Journal of the Society of Automotive Engineers Malaysia
Volume 2, Issue 1, pp 14-22, January 2018
e-ISSN 2550-2239

A Study on Gasoline Direct Injection (GDI) Pump System Performance using Model-Based Simulation

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ORIGINAL ARTICLE ***Open Access***

Article History:

Received
16 Apr 2017

Received in revised form
7 Nov 2017

Accepted
1 Dec 2017

Available online
1 Jan 2018

Abstract – *This paper focuses on a simulation study of a newly developed high-pressure fuel pump system for small engine. When an engine is operated at high speed in a typical Gasoline Direct Injection (GDI) system, its pump will perform extra pumping requiring continuous engine work. Because it is driven by the engine's camshaft, the extra pumping action is both unavoidable and parasitic. In this study, a new GDI pump has been designed and built to only operate at a constant speed, regardless of engine load and speed. The pump is driven by an electric motor via a camshaft and is intended for a four-stroke, 0.2 litre, single-cylinder, spark ignition engine. The electric motor is governed by a control unit called the Engine Control Unit (ECU). The GDI pump will supply fuel to a rail up to its maximum pressure capacity. The pump is developed in accordance with a physical model-based design approach and is simulated using Matlab-Simscape™. Based on the calculation and simulation performed, the designed pump pressure is capable of producing discharge exceeding 4.5 MPa. Theoretical calculation also shows that the pressure developed by the pump reaches 10.54 MPa when a two lobe cam is used. In addition, the pressure developed by the pump is recorded to be 11.15 MPa, with an error of 5.8 % when a similar condition is applied to the physical modelling.*

Keywords: Gasoline direct injection (GDI), demand-controlled system, fuel pump, MATLAB, model-based design simulation

1.0 INTRODUCTION

A high-pressure Gasoline Direct Injection (GDI) fuel pump is an essential component in the state-of-the-art automotive fuelling system. Such a pump type is required to pressurize the quantity of fuel at certain pressure and maintain it inside a storage called fuel rail (Achleitner et al., 2007; Hoffmann et al., 2014). There are two methods to maintain pressure in the fuel rail, namely: (i) Demand-Controlled System and (ii) Continuous-Delivery System. The

Demand-Controlled System uses a high-pressure fuel pump equipped with a solenoid valve (Hiraku et al., 2005; Husted et al., 2014; Miller, 1987). Hence, the fuel quantity delivered at different engine speed and load can be controlled by adjusting the timing for energizing and de-energizing the solenoid valve. Through this method, the fuel pump will only deliver a constant fuel charge needed for the fuel rail. In contrast, by using the Continuous-Delivery System, the pump will deliver fuel supply which is nonadjustable. Thus, the pump requires a relatively high pressure to force the fuel into the rail against the primary pressure. Since the quantity of pump delivery cannot be adjusted, the pressure control valve attached to the fuel rail is needed to depressurize excess fuel (Mahmoudzadeh Andwari & Abdul Aziz, 2012; Mahmoudzadeh Andwari et al., 2013; Shin et al., 2013).

This study shall focus on an analysis performed on a high-pressure fuel pump based on the Continuous-Delivery System. In other words, the fuel pump will not be equipped with solenoid valve. For this purpose, a high-pressure fuel pump is designed and developed based on the Demand-Delivery System requirement. The driver mechanism of the pump is also changed. Because conventional GDI pumps are driven by the engine camshaft, the pumping rate is dependent on engine speed. The pump provides an extra pumping rate at high speed resulting in a waste of unwanted energy. It is impossible to avoid extra pumping during high engine speed because the camshaft rotational speed depends on engine speed. This study proposes a new driver mechanism powered by an electric motor with constant speed (Mahmoudzadeh Andwari et al., 2014; Spakowski et al., 2013). The electric motor drives the camshaft and determines the pumping rate and will be controlled by an Engine Control Unit (ECU). Therefore, the pressure developed by the pump is delivered to a reservoir (also known as common rail) and stored at a prescribed pressure range. In other words, performance of the high-pressure fuel pump depends on the engine speed. The fuel system is intended for a four-stroke, 0.2 litre, single-cylinder, spark ignition engine, which is the reference engine in this study. The pump will be running at constant speed and is independent of the engine, as far as energy supply is concerned (Feneley et al., 2017; Ghanaati et al., 2015; Mahmoudzadeh Andwari et al., 2017a).

In summary, the objective of the study is to develop a new high-pressure fuel pump for a GDI system operating at constant pumping rate based on the Demand-Delivery System (electric motor driver mechanism). Prior to laboratory evaluation, a physical model-based design simulation will be performed using Matlab-Simscape™ to assess its initial performance.

2.0 GDI PUMP SYSTEM-DESIGN CONSIDERATIONS

The high-pressure fuel pump will supply fuel into a common rail until it reaches maximum pressure limit. When maximum pressure in the rail is achieved, a pressure sensor attached to the rail will send a signal to the ECU. The ECU will then send another signal to the electric motor to turn it off. The pressure built-up in the reservoir will decrease when the fuel injector supplies the required fuel quantity to the engine combustion chamber based on information of the engine map. The ECU will also be responsible to turn on the electric motor again when pressure inside the fuel rail reaches the lowest limit allowed (Mahmoudzadeh Andwari et al., 2015; Mahmoudzadeh Andwari et al., 2017b; Muhamad Said et al., 2014).

Figure 1 shows the components that constitute the GDI system for a reference engine. In order to examine the pressure developed inside the combustion chamber at Top Dead Center (TDC) position of the piston, the engine compression ratio is taken into consideration, as shown in Equation 1 wherein P_{TDC} = Pressure at TDC [Bar], P_0 = Pressure at Bottom Dead Center (BDC) [1 Bar], CR = compression ratio [9:1] and γ = specific heat ratio (1.4 for air).

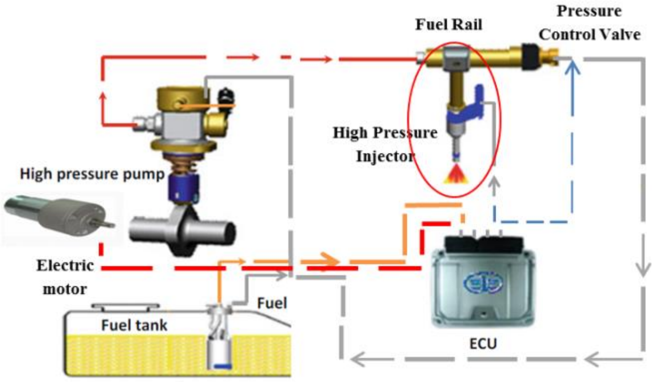

Figure 1: Schematic view of GDI system design

$$P_{TDC} = P_0 \times CR^{\gamma} \tag{1}$$

Using Equation 1, this will result in P_{TDC} = 2.167 MPa. Since the injected fuel quantity is dependent on the counter pressure in the combustion chamber, the pressure produced by the injector has to exceed more than 2.167 MPa. It is possible to inject fuel at such pressure if the maximum pressure capacity of the fuel rail exceeded 2.167 MPa. Therefore, the maximum pressure capacity of the fuel rail is set at 4.5 MPa. As such, the pump should theoretically produce higher pressure to force fuel into the rail. To avoid any damage to the pump, the output pressure must be higher than the maximum pressure developed in the fuel rail. Based on this argument, the pump will, therefore, be designed to produce pressure exceeding 4.5 MPa.

The designed GDI pump parameters have been set to be as follows: plunger diameter = 10 mm; dead volume = 1.236 e-6 m^3; Plunger stroke = 10 mm; cam rotational speed: 1000 rpm; number cam lobes = 2 lobes. The pump having these parameters should be able to generate pressure output of more than 4.5 MPa. In order to estimate pump displacement, Equation 2 is used. Wherein D = Pump displacement (l/s), A = plunger cross-section (m^2), S = Plunger stroke (m), N = camshaft speed (rpm) and n = Number of plunger. To calculate for pump volumetric efficiency (VE), Equation 3 is used. Wherein S = Slip/percentage of leakage, c = fluid chamber volume at TDC (m^3), d = volume displacement per plunger (m^3), ρ_d = fluid discharge density (kg/m^3) and ρ_s = fluid suction density (kg/m^3). And pump capacity; Q (m^3/s) is assumed by using Equation 4. The speed of plunger; υ (m/s) can be calculated by Equation 5 wherein n = Piston stroke per minute (Pumping rate or pump revolution per minute (prpm) and s = stroke length [m]. Therefore, the pump power input or mechanical power delivered to the pump input shaft P_i (kW) can be calculated based on Equation 6 wherein P_{td} = suction and discharge pressure difference (kN/m^2) and ME = mechanical efficiency (%) (Miller, 1987; Spegar, 2011; Spegar et al., 2009).

$$D = \frac{(A)\times(S)\times(N)\times(n)}{231} \tag{2}$$

$$\text{VE} = 1 - \left[S - \frac{c}{d}\left(1 - \frac{\rho_d}{\rho_s}\right)\right] \tag{3}$$

$$Q = D \times VE \tag{4}$$

$$v = \frac{n \times s}{6} \tag{5}$$

$$P_i = \frac{(Q \times P_{td})}{(1714 \times ME)} \tag{6}$$

In order to validate these parameters, theoretical calculation and physical-based simulation using MATLAB™ are employed to validate the estimated pump performance parameters. Performance of the pump is described in the following section (Spegar et al., 2009; Sun et al., 2013).

3.0 PHYSICAL MODEL-BASED SIMULATION

In this section, the model-based simulation for high pressure GDI pump system is performed using MATLAB-Simscape™ software. Physical modelling employs a physical network approach, where Simscape™ square blocks correspond to physical elements such as valves, pipe, accumulators, orifices, sensors and motors. The lines between them are physical connections which represent power transmission route. In general, the connection between these blocks indicates assemble connection of real physical components of the fuel pumping system as illustrated in Figure 2.

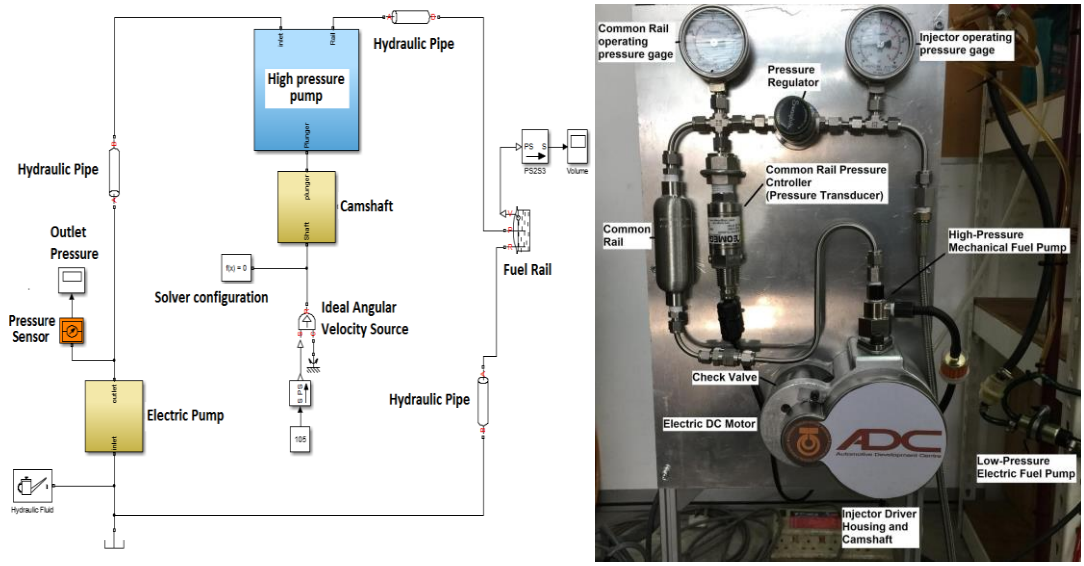

Figure 2: Physical model based simulation diagram and the physical components of the GDI kit showing the pump connected to the electric motor

The simulation starts with a Hydraulic Reference Block (HRF) which represents a fuel tank. As the name implies, the fuel tank acts as a fuel supply reservoir for the injector. The other HRF parameter input is fuel, i.e. gasoline. In the real system, low-pressure electric pump inside the tank will have to deliver fuel at constant pressure of 0.6 MPa. To deliver fuel at this constant pressure, an Electric Pump Subsystem (EPS) block has been created. In EPS Block, there are many physical elements including the inlet valve, outlet valve, orifices and single acting hydraulic cylinder. There is a sub-system inside the High Pressure Pump Block which comprises the Single Acting Hydraulic Cylinder, Sensor, Scope, Translational Spring and Translational Damper block. The Single Acting Hydraulic Cylinder block is used to set the plunger cross-sectional area, maximum plunger travel, initial distance between the plunger and cap and the dead volume. The Translational Spring and TS Block are used to represent the spring component. The spring will retract the plunger during the needle down stroke. Other blocks such as Flow Rate Sensor, Pressure Sensor and Scope are used to detect performance of the pump.

In order to define the driver mechanism of the pump, the Constant block, Simulink-PS Converter block, Ideal Angular Velocity Sources block and Solver Configuration block are used. The Constant block will represent a constant rotation of 105 rad/s. This angular speed value is the speed of the rotational camshaft, which is also equal to 1000 rpm. By using Simulink-PS Converter block, the unit of less than105 input signals will be converted to a physical signal, again in rad/s. All parameters for these physical elements have been set based on the so-called target design requirements.

Through this approach, performance of the pump can be verified based on the pre-determined parameters. The physical model is more accurate than the theoretical calculation as it includes physical structure of the pump such as its orifice size and valve maximum passage area in the prediction. Furthermore, by using this method, the physical elements such as maximum passage area, volume of the pump chamber and orifice area designed for the pump are also taken into account to predict the overall performance.

4.0 RESULTS AND DISCUSSION

Figure 3 represents the output pressure of the electrically driven fuel pump. Since the standard requirement for the pressure supply at inlet of a high-pressure fuel pump is around 0.6 MPa, the electric pump subsystem design in the simulation is assumed to deliver fuel at such pressure. As shown by the peak point of the graph, the electric pump is able to supply the target pressure of 0.68 MPa. The output of the signal from the pump is a time-averaged value, which is set at 1 second in this simulation.

Figure 4 illustrates the piston stroke movement based on the parameters calculated in the last section. Since the maximum length the plunger is able to travel inside the chamber is 12 mm, the y-axis is the representation of the traverse length. The plunger starts to move up at 11 mm because the distance between the piston and the cap at BDC has been set at 11 mm. In other words, the piston will have a 1 mm gap by the time it reaches TDC and this is also true for the 1 mm gap at the bottom of the chamber, i.e. at BDC position. The piston moves from BDC to TDC with a sweeping 10 mm distance. Accordingly, the cam stroke parameter inside the Parabolic Cam block is set at 10 mm.

Based on the simulation outcome in Figure 5, the pressure developed by the fuel pump is around 11.15 MPa. This represents the pressure developed when the pump speed is set to run at 1000 prpm. Based on calculation, the result shows that the pressure developed is 10.54 MPa when the pump speed reaches 1000 prpm. The other result based on the same speed obtained from the MATLAB shows that the pressure developed by the pump is 11.15 MPa. The difference between the calculated pressure and the pressure obtained from the simulation is marginal at around 5.8%. Therefore, the parameters used for this pump has been validated to be able to generate delivery pressure greater than 4.5 MPa.

The pump capacity (pump fuel flow rate) is shown in Figure 6. The pump is capable to deliver fuel at the rate of almost 2.07e-4 m^3/s. The pump capacity was computed using Flow

Rate Sensor block, as depicted in Figure 2. This block has been placed in several locations within the pump to study the change in volumetric flow rate. The sensors are placed directly with connection (hydraulic conserving port associated with the cylinder inlet and outlet) of a single acting hydraulic cylinder block. It is also placed after the outlet valve block. However, volume flow rate as recorded by a sensor did not indicate any difference in value. Therefore, it is clear that the volumetric flow rate in this simulation disregards the changes in orifice area as well as the outlet valve.

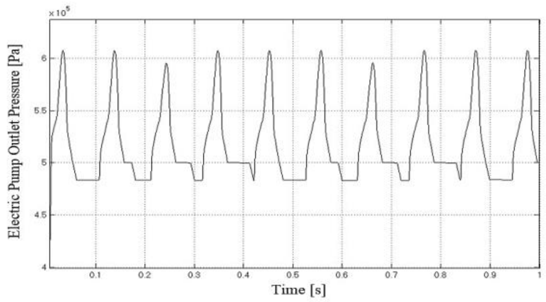

Figure 3: Electric pump outlet pressure vs. Time

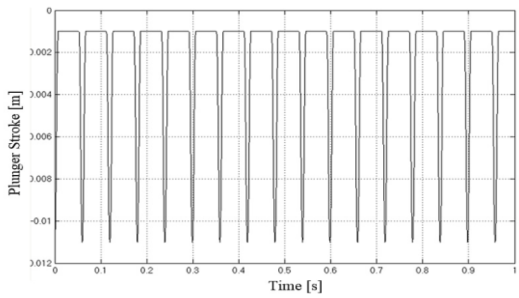

Figure 4: Pump plunger stoke vs. Time

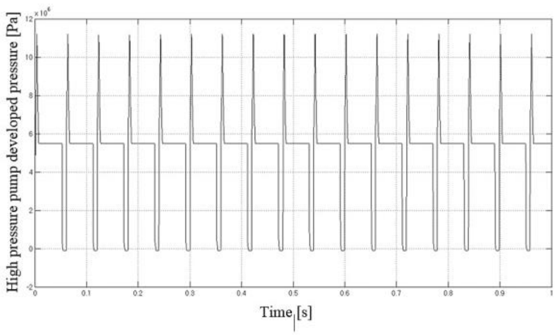

Figure 5: Pump pressure vs. Time

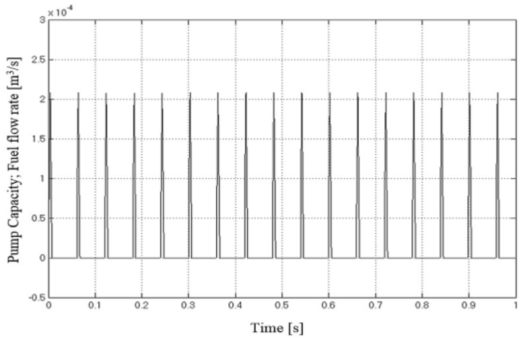

Figure 6: Fuel flow rate vs. Time

Some of the assumptions made in the simulation work include the power transmitted is of 100% mechanical efficiency, the speed of the electric motor is 1000 rpm, and the pump is driven by a two lobes cam mechanism. As such, the pump speed is taken as 2000 prpm. Based on the calculation performed, the pump capacity at this speed is 2.48e-5 m^3/s. In contrast, the simulation result shows the pump capacity equalling 2.07e-5 m^3/s. The error between the calculated pump capacity and the simulation pump capacity is around 16.0 % and there are a few reasons for this. The calculated pump capacity is obtained after considering volumetric efficiency inside the pump. The volumetric efficiency is possibly affected by leakage inside the pump.

Such leakage normally occurs due to fluid slippage passing the pump valves before they are closed. Leakage can also occur between the plunger and its seal inside the pump cavity. Due to the difference between seal system of the plunger pump and the piston pump, the volumetric efficiency can be somewhat dissimilar. In the MATLAB simulation, the block that represents the single acting hydraulic cylinder uses piston type sealing system. In a piston pump, the sealing system is attached to the piston and moves together with it during the operating strokes. In contrast, sealing system for plunger pump is stationary and the plunger moves through it during strokes. Therefore, the volumetric efficiency between these two sealing systems will obviously be different in magnitude. Since the high-pressure fuel pump uses a plunger type of sealing system, the result from the theoretical calculation is assumed to be more reliable. Table 1 is a summary to provide a better insight on the difference between theoretical and simulation results, in which the errors are tabulated.

Table 1: Comparison between theoretical calculation and simulation results

Parameters	Theoretical Calculation	Simulation	Errors [%]
Cam speed [rpm]	1000	1000	-
Input power [watt]	153.94	150	2.6
Cam lobe	2	2	-
Pump speed (prpm)	2000	2000	-
Pump displacement [m^3]	2.618e-5	2.618e-5	-
Volumetric efficiency [%]	0.95	0.95	-
Pump capacity [m^3/s]	2.48e-5	2.07e-5	16.5
Discharged pressure [MPa]	10.54	11.15	5.8

Figure 7 represents the computed volume of fluid data gathered using the physical signal port of the reservoir block. The graph shows a small change in the excess volume discharge, from the port when the pressure within the reservoir block exceeds that of the pressurization level of 4.5 MPa. The pressurization level is set based on the design requirement in this study. The optimum volume inside the fuel rail is 3.25e-4 m^3.

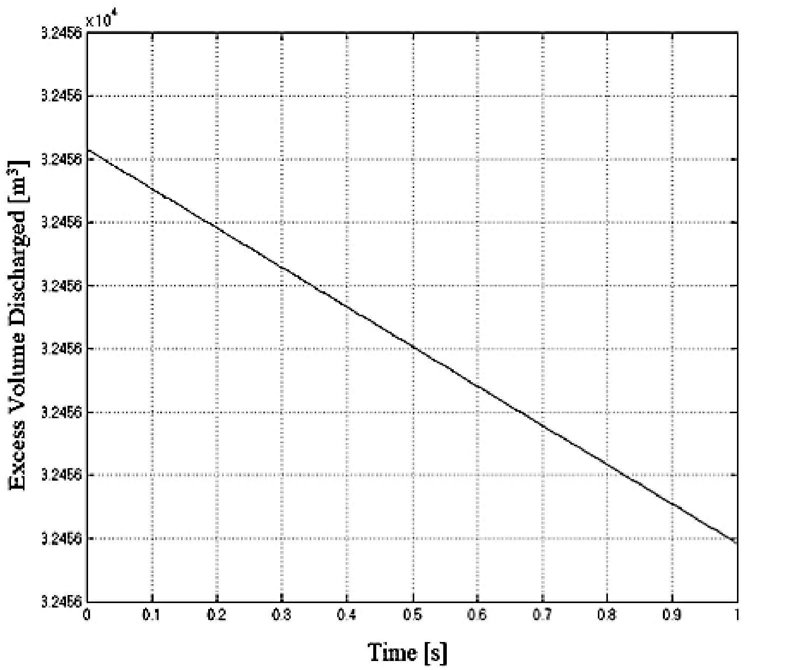

Figure 7: Excess volume discharged from the common rail

5.0 CONCLUSION

The physical based simulation using MATLAB was employed under all parameters determined to investigate performance of the newly developed high-pressure fuel pump. Based on the calculation and simulation, the designed parameters are proven to be capable of a pressure discharge exceeding 4.5 MPa, which in this case reaches 10.54 MPa. The theoretical calculation showed that the pressure developed by the pump reaches 10.54 MPa when two lobes cam is used. When the same condition is applied to the physical modelling simulation using MATLAB, the pressure developed by the pump is 11.15 MPa – indicating an error of only 5.8 %. These initial results reassured the researchers of the pump's ability to perform well in harmony with other components within the electrically driven GDI system.

ACKNOWLEDGEMENTS

The authors wish to acknowledge the Ministry of Science, Technology and Innovation (MOSTI) and Universiti Teknologi Malaysia with the project no. 03-01-06SF1370 (R.J130000.7909.4S111) for the financial support awarded to this research project.

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