Detailed modeling of common rail fuel injection process

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Detailed Modeling of Common Rail Fuel Injection Process

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ABSTRACT

Modeling of fuel injection equipment is a tool that is used increasingly for explaining or predicting the effect of advanced diesel injection strategies on combustion and emissions. This paper reports on the modeling of the high-pressure part of a research type Heavy Duty Common Rail (CR) fuel injection system. More specifically, it reports on the observed dynamics of the injection system and the capability of the model to capture this. For that reason, the total high-pressure part of the injection system, i.e. the fuel pump, rail and injector, has been modeled using the AMESim code (Imagine S.A., 2004). The reliability of the resulting hydraulic model is tested through a comparison between numerical results and actual injection measurements. This detailed comparison is based on measurements of injected mass flow rate, needle lift and pressure oscillations in the injection duct for a series of single injection events.

It is shown that the hydraulic model is able to accurately simulate the injection rate, needle lift and injection pressure for different rail pressure levels. For accurate numerical results, it is vital that the stiffness of the injector needle assembly and the discharge coefficients of the different flow restrictions in the injector (e.g. nozzle holes) are correctly modeled. Assuming a rigid injector needle results in a too early start of injection. Discharge coefficient values found in literature shows a wide spread. This makes it very difficult to simulate the injected mass flow rate accurately on the basis of literature data. Using the measured injected mass of fuel to tune the discharge coefficient, together with the inclusion of the elasticity of the injector needle, results in a good approximation of the injection rate.

KEYWORDS

Common Rail fuel injection, Modeling, Discharge coefficient

1. 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, fuel consumption and engine efficiency. In view of the ever-increasing demands on these engines, modeling of the fuel injection system has become an essential step in the fuel injection equipment design and optimization process. The large number of technical papers regarding the modeling of the dynamic behavior of the injection system (e.g. Borghi et al. [1], Catania et al. [2] and Cantore et al. [3]) confirms this.

Detailed modeling of the diesel spray development and combustion process inside the diesel engine (e.g. phenomenological or 3D CFD modeling) requires the injection rate and velocity (i.e. momentum) to be known as an input (e.g. Wickman et al. [4] Hiroyasu et al. [5], Barba et al. [6]). Therefore, correct representation of the injection rate (and velocity) for a given operating condition (rail pressure and injection duration) is of great importance. For this reason, it was decided to develop a one-dimensional hydraulic model of a Common Rail fuel injection system that is capable of correctly simulating the injection rate as a function of rail pressure.

In this article the different steps towards realizing such model are presented. First, the CR test rig that was developed will be described together with a general description of the corresponding hydraulic model. Thereafter the model validation process is discussed. A comparison between measurements and
numerical results is presented. The validation of the used injector nozzle hole discharge coefficient and the influence of cavitation on the actual value of this discharge coefficient are also discussed in great detail. Finally an overview of the main conclusions is given.

2. COMMON RAIL TEST FACILITY

The fuel injection system being analyzed is of the common rail type. It has been designed for fuel delivery to a single cylinder 2.1-litres HD diesel research engine. In figure 1, this injection system is shown schematically. The system uses a 2nd generation light-duty CR high-pressure fuel pump with an electronically controlled throttling valve to adjust the delivered mass flow rate. A frequency-controlled electrical engine drives the CR pump. The injector is connected close to the centre of the rail. 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 an inlet rounding of approx. 7%. A heat exchanger is included to control the temperature in the fuel injection system. Maximum injection pressure is 1400 bar. Single shot injections are performed into a closed chamber, the so-called Zeuch-chamber. This chamber is filled with fuel at a certain pressure (20 – 40 bar). From the measured pressure rise in the chamber during injection, the injection rate can be calculated. This method for injection rate determination is an alternative to the BOSCH method. The size of the chamber is chosen such that it fits HD like injection events: the internal volume is equal to 1.87·10⁻³ m³.

A Micro-Epsilon eddy-current needle lift sensor type U05 measures the injector’s needle lift. The pressure in the Zeuch-chamber is measured by a Kistler 7061 quartz pressure transducer and the pressure in the injection duct between rail and injector is measured by a Kistler piezoresistive pressure sensor type 4076.

![Figure 1: Common Rail fuel injection system.](image)

3. COMMON RAIL INJECTION SYSTEM MODEL DESCRIPTION

Figure 2a shows a typical HD CR fuel injector with electromagnetic fuel injection control. In closed position, the rail pressure is present in both the control chamber above the control plunger and the nozzle chamber. Because the area of the top of the control plunger is larger than the area of the needle-shoulder in the nozzle chamber, a net closing force is present. The needle tip is pushed on its seat and no injection of fuel can take place. 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 (see figure 2a), the pressure in the control chamber drops. The rail pressure is still present in the nozzle chamber and the needle is pushed upwards, starting the injection of fuel. 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.

The CR fuel injector together with the total high-pressure part of the CR injection system shown in figure 1 (high-pressure pump, rail, ducts and Zeuch-chamber) is modeled using the AMESim code [7] (Imagine S.A., 2004). In figure 2b a graphical presentation of the CR injector is shown. In the AMESim 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 modeled by isothermal and/or adiabatic submodels that are either one-dimensional (fuel lines) or zero-dimensional (restrictions, volumes). Because time scales in the injection process are very short, adiabatic submodels have been used to model the injection system.

\[ \frac{dp}{dt} = B(p,T) \left[ \frac{1}{\rho} \frac{m_{in} - m_{out}}{V} + \alpha(p,T) \frac{dT}{dt} \right] \]

where \( m_{in} \) and \( m_{out} \) are respectively the ingoing and outgoing mass flow rates of the volume \( V \). Fluid properties in equation (1) are bulk modulus \( B \) , density \( \rho \) , and cubic expansion coefficient \( \alpha \).

The temperature of the fluid in the volume is computed through solving the energy equation:

\[ \frac{dT}{dt} = \frac{(m_{in} - m_{out})}{\rho V c_p} - \frac{(dm/dt) \cdot h + \dot{Q}}{\rho V c_p} + \frac{T \cdot \alpha}{\rho c_p} \frac{dp}{dt} \]

**Figure 2a (left):** Cross-section of CR injector.

**Figure 2b (right):** AMESim CR injector model.
with $h$ the enthalpy, $c_p$ the specific heat at constant pressure and $\dot{Q}$ the heat flow from the surroundings into the volume.

In resistive elements, such as a throttle, only the mass flow rate is calculated, using the modified Bernoulli equation:

$$\dot{m} = \rho \cdot C_d(\lambda) \cdot A \cdot \sqrt{\frac{2|\Delta p|}{\rho}}$$  \hspace{1cm} (3)

where, $C_d$, is the discharge coefficient. For non-cavitating flow, $C_d$ is dependent on flow velocity, fluid density and viscosity only. This dependency is taken into account by formulating the discharge coefficient as a function of a dimensionless flow number, $\lambda$:

$$\lambda = \frac{d_h}{v} \sqrt{\frac{2|\Delta p|}{\rho}}$$  \hspace{1cm} (4)

where $d_h$ is the hydraulic diameter and $v$ is the kinematic viscosity. The cross-sectional area, $A$, can be constant or a function of lift, as in the needle-tip component.

To correct for the occurrence of cavitation, the so-called cavitation number $CaN$ is used. In this study, as in AMESim, this dimensionless number is defined as the ratio of the pressure difference over the restriction to the downstream pressure:

$$CaN = \frac{P_{up} - P_{down}}{P_{down}}$$  \hspace{1cm} (5)

In case of the injector nozzle holes, $P_{up}$ would be equal to the pressure in the sac-volume (see figure 2b) and $P_{down}$ would be equal the pressure in the volume component that models the Zeuch-chamber volume (not shown in figure 2b).

If the cavitation number is higher than a pre-defined critical cavitation number, $CaN_{critical}$, equation (3) in AMESim changes to (see [7],[8]):

$$\dot{m} = \rho \cdot C_d^c \cdot A \cdot \sqrt{\frac{2|\Delta p|}{\rho}} \cdot \sqrt{\frac{1 + CaN}{CaN}} \hspace{1cm} CaN \geq CaN_{critical}$$  \hspace{1cm} (6)

where $C_d^c$ is the discharge coefficient of the restrictive element for fully cavitating flow, i.e. at the limit of cavitation ($CaN \rightarrow \infty$). In the injector model, cavitation is modeled in the nozzle holes and the A- and Z-throttle.

Finally, all capacitive and restrictive components can be connected by long fuel lines. In these hydraulic lines pressure wave dynamics is taken into account. Assuming one-dimensional flow, the equation of motion of a hydraulic line is:

$$\frac{dq}{dt} - \frac{A}{\rho} \frac{dp}{dx} + u \frac{dq}{dx} + A \cdot g \cdot \sin(\theta) + q = 0$$  \hspace{1cm} (7)
with \( q \) the volumetric flow rate, \( \theta \) the angle the line makes with the horizontal and \( h(q) \) the viscous friction term which is dependent on the relative roughness of the pipe wall. The continuity equation is:

\[
\frac{dp}{dt} + u \frac{dp}{dx} + \rho c^2 \frac{dq}{A} \frac{dx}{dx} = 0
\]

(8)

where \( c \) is the speed of sound and \( A \) the cross sectional area of the pipe.

Fluid pressure is ‘transformed’ into a force in so-called ‘transformation elements’, such as the piston components in figure 2b. In the mass components the dynamics of motion are evaluated. In figure 2b a linear mechanical spring-damper element can be seen. This element models the elasticity of the control plunger / needle assembly in the injector. Other components in the injection system, such as the high-pressure pump and the Zeuch-chamber, are modeled in a similar manner [7].

4 MODEL VALIDATION

In the following paragraphs the model validation will be discussed. First the necessary data required for the validation will be described followed by a comparison between measurements and numerical results. Additional measurements that were performed for determining the correct values of important model parameters are described in detail.

4.1 FUEL PROPERTIES AND EXPERIMENTAL DATA

In a fuel injection system, pressures, and to a lesser extent temperatures, can vary considerably. This results in a wide variety of important fuel properties such as density, bulk modulus of elasticity and kinematic viscosity. Having good fluid property models over a large range of pressures and temperatures is therefore essential for accurate modeling. Figure 3 gives an overview of the important fluid properties of the fuel as used in the AMESim environment. The fluid properties shown in figure 3 show good correspondence with correlations presented by Rodriguez-Anton et al. [9] (max. deviation <0.3%). For ambient conditions, measured values for the density of the actual test fuel also match well (deviation < 0.7%) with the calculated values.

![Figure 3: Density (left), bulk modulus of elasticity (middle) and kinematic viscosity (right) of used diesel fuel as function of pressure and temperature. Taken from AMESim [7].](image)

Besides the fluid properties, also information about the geometry of the modeled components is necessary. As can be seen in figure 2b, in AMESim the different components in the fuel injection system are built up using several elementary building blocks, such as restrictions, volumes, etc. Most of these
building blocks require geometrical data. For example, modeling of an injector nozzle hole requires the knowledge of the diameter of the equivalent restriction. The necessary geometrical data has been obtained from work drawings or by disassembly of the component and performing actual measurements. For small critical components such as the A- and Z-throttle and the injector nozzle holes, an electronic microscope has been used.

An important element for injection timing is the electromagnetically lifted ball valve at the top of the injector. The maximum lift and lift velocity of this ball valve is difficult to determine. Therefore this process was not modeled but additional literature data on a similar valve has been used as a reference ([10], [11]).

Finally, experimental data of the injection system is required to validate and tune the model. Injection measurements have been performed into the Zeuch-chamber. During these measurements three signals were logged:

1) **Control plunger movement:** an eddy-current needle lift sensor measures the movement of the top of the control plunger. This signal is used to determine the injection timing and needle tip lift as a function of time.

2) **Injection pressure:** the pressure in the injection duct between rail and injector is measured at a position of 120 mm from the entrance of the injector. Oscillations in this pressure will influence the injection rate.

3) **Pressure in the Zeuch-chamber:** from the pressure rise in the Zeuch-chamber during the injection, the injection rate can be determined by using the following equation [12]:

\[
\frac{dm}{dt}_{\text{injected}} = \rho \cdot \left( \frac{dV}{dp} + \frac{V}{B} \right) \cdot \frac{dp}{dt}
\]

with \( V \) the volume and \( dV/dp \) the stiffness of the Zeuch-chamber. Injections are performed into a closed chamber filled with fuel at a certain pressure. The stiffness of the Zeuch-chamber, i.e. the \( dV/dp \) factor, has been determined from Finite Element Method (FEM) calculations [13] to be sufficiently high to have a negligible influence. This method represents an indirect measurement of the injection rate. From equation (9) it is clear that the accuracy of calculated mass flow rates is mainly determined by the uncertainty in density and modulus of elasticity, which are functions of both pressure and temperature. During the determination of the injection rate, the density and fluid modulus of elasticity are taken as a function of the measured Zeuch-chamber pressure. Through this, the increase in density and elasticity of the fluid inside the chamber, as a result of the injected mass into the closed volume, is taken into account. The temperature in the chamber is assumed to remain constant during an injection. Analysis has shown that the inaccuracy for determined mass flow rates is \( \pm 2.3\% \). The inaccuracy is mainly caused by the uncertainty in temperature of the fluid inside the Zeuch-chamber. This temperature is not directly measured.

In the following paragraphs the numerical results for the control plunger movement, injection pressure and injection rate will be compared with measured data. During the injection measurements the hold current of the input signal to the injector is held constant for 3.0 ms. This results in injection durations of about 3.8 ms.

### 4.2 CONTROL PLUNGER MOVEMENT

Figure 4 shows the measured and simulated displacement of the top of the control plunger for rail pressures of 1400 bar and 800 bar. In this figure it can be clearly seen that the start of fuel injection into the Zeuch-chamber (this is determined from the measured pressure rise in the chamber) does not correspond to the start of the control plunger movement. This is caused by the elastic deformation of the control plunger and injector needle by the rail pressure prior to injection. Because of this elasticity, the
measured movement of the top of the control plunger does not correspond to the actual movement of the needle tip.

As can be seen in figure 2b, the elasticity of the control plunger/needle assembly is modeled using one equivalently linear spring-damper element. The stiffness of this linear spring has been determined from experimental data. In a series of experiments, the displacement of the top of the control plunger was measured with rail pressure increased from 0 to 1400 bar. No injections were performed during these experiments. In figure 5 the elastic deformation is shown as a function of rail pressure. It can be concluded that this deformation is indeed a linear function of the rail pressure. Note that at a rail pressure of 1400 bar the initial deformation of the control plunger and needle are about 36%(!) of the total measured displacement.

In figure 5 also the measured displacement of the control plunger until the measured start of injection (SOI) is depicted. Obviously, as a result of the control plunger/needle elasticity, the actual start of injection is dependent on the rail pressure. However, because of the rapid displacement of the control plunger the actual point in time of start of injection only varies slightly with rail pressure (order of 0.01 ms).

During injection, the rail pressure also deforms the control plunger/needle assembly from the bottom, shortening the control plunger/needle. This results in a longer measured downward movement than the measured upward control plunger movement from injection start to maximum needle lift. This can also be seen in figure 4 where the level of injection end is lower than the deformation level at injection start.

After the injection of fuel into the Zeuch-chamber has ended, oscillations in measured control plunger movement can still be seen. They are the result of oscillations in rail pressure, causing oscillating elastic deformation of the control plunger/needle assembly. However, no post injections take place. From figure 4 it can be concluded that the model is well capable of capturing all the occurring phenomena.

![Figure 4 (left):Measured and simulated control plunger movement.](image1)

![Figure 5 (right):Elastic deformation prior to injection as a function of rail pressure.](image2)

### 4.3 Injection Pressure

In figure 6 the measured and simulated injection pressures are shown for 4 different initial rail pressures. In the injection pressure curves the effect of the elasticity of the control plunger/needle assembly can be seen in the time delay between the start of lift of the ball valve (SLB) and the actual start of injection (SOI). Prior to the pressure drop at the start of fuel injection, a small pressure drop can already be observed. This pressure drop originates from the pressure drop in the control volume above the control plunger when the ball valve lifts.

During the injection, pressure oscillations are present. Both the frequency and the amplitude of these pressure waves are simulated correctly. The main frequency of the pressure waves, ~850 Hz, is equal to the frequency of a standing wave in a one-sided open duct where the rail acts as the open end and the injector nozzle holes as the closed end. Accurate simulation of the frequency of the pressure waves is
mainly dependent on density and bulk modulus of the fluid (which determine the wave traveling velocity).

The rapid closing of the injector results in high-pressure oscillations known as the ‘water hammer effect’, which can be clearly seen in both the simulated and measured pressures. For the damping of the pressure waves caused by the closing of the injector, not only the rail volume but also the flow restriction formed by the edge filter (see figure 2a) was found to be of importance. The loss coefficient of this edge filter was tuned to the available injection pressure data. Both amplitude and phase of the pressure oscillation are well captured. Overall it can be concluded that a good agreement between measured and simulated injection pressures is present. Not only as a function of time but also as a function of initial rail pressure level.

![Image](image_url)

**Figure 6:** Measured and simulated pressure in the injection duct for different rail pressures. The dashed vertical lines indicate the Start Lift Ball valve (SLB), Start Of Injection (SOI) and End Of Injection (EOI).

### 4.4 Injection Rate

Finally, figure 7 shows the simulated and measured mass flow rates corresponding to the four injection pressure signals of figure 6. The correct simulation of the injector needle lift can also be seen in the mass flow rate signals. During needle lift, good correspondence exists between simulations and measurements. Figure 7 also clearly shows the influence of the oscillations in injection pressure on the injection rate: the main frequency of the fluctuations in the mass flow rate corresponds to the observed frequency of the oscillations in the injection pressure of 850 Hz. However, higher frequency oscillations are more pronounced in the injection rate. Because it was not possible to correlate these higher frequencies to standing wave frequencies in the injector, it is assumed that these frequencies arise inside the Zeuch-chamber as a result of the injection event. The rapid opening and closing of the injector for instance induces pressure fluctuations in the Zeuch-chamber. Also the implosion of cavitation bubbles inside the Zeuch-chamber is a possible explanation for the higher frequency fluctuations.

To obtain the simulated injected mass flow rates from figure 7, the discharge coefficient of the nozzle holes at maximum cavitation $C_d^c$ $(CaN \to \infty$, see equation 6) was tuned to fit the measured mass flow rates. This fit is based on the average mass flow rate during full needle lift. The value of the nozzle hole discharge coefficient is essential for accurate modeling of the injected mass flow rate during maximum needle lift. A value of 0.765 is used for all simulations in figure 7. This tuned value is compared with actual measurements of the nozzle hole discharge coefficient. Preliminary experimental results confirm the value chosen for $C_d^c$. 
5 DISCUSSION

For accurate modeling of the injection rate as a function of initial rail pressure and time, the elasticity of the control plunger/needle assembly and the value of the nozzle hole discharge coefficients are very important. In this paper, additional measurements have been performed to determine correct values for the required model parameters. However, such experimental data is not always available. In this section it will be analyzed what errors are made when this detailed information is not present.

Usually, the measured lift of the top of the control plunger (“needle lift”) and the injected fuel mass per injection are available from engine tests. Also, because the nozzle hole discharge coefficient is a much-studied subject in literature, many papers can be found that can be used to estimate the value of the nozzle hole discharge coefficient. Unfortunately, values found in literature for the discharge coefficient of an injector nozzle hole show a wide spread. For the length/diameter ratio of the used nozzle hole the correlation of Lichtarowicz [14], which is valid for non-cavitating flow through a sharp-edged nozzle hole, gives a discharge coefficient of 0.78. Nurick [15] presents values for the discharge coefficients of sharp-edged nozzles and fully cavitating flow in the order of 0.62. Discharge coefficients in the range of 0.69 – 0.73 are found for a fully cavitating flow by several authors such as Arcoumanis et al. [16], Favennec et al. [8] and von Kuensberg Sarre et al. [17]. There are also authors who use significantly higher values. Ganippa et al. [18], for instance, mentions discharge coefficients in the order of 0.78-0.8 for cavitation numbers in the range of 10 – 20. Even higher values are used by Goney et al. [19]. They use values in the range of 0.8 – 0.925 for a cavitating flow in a sharp-edged nozzle hole.

Six cases are evaluated during the analysis, as is summarized in table 1. Cases I to III represent the situation when the elasticity of the control plunger/needle assembly is not taken into account. Here, the measured movement of the top of the control plunger is used to prescribe the motion of the needle-tip. For Cases IV to VI, the elasticity of the control plunger/needle assembly is modeled using the measured value for the spring stiffness. Also a division in the nozzle hole discharge coefficient is made. Simulations are performed using a value of 0.7, which is obtained from literature. A discharge coefficient of 0.75, which is in line with the preliminary measurements, is used in cases II and V. Finally, the discharge coefficients that give the best fit between measured and simulated injection quantity are used for cases III and VI.
<table>
<thead>
<tr>
<th></th>
<th>Needle/control plunger assembly</th>
<th>Rigid</th>
<th>Elastic</th>
</tr>
</thead>
<tbody>
<tr>
<td>Literature</td>
<td></td>
<td>Case I</td>
<td>Case IV</td>
</tr>
<tr>
<td>$C_d^c$</td>
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<td>Case II</td>
<td>Case V</td>
</tr>
<tr>
<td>Measured</td>
<td></td>
<td>Case III</td>
<td>Case VI</td>
</tr>
<tr>
<td>Fit on $\Delta m_{inj}$</td>
<td></td>
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</tbody>
</table>

**Table 1:** Overview of evaluated cases.

Figure 8 shows the difference in simulated needle-tip movement for a rigid and elastic control plunger/needle assembly (cases I and IV). Neglecting the elasticity of the control plunger/needle assembly causes a deviation of about 0.2 ms in the injection start. At an engine speed of 2300 rpm, this would represent an error of 2.8 crank angles in the start of injection. This is a significant deviation since the start of injection is most frequently given with a resolution of 0.1 crank angles. Incorporation of the elasticity causes the control plunger to relax first before it actually lifts. This also results in a lower maximum lift of the needle-tip. Fluctuations in needle lift during full opening are higher because the oscillations in sac-volume pressure can deform the control plunger/needle assembly from below. The higher maximum needle lift does not have a significant influence on the injected mass flow rate as can be seen in figure 9. The higher needle lift in the non-elastic case does not result in a significant increase in flow area. Also, the flow area of the nozzle holes is rate limiting. As a result, the pressure at the entrance of the nozzle holes will be nearly identical and no real deviation in mass flow rate will be present.

In cases I to III, the oscillations in the measured control plunger movement after the start of injection, will directly result in a lift of the needle-tip. In figure 9 it can be seen that this causes post-injections to take place, which are not present in reality. In the following discussion, these post-injections will be ignored and only the main injection pulse will be considered.

![Figure 8](image1.png)

**Figure 8 (left):** Simulated needle-tip movement for rigid and elastic needle (cases I and IV).

**Figure 9 (right):** Comparison between measured and simulated mass flow rates for cases I, IV and V.

From figure 9 it can be concluded that the use of the discharge coefficient value from literature ($C_d = 0.7$) results in a too low injection rate during maximum needle lift. This causes the simulated injected quantity too differ significantly from the measured injected mass $\Delta m_{inj}$ of $0.284 \pm 0.007$ grams. It has to be noted here, that the measured injected mass is derived from the injection measurements in the Zeuch-chamber and not from actual fuel consumption measurements on an engine.
<table>
<thead>
<tr>
<th>$C_d^c$</th>
<th>Needle/control plunger assembly</th>
<th>Rigid</th>
<th>Elastic</th>
</tr>
</thead>
<tbody>
<tr>
<td>Literature ($C_d^c = 0.7$)</td>
<td>$\Delta m_{inj} = 0.277 \text{ g (}-2.50%)$</td>
<td>$\Delta m_{inj} = 0.265 \text{ g (-}5.52%)$</td>
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<tr>
<td>Measured ($C_d^c = 0.75$)</td>
<td>$\Delta m_{inj} = 0.294 \text{ g (+}3.52%)$</td>
<td>$\Delta m_{inj} = 0.282 \text{ g (-}0.95%)$</td>
<td></td>
</tr>
<tr>
<td>Fit on $\Delta m_{inj}$</td>
<td>$C_d^c = 0.726$</td>
<td>$C_d^c = 0.759$</td>
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</tbody>
</table>

Table 2: Overview of results of evaluated cases.

Table 2 presents a summary of the simulated injected mass for the different cases. Between brackets the percentage deviation with the measured mass flow rate is given. For cases III and VI, the discharge coefficients are given for which the simulated injection quantity is identical to the measured injected mass. The earlier injection start of case I compared to case IV explains the smaller deviation between the measured and simulated injected mass. This also explains the over prediction of the injected mass by case II. Using the measured value for the discharge coefficient together with an elastic needle (case V), results in a good agreement between measured and simulated injected mass. In figure 9 it can be seen that the mass flow rate during maximum needle lift is slightly under predicted for this case. Therefore, the tuned discharge coefficient from case VI is also slightly higher. It has to be noted that this discharge coefficient is lower than the earlier mentioned tuned discharge coefficient of 0.765 (see paragraph 4.4). The difference between these two tuned discharge coefficients is that the discharge coefficient from paragraph 4.4 is tuned on the flow rate during maximum needle lift. The value from table 2 is tuned on the total injection quantity. A discharge coefficient of 0.765 results in an injected mass of 0.286 grams, which corresponds to an over-prediction of 0.74%. This is still well within the measurement uncertainty for the injected mass. The fact that the two discharge coefficients are different indicates that there are slight deviations between simulated and measured injection rates during opening and closing of the injector.

In general it can be concluded that inclusion of the elasticity of the control plunger/needle assembly is essential for the simulation of the injection timing. The importance of taking the elasticity into account is expected to increase with the increase in maximum common rail injection pressures in future engines (> 200 MPa). The wide spread in discharge coefficient values found in literature makes it very difficult to simulate the mass flow rate and injected quantity with the accuracy required for further analysis, e.g. as input for CFD computations. Tuning of the discharge coefficient on actual measured injection quantities is necessary. This already gives a good approximation of the nozzle hole discharge coefficient and thus of the injection rate.

6 CONCLUSIONS

A hydraulic model of a research type Heavy Duty Common Rail fuel injection system has been developed using the AMESim code. The model is capable of accurately predicting the injection pressure, needle lift and injection rate for varying injection pressure.

For good modeling accuracy, the elastic deformation of the control plunger/needle assembly has to be taken into account. The initial elastic deformation of the control plunger, which shows a linear dependency on rail pressure, results in a time delay between start of control plunger lift and actual needle-tip lift (i.e. start of injection). For a rail pressure of 1400 bar, the initial deformation was found to be 36% of the total movement of the control plunger. However, because of the rapid rise of the control plunger, the influence of rail pressure on the time delay between start of lift and actual start of injection is only small (order of 0.01 ms). With the increase in common rail injection pressures for future engines, the importance of the inclusion of the injector needle elasticity will also increase.

Equally important for accurate results is the value for the nozzle hole discharge coefficient. This discharge coefficient is strongly influenced by cavitation, which decreases the value of the discharge coefficient. The tuned value for the nozzle hole discharge coefficient is relatively high, but is found to be in good agreement with measured values.

The use of the measured control plunger movement as a direct representation of the injector needle-tip motion, results in a significant error in the start of injection. The wide spread in values for nozzle hole
discharge coefficients found in literature makes them a inaccurate basis for injection rate simulations. Using the injected mass of fuel to tune the discharge coefficient together with the inclusion of the
elasticity of the injector needle results in a good approximation of the injection rate.

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