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Hydraulic characterization of high-pressure gasoline multi-hole injector

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Abstract

Gasoline direct injection (GDI) system are becoming a popular technology in spark ignition engines to harvest the advantage of both higher thermodynamic efficiency and less engine-out emissions. The mixture formation in a direct injection engines is an outcome of careful matching of the air-fuel mixture and accurate fuel injection process. The injection rate is a vital parameter in understanding the injector parameters, spray characteristics, and in-turn its effect on air-fuel mixture and combustion process. Therefore, this study is focused on developing an injection rate rig based on momentum flux method and then measure the injection rate and characterize the high-pressure GDI injector custom made for research purpose. The injection rate was measured over a wide range of injection pressure and total mass injected. The injection rate measurement was complemented with high-speed imaging to correct the hydraulic start and end of injection to the true value. Then the hydraulic delays and discharge coefficient were characterized for this GDI injector.

Keywords: Momentum flux, Injection rate, Hydraulic delay, Discharge coefficient

Highlights

- Injection rate measurement rig was commissioned based on momentum flux method.
- Injection rate of high-pressure multi-hole GDI injector was measured.
- Hydraulic delay during start and end of injection were characterized.
- Transient discharge coefficient was measured and reported.

1. Introduction

Gasoline direct injection (GDI) technology is not certainly novel. Hesselman engine introduced in 1925 was the first known application to use GDI technology [1]. It indeed wasn't the same technology that we use today but had the foundations of this technology. With increasing, stringent emissions legislation norms and demand for more efficient automotive engines, GDI systems in combination with turbocharging

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and engine downsizing are becoming popular in gasoline engines segment [2]. The emissions produced depends on the engine design, power output, and working load. The complete combustion of fuel reduces the harmful exhaust emissions. For direct injection engines, complete combustion is a result of accurate fuel injection process and careful matching of the air-fuel mixture [3]. Furthermore, more precise control of fuel metering is vital for operation in lean mixture mode for GDI engines. There is also a need for complete mixture preparation within short time between inlet valve opening and ignition triggering by the spark [4]. For these reasons, the injection rate is a crucial parameter to understand the injector performance, spray characteristics, air-fuel interaction and the combustion process. Therefore understanding the injection rate and hydraulic characteristics of the injector will lead to a significant improvement in design and optimization of performance and emission characteristics of the direct injection engines.

There are various methods used for injection rate measurement including Bosch method [5], Zeuch method [6], revolving disc method [7], electric charge measuring method [8], deformational measuring method [9] and momentum flux method [10]. In Bosch method, the injection rate is measured by the pressure wave generated by injecting the fuel into the tube filled with same fuel maintained at constant pressure using a check valve. On the contrary, the Zeuch method uses the measurement of pressure rise rate inside a closed chamber when fuel is injected to calculate the injection rate using the fuel density. A very comprehensive comparison was done by Bower and Foster [11] on Bosch and Zeuch measurement systems and concluded that both methods measure the magnitude and shape except that the Bosch method produces a much smoother injection rate shape. Naber and Siebers [10] used the momentum flux method to measure the injection rate of diesel injector for the first time. The underlying assumption in this approach is that the injection rate follows the same profile as that of the momentum flux. Later, this technic was adopted by various researchers to measure the mass and momentum flux together for diesel injectors [12, 13, 14, 15, 16, 17, 18]. The momentum flux method has a clear advantage over other methods regarding injection rate and transient hydraulic coefficients measurement from each nozzle holes to study the hole-to-hole variations [19, 20, 21].

Recently, momentum flux method was used to measure the rate of momentum for GDI injectors [22, 23, 24]. Payri et al. [22] used momentum flux method to measure rate of momentum and Bosch method to measure the injection rate. From rate of momentum measurement, the hydraulic parameters such as area, velocity and discharge coefficient were characterized for GDI injector which enables to understand the internal nozzle flow and establish experimental validation for modeling gasoline injectors. Furthermore, at low back pressure conditions, both lateral and frontal configurations show negligible difference in the momentum flux. Postrioiti et al. [23] used momentum flux method to measure rate of momentum and Zeuch method to measure injection rate of single hole research GDI injector. They did a parametric study on rate of momentum and injection rate by varying injection pressure, ambient pressure and fuel temperature under standard flow and flash-boiling conditions. They reported that flash-boiling onset does not affect the global momentum flux measured by the impact force. Therefore, this method can be successfully used to measure rate of momentum of GDI injectors even under flashing flows. Cavicchi et al. [24] used the same

measuring equipment as in [23] and combined the experimental rate of momentum and injection rate for different parameters such as target size, nozzle/ target distance, ambient pressure with numerical simulations. They confirmed that the combined use of numerical and experimental approaches could help to obtain an adequate knowledge of the spray evolution of GDI injectors. Though recently momentum flux method has been applied to GDI injectors, it was used only to measure the rate of momentum but not injection rate. A recent numerical study by Ge et al. [25] compared the injection rate profiles from Zeuch and momentum flux method on liquid and vapor penetration lengths. It was concluded from their research that the injection rate profile measured by momentum flux method predicts liquid and vapor penetration lengths in good agreement with the experimental results.

In the present study, the momentum flux method is used to measure the injection rate of GDI injector at high injection pressure. First of all, to achieve this, an injection rate measurement rig based on momentum flux method was constructed. Then the proposed measuring method was validated, and the GDI injector was hydraulically characterized using injection rate, hydraulic delays, and discharge coefficient under conditions over a range of injection pressures and total fuel mass injected.

2. Theoretical background

The rate of injection ($\dot{m}_f$) is defined as the instantaneous mass of fluid with density (ρ_f) flowing out through a nozzle hole of the cross-sectional area (A_0) with a mean velocity of (u_m) and is mathematically given as in Eq.1.

$$\dot{m}_f = \int_{A_0} \rho_f u_m dA \quad (1)$$

The momentum flux ($\dot{M}_f$) is defined as the mass flow rate times the velocity and is mathematically denoted as in Eq.2.

$$\dot{M}_f = \int_{A_0} \rho_f u_m^2 dA \quad (2)$$

By the conservation of momentum, the momentum flux ($\dot{M}_f$) is equal to the force (F) measured by the sensor.

$$F = \dot{M}_f \quad (3)$$

From Eqs.1, 2 and 3, the rate of injection can be calculated as

$$\dot{m}_f = \sqrt{F \rho_f A_0} \quad (4)$$

The cumulative mass m_f is then calculated as follows

$$m_f = \int_{HSOI}^{HEOI} \dot{m}_f dt = \int_{HSOI}^{HEOI} \sqrt{F \rho_f A_0} dt \quad (5)$$

where $HSOI$ is the hydraulic start of injection and $HEOI$ is the hydraulic end of injection.

The coefficient of discharge is given as follows

$$C_d = \frac{\dot{m}_f}{A_0 \sqrt{2 \rho_f \Delta P}} \quad (6)$$

The mean velocity at the outlet of the nozzle hole is given as follows

$$u_m = C_d \sqrt{\frac{2 \Delta P}{\rho_f}} \quad (7)$$

where $\Delta P = P_{inj} - P_{amb}$, P_{inj} is the fuel injection pressure and P_{amb} is the ambient pressure.

3. Experimental setup

Fig 1 shows the schematic of the experimental set-up commissioned at KAUST Spray Lab. The experimental setup consists of momentum flux measurement apparatus and fuel injection system. In the following sections, a brief description of the experimental rig and fuel injection system are presented.

3.1. Fuel injection system

The fuel injection system used in this work is common rail system consisting of a commercial common rail, solenoid multi-hole gasoline injector and compressed air driven liquid pump. The injector used in this study is a Bosch custom made 10 hole gasoline high pressure injector (HDEV5s) provided by Saudi Aramco for research purposes. The injector hole size is $165 \mu m$, and cone angle is 110° . This injector was mainly designed for high pressure gasoline direct injection applications. This injector is capable of injection pressures up to 500bar. The common rail used is a commercial gasoline common rail with a maximum pressure of 1500 bar. The accumulation pressure is precisely controlled by a pressure relief valve and a pressure transducer fitted to the ends of the rail. The compressed air driven liquid pump (Hydraulic International, Inc. 5L-SD-410) is used to pressurize the fuel to high pressures of up to 3000 bar. The inlet drive pressure of the pump is between 0-150 psi. The common rail and solenoid injector are controlled using DI driver system (National InstrumentsTM DIDS-2003) through a custom written LabVIEWTM code. The fuel used in this study is commercial E10 gasoline. The properties of the fuel used is given in the Table 1.

3.2. Momentum flux measurement rig

The spray momentum flux measurement rig consists of a chamber which can be pressurized to a maximum of 150 bar. The chamber has a provision to fit injector on one of the sides and a force sensor on the opposite side. Furthermore, it provides optical access through two windows of size $\phi 25$ mm orthogonal to the injector and sensor. Quartz glass of diameter 35 mm and 22 mm thick was used as optical windows. The force

sensor is a calibrated piezoelectric sensor (Kistler 9207). The signal from the sensor was amplified by a charge amplifier (Kistler 5018). The output voltage from the charge amplifier was acquired at a frequency of 100 kHz by a DAQ card (National InstrumentTMPCI 6040E). A low pass filter of 2kHz was applied to the obtained raw data, and it was ensured that enough information and accuracy was retained. The spray impact force is detected by means of a circular target screwed directly on the sensor head. The circular target has a smooth surface with a diameter of 3 mm to capture the spray jet. The distance between the sensor and the nozzle tip can be adjusted by adding shims of different size. For all the test cases, the gap was kept at 1.5 mm. The distance was measured using a feeler gauge which was used to position the circular target screwed to the sensor at this distance. For each test case, a total of 50 injections were conducted. The circular target was cleaned by blowing high-pressure nitrogen to prevent any residual fuel droplet after each test case. The voltage recorded was then scaled to give the force using the scale factor obtained from calibration. The scaled data from the sensor is then corrected with the half cone angle as the incident angle of the spray on the circular target is assumed to be same. The entire data acquisition and control were automated using a custom written LabVIEWTM code as shown in figure 2. The total mass of the injected fuel (m_f) was then measured using a mass balance (Sartorius CPA64) for 100 injections. This process was repeated three times for each case, and the average value is used.

3.3. Imaging system

Diffused back-lit illumination (DBI) technique was used to capture the spray jet hitting the circular target screwed on to the sensor head. A LED and a diffuser were used to create the homogeneous background incident light. A high-speed camera (Photron SA4) was employed at 80,000 frames per second (fps) to capture the spray evolution. The camera triggering event was synchronized with the fuel injection through a digital pulse generator (Stanford Research Systems DG535). The images obtained were then used to find the exact start of injection and instant at which the spray impinges on the sensor.

4. Results and discussion

4.1. Validation of the method

Figure 3 shows the solenoid current and the injection rate for a single injection event with an injector excitation time of 1.75 ms. The black line represents the mean injection rate (ensemble-averaged over 50 injections) and the lighter shaded region around the mean injection rate curve represents the standard deviation of the 50 injections. The blue line represents the injector current profile. The time zero represents the start of the excitation current, and this convention is maintained for all the results presented in this work. It can be observed from the figure that the injection rate is highly repeatable and all the results presented are similarly reproducible. It can be seen that there is a steep increase in the injection rate during the hydraulic start of injection (HSOI) as the needle lifts and after that the transients of the needle and hydraulic hammering effect were well reproduced by the injection rate profile. The slope of injection rate

during the hydraulic end of injection (HEOI) is lower than that during the HSOI which is mainly due to the slow discharge of the solenoid current and the sac pressure to avoid needle bouncing which is typical for solenoid actuated injectors [18].

Figure 4 represents the sequence of images showing the hydraulic start of injection (HSOI) and the instant at which the spray jet impinges on the circular target screwed on to the force sensor for injection pressure of 100 bar and injector excitation duration of 1.75 ms. It can be observed from the figure, that the dimension of the circular target and the distance between the target and the injector tip are good enough to capture all the sprays from the multi-hole injector. It can be observed that the hydraulic start of injection is at 0.475 ms and the spray hits the target at 0.550 ms and hence the resulting travel time is 0.075 ms. This travel time is in good agreement with the travel time calculated from the spray jet velocity of 200 m/s under the considered condition of injection pressure and ambient pressure.

Figure 5 shows the comparison between the raw and corrected injection rate curves along with the injector current profile at the injection pressure of 100 bar and excitation duration of 1.75 ms. The raw injection rate is obtained from the force measured by the force sensor using the Eq.6. The start of injection of the raw injection rate profile is adjusted with the traveling time of the spray to the target to obtain the corrected rate profile. The start of injection has been corrected using the high-speed videos for all the cases presented in this work. The time between the start of excitation and the hydraulic start of injection (HSOI) is termed as the opening delay and time between the end of excitation and hydraulic end of injection (HEOI) is termed as the closing delay. The difference between HSOI and HEOI is termed as the hydraulic duration of injection (HDOI). For this condition, the opening and closing delay are found to be 0.475 and 0.98 ms respectively.

To validate the reliability of the method proposed, the cumulative mass calculated from the injection rate curve using Eq.7 and mass measured using the mass balance are compared in the figure 6. The shaded region shows a band of 10% deviations and the symbols represents the cumulative mass against the mass from mass balance measurements. It can be observed that all the points fall well within the shaded region, which means that the error between the cumulative mass and measured mass is below 10% for all the cases reported in this study. This inference shows that the experimental rig developed in this study have high reliability and accuracy in determining the injection rate profiles.

4.2. Injection rate

Figure 7a-d compares the injection rate under different injected mass (20, 40 and 60 mg/st) and excitation duration (1.75 ms) at different injection pressures (100, 150, 300 and 450 bar). It can be observed that the hydraulic injection duration decreases as the injection pressure increases to inject the same quantity of fuel mass. A fast opening and closing transients along with moderate oscillations are observed in the injection rate profiles for all injection pressures. The mild oscillations could be due to mechanical vibrations, injector needle bounce, fluctuations in the injection pressure or combination of all these. The full needle lift operation phase gets reduced as the injection pressure increases. In particular, the full needle operating phase tends to a minimum value at injection pressures of 350 and 400 bar for injecting a total mass of 20 mg/st.

In figure 7d, it is evident that even with constant injector excitation duration of 1.75 ms, the hydraulic end of injection (HEOI) event gets advanced slightly. To confirm that this behavior is not due to any measurement uncertainty, high-speed videos were taken without the sensor in the position, i.e., free jet to capture the actual end of the injection event. Figure 8 shows the comparison between the difference in hydraulic duration of injection (HDOI) captured through injection rate profile and high-speed images. The difference in injection duration was calculated by assuming injection duration at 100 bar injection pressure as the baseline. As the injection pressure is increased to 150, 300 and 450 bar, the HEOI is advanced by 50, 125 and 200 μs respectively. This result confirms that this behavior of advancing end of injection is happening by increasing injection pressures. Furthermore, it shows that this phenomenon could be due to the inherent response of the injector.

Figure 9a-d shows the comparison of cumulative injected mass under different injected mass (20, 40 and 60 mg/st) and excitation duration (1.75 ms) at different injection pressures (100, 150, 300 and 450 bar). The cumulative mass injected was calculated using the Eq.7. It can be noted that the slope of injection rate increases with increasing injection pressures. The cumulative masses derived from injection rate profiles are in good agreement with the mass measured using the mass balance.

4.3. Hydraulic delay

Figure 10a and b shows the injector excitation duration and the hydraulic duration of injection (HDOI). It can be seen that the excitation time increases as the total mass injected increases. This increase is because longer duration is required to inject more fuel at the same injection pressure condition. Furthermore, the excitation duration decreases as the injection pressure increases for the same quantity of mass injected. This is due to the fact that higher injection pressure requires less time to inject the same amount of fuel compared with lower injection pressure. The same trend is observed for the HDOI. It should be noted that the hydraulic duration is always greater than its corresponding excitation duration for all cases of different injection pressure and total injected mass.

Figure 11a and b shows the opening and closing hydraulic delay of the injector. The opening delay is quite constant irrespective of injection pressure and total injected mass. The opening delay is constant because the initial boost current and boost duration of the injection current profile is maintained constant irrespective of the excitation duration. This makes the coil energize to same level irrespective of pressure and mass which makes it open at the same instant every time. However, the closing delay increases with total mass injected and decreases with fuel injection pressure. This follows the same trend as that of the excitation duration and HDOI as seen in figure 10a and b. This shows that the closing delay of the injector is shorter at high injection pressure and when the excitation duration is shorter. The observed trend was similar to that of solenoid-actuated diesel common rail injectors as reported by Ferrari et al [26]. This is because it takes time for the electrical energy from the control unit to overcome the resistance of the injector's closing spring. This delay time varies with the injector design. It should be noted that the closing delay is directly

proportional to the excitation duration. As the excitation duration is longer, the solenoid gets charged more and takes a longer time to discharge and hence the longer closing delay.

4.4. Discharge coefficient

Figure 12a and b shows the transient and average discharge coefficient under varying injection pressure, and total injected mass. The discharge coefficient was calculated using the Eq 6 with the actual mass flow rate $\dot{m}_f$, the geometrical cross sectional area A_0 of the nozzle orifice, the fuel density ρ_f and the differential pressure $\Delta P = P_{inj} - P_{amb}$. Similar to the injection rate, it is hypothesized that the discharge coefficient also assumes the same profile as that of the momentum flux. This transient information is beneficial for the numerical simulation to improve the fidelity of the predictions. The average discharge coefficient does not change much with the change in either injection pressure or total injected mass. Typically, the discharge coefficient of diesel injectors under high injection pressure and the non-cavitating condition can be around 0.9 [27, 17]. However, it is known that for GDI injectors, the discharge coefficient is much smaller than its diesel counterpart. The average discharge coefficient of this injector is approximately 0.5 which is equal to the manufacturer's specification. A similar range of discharge coefficient has been reported for Spray-G multi-hole GDI injector from Engine Combustion Network (ECN) [22]. The low value of discharge coefficient for GDI injectors can be attributed to the small effective cross-sectional area and low effective velocity of the spray. The small effective cross-sectional area could be due to flow not utilizing the entire orifice diameter which is typical in case of GDI injectors under non-flashing conditions [28]. The effective velocity is low due to the low operating injection pressures of GDI injectors in contrast to the diesel injectors which can operate up to 3000 bar [29, 30].

5. Conclusions

This study focused on the development of injection rate measurement rig based on momentum flux method to measure the injection rate of any type of fuel injectors. In this study, the injection rate of GDI injectors was measured. The method was well validated with the mass measured using the mass balance, and then the hydraulic start of injection (HSOI) and end of injection (HEOI) were corrected to the true value using the high-speed videos. Following observations can be drawn from the results obtained from this study

- The injection rate increases with the injection pressure to inject same quantity however the injection duration gets decreased.
- Under same excitation duration, as the injection pressure increases, the hydraulic injection duration decreases. The same was confirmed by the high-speed video of the spray exiting the nozzle.
- Both excitation duration and hydraulic duration increases with increase in total injected mass and decrease in injection pressure.

- It was found that the opening delay remains constant over total mass injected and injection pressure. However, the closing delay increases with total mass and decreases with injection pressure.
- The discharge coefficient for the GDI injector used in this study is low around 0.5 and almost constant over wide range of injection pressure and total mass injected.

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Table 1: Fuel properties

Properties	Values
RON	91
MON	83.4
AKI	87.2
Density (kg/m^3)	748.3
Kinematic viscosity (mm^2/s)	0.55
Vapor pressure (kPa)	49.6
H/C ratio	1.982
Aromatics (%vol)	22.5
Olefins (%vol)	5.7
Sulfur (ppm)	10
IBP ($^{\circ}C$)	41
10% ($^{\circ}C$)	59
50% ($^{\circ}C$)	102
90% ($^{\circ}C$)	158
FBP ($^{\circ}C$)	176
Net Heat Value (MJ/kg)	41.92

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Figure 1: Schematic of the rate of injection measurement set-up

Figure 2: LabviewTM code for control and data acquisition for injection rate measurement

Figure 3: Ensemble-averaged injection rate with standard deviation and injector current profile for 1.75 ms excitation time

Figure 4: Sequence of images showing hydraulic start of injection (HSOI) and instant at which spray impinges on sensor

Figure 5: Comparison of raw and corrected injection rate

Figure 6: Comparison of the total mass calculated from injection rate profile and measured by mass balance

Figure 7: Comparison of injection rate at different injection pressures for total mass of a) 20 mg b) 40 mg c) 60 mg and d) 1.75 ms injection duration

Figure 8: Difference in hydraulic injection duration at constant excitation duration of 1.75 ms for different injection pressures

Figure 9: Comparison of cumulative injected mass at different injection pressures for total mass of a) 20 mg b) 40 mg c) 60 mg and d) 1.75 ms injection duration

Figure 10: Comparison of a) excitation duration and b) Hydraulic duration of injection (HDOI) for total injected mass of 20, 40 and 60 mg/st at injection pressures of 100, 150, 300 and 450 bar

Figure 11: Comparison of a) opening and b) closing delay for different injection pressures

Figure 12: Comparison of a) transient and b) average discharge coefficient at different injection pressures and total mass injected