Design, Modeling and Simulation of a High-Pressure Gasoline Direct Injection (GDI) Pump for Small Engine Applications

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ABSTRACT

In this study, a plunger-type high-pressure fuel pump is designed for use in a Gasoline Direct Injection (GDI) system and intended to retrofit onto small spark ignition (SI) engines of less than 0.2 liter/cylinder. The pump is developed to generate high-pressurized fuel in range of 4-20 MPa, and is capable of supplying fuel transfer of 10-20 g/s. SolidWorks™ software was extensively used in the analysis to design the pump via 3D modeling approach. The designed model was then analyzed using ANSYS Fluent™ software to simulate for the flow pattern as well as the pressure distribution in the pump cavity during pumping process. The concept, design geometry, calculation and the testing method employed for the high-pressure fuel pump are discussed in this paper. This research aims to design a high-pressure fuel pump used in small spark ignition engines applications wherein to improve the efficiency of the engine, the fuel pump is driven by an external DC motor, relieving the engine in providing the auxiliary power supply. In general based on the simulation, during pumping process at 1000 RPM (using the two cam lobe design), the pump is able to supply fuel at 6.00 MPa of the discharge pressure. The fuel was able to be delivered by as much as 0.0196 kg/s, at a travelling

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velocity of 3.8 m/s, in which this is well in the range of the initial pump design requirements.

Keywords: *Gasoline Direct Injection (GDI), ANSYS Simulation, SolidWorks, Fuel Pump, Computational Fluid Dynamic (CFD)*

Introduction

The latest technology that has the potential to meet the entire criterion for an efficient, modern spark ignition (SI) engine is the Gasoline Direct Injection (GDI) system. GDI engine is now becoming popular due to its potential to reduce of exhaust emission and fuel consumption, increase engine performance and the ability to adapt to Ultralow Emission Vehicle (ULEV) requirements [1-5]. GDI systems are able to precisely control fuel metering and improve fuel atomization thus accelerate mixture preparation. It is a known fact that precise control of injection timing is accomplished due to well design of injector, in conjunction with the use of engine control unit (ECU) [6-11]. However, to further improve the fuel atomization and accelerating mixture preparation, high-pressure fuel pump is required. The key to the success of this technology is the efficient fuel pump performance with low power requirement to drive the pump. The role of a fuel pump for an engine is to supply fuel to operate the engine, ensure it can run at sufficient air fuel ratio at variable speeds. The basic structure of the GDI fuel rail system includes pressure control valve, high-pressure pump, fuel rail, injectors, rail pressure sensor and the Electronic Control Unit (ECU) [12-19]. Most of the GDI systems currently used in high-end automobile engines employ a single piston pump type because of design simplicity due to the few number of components involved. There are many famous manufacturers that produce high-pressure pump for GDI systems and one of them is Magneti Marelli SpA. The pump they produced is a single piston pump type that is mechanically operated by a multi-lobe cam and is designed in compact size, precision-built and lightweight, capable of generating 40 to 200 bar pressure in conjunction with a common rail [20-27]. Another well-known developer is Bosch GmbH in which they develop the HDP5 pump. The pump is basically a continuous running single piston plugged-in pump capable of generating high flow rate at high speed and has a flexible design features made of stainless steel [28-32]. The pump is equally capable to generate up to 200 bar pressure. The current high pressure fuel pump in conventional GDI system is continuously driven by a cam that is powered by the engine directly, wherein the efficiency of the engine is reduced due to the parasitic load, to enable the pump to function effectively.

In this work, in order to improve the efficiency of the engine, the fuel pump will be driven by an external DC motor, relieving the engine in providing the auxiliary power supply. This research and development project aims to design a high-pressure fuel pump of intermittent operation. It is intended for use in small spark ignition engines applications especially for mobility purposes. To be more precise, the pump is intended to be matched to a four-stroke, 0.2 liter, single-cylinder, spark ignition engine, in which it is hope to deliver improve fuel economy throughout it speed range.

Demand-Controlled Fuel System

As mentioned earlier, this special high-pressure fuel pump is designed to fulfill the fuel metering requirements of a 0.2 liter, single-cylinder, air-cooled four-stroke engine. This engine will be equipped with a pressured-swirl type injector in which the fuel source comes from the high-pressure reservoir. The aim of the pump is to charge the reservoir (common rail) capable of withstanding pressure range of 50 to 80 bar. The fuel system is classified as *demand-controlled* type in which once the pressure limit in the reservoir have been exceeded, the energy supply to the pump will be cut-off. The reservoir will continuously supply the fuel to the injector until the low preset pressure limit (50 bar) has been reached. Here the pump will be activated to recharge the reservoir until the upper pressure limit is reached. This is an ON/OFF system that relieves the energy requirement for the pump activation, which normally comes from the engine. A DC motor is used to drive the prototype pump. The motor and the pump are housed in a special casing as one complete unit. Basically this GDI conversion kit comprised of i) pressurized swirl injector ii) pump-motor assembly iii) 0.25 liter common rail iv) pressure regulator v) check valve and vi) high- and low-pressure fuel lines. The fuel system is controlled by a micro-controller mapped to provide improved fuel consumption. Figure 1 shows the components that constitute the GDI system for a reference engine

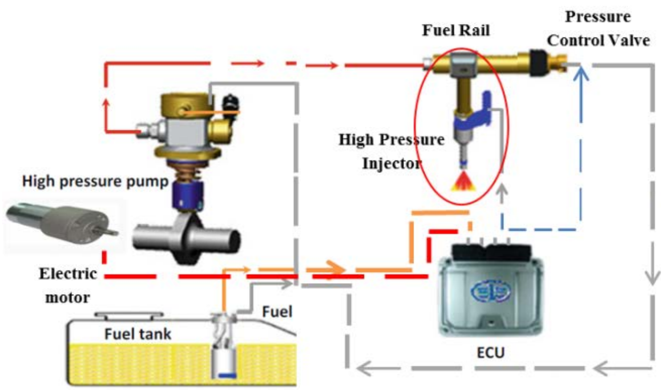


Figure 1: Schematic view of GDI system design

GDI Pump Concept

Basically, pump is a device to moves fluid to a certain level of high or supply fluid at certain pressure. The main component of a reciprocating pump is piston or plunger, cylinder, inlet and outlet port and inlet and outlet check valve. Today there are many additional components incorporated to improve the efficiency of the pump, like pressure relief to avoid pump damage due to pressure build-up, damper to reduce the pulsation flow that lead to pressure spike that can disturb the pumping process, packing to reduce slip. There also is a pump that uses a solenoid to control the opening and closing of the valve in order to control the volume of the fluid discharge and suction. The reciprocating pumps generally are driven by the external driver source that connected to a camshaft. The rotation of the cam at several speeds gives a motion to the plunger or piston rod. The motion of the piston forward and backward will build the expansion and contraction of the pump cavity. During down stroke the partial vacuum is created in the pump cavity. Due to the pressure difference, the inlet valve will open and fuel replenishes the pump cavity. Once the piston move upstroke, the pressure in the pump cavity will increase. Through a one way check valve, the inlet valve is closed and the outlet valve is open and allows the fluid to be pushed out of the pump cavity at high pressure and fluid velocity [33-35]. To design the pump the standard rating and working principle of a reciprocating pump is used. The design requires to the fuel supply at high pressure in the range 40-200 bar, at 10-20gr/s. For a GDI application a high pressure fuel is required in order to improve in fuel atomization process and accelerate mixture preparation for combustion process [3, 7, 13, 14, 20].

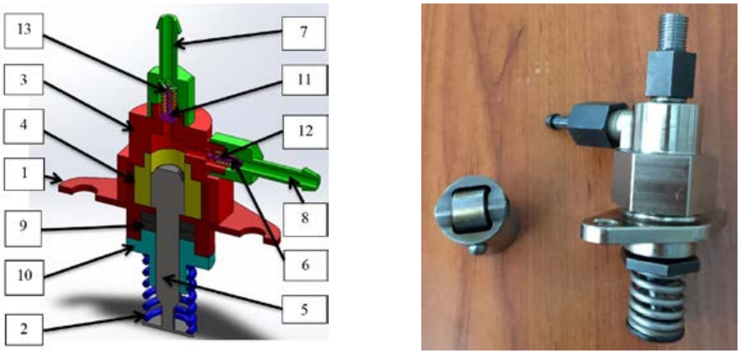
Table 1: GDI Pump specifications

Parameters	Values
Bore × Stroke	10×0 mm
Discharge Pressure	6.16 MPa
Suction Pressure	0.6 MPa
Mass flow rate	17.9 g/s
Motor Speed	1000 RPM
Cam type	2 lobe
Number of plunger	Single acting
Pump displacement	2.6182 e-5 m^3/s
Pump Capacity	2.487 e-5 m^3/s
Power input	0.20644 hp
Power Output	0.1764 hp
Working fluid	Gasoline-liquid (C_8H_{18})
Fluid temperature	26.67 C (80 F)
Pump Efficiency	85 %
Volumetric Efficiency	95 %

The single acting reciprocating plunger pump concept is selected due to the compact size and suitable to use in small SI engine applications, in platforms such as two-wheelers, range extenders and UAVs. To calculate and design the GDI pump the standard principle and equations of reciprocating pump design are employed. Design specification of the pump is made after basic analyses and design considerations were made and they are listed in Table 1. There are three springs in the pump including plunger spring, inlet check valve spring and outlet check valve spring. The advantages of this system over the conventional engine-driven type are i) relieving the engine of the pump power requirement and ii) the pump is able to operate at constant speed [28, 29, 31, 34, 35].

SolidWorks 3D Modelling

The cross-sectional view of the pump is illustrated in Figure 2(a). Figure 2(b) is the physical feature of the pump with its roller shown. It consists of 13 parts that are assembled to form a complete a unit wherein the components are as follows. 1: Body-1, 2: Plunger Spring, 3: Body-2, 4: Plunger Sleeve, 5: Plunger, 6: Suction Valve, 7: Discharge Cap, 8: Suction Cap, 9: Packing, 10: Packing Cap, 11: Discharge Valve, 12: Suction Valve Spring, 13: Discharge Valve Spring. The pump is modeled with 76 mm in length/width and 10 mm height. The material is set based on usual material used by manufacture for high-pressure pump.



(a) (b)

Figure 2: (a) Cross section view of the pump and (b) the physical features of the pump

The complete assemble model have a weight of 118.44 gm. The pump uses a single-acting working principle at which the suctioning and discharging process are occurred once in a cycle. The pump is driven by an external DC motor at 1000 RPM through a cam. In order to archive the required discharge pressure two-lobe cam type is considered in the study. During suction process the plunger will move down stroke about 10 mm stroke. The vacuum pressure is created in the pumping cavity. Due to the pressure different, the inlet check valve will open about 1 mm and let the fuel replenish to the pump cavity. During discharge process the plunger will move upstroke. At this time the inlet check valve is closed and the pressure in the pump cavity is increased due to the contracting of pump cavity through the plunger motion. Once the pressure in pump cavity reaches 6.16 MPa, the outlet check-valve will be open by 1 mm and the opening let the fuel to discharge out of the pump at high pressure and speed.

CFD Simulation Using ANSYS Fluent

By using ANSYS Fluent software, the behavior of fluid flow inside of the pump during suction and discharge is simulated. The pump cavity is extracted from SolidWorks drawing and subsequently used in the simulation, as shown in Figure 3 and Figure 4. Due to the symmetrical shape of the cavity, the model is simulated in a half section. This is to minimize the running time of the simulation and also to evaluate the contour build within the pump during pumping and suctioning processes.

Table 2: Imposed boundary conditions

Grid	Structured
Fluid	Octane-C_8H1_8
Density (kg/m^3)	720
Mass Flow rate (kg/s)	0.0179
Discharge Pressure (MPa)	6.16
Suction Pressure (MPa)	0.6
Turbulence Model	Standard k-Ɛ model
Temperature (°C)	26.66 °C

The boundary conditions for inlet and outlet are set to the designated mass flow rate and the pressure outlet respectively. The simulation condition is assumed as incompressible flow and no slip boundary condition occur. Table 2 describes the boundary conditions used for the flow simulation. Two different plunger positions in the pump sleeve are considered by using ANSYS Fluent software to simulate the condition of the fluid flow inside of the pump including suction process and discharge process [10, 15].

Results and Discussion

The motion of the plunger from top dead center (TDC) to bottom dead center (BDC) will induce fuel into the pump's cavity. The liquid transfer is due to the pressure difference developed between the pump cavity and the outlet port. Based on the results of the pressure contour depicted in Figure 5 the highest pressure developed is at the valve surface i.e. of about 0.33 MPa, as denoted by direction of the arrow. This pressure is the minimum pressure required to open the valve during suction process. In the pump and the valve cavities the pressure decreases slightly with plunger motion. Once the plunger ascend to the BDC position (i.e. 10 mm plunger down stroke) the pressure in the pump cavity would have reached 0.27 MPa. Based on the velocity vectors of Figure 6, the red streamline contours indicate the highest fuel fluid velocity region developed in the suction passage.Due to the different diameters of the cavity design the fluid flow velocity will vary during replenishing of the cavity process. The highest fluid velocity will occur when the fluid flow passes the valve cavity before entering the pump main cavity, which exceeds 7.25 m/s at the BDC position (i.e. 10 mm plunger down stroke). As denoted by arrow in the same figure, the velocity of fluid at the back of suction valve will exceed 3.25 m/s.

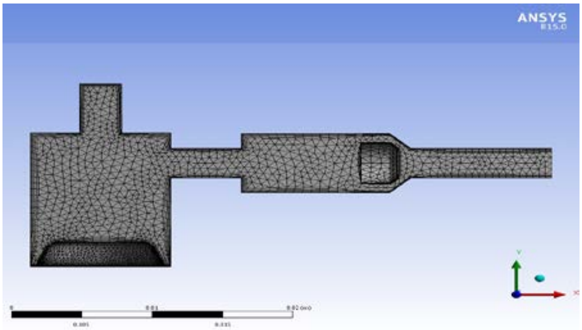


Figure 3: Extracted pump cavity in suction simulation

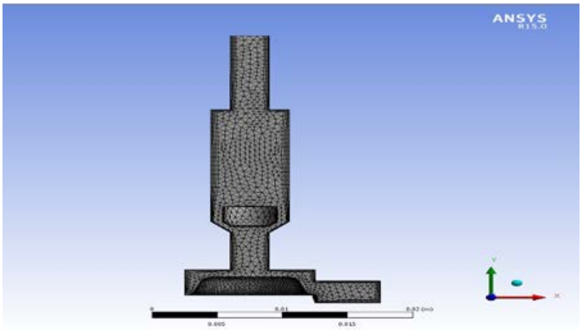


Figure 4: Extracted pump cavity in discharge simulation

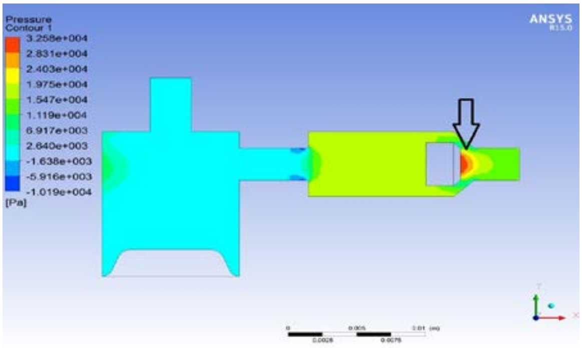


Figure 5: Developed pressure contour at BDC (Suction)

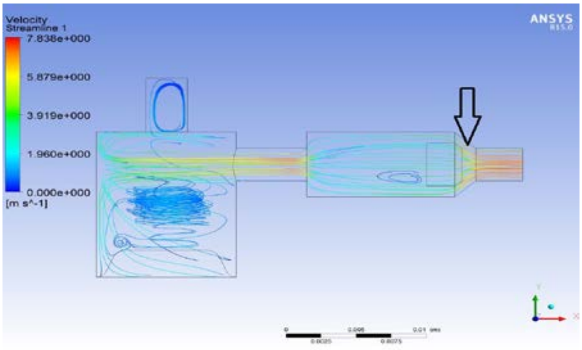


Figure 6: Velocity stream contour at BDC (Suction)

The motion of the plunger from BDC to TDC will increase the pressure inside of the cavity. This pressure increase will develop until it is able to overcome the outlet check valve pressure to discharge the fuel at high speed when needed. According to the pressure contour result of Figure 7, the highest pressure during pumping process will occur in the pump cavity as well as on the surface of the discharge valve, which is about 6.13 MPa, as denoted by the arrow. Based on the velocity streamline contour of Figure 8, the fluid flow velocity is at a maximum value when it passes through the smaller discharge passages before and after check valve, which will reach 9 m/s. The fuel flow velocity at the surface of the discharge valve is now around 3.8 m/s, just enough to lift the valve in correspond to the valve back pressure. As mentioned earlier the simulation was carried out focusing on three positions of the plunger i.e. BDC, middle and TDC. Figure 9 represents the relationship between the discharge/suction fuel flow pressure and the position of the plunger inside of pump's sleeve.

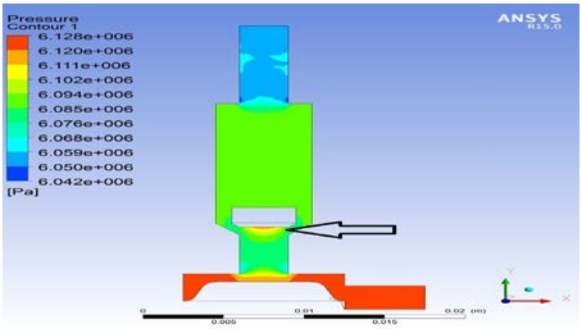


Figure 7: Pressure contour at TDC (Discharge)

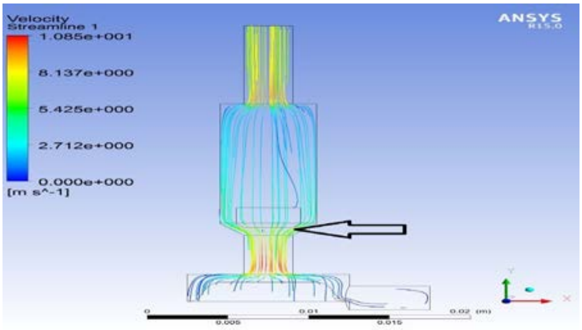


Figure 8: Velocity stream contour at TDC (Discharge)

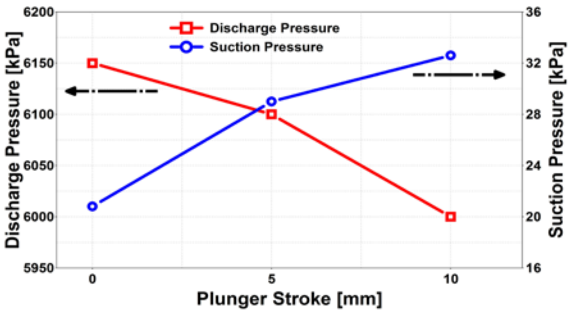


Figure 9: Variation of discharge and suction fuel flow pressure behavior against the plunger stroke

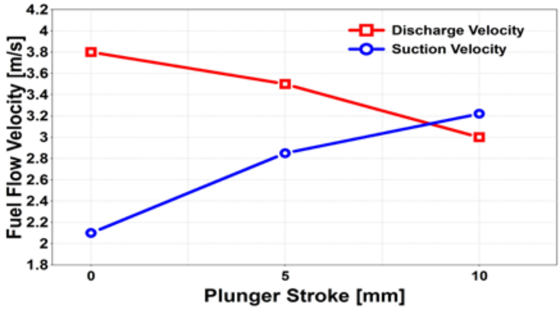


Figure 10: Variation of discharge and suction fuel flow velocity behavior against the plunger stroke

As can be clearly seen, the trend of curve representing discharge velocity will be at the highest pressure of discharge process (6.15 MPa) at BDC (i.e. 0 mm height of plunger stroke) while the highest pressure of suction process (0.325 MPa) will occur when the plunger is at TDC (i.e. 10 mm height of plunger stroke), as shown by the curve representing the suction velocity. This pressure behavior inside the pump's cavity implies that there is an inverse correlation between the position of plunger (stroke) and the pressure developed inside the pump's cavity. Thus in the suction process the pressure will develop when the plunger stroke is descending (from TDC to BDC). On the other hand for the discharging process the pressure will develop when the plunger stroke is ascending (from BDC to TDC). Moreover the maximum pressure created at the discharge process based on this simulation will be around 6.15 MPa. This is in a fair agreement with the target design pressure for the pump that was about 60 bar. The same trend can be deduced from the behavior of fuel velocity in the discharge and suction processes as illustrated in Figure 10. The figure presents the relationship between the fuel discharge/suction flow rates against the position of the plunger within the pump sleeve. From the curve it can be seen that the maximum fluid flow velocity in the discharge process (3.8 m/s) is achieved when the plunger has reached TDC. Nevertheless the maximum fluid flow velocity during the suction process is capable of reaching 3.25 m/s when the plunger is at BDC. According to the pump design requirements for the target discharge flow rate of 20 g/s, the maximum fuel flow velocity at the discharge (3.8 m/s) suggests a close agreement with that of the simulation estimation.

Conclusions

A single-acting GDI fuel pump is designed and numerically modeled using an industrial standard CFD code to predict its capability to meet the fuel supply requirements of a four-stroke, 0.2 liter, single-cylinder, SI engine. The physical components of the pump were designed using SolidWorks in which the behavior of the fuel fluid flow inside of the pump cavity was evaluated using ANSYS Fluent. Three positions of the plunger i.e. i) 0 mm of plunger stroke (BDC), ii) 5 mm of the plunger stroke and iii) 10 mm plunger stroke (TDC) were referred to in this study. The results of the CFD analysis are found to be in a close agreement to the preliminary design parameters of the pump. In general, during the fuel pumping process which is at a constant speed of 1000 RPM, at 2000 pumping rate (two lobe cam), the pump is capable delivering fuel at 6 MPa (60 bar of discharge pressure) generating 19.6 gr/s fuel flow rate, at a velocity of 3.8 m/s; wherein the pump design requirements are fairly met and warrant for further development of the conversion kit components.

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