MASTER

Fuel injection equipment characterisation for short and multiple injections
experimental setup design and model-based sensitivity analysis

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Award date:
2020

Link to publication
Abstract
Modern diesel combustion strategies rely on high-pressure direct fuel injection. The trend is towards higher injection pressures, more and shorter injection events per combustion cycle. This requires higher performance from fuel injection systems.

The aim of this project is to design an injector performance characterisation setup, capable of measuring nozzle mass and momentum flux for short and close-spaced multiple injections. The setup must also be usable for determining nozzle coefficients and injection delay. Another aim is to investigate which injection system parameters affect short-pulse injection performance.

This work was performed in the scope of heavy duty engine research at TU/e. Focus lies on heavy-duty diesel applications based on common rail injection systems equipped with solenoid injectors.

For the injection equipment characterisation setup an extensive literature study was performed to find the best suited experimental concept. Jet impingement force measurement was found to be the best experimental technique for measuring momentum flux for short and close spaced multiple injections. Under these conditions the classical ROI measurement techniques (Bosch and Zeuch methods) suffer from pressure wave interference. Mass flux can accurately be determined from the measured momentum flux curve through a simple corrective scaling based on a measured cumulative injected mass value. Individual fuel jets from a multi-orifice injector can be studied. A downside of this method is that the same systematic errors will exist in both $\dot{M}$ and $\dot{m}$ rate shape signals, one has no way of verifying accuracy.

A global setup design is proposed and data processing routines for determining nozzle coefficients and mass flux from measured momentum flux are developed. Setup sizing is approached in view of the required performance characterisation application. For determining nozzle coefficients from a long pseudo-steady state injection, the exact dimensioning of the fuel system were found not to be relevant. Allowing various injectors to be characterised with the same setup. Short injections are significantly affected by fuel line length.

A comprehensive injection system performance analysis is performed based on a developed injection system model. For this a combined zero- and one-dimensional lumped parameter hydraulic-mechanical injection system model was developed in Matlab Simscape. The electromagnetic injector control solenoid is modelled explicitly as a lumped parameter electro-magnetic-mechanical system. Baseline parameters were obtained and tuned based on literature values. A model validation based on literature values shows that the developed model is able to accurately predict the injection rate profile, rail pressure profile and needle position profile under transient and steady state conditions.

An extensive parameter sensitivity study was performed, evaluating influences of system parameters on flowrates, total flow and injection delay for a combined pilot + main injection. Baseline pilot volume 2.5 mm$^3$, baseline main volume 110 mm$^3$.

It was found that the solenoid response has a large influence on short injections where maximum needle lift is not reached. The initial and residual solenoid gap values are important for the rate of valve opening and closure. Active coil-current drawdown was identified as critical parameter to speed up injector closure, the EOI delay can be more than halved.
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1 Introduction

For over half a century, diesel has been the prime source of motive power in the heavy duty sector. With its combination of high efficiency and long service life it has long outclassed all alternative technologies. A downside to the conventional compression-ignition diesel engine is generation of particulate matter (PM) and nitrous oxides (NOx) during combustion. Concerns about public health hazards and environmental effects of these air pollutants have driven legislators to impose ever stricter limits on PM and NOx emissions.

Since the introduction of the first European and US emissions legislation in the late eighties, manufacturers and research institutions have been working on developing the diesel engine. At TU/e, research into advancements of heavy duty diesel engines is conducted by the Power & Flow group, combining novel experimental techniques with advanced model simulations.

The major challenge is to simultaneously reduce PM and NOx emissions whilst retaining high fuel-efficiency, which are generally conflicting requirements for conventional diesel combustion. Exhaust gas after-treatment systems are effective at removing pollutants but costly. As such the focus remains on reducing engine-out emissions. Advanced combustion concepts have been devised with the aim of reducing pollutant formation during combustion. These concepts are generally based on limiting the maximum flame temperature to avoid thermal NOx formation and controlling local fuel-air mixtures to avoid soot formation. Alternatively, some additional fuel can be injected after the main combustion event to burn-off soot before the gasses are exhausted. All modern diesels rely on high-pressure direct fuel injection. To enable these advanced combustion strategies higher levels of performance and controllability are required from the fuel delivery system. General trends are:

- Higher injection pressures, 2500 to 3500 bar, helps improve atomisation of the fuel spray and increase air entrainment.
- More injections per cycle. Instead of one large injection the injection is often split into multiple stages, this gives better control over combustion chemistry and rate of heat release.
- Shorter injection pulses and smaller, better defined, injected volumes. Better control over the amount of injected fuel gives more control over in-cylinder temperatures and local fuel concentrations.
- (Expected) use of a wider range of (more sustainable) fuels which have different liquid properties.

To meet these requirements common-rail injection systems equipped with electronically controlled solenoid or piezo injectors are used. All injectors are fed from a single fuel rail which is continuously kept at injection pressure by a high pressure fuel pump. This means the injectors always have an adequate supply of high pressure fuel and injections can be performed during any stage of the engine cycle. Older unit-pump based systems have a limited window during which pressurised fuel is available at each injector and therefore lack the flexibility of common-rail systems.

Development of combustion strategies is based largely on combustion modelling and simulations. The fuel spray, lying at the heart of the combustion event, is a much studied subject and a key model input. In spray research a lot of emphasis is placed on the flow through the injector orifice. Events on the interface between the injection system and combustion chamber characterise the injection.

Context

Advancements in fuel injection technology have been and still are the major enabler for advancements in combustion strategies. Better understanding of the performance of state of the art (ultra) high pressure fuel injection systems can help the development of combustion strategies for cleaner and more efficient diesel engines. The aim of this project is to provide a basis for improved understanding of short, high-pressure (above 2000 bar), injections with durations in the order of 100µs. To this end two aspects of the injection process are of interest:

1. Characterisation of the fuelspray
   Characterising the fuel flow through the injector nozzle orifice. The nozzle is the last place where the injection system can influence the spray. Nozzle (flow) characteristics form the basis of spray and combustion models, the aim is to obtain (better) model inputs by experimentally determining nozzle characteristics.
2. Realisation of the fuelspray

The entire injection system serves to realise the nozzle flow, the aim is to understand which parameters of the injection system affect the nozzle flow in which manner.

Parameters of interest to spray- and combustion researchers are nozzle coefficients \((C_a, C_M, C_v, C_d)\), fluid mass flux \((\dot{m})\) and momentum flux \((\dot{M})\). Nozzle coefficients describe the flow regime in the injector relative to an ideal nozzle. The amount of momentum carried by the fuel flow through the nozzle orifice is critical for how far the fuel spray can penetrate the combustion chamber and how well it mixes with air in the cylinder. Another important parameter is injection delay, the delay between start and end of injector actuation and the actual start and end of the injection event. When trying to realise short, precisely timed, injections knowing the discrepancy between the moment of system actuation and the moment of spray initiation is critical.

**Objective**

Currently TU/e does not have a means to experimentally characterise fuel injectors, i.e. determine nozzle coefficients, \(\dot{M}\) and \(\dot{m}\). The original idea for this project was to design and construct an experimental apparatus capable of characterising nozzle flows of state of the art (ultra) high pressure fuel injection equipment. During the first months of this project a number of physical limitations arose that made it infeasible to actually construct a system within the duration of this project. Initially renovation work seemed to limited the possibilities and of building a setup in the research lab. In February this was compounded by the nationwide Covid-19 lock-down measures that would last for several months. These events forced us to shift the project focus, development of the experimental setup was limited to the design stage. Staying within the original context of improving understanding of short injections, focus was shifted to analysing influences of the injection system on the injection characteristics.

**Scope & research questions**

Working towards the overall goal the following concrete steps will be taken.

1. Design an experimental injector characterisation setup, capable of determining mass flux, momentum flux and nozzle coefficients \(C_a, C_d, C_m\) and \(C_v\). This involves defining requirements, preferences and constraints for the experimental setup, selecting a suitable measurement concept, making a global design overview of the system and specifying required components (at this stage globally not specifically). Develop routines for determining the desired parameters from measurable quantities. Factors affecting the performance of the test setup are identified and addressed.

2. Model based injection system performance analysis. Start by developing a physics-based solenoid-type injection system model. The solenoid valve will be modelled explicitly, to this end the physics behind solenoid operation are introduced first. Define injection performance indicating values and use the developed model to identify how various aspects of the injection system affect the injection flow. Focus on transient behaviour and short duration injections.

This project focusses on heavy duty diesel applications, since it is believed that direct liquid fuel injection is likely to remain of importance in this segment for years to come given the current lack of adequate alternatives. In the modelling work, common rail systems based on solenoid injectors are investigated, mainly because solenoid technology is in widespread use due to their lower cost than the alternative piezo crystal based injector technology.

In the project, the following supportive research questions are guiding:

- 'Which experimental technique is best suited to measuring mass flux, momentum flux and nozzle coefficients?'
- 'Which factors are to be taken into account in the setup design? especially regarding the fuel supply and injection control side. What sort of setup is required and which components could be used?'
- 'What is the expected level of accuracy with which mass flux, momentum flux and nozzle coefficients can be determined? Which factors are determinative for the achieved level of accuracy?'
- 'Are there differences between the engine installed injection system and the research setup? In which ways could the experimental setup alter injection performance?'
• 'Can mass and momentum flux be adequately measured using the same experimental technique or are two separate setups required?'
• 'How can desired quantities be determined from measured data, are any other properties required, how would those properties be measured?'
• 'What are suitable performance indicators for high pressure injection systems? Which indicators can best describe transient performance?'
• 'Which mechanical aspects of the injector affect injection performance? How to model the hydraulic and mechanical aspects of a fuel injection system for performance analysis purposes.'
• 'How does the solenoid valve affect injection performance? How can a solenoid actuator be modelled for performance analysis study?'

Report structure
This report aims to document the work on developing an experimental injector characterisation setup and analysing a solenoid-based heavy duty injection system. In the first section some background information is provided on common rail injection systems and current trends in combustion injection strategies. A typical injection profile is introduced and the purposes of each sub-stage are explained. Furthermore a number of definitions used in nozzle characterisation are introduced. This information is mainly there as a reference for the reader and does not contain new work on my part.

Section 3 documents the search for the experimental technique, best suited to determining mass flux, momentum flux and nozzle coefficients. This starts by defining requirements, preferences and constraints for the eventual setup, followed by an overview of available experimental techniques. Based on the system requirements the suitability of each experimental technique is evaluated. Ultimately the experimental concept best able to full fill the requirements for the new experimental setup is selected.

In section 4 a global setup design is proposed, based on the selected experimental concept. An overview of system components is provided with suggestions on key aspects. This overview is meant as a starting point for further setup development work, suggestions on component selection and handling should be seen in this light. More detailed advise on fuel system component sizing is given in subsection 4.4, these findings are based on model analysis. Though the models on which these findings are based will be introduced later it is felt that all setup design suggestions are best presented together. Next in this section methods are introduced for determining the desired values (mass flux and nozzle coefficients) from the measured data, illustrated by examples.

Sections 5 and 6 address the development of a lumped parameter solenoid-injector based injection system model. Section 5 deals with the main hydraulic-mechanical models. Firstly the type of model and modelling environment are introduced, followed by the governing hydraulic-mechanical equations. Subsequently the developed injector and fuel pump models are discussed. Finally an overview of baseline parameters and initial model results are provided as well as a model validation based on literature and experimental data.

Given the different physical background and unfamiliarity of many mechanical engineers with electro-magnetic solenoids, this aspect of the modelling is treated in a separate section. Section 6 starts by introducing the governing electro-magnetic concepts and equations. Afterwards the developed model is introduced. To conclude this section the process of determining baseline parameter values and model validation is outlined. For the solenoid model this proved challenging due to limited available data.

The developed injection system model is used to perform a parameter sensitivity study: how does each parameter affect the injection behaviour? Of primary interest are short injections (sub 5 mg per event) and the transient response of the injection system, though longer injections (100+ mg per event) are evaluated as well. In section 7 this process is documented, starting with an outline of the process. Performance indicators, a reference injection case and an evaluation structure are provided. This is followed by the sensitivity results, tables provide data on how the key performance indicators are affected by each parameter change. Results are discussed and where necessary supported by additional figures. At the end of the section the results are interpreted with an eye on: experimental setup design and operation, model development and expected lifetime changes.

In the concluding section findings are repeated and research questions answered.
2 Context & background

This section aims to provide some general information on the subjects of (heavy-duty) common rail diesel injection systems, and a background on diesel injection and combustion strategies. Finally terminology and definitions used in spray research are introduced.

2.1 Diesel combustion strategies & injection patterns

The most basic diesel injection strategy consists of a single injection event, the duration and flow rate of which determine engine power and torque. Injection starts with the piston near top-dead-centre (TDC) when in cylinder gases are hot. Fuel jets exiting the injector entrain air and start to combust, resulting in regions of rich combustion and leaner combustion regions with very high temperatures. In the rich combustion zones soot is formed, NOx is formed in the hot regions. A second problem is ignition delay, some time is required before the fuel jet has entrained enough hot gas to reach ignition conditions. During the ignition delay some premixed charge is created, upon ignition this premixed charge combusts at a high rate (faster than the injection rate). This results in a sudden and high rate of heat release and with it a pressure spike (termed the 'premixed peak') which is responsible for high engine noise. Regarding combustion efficiency the conventional injection strategy is near optimal. Injection and combustion happens fast and around TDC. Cylinder volume change during combustion is low, allowing for a lot of post combustion expansion which results in good thermodynamic efficiency.

Advanced injection strategies for conventional diesel

Development of diesel combustion strategies is based around the following targets: reduce NOx and PM emissions, reduce combustion noise, improve drive-ability, retain fuel efficiency and improve specific engine power (kW/Litre capacity). In order to work around the limitations of conventional diesel combustion and reach the goals as stated above, new combustion and injection strategies have been devised. The following advanced injection strategies are for "conventional" diesel combustion, advanced combustion concepts like HCCI and their implications for fuel injection strategies will be treated later. Enablers of these more advanced diesel strategies are high pressure injection equipment, turbocharging and exhaust gas recirculation (EGR). High pressure injection systems are generally seen as a crucial component. Without going into too much detail, all advanced combustion concepts aim to limit maximum flame temperature to below 2000K in order to reduce NOx formation whilst creating a lean enough/hot enough flame to prevent soot formation or burn of generated soot. Most advanced injection strategies rely on a multiple separate injection stages each having a specific purpose. In general, multiple injection events can include (some or all) the following stages [1], see figure 1 for a schematic overview.

- Pre injection. A small amount of fuel is injected during the compression stroke, well before TDC. This fuel decomposes (but does not burn) during the compression stroke and serves to precondition the combustion chamber gasses. The aim is to create a more reactive environment which decreases ignition delay for the main injection event. Two mechanisms for this are increased temperature from energy released as the fuel decomposes and generation of radicals from the decomposed fuel. Reduced ignition delay means less fuel is injected during the ignition delay and the 'premixed peak' is less severe, reducing engine noise. It also helps reduce NOx formation during initial combustion [2].

- Main injection. The main injection controls the power produced by the engine, this happens through control over injection duration and fuel mass flow rates. Through the injection mass and momentum flow rates one has further control over mixing and combustion properties which affects efficiency and emissions. Rate shape optimisation is applicable to low speed and low load operation. At high speeds/loads the challenge is to inject enough fuel in the limited available time window, resulting in square rates. In these cases (cooled) EGR at high boost pressure is used to reduce NOx emissions.

- Early post injection. A short injection pulse after the main injection with the aim of burning of soot formed during the main combustion event. This requires a high injection pressure to give good mixing and a large amount of air entrainment. Fuel must break down into small droplets with a high kinetic energy to ensure adequate vaporisation and mixing. Full needle closure after this injection is critical, any large drops leaking away after this last combustion event can cause pollution or contaminate the engine oil. Good mixing and short burn duration limits flame temperature, soot is consumed and little extra NOx is produced.
Late post injection. Injection well after the main combustion event (just prior to or during the exhaust stroke), this fuel is not burnt in the cylinder. Addition of evaporated but unburned hydrocarbons to the exhaust gasses for aftertreatment purposes. Could be for thermal regeneration of diesel particulate filters or to aid in NOx conversion. If EGR is applied the unburned hydrocarbons are present in the intake charge, helping to reduce ignition delay, akin to the early injection.

Figure 1: Qualitative overview of a multiple injection event, injection pressure as function of crank angle. Various main injection possibilities are presented, ramp being typical for unit injectors, rectangular for standard common rail systems and ramp for advanced variable nozzle injectors. Source [2]

Increasing thought is given to the concept of mass and momentum flux rate-shaping. Both factors influence mixing and combustion and therefore provide a tool to reduce emissions. For example, during the main combustion the in cylinder gasses heat up, increasing the rate at which fuel can be burnt. By increasing the mass flow rate over the main injection, combustion can be optimised. Momentum of the fuel jet influences air entrainment, spray breakup and jet penetration. These mixing properties have effect on local equivalence ratio and therefore pollutant formations.

Another important reason why flexibility is desired from the fuel system is the multiple injection strategy. Each injection stage has different ideal mass and momentum fluxes. For example, at low density conditions (pilot injection) spray penetration must be limited in order to prevent wall impingement but momentum must be high enough to ensure good spray breakup and air entrainment. An early post injection must have a high momentum flux to incorporate as much as the cylinder air volume as possible to burn off the highest amount of soot.

Three main methods for mass/momentum flux rate shaping are: variable injection pressure, variable nozzle geometry (twin needle injector with two rows of nozzles) and variable needle lift. Variable needle lift has the advantage of not requiring variable pressure and is relatively simple to implement using a common rail and piezo actuated injectors but has a low nozzle efficiency (cavitation). Twin needle injectors are complex and expensive but just now finding their way into on-road applications. Variable injection pressure has the advantage of always operating at optimal nozzle efficiency but is not directly applicable to common rail systems. Pressure amplifying injectors with an intermediate pressure common rail form a solution [1]. Patented by Bosch this system is currently used in Mercedes Benz heavy duty engines.

Advanced combustion strategies

Besides improved conventional combustion a number of advanced combustion strategies have been developed to meet future propulsion goals. Most prominent are Homogeneous Charge Compression Ignition (HCCI) type strategies. The idea being to create a homogeneous (lean) charge that auto ignites in a knock type event. What results is a uniform lean combustion with a low flame temperature (low NOx and soot formation) with high thermodynamic efficiency. An interesting version of HCCI is Dual Fuel RCCI. A low reactivity fuel (compared to diesel) is used in a homogeneous premixed charge, injection of a small amount of diesel ignites the mixture. This system is suited to LNG combustion which is a relatively low carbon fuel that can help reduce specific CO2 emissions.
Heavy duty common rail injection system

In common-rail diesel injection systems the injectors for multiple cylinders are connected to a common pressurised fuel rail, figure 2 shows an overview. On V engines a separate rail might be used for each bank of cylinders though these two rails are then often interconnected. The rail is fed by an engine-driven high pressure fuel pump, typically a standalone unit or (several) unit pumps driven by a camshaft in the engine block. Fuel is supplied from the tank by a low pressure feed pump and filtered before being compressed by the HP pumps. The engine ECU controls the injectors (via an injector driver) and regulates rail pressure. Rail pressure can be regulated by altering the pumped volume (by controlling the pump control valve) and releasing fuel from the rail by a rail pressure regulator. Return fuel from the pressure regulator and injectors is returned to tank. Modern high pressure common rail system can operate at over 2500 bar, pressure can be controlled independent of engine speed. Herein lies the main advantage of HPCR systems: injection pressure is available over the entire engine cycle and is independent of engine speed, ensuring mixing and spray penetration at low speeds. Furthermore there are no limitations to the injection window and total injection volume, early and late injections are possible.

Injector properties

A large variety of different direct injection diesel injectors exists, all work on the same basic principle. Pressurised fuel is available in a volume, this volume is separated from the injection orifices by a needle. By lifting the needle the holes are exposed and fuel can flow into the combustion chamber. What goes to explain the variety in injector types are: number of holes, means of lifting the needle, type of needle seat and volume between needle and injection holes. Figure 3 shows a schematic close-up of the two most common nozzle types and gives an impression of the geometry where the jet originates.

Figure 2: Schematic overview of a 6 cylinder common rail system, indicating fuel paths, sensing and actuation signals.

Figure 3: Two most common nozzle types, Valve Covered Orifice and sac type. VCO nozzles tend to have less hydrocarbons emissions. Source
2.2 Nozzle characterisation definitions

In the field of fuel spray and combustion research, nozzle characterisation is an important and often studied subject. To this end a number of definitions and naming convention exists. The following overview of nozzle flow phenomenology and terminology is provided for completeness and does not contain new work from my part.

Characteristics of fuel sprays and nozzle flow are often expressed in terms of an ideal flow and corrective scaling factors termed nozzle coefficients (Ca, Cv, Cd and Cm). The ideal flow basis comes from the observation that, from a certain level of needle lift, the fuel discharge is governed almost entirely by the nozzle flow characteristic. Under ideal, frictionless, conditions the nozzle flow velocity can be described in terms of the pressure difference across the nozzle in terms of Bernoulli's equation:

\[ u_{th} = \sqrt{\frac{2(p_i - p_b)}{\rho_f}} \]  

The theoretical flow velocity through the nozzle channel \( u_{th} \) [m/s] is a function of fluid density \( \rho_f \) [kg/m\(^3\)] and the pressure difference over the nozzle expressed in terms of injection pressure \( p_i \) and in-cylinder back pressure \( p_b \) [Pa].

If the nozzle entrance has sharp corners and a non-converging shape the resultant sudden change in flow direction and velocity can cause the pressure to drop below the fluids' vapour pressure. When this happens gas bubbles are formed in the channel, this is termed cavitation. Vapour bubbles in the flow decrease the amount of channel area through which liquid fuel flows. A second effect the vapour bubbles have is to decrease wall friction, this means the average flow velocity can actually increase. Figure 4 provides a schematic overview of various effects and naming conventions involved in describing cavitating and (non-cavitating) nozzle flows.

![Figure 4: Schematic representation of area contraction for cavitating nozzle and effective velocity.](Adapted from [3])

The complex multi phase (vapour + liquid) flow with a non uniform density and velocity (due to wall friction) is approximated by an effective flow. Since the vapour density is much lower than the liquid density the vapour contribution to mass flux and momentum flux are often neglected. Based on this the effective flow area \( A_{eff} \) is defined as the area where liquid flows and the effective density equals the fluid density \( \rho_f \). Effective flow velocity \( u_{eff} \) is simply the flow velocity of the liquid phase averaged over the effective area. When no cavitation occurs this simplifies to: \( A_{eff} = A_0 \) and \( u_{eff} = u_{avg} \). These definitions form the basis of the nozzle coefficients. Area ratio \( \frac{A_{eff}}{A_0} \) is known as the area contraction coefficient \( C_a \). Velocity coefficient \( C_v \) is the ratio of effective velocity over ideal discharge velocity \( \frac{u_{eff}}{u_{th}} \), note that due to wall friction the \( C_v \) value for a non-cavitating nozzle flow will be below unity.

The ratio of actual mass flow rate over theoretical mass flow rate is defined as the discharge coefficient \( C_d \). This coefficient captures both the reduced liquid flow area in case of cavitation and discrepancy in actual flow velocity compared to the ideal basis case. Along similar lines the momentum coefficient \( C_M \) is defined as the ratio of \( \frac{\dot{m}_{act}}{\dot{m}_{th}} \). Following the definitions presented above the actual mass and momentum flux can be expressed as:

\[ \dot{m}_{act} = \rho_f A_{eff} u_{eff} \quad \text{and} \quad \dot{M}_{act} = \dot{m}_{act} u_{eff} \]  

Doing the same for the theoretical mass and momentum flux gives:

\[ \dot{m}_{th} = \rho_f A_0 u_{th} \quad \text{and} \quad \dot{M}_{th} = \dot{m}_{th} u_{th} \]  

Combining definitions (2) and (3) with the definition of the discharge coefficient it follows that:
\[ C_d = \frac{n_{\text{act}}}{n_{\text{th}}} = \frac{\rho_f A_{\text{eff}} u_{\text{eff}}}{\rho_f A_0 u_{\text{th}}} = C_a C_v \] (4)

Along similar lines, for the momentum coefficient this yields:

\[ C_M = \frac{M_{\text{act}}}{M_{\text{th}}} = \frac{\rho_f A_{\text{eff}} u_{\text{eff}}^2}{\rho_f A_0 u_{\text{th}}^2} = C_a C_v^2 = C_d C_v \] (5)

It must be noted that values for the various coefficients can be obtained by different methods, depending on the available data.

### 2.3 Reference engine

A lot of heavy duty engine research at TU/e is performed on a single cylinder heavy-duty test engine. This test platform is based on 12.9L, 6 cylinder heavy duty truck engine. In the test configuration only one cylinder is supplied with fuel, an electric motor is used to assist engine rotation (at low speed). Fuel is supplied to the single research cylinder from the factory common-rail. Under normal operation the fuel rail is fed by 2 unit pumps, driven by 3-lobed cams rotating at half engine speed. This results in 6 pump strokes per 2 full engine rotations, one for each injection event. On the single cylinder version only a single unit pump is actuated once per 2 full rotations, for the single required injection event. The injection system is based around solenoid injectors. For this test engine experimental datasets are available, indicating rail pressure, cylinder pressure, injector actuation data etc. Given the prominence of this setup in TU/e heavy duty engine research and the availability of experimental data this engine and its fuel system will serve as a baseline reference case for the modelling and design work performed in this thesis.
3 Measurement concept selection

Diesel injection equipment characterisation is a well-studied subject, over the years numerous experimental techniques have been proposed. In this section an overview of common experimental FIE characterisation techniques is given. The concept best able to meet the system requirements is selected, findings are based on comprehensive literature study.

3.1 Requirements, preferences, constraints

For selecting an experimental concept the relevant RPC’s are mostly functionality related. Though more relevant during the detailed design phase some practical constraints have been included.

Requirements

- The measurement technique must be suitable for measuring short (sub 0.3 ms) and long (more than 1 ms) injection events with adequate temporal resolution, and be able to deal with close-spaced multiple injections. Injection trains of upto 5 individual injections within a typical 30 CAD (crank angle degrees) injection window (at 2000 RPM → 2.5 ms window).
- Setup must be able to measure instantaneous mass flow rate $\dot{m}$ and momentum flux $\dot{M}$. Rate shape and cumulative mass should be accurately determinable.
- Setup must be able to measure start and end of injection delay, time between signal to injector and first/last nozzle flow. The temporal resolution must be in the order of $\mu$s.
- From the measured data it must be possible to determine nozzle coefficients: $C_a$, $C_v$, $C_d$ and $C_m$
- The designed setup must be safe, the experimental concept should lend itself to this. High pressure fuel poses a hazard, skin penetration of a high pressure fuel jet can be reason for amputation.
- Measurement equipment must not influence the injection properties. If interference is non-avoidable the effects must be known and accounted for. Behaviour of the apparatus must be understood and verified.
- Equipment must be able to characterise individual jets for multi-hole injectors, allowing the study of flow inequality between nozzles

Preferences

- Simplicity of the measurement setup, ideally it should not require complex components.
- Directness of measurement, a directer method is preferred.
- The process of determining desired quantities from measured data should not rely heavily on fluid properties at the nozzle. Pressure and temperature dependent fluid properties are difficult to determine at the injector nozzle.
- Low measurement noise level, high signal to noise ratio.
- Versatility of the setup, ideally it should be suitable for a wide range of flowrates, hole numbers, layouts etc. with limited alterations. Easy interchangeability of injectors is preferred.
- Optical accessibility, can help verify correct functionality

Constraints

- Availability of space. In order to construct and test the setup a suitable space is required. Since the research labs are under reconstruction currently this might prohibit the realisation of the project
- Budget, funds have to be available and will not be unlimited.
- Time, components will have to be ordered, lead times involved might prove prohibitive.

Nice to have

- Ability to measure mass and momentum flux of the fuel jet in a spatially resolved fashion, providing useful information for validating spray models and studying air entrainment. To make this relevant the spray should be injected into a pressurised environment, mimicking in-cylinder conditions.
3.2 Overview of experimental techniques

Experimental FIE characterisation methods can be divided into a number of sub-groups: mass flux focussed, momentum flux focussed and spray development focussed. For each sub-group techniques are introduced and some advantages/disadvantages are noted.

3.2.1 Mass flux measurement techniques

Mass flux of direct fuel injectors, sometimes termed ROI (rate of injection) has been studied since the 1960s. Well-established in this field are the ‘Bosch’ and ‘Zeuch’ methods, an advanced version proposed by Marcic [4] offers the ability to study individual nozzle flows. All these methods are based on interpreting the volume flow rate from the effect it has on a (closed) volume of fluid into which it is injected.

Zeuch method

The Zeuch ROI measurement technique is based on injecting into a closed fuel-filled chamber. Since the fuel has a (limited) compressibility (K [Pa N/m²]), chamber pressure will rise during injection to accommodate the added fuel mass. Injected fuel mass flow rate (\( \frac{dm}{dt} \) [Kg/s]) is linked to the derivative of chamber pressure \( p \) [Pa]:

\[
\frac{dm}{dt} = \rho_{\text{fuel}} \frac{V_{\text{chamber}}}{K_{\text{fuel}}} \frac{dp}{dt}
\]

By measuring the (static) pressure rise in the chamber one can determine the volume inflow if the compressibility value of the fuel is known. A temperature sensor is included in the chamber to help determine instantaneous fluid properties. Between injections a fast-acting solenoid valve opens to return the chamber back to initial conditions, relief pressure is set by a check valve.

Design of the chamber geometry must be such as to limit pressure waves in the fluid (limit reflections), placement of the pressure transducer is optimised to limit pressure wave interference. Generally two pressure sensors are used in tandem: one piezo and one strain type sensor. Piezo pressure transducers have good response rates but tend to drift over time. Strain type pressure sensors have less good dynamic behaviour but do not drift. Combining both allows one to work around the limitations of both types. Sometimes a second piezo sensor is added at a different position, to allow one to filter out the effects of pressure waves. The chamber volume must be chosen such that the pressure rise during the injection event is measurable with adequate resolution but the pressure rise remains limited.

Advantages, disadvantages and limitations:

- Compact setup, much smaller than Bosch apparatus.
- Cumulative mass flow follows directly from measured pressure, not from signal integration.
- Accuracy of the measurement depends on how accurate the compressibility of the fluid is known. Requires a secondary experiment to determine fluid properties over a range of temperatures. Fluid temperature is critical since compressibility is temperature dependent.
- Kinetic energy of the flow adds a thermal load and associated pressure rise: systematic error.
- Injection back pressure varies over the injection and fuel is injected into fuel not into air, introducing systematic error.
Bosch method

The Bosch ROI measurement technique is based on injecting fuel into one end of a long (approx 25m) fuel-filled tube. The injection generates a propagating wave in the tube. Assuming the flow is unidirectional, the dynamic pressure associated with the wave is directly proportional to the injection velocity. Dynamic pressure in the measurement tube is monitored just downstream of the injector with a pressure transducer. An adjustable orifice at the end of the measurement tube is used to tune the magnitude of the reflected pressure wave. By analysing the time between the first pressure rise and the reflected pressure wave the speed of sound can be determined. The length of the measurement tube is selected to prevent interference from the reflected wave during the injection event. A following tube helps to dampen out residual pressure waves. A check valve at the end of the following tube releases fuel to restore static pressure between injections. Discharged fuel can be collected and weighed to verify the integrated flow measurements. Figure 6 shows a schematic overview of the measurement apparatus.

![Schematic of Bosch ROI meter, measuring and following tube are coiled up. Source:[6]](image)

If the injector jet is aligned with the tube, the following relation for dynamic pressure holds: \( p = \frac{c \rho u}{A} \), with \( c \) the speed of sound in the fluid [m/s], \( \rho \) the fluid density [kg/m\(^3\)] and \( u \) the fluid velocity [m/s]. Combining this with the continuity equation the following relation results for mass flow rate as function of measured dynamic pressure \[^5\] where \( A \) is the cross-sectional area of the measurement tube [m\(^2\)]:

\[
\frac{dQ}{dt} = A \frac{c \rho p}{c} \rightarrow \dot{m} = A \frac{c p}{c}
\] (7)

This method has a distinct advantage over the Zeuch method namely: the relation between mass flow rate and measured pressure does not include fluid density or compressibility. Those fluid properties are temperature dependent and would have to be determined separately. The only required fluid property, the sound propagation velocity \( c \), can be obtained directly from the reflected pressure pulse delay.

Advantages, disadvantages and limitations:

- Injection rate shape follows directly from the pressure measurement, and generally has a cleaner signal than Zeuch method \[^5\].
- Less well suited to close-spaced injection trains, reflected pressure waves can contaminate the signal if insufficient time is left between subsequent injections.
- Slight temporal error due to downstream placement of the pressure sensor, might be negligible.
- Again injecting into fuel rather than engine like conditions, systematic error. Tube conditions change slightly due to momentum flux (kinetic energy absorption giving heat).
- Total mass flow is obtained through integration, generally less accurate than Zeuch method \[^5\].
Individual orifice method

Conceptually, the Bosch method is based on measuring the dynamic pressure associated with the flow wave caused by an injection into a continuous diameter tube. Marcic proposes an advanced technique based on the same principle in [4]. Rather than placing the injector at one end of a long fuel-filled tube he proposes using an injector holder in which each nozzle orifice has its own measurement chamber. The measuring chambers are fuel-filled and kept at an overpressure of 5 kPa (0.05 bar) by a pressure regulating check valve, the chamber outlet is oriented at right angles to the spray axis. Directly opposing the injector orifice a membrane is placed equipped with strain gauges which acts as a pressure sensor. Measured membrane deformation is related to instantaneous chamber pressure. Assumed the fluid pressure wave is coplanar with the membrane and uniform over the entire membrane area ($A$), equation [7] can again be used to determine the rate of injection. For this configuration $A$ corresponds to the membrane area.

Figure 7 shows the measurement device geometry.

![Figure 7: Technical drawing of measurement device geometry as presented in [4]](image)

Membrane deformation is measured using 4 strain gauges, 2 measure compressive deformation of the membrane edge and 2 measure tensile deformation of its centre. By arranging the 4 strain gauges in a Wheatstone bridge configuration with the junctions at the same temperature the measuring circuit is temperature compensated. These steps are necessary since the injected fuel heats up the apparatus. Fuel leaving each measuring chamber is collected separately for reference.

Advantages, disadvantages and limitations

- Nozzle flow-rate can be measured for individual nozzle orifices of a multi-hole nozzle
- Speed of sound in the fluid has to be known, unlike in the Bosch long-line apparatus it cannot be determined from the measured signal. An additional experiment would be required.
- Measurement chambers have to be uniquely made to fit a specific injector geometry. Injectors with a different size, hole count, hole spacing or tip geometry require a new custom measurement chamber.
- Given the dimensions of the measuring chambers one would expect interference from reflected pressure waves. Though this is not mentioned in the original paper it is something to consider.
3.2.2 Jet impingement

The most common technique used for spray momentum flux characterisation is the jet-impingement method [7, 8, 3]. The vector quantity momentum of an object is equal to the product of mass and velocity \( \vec{M} = m \cdot \vec{v} \) [kg \( \text{m/s} \)], which for a non-uniform fuel jet is best expressed as:

\[
\vec{M} = \sum_{i=1}^{i} m_i \vec{v}_i \tag{8}
\]

The momentum of an object can be altered by an external force, the impulse \( J \) exerted by an external force over a time interval equals the change in momentum over that interval:

\[
\vec{J} = \int \Delta t \vec{F} dt = \Delta \vec{M} \tag{9}
\]

The instantaneous magnitude of the external force thus equals the instantaneous rate of momentum change in the direction of the force vector. Applying this to the characterisation of fuel sprays, if the spray can be impinged such that the velocity of the spray along its axis gets reduced to zero the force required to achieve this exactly equals the instantaneous spray momentum flux along that axis. If it is assumed, as is often done, that the other spray velocity components are zero the impingement force equals the total spray momentum flux \( \dot{M} \). Figure 8 shows a schematic of the jet-impingement momentum flux measurement technique. The jet is impinged on a target, mounted perpendicular to the spray axis, attached to a force transducer. The impinged flow exits in radial direction indicating that it has no residual velocity along the spray axis.

![Figure 8: Schematic of jet impingement technique drawn in cross-section. Normally axis-symmetric, impinged flow in radial direction.](image)

To get a true reading of the momentum flux along the spray axis it is critical that the impinged flow has no residual velocity along the spray axis. If the target is too small the impinged flow can carry some forward velocity. Alternatively when the impingement is uncontrolled and rough the spray can actually be deflected back towards the injector, the resulting force reading overestimates the spray momentum. Backsplash can occur due to surface pitting on the impingement target. A flat target does not necessarily provide the best impingement surface, Lindström suggest using targets which gradually taper down from a central point[9]. This geometry makes for a more gradual deflection and a cleaner radial flow.

Advantages, disadvantages and limitations

- Momentum flux can be measured directly, both rate-shape and magnitude follow directly from the force signal
- The technique is suited to short and close-spaced multiple injections. Results are not influenced by pressure waves triggered in the measurement chamber since it is not fluid filled. The limiting factor for injection frequency is the bandwidth of the force sensor
- Fuel can be injected into (pressurised) air rather than liquid, avoiding systematic error.
- Simple and versatile, adaptable to a whole range of injectors with minimal changes.
- Individual jet only. Impingement targets have to be aligned carefully
- Mass flux cannot be determined directly but the mass flux rate-shape is linked directly to the square root of momentum flux rate shape. Mass flux can be determined from a simple scaling with total injected mass.
Impingement spatially resolved

Several authors have proposed advanced jet-impingement measurement techniques, focussed on characterising the spray momentum flux distribution in a spatially resolved fashion. Mapping the distribution of momentum flux over the spray cross-section at various distances from the nozzle can be useful in validating spray models. Two relevant works in this field are by Postriotti \[10\] and Bottega \[11\].

Postrioti shows, based on CFD results, that the pressure distribution on a flat impingement target does not reflect the momentum distribution in the free jet. A pressurised zone approximately 2 mm thick forms at the center of the target, the resulting force distribution is more spread out with a less prominent peak than the free-jet momentum flux distribution. To accurately measure local momentum flux, an isolated flow tube has to be evaluated without creating the pressure buildup associated with overall flow impingement. Two methods are suggested and evaluated, both mitigate the pressure buildup problem through use of a conical flow deflector. Fluid flow at the center of the cone is sampled, the two employed concepts are depicted in figure 10. Concept a allows a flow tube to pass through the center of the cone, underneath this flow tube is impinged and measured in the usual fashion. Concept b uses a straight guided pin protruding through the center of the cone to impinge a local flow tube and transmit the force to the force transducer. Friction and hysteresis in the pin guide is mentioned as a concern.

Findings suggest that concept a can provide a good representation of the actual momentum flux distribution. The integrated value is under predicted by 8% compared to the overall impingement force. This is attributed to dissipation in the filter orifice, if the dissipated fraction is constant the signal can be corrected by scaling to the overall impingement force value. Using concept b overall flux is over-estimated by 50%, this is attributed to pressure buildup at the central position. Too much pin protrusion gives a noisy signal due to flow interacting with the pin. Concept a is suggested to be superior.

The technique proposed by Bottega is based on flat plate impingement, impingement force is measured on a line across the plate. The spray is sectioned on a gridded plane at two non-perpendicular angles. A post processing algorithm is used to compute the spatial momentum flux distribution. The measuring line consists of a rigid strip of material, separate from the main target body. Figure 9 shows a schematic of the target design and sampling grid. Force transducers located at either end of the strip provide the line section data. Overall momentum flux can be measured simultaneously by a third transducer under the entire impingement platform. The paper proposing this technique fails to adress the pressure buildup phenomenon identified in 10. Given the flat plate impingement target pressure buildup related distortions are likely, this technique is therefore deemed inferior.

Advantages, disadvantages and limitations:

- Information on the spatial jet momentum distribution is useful for validating spray models. Spray cone angle can be determined directly
- Several measurements are required to fully characterise a spray, variation between sprays cannot be measured.
- Momentum flux values for the total jet are obtained through integration, a significant error is expected (at least 8 percent for the shielding-orifice technique)
- Spatially resolved information is less relevant for determining nozzle coefficients, overall mass and momentum flux can be determined less accurately than with a single impingement setup.
3.2.3 Spray development

Spray development is closely related to nozzle flow, mass flux and momentum flux can be correlated to spatial spray development. Local spray velocity can be determined using droplet velocimetry techniques. An alternative is to track the development of macroscopic spray characteristics cone angle and tip penetration. These can be linked to mass and momentum flux using spray correlations [12, 13].

Droplet velocimetry

For a developed spray phase Doppler anemometry (PDA) can be used to measure local droplet velocity. Phase Doppler anemometry is a non-intrusive light based velocity measurement technique. The principle is based on the phase shift a wave of light experiences when reflected off of a moving surface. A detailed explanation of the working principle is outside the scope of this work. For the technique to work discernible particles (droplets) have to be present in the flow, it can therefore only measure the spray section post primary breakup.

By sampling the spray at a range of radial positions a spray velocity profile can be created. Repeating this process at various distance from the nozzle the spray cone-angle can be determined as well as the spatial velocity development. Spatial velocity development provides information on air entrainment, the faster momentum energy is exchanged with ambient gas the faster the spray velocity decreases. Determining a momentum flux distribution from the measured velocity distribution requires information on local density, the PDA technique does not provide any information on this. It is possible to relate the measured velocity profiles to an initial mass and momentum flux by using spray models [13]. This relies heavily on correlations and correction factors and therefore have a considerable uncertainty.

Advantages, disadvantages and limitations

- The method is non-intrusive, fuel can be injected into an engine-like ambient.
- Local droplet velocity can be measured exactly.
- Mass and momentum flux can only be obtained through correlation using spray models, not exact enough for nozzle flow characterisation.
- Not suited to transient measurement, the spray has to be measured at a range of positions requiring a high level of repeatability.

DBI optical

Studying macroscopic development of non-combusting sprays can be done by means of diffuse-backlight illuminated photography. In a DBI photograph the liquid fuel droplets stand out as dark against a white background. Spray cone angle and tip penetration can easily be tracked over time if an adequate camera frame rate is used. For this experimental technique a simple setup is sufficient. The setup can consist of a spray vessel with clear walls, a light source with diffuser screen and a camera. Another advantage is that the spray can be injected into an engine-like high density ambient, avoiding systematic error.

![Figure 11: Non-combusting spray development phenomenology. Source [12]](image)

Spray models can be used to correlate cone angle and tip penetration to mass and momentum flux, taking into account ambient density [12]. The heavy reliance on theoretical spray models makes this technique less suited to fundamental nozzle flow characterisation. Furthermore this technique cannot provide information on short injections.
3.3 Evaluation

Based on the requirements, preferences and constraints defined for the new injection characterisation setup, the suitability of each technique can be assessed. Table 1 shows an overview of all introduced experimental techniques and the defined system requirements and preferences. The extent to which each technique is able to satisfy a given requirement or preference is indicated in a relative fashion. Overall capabilities (for example the ability to measure mass flux) are judged with a simple yes-no answer, when the technique relies heavily on correlations and models the capability is noted as ‘inferred’. More specific or subjective parameters are indicated in a qualitative fashion using a ++ to - - scale with ++ indicating a high level of suitability and - - indicating it is not suited. Intermediate values + or - signify that it is possible/suited but alternatives perform better.

Some remarks on the assessment presented in table 1:

All presented experimental techniques can be used to some extent to measure mass flux. Since the optical and velocimetry methods rely heavily on spray correlations to infer mass flux from spray development they are not suited for exact quantitative measurements. For long, pseudo-steady, injections the Zeuch method is preferred since it can give a clean signal and the measured quantity is directly related to the cumulative injected mass. With the Bosch and impingement method a cumulative mass value can only be obtained through signal integration or measuring a collected sample. For transient mass flux measurement the Bosch method is deemed the most accurate, under these conditions the Zeuch method suffers from pressure wave interference. Transient injections consisting of multiple events spread over a longer timespan are best measured with the impingement method, ultimately reflected pressure waves will affect the Bosch measurement.

Momentum can only be determined directly using the impingement methods, full jet impingement provides a direct rate shape trace. Measuring spatially resolved momentum flux introduces a cumulative error in the steady state measurement and is not suitable for studying transient conditions since the jet has to be sampled on a series of locations. Though the optical and velocimetry techniques can provide a global steady state value they are not suited to quantitative analysis.

For studying developing fuel sprays the optical, velocimetry and spatially resolved impingement methods can be used. Droplet velocimetry offers an non-intrusive method of directly measuring the spatial velocity distribution. Quantitatively mapping the momentum flux distribution can only be done through sampled flow-tube impingement.

For the Bosch and impingement methods the measured start/end of injection delay value depends on proximity of the sensor to the nozzle. Pressure rise in the closed Zeuch vessel is instantaneous and uniform so not dependent on sensor proximity.

The classic ROI measurement methods make it possible to determine the overall nozzle discharge coefficient only, no distinction can be made between individual orifices since no individual mass flow rates are known. This possibility is offered by the individual orifice method. Impingement methods provide a momentum flux value for an individual orifice thus a momentum coefficient can be determined per orifice. Combined with a total mass measurement the discharge-, area- and velocity-coefficients can be computed.

Regarding the preferences, the Bosch line and impingement force techniques are arguably the simplest of the quantitative techniques. Both incorporate only one pressure or force sensor and no other active components. The Zeuch vessel requires a fast-acting relief valve and the individual orifice method a complex housing and multiple measuring membranes. Bosch and impingement also offer the most direct measuring method, not requiring information on fluid compressibility (unlike the Zeuch method) or (temperature dependent) speed of sound (unlike the individual orifice method).
Table 1: Overview of experimental mass and momentum flux measurement techniques against various criteria

<table>
<thead>
<tr>
<th>Targeted properties</th>
<th>Dynamic p (Bosch)</th>
<th>Closed Volume p (Zeuch)</th>
<th>Individual orifice (membrane def)</th>
<th>Impingement overall force</th>
<th>Impingement spatially resolved</th>
<th>DBI optical (PDPA)</th>
<th>Droplet velocity (PDPA)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flux capable</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>inferred</td>
<td>inferred</td>
<td>inferred</td>
<td>inferred</td>
</tr>
<tr>
<td>Steady state ( \dot{m} )</td>
<td>+</td>
<td>++</td>
<td>+</td>
<td>+</td>
<td>+</td>
<td>+</td>
<td>+</td>
</tr>
<tr>
<td>Transient ( \dot{M} + ) rate shape</td>
<td>++</td>
<td>+</td>
<td>+</td>
<td>++</td>
<td>+</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Total injected mass accuracy</td>
<td>+</td>
<td>++</td>
<td>+</td>
<td>+</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Momentum flux capable</td>
<td>no</td>
<td>no</td>
<td>no</td>
<td>yes</td>
<td>yes</td>
<td>inferred</td>
<td>inferred</td>
</tr>
<tr>
<td>Steady state ( \dot{M} )</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>+</td>
<td>+</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>Transient ( \dot{M} + ) rate shape</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>+</td>
<td>+</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>Spatial jet development</td>
<td>no</td>
<td>no</td>
<td>no</td>
<td>no</td>
<td>inferred</td>
<td>yes</td>
<td>yes</td>
</tr>
<tr>
<td>Velocity distribution</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>+</td>
<td>++</td>
<td>++</td>
</tr>
<tr>
<td>Mass flux distribution</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>+</td>
<td>+</td>
<td>+</td>
</tr>
<tr>
<td>Momentum flux distribution</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>+</td>
<td>+</td>
<td>+</td>
</tr>
<tr>
<td>Timing, duration + delay</td>
<td>+</td>
<td>++</td>
<td>+</td>
<td>++</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Multiple-injection capable</td>
<td>+</td>
<td>+</td>
<td>+</td>
<td>+</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Individual hole capability</td>
<td>no</td>
<td>no</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
</tr>
<tr>
<td>In-engine conditions match</td>
<td>no</td>
<td>no</td>
<td>no</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
<td>yes</td>
</tr>
<tr>
<td>Determinable coefficients</td>
<td>( C_d )</td>
<td>( C_d )</td>
<td>( C_d )</td>
<td>( C_d )</td>
<td>( C_d )</td>
<td>( C_d )</td>
<td>( C_d )</td>
</tr>
<tr>
<td>Measurement noise level</td>
<td>low</td>
<td>high</td>
<td>unknown</td>
<td>low</td>
<td>medium</td>
<td>high</td>
<td>high</td>
</tr>
<tr>
<td>Simplicity setup</td>
<td>++</td>
<td>-</td>
<td>-</td>
<td>++</td>
<td>-</td>
<td>+</td>
<td>-</td>
</tr>
<tr>
<td>Directness of method</td>
<td>++</td>
<td>+</td>
<td>+</td>
<td>++ / - ( \Box )</td>
<td>+ / - ( \Box )</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Fluid properties required</td>
<td>+</td>
<td>-</td>
<td>-</td>
<td>+</td>
<td>+</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Versatility, different injectors</td>
<td>++</td>
<td>++</td>
<td>-</td>
<td>++</td>
<td>++</td>
<td>+ ( \Box )</td>
<td>+ ( \Box )</td>
</tr>
</tbody>
</table>

1. Typically applied for overall injector measurement, could theoretically be used for individual hole measurement but is highly impractical.
2. Injection into pressurised gas is possible, an evaporative high density environment can distort transient measurements.
3. Noise level for this technique is highly dependent on the type of force transducer used but can be very low.
4. Spray coefficients are: \( C_d, C_a, C_v, C_M \), a \( \bar{C} \) signifies that the coefficient is injector averaged (rather than hole specific) or time averaged.
5. Impingement force directly equals momentum flux, to derive mass flux additional information is required.
6. Versatility is lower than for other methods since experimental parameters for the used spray models are injector specific and have to be measured.
7. Fewer fluid properties required is preferable.
3.4 Concept selection

Based on the evaluation I would suggest using the spray impingement force measurement concept for the new injector characterisation setup. This technique is the only one suited for directly measuring momentum flux and determining all nozzle coefficients. Mass flux can accurately be determined from the measured momentum flux curve through a simple corrective scaling based on a measured cumulative injected mass value. It is suitable for characterising short and multiple (close spaced) injection events unlike the classical ROI measurement techniques which suffer from pressure wave interference. On this basis I would not advise using a separate ROI measuring device. The method is simple (requiring only a single force transducer), direct (not relying on any measured values for fluid properties) and fuel is injected into air rather than liquid, avoiding systematic error. Furthermore, individual fuel jets from a multi-orifice injector can be studied.

A downside of this method is that the same systematic errors will exist in both $\dot{M}$ and $\dot{m}$ rate shape signals, one has no way of verifying accuracy.
4 Experimental setup design

In the previous section an experimental technique was selected to form the basis of the new experimental FIE characterisation setup. In this section a global setup design is proposed. This process starts with outlining the required functionalities. For each function sub-group an overview of the required hardware is introduced. Later these component groups are discussed in more detail, requirements are specified on component level and where necessary supported with relevant simulations and calculations.

Note: given the decision not to construct the setup during this project and shift the focus towards modelling, not all aspects of the setup design have been worked out in detail. It is felt that some of the more practical design decisions are best taken when parts start to materialise (fixings, housings etc.), the following design overview is meant as a starting point for subsequent development work.

4.1 Setup design overview

Since the injector characterisation setup will be designed to operate as a standalone unit it must perform two main tasks: facilitate the functioning of the injector and characterise the created fuel spray. Performing these two tasks requires several main system functionalities:

- Fuel management
  The injector must be supplied with clean fuel, pressurised to a set value at acceptable temperature. Returning flows must be managed. To prevent systematic error the fuel delivery system should not alter the injection behaviour compared to engine fuel system. Safety is paramount, both for the operator and the injector under test. Regarding the injector this means adequate filtration to prevent damage from particulates, pressures must be kept in the specified working range.

- Spray and impingement conditions
  For an accurate measurement the entirety of the fuel spray has to be impinged at right angles to the jet direction. Conditions the spray encounters between the orifice and impingement plate also affect the measurement. If the distance is too large the spray will lose momentum to the ambient, when it is too small the impinged spray can affect nozzle flow and momentum is lost due to viscous friction.

- Data acquisition and control
  Given the time-scales involved with injection events the data acquisition can only be done electronically. Acquisition resolution in the order of $\mu$s is required. Meanwhile a number of processes have to be controlled actively in order to create the required injection conditions. Rail pressure must be managed, for this the rail pressure regulator and pump control valve have to be actuated. Furthermore the solenoid of the injector must be actuated, the achieved current profile must match that obtained by the engine management electronics.

- Sensing
  Various parameters must be measured accurately, both in magnitude and in time. Inertia in the sensing equipment must be understood, especially when studying short injections. Besides the obvious impingement force, the conditions that lead to those spray characteristics must also be captured. This includes fuel pressure and temperature measurements but might also require information on needle position and solenoid coil current.

Global design overview

Realising the functionality requirements requires quite a number of physical components. Figure 12 shows a schematic overview of the system components outlined below. In this figure the fuel system is displayed in blue, to distinguish high pressure fuel lines from low pressure ones they are printed bold. Control signals and controlled actuators are shown in green, sensed signals in orange.

- Fuel system
  Consisting of low pressure (LP) supply system, high pressure (HP) section feeding the injector and low pressure return. On the supply side the LP system includes a fuel tank with internal or external feed pump, filtration and water separation. The high pressure section consists of a HP pump, fuel rail, high pressure fuel lines and the injector of interest. Return fuel from the injector and rail pressure regulator is fed back via low pressure lines to the fuel filter inlet or directly to tank possibly via a cooler.
• Impingement + force measuring section
The spray impingement and force measurement section must house the injector, provide a positionable mounting point for the impingement force sensor and enclose the impinged spray. Spray containment is necessary both for safety reasons and collecting all injected fuel. Since most injectors have multiple holes arranged around the perimeter of the nozzle it beneficial if the positioning stage can rotate around injector axis, a second tilt axis allows for alignment with the hole. Given the large range of motion required (360°) a combination of coarse and fine adjustment stages could be useful. The spray chamber itself must be shaped such that the injected fuel can be easily collected and weighed.

• Data acquisition + control
Control and data acquisition are best combined and managed from a single computer with all relevant control and data processing software installed. To interface the computer with various sensors and hardware a real-time input and output device can be used (ROI). This ROI device must be equipped with relevant sensor input blocs (combined DAC modules and sensor drivers) and electronic driver blocks. Controlled hardware includes: HP fuel pump flow control valve, rail pressure regulator, HP pump drive (electric motor speed), LP feed pump (these generally have a built in electric motor) and the injector solenoid. Last in this category is all connecting hardware: cables, connectors etc.

• Sensing
For the characterisation measurements a fuel pressure sensor and impingement force sensor are required. Sensitivity, linearity and repeatability of these sensors must be high since they provide the characterisation data. The fuel pressure sensor used for injector characterisation is best placed as close to the injector as possible to get the most direct readings. For rail pressure control a separate rail pressure sensor can be used, this could be an OEM sensor. Addition of thermocouples to the rail and possibly in the fuel line near the injector provides information on fuel temperature. Since diesel fuel density and viscosity show a clear temperature dependency this information can be used to improve the accuracy of the characterisation. Ideally the injector is equipped with a needle lift sensor, this allows one to validate the needle motion aspects of a solenoid injector model. For the same purpose it is useful to measure the solenoid coil current as well.

Figure 12: Schematic overview of the impingement-force based momentum flux measurement setup.
4.2 System components, general guidelines

Rather than specify a list of which components are best suited, advice on component selection has been formulated in a guiding sense: aspects to consider when selecting components for the setup. Providing exact make and model information on which components to use would limit 'shelf-life' of this information.

4.2.1 Fuel system

Since the research setup is designed as a standalone system, not connected to a physical engine, a custom fuel system is required. Sizing of the fuel system as well as expected effects on injection behaviour are discussed in subsection 4.4. Here some general advice is given on fuel system components, providing a starting point for a future detailed design stage. For background information on diesel fuel systems and in-depth explanations of component functionalities the reader is referred to [14] and [15].

Storage, supply and filtration

A basic yet important component of the fuel system is a fuel storage tank. The container itself must be made of a fuel-safe non-corrosive material. Tank volume is mainly a function of heat capacity, when running a high return fuel load without a fuel cooler a large tank volume will slow down the heating process. If the return flow is cooled properly before returning to tank, or directly returned to the fuel filter, the tank volume can be lower. When dealing with precious research fuels a lower required volume is beneficial. Fuel supply outlet and return fittings should be positioned near but slightly above the bottom of the tank. Any water in the fuel will collect at the bottom, positioning fittings slightly higher could be beneficial. Adding a fuel drain point makes changing fuels easier. When draining the entire system it might be necessary to bleed all fuel lines.

Fuel flow from the tank through the filter to the HP pump is driven by a supply pump, either integrated in the HP-pump, in the tank or placed somewhere in-between. When using a HP-pump-integrated supply pump enough fluid head must be available at its inlet to prevent cavitation. Placing the tank some height above the fuel filter and HP pump should be sufficient.

Filtration is critically important to avoid damage to the HP-pump and injector itself and ensure consistency of results. The two main harmful contaminants found in diesel fuel are abrasive particulates and water. Abrasive particulates can wear an injector, enlarging fluid passages which alters the flow response. Water contamination is undesired since water does not have the same lubricating properties as diesel (low shear viscosity). When water enters a fuel-lubricated HP-pump the various high stress slide contacts (plunger-cam contacts) in the pump are no longer lubricated an metal particles wear off, (oil-lubricated versions exist but add undesired complexity through requiring an oil system). If these particles are ingested into the compressed fuel stream they can clog the injector and contaminate the entire system. A water-separator must be included in the filtration setup, together with a high-quality diesel filter capable of separating out sub-micron particles. Commercial vehicles are generally equipped with separate coarse and fine filters to prevent premature clogging of the fine filter. Given the low volume of fuel used in the experimental setup a single fine filter should be adequate.

Fuel rail and HP fuel-lines

The fuel rail serves a number of purposes, on a multi-cylinder engine it acts as a manifold (distributing fuel to all injectors), it provides storage of high pressure fuel, helps even-out pressure fluctuations from supply and demand variations and provides a place to mount pressure sensing and regulating hardware. Rail sizing is addressed in [14]. Fuel rails from engines are typically cylindrical, with a rail spanning the full engine length each injector feed line can be of similar length. For the research setup the 'rail' only has to feed one injector and total length is therefore not relevant (altering rail geometry can affect pressure pulses but this should not cause problems, see [14]). A custom rail can be made, this must include fittings for the pump and injector line, an attachment point for a pressure regulator and a pressure sensor. Given the high operating pressure safety is key, wall thickness and line + pressure sensing fittings have to be sized appropriately.

An easily overlooked component are the high pressure fuel lines making the pump-rail and rail-injector connections. Given the high operating pressure these lines are subject to, their structure is a critical safety aspect. Line structure and length also affects injection behaviour, this is again addressed later. High
pressure fuel lines are often pre-stressed during their manufacturing process, pressurising to well above operating pressure and suddenly discharging deforms the walls and improves stiffness. High pressure lines are best purchased, not custom made, pay attention to pressure ratings, length and fitting type.

**High pressure pump**

Heavy duty common rail systems are often fed by unit fuel pumps driven by a cam shaft located in the engine block. The number of cam lobes and unit pumps determines the number of pump strokes per full engine rotation. Typically 1 pump stroke per cylinder per 2 full engine rotations i.e. one pump stroke per injection. Unit pumps are less suited for the standalone research setup since they would require a custom cam shaft and custom (rigid!) cam + pump housing. Given the lower fuel volume demand for the single injector setup an alternative is to use a separate, light duty fuel pump (of a 2-2.5 L engine), driven by an electric motor. For long injections it might take two pump strokes to restore rail pressure rather than a single stroke of a larger pump but this is irrelevant when pump strokes fall between injections (see [4.3]).

Light duty injection pumps vary in mechanical layout, number of plungers and pressure rating though all are based on plungers driven by some form of cam and check valves to control in and outflow. Smaller combustion chambers, wider engine speed ranges and cost + weight concerns mean that the maximum rail pressures seen in light duty systems (up to 2700 bar currently) are lower than those in heavy duty applications (currently exceeding 3000 bar). Since the aim is to study high pressure injections the pressure rating of the pump is one the decisive aspects. To the best of my knowledge some variants of the Bosch CP-4 pump offer the highest rated pressure of all current light duty fuel pumps at 2700 bar.

High operating pressures result in high surface loads on cam and cam follower, lubricating films are critical for reducing wear and friction. The lubricating flow also serves as a cooling agent for the pump (prevent excessive plunger piston expansion and possible binding) and as a sealing agent for the plungers. Light duty pumps are typically fuel lubricated so attention should be paid to the lubricity and cleanliness of the fuel supply (this is especially important when working with novel research fuels). Activation of the pumps should be sequenced such that the feed pump starts before the HP pump, preventing it from running dry and wearing prematurely.

**Pump flow monitoring and rail pressure control**

For shot to shot consistency and measurement repeatability the rail pressure at the start of each injection should be identical. Common rail pressure control strategies take two main forms:

- Always operate the HP pump at full capacity, the rail pressure regulator drains excess fuel to reach the desired pressure level. This is the simplest strategy but results in a high return flow and excess heat generation which needs to be cooled to maintain a constant supply temperature.
- Use some form of pump flow control to adjust the pump delivery quantity to the injected quantity. Sometimes a slight over capacity is maintained and ‘trimmed’ by the rail pressure controller to speed up the system response. With this strategy the return load is reduced or even eliminated. A downside is the increased complexity and typically a longer response time to changes in pressure setpoint. For the research setup this is not a concern since the experiments can be designed to avoid rapidly changing rail pressure setpoints.

Controlling the pumped quantity can be done on the pump inlet or outlet side. Outlet monitoring uses a pump control valve (PCV) to actively close of a bypass channel. To limit the pressurised quantity per stroke the valve remains opened for a specified duration of the compression stroke. When the valve closes the bypass is blocked off, initiating the start of compression. The amount of fuel present in the compression chamber at the start of compression is the amount that can ultimately be pumped to the rail. Small pumps tend not to have PCV’s this feature is more common on unit pumps in heavy duty applications.

Flow monitoring on the intake (suction) side relies on restricting the pump intake. An intake restriction means the pump element does not fill to maximum capacity during the intake stroke. For pumps where cam-plunger contact is not maintained by a spring preload (as is common on radial cam-ring type pumps) the plungers lose contact with the camp. When the plunger position is positively prescribed an inflow restriction can drop the chamber pressure to below vapour pressure. The chamber partially fills with
vapour which means the total liquid volume in the pump chamber is lower than the chamber volume. These two styles of intake monitoring rely on different types of intake monitoring valve (IMV). There are proportional IMV’s where the relative valve area determines the inflow restriction and oscillating IMV’s which open and close for each pump stroke to meter the amount of ingested fuel. Proportional IMV requires less complex drive hardware, only a control voltage proportional to required amount of flow. Oscillating IMV’s are potentially more accurate but require a positive closing signal for each pump stroke.

For the new setup I would advise using a rail pressure control strategy with intake monitoring, combining a slow-acting proportional IMV and a direct acting rail pressure regulating valve. The use of intake monitoring reduces the heat load on the fuel tank and potentially eliminating the need for a return flow cooler. A proportional IMV drastically simplifies the drive hardware requirements compared to an oscillating IMF. For optimal rail pressure accuracy the IMV can be set to deliver a slight overcapacity, a rail pressure regulator can then reduce the rail pressure to the desired value before the start of injection.

4.2.2 Impingement target

Though the impingement method seems decidedly simple, care must be taken to ensure the spray is impinged correctly. Key are: impingement target geometry, size and positioning.

**Target geometry and material**

For an accurate momentum flux measurement the spray impingement is critical. An ideal target deflects the jet perfectly in radial direction, meaning the jet has no residual momentum in its initial direction. Rather than a flat target an axisymmetric concave-peaked target might be used. Such a shape will give a more gradual jet deflection and help the flow leave the target in a more uniformly radial fashion. Rather than reflect/bounce back the spray is gradually deflected.

![Figure 13: Schematic representation of two axisymmetric impingement plate geometries. The pointed design on the right provides a more gradual flow diversion.](image)

Material choice for impingement targets is important, a high pressure fuel jet can easily erode soft materials like aluminium. If pitting occurs on the impingement surface the flow deflection becomes rough and some of the spray may be deflected back towards the nozzle, overestimating the momentum flux. A hard steel can be used to make the targets, care must be taken to leave a smooth surface finish.

**Target sizing and positioning**

Wrong target size or proximity can introduce systematic errors. Obviously too small a target will not completely impinge and radically deflect the entire jet, underestimating the momentum flux. There are however some more complex effects to consider. During the initial stages of a spray event the tip of the spray is moving towards the target. In this process it is slowed down by air entrainment, spreading and drag. The oncoming liquid accumulates at the head of the spray during this opening transient, resulting in an increased magnitude of the initial spray impact. The tip-accumulation effect gives an non-physical momentum peak at the opening stage and some temporal shift. This effect is especially prominent when injecting into a hot, dense environment and when a large target-nozzle distance is used (over 20 mm), allowing the jet to partially evaporate between nozzle and target. Behind the tip, surrounding the liquid core, a low pressure gas region forms, when this impacts the target a drop in force is detected which is again not representative of the actual momentum flux. Figure 14 shows the resulting force plot, adapted from [8].
The low pressure hit can be avoided by selecting a smaller target, allowing the low pressure air to pass around the target. A better workaround is to reduce target-nozzle distance to below 5 mm as is done by many authors ([16], [17], Embersson [7]) goes as small as 0.5mm (!). This way the jet has less time to evaporate, entrain air and accumulate momentum. Working with a less dense ambient also helps but might affect the injector behaviour.

For the impingement force measurement it is possible to use a force sensor or adapt a pressure sensor by gluing the strike cap directly to the pressure sensing face ([9] & [17]). When adapting a pressure sensor the impingement target base must be narrower than the pressure sensing face diameter. A narrower interface gives higher sensitivity and more accuracy since transfer to the measurement membrane is improved (at the edges force is transferred directly to the sensor body). Knox [17] presents a direct comparison of impingement force measurements made with a piezoelectric force transducer (Kistler 9215) and a piezoresistive pressure transducer (PCB113B26). His findings suggest that the pressure transducer provides a cleaner, more precise signal than the force transducer. This is attributed to electrical noise in the cables (1m long) leading to the charge amplifier used with the Kistler sensor. The PCB pressure sensor has an integrated charge amplifier. When using an external charge amplifier the sensor-amplifier cables must be shielded and as short as possible to reduce electric noise. In the majority of publications on impingement based momentum flux measurement a Kistler 9251 force transducer is used.

4.2.3 Miscellaneous

Regarding the rest of the designs some aspects could be considered:

- Rail pressure can be measured by a combination of an OEM membrane based rail pressure sensor and a piezoresistive research type pressures transducer. The membrane sensor is free of drift and can be used for rail pressure control. A piezoresistive sensor is better for dynamic measurements (faster response rate) but does suffer from drift, which can be corrected for based on the other sensor reading. Placed at the injector line inlet it can provide information on pressure pulse timing. For more information on pulse dynamics a third pressure sensor can be placed at the injector inlet by adding an adapter block between the fuel line and injector inlet. In this adapter a thermocouple can be added to track fuel temperature. The advantage of an adapter block is that the fuel lines nor injector have to be altered, for each new injector fitting a new adapter block can be made (cheaply)

- To facilitate spray collection the injection chamber can be tapered towards the bottom and a fuel drain (tap or screw) added. Coating the walls of the chamber in a liquid repellent medium helps improve collection. Mounting the force transducer from the top of the chamber means the walls can be kept completely clean and smooth. Adding a viewing window to the chamber helps verify target alignment and impingement quality.

- The positioning stage can be reduced to 2 rotation axes and a translation axis. Combining a rotation axis coincident with the centreline of the injector and one centred at the injector tip a sphere around the nozzle can be described. A translation stage then sets the impingement distance.
4.3 Data processing

With the jet impingement force measurement setup jet momentum can be measured directly. For spray and nozzle characterisation quantities of interest, besides momentum flux, are: nozzle coefficients, mass flux and injection delay. In this subsection, methods are discussed for determining these factors from a measured force/momentum flux signal. All used quantities were introduced in section 2.2. Data processing examples using injection model data are presented to illustrate the concepts and illustrate the influence of potential error sources.

**Mass-flux from momentum flux**

Expressions for mass flux and momentum flux (eq:2) are identical barring a factor $u_{\text{eff}}$. Assuming the effective velocity remains constant over an injection it is possible to base the scaling ($\dot{M} \rightarrow \dot{m}$) on a measurement of overall injected mass. In case $u_{\text{eff}}$ is constant the following relation holds:

$$\dot{m}(t) = \frac{\dot{M}(t)}{\int_0^t \dot{M} \, dt} = \left(\frac{\dot{M}}{\int_0^t \dot{M} \, dt}\right)^{1/2} m_{\text{tot}}$$

Rearranging terms gives the desired expression for instantaneous $\dot{m}$ as function of instantaneous $\dot{M}$ in terms of measured overall injected fuel mass and time integrated momentum flux [3]

$$\dot{m} = \sqrt{\frac{M \rho_f A_0}{\int_0^t \dot{M} \rho_f A_0 \, dt}} m_{\text{tot}}$$

Applying a single corrective scaling has an appealing simplicity but there are some downsides: it omits any distinction between the various discharge effects and assumes $C_a, C_v, C_d$ and $\rho_f$ are constant during the entire measurement interval. Furthermore accuracy of the results is limited to the accuracy with which all injected fuel can be collected and weighed. Collection/weighing errors have a linearly proportional effect on the calculated mass flux values.

Figure 15 shows results from the data processing routine applied to model data. When all nozzle coefficients and liquid density remain constant over the measurement, mass flux can be reproduced exactly from measured $\dot{M}$, the computed line exactly coincides with the baseline. A third line shows computed mass flux with a 5% error in measured total mass, this has the effect of scaling the entire result by the same value. Variable nozzle coefficients have a more complex effect. To illustrate this a case has been simulated where the $C_a$ is proportional to flow-rate, following relation 12. This could represent a non-constant cavitation effect: maximum cavitation occurs at maximum flowrate, no cavitation occurs at low flowrates. For the processed mass flux signal this leads to an under-estimation at low flowrates and an over-estimation at large $\dot{m}$, skewing the results rather than merely scaling them. A lower than average $C_a$ results in a higher velocity for the same volume flowrate and therefore more momentum. When this is not taken into account and merely the average area coefficient is used that momentum flux will be associated with a higher mass flux. This will mainly cause problems when a the volume flowrate varies significantly during the experiment as is illustrated by the pilot + main signal of the example dataset.

$$A_Q = A_0 \left(1 - (1 - C_a) \frac{Q}{Q_{\text{max}}}\right)$$

Other potential error sources include:

- Force sensor drift, this will skew the results potentially underestimating initial flow whilst overestimating flow at the end or vice-versa.
- Measured momentum flux values, spray tip accumulation, oversized targets and target-nozzle distance introduce errors between the instantaneous nozzle momentum flux and measured force signal. Errors in measured momentum flux are reflected in the computed mass flux signal by their square root.
Figure 15: Mass flux traces computed from momentum flux data based on total mass scaling, showing baseline result, effect of mass weighing error and effect of flowrate dependent area coefficient.

**Nozzle coefficients from momentum flux**

In section 2.2 all relevant definitions were established the next step is to obtain expressions in terms of measured quantity \( \dot{M} \). For the moment it is assumed that all parameters required to compute the ideal solution \( (u_{\text{eff}}) \) are measurable, though it must be mentioned that measuring the injection pressure according to the definition used in the derivation of relation 1 is not trivial as will be addressed later.

When momentum flux can be measured directly determining \( C_m \) is simple:

\[
C_m = \frac{\dot{M}_{\text{measured}}}{2A_0(p_i - p_b)}
\]  

By combining the definitions for actual mass and momentum flux given in equation 2 it is possible to express both \( u_{\text{eff}} \) and \( \dot{m}_{\text{act}} \) in terms of measured momentum flux \( \dot{M}_{\text{measured}} \):

\[
\dot{M}_{\text{measured}} = \frac{\rho_f}{\sqrt{2A_0}} (p_i - p_b)
\]

\[
\dot{m}_{\text{act}} = \frac{\sqrt{\rho_f A_{\text{eff}}}}{\sqrt{2A_0}} \dot{M}_{\text{measured}}
\]

Combining expressions 15, 4, and 3 it is possible to define the discharge coefficient in terms of the area coefficient and measured momentum flux:

\[
C_d = \frac{2A_0C_a^2(p_i - p_b)}{\frac{\Delta m_{(0-t)}}{\Delta t A_0 \rho_f u_{\text{th}}}} = \frac{2A_0C_a^2(p_i - p_b)}{2A_0(p_i - p_b)}
\]

Rearranging yields an expression for \( C_a \) in terms of \( C_d \):

\[
C_a = \frac{2A_0 C_d^2(p_i - p_b)}{\dot{M}}
\]

This is useful since the (time averaged) discharge coefficient can be determined by measuring the overall injected fuel mass after a number of cycles and computing the theoretically injected fuel mass over the same duration, see equation 18.

\[
C_d = \frac{\Delta m_{(0-t)}}{\Delta t A_0 \rho_f u_{\text{th}}}
\]

In cases where no cavitation occurs, \( C_a \) is constant at 1 and relation 16 simplifies to relation 19. This makes it possible to determine \( C_d \) without knowledge of (time integrated) injected mass.

\[
C_d|_{\text{no cavitation}} = \sqrt{\frac{\dot{M}}{2A_0(p_i - p_b)}}
\]
Data processing example

Computing the discharge coefficient in accordance with definition [18] from model data the results presented in figure [16] are obtained. As reference case a pilot + main injection is used, since needle lift is not constant the discharge coefficient remains zero for the majority of the event.

Figure 16: Area and discharge coefficients determined from momentum flux and theoretical discharge based on rail pressure for a simulated pilot+main injection

Since the coefficients are defined at full needle lift this section of data has to be filtered from the overall dataset. When no needle lift signal is available the experimental injection has to be long enough to ensure maximum needle lift is reached, start and end effects have to be cut from the dataset. Figure [17] shows the computed discharge and momentum coefficient for the momentum flux data where needle lift >0.7 max needle lift. Both coefficients are computed for two cases: the injection pressure difference (p_i − p_b) with p_i measured at the rail and p_i measured at the nozzle chamber. According to the definition the ideal velocity is based on the limit case of Bernoulli flow through the nozzle. This means that p_i should be measured in the sac volume at the nozzle entrance. On a physical injector the nozzle chamber/sac volume pressure can not realistically be measured, inserting a pressure sensor at that location would compromise the structural integrity of the injector. Rail pressure can easily be measured but is not directly representative of nozzle chamber pressure.

The results show a number of interesting trends. Firstly the coefficients computed from rail pressure show a lot of variation over time whilst the nozzle chamber pressure based coefficients are nearly steady. This difference is the result from pressure waves and other pressure effects that occur between the rail and nozzle chamber. Pulses in outflow are out of phase with pulses in rail pressure, whereas they are directly related to variations in nozzle chamber pressure. When averaged over the duration of the dataset coefficient values obtained by both methods match to within 1 %. Determining nozzle coefficients based on rail pressure is therefore a viable option if full needle lift is sustained for several ms and an averaged value is taken. Note the slight rise and dip at the start and finish of the dataset for the nozzle chamber pressure based coefficient values. This is indicative of non-steady state full-needlelift conditions.

Figure 17: Area and discharge coefficients at full needle lift, theoretical discharge based on rail pressure and nozzle chamber pressure.
For completeness the non-cavitation corrected method of determining $C_d$ has been evaluated in the same way. Results are shown in figure 18 for both methods 18 and 19 applied to cases with known Ca values of 1 (non-cavitating flow) and 0.8 (cavitating flow). The cavitation-corrected method gives the same discharge coefficient for both cavitating and non-cavitating flows. Under non-cavitating conditions the second method yields exactly the same $C_d$ values, indicating the validity of that method under those circumstances and the validity of the mass-corrected method under all circumstances. Analysing a cavitating flow with the non-cavitation corrected method results in an over-prediction of discharge coefficient. This is due to momentum originating from a higher velocity being interpreted as a higher nozzle mass flow in absence of volume flow-rate correction.

![Figure 18: Discharge coefficients at full needle lift, computed with cavitation corrected and non-cavitation corrected methods for cavitating and non-cavitating flows.](image)

It should be noted that for ideal data the area coefficient can be determined exactly by using the overall mass based discharge coefficient and relation 17.

**Injection timing**

When analysing injection timing related aspects from the measured force data a correction has to be made for the impingement distance. It will take a finite amount of time for the spray to cross the distance between nozzle and impingement plate, the force signal lags behind the actual nozzle momentum flux. Computed discharge velocity can be used to predict delay between initial impact and start of nozzle flow.
4.4 Fuelsystem sizing

Injection behaviour is influenced by the design and sizing of the fuel system, both on an engine and in the experimental test setup. The new experimental setup should be usable for characterising a variety of heavy-duty (typically between 1.5 and 2.5 l/cylinder) sized injectors from different manufacturers and from different engine applications. The fuel system on each engine will have a specific combination of rail volume, injector line length and pump timing. Ideally different injectors should be characterisable without having to make changes to the fuel system on the experimental setup (barring some fittings). This calls for a universally sized fuel system and an experimental routine able to work around any performance alterations introduced by the different fuel system.

The setup is primarily designed to measure nozzle coefficients and characterise injector behaviour for short, transient, and close-spaced multiple injections where maximum needle lift is not reached. Experimentally obtained nozzle coefficient values are input parameters for spray models. Measured dynamic injection behaviour mainly serves as validation material for developed injection system models. Fuel system sizing will be approached in view of these two applications. To this end the developed fuel system model is used to evaluate differences in obtained coefficient values and injection response for varying component sizing. Model development and validation are addressed in sections 5 and 6.

Nozzle coefficients

Nozzle coefficients are used to relate the actual nozzle flow to a theoretical ideal nozzle flow based on the injection-pressure difference. As discussed in section 4.3 the pressure difference over the actual nozzle is difficult to determine, rather the difference between ambient and rail pressure is typically used to determine the theoretical flow. This introduces an error since the pressure pulses at the nozzle are typically out of phase with those at the rail and differ in magnitude, introducing a variation in the computed instantaneous discharge coefficient. To arrive at constant values for the nozzle coefficients a long injection is performed and the computed nozzle coefficients are time-averaged. Figure 19 shows the effects of varying rail volume and injector feed-line lengths on nozzle flowrate and computed discharge coefficient (based on rail pressure) for a typical range of HD rail volumes (20 to 60 cm$^3$) and line lengths (20-40 cm). For these simulations pump and injection do not overlap. An increased fuel-rail volume decreases the amount of pressure drop over the injection event, resulting in a lower drop in outflow rate over the duration of the event. Pressure pulses in the fuel line cause a ripple in the injection rate trace, halving the line length halves the frequency of this ripple.

![Figure 19: Nozzle flowrate for various rail volume and injector feedline length combinations, and corresponding nozzle discharge coefficients based on rail pressure. Time averaged discharge coefficients for all cases are shown in red, mean values range from 0.8323 to 0.8328. Injector actuation 5 ms.](image-url)
Despite the clear differences in injection rate and instantaneous discharge coefficient, the time-averaged discharge coefficient values for the 4 evaluated cases range between 0.8323 and 0.8328, an expected variation of less than 0.1%. This low sensitivity to fuel system variations can be explained as follows: the effect of rail pressure drop is accounted for in the theoretical flow value and pressure fluctuations at the nozzle average out over time. From these results it can be concluded that fuel system sizing is irrelevant for experimentally determining nozzle coefficients, provided that the measurement is long enough to include a whole number of pulse cycles.

**Short injections**

Short and close-spaced multiple injections are primarily affected by pressure pulses in the injector feed line. To study the expected effects a model simulation was performed for a sequence of two short injections, actuation duration of 0.3 ms, spaced 1 ms apart. Maximum achieved needle lift was around 10% of maximum travel. Cases were simulated for 20 and 40 cm injector feed line lengths. Figure 20 shows nozzle flow-rate and nozzle chamber pressure traces as well as needle lift. Solenoid actuation timing and needle lift are included to give an impression of delays.

Figure 20: Model predicted nozzle flow-rate, needle lift and nozzle chamber pressure for a sequence of two short close spaced injections, actuation duration of 0.3 ms spaced 1 ms apart. Simulated for 20 and 40 cm injector feed line length.

From the results it can be seen that first short injection is near identical for both line-lengths, this makes sense since no pressure waves have yet been created. Waves triggered by the first event do affect the second, for both line-lengths a lower maximum needle lift and flowrate are achieved on the second injection. Here line-length does matter especially actuation timing relative to pulse timing. To illustrate the importance of relative timing another simulation was performed on the 20 cm line model with the second actuation delayed by half a pulse duration (0.4 ms). The resulting maximum needle lift and nozzle flow-rate are higher and nearly the same as for the 40 cm line, indicating the actuation occurs at a similar stage in the pressure pulse cycle.

Though the injector line length has a clear influence on close-spaced injection performance, the relevance of these effects should be evaluated in the context of dynamic injector characterisation. Measured short-actuation ROI responses serve as model validation material. To this end the model can be used to simulate the properties of the experimental setup and be validated with the measured data. Having
tuned and validated that model actual on-engine performance can be predicted by simply altering the injection system parameters where necessary. It is therefore not critical to replicate the exact on-engine system. For characterising and tuning the short-actuation response of an injector it is advisable to study the first actuation starting from a pseudo-steady state.

**Additional considerations**

For determining nozzle coefficients it was shown that maintaining rail pressure during the 5ms injection is not required to obtain a consistent $C_d$ value. This means that it is not necessary to feed the fuel rail during the injection, pump events can be timed between subsequent injections. Given the model-based nature of short-injection performance analysis effects of altered pump-timing can easily be accounted for in the validation model. Between injections the set-point rail pressure can be restored by the pump, when necessary with multiple pump strokes. Taking the 5 ms coefficient determining experiment as a baseline, a pump capacity of at least 0.6 ml per stroke is required when allowing for a single stroke between injections. The capacity requirement can be halved when two pump strokes are allowed between injections. Most light-duty automotive pumps fall within this range, the pump selection can thus be based on other factors like maximum pressure rating and flow control.

A smaller rail volume is inherently safer since less energy is stored, in case of component failure this limits the potential damage. A downside to a smaller rail is that pressure fluctuates more for a given fuel mass addition or extraction. Controlling the pre-injection rail pressure to the same level of accuracy requires a faster, more precise, control strategy.

**Conclusion**

From the performed analysis the following conclusions can be drawn regarding system sizing and evaluation strategies:

- The first short injection from a fuel system at pseudo steady state is not affected by fuel system sizing and offers a good baseline injector response for model validation. Subsequent short injections are affected by pressure waves in the injector feed line, the magnitude of these effects can be ascertained by altering injection spacing.
- For determining nozzle coefficients fuel system sizing is irrelevant (within the typical heavy duty range). Coefficient values averaging should be done for an interval length equal to a whole number of injection line pulse cycle times, this can be done in post processing.
- Pump timing is not a concern for the experimental characterisation. Pump strokes should be separated from the injection events, starting each injection from a steady rail pressure.
- I would suggest using a rail volume of around 40 cm$^3$, in line with a typical heavy duty average.
- I would suggest using a short injector feed line of around 20 cm, faster pulse dynamics mean they are more clearly identifiable in shorter injections. Though this parameter can be altered to suit practical requirements.
5 Hydraulic-mechanical fuel system model

One of the critical aspects of the experimental momentum flux measurement setup is the fuel injection system. The consistency, accuracy and predictability of the generated fuel sprays determine the usability of the setup. In order to gain insight into the performance of the fuel injection system a model of it is developed. This model can be used to perform sensitivity studies on various system components, predict the response of the constructed setup and help guide and support sizing decisions and eventual component selection. In this section the developed hydraulic-mechanical injection system model is presented and modelling choices are explained.

Type of model & requirements

Formation of the fuel spray is governed by two main parameters: opening dynamics of the injector and hydraulic pressure in the system. In the heavy duty reference engine solenoid injectors are used, therefore the opening dynamics also depend on hydraulic pressures in various sections of the injector. For accurate predictions the fuel spray model has to take into account:

- Fuel pressure in various sections of the injector most notably the control chamber and nozzle chamber. These pressures are affected by wave dynamics originating from injector opening and closing events, which have to be accounted for.
- Mechanical force balance on the injector needle, this is a result of hydraulic + spring forces and must include mechanical deformation of the needle and needle seat.
- Pressure in the fuel rail and injector lines, these are a function of flow pulses coming from the high pressure pump and fuel demand from the injector. Pressure pulses in fuel lines are included.
- Pump dynamics, outflow of the high pressure fuel pump is the result of pump speed, geometry and control. Dynamics of pump valves are important and cause pressure waves in downstream fuel lines.

An efficient way of modelling such a system with multiple different components and physical domains is to use a lumped approach. For the mechanical aspects this means a combination of mass-spring-damper models with component stiffness. Discrete fluid capacity blocks at uniform pressure connected with 1D lines and 0D resistances make up the hydraulic model. Fuel lines are assumed one dimensional allowing the inclusion of pressure-wave dynamics.

The full fuel system model consists of hundreds of conservation equations for all individual discrete components. Several commercially available programs can be used to construct and solve such models, most notable are Siemens AMESIM, (open) modelica and Matlab Simscape. These programs are all GUI based and provide similar capabilities. It was decided to use Simscape since the full academic license is available to TU/e staff and students. Of AMESIM and modelica only restricted student versions are available, these lack some functionalities (especially solver selection options and several hydraulic components). It must be mentioned that the full commercial version of AMESIM does offer a more extensive library and range of models than Simscape.

An alternative approach would be to implement all governing equations in a programming language and use that to solve for the system response. This gives full control over the implemented governing equations and provide a lot of insight. It was decided not to do so since the commercial packages probably have better optimised solving routines, are less error prone and are ultimately more transferable than custom code. Furthermore, model implementation can be done a lot faster.

Background information Simscape

Simscape is a GUI based physical modelling environment available in Matlab Simulink. The generic Simscape library includes multiple physical domains, each has a specific colour associated with it (hydraulic = yellow, mechanical = green). For this work the extended fluids library was also used. Components blocks are connected with physical signals (in red), these include all appropriate parameters for the governing equations associated with the connected blocks. In the block components initial conditions can be specified with a level of priority. At the start of a simulation the system of initial conditions is solved first, when necessary low priority IC’s are violated. Simscape blocks can be combined with Simulink blocks in a common model. The user has control over which solver is used, in the same way as in a standard Simulink model.
5.1 Governing equations

Each Simscape building block is comprised of a series of conservation equations for the parameters relevant to that physical domain. The developed models consist of mechanical and hydraulic elements. Conservation equations for the mechanical systems are assumed common knowledge (mass-spring-damper systems), the used hydraulic equations will be introduced here.

An important decision regarding the hydraulic models is whether to include thermal aspects or not. Changes in fuel pressure (due to pumping work or resistance to flow) cause changes in fuel temperature. Since the diesel has a non-zero thermal expansion coefficient the temperature change will have an affect on the pressure-density relations. Furthermore other fluid properties like viscosity and bulk modulus show a temperature dependency. For ultimate accuracy one would have to model the hydraulic system as a thermal-liquid model. Timescales for the injection events are in the order of ms, much lower than those for heat exchange, thus the injection events can be approximated as adiabatic. Using a full adiabatic thermal-liquid model does have a number of downsides. Firstly the number of equations and required boundary conditions increases significantly, this makes solving the initial value problem and subsequent simulation more expensive. Furthermore the added complexity requires more knowledge about the system and more extensive datasets to validate the implementation. Therefore it was decided to use an isothermal hydraulic model, validity of this simplification will be addressed during model validation.

The hydraulic components can be divided into two main types of element: resistive and capacitive. Capacitive elements represent fluid volumes and allow for accumulation and pressure rise. The generic governing equation for capacitive elements is given by equation 20.

\[
q_{\text{acc}} = \frac{d}{dt} \left( \frac{\rho}{\rho_0} V \right)
\]  

(20)

Instantaneous fluid accumulation \(q_{\text{acc}}\) is a function of vessel volume change \(\frac{dV}{dt}\) and density change \(\frac{d\rho}{dt}\), where vessel volume change can be the result of wall stiffness or (in the case of hydraulic cylinders) piston movement. Fluid density is a function of pressure and depends on the relative gas content as per equation 21, a limited amount of entrained gas is a means of approximating cavitation behaviour.

\[
\rho = \frac{\left( \frac{\alpha}{1 - \alpha} \right) \rho_g^0 - \rho_l^0}{\left( \frac{\alpha}{1 - \alpha} \right) \left( \frac{\rho_0}{\rho} \right)^{1/\gamma} + e^{-\frac{\rho - p_0}{\beta_l}}}
\]  

(21)

With liquid bulk modulus \(\beta_l\), relative gas content \(\alpha = \frac{V_{\text{gas}}}{V_{\text{liq}}}\) and gas + liquid densities at ambient conditions \((\rho_l, \rho_g)\) and ambient pressure \(p_0\). For accumulators with flexible walls the container volume change is generally expressed in terms of proportionality constant \(K_p\) [m/pa]. In Laplace domain \((s)\) the chamber diameter then becomes:

\[
d(s) = \frac{K_p}{1 + \tau_s} p(s)
\]  

(22)

With \(\tau\) the characteristic time scale of wall displacement. Flowrate over a local (non-dimensional) resistance is a function of the available pressure drop over that resistance \((p)\), flow area \((A)\) discharge coefficient \((C_d)\) and the current flow regime (turbulent or laminar). The flowrate correlation used for both the constant and variable flow resistances in Simscape is given by equation 23.

\[
q = C_d A \sqrt{\frac{2}{\rho}} \left( \frac{p}{(p^2 + p_{cr}^2)^{1/4}} \right) \quad \text{with} \quad p = p_a - p_b
\]  

(23)

Here the flow regime is accounted for through the pressure ratio compared to the pressure ratio at critical Reynolds \((p_{cr} = p(Re_{cr}))\) as per equation 24 where \(D_h\) is the hydraulic diameter (equivalent circular pipe diameter).

\[
p_{cr} = \frac{p}{2} \left( \frac{Re_{cr} \nu}{C_d D_h} \right)^2 \quad \text{with} \quad D_h = \sqrt{\frac{4A}{\pi}}
\]  

(24)

Pressure drop due to fluid inertia is dependent on the instantaneous flow acceleration and pipe length and crossectional area:

\[
\Delta p = \rho \frac{L dq}{A dt}
\]  

(25)
5.2 High pressure fuel pump model

In subsection 4.2.1 types of HP injection pump and methods of flow monitoring were introduced. All conventional systems are based on plunger elements displaced by an engine-driven cam with valves to control in- and outflow. In the following section the developed Simscape model of a generic plunger-pump element is discussed. With this basic model simulations can be constructed for a range of pumps. Figure 21 shows the developed generic pump element model, in Appendix parameter values are provided for various simulated pumps.

Figure 21: Overview of developed single element plunger pump model with PCV

Valves

Flow into and out of the pump chamber is controlled by check valves, the amount of valve opening is a function of the pressure difference over the valve. For example: upon compression the outlet valve starts to open after the plunger chamber pressure has reached rail pressure + the required cracking pressure. When a specified maximum pressure difference is reached the valve is fully opened. In this application the amount of valve opening is not simply a linear function of the existing pressure differential, valve opening and closing dynamics have been included. During the opening and closing transients the rate of area change is governed by the valves dynamical timescale as per equation 26.

\[ \frac{dA}{dt} = \frac{A_{(p)} - A}{\tau} \]  

\(A_{(p)}\) is the steady state opening area at a given pressure difference. In this application it is important to include realistic valve transient characteristics, otherwise there would be high derivatives of valve open area resulting in high derivatives of flow rate. Such high changes in flowrate would result in non-physical, high magnitude, pressure waves in the fuel lines connected to the pump inlet and outlet. A pump outlet chamber volume, between the outlet check valve and high pressure fuel line, has been modelled explicitly since it provides further pressure pulse damping.

Flow monitoring

As discussed pump flow metering concepts for constant-contact plunger pumps are based on creating a variable amount of ‘dead’ pump stroke. Ways of doing this include valve control (on either the inlet valve or a bypass valve to the return line) or fluid cavitation methods. The latter relies on creating high inflow resistance, pressure drops below the vapour pressure and the plunger chamber is partially filled with vaporised fuel. During the initial pump stroke the charge first condenses, compression starts when the chamber volume has reached the total enclosed fluid volume. Even though the actual mechanisms of flow monitoring are different the resulting fuel delivery to the rail is the same. Whether the start of compression is delayed by leaving a valve open or by fluid condensation, the dynamics after compression starts are only determined by the outlet valve and cam profile. Since we are only interested in the injection event only the pump discharge characteristic is of interest. In the model flow monitoring is implemented by means of a bypass valve connecting the plunger chamber to the inlet circuit. By keeping this valve open during the initial pump stage or opening it some part through the pump stroke it is possible to simulate the working of a PCV, cavitation based intake monitoring or an outlet bypass valve.
Prescribed plunger displacement

Most plunger based high pressure fuel pumps use springs to preload the plungers against the drive cam\(^1\). Assuming infinite drive stiffness and adequate preload, the plunger motion is directly determined by the cam profile and cam position: plunger displacement is prescribed. In the Simscape model the fuel pump plungers are represented by a hydraulic-mechanical converter block whose position is determined by an ideal position source. Ideal position sources are not default components in Simscape (version 2017b), a workaround was created based on an ideal velocity source with a feedback control loop as shown by Figure 22.

![Figure 22: Ideal position source](image)

Input for the ideal velocity block is the position error signal (difference between current plunger position and target plunger position) multiplied by a constant gain. By using high gain feedback (gain = 1e6) and small simulation stepsize the system shows excellent position signal tracking. Observed position errors are in the order of 1e-6mm for stroke lengths in the order of millimetres. Fluid compressibility in the pump chamber has been modelled explicitly, this is not accounted for in the standard hydraulic-mechanical converter block.

High pressure fuel line

Connecting the pump outlet to the fuel rail is a high pressure fuel line. Since the pumps are often remote mounted (near the engine drive) the lines are typically several tens of centimetres long. Combined with typically internal diameters of several mm and abrupt changes in flow rate when the pump outlet valve opens or closes this results in pressure waves in the fuel line. In order to be able to represent this aspect, segmented line models with flexible walls were used (available as standard components in Simscape). Each line is discretised into segments (for this simulation 20 segments were used), figure 23 shows the equivalent circuit for a line segment.

![Figure 23: Schematic equivalent circuit of a segmented fuel line segment, ports N and N+1 connect to adjacent segments.](image)

Each segment consists of a local resistance and fluid inertia in series together with an accumulator volume that includes wall flexibility and fluid compressibility. By increasing the number of segments the magnitude and velocity of pressure waves in the line can be computed more accurately.

---

\(^1\)One notable exception being internal cam opposing plunger pumps, where the plungers are pressed against the cam by fuel pressure + centripetal force. This design is typified by intermittent cam-plunger contact.
5.3 Injector model

Solenoid injectors use an electro-magnetic solenoid valve to control needle lift and with it opening and closure of the injector. Figure 24 shows a schematic overview of a Bosch solenoid injector, the developed model is based on this injector design and adopts the same naming convention. Before discussing the developed model we will first briefly introduce the working principle of a solenoid injector.

Fuel flow through the nozzle is controlled by a needle valve, when the needle lifts from its seat high pressure fuel can discharge through the nozzle. The needle is preloaded onto its seat by a spring and fuel pressure exerts forces on two opposing faces/shoulders of the control plunger-needle combination. In steady state, with the solenoid valve closed, pressure in the control chamber equals system pressure and the combined hydraulic and spring preload forces on the needle keep it pressed onto its seat. When the electromagnetic solenoid is actuated it pulls the solenoid valve off its seat (against its preload-spring). Opening this valve allows fuel to be discharged from the control chamber through the A-throttle (flow restriction) to the return line. Fuel flow into the control chamber from the high pressure system is restricted by the Z-throttle, resulting in a pressure drop in the control chamber. Hydraulic force exerted on top of the control plunger drops whilst the force exerted in opposite direction on the needles' pressure shoulder remains the same. The resulting hydraulic force pushes the needle of its seat, initiating the injection. When the solenoid valve is de-energised it is closed by its preload spring, control chamber pressure rises and the needle is pushed back onto its seat.

Figure 24: Solenoid injector schematic  
Figure 25: Overview of developed simscape injector model
Model basics

Figure 25 shows the developed Simscape hydraulic-mechanical injector model (appendix A provides an enlarged copy). Both the solenoid valve and needle+control plunger assembly have been modelled as mass-spring-damper systems. The electromagnetic electromagnetic actuation force is represented by an ideal force source, this is a simplification of the complex electromagnetic charge build-up in the coil. Hydraulic pressure forces on the top of the control plunger and on the needles’ pressure shoulder(s) are represented by two hydraulic-mechanical converter blocks. These blocks also account for the volume change of the control chamber and nozzle chamber upon needle motion. All hydraulic components are lumped volumes and resistances, only exception being the connection between inlet and nozzle chamber. In the reference injector this is a drilling of 13 cm long. To account for the occurrence of pressure waves, the drilling is represented by a segmented tube.

Needle motion end of travel

If the system is pressurised and the injector at rest, the combined spring preload and hydraulic forces push the needle onto its seat. Both the needle and needle seat (injector housing) have limited stiffness thus the applied load results in an elastic deformation of needle + seat in negative direction. Operating at a system pressure of 1400 bar, the expected negative needle displacement is in the order of 0.15 mm [15]. When the injector is actuated the needle first has to overcome this deformation before the injector actually opens. Typical positive needle lift values are around around 0.3 mm so overcoming the initial deformation significantly increases the required needle travel. The elastic deformation thus changes the effective start of injection relative to the start of actuation. Needle and seat stiffness also play an important role during injector closure. If the needle were to bounce or oscillate, as would be the case for an infinitely stiff needle, some residual fuel would escape after the injection event.

In the model the needle and needle seat stiffness have been combined into a single ‘translational hard stop’ block which comes into effect if the zero position is reached. The end-stop has a finite stiffness and damping, damping is also applied upon rebound. To prevent discontinuous derivatives the stiffness and damping are gradually applied through a transitional region of 0.01 mm, so one order of magnitude lower than the steady state deformation. Figure 26 shows an example of needle lift and flow rate, note the initial negative needle displacement and oscillations around steady state (resulting from water hammer in the nozzle tube). The injection starts as soon as the needle position becomes positive, due to the needle valve geometry the transient rate gradients are much steeper than those of the displacement transients.

![Figure 26: Needle lift and injection flow rate at 1200 bar rail pressure](image)

Unlike full closure, maximum needle lift is not determined by a hard mechanical stop. When the control plunger lifts it starts to restrict outflow through the A-throttle, this changes the balance between inflow and outflow resistance and thus the pressure in the control chamber. Rising pressure increases the closing force on the control plunger until an equilibrium position is found. In the model this needle lift limiting mechanism is implemented using a second ball valve upstream of the A-throttle. Valve position is linked to the needle position with an offset, when this offset is overcome the A-throttle is blocked off fully. A large ball diameter and square edged seat are used to represent the top of the control plunger.

Nozzle model

The reference heavy duty injector uses a sac-type nozzle with 7 nozzle holes, orifice diameters and tube length are known. In the model the nozzle has been implemented as 7 resistive tubes in parallel, the entrance to each nozzle hole is a sudden area change block. The sudden area change represents flow resistance due to the change in direction and vena-contracta effect. Properties of the resistive tubes have been set to those of the nozzle, the overall flow rate at reference conditions (100 bar steady pressure) has been tuned to experimental data using the discharge coefficient in the sudden area-change block.
5.4 Baseline parameter values & model validation

For optimal relevance of the sensitivity study results, ideally all injection system model parameters would be based on the reference heavy duty research engine. A solenoid injector has a lot of critical internal properties, values for which are not generally available and not released by the manufacturer. Given the limitations on physical lab access, physically measuring properties of injectors from the reference engine was not possible during this project. As a workaround for the current study, an available set of measured values from another heavy duty solenoid-injector were used. Values were taken from an injector of a heavy duty 6 cylinder engine (slightly smaller than the reference engine at 1.53 l/cylinder rather than 2.15 l/cylinder), as measured by X.Seykens on a disassembled injector [19]. This dataset includes almost all mechanical and hydraulic injector parameters: spring stiffness’s + preload values, component masses, hydraulic line lengths and diameters, throttle , orifice diameters and chamber volumes.

Nozzle tuning
Nozzle orifice dimensions, as specified by the manufacturer, were available for the reference injector. This injector has a nozzle diameter of 0.195 mm, the nozzles of the 1.53 l/cyl injector are 0.183 mm, reflecting the higher volume flow demand from the larger reference engine. Besides the nozzle dimensions a 100 bar steady nozzle-flow value is provided: 1.7 l/min @ 100 bar. The 100 bar steady flowrate measurement is a standard injector characterisation measure, it corresponds to the flow of fuel through the nozzle when a steady pressure difference of 100 bar is applied over the nozzle (between sac volume and the outer face of the injector). To tune this flowrate, the correction coefficient in the sharp edged area contraction block of the nozzles was used. Without correction the flow rate was overestimated by around 20%. Note that the nozzle model is purely pressure based, fluid cavitation is not explicitly modelled. The entrance effect correction coefficient therefore does not directly relate to the discharge coefficient. It merely increases the total nozzle resistance to reduce the steady state 100 bar flow rate to the rated level.

Needle and seat stiffness
The set of measured parameter values does not contain information on needle or needle-seat stiffness. What is available is a paper on injector modelling from the same author in the same period which presents experimental data on nozzle displacement for various rail pressure levels [18]. For this work the same heavy-duty injector was used from which the dimensions are provided, making this as near to a direct comparison as possible under the current circumstances. Figure 28 is a direct copy of the needle displacement over time for two rail pressure levels presented in [18]. For practical reasons needle displacement is measured on the upper part of the injector (generally on a shoulder of the control plunger), the displacement value is therefore affected by both the needle and needle seat stiffness. It is therefore impossible, and for the results irrelevant, to assign values to these parameters separately. In the model needle and seat stiffness are lumped and tuned to obtain the same initial pressure based displacement. Maximum needle lift has also been set based in line with the data from this figure, this is done through changing the displacement offset on ballvalve modelling A-throttle shielding by the control plunger. Figure 27 shows needle lift traces for the tuned model.

![Figure 27: Simscape simulated needle displacement at various rail pressures.](image1)

![Figure 28: Measured and simulated needle displacement at various rail pressures, source: [18](image2)
Comparing the Simscape model results with the reference experimental results, the initial and maximum needle displacements are identical for both rail pressure levels indicating that the nett hydraulic force/s-tiffness combination is set correctly. The opening and closing times also match for both rail pressure levels, from this it can be concluded that the hydraulic force to spring force and damping ratios are accurate as well. A notable difference between the responses is the frequency at which the needle oscillates after full closure, Simscape predicts it is twice as high as in the literature example. Oscillation magnitude is comparable. A higher oscillation frequency can be caused by a higher seat stiffness or lower needle mass. In tuning the seat stiffness focus was on initial deformation. To obtain the same pressure related deformation for a lower seat stiffness the injector used in literature must have a lower nett pressure force i.e. a lower nett control plunger area/ needle-shoulder area ratio.

**High pressure pump**
Parameter values for the high pressure pump are set to those on the research setup. In a truck application the fuel rail on this reference engine is fed by 2, cam driven, unit pumps. Each cam has 3 lobes and rotates at half engine-speed, resulting in 6 pump strokes per 2 revolutions i.e. 1 pump stroke per injection event. On the test setup the fuel rail feeds only one injector and the rail is fed by one unit pump. The pump control valve of this one pump (variable dead stroke type) is only activated for 1 in 3 strokes, replicating the single pump stroke per injection event. Plunger diameter and stroke of these unit pumps are know and used in the simulations.

**Rail volume, pump timing and pump line length**
A ballpark value for the reference heavy duty common-rail volume was known to be around 40 cm$^3$. On the research engine the single research cylinder is fed from the normal engine rail at its original volume. Experimentally obtained rail pressure traces were available from the research setup, together with information on pump timing, fuel flow-rate, engine speed and injector actuation timing. Based on this data the pump phasing and rail volume were tuned. From the fuel flowrate values (expressed in g/s) and engine speed the injected volumes were determined. Injector actuation was adjusted to obtain the same injection flows for this comparison. Figure 29 shows the experimental rail pressure traces together with model results obtained after parameter tuning. Note that the rail pressure was measured using the factory original rail pressure sensor, not a calibrated research sensor. On the 1115 bar measured pressure trace clear peaks can be seen at t = 0.107 sec. These are probably due to measurement noise and not physical. Comparing the experimental and modelled results it can be seen that pump and injection timing are the same. The resulting rail pressure rise and drop match both in terms of magnitude and duration. When the pump check valve opens the sudden discharge of fuel causes pressure fluctuations in the high pressure connecting line, this effect can be seen in both the experimental and modelled rail pressure traces. The magnitude and frequency of the pressure waves are matched to within approximately 10%, indicating that the pump check valve dynamics and pump-rail line length are set correctly.

![Figure 29: Measured rail pressure on the single cylinder heavy-duty test engine together with rail pressure data from tuned model simulation. Model predicted nozzle flowrate is shown to illustrate timing.](image)

Experimental data courtesy of Robbert Willems
Pressure pulses triggered by the end of injection are not matched to the same level. These pulses originate in the line connecting pump to injector, the length of which was tuned based on another data set as will be discussed next. On the research engine the injector line length might be different. Another possible explanation for the discrepancy is the location of the pump and injector lines on the fuel rail relative to the pressure sensor. A rail length distance between line and pressure sensor will affect the measured pressure waves.

**Overall response and line-length effect**

In the paper that was used for the baseline needle and seat stiffness values, experimentally determined nozzle flow traces are provided for various rail pressure levels. As an overall validation of the injection system models, the same case was evaluated with the developed Simscape model. Results from the paper are shown in figure [31]. Simscape model predictions in figure [30]. The reference case used in the paper is a single long (5 ms) injection event, resulting in around 330 mg of fuel per injection. A characteristic feature of the injection flowrate traces is the ‘wave’ effect at full needle lift with a cycle time of around 2 ms. This effect is caused by pressure waves in the injector feed line (connecting rail to injector). The injector line length in the model has been tuned to obtain the same pulse frequency as in the experimental results (35 cm). To illustrate the line-length effect a case was simulated with a line of 20 cm at an intermediate pressure value of 1100 bar. The resulting pulse frequency is nearly twice as high (approximately 1.2 ms cycle time) as can be seen in the flow-rate trace of figure [31]. The different pulse timing slightly affects the opening transient since the first positive pressure pulse falls slightly earlier than for the other cases, at t = 2 ms the nozzle flow-rate is higher than it would have been for shorter line lengths. In the context of a 1 ms injection event (∼100 mg per injection) a shorter line would thus result in a higher maximum injection rate since the higher pulse flow level is reached earlier.

![Figure 30: Simscape simulated injection rate profiles at various injection pressures.](image)

![Figure 31: Measured and simulated injection rate profiles at various injection pressures, source: [18](https://example.com)](image)

Overall the results show a clear match with the experimental results. Both the opening and closing rates are the same as well as the pulse related pressure fluctuations. Effects of rail pressure are of the same type and magnitude. Based on these results it is felt that the selected baseline parameters result in realistic injection behaviour and the developed model is capable of accurately predicting the nozzle flow-rate for a heavy duty-injection event.
6 Solenoid valve model

In the previously developed injector model the solenoid valve was simplified to an ideal force source and mass-spring-damper system. Though a good starting point this simplification is insufficient for accurately predicting the dynamic response of the injector. The solenoid valve dynamics are especially relevant for short (pilot) injections and close-spaced multiple injections where the injector possibly does not fully open. A dynamic solenoid model is investigated and constructed as an extension to the main injector model and will be used in the dynamic sensitivity analysis. Given the physical limitations during this project the upcoming section is meant as a phenomenological study rather than a quantitative one. The structure presented below can serve as a starting point for future electric-magnetic-mechanical (FEM) based modelling.

6.1 Solenoid actuator dynamics

Solenoid-based magnetic flow control valves used in common rail injectors are a type of variable-reluctance actuator. Magnetic reluctance is a concept that can be viewed as the opposition of a section of material to magnetic flux passing through it. Geometries vary but the basic function is the same for all solenoid injectors. Figure 32 shows a schematic representation of a solenoid actuator.

![Schematic representation of the electromagnetic solenoid valve from a common rail injector, drawn axis-symmetric](image)

In this version the valve consists of a ball sealing on a conical seat (not drawn), more elaborated pressure-balanced versions exist. The ball is backed up by the valve body, termed armature, which is preloaded closed by a spring and can move relative to the main injector body. A coil of wire in the main body produces a magnetic field when current is passed through it. This magnetic fields creates a magnetic flux looping around the coil (passing through the coil center). Around the top and sides of the coil the flux loop passes through the highly permeable injector body (termed yoke), this yoke does not close the loop around the bottom of the coil. Since the bottom of the coil is not encased in a highly permeable material, the path of least reluctance passes through the gap separating armature and main valve body and through the armature itself. The lowest energy state of a magnetic system it that were the flux path has the lowest reluctance. In the flux carrying bodies this gives rise to a force acting in the direction that would decrease the overall flux path reluctance, which in this case means a force acting to close the gap between armature and yoke. Since the yoke is fixed to the injector body the resultant motion, after overcoming the preload force, is lift of the armature and with it opening of the ball valve.

Before looking at the governing equations let us introduce the engineering concept of Magneto Motive Force (MMF), symbol $F$ [A]. The concept of MMF is a tool used to determine fluxes in magnetic circuits. It can be thought of as the magnetic circuit equivalent of EMF for electric circuits in describing the ‘driving force’ of a magnetic field. Mathematically the MMF experienced on a closed flux loop of length $l$ through a magnetic field $H$ [A/m] is defined in terms of loop integral\(^{27}\)

$$ F = \oint Hdl $$

(27)
When a current $I$ is passed through a coil with $N$ windings the MMF that is created by the coil is given by expression (28).

$$\mathcal{F} = NI$$

(28)

From expression (28) it is clear that the unit of MMF is indeed amperes since the number of coil turns is dimensionless. To avoid confusion with electric current the unit is often termed ampere-turns instead though the physical relevance is the same.

The magnitude of the resulting magnetic flux $\phi$ [Wb] over a flux path is a function of the driving MMF and the overall magnetic reluctance of that path $R_{tot}$ [H$^{-1}$] as per definition (29).

$$\mathcal{F} = \phi R$$

(29)

Definitions (28) and (29) can be combined to find the magnetic flux generated by a solenoid coil, expression (30). The total flux is a function of the number of coil turns and driving current as well as the total reluctance of the flux loop $R_{tot}$.

$$\phi = \frac{NI}{R_{tot}}$$

(30)

For a section of material of constant cross-sectional area $A$ [$m^2$] and magnetic path length $l$ [m], the magnetic reluctance is given by relation (31). Here the materials permeability $\mu$ [H/m] is expressed relative to the permeability of free space $\mu_0$ [H/m] with $\mu_r$ a dimensionless proportionality factor.

$$R = \frac{l}{\mu_0 \mu_r A}$$

(31)

When the flux path of a loop consists of multiple materials or varying cross sections the reluctances of individual sections can simply be added to obtain the total reluctance.

The magnitude of the reluctance force exerted on the armature is a function of magnetic flux $\phi$ and the relative change in reluctance for a relative armature displacement $dx$:

$$F = -0.5 \phi^2 \frac{dR}{dx}$$

(32)

Based on expression (31) and assuming a constant cross sectional area of the flux path, the derivative of reluctance for the gap between armature and yoke becomes:

$$\frac{dR}{dx} = \frac{1}{\mu_0 \mu_r A}$$

(33)

Note that neither the direction nor the magnitude of the reluctance force are a function of the magnetic field direction. The reluctance force always acts in the direction that would minimise the gap reluctance, therefore it is not possible to actively close the solenoid valve by reversing the current direction.

Equations (27) till (33) provide the basis for valve opening force as function of solenoid properties. In order to obtain the total dynamic response of the solenoid two electromagnetic effects have to be considered. Firstly the gap reluctance varies as a result of armature displacement, leading to changes in magnetic flux and with it changes in reluctance force. The second complicating factor is that of inductance.

According to Faraday’s law, when the coil is subject to a change in magnetic flux an electromotive force $E$ [V] is induced which acts to oppose the change in magnetic field. The induced voltage limits the current (or generates a reverse current when no forward voltage is applied). Equation (34) provides the relation linking magnetic field change and induced current $I$.

$$E = -\frac{d\phi(t)}{dt} = L \frac{dI}{dt}$$

(34)

Parameter $L$ is the inductance [H], for a long coil of wire $L = \mu_0 \frac{N^2 A}{l}$ with $l$ the height of the winding stack [m] and $A$ the area enclosed by the coil wire. An important realisation is that the coils self inductance will limit the rate with which the magnetic field strength rises when the solenoid is activated. Furthermore when the armature lifts the gap reluctance decreases, the resulting increase in magnetic flux induces a back EMF in the coil, again limiting the available MMF.
Additional effects
The governing equations introduced in the previous paragraph describe an idealised solenoid-based variable-reluctance actuator. In order to accurately describe the response of an actual injector a number of additional magnetic and mechanical effects should be taken into account. Especially under transient operating conditions these factors become significant.

• Eddy currents
  The inductance phenomenon does not only occur in the solenoid coil itself, the metal yoke and armature also exhibit a type of inductance. When subject to a changing magnetic field, current loops form in the material, so-called eddy currents (sometimes called Foucault currents). The magnetic field associated with these current loops acts to oppose the change in external magnetic field. Eddy currents in the yoke and armature thus diminish the amount of flux through the actuator gap and with it the available reluctance force. Since eddy currents are related to changes in flux density they will occur upon activating and deactivating of the solenoid and during armature travel, in all cases increasing the response time. In 0 dimensional models eddy currents are typically modelled by a single shorted winding (20, 21). The eddy current winding is inserted in the magnetic flux path, changes of flux induce an EMF in the winding. A limited resistance in the eddy current loop determines the magnitude of the effect. This way of modelling eddy currents stays true to the actual eddy current phenomenon and is included in the developed model.

For more advanced modelling of eddy current effects one has to consider the so-called skin effect and unbroken loop length. In current carrying devices current density is highest at the outside skin of the body unlike for magnetic flux carriers. The limited penetration depth will affect the magnitude of the eddy currents. Apart from the rate of change of magnetic field strength the unbroken eddy current loop length also affects its magnitude. By segmenting magnetic flux carriers the length can be limited, reducing the undesirable effects (22). When the armature or yoke of an injector are slotted, this might be done to limit eddy currents and should be modelled explicitly. Modelling such complexities is best done with 3D FEM simulations. Given the 1D nature of the models developed in this study and limited validation data, the skin end loop length effects have not been included.

• Magnetic saturation, hysteresis and residual magnetism
  In general the flux yoke and armature are made of highly permeable ferro-magnetic materials. Ferromagnetic materials are non-ideal flux carriers, suffering from saturation and hysteresis. Their B-H curve (flux density V external magnetic field strength) is not a straight line. The material has a saturation limit, from a certain point increasing the field strength does not significantly increase magnetic flux. Actuation currents are generally designed to avoid saturation, limiting unnecessary coil heating and stress on drive electronics. Hysteresis means that a given field strength yields a lower flux when the field is increasing than when the field is decreasing: the B-H curve depends on loading history. After completely removing the external field the material can have some residual magnetic field. The cause for these effects can be found in the re-aligning of internal permanently magnetised domains, hysteresis and residual magnetism are caused by residual alignment of domains. To accurately model hysteresis effects extensive calibration (possibly based on 3D FEM simulations) is required. Such work is outside the scope of this project. To avoid introducing complexities that cannot be validated the saturation and hysteresis effects were not included in the model.

• Flux leakage
  The magnetic permeability (flux carrying capacity) of the metal yoke and armature is higher than that of air ($\mu_r \approx 4000$) but some magnetic flux will still pass around the yoke and armature. Therefore the reluctance force will be somewhat lower than if all flux lines would intersect the armature as is assumed in the basic model. The amount of flux leakage can be predicted using an (axis symmetric) FEM model if the actuator geometry and material properties are known. In the developed model flux leakage is not included, since no information is available on total flux and gap flux it is not relevant to include a leakage parameter. Rather the overall response is tuned to obtain the desired behaviour.
Fluid squish

In most common rail solenoid injector designs the magnet valve itself is submerged in fuel, often the return drainage passes through the valve body. Typically the gap between the solenoid coil/yoke and valve armature is in the order of 0.1 mm. When the valve is actuated the armature is pulled towards the yoke and the gap closes. During this motion fluid is expelled from the gap. Since the ratio of radial length to gap height is large ($\approx \frac{7}{0.1}$) and diesel has a relatively high viscosity a significant pressure differential is required to realise the radial outflow rate. Increased fluid pressure in the gap provides a force that resists armature lift. Faster armature lift means the fluid has to be expelled faster and more pressure is built up. This mechanism effectively acts as a viscous damper for the armature motion, though its action is non-linear. For optimal accuracy the viscous radial flow and pressure buildup should be modelled explicitly [23]. In this study fluid squish was not modelled in detail, in view of limited validation possibilities it was felt that unnecessary complexity is best left out. In view of the phenomenological nature of the study the viscous damping effect was included in the model but by means of a simple linear viscous damper.
6.2 Simscape model

In the simscape foundation library zero dimensional (lumped) electric and magnetic component blocks are available. These have been used to construct a lumped parameter, zero dimensional electro-magnetic solenoid model. Mechanical aspects of the solenoid valve have remained the same as in the original injector model apart from the addition of a hydraulic pressure force acting on the valve ball. Figure 33 shows an overview of the developed model, different colors signify the various domains (electrical, mechanical, magnetic and hydraulic).

![Diagram of the developed solenoid model in Simscape](image)

**Figure 33**: Overview of solenoid model developed in Simscape, includes electrical, magnetic, mechanical and hydraulic submodels.

Starting with the solenoid coil, this is represented by a coil model which interacts with the magnetic domain according to equations 28 and 34 regarding the generated MMF and induced back EMF. Coil resistance is set by a separate resistance block. The supply voltage is regulated by a controller (as will be introduced later) and adjusted depending on instantaneous coil current, to this end a current sensor is included for feedback purposes.

The magnetic circuit starts at the solenoid coil which provides the driving MMF. Constant reluctance blocks describe the reluctance of the yoke and armature. It has been split into three segments, each representing a section of yoke/armature body with a constant cross-section, in line with what was done by Payri [116]. Eddy current effects are included by means of a single shorted winding. The resistance of this winding determines the induced current in the winding for a given flux change and with it the generated reverse MMF. Reducing the winding resistance increases the magnitude of the eddy current effects. Completing the flux loop is the reluctance actuator gap, modelled by a reluctance force actuator block. This block implements equations 31, 32, and 33 to model the force exerted on the armature and position dependent gap reluctance. In the reluctance force actuator block the initial and residual gap height are included as well as the end-stop stiffness at full armature lift. End-stop stiffness at full armature closure is included in the mechanical model.

On the interface of the magnetic and mechanical domains lies the reluctance force actuator model block. For the rest the mechanical domain consists of a simple mass-spring damper system and a hard end-stop block. Force exerted by fluid pressure on the valve ball is included by means of a hydraulic-mechanical converter block. The pressure area is equal to the A-throttle area and mainly contributes an opening force when the valve is completely closed.

**Injector driver**

Valve opening and closure is controlled by means of a voltage supplied to the coil by an injector driver. Solenoid actuation protocols are often expressed in terms of target coil current values, the injector driver switches the supply voltage to achieve and maintain the desired currents. Actuation protocols have two distinct stages: opening and hold. During the opening stage a high coil current is used to quickly open the
valve. Typical opening-current set-points are around 15A and last for around 200 µs, to reach and sustain the desired current level the driver supplies a 'boost' voltage, typically 50V or higher, generated by an internal boost circuit. During the hold stage the solenoid force must be high enough to hold the valve body against the spring force. For this a lower current setpoint is used (typically around 8A), maximum driver output is generally limited to the available battery voltage (24V). Coil current is reduced during the hold stage to limit the amount of power required, strain on drive electronics and heat generation in the solenoid coil. When, during either stage, the specified maximum current threshold is exceeded the controller reduces the driving voltage to a lower level, for the boost stage this is typically battery voltage, during hold it generally switches to ground.

At the end of actuation a reverse voltage is applied to the coil, this serves to quickly draw down the existing magnetic field and speed up the closing response. A powered inductor stores energy, this energy is drawn out through the coil. As mentioned previously the direction in which the reluctance force acts is independent of the magnetic field direction. Therefore a reverse current cannot generate an active closing force. The active reverse voltage is switched off as soon as zero current is reached.

For the Simscape model a simple injector 'driver' has been developed. Upper and lower current thresholds can be specified for both the boost and hold stage. When the measured coil current falls below the lower threshold the driver switches to the maximum voltage specified for that stage (boost or battery). If the current rises above the upper threshold the driver switches to the lower specified voltage (battery or 0). At the end of actuation reverse boost voltage is applied until the coil current has decayed to zero. The overall result is a characteristic saw-tooth current pattern. For non-equal current rise- and fall rates it might be necessary to set the upper and lower thresholds with a different offset from the desired mean value. Figure 34 shows the workings of the designed injector driver, for this demonstration it was connected to a simple test circuit consisting of a resistor and inductor placed in series.

![Supply voltage](image1)

![Coil current](image2)

Figure 34: Supply voltage and circuit current traces showing the workings of the designed power supply. Actuation consists of an opening (boost) phase, hold stage and active current draw-down. Tested on a simple circuit consisting of 2.5 Ω resistor and 0.03 mH inductor in series.
6.3 Baseline parameter values & model tuning

A major difficulty in developing the solenoid model was obtaining a representative set of parameter values. Given the limitations on lab work, analysing an actual injector was not possible during this project and unlike for the mechanical injector properties no complete set of data was available from other works. The following sources of data were available and together formed the basis on which the model parameters were selected:

- For the actual reference injectors there is data on target driving voltages and resulting solenoid currents but no information regarding the internal structure (number of windings, coil resistance, reluctance gap etc.).
  
  Target actuation is as follows: during the boost phase driving voltage should be 54 V, coil current should rise to 17 A within 55 $\mu$s. Boost voltage is sustained for 200 $\mu$s, at which point the valve should be fully opened. To hold the valve open a hold current of 7A is specified with a range of plus and minus 2 A, during the hold phase driving voltage should switch between boost (54 V) and battery voltage at 24 V. To increase valve closure speed a negative boost voltage is applied (-54 V) until the coil current has reduced to zero. It is believed that the specified hold-voltage specifications are erroneous, if 54V is enough to exceed 17A (otherwise no switching would be required during the boost phase) then 24V applied to the same coil cannot result in less than the average hold current of 7A. A more likely scenario is that during the boost phase the power supply switches between boost and ground or battery and ground.

- The dataset on the older HD-solenoid injector, measured by X.Seykens \[19\], that forms the basis of the rest of the injector model also contains some information on the solenoid valve of that injector. Available data: armature mass, spring stiffness and spring preload. No information is given on electrical aspects like coil resistance, windings and actuation voltages/currents.

- Payri et.al \[24\] developed a full 1D injection system model including a solenoid valve model and provide detailed information on internal structure including parameter values. Provided information is: number of windings, coil resistance, magnetic flux path dimensions, initial reluctance gap and mechanical properties. This paper however fails to specify driving voltages, resulting solenoid currents and most importantly the rate with which the valve actually opens. Key information taken form this study is the initial air gap value (80 $\mu$m) and the flux path reluctances.

- Huber \[25\] provides experimentally obtained solenoid position data and modelling work including parameter values, but fails to specify number of coil windings and actuation voltage/current values. The presented valve opening speed matches that specified for the reference injector (sub 200 $\mu$s to open fully).

- Another work that provides detailed solenoid parameter values is the recent paper by Zhao et.al on eddy current effects in the dynamic response of high speed solenoid valves \[22\]. Information on coil resistance, number of windings, spring stiffness + preload values, initial and residual gap values and armature mass are given as well as coil current and driving voltages. Though interesting the provided values are not representative of those for the reference injector. Reason to believe this is that the presented current response differs from those specified for the research engine. The rate of current rise is much slower, taking 200 $\mu$s to reach 17A rather than the reference target of 55 $\mu$s. Boost voltage is sustained for 600 $\mu$s rather than 200 $\mu$s and maximum opening takes almost 400 $\mu$s from the start of actuation.

Combining all available information, parameter values were set according to:

- Valve spring stiffness and preload values from the measured dataset by Seykens were used directly, they should work well with the rest of the injector model. Armature mass and valve geometry (ball diameter, orifice diameter and cone angle) are also taken from this dataset.

- For the initial (110$\mu$m) and residual reluctance gap (30$\mu$m) an intermediate value was chosen in the range presented in the works of Payri \[24\], Zhao \[22\] and Huber \[25\]. The resultant total armature lift (80$\mu$m)is enough to reach maximum flow area with the 1.35 mm valve ball in a conical seat sealing a 0.4 mm orifice.
• Reluctance gap area and reluctances for the yoke and armature were set to values presented by Payri [24]. Given the high relative permeability of the yoke material (4000) this is not a critical factor.

• Representative values for the lumped eddy current loop resistance were difficult to obtain. Li [21] proposes values between 0.27 and 0.37 for a gasoline-direct injector which has a different design than the solenoid valve on a diesel injector (larger coil, 93 turns, and lower coil resistance, 1.09Ω, thus more flux). As a starting point the eddy current loop was initially set at 1 Ω, this is likely to underestimate the eddy current effect. In the sensitivity study this will be addressed by exploring values spanning a larger range than for other parameters.

• Coil resistance is set to obtain the desired steady hold current. A value of 3 Ω was selected, together with a battery voltage value of 24V applied during hold this gives a steady hold current around 8A, in line with specifications. If during hold the powersupply is switched between boost and ground with a duty cycle of 0.5 the average voltage will be around 26V, keeping the mean current value in the correct ballpark.

• Having established all other parameters, the number of coil windings was tuned to obtain the desired response. The number of windings affects the amount of flux, and with it the amount of force, generated for a given current value. Besides generated force it affects the total inductance, which in turn controls the current rise rate. These effects combined determine the transient opening response. The goal was to reach 17A coil current during the boost phase and completely open the valve in sub 200µs. To achieve these fast transient responses the number of coil windings had to be set at a low value of 17 turns. For reference, the valve used by Zhao has 52 coil turns but this results in the aforementioned 600µs opening time [22].

A full overview of all baseline values (and value references) is presented in table 8. Figure 35 shows the resultant coil current and valve lift traces for a 2 ms total actuation and 200 µs boost duration, actuation starts at t=0 ms. Maximum valve opening is reached after 178 µs, well within the specified 200 µs opening window. During the opening stage a maximum coil current of 15.6 A is reached, this is lower than the specified 17A. In order to reach the specified value the number of coil windings would have to be reduced to around 12, this is much lower than any value encountered in literature. Since the opening transient is accurate it was decided not to pursue the coil current value by further reducing the number of windings. Looking at the coil current response note that the coil current reaches a maximum just as the valve starts to open, this is the result of inductance due to induced flux as the overall reluctance decreases. After the armature has reached full lift the current starts to rise again, at 200 µs the driver switches to hold voltage, allowing the coil current to stabilise at 8A.

![Figure 35: Coil current and valve position traces for the baseline Simscape solenoid model, 200 µs opening boost, total actuation of 2 ms.](image-url)
In order to verify that the developed model and selected baseline parameters give the correct valve response, results are compared to results presented by Huber [25]. Though the solenoid valve used in his paper might differ slightly from the reference solenoid in this study the nature of the behaviour should be comparable. Figure 37 shows a measured and modelled armature position response from literature for a 2 ms total actuation. Figure 36 shows simulation results from the developed model. Note that the developed model uses a larger total armature lift \( 80 \mu m \) compared to the \( 55 \mu m \) for the literature case. Comparing the overall responses it can be seen that the simscape model predicts very similar behaviour to that found by Huber. Opening and closure delays are comparable (180 \( \mu s \) opening delay and 80 \( \mu s \) closure delay) and during the hold phase the armature remains steadily at full lift. End-stop stiffness at either end of full armature travel is slightly lower in the model than in the literature example, this results in a larger oscillation amplitude and lower oscillation frequency. In the simscape model endstop damping is slightly stronger, the oscillations are damped out completely in 0.5 ms \( V \) the 0.7 ms found by Huber. Since stiffness and damping values will differ for each injector design the observed discrepancy is not worrying, phenomenologically the behaviour is the same indicating stiffness and damping are implemented correctly.

Given the close match of simulated behaviour with that presented in literature and the close match of opening rate with that specified for the injector, it is felt that the developed model is phenomenologically correct and the selected baseline values are in the correct ballpark. The selected baseline should be an adequate starting point for the sensitivity analysis. Remember that at this stage the solenoid model is not meant to exactly replicate the behaviour of the valve on the reference injector. Given the lack of available data and limitations on physical measurements for characterisation and validation this would be an unrealistic objective.
7 Sensitivity study

Given the limitations on physical experimentation and construction that plagued this project the developed Simscape injection-system model has become the core analysis tool. An interesting possibility the model offers is to perform a parameter sensitivity study, mapping the effect each individual parameter has on performance quantities of interest. A sensitivity study can improve the understanding of a system and provide useful insights for future modelling, design and analysis work.

For the current study the overview of sensitivities will be evaluated in three main contexts:

- **Experimental setup design and operation**
  Staying within the original project context, the experimental injection characterisation setup will to some extent differ from an on-engine injection system. From the sensitivity study it could tell us which aspects are critical to the system operation and which aspects can be dealt with more loosely. Main points of interest are the fuel system (pump, fuel lines rail etc.), which will differ markedly from the engine version, and the actuation signal which will be generated by different electronics.

- **Injector characterisation for model input**
  Accurately modelling a physical injector depends largely on being able to measure the relevant mechanical properties, think of hydraulic diameters, volumes, masses, stiffness’s and preload forces. Determining these properties is not trivial, especially when the injector cannot for whatever reason be disassembled. In this context the sensitivity analysis can help identify which quantities should be measured especially accurately. Furthermore it can help identify the most likely causes for any discrepancies between model and measurements should those occur. Properties that are difficult to measure accurately but have a large influence on any particular result will be prime suspects when experimental results differ from model predictions. Having a cause-effect maps can help direct and support model parameter tuning decisions.

- **Relevance of operating conditions and expected lifetime changes**
  Operating an injector for extended periods of time changes will occur, components heat up causing properties to change spring stiffness and electrical resistance being two notable examples. When performing experiments analysing individual injection events, the number of injections per experiment is much smaller than for a running engine. This means the system might not reach a thermal steady state during experiments resulting in changing performance over the experiment. Having an idea of what effects could be expected from changing spring stiffness and solenoid coil resistance can help when interpreting experimental data. Along similar lines, changes can occur in an injector over time. Parts will wear, materials can work-harden, electrical properties can change, fouling can occur, fluid passages can clog or wear wider. When testing used injectors and comparing them to new examples differences might show up. Understanding the different performance changes expected from various wear mechanisms can help with interpretation. This information can also be interpreted with an eye on preventative maintenance, linking performance changes to internal wear.

**Limitations**

Important to state upfront is that this sensitivity study is based purely on the injection models developed in Simscape. As mentioned previously the baseline model parameters are based on values used by others ([18], [24]) and tuned to obtain responses similar to what is presented in literature. Given the limitations on practical experimentation and availability of experimental data this is the best that could be done during this project. Results of this sensitivity study should be interpreted with these limitations in mind, focussing on relative changes and relative importance of parameters rather than absolute values.

7.1 Process

The basic principle of a sensitivity study is simple: for a set simulation case define characteristic quantities and log how these values change for alterations to the model input variables. With characteristic quantities are meant aspects of the simulation result that characterise the behaviour of interest.

Given the multi-physics nature of the injection system (hydraulic, mechanical, electrical and magnetic) the quality of the results benefits from a systematic and structured approach. To this end the sensitivity study is performed in three stages:
1. Fuel system + injector model with simplified solenoid valve
Solenoid injectors use hydraulic pressure forces to lift and close the needle, this hydraulic-mechanical coupling introduces complexity. Apart from the needle a hydraulic pressure force is also exerted on the solenoid valve ball, altering the solenoid valve opening response. In order to isolate and zoom in on the effects of needle dynamics and hydraulic performance, the solenoid valve dynamics are simplified and its motion is set constant. To this end the solenoid force is approximated by a constant force driven by an external signal, hydraulic forces on the valve ball are neglected. With reproducible ball valve motion the effects of hydraulic properties (diameters, line lengths, absolute pressure) and mechanical properties of the needle (stiffness, spring stiffness, preload etc.) can be investigated.

2. Standalone solenoid valve model
This analysis is only concerned with the dynamic displacement response of the solenoid valve, actual injection behaviour is not considered. The interaction of electric, magnetic and mechanical 'forces' result in complex dynamics. Effects of input changes are best expressed in terms of a different set of characteristic parameters than the injection itself. Quantities of interest are: opening delay, coil currents, closing delay etc.

3. Full model
With the full injector model effects of solenoid valve dynamics on injection behaviour can be studied. Only parameters that significantly affect solenoid valve motion (as identified during the solenoid valve motion) will be investigated at this stage.

Figure 38: Schematic overview of the structure of the sensitivity study: Goal, process, methods etc.
Actuation reference case

In order to investigate the relevant injection system behaviour the correct actuation reference case has to be used. Of primary interest are short and close-spaced multiple injections, these provide a lot of information about the dynamic performance of an injector. A second interesting operation condition are long (main) injections during which a pseudo steady-state is reached (at maximum needle lift). To this end the following engine-operation reference case was chosen:

- Single pilot + main injection, no post injection. This corresponds to typical operation of the research engine and should showcase effects on both transient and steady state injector operation.

- 100 µs between pilot and main injection, corresponding to 1.2 crank angle degrees (CAD) at 2000 RPM. If subsequent injections are affected by pressure waves in the system, triggered by injector closure, this will show at the start of the main injection. By comparing the opening stages of the pilot and main injection any effects of pressure waves, triggered by valve closure at the end of injection, can be studied.

- Baseline pilot-injection volume of 2.5 mm$^3$ and main-injection volume of 110 mm$^3$ per cycle, corresponding to a typical heavy-duty highway load point. The large main volume is mainly meant to show up changes due to pressure drop in the fuel rail. Pilot volume is chosen such that the needle does not reach full lift during the pilot, highlighting transient performance.

Figure 39 shows the resulting nozzle flowrate over time for the selected reference case as obtained from the full simscape model (100 µs between first solenoid actuation for pilot and main).

![Figure 39: Reference case nozzle flowrate, pilot and main injection](image)

For the standalone solenoid model a single main injection reference case is used. This provides information about transient opening and closure from steady state. Fully transient operation and possible after effects on subsequent actuations will be evaluated during the full model simulations.

Characteristic quantities

From an engine-performance point of view the two most important characteristics of the injection system are: amount of fuel delivered and injection timing. Taking this as the basis for evaluating injection system performance the following characteristic quantities have been defined:

- **SOI- & EOI-delay.** Start of injection and End of injection delays, defined as the time between start of actuation (SOact) and first positive nozzle flow respectiveley EOact and last nozzle flow. With SOact actuation of the solenoid is meant, generally this means nonzero driving voltage, for the simplified solenoid case this corresponds to the first positive force exertion. SOI corresponds to the moment of first positive needle lift, assuming the undeformed needle seat is at zero displacement. These delays are expressed in µs and crank angle degrees @ 2000 RPM, the latter being more relate-able in terms of combustion timing. SOI delay is measured at the pilot since the system comes from complete rest and the fuel rail is in a pseudo-steady state. EOI delay is measured at the end of the main injection since the needle will have reached maximum lift here, unlike during the pilot. Since injection events have to be timed accurately for optimal combustion the delay between actuation and first result is important for injector control.

- **Vtot pilot & Vtot main.** Total injected volumes for pilot and main injection, determined by integrating nozzle flow-rate from start to finish of either signal. Volumes are expressed in mm$^3$. 
• **Q max main & Q max pilot.** Maximum nozzle flow-rates during pilot and main injection, summed for all 7 nozzle orifices. Expressed in mm³/ms since main injections have durations in the order of ms and typical values around 100 mm³. Maximum flow-rates are an indication of overall hydraulic efficiency and are directly related to momentum flow-rate. Since the nozzle orifice diameters are constant this provides a measure of relative spray momentum variations.

• **max needle lift pilot.** Maximum needle lift achieved during the pilot injection expressed in mm. During short injections like a pilot the needle typically does not reach maximum lift. By logging maximum needle lift observed differences in pilot quantity can be understood in more detail.

Figure 40 shows baseline results for the hydraulic-mechanical model, indicating the characteristic points.

![Figure 40: Needle lift, nozzle flowrate and actuation force for the baseline conditions, indicating all characteristic points. Quantities are normalised and scaled for clarity](image)

For the standalone solenoid valve analysis a different set of performance indicators are used. The solenoid valve provides the interface between the control signal and hydraulic injector circuits. An ideal solenoid valve responds instantaneously to control signals, characteristic values have been chosen to reflect this:

• **SOV lift delay & SOV closure delay.** Delay between SOact and Start of valve (SOV) lift respectively end of actuation and start of valve closure, expressed in µs. Valve closure is defined as the first instance the valve reaches zero position. Given inertia and finite stiffness of valve body and seat the valve can bounce, by its definition any bounce is not apparent in SOV closure. This parameter provides the a measure of the valve’s response rate.

• **Delay full valvelift.** Delay between SOact and first full valvelift, again expressed in µs. This parameter provides more information about the ratio of solenoid force to spring-, damping- and inertia-forces.

• **Current drawdown delay.** Time between end of positive actuation (positively opening the valve) and reaching zero current in the coil. Coil inductance means the coil current does not instantaneously drop to zero after removing the driving voltage, a reverse voltage can speed up current drop rates. This parameter serves to indicate the current drop rate and linked to overall inductance.

• **delay max coil current.** Time between SOact and reaching maximum coil current in µs. An extra parameter, mainly meant to give more insight into the opening transient.

• **Max current & Hold current.** Maximum coil current achieved during opening transient and mean hold current while valve is at constant opening [A]. The reference injector is current controlled, the driver switching the supply voltage to keep coil current between thresholds. Monitoring this parameter helps monitor controller performance.
Figure 41 shows baseline results for the solenoid model, normalised and scaled for clarity, indicating all characteristic points. Note the ballvalve bounce at closure and the indicated valve closure point.

![Normalized quantities chart](chart.png)

Figure 41: Ballvalve lift, coil current and source voltage for baseline solenoid model, indicating all characteristic points. Quantities are normalised and scaled for clarity.

For the full model analysis the same characteristic parameters are used as in the hydraulic-mechanical analysis complemented by some indicators from the solenoid analysis.

**Evaluation step and presentation of results**

For consistency and ease of comparison the target parameter value step is 10% of its initial value. A 10% increase or decrease is felt to represent a physically realistic range of change and should result in perceptible change of the characteristic parameters. Some exceptions have been made, mainly to ensure a positive pilot is retained. Changing some parameters by 10% would result in zero flow during the pilot, in these cases the values are tuned to obtain a minimum but positive pilot flow. Other exceptions are made for parameters subject to a larger range of conditions under normal engine operation (rail pressure and fluid temperature being most notable), parameters where the ballpark value is not well known (eddy current loop resistance, viscous damping) or the change does not lend itself to being expressed as a percentage (reluctance gap: expressed as percentage of initial or residual gap gives wildly different values). The relative change in parameter values is indicated in the results tables: 2, 3 and 4.
7.2 Results injector & fuel system analysis: hydraulic-mechanical

For clarity results from the different sensitivity analysis segments are presented in individual subsections. Tables showing an overview of all simulation results are presented at the end of each section. Presented results are characteristic values changes relative to the baseline case, expressed in the same units as the baseline values. It was decided not to present changes in percentages since the base units (seconds, mm$^3$ etc.) provide more feeling for their physical relevance. Conditional formatting is used to highlight the most important parameter changes for selected characteristic values, yellow being used for a value decrease and blue for a value increase. In the rest of each subsection the results are discussed and where relevant supported using further simulation results.

This section deals with the hydraulic-mechanical system analysis (without solenoid effects), results are presented in table 3, baseline parameter values can be found in tables 5 and 6.

**Mechanical parameters**

Results show that the following mechanical parameters have little effect on injection performance:

- **Pump-injection phasing.** Baseline pump timing places the entire pump stroke before SOAct, during the injection event rail pressure drops. The alternative is to have the pump stroke and injection event coincide, limiting rail pressure drop during the injection. Rail pressure at the start of injection was kept identical. From the results it appears that pump phasing hardly affects injection performance.

- **Needle spring stiffness & preload.** Needle spring stiffness influences initial deformation and the amount of force required to achieve needle lift. Results show that 10 % changes to the needle spring stiffness and preload result in sub-microsecond alterations in timing and negligible influence on injected volumes. Predicted effects would not be practically measurable. Needle stiffness can realistically be measured to within 10 % accuracy.

- **Control plunger mass.** Has a slightly larger influence on injection timing than the previous two parameters, influence on injected quantities is negligible. More mass means more inertia which causes the observed injection delays.

Parameters that do have a large effect on both injection timing and injected quantities (especially pilot) are the pressure shoulder areas of the needle and control plunger. Increasing the pressure area of the control plunger by 5 % results in zero nozzleflow during the pilot and delays SOI by 45.4 µs, corresponding to 0.5 CAD @ 2000 RPM. Meanwhile EOI delay decreases, which, combined with the increased SOI-delay, means the main volume decreases by almost 5%. Figure 42 shows needle positions and nozzle flow-rates. Both effects are caused by a stronger closing force exerted on the needle. Firstly the initial needle deformation increases which means more deformation has to be overcome before the needle reaches positive lift. Closing response speeds up since the force balance shifts to the closing direction at a lower control-chamber pressure and thus is reached earlier + a higher closing force is achieved. Increasing the needle pressure shoulder area has the opposite effects: faster opening, later closure and larger pilot volume.

![Figure 42: Nozzle flow-rate and needle position for increased needle and control plunger pressure areas](image)

Figure 42: Nozzle flow-rate and needle position for increased needle and control plunger pressure areas
Hydraulic parameters

Again starting with the parameters that have limited effect on performance:

- Nozzle chamber volume. Size of the collection chamber directly before the needle seat, having a volume of fluid here helps damp pressure pulses between injector inlet and nozzle. On the overall response the effects are negligible.

- Tube to nozzle chamber diameter. Has a negligible influence on overall hydraulic efficiency.

- Rail volume. Small variations of fuel rail volume have negligible effect on the injection event. A larger rail would result in lower rail pressure variation during the injection. It appears that the ratio of injected volume to rail volume is low enough at the chosen design point and 10% changes are within tolerance. The robustness to rail pressure variation during injection corresponds to that observed for pump phasing.

- Rail-injector line length. Effects on flows are negligible, SOI delay increases notably for decreased line length. This is likely the result of pressure pulse phasing.

- Injector line area. Negligible influence.

Hydraulic factors that do significantly affect performance are orifice diameters namely: Z-throttle, A-throttle and the nozzle orifices. Diameters of the nozzle orifices only affect nozzle flow, not the dynamic response of the injector (opening and closing transients). As expected an increase in nozzle orifice diameter increases the maximum flow rate during the main injection and main injection volume, indicative of a lower overall hydraulic resistance. No effect is seen for the pilot injection, limited needle lift means the hydraulic flow resistance remains dominated by the needle passage during the pilot.

Opposite is true for the A- and Z-throttle areas, they affect the dynamic response without altering the hydraulic resistance experienced by the injection flow. In the results this shows in the negligible changes in Q-max for the main injection. Interesting to note that the Z-throttle influences both the opening and closing speed whereas the A-throttle only affects the rate of opening. What this indicates is that the A-throttle flow does not play a significant role during needle closure. Figure 43 shows needle lift over time for various throttle areas, to get a sense of opening and closing delay ball valve position is also plotted. From this figure a clear difference between opening and closing transients can be discerned. Start of needle lift nearly coincides with start of ballvalve lift whereas needle closure starts after the valve has closed entirely. This supports the notion that the A-throttle diameter does not affect closing performance: the entire A-throttle is blocked off during valve closure. Meanwhile the rate of control chamber pressure rise is determined by inflow through the Z-throttle, explaining its large influence on EOI-delay (around 2 µs or 0.3 CAD @ 2000 RPM for a 10 % change in area). Both throttles do affect the opening response with opposite effects. Larger A-throttle and smaller Z-throttle result in faster opening, decreasing these flow areas has the opposite effect. Enlarging the outflow or restricting inflow reduces the chamber pressure at full valve opening, which means the nett lift force on the needle increases. It must be noted that the A-throttle has a lesser influence than the Z-throttle, this is probably due to its larger initial diameter (0.4 V 0.25 mm).

Figure 43: Ball valve and needle positions for varying A- and Z-throttle flow areas.
Rail pressure effect

Magnitude of rail pressure has a number of seemingly contradictory though logically explicable, effects. Start of injection delays with increasing rail pressure whereas end of injection decreases. Fully closed the area ratio of control plunger and needle shoulder means a nett closing force is generated. With increased pressure magnitude the nett closing force also increases. Since the needle + seat have finite stiffness a higher closing force results in more deformation. Figure 44 shows needle position and nozzle flowrate for various rail pressures. It can be seen that a higher rail pressure results in a larger negative needle deformation in closed position. Since this deformation has to be overcome before SOI a larger initial deformation delays SOI. Interesting to note is that the rate of needle lift actually increases with rail pressure due to the larger opening pressure force. This effect can be seen in the main injection where higher pressure cases overtake in terms of positive needle lift from 0.1 mm positive lift on. During the pilot injection this point is not reached, therefore lower pressure values result in earlier SOI. Upon closure the higher closing pressure force takes effect immediately (no initial deformation to overcome) and higher pressure results in faster needle closure.

Regarding flow-rates a similar trend can be seen. Steady state flow-rate at full needle lift increases with rail pressure as a result of higher hydraulic driving potential. Main injection volumes increase despite the faster closing transient. Pilot volumes actually decrease with pressure since the total needle open time is decreased from longer SOI delay and faster needle closure. On the whole higher rail pressure results in a sharper (more square) injection profile.

Figure 44: Nozzle flow-rate and needle position over time for a range of rail pressures
Table 2: Sensitivity study results of hydraulic-mechanical analysis. Presented values are differences compared to baseline case in specified units. Conditional formatting highlights the largest changes for selected characteristic values, yellow indicating a value decrease and blue a value increase.

<table>
<thead>
<tr>
<th>parameter</th>
<th>SOI delay</th>
<th>EOI delay</th>
<th>SOV closure delay</th>
<th>pilot max needle lift change</th>
<th>Q-max pilot</th>
<th>Q-max main</th>
<th>Vtot pilot</th>
<th>Vtot main</th>
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<td>unit baseline</td>
<td>µs</td>
<td>µs</td>
<td>µs</td>
<td>mm</td>
<td>mm³/ms</td>
<td>mm³/ms</td>
<td>mm³</td>
<td>mm³</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>simultaneous pump-injection</td>
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<td>0.4</td>
<td>-0.2</td>
<td>0.13</td>
<td>-0.001</td>
<td>-1.18</td>
<td>-0.06</td>
<td>-0.10</td>
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<tr>
<td>needle spring stiffness +10 %</td>
<td>0.8</td>
<td>-1.1</td>
<td>0.37</td>
<td>-0.001</td>
<td>-0.91</td>
<td>-0.01</td>
<td>-0.10</td>
<td>-0.22</td>
</tr>
<tr>
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<td>-0.8</td>
<td>1.5</td>
<td>0.07</td>
<td>0.001</td>
<td>0.90</td>
<td>0.00</td>
<td>0.10</td>
<td>0.22</td>
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<td>-0.01</td>
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<td>-0.029</td>
<td>-40.18</td>
<td>-0.52</td>
<td>-2.46</td>
<td>-5.39</td>
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<td>needle pressure shoulder area +3 %</td>
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<td>24.2</td>
<td>0.16</td>
<td>0.015</td>
<td>13.58</td>
<td>0.01</td>
<td>1.82</td>
<td>3.21</td>
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<tr>
<td>- 5 %</td>
<td>27.3</td>
<td>-29.9</td>
<td>0.34</td>
<td>-0.023</td>
<td>-30.79</td>
<td>-0.45</td>
<td>-2.21</td>
<td>-5.65</td>
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<tr>
<td>control plunger mass +10 %</td>
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<td>3.2</td>
<td>0.01</td>
<td>0.000</td>
<td>0.44</td>
<td>-0.02</td>
<td>0.09</td>
<td>0.05</td>
</tr>
<tr>
<td>- 10 %</td>
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<td>-2.5</td>
<td>0.05</td>
<td>0.000</td>
<td>0.27</td>
<td>-0.13</td>
<td>-0.01</td>
<td>-0.14</td>
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<td>needle + needle seat stiffness +10 %</td>
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<td>0.8</td>
<td>0.07</td>
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<td>-0.11</td>
<td>-0.79</td>
<td>-0.91</td>
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<td><strong>Hydraulic</strong></td>
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<td></td>
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<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>Z-throttle area +10 %</td>
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<td>-0.2</td>
<td>-0.014</td>
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<td>-0.12</td>
<td>-1.49</td>
<td>-3.04</td>
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<td>-0.010</td>
<td>-11.74</td>
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<tr>
<td>tube to nozzle chamber diameter +10 %</td>
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<td>0.2</td>
<td>0.07</td>
<td>-0.001</td>
<td>-1.43</td>
<td>-0.31</td>
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<td>0.001</td>
<td>0.68</td>
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<td>0.07</td>
<td>0.14</td>
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<td>nozzle chamber volume +10 %</td>
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<td>0.3</td>
<td>0.02</td>
<td>0.000</td>
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<td>-0.09</td>
<td>0.04</td>
<td>-0.04</td>
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<td>0.4</td>
<td>0.24</td>
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<td>-0.05</td>
<td>0.04</td>
<td>-0.03</td>
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<td>nozzle orifice areas +10 %</td>
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<td>-0.55</td>
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<td>0.01</td>
<td>-11.51</td>
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<td>- 10 %</td>
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<td>0.12</td>
<td>0.000</td>
<td>0.96</td>
<td>8.69</td>
<td>0.06</td>
<td>8.55</td>
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<tr>
<td>common rail volume +10 %</td>
<td>-0.3</td>
<td>-0.1</td>
<td>0.17</td>
<td>0.000</td>
<td>0.34</td>
<td>0.09</td>
<td>0.03</td>
<td>0.09</td>
</tr>
<tr>
<td>- 10 %</td>
<td>-0.5</td>
<td>1.3</td>
<td>0.17</td>
<td>0.000</td>
<td>0.36</td>
<td>-0.34</td>
<td>0.04</td>
<td>-0.16</td>
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<tr>
<td>rail-injector line length +10 %</td>
<td>-0.5</td>
<td>-2.1</td>
<td>0.18</td>
<td>0.001</td>
<td>0.87</td>
<td>0.46</td>
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<td>- 10 %</td>
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<td>6.9</td>
<td>0.22</td>
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<td>0.15</td>
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<td>0.000</td>
<td>-0.41</td>
<td>0.16</td>
<td>-0.04</td>
<td>-0.13</td>
</tr>
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<td><strong>Rail pressure</strong></td>
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<td></td>
<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>rail pressure 1000 bar -200 bar</td>
<td>-3.8</td>
<td>22.8</td>
<td>0.27</td>
<td>0.000</td>
<td>-3.26</td>
<td>-8.24</td>
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<td>-8.06</td>
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<td>rail pressure 1400 bar +200 bar</td>
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<td>-15.7</td>
<td>0.15</td>
<td>-0.001</td>
<td>1.89</td>
<td>7.78</td>
<td>-0.10</td>
<td>7.56</td>
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<td>rail pressure 1600 bar +400 bar</td>
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<td>0.30</td>
<td>-0.004</td>
<td>0.86</td>
<td>15.12</td>
<td>-0.40</td>
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<tr>
<td>rail pressure 1800 bar +600 bar</td>
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<td>-37.0</td>
<td>0.21</td>
<td>-0.008</td>
<td>-2.82</td>
<td>21.99</td>
<td>-0.83</td>
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<tr>
<td>rail pressure 2000 bar +800 bar</td>
<td>19.3</td>
<td>-44.8</td>
<td>0.18</td>
<td>-0.013</td>
<td>-9.11</td>
<td>28.46</td>
<td>-1.32</td>
<td>25.47</td>
</tr>
<tr>
<td>rail pressure 2200 bar +1000 bar</td>
<td>25.8</td>
<td>-51.9</td>
<td>0.28</td>
<td>-0.019</td>
<td>-17.97</td>
<td>35.18</td>
<td>-1.82</td>
<td>30.61</td>
</tr>
<tr>
<td>rail pressure 2400 bar +1200 bar</td>
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<td>-59.0</td>
<td>0.43</td>
<td>-0.025</td>
<td>-31.14</td>
<td>42.06</td>
<td>-2.30</td>
<td>35.70</td>
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</tbody>
</table>
7.3 Results Solenoid valve analysis: electro-mechanical

Operation of the solenoid valve is based on electrical, magnetic and mechanical effects. To reflect this the studied parameters have been clustered accordingly. Table 3 shows the obtained results.

**Electrical**

The results of the electrical analysis show a few interesting trends. All studied factors have a notable influence on full valvelift delay, effects on valve closure delay are present but generally less strong. Maximum coil current (reached during the opening transient) does not vary by more than 1 A.

In order to get a feeling for the influence of the drawdown voltage at EOAct a case was simulated without it. Results show that the valve closure delay increases by 0.14 ms (equivalent to 0.167 CAD @ 2000 RPM), nearly doubling the baseline value. For the coil current to decay to zero takes over .8 ms! longer. This indicates that the active draw-down is a highly effective performance enhancer.

Coil resistance has a relatively strong effect on opening delay and a somewhat lesser effect on closure delay. Increased coil resistance decreases coil current and with it the generated MMF which in turn limits the amount of force generated by the reluctance actuator. The combination of lower current and less MMF means the total system inductance is lower, resulting in a faster closure as can be seen in the current draw-down delay and valve closure delay. Increased resistance naturally decreases maximum coil current though the change is not linearly proportional (6% lower maximum current for 10 % higher resistance). This can be explained on the basis that maximum current is reached during the opening transient and not at steady state. Back EMF from change in magnetic flux during valve opening affects the coil voltage. The hold current, representing a steady state, does change (nearly) linearly with coil resistance.

Increasing the number of coil windings has a number of conflicting effects but results in faster valve opening and a longer closure delay. Note that in the simulation coil resistance is independent from the number of windings. Figure 45 shows coil current and valve position during the opening transient. More windings mean a stronger MMF is generated for the same coil current, hence more force is generated and the valve opens faster. Meanwhile the coil current reached during the opening transient is lower than when fewer windings are used. For a given rate of current rise the MMF rise-rate is higher and therefore more back-EMF is generated, limiting the current rise-rate achieved for a given electrical potential. Upon closure the larger inductance results in a longer delay. Interesting to note is that start of valve lift actually gets delayed (on a microsecond level) when more windings are used. This increased initial 'hesitation' can be attributed to inductance.

![Figure 45: Coil current and valve position during the opening transient for varying numbers of windings](image)

Effects of driving voltage are again similar and based on the same principles. Increased voltage decreases both SOVlift delay and full valve lift delay. Higher driving EMF overpowers the induced back EMF, so it does not show the same initial hesitation seen with increased number of windings. Contrary to the previous two cases valve closure speeds up as well, the larger reverse voltage speeds up current drawdown and with it valve closure.
Magnetic
Within the \( \pm 10\% \) range altering the permeability of the flux path, excluding the reluctance gap, has negligible effect on solenoid performance. This indicates that the flux yoke is sized with adequate safety margin such as not to affect performance.

Eddy currents have a damping effect on the magnetic response of the system, fast changes in magnetic flux create eddy currents that generate an opposing MMF. Figure 46 clearly illustrates this, when the magnetic flux increases eddy currents generate a negative MMF. Eddy current effects are implemented as a shorted winding model with a finite resistance, decreasing winding resistance increases induced ‘eddy’ current for a given \( \frac{\Delta \phi}{\Delta t} \). It can be seen that for the case with stronger eddy currents the magnetic flux lags behind during the opening transient which results in slower valve lift (not shown in figure 46). Upon closure eddy currents resist dissipation of the magnetic field, sustaining the lift force and delaying valve closure. An interesting side effect of stronger eddy currents are faster responses of coil-current to steps in driving voltage. Eddy currents limit the rate of magnetic flux change, limiting back EMF in the coil which facilitates faster current fluctuations. This is evidenced during the initial boost stage and at the voltage stepdown from boost to battery \((t = 0.1002s)\).

Note that a 90% reduction in resistance was required to obtain performance changes of the same order of magnitude as other parameters being varied by 10%. This indicates that the eddy current effect is slight or the initial resistance value was set too high to introduce significant eddy current effects.

![Figure 46: Coil current, magnetic field strength and MMF from coil and eddy current loop for baseline case and stronger eddy currents.](image)

Looking at the reluctance actuator the influence of the initial and residual reluctance gap was evaluated. Overall armature travel was kept constant by simultaneously increasing/decreasing both the initial and residual gap heights. It was found that a smaller initial + residual gap speeds up valve opening whilst delaying valve closure. This is the result of a lower overall reluctance resulting in a stronger magnetic flux, as can be seen in figure 47. More magnetic flux is generated from the start of actuation, though the difference increases after the initial transient (inductance effect). A stronger opening force results which speeds up valve lift. Valve closure is delayed since a stronger magnetic field has built up that needs to dissipate, inductance means this takes more time than when starting from a lower magnetic flux. Meanwhile the residual opening force remains larger.
Increased relative permeability and surface area of the reluctance gap have the same effect on reducing the overall reluctance of the flux path. This means a higher flux for a given MMF. Resulting performance changes are analogous to those of adding more coil windings: higher flux increase for a given MMF/coil-current increase gives faster opening and higher inductance. Higher inductance creates initial hesitance (slightly higher SOVlift delay), full valvelift is reached earlier due to the stronger reluctance force, valve closure is delayed and maximum achieved coil current decreases slightly. It must be noted that the effects are approximately half as strong as those from altering the number of windings.

**Mechanical**

Regarding mechanical influences on solenoid valve performance the effects are more straightforward. Increased valvespring stiffness delays the opening response (almost linearly) and speeds up the closing response. An interesting nuance is that the effect of decreased stiffness on valve closure seems linearly proportional whereas higher stiffness gives a slightly less strong effect. This is probably attributable to viscous damping and associated fluid displacement. A stronger spring force resulting in faster closure which increases the experience damping forces.

Note that spring preload is defined in terms of initial deformation (since deformation can realistically be measured when characterising an injector, actual preload force is hard to determine on an assembled valve), therefore increased spring stiffness also increases preload. It therefore is not surprising that increased preload deformation has a near identical effect as increased spring stiffness. Given the limited actual valve travel the higher spring rate does not amount to a significant force difference over the full valve travel.

Translating mass has an overall inertial effect, increased mass slows down the opening and closing transients without affecting the start of motion. Viscous damping has a similar effect, increasing the length of transients, though approximately a factor 10 less strong.

When control chamber pressure is larger than return line pressure a pressure force acts on the valve ball in opening direction. Both increasing the pressure surface area and pressure magnitude have an identical effect, as is to be expected. A 10% increase in hydraulic force speeds up valve opening by 5 µs and delays valve closure by 7 µs. The slightly larger effect on valve closure is due to the different opposing forces, closing spring pressure relatively smaller than the net opening solenoid force (minus spring force).
Table 3: Sensitivity study results of electro-magnetic-mechanical solenoid analysis. Presented values are differences compared to baseline case in specified units. Conditional formatting highlights the largest changes for selected characteristic values, yellow indicating a value decrease and blue a value increase.

<table>
<thead>
<tr>
<th>parameter</th>
<th>relative change</th>
<th>SOV lift delay</th>
<th>delay full valvelift</th>
<th>Delay max coil current</th>
<th>current drawdown delay</th>
<th>SOV closure delay</th>
<th>max coil current</th>
<th>hold current</th>
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</thead>
<tbody>
<tr>
<td><strong>unit</strong></td>
<td></td>
<td>µs</td>
<td>µs</td>
<td>µs</td>
<td>µs</td>
<td>A</td>
<td>A</td>
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</tr>
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<td>baseline values</td>
<td>0</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
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<tr>
<td>no active drawdown (backboost)</td>
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<td>0.0</td>
<td>0.0</td>
<td>823.2</td>
<td>139.3</td>
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<td>15.4</td>
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<td>0.9</td>
<td>0.9</td>
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<td>more coil windings</td>
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<td>-5.3</td>
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<td>6.5</td>
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<td>-10.2</td>
<td>-1.2</td>
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<td>higher voltage boost + battery</td>
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<td>-17.2</td>
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<td>25.1</td>
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<td>-1.0</td>
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<td>6.9</td>
<td>6.9</td>
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<td>-6.2</td>
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<td>8.4</td>
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<td>-0.4</td>
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<tr>
<td></td>
<td>- 10 %</td>
<td>4.1</td>
<td>4.1</td>
<td>-0.5</td>
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<td>-3.6</td>
<td>0.4</td>
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<td>4.2</td>
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<tr>
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<td>4.1</td>
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<td>0.4</td>
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<td>+10 %</td>
<td>11.6</td>
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</tr>
<tr>
<td></td>
<td>- 10 %</td>
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<td>-9.8</td>
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<td>0.3</td>
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</tr>
<tr>
<td></td>
<td>- 10 %</td>
<td>-9.8</td>
<td>-9.8</td>
<td>0.1</td>
<td>0.0</td>
<td>14.3</td>
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<td>valvebody mass</td>
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<td>0.0</td>
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<td>0.1</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>- 10 %</td>
<td>-5.2</td>
<td>-5.2</td>
<td>-3.6</td>
<td>0.0</td>
<td>-4.5</td>
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<td>viscous damping</td>
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<td>2.8</td>
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<td>5.5</td>
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<td>-5.3</td>
<td>-5.3</td>
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<td>-0.2</td>
<td>0.0</td>
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<tr>
<td></td>
<td>- 10 %</td>
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<td>0.8</td>
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<td>-6.0</td>
<td>0.2</td>
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<td>hydraulic pressure</td>
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<td>-0.2</td>
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</tr>
<tr>
<td></td>
<td>- 10 %</td>
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<td>5.8</td>
<td>0.8</td>
<td>0.0</td>
<td>-6.0</td>
<td>0.2</td>
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7.4 Results full model analysis

Completing the injection system model by the addition of the full solenoid model allows us to study solenoid effects on the injection event. Parameters of primary interest are those that significantly affect the solenoid response as identified in the previous analysis. Another parameter to be investigated is the fuel, its properties might affect the injection event. Since the physical backgrounds of all solenoid effects have been discussed with the solenoid analysis results these will not be repeated here. Rather a brief evaluation is given on how the effects manifest themselves on the injection flows and the relative importance of each parameter is discussed with regards to its effect and uncertainty. Results are presented in table 4.

Solenoid effects

The observed increase in valve closure delay from lack of an active current drawdown is reflected in the EOI delay with nearly the same value (111 and 107 µs respectively). Pilot volume increases by a factor 5, main volume increases by around 10%. Figure 48 shows rail pressure, supply voltage, nozzle flowrate and ball + needle positions for cases with and without active drawdown.

![Figure 48: Effect of reverse voltage at end of actuation (current drawdown voltage) on rail pressure, nozzle flow-rate, ballvalve lift and needle lift.](image)

The most striking difference between both cases is the effect on pilot volume, both its duration and maximum flowrate double. Larger maximum flowrate is due to higher needle lift, for a standard pilot needle lift is halted well before maximum lift is reached. Since the solenoid valve closes slower the needle has more time to open, achieving a higher maximum lift. Slower valve closure means control chamber pressure rises more gradually, resulting in further delayed and slower needle closure.

An interesting side effect of slower needle closure is the magnitude of triggered pressure waves is lower. Looking at the rail pressure trace this effect can be seen clearly at the end of the main injection. After the pilot the pressure wave is actually stronger for the no-drawdown case. This can be explained from the larger total pilot flow, more flow means the fluid has built up more momentum.

Overall it must be concluded that without the sharp solenoid valve response, accurately dosing the pilot injection is much more challenging. Thought the effects of failing to apply an active drawdown voltage are significant this should not be a problem since solenoid actuation is directly controllable and measurable.
Actuation duration has a linear effect on main volume and a larger (non-linear) effect on pilot volume. The non-linear effect on pilot volume is due to the transient character of that injection, no steady state is reached during the injection. Actuating longer results in more needle lift, higher maximum flowrate and longer closing time. Since the bulk of the main injection flow occurs during a pseudo steady state of maximum opening the longer actuation acts to linearly increase the injected volume. SOI delay decreases slightly, this is the result of the reference case used to determine that factor. SOI delay is measured at the start of the pilot injection, normally SOI falls post EOAct for the pilot. Prolonging actuation means the control chamber pressure decreases further, increasing the opening force during the initial stages.

Coil resistance delays valve opening and speeds up valve closure, this is reflected in the SOI and EOI delays. The effect on SOI delay is most prominent in line with the valve opening delay. The delay in terms of CAD is significant (0.3 CAD @ 2000 RPM for a 10% increase in resistance). Furthermore the same 10% resistance increase is enough to nearly eliminate the pilot injection. Different operating conditions can affect the resistance of a coil (especially temperature) so this parameter can have relevant practical implications.

Effects of varying power supply voltages are similar to those of coil resistance, both in magnitude and type. The power supply is however easier to monitor and directly controllable so the practical relevance is lower. It would be possible to offset identified coil resistance changes by altering the supply voltages.

Both the initial and residual reluctance gap significantly affect timings and injected volumes. Contrary to the solenoid-only analysis initial and residual gap have been studied separately, overall valve travel does not remain constant. It is felt this situation more accurately represents the effects of contamination/fouling of components. Since the valve opening area is directly related to armature travel, the maximum valve opening will differ when the residual or initial gap is changed. The results show that residual gap has a stronger effect on opening and closing delays than initial gap. Important to remember is that even when the needle does not reach full lift (during short injections) the ballvalve does fully open. Different maximum valve areas therefore strongly affect the flow balance of the control chamber and with it fluid pressure force on the needle. The initial gap does not affect the maximum outflow rate and therefore has a less strong effect. Figure 49 shows nozzle flow-rate + valve and needle positions for an increased initial and residual reluctance gap.

Figure 49: Effect of increased initial and residual reluctance gap on valve lift, needle lift and nozzle flow.
Looking at the nozzle flow-rate it can be seen that an increased initial gap advances the pilot without notably increasing its volume. This is the result of a larger maximum opening, allowing the control chamber pressure to drop faster, and faster valve closure due to lower inductance resulting from lower maximum flux (due to the larger overall reluctance). A smaller residual gap gives faster valve opening and higher maximum flux thus longer valve closure delay. This elongates the pilot and main, increasing injected volumes.

Note that 10 microns of initial/residual displacement can halve or double the pilot volume, making this a highly significant factor.

Increased spring stiffness acts to delay valve opening and speed up valve closure, this is reflected directly in the increased SOI delay and decreased EOI delay, shortening the open time. The volume decreases on the pilot and main injection are similar. With 10% increased spring-stiffness the pilot is nearly eliminated. Since spring-stiffness changes over time and with temperature this is a relevant parameter to keep track of. From the previous analysis it is known that spring preload has an identical effect on valve behaviour, the same is expected for injection behaviour.

Ballvalve seat stiffness does not significantly alter injected volumes or timing, any changes are the result of limited valve travel in tandem with different initial deformation. If more of the limited armature travel is required to overcome initial seat deformation the maximum valve opening decreases which is reflected in performance.

Fluid properties
Regarding fluid properties the two parameter that have been looked at are viscosity and temperature effects. Viscosities of different fuels can vary, form the results it seems that a 10% range does not significantly alter timings or flow-rates.

Temperature dependent fuel property changes have a slightly larger effect. Compression of the fuel in the HP pump heats it up significantly, under normal operating conditions fuel temperatures can reach 100 °C. Increasing the system liquid temperature from 60°C to 100°C changes the pilot and main volumes in the order of 1.5%. Though not that significant in and of itself it must be remembered that temperature dependent changes in flow regime are to be expected that have not been included in the current model. Furthermore electric resistance and spring stiffness show a temperature dependence which can affect results in a more significant way.
Table 4: Sensitivity study results of full system analysis. Presented values are differences compared to baseline case in specified units. Conditional formatting highlights the largest changes for selected characteristic values, yellow indicating a value decrease and blue a value increase.

<table>
<thead>
<tr>
<th>parameter</th>
<th>relative change</th>
<th>SOI delay</th>
<th>EOI delay</th>
<th>SOI delay</th>
<th>EOI delay</th>
<th>SOV lift delay</th>
<th>SOV closure delay</th>
<th>max needlelift pilot</th>
<th>Vtot pilot</th>
<th>Vtot main</th>
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<td>unit</td>
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<td>µs</td>
<td>µs</td>
<td>CAD @ 2000 RPM</td>
<td>µs</td>
<td>µs</td>
<td>mm</td>
<td>mm²</td>
<td>mm²</td>
<td></td>
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<td>baseline values</td>
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<td>0.00</td>
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<td>-0.58</td>
<td>0.002</td>
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<td>4.01</td>
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7.5 Physical interpretation

Having explored a large range of parameters, quantified their effects on injection performance and explained the mechanisms behind them, it is possible to identify some trends and highlight which parameters are relevant in the three areas of interest outlined at the start of this section. Findings, conclusions and recommendations for future work are presented in a point-wise fashion.

Injector modelling and injector identification

Through the modelling process and sensitivity study an overview has emerged of which factors warrant particular attention, especially with respect to determining model inputs from injector identification. Findings and recommendations are:

- Accurately determining the equivalent area of the pressure shoulders on needle and control plunger is important for this determines the force balance on the needle. A complicating factor is the stepped and chamfered nature of the pressure shoulders on the needle. Attention should be paid to which edge the needle actually seats on as this limits the pressure area exerting force in opening direction. Misidentification will result in timing errors (opening + closure delays).

- Areas of the A- and Z-throttle affect opening + closing delays and overall injected volumes. Ultimately the Z-throttle has a slightly larger influence than the A-throttle. Important in modelling is to consider the influence of area contraction on the overall flow resistance, not just the restriction itself. For optimal accuracy flow regimes and resistances could be studied with (axis-symmetric) CFD simulations and serve to improve model inputs.

- Combined needle and needle seat stiffness affect SOI delay and injected quantity, this becomes more relevant for short injections. In some previous works combined needle + seat stiffness has been tuned to match SOI delay to experimental data. Based on the relatively limited influence of this parameter on SOI delay compared to others I would advise against this. Ideally this parameter would be a measured model input. This would be possible if the test injector is equipped with an absolute needle position sensor. When the system is pressurised the needle displaces relative to its unpressurised state. If the pressure areas are known the nett change in closing force is known and combined with measured displacement the combined stiffness can be determined. Alternatively bench tests could be performed if the injector can be disassembled. Fixing the lower injector body and using a press to load the needle the same displacement V force curve can be obtained.

- From the sensitivity study one gets a general sense that the solenoid valve has a more delicate balance than the needle. A number of parameters can significantly alter the armature response whereas the needle displacement is dominated by hydraulic forces. When varied within a plus and minus 10 % range the following solenoid parameters can almost double or reduce to zero the pilot volume: initial and residual solenoid gap, coil resistance and valve spring stiffness.

- The solenoid-valve spring has a much greater influence on timing and injected quantities than the needle spring. This is a direct result of the more delicate solenoid valve balance, the needle spring being heavily overpowered by hydraulic forces where the magnetic solenoid force is less prominent. From a mechanical point of view this should come as no surprise since the hydraulic forces can actively open and close the needle whereas the reluctance force cannot be reversed. Valve closure is entirely driven by its spring, closure can only be sped up by increasing its preload/stiffness. During injector identification it is important to accurately determine valve-spring preload, more so than spring stiffness. Given the small displacements preload is responsible for the bulk of spring force.

- For a representative solenoid closure response the active drawdown (reverse voltage applied at the end of actuation) must be included in the model.

- Regarding the magnetic side of the solenoid model point of primary importance are the number of coil windings and the initial + residual reluctance gap. Determining the number of coil windings without disassembling an injector might be tricky, perhaps this data could be obtained from the manufacturer. Otherwise x-ray scanning might be a possibility. Failing to correctly identify the number of windings will affect the valve response rate (opening and closing delay) and alter the voltage-current response. Comparing the measured and modelled solenoid voltage-current response
will show. Remember that magnetic saturation and eddy currents significantly affect inductance, this will also affect the voltage-current response and should not be confused with other factors.

- Initial and residual reluctance gap have a strong effect on opening + closing speed and injected quantities. For fast acting injectors total valve travel is in the order of tenths of mm and the residual gap could be in the order of microns. Particular attention should be paid during injector identification to accurately determining the initial gap and overall armature travel.

- Reluctance of the flux yoke is not of primary significance, overall reluctance remains dominated by the airgap even at full valve closure (factor 4000 difference in relative permeabilities of the yoke V gap). Factors to pay attention to are magnetic saturation and eddy currents in the yoke + armature. Performing FEM simulations of the magnetic system can help improve understanding.

Potential operating and lifetime effects

For different operating conditions the injection response may vary. Furthermore wear can alter performance over the lifetime of an injector. The following effects could occur:

- Copper has a positive thermal coefficient of resistance (TCR), the TCR value around room temperature for pure copper is approximately 0.39e-2 [°C]. This means that coil resistance increases by over 10% when the coils’ temperature increases by 30 °C. Given the large amount of current passed through the coil such temperature increase values are not unrealistic. If the injector driver is not able to follow the target current trajectory at a higher resistance injection delay will increase and short injections. Short injections are affected most severely.

- Spring stiffness tends to decline with temperature, over the lifetime of an injector the spring stiffness can also decay. Both the solenoid valve spring and needle spring are exposed to heated fuel (fuel heats up when it is compressed to injection pressure), they therefore heat up during operation. The solenoid should not be overlooked when trying to explain observed changes in injection performance, decreased valve spring stiffness elongates injections.

- Contamination can clog fuel passages and increase drag on components. Keep in mind that increased frictional damping on needle or armature travel has the same effects on performance as decreased spring stiffness. Internal scratching from debris in fuel can destroy injectors, fuel filtration is of critical importance.

Experimental setup design and operation

Regarding the construction and operation of a standalone experimental injector test setup the following aspects should be considered:

- Relative pump-injection timing does not significantly affect injection behaviour. Pump capacity must be adequate to restore rail pressure between injections. In case of insufficient single stroke pump capacity one could use multiple pump strokes between, in the simulations this does not affect injection performance.

- Include a drawdown voltage at the end of solenoid actuation, otherwise performance cannot be representative.

- Coil resistance has a temperature dependency, variations are to be expected over the duration of an experiment. It is advisable to monitor coil resistance and possibly account for changes in coil resistance by altering the supply voltage accordingly.

- Fuel rail volume is best kept the same as is used on the reference engine. Despite the rail normally feeding multiple injectors, injection events are isolated in time (duration much shorter than intervals between injections). Some deviation in rail volume should not affect results.

- Fuel line length and stiffness are best kept identical to the reference engine, line length can affect pressure pulse timing (simulations foresee no major effects). Line stiffness + length and rail volume + stiffness affect the elastic energy storage. Less energy stored leads to larger pressure variations over an injection, something to keep in mind when designing a rail-pressure controller. Faster system reactions require a higher controller bandwidth.
To obtain the same injected volumes under various experimental conditions (rail pressure, temperature) different actuation durations are required, this is best mapped experimentally. Especially pilot volume scales non-linearly with actuation duration.

Diagnostic possibility: armature displacement can be inferred from coil current measurements. During the boost phase current first rises but subsequently peaks and drops. As the reluctance gap decreases the magnetic flux can increase. Coil inductance means a back emf is generated opposing this positive change in flux, resulting in a lower current. Therefore a drop in current signifies armature lift, peak current corresponds to initial armature travel. Upon full armature lift the reluctance gradient returns to zero and the magnetic flux stabilises, induced EMF disappears and current rises again. The point where coil current starts to increase post the initial dip therefore indicates the point of full armature lift. Figure 45 indicates the expected current V displacement characteristic, note the points of initial valve opening and maximum valve opening and their corresponding points on the current characteristic. Coil current can easily be measured (at high frequency) whereas valve travel cannot.

Importance of operating temperature: continued operation gives heatsoaking from high temperature compressed fuel. Heat affects spring stiffness and coil properties + fluid properties. Fluid temperature can affect EOI timing and increase flowrate at max needle lift. Ways to deal with this experimentally: start performing experiments after the injector had time to warm up to equilibrium temperature. Monitor body temperature of injector. Keep track of voltage-current traces of the coil, changes in profile (initial transient and maximum reached current during opening) can point to changing resistance properties.
8 Conclusion and recommendations

From the performed literature study, modelling and analysis work a number of conclusions can be drawn regarding the two main topics of this work: designing (conceptually) an experimental injector characterisation setup and analysing (based on models) which factors affect transient injection performance. Where relevant some recommendations on setup design, model development and injection performance analysis have been added. To help structure the findings the supportive research questions have been repeated here, note that the conclusions are not written in a direct question-answer style.

Experimental setup

**Which experimental technique is best suited to measure mass flux, momentum flux and nozzle coefficients?**

**Can mass and momentum flux be adequately measured with the same technique? How can the desired quantities be determined from measured data?**

From analysing a large range of experimental injector flow characterisation techniques found in literature, jet impingement force measurement was found to be the most versatile and best suited to studying short and close-spaced-multiple injections. With the jet-impingement force measurement, time-resolved momentum flux is measured directly, it therefore provides excellent information on rate shape. Mass flux is linked to the square root of momentum flux. Through a single corrective scaling based on an overall injected mass measurement, the time-resolved mass flow-rate can be determined from measured momentum flux. All nozzle coefficients $C_a$, $C_m$, $C_d$ and $C_v$ can be determined from the momentum flux measurement, again relying on a scaling with an overall discharged mass measurement. To determine overall discharged mass the injected fuel from a known series of injection events has to be collected and weighed. The only other required information for determining the nozzle coefficients is liquid density.

A distinct advantage of the jet-impingement technique is that fuel is injected into air rather than liquid, avoiding potential systematic error. The traditional rate of injection measurement techniques (Bosch long-line and Zeuch chamber) are based on injecting into liquid. Though they can provide a more direct mass flux measurement they are not well suited for characterising close-spaced multiple injections. Under highly dynamic injection conditions these methods suffer from pressure wave interference, the impingement method does not suffer from this problem and can accurately characterise highly dynamic injections. Thus, having a separate injection mass flow rate measuring setup is not beneficial for studying short, close-spaced injections.

**What sort of setup is required, which components could be used and which factors are to be taken into account in the setup design?**

A standalone injector characterisation setup based on jet impingement force measurement requires the following functionalities:

- A high pressure fuel supply consisting of tank, feed pump, filtration, high pressure pump, fuel rail and fuel lines.
- A force sensor equipped with hard wearing impingement target, mounted on a positioning device.
- Some form of enclosed chamber into which the impingement force sensor is mounted, this chamber serves to contain and collect all injected fuel as well as ensure operator safety.
- Driving electronics for the solenoid injector and fuel system.
- Sensing + data acquisition equipment for impingement force, fuel pressure, fuel/injector temperature and possibly coil current.

Critical aspects to the design are:

- Fuel filtration and water separation, especially when using a fuel-lubricated injection pump, water in the fuel can cause high pressure pump failure and debris can wear or clog the internals of an injector.
- Particular attention should be paid to shielding the impingement force sensor from mechanical shocks. Its mounting should be mechanically insulated from the injector mounting frame, otherwise mechanical shocks caused by needle impact will cause noise in the measured force signal.
• Impingement distance should be kept minimal (approximate maximum of 2 mm) and the impingement target no larger than what is required to ensure full radial deflection of the fuel jet. What target size this requires differs per jet and has to be determined experimentally. When using an oversized target or excessive nozzle-target distance a systematic error is introduced in the form of a clear non-physical dip-peak combination at the opening momentum transient. This is caused by mass accumulation at the spray tip before first target impact.

• Impingement force can be measured with a piezo-electric force transducer or alternatively with a piezo-resistive pressure sensor to which the impingement target is glued. When using a pressure sensor, the target base diameter should be smaller than the pressure sensing membrane diameter, this improves the signal to noise ratio.

Are there differences between the engine installed injection system and the research setup? How could the setup alter injection performance?

The new injector characterising setup should be useable for a variety of heavy duty injectors from various manufactures and applied on a range of engines. It is therefore not possible to replicate the exact fuel system dynamics found on each reference engine. From the performed analysis the following conclusions can be drawn regarding system sizing and evaluation strategies:

• The first short injection from a fuel system at pseudo steady state is not affected by fuel system sizing and offers a good baseline injector response for model validation. Subsequent short injections are affected by pressure waves in the injector feed line. Actuating at a different point in the pressure pulse cycle can halve the achieved maximum needle lift for the same actuation duration. Experimentally the magnitude of these effects can be ascertained by altering injection spacing, actuating at different stages of the pressure pulse cycle.

• For determining nozzle coefficients fuel system sizing is irrelevant (within the typical heavy duty range). A varying rail pressure drop is accounted for in the definition of the nozzle coefficient. Pressure pulse effects on nozzle flow can be averaged out of the signal. Make sure to use an evaluation interval length equal to a whole number of injection line pulse cycle times, this can be done in post processing. The expected difference in computed discharge coefficient is below 0.1 % when halving rail volume from 40 to 20 cm$^3$ and injector line length from 40 to 20 cm.

• Pump timing is not a concern for the experimental characterisation. Pump strokes should be separated from the injection events, starting each injection from a steady rail pressure at the desired setpoint.

• Between long injections multiple pump strokes can be used to restore rail pressure. This means a smaller light duty sized pump can be used. Of the currently produced externally driven high pressure injection pumps the Bosch CP-4 pump offers the highest injection pressure rating at 2700 bar for some variants.

• I would suggest using a rail volume of around 40 cm$^3$, in line with a typical heavy duty average.

• I would suggest using a short injector feed line of around 20 cm, faster pulse dynamics mean they are more clearly identifiable in shorter injections. Though this parameter can be altered to suit practical requirements.

What level of accuracy is expected determining mass flux, momentum flux and nozzle coefficients? Which factors determine the achieved level of accuracy?

The accuracy with which momentum flux can be determined depends on the accuracy of the force sensor and exactness of the radial impingement. Sensor errors linearly affect the momentum flux magnitude error. Provided the sensor is linear, the rate-shape (both mass and momentum) should be accurate. Collection and weighing errors in the overall injected mass measurement linearly affect computed mass flux and discharge coefficient. To limit these errors the fuel from a larger set of injections can be collected, a downside to this approach is that the integrated forces signal error also increases. If the impinged jet has a residual forwards velocity the measured force signal is not equal to the original jet momentum. The corresponding error scales with the ratio of the squares of residual to initial forwards velocity.
Model analysis

How can the hydraulic and mechanical aspects of the fuel injection system and a solenoid actuator be modelled? What are suitable performance indicators for high pressure injection systems?

A lumped-parameter injection system model was developed in Simscape, the physical modelling environment of MATLAB Simulink. The hydraulic model consists of local resistances and volumes, longer fuel lines (pump-rail, rail-injector and injector inlet-nozzle chamber) are modelled in a 1-D fashion to include pressure pulse dynamics in the system. The mechanical model consists of a simple mass-spring-damper system with hydraulic-mechanical converters forming the interface between the hydraulic and mechanical model. Needle and needle-seat stiffness has been lumped into a single value. The solenoid valve has been explicitly modelled by means of a lumped-parameter electro-magnetic-mechanical model, consisting of a solenoid coil with separate resistance, magnetic flux path reluctances and variable reluctance actuator model. Eddy current effects are included as a shorted winding model. Physical limitations meant that parameter values could not be measured on a real injector. As a workaround baseline values for the hydraulic-mechanical model were taken form previous works. The solenoid was tuned to obtain the coil-current response specified for the injectors on the research engine. The developed model was verified and to the best possible extent validated based on experimental results presented in literature. It was shown that the developed model is able to accurately predict the injection rate profile, rail pressure profile, ball-valve position profile and needle position profile under transient and steady state conditions.

Based on the developed model an extensive parameter sensitivity analysis was performed, mapping how each parameter affects critical aspects of the injection. This was done for a single pilot + main injection (baseline 2.5 mg pilot and 110 mg main). Influences on start of injection delay, end of injection delay, total pilot volume, total main volume and maximum pilot flow rate were investigated. The pilot flowrate, volume and injection delay are key indicators for short-injection performance.

How does the solenoid valve affect injection performance? Which mechanical aspects of the injector affect injection performance?

Short injections are affected strongest by the solenoid valve. In general the force balance of the solenoid is more delicate than that of the control plunger and needle. A number of parameters can significantly affect the solenoid response whilst the needle lift response is dominated by hydraulic forces. When varied within a plus and minus 10 % range a number of solenoid parameters can almost double or reduce to zero the pilot volume. Most prominent are: initial and residual solenoid gap, coil resistance and valve spring stiffness. Valve spring stiffness and preload have much more influence on the overall injection than those of the needle spring. Valve closure time can be more than halved when a reverse voltage is applied at the end of actuation to actively draw down the coil current to zero.

Other parameters that significantly affect short injections are the A and Z-throttle diameters, the ratio between these determines the rate of needle lift through how fast the control chamber pressure drops. Needle and needle seat stiffness determine the initial deformation which has to be overcome before needle lift results in nozzle flow, though this influence is around four times less strong than the solenoid influence.

Overall it can be concluded that more parameters can significantly affect short injections than long injections, small changes in delays and flow rate more easily amount to tens of percent variations in total outflow. Particular attention should be paid to performance of the solenoid valve when working with short injections on solenoid-based injection systems.

An interesting observation is that solenoid valve opening can be inferred from the coil-current response. During first valve actuation a drop in current can be observed (assuming drive voltage remains at a constant level) when the valve starts opening. This is due to (increased) back EMF caused by an increase in coil flux associated with the decrease in gap reluctance resulting from armature travel. When the valve is fully opened the coil current starts to increase again. Accurate coil current measurements with a high temporal resolution can therefore be used to identify solenoid valve timing as reference for model validation. A useful workaround since solenoid valves are not typically equipped with position sensors.
References


A  Simscape model parameters

Figure 50: Overview of developed Simscape injector model with full showing full nozzle construction
<table>
<thead>
<tr>
<th>Component</th>
<th>property</th>
<th>value</th>
<th>units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Z throttle</td>
<td>internal diameter</td>
<td>250</td>
<td>um</td>
</tr>
<tr>
<td></td>
<td>length</td>
<td>100</td>
<td>um</td>
</tr>
<tr>
<td>pressure shoulder control plunger</td>
<td>area</td>
<td>15.2</td>
<td>mm²</td>
</tr>
<tr>
<td>hydraulic-mechanical converter</td>
<td>initial position</td>
<td>0.43</td>
<td>mm</td>
</tr>
<tr>
<td></td>
<td>dead volume</td>
<td>4.4</td>
<td>mm³</td>
</tr>
<tr>
<td>pressure shoulder needle</td>
<td>area</td>
<td>12</td>
<td>mm²</td>
</tr>
<tr>
<td></td>
<td>initial position</td>
<td>1</td>
<td>mm</td>
</tr>
<tr>
<td></td>
<td>dead volume</td>
<td>5</td>
<td>mm³</td>
</tr>
<tr>
<td>needle valve</td>
<td>orifice diameter</td>
<td>2</td>
<td>mm</td>
</tr>
<tr>
<td></td>
<td>cone angle</td>
<td>90</td>
<td>deg</td>
</tr>
<tr>
<td></td>
<td>discharge coefficient</td>
<td>0.65</td>
<td></td>
</tr>
<tr>
<td>tube to nozzle chamber</td>
<td>length</td>
<td>131</td>
<td>mm</td>
</tr>
<tr>
<td></td>
<td>diameter</td>
<td>2.83</td>
<td>mm</td>
</tr>
<tr>
<td></td>
<td>stiffness</td>
<td>0.1</td>
<td>um/bar</td>
</tr>
<tr>
<td></td>
<td>internal surface roughness height</td>
<td>8</td>
<td>um</td>
</tr>
<tr>
<td>nozzle chamber</td>
<td>volume</td>
<td>6</td>
<td>mm³</td>
</tr>
<tr>
<td>A throttle ball valve</td>
<td>ball diameter</td>
<td>1.35</td>
<td>mm</td>
</tr>
<tr>
<td>(sharp edged seat)</td>
<td>orifice diameter</td>
<td>400</td>
<td>um</td>
</tr>
<tr>
<td></td>
<td>discharge coefficient</td>
<td>0.75</td>
<td></td>
</tr>
<tr>
<td>A throttle block by needle lift (ball valve)</td>
<td>ball diameter</td>
<td>2</td>
<td>mm</td>
</tr>
<tr>
<td>(sharp edged seat)</td>
<td>orifice diameter</td>
<td>1</td>
<td>mm</td>
</tr>
<tr>
<td></td>
<td>displacement offset</td>
<td>0.4</td>
<td>mm</td>
</tr>
<tr>
<td>control chamber</td>
<td>piston area</td>
<td>15.2</td>
<td>mm</td>
</tr>
<tr>
<td></td>
<td>dead volume</td>
<td>4.42</td>
<td>mm³</td>
</tr>
<tr>
<td></td>
<td>initial position</td>
<td>0.43</td>
<td>mm</td>
</tr>
<tr>
<td>area change inlet to Z throttle 4 mm-0.25 mm</td>
<td>cone angle</td>
<td>40</td>
<td>deg</td>
</tr>
<tr>
<td>nozzle orifice</td>
<td>tube diameter</td>
<td>195</td>
<td>μm</td>
</tr>
<tr>
<td>nozzle consists of 7 orifices in parallel</td>
<td>tube length</td>
<td>1</td>
<td>mm</td>
</tr>
<tr>
<td></td>
<td>internal surface roughness height</td>
<td>10</td>
<td>μm</td>
</tr>
<tr>
<td>nozzle hole entrance</td>
<td>large diameter</td>
<td>1000</td>
<td>μm</td>
</tr>
<tr>
<td>semi-empirical correlation + correction coefficient</td>
<td>small diameter</td>
<td>195</td>
<td>μm</td>
</tr>
<tr>
<td>discharge tuned to 1.7 l/m 100 bar pressure drop</td>
<td>correction coefficient</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>return line</td>
<td>internal diameter</td>
<td>2</td>
<td>mm</td>
</tr>
<tr>
<td></td>
<td>line length</td>
<td>20</td>
<td>cm</td>
</tr>
<tr>
<td></td>
<td>return back pressure</td>
<td>20</td>
<td>bar</td>
</tr>
<tr>
<td>solenoid valve</td>
<td>mass</td>
<td>6</td>
<td>g</td>
</tr>
<tr>
<td></td>
<td>translational damping</td>
<td>10</td>
<td>N/(m/s)</td>
</tr>
<tr>
<td></td>
<td>actuation force</td>
<td>160</td>
<td>N</td>
</tr>
<tr>
<td></td>
<td>endstop stiffness</td>
<td>4000</td>
<td>N/mm</td>
</tr>
<tr>
<td>solenoid valve spring</td>
<td>stiffness</td>
<td>23.3</td>
<td>N/mm</td>
</tr>
<tr>
<td></td>
<td>preload</td>
<td>140</td>
<td>N</td>
</tr>
<tr>
<td>control plunger + needle</td>
<td>mass</td>
<td>11.8</td>
<td>g</td>
</tr>
<tr>
<td>needle spring</td>
<td>stiffness</td>
<td>16.3</td>
<td>N/mm</td>
</tr>
<tr>
<td></td>
<td>preload</td>
<td>25</td>
<td>N</td>
</tr>
<tr>
<td>needle / needle seat contact</td>
<td>stiffness</td>
<td>4000</td>
<td>N/mm</td>
</tr>
<tr>
<td>stiffness &amp; damping applied through transition region</td>
<td>damping</td>
<td>150</td>
<td>N/(m/s)</td>
</tr>
<tr>
<td></td>
<td>transition region</td>
<td>0.01</td>
<td>mm</td>
</tr>
</tbody>
</table>

Table 5: Simscape solenoid-injector model, parameter values
Figure 51: Overview of developed plunger pump model with PCV, model represents one plunger element with its own inlet and outlet valves.

<table>
<thead>
<tr>
<th>Component</th>
<th>property</th>
<th>dimensions</th>
<th>units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet feed</td>
<td>pressure</td>
<td>10</td>
<td>bar</td>
</tr>
<tr>
<td>Inflow restriction</td>
<td>area</td>
<td>2</td>
<td>mm$^2$</td>
</tr>
<tr>
<td></td>
<td>discharge coefficient</td>
<td>0,7</td>
<td></td>
</tr>
<tr>
<td>Inlet check valve</td>
<td>maximum passage area</td>
<td>6</td>
<td>mm$^2$</td>
</tr>
<tr>
<td>include opening dynamics</td>
<td>cracking pressure</td>
<td>0,5</td>
<td>bar</td>
</tr>
<tr>
<td></td>
<td>maximum opening pressure</td>
<td>2</td>
<td>bar</td>
</tr>
<tr>
<td></td>
<td>discharge coefficient</td>
<td>0,7</td>
<td></td>
</tr>
<tr>
<td></td>
<td>opening time constant</td>
<td>0,015</td>
<td>ms</td>
</tr>
<tr>
<td>Outlet check valve</td>
<td>maximum passage area</td>
<td>6</td>
<td>mm$^2$</td>
</tr>
<tr>
<td>include opening dynamics</td>
<td>cracking pressure</td>
<td>0,5</td>
<td>bar</td>
</tr>
<tr>
<td></td>
<td>maximum opening pressure</td>
<td>2</td>
<td>bar</td>
</tr>
<tr>
<td></td>
<td>discharge coefficient</td>
<td>0,7</td>
<td></td>
</tr>
<tr>
<td></td>
<td>opening time constant</td>
<td>0,01</td>
<td>ms</td>
</tr>
<tr>
<td>Pump control valve</td>
<td>discharge coefficient</td>
<td>0,7</td>
<td>mm$^2$</td>
</tr>
<tr>
<td></td>
<td>max opening</td>
<td>2</td>
<td>mm</td>
</tr>
<tr>
<td>Plunger element</td>
<td>diameter</td>
<td>conf</td>
<td>mm</td>
</tr>
<tr>
<td></td>
<td>dead volume (height)</td>
<td>1</td>
<td>mm$^2$</td>
</tr>
<tr>
<td>Pump outlet chamber</td>
<td>volume</td>
<td>1</td>
<td>cm$^3$</td>
</tr>
<tr>
<td>Cam profile (type = eccentric shaft)</td>
<td>cam circle diameter</td>
<td>20</td>
<td>mm</td>
</tr>
<tr>
<td>3 pump cycles per 2 engine revolutions</td>
<td>stroke/offset</td>
<td>conf</td>
<td>mm</td>
</tr>
</tbody>
</table>

Table 6: Simscape unit pump model, parameter values. Values are based on the single cylinder heavy duty research engine at TU/e, three cam lobes per cam, rotating at half engine speed. Plunger diameter and stroke are deemed confidential but known to the author.
<table>
<thead>
<tr>
<th>Component</th>
<th>property</th>
<th>dimensions</th>
<th>units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel rail</td>
<td>volume</td>
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<td>cm$^3$</td>
</tr>
<tr>
<td>Fuel line pump-rail</td>
<td>length</td>
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<td>cm</td>
</tr>
<tr>
<td></td>
<td>internal diameter</td>
<td>3</td>
<td>mm</td>
</tr>
<tr>
<td></td>
<td>wall stiffness</td>
<td>0.1</td>
<td>mm/1000 bar</td>
</tr>
<tr>
<td></td>
<td>internal surface roughness</td>
<td>4</td>
<td>um</td>
</tr>
<tr>
<td></td>
<td>rail side restriction diameter</td>
<td>2</td>
<td>mm</td>
</tr>
<tr>
<td></td>
<td>rail side restriction length</td>
<td>2</td>
<td>mm</td>
</tr>
<tr>
<td>Injector feed line</td>
<td>length</td>
<td>20</td>
<td>cm</td>
</tr>
<tr>
<td></td>
<td>internal diameter</td>
<td>4</td>
<td>mm</td>
</tr>
<tr>
<td></td>
<td>wall stiffness</td>
<td>0.1</td>
<td>mm/1000 bar</td>
</tr>
<tr>
<td></td>
<td>internal surface roughness</td>
<td>4</td>
<td>um</td>
</tr>
<tr>
<td></td>
<td>rail side restriction diameter</td>
<td>2</td>
<td>mm</td>
</tr>
<tr>
<td></td>
<td>rail side restriction length</td>
<td>2</td>
<td>mm</td>
</tr>
<tr>
<td>Fluid properties</td>
<td>type</td>
<td>diesel fuel</td>
<td></td>
</tr>
<tr>
<td></td>
<td>temperature</td>
<td>60</td>
<td>°C</td>
</tr>
<tr>
<td></td>
<td>density</td>
<td>795.9</td>
<td>kg/m$^3$</td>
</tr>
<tr>
<td></td>
<td>viscosity</td>
<td>1.36</td>
<td>cSt</td>
</tr>
<tr>
<td></td>
<td>bulk modulus</td>
<td>1.53e9</td>
<td>Pa</td>
</tr>
<tr>
<td></td>
<td>relative amount of trapped air</td>
<td>0.01</td>
<td>%</td>
</tr>
</tbody>
</table>

Table 7: Additional Simscape injection model parameters
### B Solenoid model parameters

Table 8: Simscape solenoid valve model, baseline parameter values. Values are not derived from one single model of injector, rather values fall within a realistic range found in literature. Last column indicates the bases on which each parameter was selected.

<table>
<thead>
<tr>
<th>Component</th>
<th>property</th>
<th>dimensions</th>
<th>units</th>
<th>Value based on</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power supply</td>
<td>Boost</td>
<td>52.4</td>
<td>V</td>
<td>Specs research engine</td>
</tr>
<tr>
<td>active drawdown</td>
<td>Drawdown boost</td>
<td>-52.4</td>
<td>V</td>
<td>Specs research engine</td>
</tr>
<tr>
<td></td>
<td>Battery</td>
<td>24</td>
<td>V</td>
<td></td>
</tr>
<tr>
<td></td>
<td>boost time</td>
<td>200</td>
<td>µs</td>
<td>tuned for 110 mm³ delivery</td>
</tr>
<tr>
<td></td>
<td>total actuation time (incl boost)</td>
<td>2000</td>
<td>µs</td>
<td></td>
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C Sensitivity study

Pump timing on test engine

Figure 52: Nozzle flowrates and railpressure data for: rail pressure set at constant value, simultaneous pump stroke and injection, pump stroke and injection 120 CAD out of phase
Declaration concerning the TU/e Code of Scientific Conduct

I have read the TU/e Code of Scientific Conduct¹.

In carrying out research, design and educational activities, I shall observe the five central values of scientific integrity, namely: trustworthiness, intellectual honesty, openness, independence and societal responsibility, as well as the norms and principles which follow from them.

Date
08-09-2020

Name
J.C. Vermeiden

ID-number
0955618

Signature

Submit the signed declaration to the student administration of your department.

¹ See: https://www.tue.nl/en/our-university/about-the-university/organization/integrity/scientific-integrity/
The Netherlands Code of Conduct for Scientific Integrity, endorsed by 6 umbrella organizations, including the VSNU, can be found here also. More information about scientific integrity is published on the websites of TU/e and VSNU