A Physics Based Simulink Model for Common Rail Fuel Injection System for Rail Pressure Controller Development

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ABSTRACT

In common rail fuel injection system, the quantity of fuel injected depends on the rail pressure during injection. The common rail experiences pressure pulsations even during steady engine operations due to the delivery of high-pressure pump and the rapid opening and closing of injectors. The drop in rail pressure due to pilot injection affects the main injection. Therefore, pressure pulsation is an important aspect that needs to be considered in the process of injecting the required amount of fuel. In order to minimize pressure pulsations, rail pressure is controlled in closed loop. This is achieved through the fuel Inlet-Metering Valve (IMV) of the high-pressure pump or Pressure Control Valve (PCV) on the rail. In order to develop simulators that can enable development of such controllers, simple yet accurate physics-based models of subsystems like high-pressure pump including IMV, Rail and Injector are required. Hence in this work, a common rail injection test bench is used to study the system and IMV is used to control rail pressure. In this work, a three-cylinder pump with IMV was modeled starting from first principles in MATLAB Simulink and validated experimentally.

INTROCUCTION

The Common Rail fuel injection system is widely used because of its potential to lower emissions and noise levels and also improve drivability in automotive diesel engines. The main advantage of this system over traditional fuel injection systems is that the fuel injection pressure is independent of the speed of the engine. Further the ability to inject the fuel in successive well defined pulses within one cycle allows combustion to be tailored to meet stringent requirements. In modern diesel engines with high boost pressures and with the need for extremely low levels of engine out emissions very high injection pressures are required to mix the fuel effectively with air. Properly timed fuel injection and well mixed charge are important for high power output and low levels of smoke and NOx emissions. In common rail injection systems this is achieved by setting the injection pressure, injection timing, injection duration and injection sequence like pilot, main and post based on the operating conditions.

The common rail system consists of a high pressure positive displacement (plunger) pump that feeds fuel into a rail or accumulator wherein the pressure pulsations are damped and regulated. This fuel is admitted into the injector through a high-pressure tube. The injector may use a solenoid or piezoelectric actuator to time the injection pulses that are controlled by the Engine Control Unit (ECU). The precision of the common rail fuel injection system is mainly influenced by the rail pressure, as the quantity of fuel injected depends mainly on the rail pressure during injection. The pressure in the common rail depends on rate of fuel delivered by the high-pressure pump and the rate of fuel injected by the fuel injectors. The rail pressure is never constant in a cycle because of the intermittent flow of the pump and injectors. For steady operation of the engine, a constant average rail pressure is desired for reducing the cycle to cycle variations, which requires a closed loop control of the fuel pumped into the rail or the fuel leaked from the rail or both.

In the present common rail engines, closed loop control of rail pressure is achieved by two ways namely, by control of IMV (Inlet Metering Valve) on the high pressure pump or control of PCV (Pressure Control Valve) on the common rail. In case of control using PCV, more than the required amount of fuel is always pumped to high pressures into the rail and the excess fuel is drained to the tank to maintain the rail pressure. Thus the work done to pump the excess fuel which is eventually drained from rail to tank is actually wasted and this work is lost as heat to the drained fuel. In case of control using the IMV, the fuel flow into the high-pressure pump and thus into the fuel rail is varied as per the requirement and the desired rail pressure is maintained. The high-pressure pump is a positive displacement pump and hence only the restricted quantity of fuel is pumped. Thus closed loop control of rail pressure using IMV is more efficient as no fuel from the rail is drained.

In the case of rail pressure control using an IMV the amount of fuel entering the pumping chamber is controlled and this means for most operating conditions the pump is never fully filled. That is, at the end of the suction stroke of the plunger the volume of liquid fuel that is present is considerably lower than the volume of the chamber where it is present. Thus a part of the liquid fuel vaporizes to fill the entire chamber. During the pumping stroke this fuel vapor condenses and the liquid fuel is then pressurized and pumped. Though the high pressure pump incorporates a simple plunger arrangement, these complex phenomena have to be incorporated during the development of simulation models that can be used for development of controllers.

Model based control strategies for fuel injection could substantially reduce development time, improve accuracy of controllers, their performance and response time. For this accurate physics based models of the components like high-pressure pump with IMV, common rail and injector are required. Literature on physics-based models of common rail injection system components is scarce. The common rail high-pressure pump with no inlet metering valve was modelled using a commercial code [1] and good agreement between the experimental values was found. A detailed model of a common rail injector which predicts the needle lift is presented in [2]. This model uses force balance on the injector needle. In [4], the main components of gasoline common rail injection system like high-pressure pump, fuel metering valve and injector were modelled and transfer functions for the rail and injector were obtained. The modelling of high-pressure pump and common rail were done using equations involving the bulk modulus. However, the working of fuel metering valve is different from that of the diesel common rail injection system. In case of gasoline direct injection systems, the fuel inlet metering valve is purposefully kept open during the delivery stroke to pump the surplus fuel back in to the inlet. In case of diesel, however, the high-pressure pump is not allowed to be filled completely resulting in the formation of fuel vapour. Very little information is available in literature about the formation of vapour in the diesel common rail high-pressure pump and its modelling.

This work was aimed at developing a model of the high-pressure pump with IMV and validating the same through bench experiments. Further the developed model was coupled with a model of the common rail and a simple yet useful model of the injector, in order to obtain a full system simulation program. The development of the pump model, its validation and the results of modeling the full system are presented and discussed. The developed model could have applications in HIL simulators.

INLET METERING VALVE

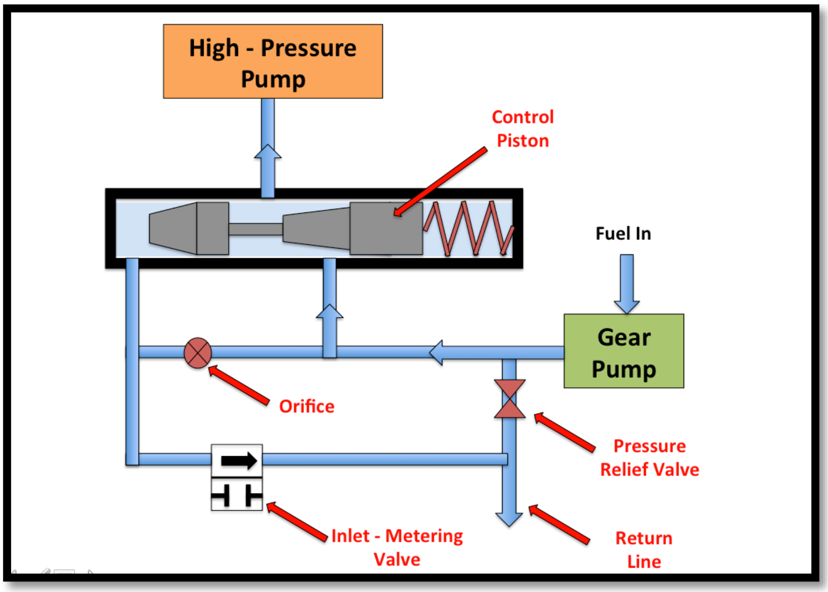

Fig.1 Schematic of the common rail pump with IMV

The layout of common rail pump that was used for the simulation model is indicated in Fig.1. The main components are the supply pump (gear pump), the high-pressure pump, control piston and inlet-metering valve. The control piston actually varies the area of opening that allows fuel into the pump. The position of the control piston depends on the pressure on its left end, which applies a force to balance the spring on the right. The IMV attains different positions based on the PWM signal it receives. The position of the IMV corresponds to a position of the control piston provided the supply pressure as regulated by the relief valve is kept constant. The supply is from the gear/vane pump whose outlet pressure maintained by the relief valve has been assumed as 5 bar in this simulation. Thus the duty cycle of the PWM signal given to the IMV varies the area of the orifice that allows fuel into the pump and controls its delivery.

HIGH PRESSURE PUMP

The high-pressure pump is a reciprocating piston pump driven by an eccentric. Three pumping elements are radially located at a spacing of 120 degrees and the eccentric pushes their plungers as it rotates. A spring is assumed to push the plunger against the eccentric in order to make it follow the same. Thus the stroke of the plunger is always constant but the amount of fuel that fills the plunger will depend on the position of the control piston, which in turn will depend on the IMV duty cycle. Of course the supply pressure and time allowed for filling based on pump speed will have an influence.

SIMULINK MODEL

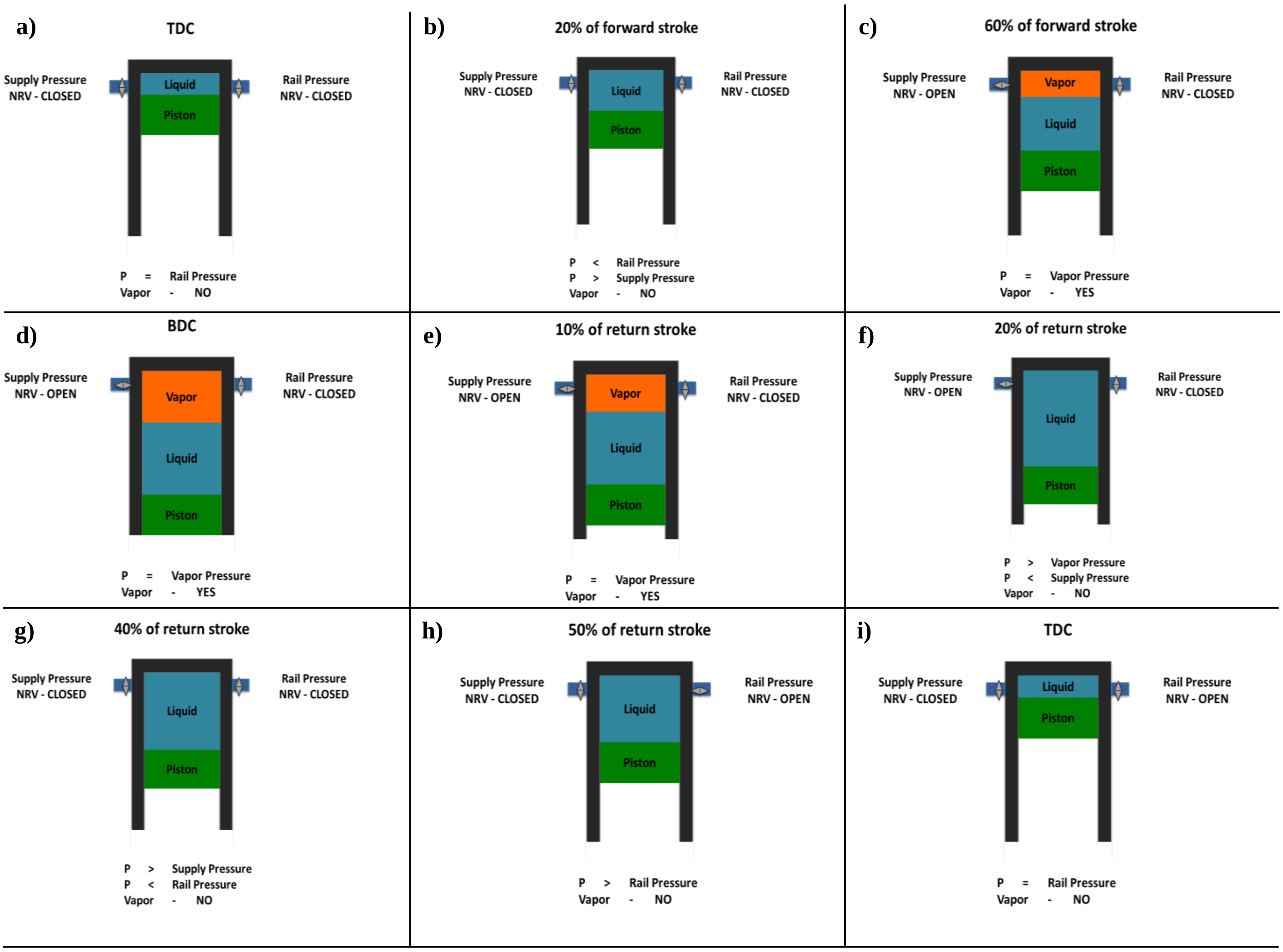

Fig.2 Stages of vapor formation inside the cylinder in one pump revolution

The process of fuel flow into the cylinder, the corresponding fuel expansion and compression in the cylinder of high pressure pump is modeled as described below. One pump revolution consists of a suction and a delivery phase. At the beginning of the suction phase, the clearance volume is occupied by liquid fuel at a pressure equal to rail pressure (Fig.2a). As the piston moves down during the suction phase (Fig.2b), the volume of the cylinder increases and the liquid fuel left in the cylinder expands to accommodate the additional volume. This reduces the pressure of the fuel inside the cylinder. As the pressure goes down, at some position of the piston, the fuel pressure falls below the supply pressure. This pressure difference between the supply line and cylinder results in fuel flow into the cylinder. The position of the inlet metering valve determines whether the cylinder is fully filled or partially filled. The flow rate into the cylinder is based on orifice equations. If the cylinder is partially filled (Fig.2c and 2d), at some position of the piston the cylinder pressure falls to saturation pressure of the fuel and on further expansion, part of the liquid fuel then will get converted to its vapor and the entire cylinder will remain at the saturation pressure of fuel at that temperature.

For a given IMV position, the cylinder will contain the maximum volume of vapor when the piston is at its bottom dead centre (Fig.2d). As the piston moves up, the volume of the cylinder starts decreasing and the vapor starts condensing to a liquid (Fig.2e). The pressure in the cylinder is still at its saturation pressure for the given temperature and is below the supply pressure of 5 bar. Hence fuel continues to flow in to the cylinder. As the piston moves further up, all the vapor gets converted to liquid and the pressure starts increasing to above saturation pressure (Fig.2f). Once the pressure increases above the supply pressure, inlet is closed by the non-return valve and the piston starts compressing the fuel inside the cylinder (Fig.2g). Once the pressure reaches the rail pressure, the non-return valve on the outlet opens and fuel flows in to the rail (Fig.2h). The fuel delivery to the rail ends when the piston reaches the top dead centre (TDC) (Fig.2i).

The bulk modulus equation is used to simulate the fuel pressure inside the cylinder in both suction and delivery phases. As per the equation (1), the rate of change of pressure at any instant is a function of bulk modulus of the fuel, volume of fuel and rate of change of volume. The rate of change of volume is a result of four different terms as given in equation (2). Here Q_{out} is the volume of fuel delivered by the high-pressure pump to the common rail. Q_{leak} is the effective volume of fuel leaked through the non-return valve in inlet and through the clearance between piston and cylinder. dV/dt is the change in volume of the cylinder due to piston motion and is obtained by geometry. Q_{in} is the volume of fuel added to the cylinder through the IMV. The differential equation was then solved at every time step to find the pressure inside the cylinder.

$$\mathbf{B} = -\mathbf{V}\frac{\mathbf{dP}}{\mathbf{dV}} \qquad \text{...(1)}$$

$$\frac{\mathbf{dV_{pump}}}{\mathbf{dt}} = \left[\mathbf{Q_{out}} + \mathbf{Q_{leak}} + \frac{\mathbf{\Delta V}}{\mathbf{\Delta t}} - \mathbf{Q_{in}}\right]_{\mathbf{pump}} \qquad \text{...(2)}$$

The fuel pressure inside the common rail was also modeled using the same equations with the leakage term set to zero and the volume change of the rail obtained from equation (3). The enlargement of the fuel rail volume in equation (3) was modeled using thick cylinder considerations. The rate of change in volume in the bulk modulus equation is a result of three different terms in equation (3). Here Q_{out} is the volume of fuel removed by the injectors from the rail. dV/dt is the change in common rail volume (expansion) due to high pressures. Lame's theory was used to find the radial, hoop and longitudinal stresses in the walls of rail. Hooke's law is used to compute the radial, hoop and longitudinal strains and finally the change in volume of rail. Q_{in} accounts for the volume of fuel added to the common rail by the high-pressure pump.

$$\frac{\mathbf{dV_{rail}}}{\mathbf{dt}} = \left[\mathbf{Q_{out}} + \frac{\mathbf{\Delta V}}{\mathbf{\Delta t}} - \mathbf{Q_{in}}\right]_{\mathbf{rail}} \qquad \text{... (3)}$$

These equations were converted into MATLAB Simulink blocks and solved using discrete solver with a time step of 50 nano seconds. The differential equation (3) was created in Simulink as seen in Fig. 3. It receives the inputs of rail pressure, inlet orifice area (IMV), and engine speed from the user. Crank angle was computed based on the engine speed. Engine speed and crank angle were given as inputs to the piston motion where the positon of piston, velocity of piston and volume of cylinder were calculated. The dV/dt block seen in Fig. 3 represents the equation (2). The four terms in equations (2) were computed at every time step. Thus rate of change in pressure was obtained at every time step. The pressure inside the cylinder was then calculated by integrating the rate of change in pressure. If the calculated pressure was observed to be less than the saturation pressure of fuel, the pressure was set as saturation pressure. The cycles were repeated till convergence was obtained.

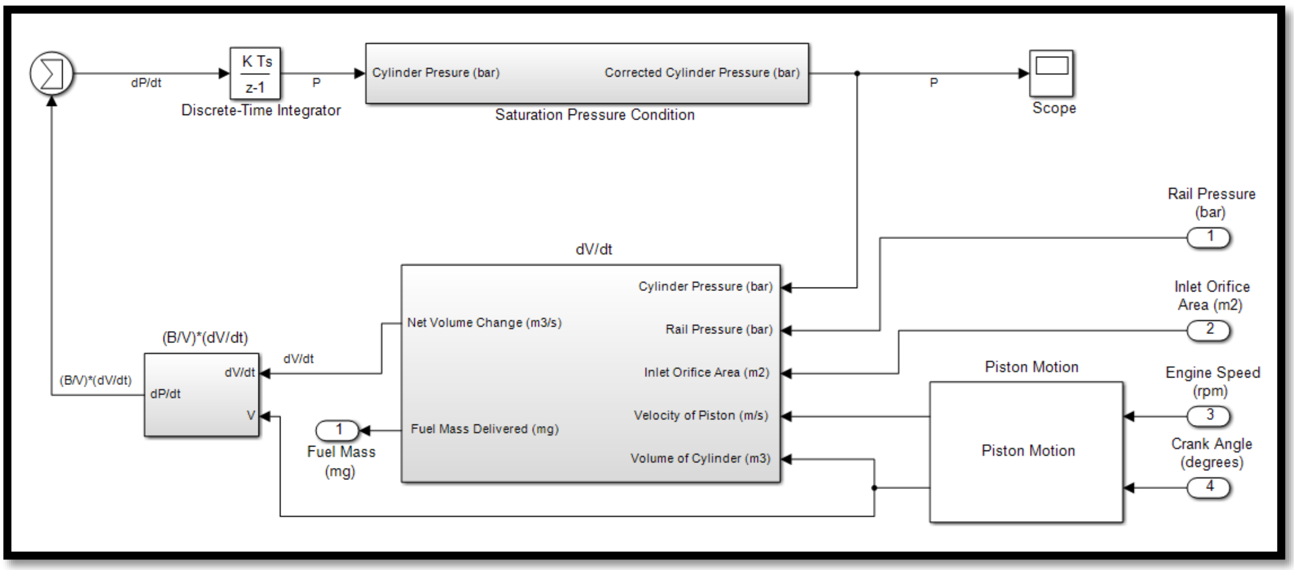

Fig.3 Simulink model of high-pressure pump (one cylinder)

EXPERIMENTAL SET UP

A special common rail fuel injection test bench was used to validate the high-pressure pump model. The test bench consists of a motor, which runs at one-fourth of the engine speed. The motor consists of a flange with two lobes separated by 180 degrees. The hall-effect sensor on this flange generates the cam signal, same as that generated on the engine. The motor drives a shaft with a gear ratio of 1:4 and the shaft runs at engine speed. The shaft consists of a 60-2 trigger wheel and an angle encoder to generate the engine crank signal. The cam signal and crank signal can be used by any engine control unit for synchronization in order to perform injection. This shaft drives the common rail pump with a gear reduction of 3:2 (same as that on the engine). The drive from the motor to the common rail pump can be seen in Figs. 4 and 5.

Fig.4 Motor and Angle Encoder

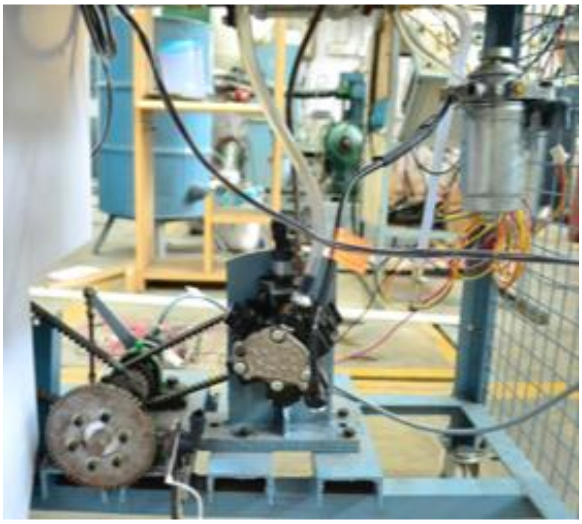

Fig.5 High-Pressure Pump and Trigger Wheel

EXPERIMENTS

Experiments were done to validate the high-pressure pump model. This was done by measuring the mass of fuel delivered by the high-pressure pump in one cycle in the test bench and comparing with simulations. The pressure control valve on the common rail was set at 600 bar and the mass of fuel leaked from the rail was measured when there was no injection. The duty cycle of IMV was then set at a value so that the rail pressure was not below 600 bar. Once the average rail pressure stays at 600 bar, the system was at steady state and whatever fuel leaks through the rail was the fuel pumped by the high pressure pump. The experiment was repeated for different IMV duty cycles, rail pressures and the mass of fuel leaked was measured at every duty cycle. The following results were obtained based on the experiments and simulations conducted at 600 bar. A good agreement between the results is seen. This was taken as the validation of the pump at different IMV positions.

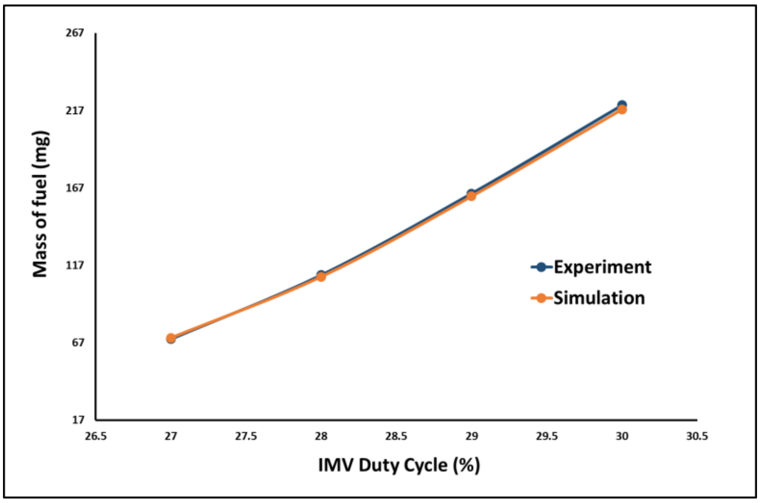

Fig.6 Comparison of mass of fuel delivered by pump in 1 cycle

This high-pressure pump model was connected to the common rail model and the rail pressure was plotted for the entire cycle. The effect of pumping and injection on rail pressure can be seen in Fig.7. The simulations were performed at 30% duty cycle of IMV, 1000 bar base rail pressure and injection duration of 700 micro seconds. The simulation starts at zero Crank Angle (CA) degrees and one revolution of pump takes 540 ^{0} CA. The simulation of cylinder 2 starts at 180 ^{0} CA and cylinder 3 starts at 360 ^{0} CA. The pressure and mass of fuel at TDC in cylinder 1 at the end of first cycle were given as initial conditions to the next cycle. The conditions at TDC converge after a few iterations. The converged rail pressure for one cycle is shown in Fig.7, where the influence of pumping and injection on rail pressure can be seen. In Fig.7, 1 is the start of pumping of cylinder 1 and 2 is the end of pumping. Point 3 is the start of injection and 4 is the end of injection. The same 4 actions repeat for cylinder 2, cylinder 3 and again cylinder 1 in one engine cycle (720 degrees).

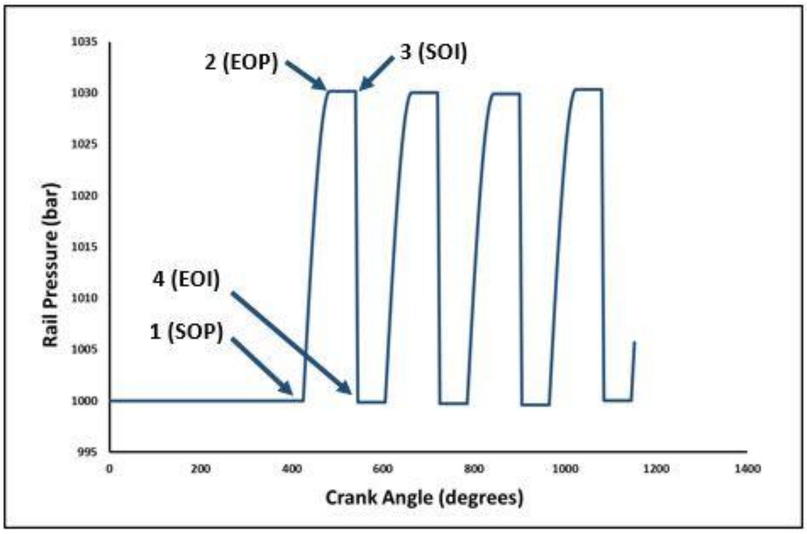

Fig.7 Variation of rail pressure in one engine cycle

Experimental data from mHawk 2.2L 4 cylinder common rail diesel engine at 2100 rpm and 80% load were shown in Fig.8. The duty cycle of IMV during this cycle was 32%, pilot duration was 235 micro seconds and the main injection duration was 830 micro seconds. From Fig.8, the base pressure was 1050 bar and the pressure rise was 40 bar. At high duty cycles, pressure rise will be high and correspondingly injection duration also will be high. The trend of pumping action and injection was correctly predicted by the simulations. The experimental data was used to partly validate the model. Simulations predicted a pressure rise of 30 bar for 30% duty cycle and 700 micro seconds injection duration. The pressure rise in simulations is 10 bar less than the experimental data because of smaller duty cycle of IMV. The pump used in validating the model for different IMV positions was different from the pump present in the mHawk engine which is also possible for deviation of simulations from experimental data.

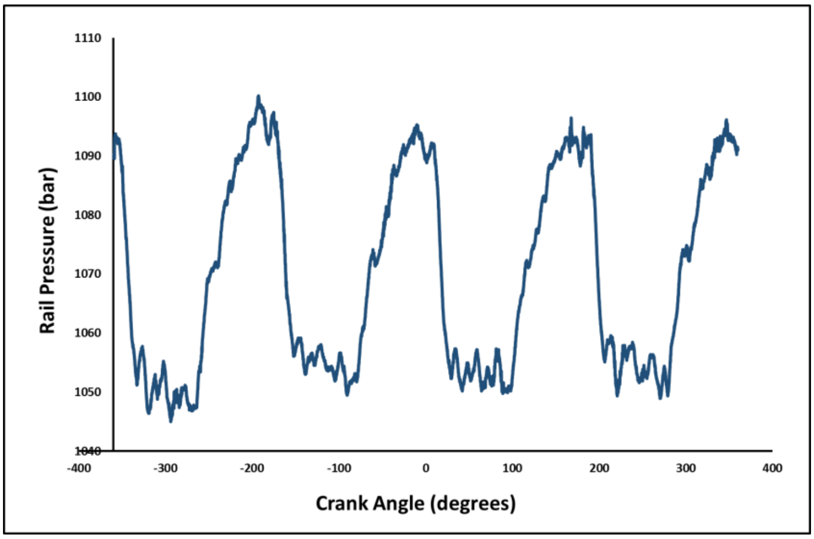

Fig.8 Variation of Rail Pressure in one cycle in mHawk Engine

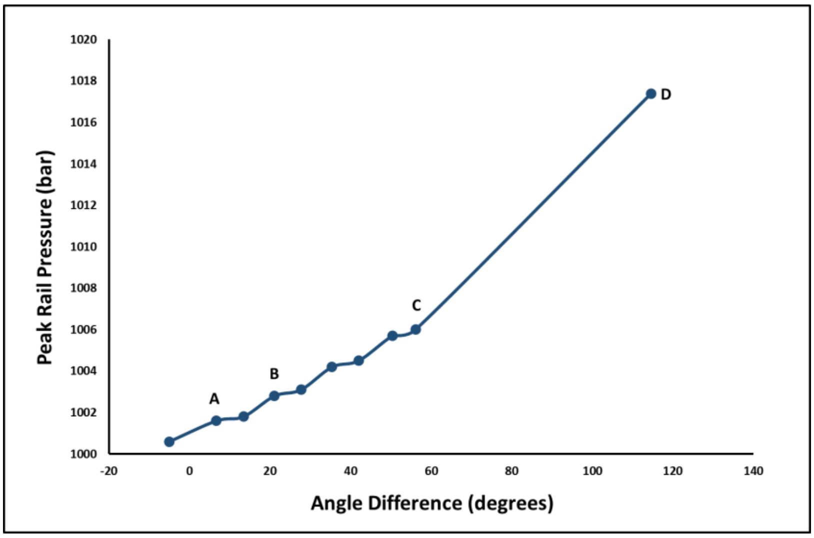

Fig.9 Variation of Average Rail Pressure with Injection Timing

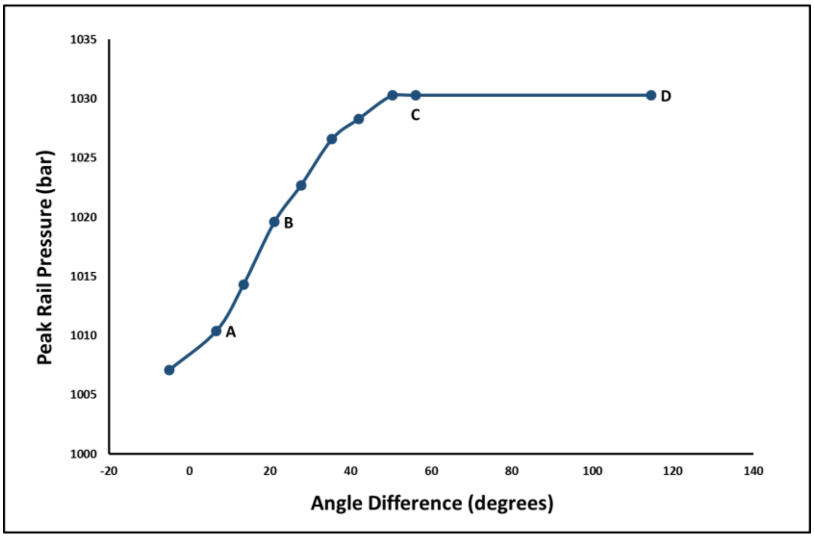

Fig.10 Variation of Peak Rail Pressure with Injection Timing

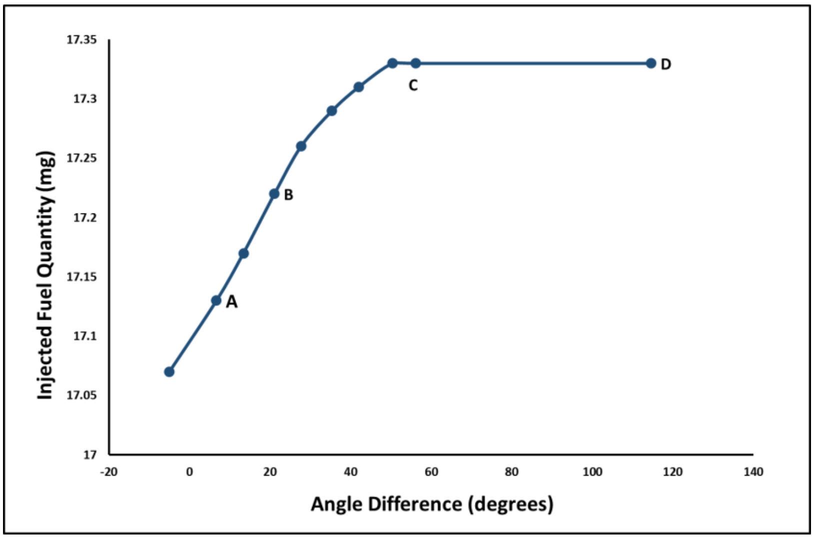

Fig.10 Variation of Fuel Quantity Injected with Injection Timing

The effect of injection timing on the quantity of fuel injected, average rail pressure and peak rail pressure was studied by keeping the duty cycle of IMV, base rail pressure and injection duration constant. The x axis for the Figs.9, 10 and 11 is the difference in the angle between the start of pumping and the start of injection for cylinder 1.When the start of injection is retarded from the end of pumping, the average rail pressure, peak rail pressure and injected fuel quantity are high as seen in Figs.9, 10 and 11. It is also seen that if the start of injection is advanced with respect to the start of pumping, average rail pressure, peak rail pressure and fuel injected fuel quantity decrease. Thus injection timing with respect to the start of pumping plays a key role in determining the amount of fuel injected and injection pressure. These sort of small variations can be modelled and accommodated in the ECU. This indicates the usefulness of the present model of the injection system.

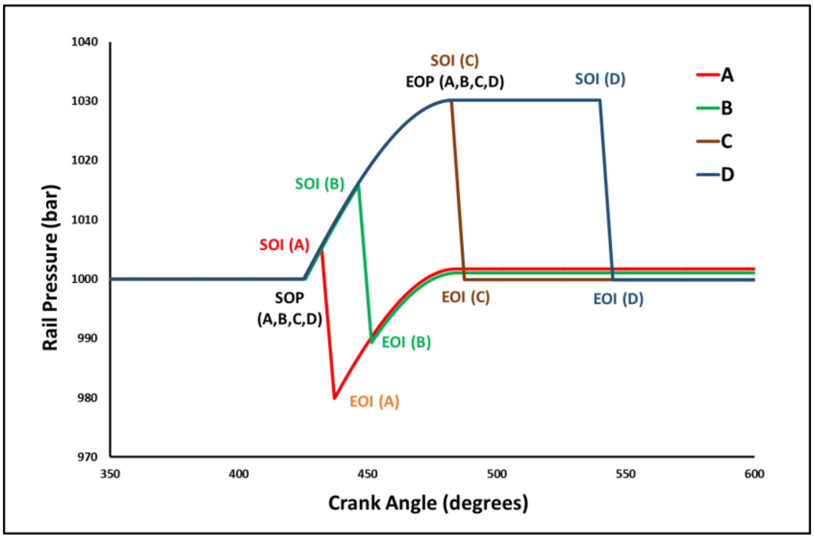

Fig.12 Variation of Rail Pressure (one fourth cycle) for 4 Injection Timings

In Fig.12, simulation results for rail pressure for one fourth of the engine cycle were shown for four injection timings A, B, C and D. The injection timing was retarded with respect to the start of pumping from A to D and the start of pumping was same for all the four timings. The variation of rail pressure around the base rail pressure was nearly the same for the 4 timings. As the injection timing was retarded from A to D, the peak rail pressure, average rail pressure and the quantity of fuel injected was increasing.

CONCLUSION

A pump and rail model was developed for a common rail injection system. The mass of fuel delivered by the high-pressure pump model was validated. The developed model can be tuned with limited experimental data and used for HIL applications. The effect of injection timing on the average rail pressure, peak rail pressure and injected fuel quantity were studied. The variation in the rail pressure is influenced by the timing of the injection process with respect to pumping. However, the amplitude of fluctuations is not significantly influenced. The mass of fuel delivered is affected by the timing of the injection process due to the nature of variations in the rail pressure. Thus corrections for this phenomenon may be needed in controllers.

REFERENCES

1. Baur, R., Blath, J., Bohn, C., Kallage, F. et al., "Modeling and Identification of a Gasoline Common Rail Injection System," SAE Technical Paper 2014-01-0196, 2014, doi:10.4271/2014-01-0196.
2. Chiavola, O. and Giulianelli, P., "Modelling and Simulation of Common Rail Systems," SAE Technical Paper 2001-01-3183, 2001, doi:10.4271/2001-01-3183.
3. Huhtala, K. and Vilenius, M., "Study of a common rail fuel injection system," SAE Technical Paper 2001-01-3184, 2001, doi:10.4271/2001-01-3184.
4. A technical overview of Common Rail Diesel Fuel System presented by Tony Kitchen (AK Training)

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

IMV	–	Inlet Metering Valve
PCV	–	Pressure Control Valve
TDC	–	Top Dead Centre
BDC	–	Bottom Dead Centre
PWM	–	Pulse Width Modulation
SOI	–	Start of Injection
SOP	–	Start of Pumping
EOI	–	End of Injection
EOP	–	End of Pumping
CA	–	Crank Angle