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Original Research Article

Determination of the injector type and location for a direct injected Wankel engine

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ABSTRACT

Direct injection on the Wankel engines has been practiced since the 1970's. By applying direct injection in the Wankel engine, the specific fuel consumption and HC emissions, which are seen as disadvantages for this engine, might be reduced and the mixture formation is improved. In order to obtain a better mixture formation, the fuel injector must be located in a proper place and a correct direction. In addition, the most suitable injector for the engine structure should be selected. In this study, direct injection was applied for the RENESIS 13B Wankel engine and injector selection for the engine and the location of the injector on the housing were examined. In addition, the diameter of orifice and flow rate characteristics of the injector were investigated. According to the results, two types of injectors were selected. These injectors were called as low speed and high speed. An injector with a narrow nozzle angle (<30°) was used to deliver the fuel to the leading of the chamber, and the fuel injected after the intake ports closed. The orifice diameters of low-speed and high- speed injectors were measured 0.33 and 0.45 mm, respectively. In addition, both injectors have low speed rates when injection duration was below 2 ms. This flow rate was increased by providing high voltage.

Keywords: Injector selection; Injector flow rate; Orifice diameter; Direct injection; Wankel engine.

1. Introduction

The main advantages of the Wankel engine might be listed as the low number of moving parts and vibration, the high power obtained from the unit volume and the lightness. With these qualities, it becomes a promising propulsion option in newly developing applications such as unmanned aerial vehicles in aviation and range extenders in hybrid vehicle systems [1, 2]. In addition, due to its different features in its structure, it provides superiority over conventional engines in the use of certain alternative fuels (such as hydrogen, H_2, and natural gas, CH_4). However, despite the mentioned positive features, the Wankel engine generally lags behind the level reached by today's piston engines in terms of fuel consumption, hydrocarbon (HC) emissions and engine life [3].

One of the main objectives of this study is to reduce the specific fuel consumption and HC emissions to a lower level by applying direct injection in a Wankel engine. Many studies have been carried out on direct injection application. Peden et al. [4] used a 225CS model, 225 cm^3 single-rotor Wankel engine for unmanned aerial vehicles developed in England. A single dimensional model to this engine was created in the AVL BOOST software and verified with the results obtained from the model. However, this verification was made with a homogeneous mixture engine and does not include direct injection. Within the scope of the study, the computer model was used to analyze the relationships between injector position, injection pressure and engine speed and to see their effects on engine performance. The best injector position was obtained by placing it on the relatively colder part of the housing. The engine was operated at full load and between 5000 and 7500 rpm. It has been observed that fuel consumption and engine torque were improved at 4.5 bar injection pressure under all operating conditions. In addition, the best results in engine performance were obtained when the injector sprays at an eccentric shaft angle of 597° (Ignition 483° before TDC). Roberts et al. [5] developed two different thermodynamic models for a 573 cm^3 two rotor Wankel engine, both of manifold injection and direct injection. Results of model was close to the experimental data at 2000 rpm, 66% throttle opening and lambda: 0.83 (Excess air coefficient) conditions. In addition, losses were observed by calculating the crevice volume and leakage with this modeling. A similar study was developed by Nguyen et al. [6] for a 648 cm^3 single rotor direct injection stratified charge Wankel engine used in boats. As a result, it has been stated that better combustion and higher performance will be achieved if the leakages were reduced in the stratified charge Wankel engine and the burning speed was increased with the application of turbocharging in the modeling. Ji et al. [7] numerically investigated using CONVERGE software, which the effects of different injection positions and injection angles on the mixture formation and combustion process in a 654 cm^3 volume, side-port and direct injection Wankel engine. In the analysis, the rich mixture formed in the combustion chamber came from the rear region to the front region as the spray position approached the TDC and the spray angle increased. Spray locations were designated as low (30 mm before BDC), medium (BDC), and high (30 mm after BDC). High position injection strategies show that the mass fraction (fraction) of the fuel is burned above 99% and is suitable for complete combustion. Maximum cylinder pressure increased by 12.39% with 0° spray angle and high spraying position.

Abraham et al. [8, 9] analyzed using Computational Fluid Dynamics (CAD), which the injection strategy and bowl geometry in a 663 cm^3 cylinder volume, direct injection stratified charge and peripheral port engine. The fact that the cavity on the rotor is ahead of the direction of movement results in higher cylinder pressure and higher combustion speed and rate closer to the TDC than other geometries. In addition, thanks to pilot injection, the thermal efficiency increased by 6-8% and a two-hole pilot injector gave better results than other spraying methods [8, 9].

Chen et al. [10] aimed to increase the combustion efficiency of a direct injection diesel rotary engine (DI-DRE) by adjusting the injection strategy. ANSYS Fluent software was used in the numerical study. In the study, a single rotor and peripheral port engine with a cylinder volume of 648 cm^3 was used. The effects of injection time and angle on the air-fuel mixture and combustion process were investigated. The fuel distribution in the chamber was realized in a narrower and more concentrated manner due to the delay of the injection time and the increase of the injection angle. It was observed that the combustion efficiency increases due to the formation of this fuel density in the rear region of the combustion chamber. When the optimum angle was 90° and the injection time was 80° before the TDC, the maximum cylinder pressure increased by 16.82%, soot decreased by 41.48% and CO by 32.87%, while NO and CO_2 emissions slightly increased. Hamadaty et al. [11] investigated the flow properties of the kerosene/air mixture using a high-speed camera in a direct-injection rotary piston engine. it was stated that the effect of turbulence intensity on the formation of swirl in the initial process of combustion is great, and also, injection parameters are critical because large fuel droplets can cause incomplete or delayed combustion. Votaw [12] investigated the effects of pilot injection parameters such as injection angle, injection position and injector retraction distance on a direct injection rotary piston engine with ANSYS Fluent software. In the study, AR747 model rotary piston engine with 208 cm^3 volume used in UAVs (Unmanned Aerial Vehicles) was selected. As a result, it has been seen that the engine can achieve ideal combustion characteristics when the injection angle is 65° and the injector retraction distance is 6.6 mm. Boretti et al. [13] modeled the RENESIS 13B Wankel engine using GT-SUITE software. In the study, both spark plug slots were kept wide and jet ignition (spark plug and injector side by side) and direct injection were compared with port fuel injection (PFI). Gasoline and hydrogen were tested as fuels. The engine was operated from 1000 rpm to 10000 rpm with 1000 rpm intervals at the full load. Compared to the PFI system, the combustion speed was increased and it became more suitable for complete of combustion. Maximum pressure and temperature in the cylinder increased.

Meyer and Shoemaker [14] designed a prototype high pressure common rail fuel injection system and tested it on a single and two rotor rotary engines. The highlight of the system is the achieved high injection frequency (8000 injections/min/injector) and the use of a pilot injector. HSEFI-(High Speed Electronic Fuel Injection) fuel injection system is provided with a pump line-nozzle-gearbox at full load to obtain optimum injection time and lower fuel consumption. Hasegawa and Yamaguchi [15] designed a transparent single-rotor engine to observe the air-fuel mixing process and the flow fields from intake to compression stroke of a side-port 13B Wankel engine. An injector was placed in the intake and compression area and five different combustion chamber geometries were used. Setting the recess close to the L side creates a counterclockwise vortex on the L (Leading) side instead of the clockwise swirl that dominates the center of the combustion chamber. It has been stated that if the fuel is fed in this vortex, engine stratified charge is achieved.

Taskiran [16] investigated the early direct injection situation using CFD techniques in a side port RENESIS 13B Wankel engine. The fuel was injected inside by placing a fuel injector, which was assembled into the oil injection hole on the housing. Simulations were made at the part-load condition and 2000 rpm. The results showed that the swirl-like motion prevented the fuel jet (spray) from accumulating in the middle of the combustion chamber. Fuel droplets were thrown to the opposite side of the inlet wall by the centrifugal force of the inlet air. This effect decreases as the swirl flow decreases due to the sweeping action of the rotor. In another study [17], it was stated that the use of DISC (Direct-Injection Stratified-Charge Rotary Engine) provided significantly improved fuel consumption and lower exhaust emissions. The use of the exhaust gas recirculation system was investigated and found to be beneficial in NO_x reduction. Muroki et al. [18] studied the DISC-RE with pilot flame ignition system to find the possibility of reduction in fuel consumption and HC exhaust gas emissions. As a result, it was determined that a more active and better ignition was obtained in leaner mixtures than the spark ignition system. Fan et al. [19] numerically investigated the effect of injection timing (IT) and injection angle (IA) on mixture formation and combustion process of a direct-injection natural gas Wankel engine. The simulation results emphasized that the injection angle should be chosen rather small when the injection is made at the intake stroke to ensure high combustion rate and ideal fuel distribution, and it should be a larger injection angle when the injection is made at the compression stroke. Ochere et al. [20] used a 648 cm^3 peripheral ported Outboard Marine Corporation (OMC) in a direct injection rotary engine (DIRE). The numerical study was carried out on Ansys Fluent. Injection time and ignition time were investigated in a direct injection Wankel engine. Optimum combustion process was obtained in terms of emissions when injection timing is 80°CA (BTDC) and spark time is 35°CA (BTDC). In another study, Ochere et al. [21] examined the effect of ignition advance on the combustion process by using biodiesel fuel in the same engine. The simulation results showed that properly advancing injection timing increases the in-cylinder pressure and the peak pressure. Many studies have been done in the literature on direct injection in a Wankel engine [22-24].

As can be understood from the literature studies, most of the studies on direct injection in the Wankel engine are numerical analysis (One-dimensional models and many software such as Ansys Fluent, Converge, AVL Boost and GT-Suite). In the studies, the position of the injector on the housing, injection pressure, injector type, air-fuel mixture process, flow fields were examined in different rotary engines. Optimum injection process was investigated by examining parameters such as power, fuel consumption, thermal efficiency, combustion characteristics and exhaust emissions. Experimental studies on the selection of injectors and the determination of the injector location on the housing are very limited in the literature. In this study, the injector selection criterions for the Wankel engine and the location of the injector on the housing were investigated. Two types of injectors were selected. These are Bosch - 0 280 158 038 injector for low speed/loads and Bosch - 0 280 158 040 injector for high speed/loads. In addition, the measurement of orifice diameter and flow rate characteristics of the selected injector were tested.

2. Experimental Study

2.1. Injector location

As could be obtained from the literature, location of the injector plays an important role in mixture form. Besides, there is another important point in injector location arising from Wankel engine unique geometry; the hole drilled on the housing body of the engine must not have large diameter in the areas that the pressure difference between the chambers is high; which causes leakage between chambers. Fig. 1 shows the pressure difference between the chambers [25].

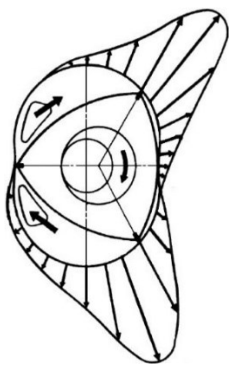

Figure 1. Pressure difference between the combustion chambers [25]

As shown in Fig 2, the injector location was selected. This zone is in a position after the intake ports and it could be a safe place in terms of leakages. Because the pressure difference is not much high at this region when the rotor apex passing from here. For instance, at the operating conditions of 3000 rpm and at a mean effective pressure of 3 bar, while one of the chambers is in intake stroke and the inside pressure is 0.5 bar, the other chamber is at the first of compression stroke and the inside pressure is 1.2 bar. Thus, the pressure difference was 0.7 bar between this two region and this value is not a large number causing to leakage through the injector recess.

Another advantage of this region is that the cylinder pressure is low in this region. The chamber pressure in the around this area is just like intake manifold and it is approximately atmospheric pressure. So, there is no need to use high pressure direct injection injectors to deliver the fuel in this area and normal PFI injectors could be used here thanks to not having high back pressure.

But, there is enough time and space for the fuel to mix homogeneously with the air in this location if the injector spread the fuel widely. For this purpose, an injector with a narrow spray angle (<30°) should be selected to send the fuel to the leading side (Fig. 2). Otherwise, there is a possibility that the fuel will not mix with the air or reach the sidewalls. In our previous study on the engine control unit, according to the results obtained with the oscilloscope and high speed camera, there was no delay in opening time of the injector with the solid state relay. However, there was 0.6 milliseconds delay in closing time of injection. This situation caused the engine to run unstable at high speeds. In the circuit established with transistor (mosfet), there was no delay at the opening and closing times of the injector [26].

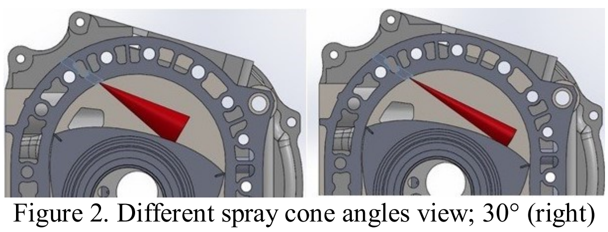

Figure 2. Different spray cone angles view; 30° (right) and 10° (left)

2.2. Selection of injectors

In choosing a proper injector for this engine, the injector, which is used in direct injection system, must have a high speed rate to inject enough amount of fuel. For example, a PFI injector, which was used in Mazda RX8 engine

(Fig. 3), has a flow rate of 420 cm^3/min at a line pressure of 3.0 bar.

Figure 3. Mazda RX-8 original PFI injectors (Denso)

Another criterion in selection of an injector is its dimensions. There are cooling water jackets surrounding the housing. The size of the injector should be small enough to not having interference with neither coolant jacket nor cross screw (Fig. 4). The injector should be slim so injectors with extension were selected (Fig. 5). In addition, another criterion of the injector selection is spray angle. The spray angle of the injector should be small to prevent a contact between fuel and walls. However, some of the MPI injectors have dual cone spray shape in order to use in 4 valve per cylinder engine in which the fuel is sent for every intake valve separately (Fig 6), which is not proper for this investigation.

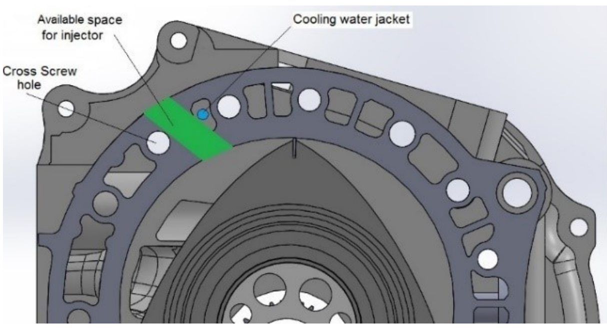

Figure 4. Available space for placing injector

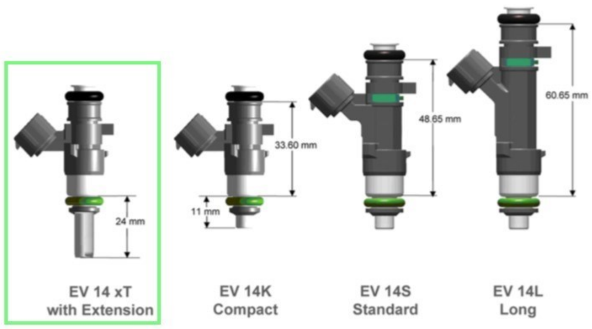

Figure 5. Different injector constructions for Wankel engine

There are some injectors that have criterions, which are mentioned above selected among the Bosch EV 14 injectors. These injectors are shown in Fig 7. The first injector (Bosch - 0 280 158 038) has enough flow rate for low-middle speed/loads and the second one (Bosch - 0 280 158 040) has enough flow rate for higher speed/loads. So, these two injectors are selected. As shown in Fig 8, firstly, a proper place was selected to drill injector hole on the housing, by having the injector dimension.

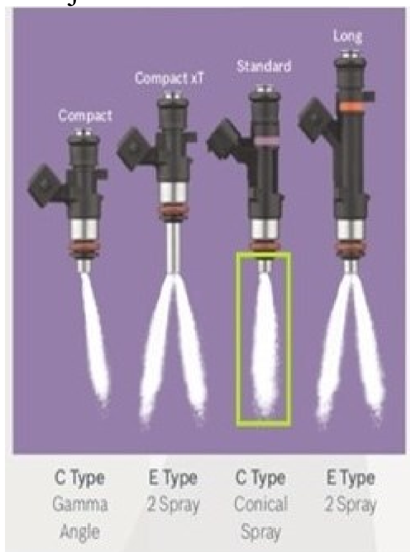

Figure 6. Selection of the proper spray type and spray cone

Figure 7. Selecting direct injection injectors for Wankel engine [3]; 347cc (up), 980cc (down)

3. Results and Discussion

3.1 Preliminary injector tests

Although each injector has technical specifications in its datasheet, to determine the accurate characteristics of them, some measurements and experiments were done in to calculate the orifice diameter, flow rate and spray geometry.

3.1.1. Measurement of orifice diameter

In order to estimate the spray penetration of injectors, it was needed to measure the orifice diameter of the injectors. By taking close up picture the injector tip, it is possible to see the orifices clearly. Orifice diameter could be measure as counting the quantity of pixels (Fig. 9).

The outer diameter of both injector was measured 6.7 millimeters. By calculating the ratio of the pixel number, the diameters were obtained 0.33 and 0.45 millimeters for low speed and high speed injectors, respectively.

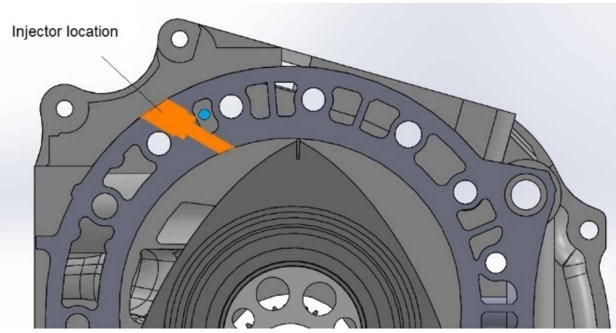

Figure 8. Location of the direct injection injector

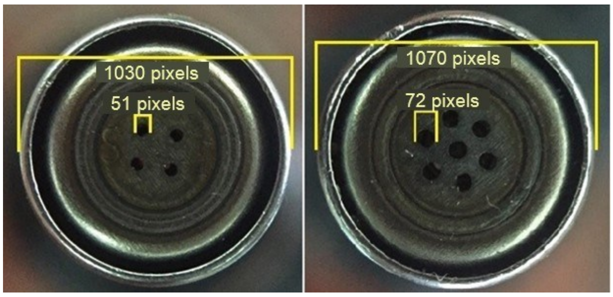

Figure 9. Determination of orifice diameters of injectors (low speed, left, high speed, right)

3.1.2. Flow rate characteristics

In datasheet of each injector, there is information about the flow rate; but, to ensure the accurate flow rate characteristics of each injector, some experiments were conducted in different injection durations and pressures. For low speed injector the flow rate diagrams are shown in Fig. 10 and Fig 11. In general, the mass flow rate remained constant at injection times after 2 ms (Fig. 11). It has been experimentally determined that the mass flow rates of the two different injectors selected are quite different.

As seen in Fig. 11, both injectors have very low mass flow rates when the injection duration was below 2 ms. This is due to not having enough time for mechanical (Solenoid-Spring) system to react for opening and closing the injector. In order to increase the reaction speed, we increased the voltage of power supplier (Fig. 12 and Fig 13).

As seen in by increasing the injector voltage, the amount injected of fuel increases. This is due faster response of solenoid and shortening the injection duration. The increase of voltage is effective on mass of the injected fuel when injection duration is below the 2ms. However, the effect of voltage change is negligible in comparison to total amount of injected fuel as the injection duration increased.

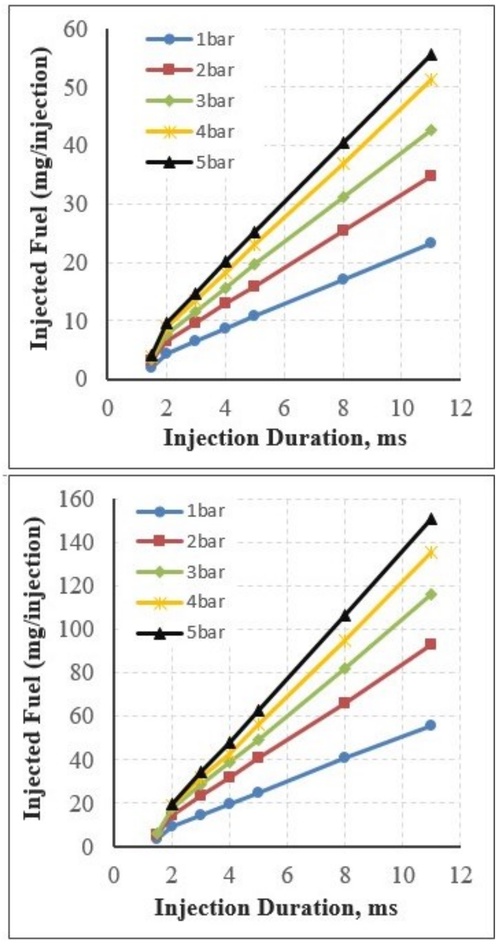

Figure 10. Injected fuel mass of the low speed (up) and the high speed (down) injectors

3.1.3. Spray geometry

A high-speed camera (Photron Fastcam 512 PCI) was used to measure the injector spray geometry. The speed of the camera was set at 4000, 8000 and 16000 frame per second. The lower speeds were used to capture the spray angle of injector because the camera could take wider pictures at lower frame speeds. Higher speed was used for measuring the spray tip penetration, injection delay and observing the small droplets.

Injection pressure was set on 5.0 bar relative to the atmospheric pressure (6 bar absolute) in all experiments. Schematic map for camera setup was given in Fig 14. The trigger of camera and injector driving unit (Solid state relays) were connected in a parallel line, which is controlled by Arduino (5v digital output). In this way, the injector and the camera were operated simultaneously. In addition, the delay time in opening the injector was also determined.

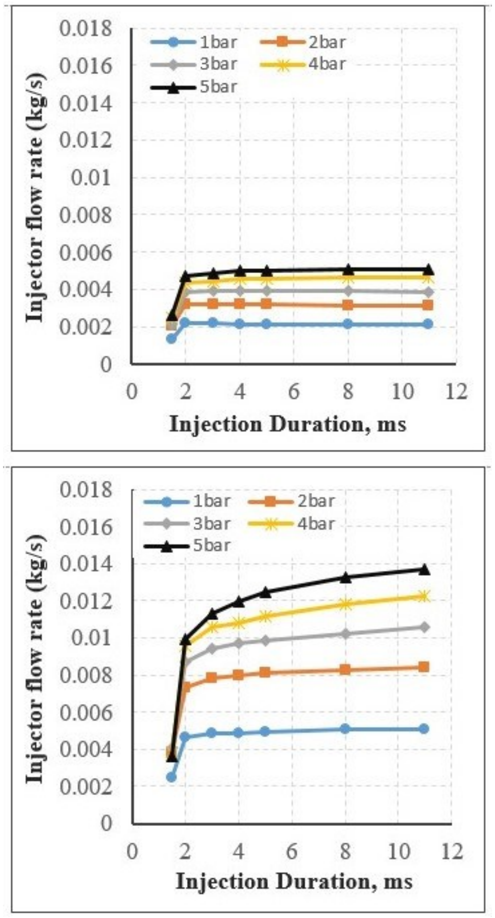

Figure 11. Mass flow rate of selected injectors, low speed (up) and high speed (down)

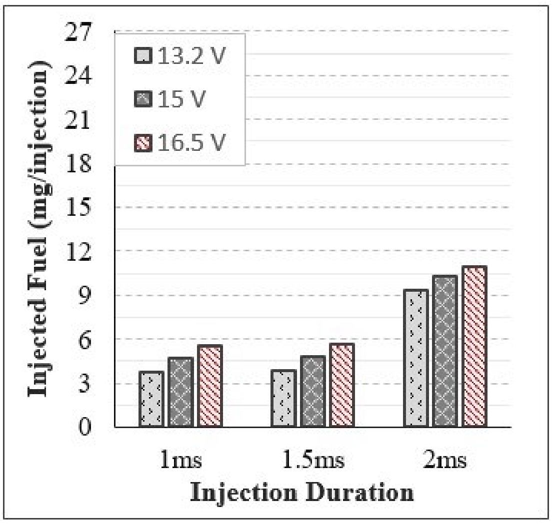

Figure 12. Effect of voltage changing on the injected fuel for low speed injector

3.1.3.1. Spray tip penetration

The penetration of injector spray is needed to determination of spray geometry. This parameter shows how far the injected fuel travels. Penetration is defined as the distance of spray tip from the injector nozzle at the time that the injection ends. The results are shown Fig. 15 for low speed injector and Fig. 16 for high speed injector.

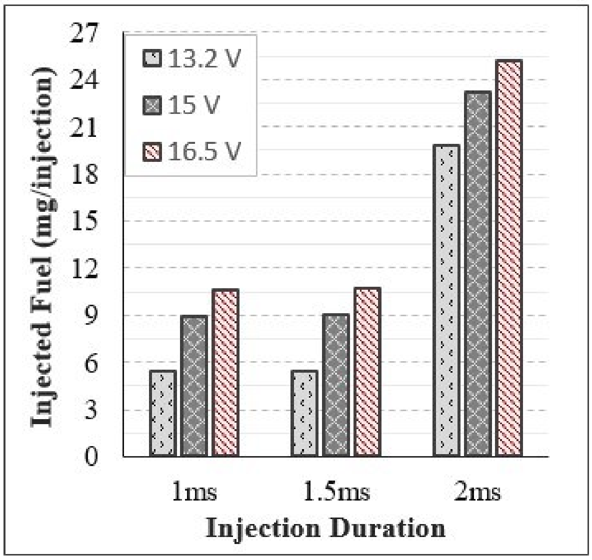

Figure 13. Effect of voltage changing on the injected fuel for high speed injector

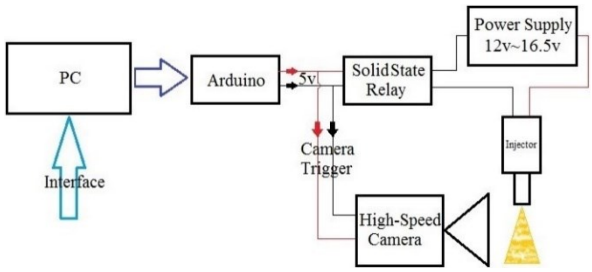

Figure 14. Schematic view of camera and injector test systems

If we analyze the penetration results, in Hiruyasu's equation [27], the penetration length is divided in two sections; before and after break-up. The equation 1.1 and 1.2 for describing the spray penetration is more realistic. When the spray first injected into the chamber, the initial velocity of the spray tip is much larger than the surrounding air. According to Hiroyasu and Arai [27], at this zone the penetration is described by an expression proportional to time, t, in other words; it has constant velocity. At the next time step, when the spray velocity slows downs, the penetration at this zone is proportional to square root of time. Hiroyasu and Arai [27] describe the distinguishing time between these two time steps as breakup time [28]:

According to Hiruyasu's equation, for $t < t_b$, the penetration could be calculated from:

$$S = 0.39\left(\frac{2\Delta P}{\rho_L}\right)^{0.5} t \tag{1.1}$$

and for $\gg t_b$:

$$S = 2.95 \left(\frac{\Delta P}{\rho_G}\right)^{0.25} (d_n t)^{0.5} \qquad (1.2.)$$

and break-up time could be calculated from:

$$t_b = \frac{29 \rho_L d_n}{\rho_G^{0.5} \Delta P^{0.5}} \qquad (1.3.)$$

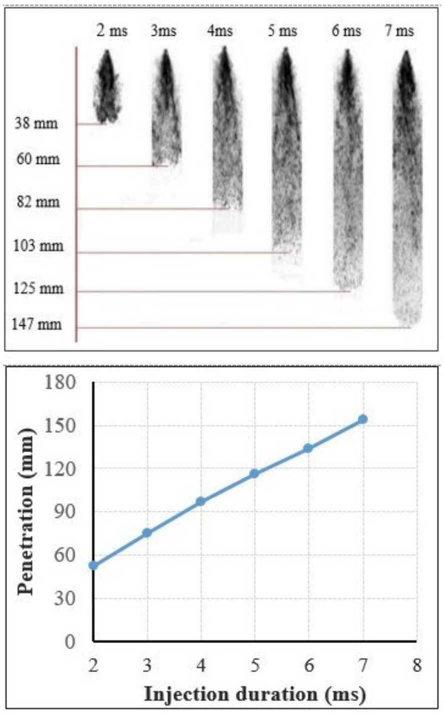

Figure 15. Penetration of low speed injector in different injection durations

As could be seen from the equations, the penetration increases linearly by increasing the time before the break-up and after that increases by square root of time. For our injectors, the break-up time could be calculated with the injection and ambient pressure and injector nozzle diameter. The break-up time will be obtained about 9.2 and 12.5 milliseconds for low speed and high speed injectors, respectively. Therefore, when ınjection duration is below these values, the penetration changes linearly with time, which was happen in this study.

3.1.3.2. Spray angle

Lower frame rates of camera (4000 fps) was chosen in order to have wider view of spray to measure the spray angle (Fig. 17). After converting the spray photos to negative mode and neglecting splashed droplets, the main spray cone and its angle could be seen in Fig. 18. In this figure, upper side picture shows the measurement of spray angle for low speed injector and down side picture shows measurement for high speed rate injector. By the way, the spray angles were obtained 14.1° and 25.8° for low speed and high speed injectors, respectively.

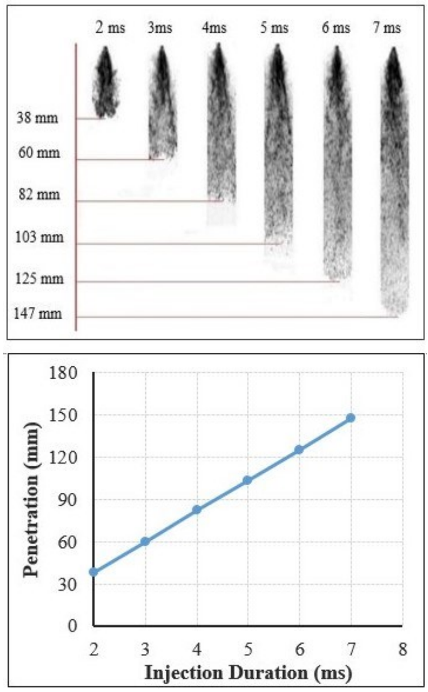

Figure 16. Penetration of high speed injector in different injection durations

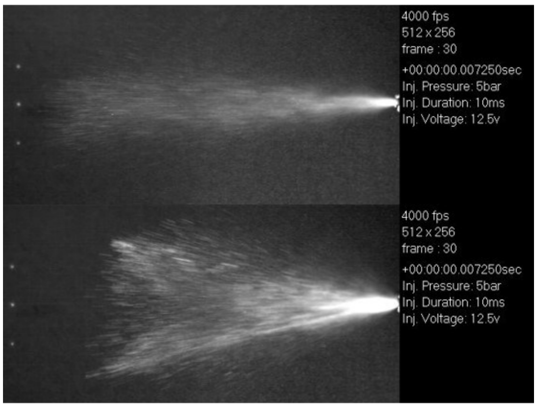

Figure 17. Spray angle of the low speed (up) and high speed (down) injectors

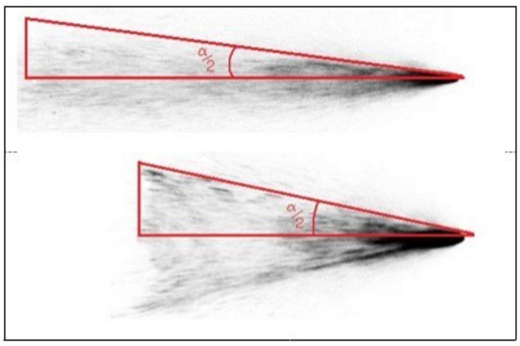

Figure 18. Determination of spray angle for low speed (up) and high speed (down) injector

4. Conclusions

The position of the injector is very important for the formation of the mixture. The pressure difference must not be high between the chambers due to the leakages. The location of the injector was positioned towards the end of the intake stroke. This region is right after the intake ports and could be a safe place in terms of leakages. Because the pressure difference is not much high at this region when the rotor apex passing from here. An injector with a narrow nozzle angle (<30°) had used to deliver the fuel to the leading of the chamber. The dimension of the injector was chosen so that neither the coolant jacket nor the cross screw is obstructed. Two types of injectors were selected. These are Bosch - 0 280 158 038 injector for low speed/loads and Bosch - 0 280 158 040 injector for high speed/loads. Some measurements and experiments were carried out to calculate the nozzle diameter, flow rate and spray geometry of the injectors. The orifice diameters of low-speed and high- speed injectors were calculated 0.33 and 0.45 mm, respectively.

When the injectors were tested in terms of injection duration, both injectors were found to had low speed rates below 2 ms. This is because the mechanical (Solenoid-Spring) system does not have enough time to react to open and close the injector tip. The voltage of the power supplier was increased to overcome this problem. As a result, it was concluded that both injectors can be used. In the continuation of the study, these injectors might be tested in the Wankel engine at different load and speeds. In addition, a new research should be conducted with piezo injectors for Wankel engines.

Nomenclature

ABDC	: After bottom dead center
BDC	: Bottom dead center
BTDC	: Before top dead center
CFD	: Computational fluid dynamics
CO_2	: Carbon dioxide
DI-DRE	: Direct injection Diesel rotary engine
DISC	: Direct injection stratified charge rotary engine
IA	: Injection angle
IT	: Injection timing
HC	: Hydrocarbon
L	: Leading
MPI	: Multi point injection
NO	: Nitrogen oxide
PFI	: Port fuel injection
T	: Trailing
TDC	: Top dead center
UAVs	: Unmanned aerial vehicles
λ	: Excess air coefficient

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CRediT authorship contribution statement

Majid Javadzadehkalkhoran: Materials and Method, Experimental Study, Results and Discussion, Injector tests, Investigation and Validation.
Osman Akın Kutlar: Conceptualization, Data curation, Materıals and Method, Spray tip penetration, Spray angle, Conclusions and Supervision.
Ömer Cihan: Introduction, Experimental Study, Results and Discussion, Conclusions, Conceptualization, Supervision and Writing-Reviewing.
Hüseyin Emre Doğan: Preliminary Injector Tests, Results and Discussion, Injector tests, Conclusions, Conceptualization, Data curation and Writing- Reviewing and Editing, Validation and Supervision.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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