MODELLING OF COMMON RAIL FUEL INJECTION SYSTEM AND INFLUENCE OF FLUID PROPERTIES ON INJECTION PROCESS

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**ABSTRACT**

This paper focuses on the modelling of a research type Heavy Duty Common Rail (CR) fuel injection system. More specifically it reports on the observed interaction between fuel properties and injection and on the capability to model this. For that reason a hydraulic model of the fuel injection system has been developed using the AMESim code (Imagine S.A., 2003).

The reliability of the numerical results is tested through a comparison between numerical and experimental results when using regular diesel fuel. Basis for this detailed comparison are measurements of injected mass flow rate, needle lift and pressure oscillations in the injection duct for a single injection. Simulation results for regular diesel show good agreement with measured data for pressure oscillations in the injection duct, needle lift and injected fuel mass flow rate.

A comparison of experimental and simulated results for Rapeseed oil Methyl Ester (RME) also shows good correspondence, which proves the capability of the model to capture the influence of different fuel properties.

Keywords: Fuel injection, Modelling, Fluid properties

\section{INTRODUCTION}

With modern diesel engines the injection process (i.e. the injection rate and injection pressure) has a major impact on noise production, exhaust gas emissions and fuel consumption. In view of the ever-increasing demands on these engines, modelling of the fuel injection system has become an essential step in the fuel injection equipment design and optimisation process.

As part of the move towards greenhouse gas reduction and diversification of energy supply there is a growing interest to test and enhance the ability of diesel engines to run on alternative fuels.

Because fuel properties such as density, bulk modulus of elasticity and viscosity influence the injection behaviour, the use of an alternative fuel will affect the injection process. Szybist et al. [1] found that injection timing advanced and injection duration shortened with increased biodiesel content.

The higher viscosity of biodiesel is believed to be responsible for the change in injection timing by Choi et al. [2], but no mechanism is offered for this conclusion. Arcoumanis et al. conclude that only fluid bulk modulus of elasticity is responsible for a change in injection timing. Rapokoulos and Hountalas [5] state that the only fluid properties that are of importance for modeling of fuel line pressure are fluid density and fluid bulk modulus of elasticity. Again, fuel viscosity was not found to affect injection timing. All of the above analyzed pump-in-line fuel injection systems.

Ziejewski et al. [6] studied the discharge coefficient of a diesel injector nozzle for laminar and turbulent flows and for different (alternative) fuels. He found that for a specific injector nozzle geometry a relation between Reynolds number and discharge coefficient could be established that was valid irrespective of fuel properties.

The goal of the present study is to validate our hydraulic, one-dimensional, model of a Common Rail (CR) type injection system using both standard diesel and RME and to analyze differences in the fuel injection behaviour related to the difference in fluid properties. The capability of the model to capture these differences is tested.

\section{COMMON RAIL INJECTION SYSTEM}

The studied fuel injection system is of the Common Rail (CR) type and is designed for the fuel delivery to a one-cylinder 2.1-litres Heavy Duty (HD) research diesel engine. In figure 1, this injection system is shown schematically.

The system comprises a \textsuperscript{2}nd generation light-duty CR high-pressure pump with an electronically controlled throttling valve to adjust the delivered...
mass flow rate. The injector is an 8-hole HD diesel engine CR sac-hole nozzle injector. The nozzle holes have a nominal diameter of 0.184 mm, a length/diameter ratio of \( \approx 5 \) and \( \approx 7\% \) inlet rounding. Since in this study only stationary conditions are to be tested, an adjustable mechanical pressure limiter is used to control the pressure in the system. A heat exchanger is included to control the temperature in the fuel injection system. Maximum injection pressure is 1400 bar.

3 MODEL OF CR INJECTION SYSTEM

The high-pressure part of the CR injection system depicted in figure 1 is modelled using the AMESim code [6] (Imagine S.A., 2004). In this code each physical component of the system is represented by an appropriate icon, and is associated to one or more lumped parameter models (called submodels). Principally, in AMESim hydraulic systems can be modelled by isothermal and/or adiabatic submodels that are either one-dimensional (fuel lines) or zero dimensional (restrictions, volumes). Because of the fast nature of the injection process, adiabatic submodels have been used in this study.

In figure 2 a schematic representation of the CR injector is given together with details from the corresponding hydraulic AMESim model. When the solenoid in the top of the injector is energized, the resulting magnetic force lifts the ball valve from its seat. Because the flow rate through the Z-throttle is smaller than the flow rate through the A-throttle, the pressure in the control chamber drops. Because the rail pressure is still present at the needle tip, the needle is pushed upwards, starting the injection. As the current through the solenoid is stopped, the solenoid spring forces the ball valve back on its seat. As a result, the pressure in the control chamber increases again and the needle is pushed down on its seat, thus stopping the injection.

In the graphical representation of the AMESim model in figure 2, the physical elements of the

\[
\dot{m} = \rho \cdot C_d(\lambda) \cdot A \cdot \frac{2|\Delta p|}{\rho} \sqrt{\frac{1 + CN}{CN}} \tag{2}
\]

with \( C_d \) the discharge coefficient at the limit of cavitation \( (CN \to \infty) \). In the injector model, cavitation is modelled in the nozzle holes and A- and Z-throttle.

All capacitive and restrictive components can be connected by long fuel lines. In these one-
Dimensional hydraulic lines pressure wave dynamics is taken into account. Dynamics of motion is evaluated in mass components. The spring-damper component in figure 2 models the elasticity of the control plunger and injector needle. In so-called ‘transformation elements’, such as piston components, a pressure is transformed into a force. Other components in the injection system are modelled in an analogous manner [6].

4 INJECTION MEASUREMENTS

Injection measurements with both standard diesel and Rapeseed oil Methyl Ester (RME) have been performed to validate the model. In table 1 the relevant physical properties of both fuels are presented.

Table 1. Fluid properties of used fuels.

<table>
<thead>
<tr>
<th>Fluid property (20°C)</th>
<th>Diesel</th>
<th>RME</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density [kg/m³]</td>
<td>823.4</td>
<td>882.5</td>
</tr>
<tr>
<td>Bulk modulus [MPa]</td>
<td>1393.0</td>
<td>1522.5</td>
</tr>
<tr>
<td>Kinematic viscosity [cSt]</td>
<td>4.59</td>
<td>6.57</td>
</tr>
</tbody>
</table>

The signals used to validate the model are the measured pressure in the injection duct, at a position 120 mm from the injector entrance, the needle lift and the pressure inside the Zeuch chamber, see also figure 1. From the latter, the injected mass or volumetric flow rate can be determined using the Zeuch method [8]. In this method injections are performed into a pressurized chamber (Zeuch-chamber) filled with fuel. Since geometrical parameters are known, the only unknown parameters are discharge coefficients and loss coefficients of the different restrictive components. These parameters are obtained by tuning the results from the AMESim CR model to the injection measurement data.

4.1 Results for standard diesel

In figure 3 the measured and simulated injection rate is depicted for a range of rail pressures, typical for modern diesel engines. The hold current of the input signal to the injector is held constant at 3 ms for all rail pressures, resulting in 3.8 ms injection duration.

Both the simulated start and end of the injection correspond well to the measurements. Also, the main fluctuations during full needle lift are captured well by the model.

The higher order frequencies in the measured injection rates are caused mainly by pressure fluctuations that arise inside the Zeuch chamber during injection. A possible explanation for this is the presence of cavitation inside the Zeuch chamber. Also, rapid opening and closing of the injector induces pressure waves inside the chamber.

The model of the Zeuch chamber does not capture these phenomena.

The lower order frequencies in the injection rate are caused by fluctuations in injection pressure. The link between rail pressure and injection rate is of course through the different obstructions and lines in the injector. Figure 4 shows the measured and simulated pressure in the injection duct.

The main frequency of the pressure waves, ~850 Hz, is equal to the frequency of a standing wave in a one-sided open duct when the rail acts as the open end and the injector nozzle holes as the closed end. As shown in figure 4, the end of injection is characterised by a fast pressure increase caused by the rapid closing of the injector (‘water hammer-effect’). From figure 4 it can be concluded that simulations and measurements correspond well. Both frequency and amplitude of the pressure fluctuations are determined well by the model.
waves are calculated well. Accurate modelling of the frequency of the pressure waves is mainly dependent on the density and bulk modulus of the fluid, which determine the wave travelling velocity. For the amplitude (i.e. damping) of the pressure waves between rail and nozzle exit, the loss coefficient of the edge filter (see figure 2) was found to have a large influence.

The critical cavitation number is taken equal to 3, which is a common value for diesel injectors, see e.g. [7]. Mean cavitation numbers during the injection range from 13 for 800 bar rail pressure to 20 for 1400 bar rail pressure (downstream pressure 40 bar). These values are well above the critical cavitation number indicating that the flow is cavitating. Therefore, for modern diesel engines the operating range is well outside the range discussed by Ziejewski et al. [5] and discharge coefficients are not a function of Reynolds numbers only.

Injection start is determined by the elasticity of the control plunger and injector needle (see figure 2). As can be seen in figure 4, a slight decrease in pressure is already present before the actual injection starts. When the pressure in the control chamber of the injector drops as a result of the lifted ball valve, the control plunger will first elongate before it actually lifts.

![Figure 5. Measured and simulated control plunger displacement.](image)

Figure 5 shows the measured and calculated displacement of the control plunger for two different rail pressures. At zero rail pressure the maximum lift of the control plunger is 0.25 mm. During injection, the control plunger and needle are still deformed by the rail pressure working at the bottom of the injector needle. This compression causes the injection end to correspond with larger displacements than injection start, as is shown in figure 5. This is because the displacement of the control plunger is measured at the top of the plunger and does not correspond to the actual displacement of the needle tip.

From figure 5 it can be concluded that the control plunger displacement is simulated well for 800 bar rail pressure, but for the case of 1400 bar rail pressure, the calculated lift is slightly slower. Calculations at other rail pressures confirm a trend of greater differences in lift at higher rail pressures. The slower calculated needle lift causes the injection rate to increase slower than the measured rate during needle lift, as can be seen in figure 3. The important model parameters here are the discharge coefficients of the A- and Z-throttle. Calculated cavitation numbers indicate that the flow is non-cavitating in both throttles. Tuned discharge coefficients, corresponding to fully turbulent flow, are 0.84 for the A- and 0.75 for the Z-throttle.

### 4.2 Results for RME

Injection measurements have also been performed using Rapeseed oil Methyl Ester (RME). The AMESim CR model is again tuned to these results, now using RME fluid properties.

![Figure 6. Measured and simulated injection rate for RME at different rail pressures.](image)

Figure 6. Measured and simulated injection rate for RME at different rail pressures.

Also for RME good correspondence between measured and simulated injection rates is found as can be concluded from figure 6 and figure 7. Start and end of injection match well. Amplitude and main frequency of the fluctuations in the injection rate are well captured by the model. This indicates that the injection pressure is simulated correctly. The amplitude of the higher order frequencies seem be higher than for standard diesel. This can be an indication for more severe cavitation.

Control plunger displacement shows the same trend as standard diesel fuel. For higher rail pressures the calculated control plunger lift lags behind, but more than for standard diesel fuel.
4.3 Discussion

For modelling of the injection rate, discharge coefficients of the nozzle holes and injector needle tip passage are key parameters. Tuned discharge coefficients for the needle tip and injector nozzle holes are respectively 0.9 and 0.835. These values are relatively high. A critical cavitation number of 3 has been used, which means that the discharge coefficient of the nozzle holes would be equal to 0.96 for fully turbulent and non-cavitating flow, see equation (4). For the length/diameter ratio of the used nozzle hole geometry the correlation of Lichtarowicz [9], for non-cavitating flow and sharp-edged nozzle holes, gives a discharge coefficient of 0.78. For fully cavitating flows discharge coefficients of ~0.73 are found, e.g. by Favennec et al. [7].

No real improvement could be made to lower the tuned values of the discharge coefficients. Since the pressure at the injector entrance is simulated correctly, lower values for the discharge coefficient of the nozzle holes can only be obtained by a smaller pressure drop inside the injector. However, even when no losses are assumed, a decrease of only 3.6% in discharge coefficient is realised.

Also the uncertainty in measured injection rate (<3.8%), which is merely present through an uncertainty in used fluid bulk modulus of elasticity, is too low to be significant. Because the results for RME also give relatively high discharge coefficients, the bulk modulus of the fluid is not taken as the error source. On the other hand, Goney et al. [8] also found high values for the discharge coefficients (0.8 – 0.925) for cavitating flows in a sharp-edged nozzle hole and 0.85 – 0.975 for non-cavitating flows through a nozzle hole with high inlet rounding. Our nozzle geometry lies somewhere in between, as do our tuned discharge coefficients. In contrast to many other authors Goney et al. use actual sac-pressure measurements. Gianippa et al. [11] found discharge coefficients around 0.78-0.8 for the same range of cavitation numbers as calculated in our measurements. In view of the above and for lack of more data, tuned discharge coefficients were considered to be acceptable and they were retained. Experiments are being conducted for further evaluation of the capability of the AMESim code to model the injection system.

5 INFLUENCE OF FLUID PROPERTIES

The influence of the difference in fluid properties between standard diesel and RME are most evident when the volumetric injection rates are compared, see figure 8. For RME a decrease in mean injection rate of 6.9% with respect to diesel fuel is present.

The higher density of RME causes the volumetric injection rate to be lower. From the lower flow velocities due to the higher viscosity and density of RME it is expected that cavitation is less severe and discharge coefficients are higher. However, in the CR model, the discharge coefficient for maximum cavitation of the injector nozzle holes had to be lowered by 3.2%. A possible explanation for this decrease can be a higher air release during injection because of more dissolved gas in the RME.

In contrast to Szybist et al. [1] and Arcoumanis et al. [3], no difference in injection start can be distinguished between standard diesel and RME. Because in CR systems the injector needle is not lifted through a pressure increase by the pump (as in pump-in-line systems), injection timing is not influenced by the compressibility of the fuel. The increase in flow rate during needle lift is slightly slower for RME. This is the result of a
slower needle lift, as can be seen in figure 9. The slower lift is caused by lower volumetric flow rates through the A- and Z-throttle as a result of the higher density of RME. This results in a slower pressure drop in the control chamber of the injector. Also, the flow rate into the sac-volume is lower. The pressure rise in the sac-volume is therefore slower and the needle is pushed upwards with less force. Because the flow through the A- and Z-throttle is turbulent almost immediately, the viscosity of RME only has a marginal influence.

Figure 9. Measured control plunger movement for standard diesel and RME.

In general, differences are small and it can be concluded that the use of RME does not have a large effect on the injection behaviour. To observe more pronounced differences, a wider spread of fuel viscosity and density must be used.

6 CONCLUSIONS
The model of the CR injection system presented in this study gives good results for both standard diesel fuel and Rapeseed oil Methyl Ester. Though, differences between RME and diesel fuel are small. It is therefore expected that also blends of standard diesel and RME will be simulated correctly. Used discharge coefficients for the injector nozzle holes and needle tip passage are found to be quite high, but resulted in acceptable calculation results. For an optimal simulation when using a different fuel, experimental results are necessary to determine fluid properties and discharge coefficients of the different components.

In CR systems injection timing is not influenced by fluid bulk modulus of elasticity and in the normal operating range of diesel engines, fluid viscosity only has a marginal influence on needle lift because the flow is turbulent almost immediately.

For research on the influence of fluid properties, a wider spread in fluid properties must be present.

This research will be continued for other alternative fuels with a wider variation of fuel properties.

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REFERENCES