, compared with GPI only. When the ethanol dual injection strategy was used, the $ISNO_X$ reduced with the percentage of EDI because of the cooling effect enhanced by direct injection. The largest reduction was recorded when a 100% of EDI was used at both engine loads. Compared with GPI only, a 55.1% and 58.46% were the amounts of the $ISNO_X$ reduction at light and medium engine loads respectively. This could be attributed to two main reasons. Firstly, concerning the ethanol fuel properties, the high ethanol's latent heat of vaporization could significantly contribute in the ISNOx reduction. This could explain the ISNOx reduction when the ethanol PI only used at light engine load. Secondly, the direct injection strategy could play an important role in fully utilizing of the ethanol fuel properties such as the latent heat of vaporization compared to ethanol PI only which resulted in further decrease in the combustion temperature [6, 13, 16].

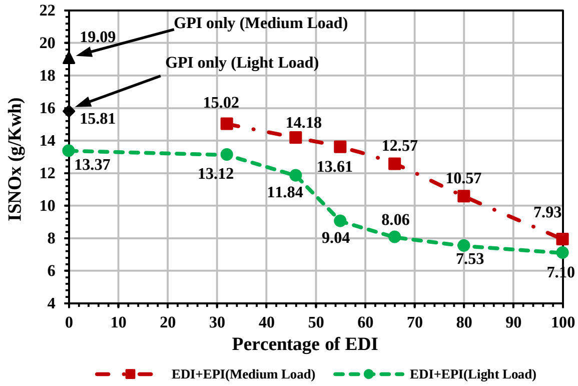


Figure 5. Variation of ISNOx with EDI percentages.

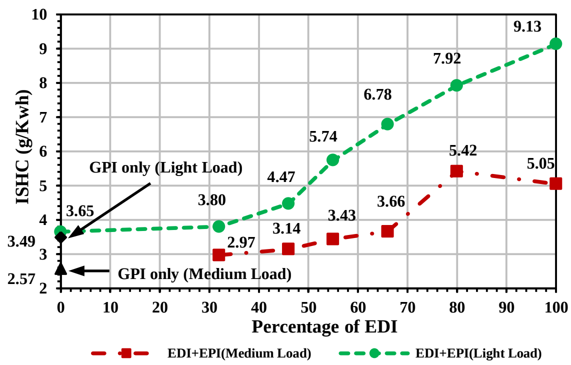


Figure 6. ISHC variation with the EDI Percentages.

The dual ethanol injection strategy effect on the indicated specific hydrocarbon (ISHC) at two engine loads is shown in figure 6. The ISHC was increased when the dual ethanol injection started at EDI percentage at 32% for both engine loads.

The ISHC percentage was continued increasing along with the percentage of EDI reaching the maximum value at 100% of ethanol. The incomplete combustion could be the main reason for this unfavourable tendency of the ISHC which might be caused by two main reasons. Firstly, a poor mixture quality which could be caused by a high fuel impingement on the combustion chamber walls, forming a fuel film, resulting in the nonhomogeneous distribution of ethanol fuel which is directly injected into the combustion chamber [16]. Secondly, the low combustion temperature and the consequent incomplete combustion could be caused by not only the ethanol's latent heat of vaporization but also the cooling effect enhanced by the direct fuel injection [15, 21].

Figure 7 shows the effect of dual ethanol injection strategy on the indicated specific carbon monoxide (ISCO) emission at the two defined engine loads. The lack of oxygen and incomplete combustion are the two main reasons for incomplete combustion resulting in ISCO formation [18]. As shown in figure 7, at light engine load, the ISCO is significantly reduced in the range of EDI of 0% to 46%, compared with the ISCO in the GPI only. This tendency could be attributed to the ethanol's oxygen content and combustion speed combined with mixture quality that possibly improved due to injecting a relatively small amount of ethanol directly to combustion chamber [2]. However, when the EDI percentage goes over 46% at light load and 32% at medium load, the over cooling effect might reduce the combustion temperature to be too low result in incomplete combustion and thus a greater quantity of ISCO emission [18]. Furthermore, a poor mixture quality due to the high level fuel impingement of ethanol fuel on to the combustion chamber walls and the lower ethanol's vapor pressure at high EDI percentages for both engine loads might adversely affect the combustion quality which led to the increase of the ISCO emission.

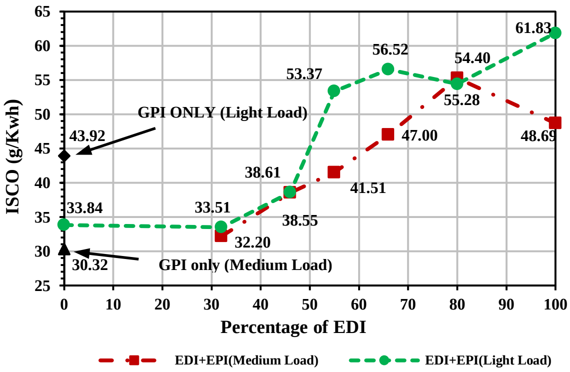


Figure 7. Variation of ISCO with the EDI percentages.

Conclusion

In order to optimize the benefits of ethanol fuel properties to the spark ignition engine, a new injection strategy, ethanol port injection plus ethanol direct injection was experimentally investigated. Based on the above results and discussions, number of conclusions can be drawn as the following:

1. In light and medium engine load conditions, when the total fuel heating energy was kept the same, the IMEP with dual ethanol injection was greater than that with GPI only. Analysis of the results showed that this improvement in IMEP was mainly due to the ethanol fuel properties such as oxygen content, fast combustion speed and adiabatic flame temperature rather than the direct injection strategy.
2. The combustion duration ($\theta_{10\text{-}90\%}$), at medium engine load, was reduced when the EDI in the range of 32% to 66%. This shows the positive effect of ethanol's higher combustion speed and the enhanced mixture quality when a certain percentage of the ethanol was injected directly into the combustion chamber. However, because of the over cooling effect and mixture quality deterioration, the $\theta_{10\text{-}90\%}$ increased when the EDI percentage is over 80%.
3. The EDI strategy significantly reduced the emission of the indicated specific nitric oxide ($ISNO_X$). This was due to the combined of enhanced cooling effect of ethanol's latent heat of vaporization and direct injection strategy.
4. As a result of incomplete combustion and a non-homogeneous air-fuel mixture when the percentage of EDI was greater than 32%, the ISHC increased with the increase of EDI percentage compared to GPI only. The maximum value of ISHC was recorded at 100% at light load and 80% of EDI at medium load respectively.
5. At light engine load, the ISCO was reduced when a relatively small amount of EDI percentage was used. This performance could be imputed to a better combustion and mixture quality which could reduce the amount of ISCO emission. On the other hand, at both engine loads, the ISCO was increased when a high percentage of EDI was used which could be attributed to the over cooling effect and poor mixture quality. The maximum value of ISHC was recorded at 80% and 100% of EDI at light and medium loads respectively.

References

[1] F. Bonatesta; E. Chiappetta; A. La Rocca, Applied Energy 124 (2014) 366-376.
[2] D. Turner; H. Xu; R. F. Cracknell; V. Natarajan; X. Chen, Fuel 90 (5) (2011) 1999-2006.
[3] M. Hedge; P. Weber; J. Gingrich; T. Alger; I. Khalek, SAE Int. J. Engines (2011).
[4] P. Bajpai, Advances in Bioethanol, Springer New Delhi Heidelberg New York Dordrecht London, India, 2013, p.^pp. 100.
[5] A. S. Ramadhas; P. K. Singh; P. Sakthivel; R. Mathai; A. K. Sehgal in: *Effect of Ethanol-Gasoline Blends on Combustion and Emissions of a Passenger Car Engine at Part Load Operations*, SAE International, 2016; SAE International: 2016.
[6] R. A. Stein; J. E. Anderson; T. J. Wallington, SAE Int. J. Engines 6 (1) (2013) 470-487.
[7] P. Bielaczyc; A. Szczotka; J. Woodburn, in: SAE International: 2011.
[8] R. C. Costa; J. R. Sodré, Fuel 89 (2) (2010) 287-293.
[9] C. J. H. a. T. G. L. Robert A. Stein; F. M. Company, SAE Int. J. Engines (2009).
[10] M. Koç; Y. Sekmen; T. Topgül; H. S. Yücesu, Renewable Energy 34 (10) (2009) 2101-2106.
[11] M. Sjöberg; W. Zeng; D. Singleton; J. M. Sanders; M. A. Gundersen, SAE Int. J. Engines 7 (4) (2014) 1781-1801.
[12] P. G. Aleiferis; J. Serras-Pereira; Z. van Romunde; J. Caine; M. Wirth, Combustion and Flame 157 (4) (2010) 735-756.
[13] E. Kasseris; J. B. Heywood, SAE Int. J. Engines (2012).
[14] E. Kasseris; J. B. Heywood, SAE Int. J. Engines (2012).
[15] Y. Zhuang; G. Hong, Fuel 105 (2013) 425-431.
[16] Y. Huang; G. Hong; R. Huang, Applied Energy 160 (2015) 244-254.
[17] Y. Huang; G. Hong; R. Huang, Energy Conversion and Management 92 (0) (2015) 275-286.
[18] J. B. Heywood, Internal Combustion Engine Fundemantales McGraw-Hill, Inc., 1988, p.^pp.
[19] P. Zoldak; J. Naber, in: SAE International: 2015.
[20] B. M. Masum; H. H. Masjuki; M. A. Kalam; I. M. Rizwanul Fattah; S. M. Palash; M. J. Abedin, Renewable and Sustainable Energy Reviews 24 (2013) 209-222.
[21] G. Zhu; D. Hung; H. Schock, Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering 224 (3) (2010) 387-403.