Conceptual Development of a Pressure-swirl Injector For a Two-stroke GDI Engine

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Abstract: Numerous attempts have been made by researchers and developers around the globe to rectify the fuel "short-circuiting" problem of two-stroke engines by employing gasoline direct injection. Almost all published literature showed promising outcome but there are a lot more work needs to be done before the GDI method came in widely practical within two-stroke engine application. Given such challenge, this paper presents a conceptual development of a pressure-swirl direct fuel injector for an arbitrary two-stroke engine application. The use of several empirical correlations as well as computational fluid dynamics calculations are also highlighted as useful tools for design selection and spray predictions.
Keywords: two-stroke, gasoline, direct injection

1. Introduction

The advantages of gasoline two-stroke engine over its four-stroke counterpart are higher power-to-weight ratio, less components, simpler construction, and lower cost. Two-stroke engine has fewer components than four-stroke engine typically because two-stroke engine uses no valves. The piston of the two-stroke reciprocating engine takes over any valve functions in order to obtain a power stroke each revolution of the crankshaft. The air and mixture flow in and out of the combustion chamber through several ports on the cylinder walls. The piston movement will cover and uncover the ports at correct time for maximum fluid exchange inside the combustion chamber. The process of emptying the cylinder of burned gases and replacing them with a fresh mixture (or air) is called scavenging.

During the scavenging process, the intake and exhaust ports (at certain duration) are both open at the same time and some of the fresh air/fuel charge is lost out the exhaust port. This loss of fresh fuel is called short-circuiting. The typical two-stroke engine loses 30%-40% of its fuel during this scavenging process, with losses of up to 70% under idle conditions [1]. Two major problems associated with short-circuiting is high fuel consumption, and high percentage of hydrocarbons (HC) released with exhaust emission.

With the ever increase of crude oil price, and more stringent emission level standard regulation throughout the continent, the two-stroke engine is facing greater challenge to remain competitive with its four-stroke counterparts. To counter such problem of two-stroke engine, the foremost measure is to minimize the problem of fresh fuel "short-circuiting" thus reducing part of exhaust emission level, and part of fuel consumption. Still under extensive development, a novel method of gasoline direct injection (GDI) is a known permanent solution to the fuel "short-circuiting" problem. GDI is a method of injecting gasoline directly into the combustion chamber of spark ignition (SI) engine. Results from numerous recent researches done in this particular field such as Casarella et al. [2], Badami et al. [3], Morikawa et al. [4], Stan et al. [5], Pontopiddan et al. [6], Ramakrishanan et al. [7], Mitianiec [8], Nuccio and Marzano [9], and others showed promising outcomes.

2. Technical Requirements

The main requirement of GDI application is high atomization quality. There are a wide range of atomization approach but the most popular for GDI application is pressure-swirl atomizer or sometimes known as simplex atomizer. The basic components of a pressure-swirl injector are (i) needle, (ii) swirler and (iii) nozzle as shown in Fig. 1. The fuel will flow through the needle seat passage when the needle is actuated. Before the fuel reaches the needle seat passage, it is forced to flow though a swirler to create angular and swirl momentum. Due to the both momentum, a thin liquid film will formed on the wall of the nozzle orifice. This occurrence creates low-pressure region near the central axis of the nozzle orifice thus forming an air core. Thus, the thin liquid sheet will exit the nozzle orifice at a certain angle depending on the ratio of axial velocity and radial velocity of the fuel.

The purpose of a GDI injector is to supply correct amount of atomized fuel for a given engine requirement. Thus, it is necessary to identify the engine in which the injector will be applied before any design work commences. In this study, a prototype pressure-swirl injector was conceptually developed to satisfy arbitrary 125 cc two-stroke gasoline engine. The diagram of the engine is shown in Fig. 2 and its specification is shown in Table 1.

The application of GDI in a two-stroke engine is more challenging than four-stroke engine. Two-stroke engine has power stroke on every cycle, which means fuel needs to be injected in every cycle. This limits the injection period to 360° CA unlike 720° CA as in four-stroke engine. To solve the fuel short-circuiting problem during scavenging process, fuel needs to be injected during all ports are closed (i.e. during compression stroke after scavenging process ends). The duration between exhaust port close and ignition point is known as static allowable injection period [4]. This prerequisite set further constraint the atomization process within two-stroke GDI engines. Thus, this paper focuses on the following considerations: (i) injector duty cycle, (ii) injector static flow rate, (iii) spray angle, (iv) mean droplet size, (v) fuel operating pressure, and (vi) spray pattern.

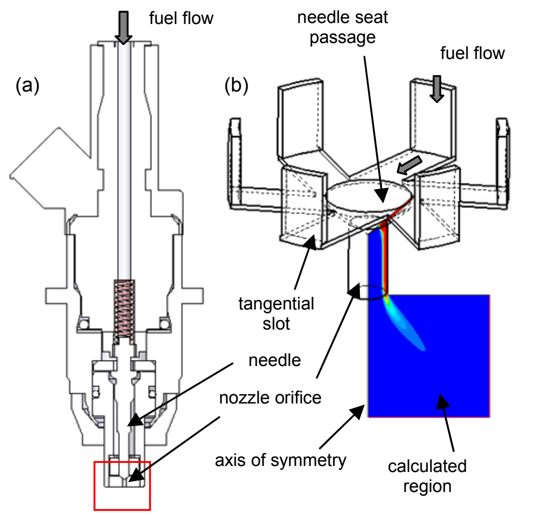

Fig. 1. (a) Cross-section of a typical pressure-swirl injector, (b) volume of fluid of the tangential slot and near nozzle exit; also shown is the 2D section of the CFD calculated region.

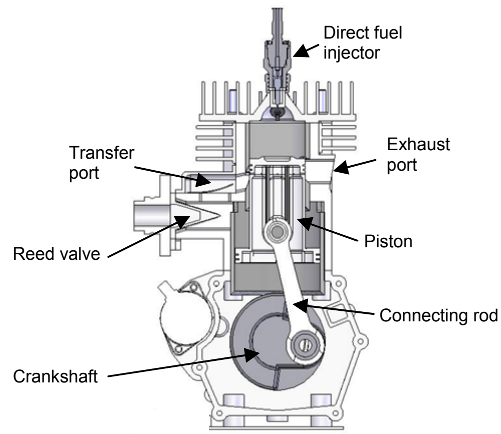

Fig. 2. Prototype two-stroke step piston gasoline engine cross-section

Table 1. Base engine specifications

Engine type	2-stroke, air-cooled	**Chamber geometry**	pent-roof
No. of cylinder	1	**Compression ratio**	8.5
Displacement	125 cc	**Exhaust port close**	99° BTDC
Bore x Stroke	54.2 x 54.2 mm	**Ignition**	10° BTDC (approx)
Connecting rod length	110 mm	**Power**	9.3 kW@7500 rpm
Piston type	bowl, step piston	**Torque**	15.7 kg/m@5600 rpm

2.2 Injector Static Flow Rate

The static flow rate of the injector should allow the injector to disperse adequate fuel within the limited duty cycle. The static flow rate was calculated by considering:

1. Engine displacement of 125 cc
2. Estimated scavenging ratio of 0.6
3. Stoichiometric AFR at engine speed up to 10,000 RPM
4. Injector duty cycle constraint limited to 20%
5. Orifice diameter of 0.80 mm
6. Estimated discharge coefficient of 0.13

The calculated minimum static flow rate of the injector should be 118 cc/min measured at 0.3 MPa or 480 cc/min at 5.0 MPa. The calculation was performed using several formulas suggested by Heywood [1].

2.1 Injector Duty Cycle

Duty cycle of the injector should be limited within static allowable injection period. Within this limited time, adequate fuel should be injected directly into the combustion chamber to realize the following advantages:

1. solve the problem of fuel short-circuiting
2. better fuel consumption through lean burn stratified charge approach
3. better volumetric efficiency

The typical port timing of two-stroke engine is shown in Fig. 3. From the Fig., the static allowable injection period is approximately 90° CA which gives injector duty cycle of 25%. However, static allowable injection period decreases as engine speed increases. In order to provide design allowance for advance ignition point, the injector duty cycle should be further reduced to 20%. Fig. 4 shows the static allowable injection period of a typical two-stroke engine.

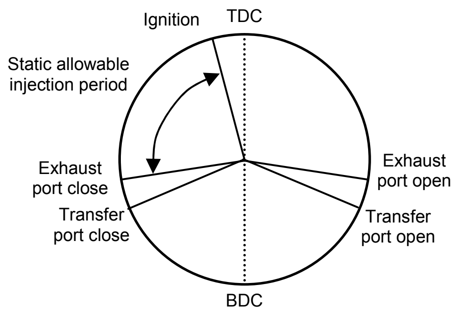

Fig. 3. Typical port timing of two-stroke engine.

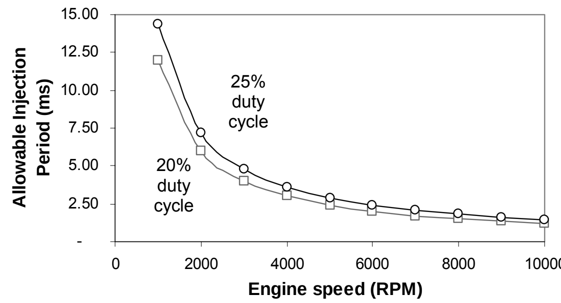

Fig. 4. Allowable injection period for typical two-stroke gasoline engine.

2.3 Spray Angle

The spray angle of the droplet should be optimized to provide the best atomization while avoiding cylinder liner wetting. Wang and Lefebvre [10] droplet size empirical correlation showed that wider spray angle produce better atomization. Lefebvre [11] regards cylinder liner wetting as fuel wastage. Stanglmaier, et al. [12] work indicate that spray impingement upon cylinder liner is an important source of HC emission in GDI engines. Thus, to get the best atomization, to avoid fuel wastage, and to minimize HC emission, the practical limit of the spray angle should be the maximum point before it wet the cylinder liner. On the contrary, for two-stroke engine application, where gasoline is premix with lubricant, a slight cylinder liner wetting might be a wise approach to lubricate the system.

For the specified engine, at 6000 RPM spray impingement is likely to occur at 60° BTDC. With injector placed at the center of the pent-roof, the maximum spray angle at that particular crank angle should not exceed 38.28°. However, the optimum spray angle for wall-guided approach should depend on the geometry of the piston bowl. Thus, as shown in Fig. 5, at any crank angle, optimum spray angle is given by arc tan (b/a), where a is a function of crank angle.

2.4 Mean Droplet Size and Fuel Pressure

The mean droplet size produced by the injector should be less than 20 micron Sauter Mean Diameter (SMD). Generally, smaller fuel droplet size promotes better combustion because of larger surface area [13]. Based on several experiments, Zhao, et al. [14] stated that a GDI system have significant advantages over comparable port fuel injection (PFI) system when operated with droplet SMD of 20 micron and below. However, Blair [15] suggested that two-stroke GDI engine should have droplet SMD of 10-15 micron based on works from Beck et al. [16], Nuti [17], Sato and Nakayama [18], and Plohberger et al. [19].

Previous works by Radcliffe [20], Jasuja [21], Lefebvre [22], Wang and Lefebvre [23] showed that the droplet SMD for pressure-swirl atomizer is inversely proportional to the fuel-ambient pressure difference [24]. Correlation of droplet SMD by Lefebvre [22] shown in Fig. 6 suggests that 5.0 MPa fuel pressure is sufficient to produce droplet SMD of much less than 20 micron. Further increase of the fuel pressure no doubt will reduce the droplet SMD but will also increase the cost of the associated components and reduce its reliability.

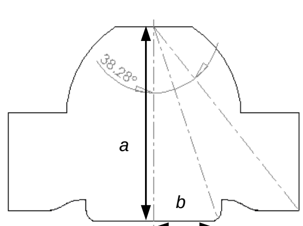

Fig. 5. Combustion chamber cross-section.

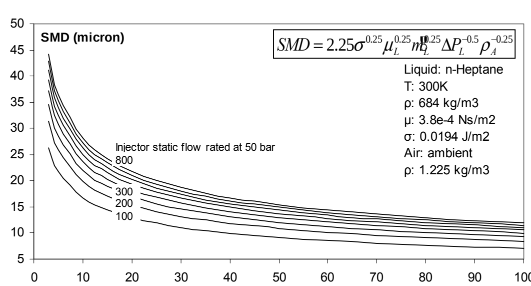

Fig. 6. Mapping of Lefebvre [22] droplet SMD correlation for pressure-swirl atomizer.

2.5 Spray Pattern

Theoretically, the optimum spray configuration is subjected to injector location and orientation on the engine, combustion chamber roof, piston bowl shapes, and spark plug location. Thus, the injector should generate spray pattern that takes into account the above environment. However, Sogawa and Kato [25] experiment showed that the best injector location is at the center of the combustion chamber with vertical orientation. Under such configuration, their result showed considerable reduction in exhaust emission.

3. Calculation

The variations of GDI injector such as fuel injector type, nozzle shape, fuel injection pressure, position & orientation of injector, etc. have different effect on spray performance. Other factors such as piston shape, cylinder size, compression ratio, position of spark plug, engine speeds, etc. will also significantly affect the performance of the whole system. Combination of all significant factors will results in millions of probable outcome especially in terms of injector performance, exhaust emission and fuel consumption. Due to these factors, to conduct accurate experiment procedure for an analysis of GDI within the cylinder of an engine would be costly. In contrast, an adequate model of the process with the aid of computational fluid dynamics (CFD) software was proved otherwise. A validated model of the system even extend the “what if” possibility even beyond the practicality of an actual experiment.

To verify the compliance between the design and the requirements specified, several CFD calculations were performed. The calculations consist of Eulerian multiphase model and Lagrangian discrete phase model. For hollow-cone pressure-swirl atomizer, the use of Lagrangian approach to simulate the discrete phase is favorable because of lower computing cost compared to multiphase Eulerian approach. Each has their own important representation of the calculated spray. However, where Eulerian approach thrives, the Lagrangian approach fails to give prediction of flow inside a pressure-swirl atomizer and the near nozzle spray characteristics, particularly: (i) the thin liquid film and air cavity due to swirling velocity of the fuel, and (ii) the spray angle as part of the input parameter for the Lagrangian discrete phase approach [26].

3.1 Eulerian multiphase model

The Eulerian multiphase model was used to calculate the internal flow of the injector nozzle and near vicinity of the nozzle exit. The outputs of this calculation are liquid sheet thickness at nozzle exit, and initial spray angle prior to droplet breakup. A static analysis of the flow was calculated with the following assumptions:

1. the ambient air is initially quiescent
2. the transient effect of the needle lift is negligible,
3. needle lift is at maximum and its distance is constant at 100 micron
4. pressure drop from high pressure fuel line to the injector tangential slot is virtually negligible
5. all fuel entering from all the tangential slots has the same vector relative to the axis of symmetry

The calculation was performed using setting shown in Table 2. The calculation was done in 2D due to assumption made earlier. The calculated region is shown in Fig. 1(b). The domain size, boundaries and output is shown in Fig. 7.

Table 2. Eulerian multiphase calculation setting.

Software	Fluent 6.1.22
Solver	2D axisymmetric swirl
No of phases	2 (air, n-heptane)
Viscous	RNG k-epsilon turbulence model; Swirl dominated flow
Min 2D face area	2.295e-10 m^2
Max 2D face area	6.250e-08 m^2
No of cells	10,984
Pressure inlet	5.0 MPa (axial= 0, radial= -0.617, tangential= -0.787)

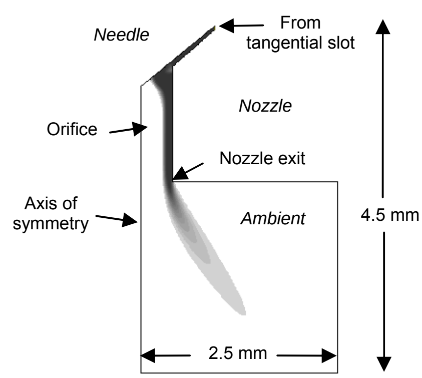

Fig. 7. Volume fraction of liquid fuel at 0.32 ms after SOI, plotted in grayscale. The color black represent liquid fuel volume fraction 100%.

It was observed in the calculation that the thin liquid sheet has fully developed and reached its steady state in less than 0.16 ms after start of injection (SOI). However, the analysis was done subjectively at 0.32 ms after SOI. Half spray cone angle (θ) is given by the direction of densest liquid sheet exiting the nozzle, which in this case was found to be 26°. Since the exact boundary between liquid fuel and air phase is difficult to determine, the liquid sheet thickness data for each case were judged at 50% volume fraction of each phase. The liquid sheet thickness at nozzle exit was found to be approximately 123.5 micron.

Using a SMD empirical correlation found by Wang and Lefebvre [23], such liquid sheet thickness would yield a calculated SMD of 17.62 micron. The equation is given by

$$\mathrm{SMD} = 4.52\left(\frac{\sigma\mu_L^2}{\rho_A \Delta P^2}\right)^{0.25}\left(t_f \cos\theta\right)^{0.25} + 0.39\left(\frac{\sigma\rho_L}{\rho_A \Delta P}\right)^{0.25}\left(t_f \cos\theta\right)^{0.75} \quad (1)$$

where c is the surface tension, μ_L is the fuel viscosity, ρ_A is the air density, ΔP is the pressure differential between fuel and air, t_f is the liquid sheet thickness at nozzle exit and θ is the half spray cone angle.

3.2 Lagrangian discrete phase model

The Lagrangian discrete phase model was used to calculate the spray dispersion. The outputs of this calculation are spray pattern and mean droplet size. To predict the spray characteristic inside the combustion chamber during the compression stroke, 3D dynamic mesh was used. The setting used in this calculation is shown in Table 3.

From Fig. 8, it can be seen that the droplet SMD increased as the piston gets nearer to top dead center (TDC). Although the same injector configuration was used, the initial droplet SMD of the two cases differed. This observation was supported by empirical correlation that droplet SMD is inversely proportional to the fuel-air pressure differential [20, 21, 22, 23]. As injection ended after 1 ms pulse width, the resulting droplet SMD for both cases is almost similar. The calculation also indicated that droplet impingement on the piston surface does not contribute much on the increment of the mean droplet size. The increase of the mean droplet size is likely due to the compression of the cylinder which promote the droplet coalesces. Thus, it suggests that the ignition timing should be placed as soon as the end of injection to enhance better combustion.

Table 3. Lagrangian discrete phase calculation setting.

Software	Fluent 6.1.22
Solver	3D in-cylinder dynamic mesh
Viscous	RNG k-epsilon turbulence model; Swirl dominated flow
Liquid	n-heptane
Droplet breakup	Wave model
Half-cone angle	26° (from Eulerian calculation)
Fuel static flow rate	7.6 g/s
Pulse width	1 ms

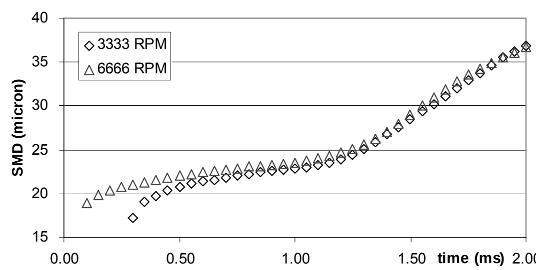

Fig. 8. Calculated SMD droplet temporal analysis.

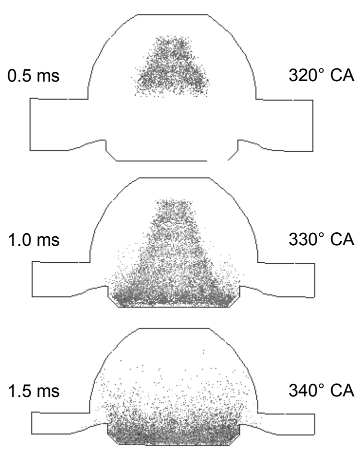

Fig. 9. Calculation of spray characteristics inside combustion chamber during compression stroke at 3333 RPM. Injection starts at 316° CA and ends at 336° CA.

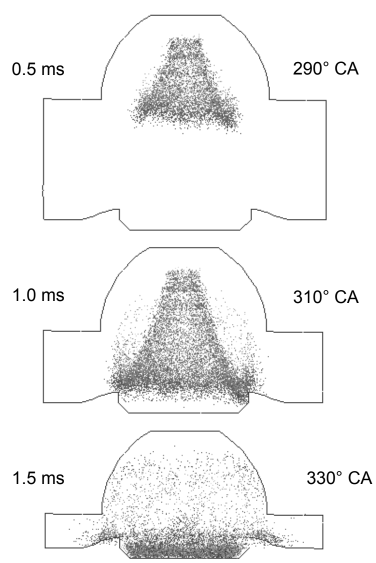

Fig. 10. Calculation of spray characteristics inside combustion chamber during compression stroke at 6666 RPM. Injection starts at 272° CA and ends at 312° CA.

Fig. 9 and 10 shows particle track of the in-cylinder injection at several instant after SOI during compression stroke at 3333 RPM and 6666 RPM respectively. The particle tracks are shown in grayscale where darker shades represent larger particle diameter. Fig. 10 shows temporal analysis of the droplet SMD under the both conditions.

From Fig. 9 and 10, the initial half spray cone angle of 26° appear to suit the spray characteristics required at tested engine speed with minimal amount of wall-wetting. By comparing both Fig. 9 and 10 at time 1.5 ms after SOI, at higher engine speed, more particle were dispersed outside the piston bowl. Thus, indicating that the amount of wall-wetting is expected to increase as the speed increases because of the limited static allowable injection period available. At both speed, rich mixture seem to be accumulated inside the piston bowl even though the allowable injection period nearly reach its end. This occurrence suggested that much of the droplet lost their momentum upon impingement with the piston surface. Still, some of the particles rebound and reached the roof of the combustion chamber.

4. Conclusions

This paper presents several important technical requirements of a GDI pressure-swirl atomizer for an arbitrary two-stroke engine. From such requirements, a conceptual design specification was obtained. The design is then subjected to several calculations in order to predict its probable spray characteristics. The predictions were crucial to ensure the specified technical requirements were being complied. From this study, several conclusions can be made:

1. Injector technical requirement of GDI two-stroke engine is more stringent than four-stroke due to the static allowable injection period.
2. The Eulerian multiphase model calculation shows good representation of the thin liquid sheet formation inside the nozzle orifice and the resulting initial spray angle.
3. The Lagrangian discrete phase calculation, which depicts spray characteristics during in-cylinder injection, showed that the mean droplet size increased dramatically due to compression.
4. Due to droplet impingement, piston movement in the opposite direction of the spray, and the loss of droplet momentum, relatively rich mixture was observed to remain inside the piston bowl.
5. Cylinder wall-wetting occurrence is likely intensified as engine speed increases because of the limitation of allowable injection period, the initial spray cone angle, and the shape of the designated piston bowl.
6. As a result of this study, a general specification of the conceptual GDI injector to suit the 125 cc two-stroke engine was obtained. The specification of the conceptual injector is shown in Table 4.

Table 4. Conceptual injector specification

Orifice size	0.80 mm
Duty cycle	20% (during compression)
Static flow rate	480 cc/min at 5.0 MPa
Half cone angle	26° (±2°)
Mean droplet size	17 micron SMD (approx)
Fuel pressure	5.0 MPa (minimum)
Spray pattern	Symmetrical hollow-cone

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