Research study of HONDA NHX 110 powered by an alternative fuel

A Chmielewski1,*, R Gumiński1, T Mydłowski1, A Małecki1, K Bogdziński1

1Warsaw University of Technology, Faculty of Automotive Construction Machinery Engineering, Institute of Vehicles, 02–524, POLAND

E-mail: a.chmielewski@mechatronika.net.pl

Abstract. Nowadays more and more attention is focused on distributed energy generation (e.g. distributed generation sources which are widely promoted by the European Union). The view of 2020 assumes purposes for the each member states of the European Union, connected with respect for energy and alternative fuels. One of the assumptions is to achieve a 10% share of alternative fuels in transport. This objective can be achieved by using internal combustion engines powered by an alternative fuels (e.g.: compressed natural gas – CNG, Liquid natural gas – LNG, hydrogen, biogas, the mixture of bioethanol with gasoline –E85 and others). In the present work, the research study of HONDA NHX 110 fueled with gasoline Pb95, CNG, biogas and E85 have been presented. The experiment was conducted on the author's test stand located on the equipment Institute of Vehicles of the Faculty of Automotive Construction Machinery Engineering of the Warsaw University of Technology. The research study for different ignition advance angles at maximum internal combustion engine load (electricity generation state) have been shown. The influence of the powered by gasoline Pb95, CNG, biogas and E85 on the value of mean indicated pressure, mechanical power generated by internal combustion engine at different angles of the ignition advance have been presented. Moreover, the influence of emission of hydrocarbons – HC and nitrogen oxides – NOx at different angles of the ignition advance have been analysed.

1. Introduction

Currently the European Union is in the process of adopting the framework of the climate–and–energy policy for the 2030 [1] as well as the 2050 perspective. The 2030 perspective [1–3] sets new targets which include: improvement of energy efficiency to the level of 27%, growth of the share of Renewable Energy Sources (RES) to 27% of the total energy consumed in the EU and reduction of greenhouse gases to 40%. The above listed goals will have decisive influence on the development of low–emission economies in respective countries, especially the economies in which important role is played by distributed generation devices [1–14]. As regards Poland, the development of distributed generation devices will have direct influence on competitiveness of the economy. Distributed generation devices include sources which produce energy from alternative fuels (e.g. from biogas). These devices include gas engines. It is the low power engines, which can be powered by alternative fuels, that are of particular importance.

The paper presents a study of a low–power Honda NHX combustion engine powered by Pb95 unleaded gasoline, CNG, E85 and biogas.
Chapter 2 presents and discusses a test bench (in particular the respective components, the measuring circuit and the controlling system). Chapter 3 presents a broad spectrum of conducted tests, while chapter 4 contains the major findings which have been formulated on the basis of the results of the analyses presented in this paper.

2. Description of the testbed

The dynamometer test bench, which is presented in Figure 1, consists of a Honda NHX 110 four-stroke combustion engine, an electrical machine, a programmable ECU MASTER EMU (EMU – Engine Management Unit) [15] controller as well as the measurement circuit and the controlling system. The engine was equipped with a three–way catalyst. In the case of this dynamometer test bench the torque was conveyed to a brushless DC electric motor with fixed magnets, operating as a generator). The generated electrical power was received by such components as a three–phase bridge rectifier (with the following parameters: maximum voltage: 400V, maximum current: 300A), a transistor module, a resistor with the resistance of 0.05 Ω, a brushless electrical motor with fixed magnets (having the following parameters: resistance: 0.0004 Ω, supply voltage 30–70 V, rotational speed: 150 rpm per each 1 V of supply voltage, the maximum rotational speed, corresponding to the maximum supply voltage, was 10500 rpm, power consumption for a motor operating without a load: 13 A when powered with 20 V current). The transistor module was controlled while using a proprietary microchip controller which is described in detail in [16]. The torque from the combustion engine’s shaft was conveyed to the generator while using a belt drive with a toothed (timing) belt (gear ratio i=1.42). Torque was measured with the use of L6N tensometric meter manufactured by Zemic (precision class C3). The rotation angle of the crankshaft was registered with the use of a digital 14bit absolute single–turn encoder. The encoder communicated with the measurement board in SSI standard, having the clock frequency of 44.9kHz, which ensured measurement precision of 0.5 CA for the rotational speed of 3800 RPM (respectively eg. 1 CA for 7600 RPM). The detailed description of the structure of the test bench is presented in [7, 15–19].

![Figure 1. Photograph of the test bench (a), Diagram of the test bench (b).](image)

Chapter 3 presents the results of the tests performed at the above discussed dynamometer test bench.

3. Results of tests carried out at the test bench

The tests were conducted in such a way so as to ensure that in the case of each fuel used for powering the engine the surplus air–fuel ratio was maintained at $\lambda=1$. The temperature of intake air was $26\pm2^\circ$C, atmospheric pressure was identical for all the measurements i.e. 1009 hPa, while the temperature of
the coolant in the engine oscillated in the range of 90–95°C during all the tests. The tests were conducted for the rotational speed of 4500rpm (in accordance with the specification obtained from external sources, this is the rotational speed at which the engine achieves maximum torque) and with fully open throttle.

3.1. Opened diagrams of indicated pressure

This subchapter presents the influence that ignition advance angle has on the values of indicated and opened pressure for HONDA NHX 110 engine powered respectively with: gasoline, CNG, biogas and E85. In the case of gasoline, the tests were conducted for ignition advance angles from IAA=10 to IAA=40, with measurements taken at 10–degree intervals before the Top Dead Center (TDC). Figure 2 presents the influence that the ignition advance angle has on the values of indicated pressure for a gasoline–powered engine. The figures present several hundred consecutive working cycles for each ignition advance angle.

![Figure 2. Influence of the ignition advance angle on the values of pressure for an engine powered with unleaded gasoline (Pb95).](image)

The analysis of Figure 2 shows that the maximum indicated pressure in a cylinder increases along with the growth of the ignition advance angle. The tests demonstrated high scatter of the results in the interval beginning from the crankshaft revolution angle corresponding to the start of the combustion process (for the ignition advance angle of IAA=10, the beginning of the combustion process was observed at 359 CA) until the end of the main combustion process, which corresponds to the fast movement of the combustion front across the combustion chamber [10, 12, 19], together with intense emission of heat and growth of the pressure in the cylinder (for the ignition advance angle of IAA=10, the end of the main combustion process was observed at 445 CA). The maximum indicated pressure was 3.03MPa. The average torque at the engine’s shaft was 8.241 Nm. For the ignition advance angle of IAA=20 before TDC, the beginning of the combustion process was observed for the angle of 349.9 CA. The end of the main combustion process was observed for the angle of 349.9 CA. The end of the main combustion process was observed at 418 CA. The maximum indicated pressure was 4.93 MPa. The average torque at the engine’s shaft was 9.813 Nm. For the ignition advance angle of IAA=30 before TDC, the beginning of the combustion process was observed for the angle of 341.6 CA. The end of the main combustion process was observed at 398.5 CA. The maximum indicated pressure was 6.56 MPa. The average torque at the engine’s shaft was 10.38 Nm. For the ignition advance angle of IAA=40 before TDC, the beginning of the combustion process was observed for the angle of 334 CA. The end of the main combustion process was observed at 395.4 CA. The maximum indicated pressure was 7.48MPa. The average torque at the engine’s shaft was 10.36 Nm. The delay of the ignition increased along with the growth of the ignition advance angle (for IAA=10 it was 9, for IAA=20 it was 9.9, for IAA=30 it was 11.6, while for IAA=40 it reached the highest value of 14).

Figure 3 presents the influence that the ignition advance angle has on the values of the indicated pressure for the combustion engine powered with CNG.
The analysis of Figure 3 shows that the maximum indicated pressure in a cylinder increases along with the growth of the ignition advance angle. The conducted tests demonstrated high scatter of the results in the interval beginning from the crankshaft revolution angle corresponding to the start of the combustion process (for the ignition advance angle of IAA=10, the beginning of the combustion process was observed at 364.9 CA), until the end of main combustion process [20], corresponding to fast movement of the combustion front across the combustion chamber, together with intense emission of heat and growth of the pressure in the cylinder (for the ignition advance angle of IAA=10, the end of the main combustion process was observed at 463.6 CA). The maximum indicated pressure was 2.415 MPa. The average torque at the engine’s shaft was 4.919 Nm. In the case of the ignition advance angle of IAA=20 before TDC, the beginning of the combustion process was observed for the angle of 356.9 CA. The end of the main combustion process was observed at 438.5 CA. The maximum indicated pressure was 3.693 MPa. The average torque at the engine’s shaft was 6.652 Nm. For the ignition advance angle of 30 before TDC, the beginning of the combustion process was observed for the angle of 347.8° CA. The end of the main combustion process was observed at 410.9 CA. The maximum indicated pressure was 5.913 MPa. The average torque at the engine’s shaft was 7.405 Nm. In the case of the ignition advance angle of IAA=40 before TDC, the beginning of the combustion process was observed at 343.2° CA. The delay of the ignition increased along with the growth of the ignition advance angle and for IAA=10 it was 14.9 , for IAA =20 it was 16.9 , for IAA=30 it was 17.8 , while for IAA=40 it reached the highest value of 23.2).

While observing the changes of pressure in the cylinder (Figure 3) it should be noted that as long as the changes of the pressure are to a great extent associated with charge compression, the process is much more repetitive than in the situation when the changes of pressure associated with combustion become dominant.

Figure 4 presents the influence that the ignition advance angle has on the values of indicated pressure for the combustion engine powered with biogas.
While analyzing figure 4, we have concluded that in the case of the ignition advance angle of $\text{IAA}=20$ before TDC, the beginning of the combustion process was observed for the angle of 352.8 CA. The end of the main combustion process was observed at 433.3 CA. The maximum indicated pressure was 4.329 MPa. The average torque at the engine’s shaft was 5.918 Nm. It should be stressed that incorrect combustion was observed in the case of some work cycles due to too late ignition as well as due to total absence of ignition of the fuel–air mix. In the case of the ignition advance angle of $\text{IAA}=30$ before TDC, the beginning of the combustion process was observed for the angle of 348.4 CA. The end of the main combustion process was observed at 405.2 CA. The maximum indicated pressure was 6.17 MPa. The average torque at the engine’s shaft was 7.415 Nm. For the ignition advance angle of $\text{IAA}=40$ before TDC, the beginning of the combustion process was observed for the angle of 345.6 CA. The end of the main combustion process was observed at 393.3 CA. The maximum indicated pressure was 6.884 MPa. The average torque at the engine’s shaft was 7.573 Nm. The delay of the ignition increased along with the growth of the ignition advance angle, and it was respectively: for $\text{IAA}=20$ – 12.8, for $\text{IAA}=30$ – 18.4, while for $\text{IAA}=40$ – 25.

In the case of biogas, the tests were not conducted for the ignition advance angle of $\text{IAA}=10$ due to the fact that the engine would not start in such circumstances. In the case of the ignition advance angle of $\text{IAA}=20$, we witnessed misfiring, which manifested itself by much lower, than in the other cases, values of indicated pressure.

Figure 5 presents the influence that the ignition advance angle has on the values of the indicated pressure for the combustion engine powered with E85.

The analysis of Figure 5 shows that the maximum indicated pressure in a cylinder increases along with the growth of the ignition advance angle.

In the case of the ignition advance angle of $\text{IAA}=20$ before TDC, the beginning of the combustion process was observed for the angle of 349.5 CA. The end of the main combustion process was observed at 428.3 CA. The maximum indicated pressure was 4.326 MPa. The average torque at the engine’s shaft was 8.94 Nm. For the ignition advance angle of $\text{IAA}=30$ before TDC, the beginning of the combustion process was observed for the angle of 343.8 CA. The end of the main combustion process was observed at 415.3 CA. The maximum indicated pressure was 6.03 MPa. The average torque at the engine’s shaft was 9.914 Nm. The delay of ignition increased along with the growth of the ignition advance angle (for $\text{IAA}=20$ it was 9.5, for $\text{S.A.}=30$ it was 13.8).

3.2. Graphs presenting the torque
This sub–chapter presents the results of the tests demonstrating the influence of the ignition advance angle on the achieved torque in the case of the combustion engine powered respectively with gasoline, CNG, biogas and E85.

Figure 6 presents the influence of the ignition advance angle on torque (Figure 6a) as well as on mechanical power (Figure 6b) for the combustion engine powered respectively with gasoline, CNG, biogas and E85.
Figure 6. Influence exerted by the ignition advance angle on: achieved torque (a), mechanical power of an engine powered with gasoline, CNG, biogas and E85 (b).

Analysis of Figure 6a demonstrates that the highest value of torque was achieved for the ignition advance angle of $\text{IAA}=30$ (gasoline – 10.38 Nm, CNG – 7.405Nm and E85 – 9.914 Nm) as well as for $\text{IAA}=40$ (biogas – 7.573 Nm). The highest value of mechanical power (figure 6b) was achieved for the ignition advance angle of $\text{IAA}=30$ (gasoline – 5.121 kW, E85 – 4.593 kW) and for $\text{IAA}=40$ (CNG – 3.658 kW, biogas – 3.73 kW).

3.3. HC and NOX emission graphs
This sub–chapter presents the findings related to the influence that ignition advance angle has on the emission of HC hydrocarbons and NOX in the case of a combustion engine powered respectively with gasoline, CNG, biogas and E85.

Figure 7 presents the influence of the ignition advance angle on the emission of hydrocarbons (figure 7a) and on the emission of nitrogen oxide (Figure 7b) in the case of a combustion engine powered with gasoline, CNG and biogas.

Analysis of figure 7a demonstrates that the highest value of emission of HC has been obtained for the ignition advance angle of $\text{IAA}=40$ (gasoline – 115 ppm, CNG – 144 ppm, Biogas –158 ppm). Emission hydrocarbons increased along with the growth of the ignition advance angle for all the examined fuels (Pb95 unleaded gasoline, CNG and biogas). The highest value of emission of nitrogen oxide (Figure 6b) was obtained for the ignition advance angle of $\text{IAA}=40$ (gasoline – 638 ppm, CNG – 299ppm, biogas – 189ppm). Biogas had the lowest NOX emission but due to the high CO2 content combustion was slowed down, and thus the combustion temperature was lower, resulting in the highest emission of hydrocarbons.

4. Conclusions
Based on the conducted tests, we have concluded that as the ignition advance angle increased, so did the maximum pressure in a cylinder during the gasoline combustion process (the maximum indicated pressure was 7.48 MPa for the ignition advance angle of $\text{IAA}=40$), CNG (the maximum indicated
pressure was 6.72 MPa for the ignition advance angle of IAA=40, biogas (the maximum indicated pressure was 6.884 MPa) and E85 (the maximum indicated pressure was 6.03 MPa for the ignition advance angle of IAA=30). Moreover, as the ignition advance angle increased, so did delay of ignition. For the ignition advance angle of IAA=40, the ignition delay was respectively: Pb95 – 14, CNG – 23.2, biogas–25.6. For E85, for the ignition advance angle of IAA=30, the ignition delay was 13.8.

The highest value of torque was achieved for the ignition advance angle of IAA=30 (gasoline – 10.38 Nm, CNG – 7.405 Nm and E85 – 9.914 Nm) as well as for IAA=40 (biogas – 7.573 Nm). The highest value of mechanical power was achieved for the ignition advance angle of IAA=30 (gasoline – 5.121 kW, E85 – 4.593 kW) and for IAA=40 (CNG – 3.658 kW, biogas – 3.73 kW).

Based on the research, it was concluded that growth of ignition advance angle is accompanied by both, growth of emission of hydrocarbons and nitrogen oxide in the case of all examined fuels. The highest value of emission of nitrogen oxide was recorded for the ignition advance angle of IAA=40 for gasoline – 638 ppm, with the lowest value seen in the case of biogas –189 ppm. In the case of biogas, due to the high CO2 content there occurred higher emission of hydrocarbons, which for the ignition advance angle of IAA=40 amounted 158 ppm. Growth of emission of nitrogen oxide, occurring together with the growth of the ignition advance angle, is undoubtedly caused by the occurrence of higher pressures, and consequently of higher temperatures during the combustion process, which is conducive to formation of nitrogen oxide. Moreover, as the ignition advance angle increases, the temperature of exhaust gases decreases, which directly translates to the lower temperature of the catalyst and to the lower efficiency of catalytic conversion (hence into higher emission of hydrocarbons (HC) as well as other toxic substances).

References

[1] Chmielewski A, Gumiński R, Mączak J, Radkowski S, Szulim P, Aspects of balanced development of RES and distributed micro–cogeneration use in Poland: Case study of a mCHP with Stirling engine, Renewable & Sustainable Energy Reviews, 60 (2016) 930–952.

[2] Milewski J, Szabłowski Ł, Kuta J, Optimal control strategy of NG piston engine as a DG unit obtained by an utilization of Artificial Neural Network, Power Engineering and Automation Conference (PEAM 2012), IEEE (2012) 410–416.

[3] Chmielewski A, Gumiński R, Radkowski S, Szulim P, Aspects of support and development of distributed microcogeneration in Poland, Rynek Energii, 5 (2014) 94–101.

[4] Milewski J, Szabłowski Ł, Kuta J, Control strategy for an internal combustion engine fuelled by natural gas operating in distributed generation, Energy Procedia, 14 (2012) 1478–1483.

[5] Szabłowski Ł, Milewski J, Kuta J, Badyda K, Control strategy of a natural gas fuelled piston engine working in distributed generation system, Rynek Energii, 3 (2011) 33–40.

[6] Chmielewski A, Gontarz S, Gumiński R, Mączak J, Szulim P, Research on a Micro Cogeneration System with an Automatic Load–Applying Entity, In Springer, Cham: Szewczyk R., Zieliński C., Kaliczynska M. (eds) Challenges in Automation, Robotics and Measurement Techniques. Advances in Intelligent Systems and Computing, 440 (2016) 387–395.

[7] Małecki A, Mydlowski T, Dybała J, Badania wpływu zanieczyszczeń biopaliw na sprawność silnika ZI (Research on the impact of biofuels pollutants on the SI engine efficiency), Zeszyty Naukowe Instytutu Pojazdów Pojazdów 99 (2014) 89–97. [In Polish]

[8] Sendzikienė E, Rimkus A, Melaika M, Makarevičiene V, Pukalskas S, Impact of biomethane gas on energy and emission characteristics of a spark ignition engine fuelled with a stoichiometric mixture at various ignition advance angles, Fuel, 162 (2015) 194–201.

[9] Chmielewski A, Gumiński R, Radkowski S, Szulim P, Experimental research and application possibilities of microcogeneration system with Stirling engine, Journal of Power Technologies, 95 (2015) 14–22.
[10] Kukharonak H, Ivashko V, Pukalskas S, Rimkus A, Matijošius J, Operation of a Spark–Ignition Engine on Mixtures of Petrol and N–Butanol, Procedia Engineering 187 (2017) 588 – 598.

[11] Chmielewski A, Gumiński R, Maczak J, Szulim P, Model–based research on a micro cogeneration system with Stirling engine, Journal of Power Technologies, 96 (2016) 295–305.

[12] Nunes de Faria M M, Vargas Machuca Bueno J P, ElmasIAAlami Ayad S M M, Pereira C R Belchior, Thermodynamic simulation model for predicting the performance of spark ignition engines using biogas as fuel, Energy Conversion and Management 2017 [In print].

[13] Chmielewski A, Gontarz S, Gumiński R, Maczak J, Szulim P, Research study of the micro cogeneration system with automatic loading unit, In Springer, Cham: Szewczyk R., Zieliński C., Kaliczyńska M. (eds) Challenges in Automation, Robotics and Measurement Techniques. Advances in Intelligent Systems and Computing, 440 (2016) 375–386.

[14] Małecki A, Chmielewski A, Mydłowski T, Gumiński R, Dybała J, Silniki spalania zewnętrznego w układach mikrokogeneracji (External combustion engines in micro-cogeneration systems), Zeszyty Naukowe Instytutu Pojazdów 98 (2014) 147–156 [In Polish].

[15] Mydłowski T, Nader S, Biskup K, Jasiński M, Wykorzystanie urządzenia ECU Master EMU do sterowania silnikami z zapłonem iskrowym (Using the ECU Master EMU device to control spark ignition combustion engines), Zeszyty Naukowe Instytutu Pojazdów, 86 (2011) 125–129 [In Polish].

[16] Małecki A, Mydłowski T, Dybała J, Stanowisko hamowniane do badań silników spalinowych o małych mocach (The engine dynamometer to test internal combustion engines with low power), Zeszyty Naukowe Instytutu Pojazdów, 96 (2013) 55–66 [In Polish].

[17] Dybała J, Mydłowski T, Małecki A, Bogdziński K, Dynamometer and test stand for low power internal combustion engine, Combustion Engines, 162 (2015) 996–1000.

[18] Małecki A, Mydłowski T, Radkowski S, Przegląd uniwersalnych sterowników do silników ZI (Review of programmable electronic fuel injection controllers), Zeszyty Naukowe Instytutu Pojazdów, 93 (2013) 93–101.

[19] Pham P X, Vo D O, Jazar R N, Development of fuel metering techniques for spark ignition engines, Fuel, 206 (2017) 701–715.

[20] Rychter T, Teodorczyk A, Teoria silników tłokowych (The theory of piston engines), Wydawnictwo Komunikacji i Łączności WKŁ, 2013 [In Polish].