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Comparing the performance of the common rail fuel injection system with the traditional injection system using computer aided modelling and simulation

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SYNOPSIS

In this paper, models of a traditional cam-driven plunger diesel injection system and a common rail injection system are presented. The models are developed using bond graphs, a modelling tool that for many years has shown its excellence in modelling systems consisting of sub-models from several energy domains in a unified approach. Through simulations, the performance of the common rail fuel injection system and the traditional injection system will be analysed and discussed. Of particular interest is the pressure differential over the nozzle bores.

INTRODUCTION

Traditional fuel injection systems for diesel engines are designed with the objective to secure acceptable fuel spray characteristics during the combustion process at all load conditions. Incorrect injection causes reduced efficiency and increased emission of harmful species. In the later years, the common rail injection system with electronic controls has been promoted as the future standard in fuel injection systems for diesel engines. Among the advantages claimed with respect to the common rail concept are injection rate shaping, variable timing and duration of the injection, in addition to variable injection pressure, enabling high injection pressure even at low engine loads.

Medium speed diesel engines are different from the automotive diesel engines, especially in that the majority of them operate at constant load and speed most of the time, and the advantages of the more complicated common rail system may not be justified. The common rail injection system is not capable of supplying all possible rate shapes, and rate shaping is mostly restricted to delivering a pre injection prior to the main injection. The major benefits from the common rail injection system on medium speed diesel engines are thus the individual adjustment of the timing and duration of the injection for each cylinder.

The rate of energy conversion in the cylinder, commonly called the rate of heat release, is closely related to the rate of injection. When the rate of injection is the key to an effective combustion process, it is vital to determine how the rate of injection from the common rail system compares to the rate of injection from a traditional injection system. The characteristics of these injection systems will be discussed in this paper, using computer aided modelling and simulation.

Today's fuel injection systems consist of components from several energy domains working together in a highly dynamic system. The bond graph modelling approach[1] has for many years shown its excellence in representing multi-domain systems. By using bond graphs and matching software for modelling and simulation, it will be shown how this can be a powerful tool for engineers when evaluating the performance of physical systems. Converting a bond graph model into a set of first order differential equations is a straightforward systematic process that can easily be computerised. The models presented in this paper are developed in the modelling tool MS1[2], generating simulation code for ACSL[3] (Advanced Continuous Simulation Language).

Author's Biography
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P.M. Einang is MSc (1979) from the Department of Mechanical Engineering, The Norwegian Institute of Technology. He holds the position as Research Manager (internal combustion engines), and has been involved in research and development of diesel and gas engines since 1981. He is also responsible for the Machinery Laboratory comprising a number of medium speed and high speed engines as well as different constant volume combustion rigs for studying fuel injection and combustion.

FUEL INJECTION AND FUEL INJECTION SYSTEMS

The diesel fuel injection system consists of an injection pump, delivery pipes and fuel injector nozzles. Several different types of injection pumps and nozzles are used. In this paper only the traditional cam driven mechanical injection system and the common rail system is discussed.

The task of the fuel injection system is to meter the appropriate quantity of fuel for the given engine speed and load to each cylinder, each cycle. Further, the injection system should inject that fuel at the appropriate time in the cycle at the desired rate with the spray configuration required for the particular combustion chamber employed. It is important that the injection begins and ends cleanly, avoiding any secondary injections. The fuel is introduced to into the cylinder of a diesel engine trough a nozzle with a large pressure differential across the nozzle orifice. This large pressure differential is required so that the injected liquid fuel jet will enter the chamber at sufficiently high velocity to atomise into small-sized droplets to enable rapid evaporation. The plume should traverse the combustion chamber in the time available and fully utilise the air charge. Thus, the fuel injection process controls only four basic functions:

1. *Timing and duration of injection*
2. *Injection rate shape*
3. *Injection quantity*
4. *Spray quality*

Traditional injection system

An auxiliary cam on the engine camshaft drives a single-cylinder injection pump. Early in the stroke of the plunger, the inlet port is closed and the fuel trapped above the plunger is forced through a check valve into the injection line. The injection nozzle has several holes through which the fuel sprays into the cylinder. A spring-loaded injection needle keeps the injection valve closed until the pressure in the injector volume, acting on parts of the needle surface, overcomes the spring force and opens the valve. Thus, the phase of the pump camshaft relative to the engine crankshaft controls the start of injection, while the force given by the initial displacement of the spring gives the opening pressure. Injection is stopped when the inlet port of the pump is uncovered by a helical groove in the pump plunger, and the high pressure above the plunger and in the injector volume is released. The injection pump cam design and the position of the helical groove determine the amount of fuel injected into the cylinder. Thus for a given cam design, the rotating plunger and its helical groove controls the load.

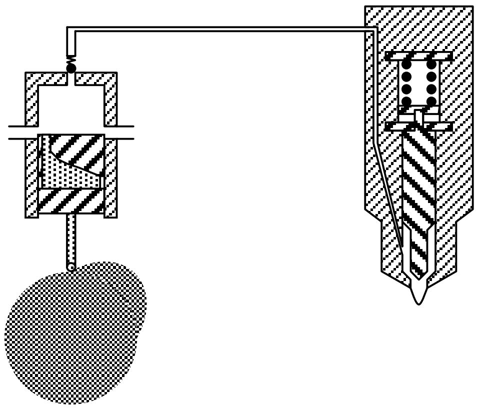


Fig. 1 Traditional cam-driven fuel injection system

Common rail injection system

Contrary to the traditional injection system, pressure generation and injection are decoupled in the common rail system. The injection pressure can even be generated independently of engine revs and the injected fuel quantity can be freely selected within limits. A prerequisite for this decoupling of pressure generation and injection in common rail systems is the high-pressure accumulator, which consists of the rail and the high pressure fuel lines to the nozzles. The key component of the system is the solenoid valve controlled injector. In order to control the opening and closing time of the needle, a small chamber of pressurised fuel is present at the top of the needle. This volume is connected to the rail through a small orifice that assures that same pressure between the nozzle and the chamber when the valve is closed. At the same time, a small solenoid balance valve is in operation in this area that can open at an accurately specified time, thus creating a pressure drop. This will result in a negative net force on the valve needle, and the injection is initiated. As soon as the solenoid closes, the pressure in this chamber will increase again resulting in the closure of the needle. Shaping the injection rate, obtaining pilot injection and multiple injections are done by controlling the nozzle needle movement.

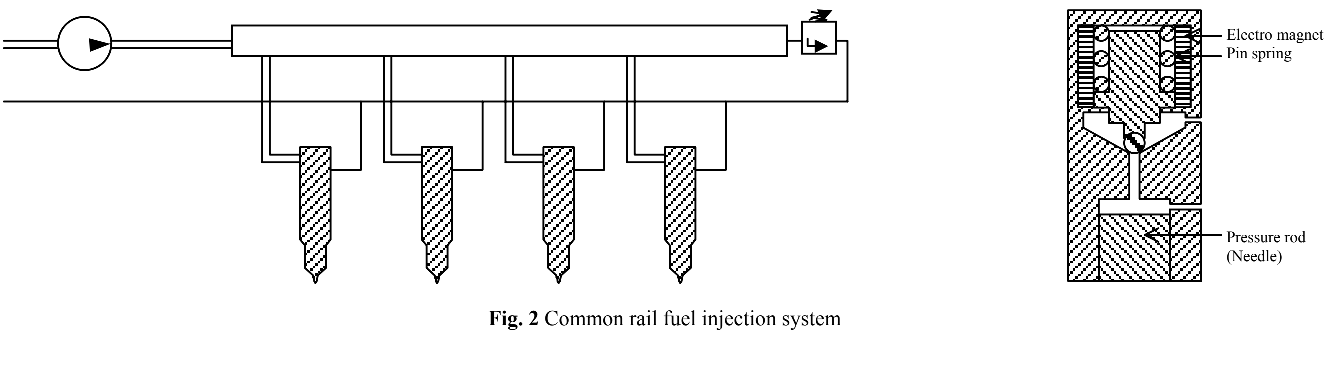


Fig. 2 Common rail fuel injection system

Fuel injection and combustion

The direct ignition combustion process can be divided into four phases:

1. *Ignition delay: The time between start of injection and start of combustion*
2. *Premixed combustion phase: Combustion of the fuel that has mixed with air within the flammability limits during the ignition delay*
3. *Mixing controlled combustion phase: Combustion of fuel at a rate which mixture becomes available for burning*
4. *The late combustion phase: Combustion of the final amounts of fuel, soot and other fuel rich areas*

It is desirable to have the shortest possible ignition delay and to minimise the amount of fuel injected during this time, and thereby limit the rapid energy release in the premixed combustion phase. Obviously, it is also important to limit the energy release in the late combustion phase, meaning that as much as possible of the energy release should be mixing controlled. Several processes controls the mixing controlled combustion phase:

- *Liquid fuel atomisation*
- *Vaporisation*
- *Mixing of fuel vapour and air*
- *Pre-flame chemical reactions*

The vapour-air mixing process is primarily controlling combustion in this phase.

The authors believe that in a more "ideal process", the pressure in the sac volume should rise very distinct to a sufficient high level during start of injection to secure good atomisation and rapid mixing between fuel and air. This results in a short ignition delay and equally short premixed combustion phase. The initial pressure should be limited in order to reduce the premixed heat release and thereby avoiding unnecessary large pressure gradients and local temperature peaks. During the mixing controlled combustion phase, the rate of injection should be progressive in order to maintain the cylinder pressure as the cylinder volume is increasing and less oxygen is available for combustion. The closing of the injection valve should also be distinct to minimise the period with decreasing pressure, thereby reducing the amount of fuel injected with reduced penetration.

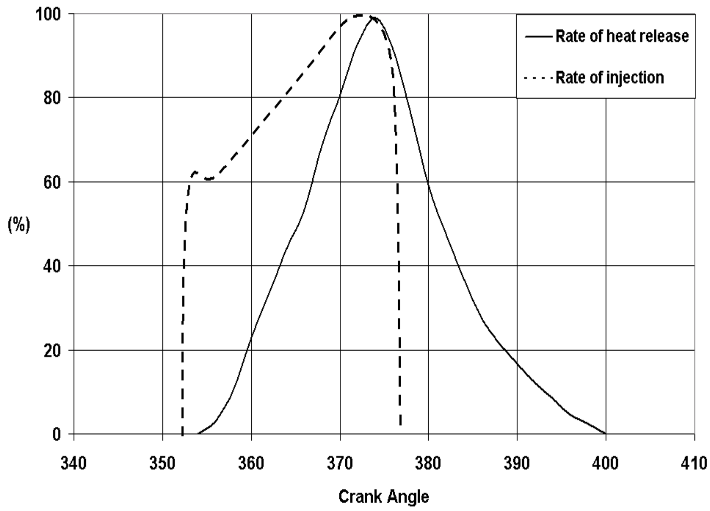


Fig. 3 Ideal injection rate (dotted) and rate of heat release (solid)

BOND GRAPHS

Bond graphs are a unified graphical notation for the representation of physical systems. The basic concept is that power is the fundamental constraint in a physical system and is the one variable that is common to the whole system. Bond graphs consist of letters and numbers representing the components connected by lines or bonds standing for power interactions.

The bond is considered to conduct power between the ports instantaneously and without loss. Each bond has an effort, e, and flow, f, signal and in true bond graphs the product of the two signals is the instantaneous power between system A and B shown below.

$$A \underset{f}{\overset{e}{\rightharpoonup}} B \qquad \text{Power} = e \cdot f$$

The half arrow shows the direction in which the positive power is flowing. Two other types of variables that are important are momentum, p, and displacement, q, in generalised notation. Momentum is defined as the time integral of an effort and the displacement variable is the time integral of a flow variable. Momentum and displacement variables can be used to represent system energy and thus state variables.

Using the classification of power and energy variables presented previously, it turns out that only nine basic types of multi-port elements are required in order to represent models in a variety of energy domains. These multi-ports function as components of subsystem and system models. They are in many cases idealised mathematical versions of real components, in other cases they are used to model physical effects. The nine basic multi-ports are classified in Table 1, according to the way they process energy. Roughly speaking, these elements account for energy dissipation, storage, supply and conversion of energy from one form to another.

Table I Basic bond graph elements

Bond graph element	*Elemental relations*	*Examples (Mechanics, hydraulics, electricity)*
$\frac{e}{f}\rightharpoondown$ R — Resistor	General: $e = \Phi_R(f)$ Linear: $e = R \cdot f$	Damper, friction, resistor
$\frac{e}{f}\rightharpoondown$ C — Capacitor	General: $q = \Phi_c(e)$ Linear: $q = C \cdot e$	Spring, volume, capacitor
$\frac{e}{f}\rightharpoondown$ I — Inertia	General: $p = \Phi_I(f)$ Linear: $p = I \cdot f$	Mass, inertia, inductance
Se $\frac{e}{f}\rightharpoondown$ — e-source	$e = e(t)$	Source of force, pressure, voltage
Sf $\frac{e}{f}\rightharpoondown$ — f-source	$f = f(t)$	Source of speed, flow, current
$\frac{e_1}{f_1}\rightharpoondown$ TF $\frac{e_2}{f_2}\rightharpoondown$ — Transformer	$e_1 = e_2 \cdot m$ $f \cdot m_1 = f_2$	Rocker arm, hydraulic ram, transformer
$\frac{e_1}{f_1}\rightharpoondown$ GY $\frac{e_2}{f_2}\rightharpoondown$ — Gyrator	$e_1 = f_2 \cdot r$ $f_1 \cdot r = e_2$	Gyrator, -, voice coil transducer
$e_3 \mid f_3$ ↓ ; $\frac{e_1}{f_1}\rightharpoondown$ 0 $\frac{e_2}{f_2}\rightharpoondown$ — 0-junction	$e_1 = e_2 = e_3$ $f_1 - f_2 + f_3 = 0$	Common force, pressure, voltage
$e_3 \mid f_3$ ↓ ; $\frac{e_1}{f_1}\rightharpoondown$ 1 $\frac{e_2}{f_2}\rightharpoondown$ — 1-junction	$f_1 = f_2 = f_3$ $e_1 - e_2 + e_3 = 0$	Common speed, flow, current

Using these basic elements, a graphical model can be constructed of a system. It is a remarkable fact that models based on apparently diverse branches of engineering science all can be expressed using the notation of bond graphs. This allows one to study the structure of a system model. The nature of the parts of the model and the manner in which the parts interact can be made evident in a graphical format.

Using bond graphs, the differential equations can be constructed in an algorithm manner from the graph. That is the graph is perfectly equivalent to the differential equations of the form

$$\frac{d\vec{\chi}}{dt} = f(\vec{\chi}, \vec{u}, t) \qquad (1)$$

with state vector, $\vec{\chi}$, and input vectors $\vec{u}$. The state equations can thus be written unambiguously in an algorithmic way.

However, before the equations can be written from the bond graph, causal information must be added to the graph. Causality means obtaining an explicit indication of which variables for an element is to be considered independent and which are to be considered dependent. Said another way, which are the inputs and which are the output variables. The problem of assigning a particular causality to a system model is related directly to the manner in which the system can be simulated, or the manner in which the system equations can be written without difficulty.

The way both the effort and flow signals travel are indicated directly on the graph by drawing a perpendicular line at one or the other end of the bonds.

The notation above tells us that the direction of which positive power flows, is from A to B. Input and output of system A is flow and effort, respectively. For system B, effort is input while flow is output.

The simultaneously representation of physical and computational structure is a unique and very powerful property of the bond graph which other representations like the linear graph or the block diagram do not have.

MODELLING

From a modellers point of view, fuel injection systems consist of more or less the same components, enabling the modeller to use and re-use basic sub models. This re-usability of models is crucial in an effective modelling environment. We will now present the sub-models required for assembling overall models for both the traditional injection system[4] and the common rail injection system.

Nozzles

Hydraulic nozzle flow is modelled using the general equation:

$$Q = C_f \cdot A \cdot \sqrt{\frac{2}{\rho} \cdot (p_u - p_d)} \tag{2}$$

where C_f is the flow coefficient, A is the flow area, ρ is the density and p_u and p_d are upstream and downstream pressure. The nozzle is modelled as a resistor (R-element). Input to this model is upstream and downstream pressure, while the output is flow rate.

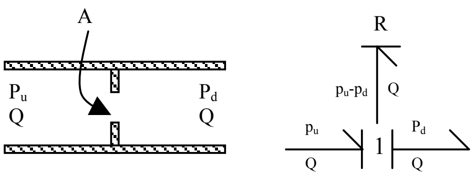


Fig. 4 Resistor element

Hydraulic flexibility of volumes

Hydraulic flexibility must be included in a model where high frequency dynamic behaviour is to be studied. The hydraulic flexibility is given by the bulk modulus of the fluid at constant temperature, defined as:

$$\beta = \rho \cdot \left(\frac{\partial p}{\partial \rho} \right)_T \tag{3}$$

For a constant volume element, the constitutive relation becomes:

$$p = p_0 + \frac{\beta}{V} \cdot \int (Q_u - Q_d) dt \tag{4}$$

and for a variable volume element like the cylinder of the fuel pump, the relation becomes:

$$p = p_0 + \int \frac{\beta}{V} \cdot (\dot{V} - Q) dt \tag{5}$$

The flexibility of the volume elements shown next is given by the relation:

$$C = \frac{V}{\beta} \tag{6}$$

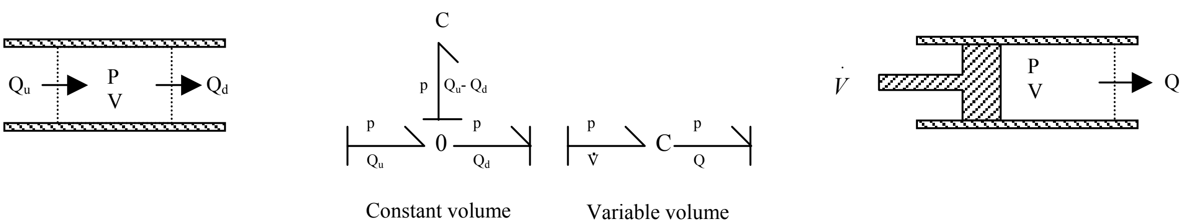


Fig. 5 Flexibility elements

Inputs to these models are upstream and downstream flow rate (or rate of volume change), while the output is pressure.

Pipe model

The pipe model is obtained by solving the continuous flow case where the flow rate can be represented by Newton's equation and the continuity equation. The two equations can be combined to form the well-known wave equation.

$$F = m \cdot a$$
$$\rho \cdot \frac{\delta Q}{\delta t} = -A \cdot \frac{\delta P}{\delta z} + g(z,t) \tag{7}$$
$$Q - \left[Q + \frac{\partial Q}{\partial z} \cdot dz \right] + q(z,t) = \frac{\partial V}{\partial t}$$
$$\frac{A}{\beta} \cdot \frac{\delta P}{\delta t} = -\frac{\delta Q}{\delta z} + h(z,t) \tag{8}$$

where $g(z,t)$ represents the external forces on the fluid element and $h(z,t)$ represents flow sources. Using standard separation of variables technique, one may express Q as a product of $\xi_i(t)$ and $G_i(z)$. $\xi_i(t)$ represent model general co-ordinates and $G_i(z)$ represent the mode shapes given by the homogenous solution of equations 7 and 8. Since the model functions are orthogonal, a set of de-coupled ordinary differential equations is obtained for each of the normal modes.

$$Q = \sum_{i=0}^{\infty} G_i(z) \cdot \xi_i(t) \tag{9}$$
$$G_i(z) = \cos\left(\frac{i \cdot \pi \cdot z}{L} \right) \qquad i = 0,1,2,3.... \tag{10}$$
$$m_i \cdot \ddot{\xi} + k \cdot \dot{\xi} = \dot{P}_0 \cdot G_i(0) - P_L \cdot G_i(L) \tag{11}$$

P_0 and P_L are the pressures at the respective pipe ends. Each equation can then be solved separately as a result of the orthogonality. The bond graph model of the modal pipe flow consisting of R-, C- and I-elements in addition to transformers is illustrated in Figure 6. The bond graph parameters are given as:

$$m_0 = \frac{\rho \cdot L}{A} \qquad k_0 = 0$$
$$m_i = \frac{\rho \cdot L}{2 \cdot A} \qquad k_0 = \frac{\beta \cdot (i \cdot \pi)^2}{2 \cdot A \cdot L} \qquad i = 1,2,3,... \tag{12}$$
$$I_0 = \frac{\rho \cdot L}{A} \qquad C_0 = \infty \qquad R_0 = \frac{8 \cdot \upsilon \cdot \rho \cdot L}{A^2} \qquad m = \cos\left[\frac{i \cdot \pi \cdot z}{L} \right]$$
$$I_0 = \frac{\rho \cdot L}{2 \cdot A} \qquad Ci = \frac{2 \cdot A \cdot L}{\beta \cdot (i \cdot \pi)^2} \qquad R_0 = \frac{8 \cdot \upsilon \cdot \rho \cdot L}{2 \cdot A^2 \cdot i} \tag{13}$$

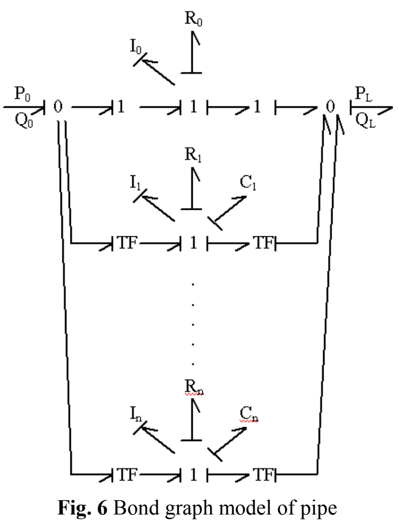


Fig. 6 Bond graph model of pipe

Input to the pipe model is upstream and downstream pressure, while the output is upstream and downstream flow rate

Fuel pumps
The fuel flows from the lower pressure system (Se-element) through the inlet port (R-element) into the pump cylinder (C-field). The fuel then flows through the discharge valve (R-element) and into the fuel line. The volume of the pump cylinder is given by the displacement of the cam (Sf-element). Between the cam and the pump cylinder transformers converts power from the mechanical to the hydraulic energy domain. The flow through ports are dependent upon the position of the plunger, Z, and also the fuel rack position, X.

For the common rail system, the high-pressure pump is modelled as a flow source. These pumps are mostly multi-cylinder displacement pumps where the flow is governed in very much the same manner as for the traditional pumps by controlling the effective pump stroke. The effective pump stroke is given by a governor with the purpose of achieving a given rail pressure. In this case there is little to be gained with respect to accuracy by modelling the individual pump cylinders.

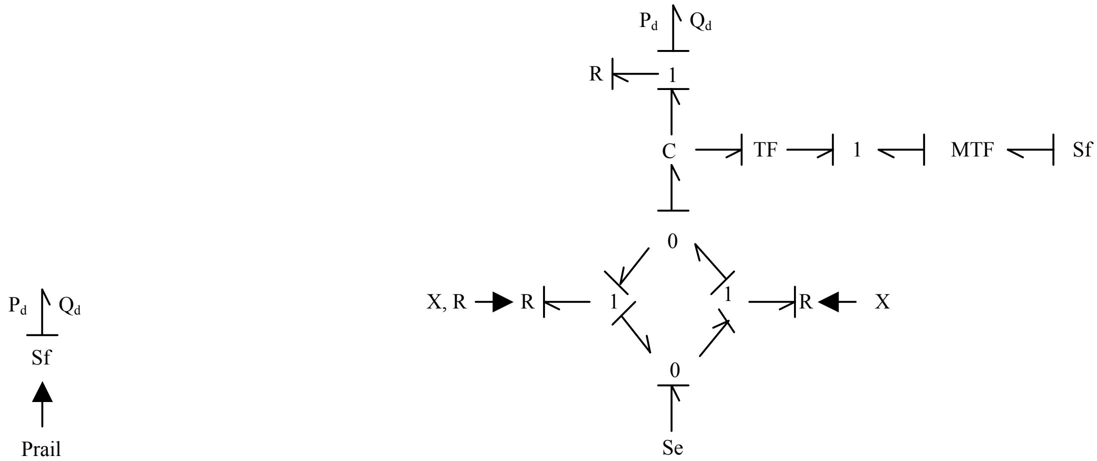


Fig.7 Bond graph models of pumps

Output from the pump models is flow rate.

Injectors
In addition to the hydraulic force (Through the transformer), forces from the return spring (C-element) and friction (R-element) are acting on the valve needle (I-element). When the valve is in closed position, the needle rests at the valve seat and when the valve is fully open, the needle hits a restriction. These contact forces are shown as "bumpers", a spring and damper system (C- and R-element).

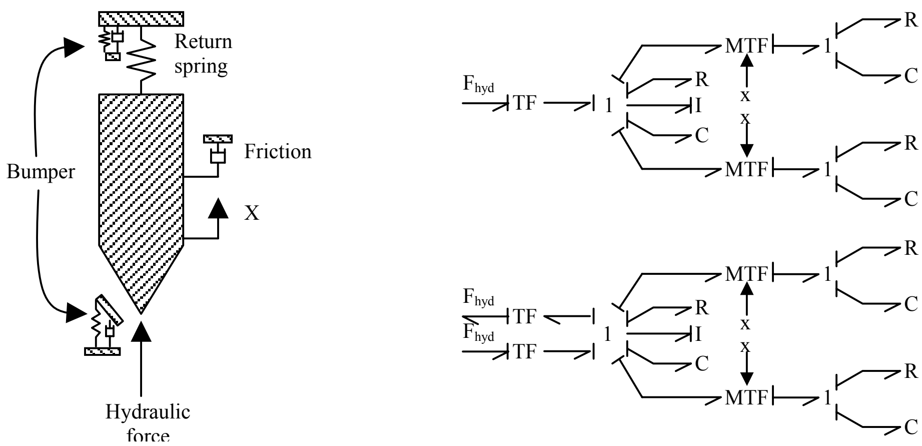


Fig. 8 Bond graph model of mechanical parts of the injectors (Common rail injector bottom)

The common rail injector is modelled much in the same manner as the traditional injector. The only difference is the additional force originating from the pressure in the volume above the pressure rod.

In order to determine the motion of the valve needle, the pressure distribution along the needle must be determined accurately since it varies due to the acceleration of the fluid as it enters the sac volume.

The flow is considered as steady state without friction with known pressure in the injector volume. The local pressure can then be found by using Bernoulli's law.

$$p(x) = p_0 - \frac{\rho \cdot v^2}{2} \tag{15}$$

The conical surface perpendicular to the flow velocity between the valve and the valve needle is given by[5]:

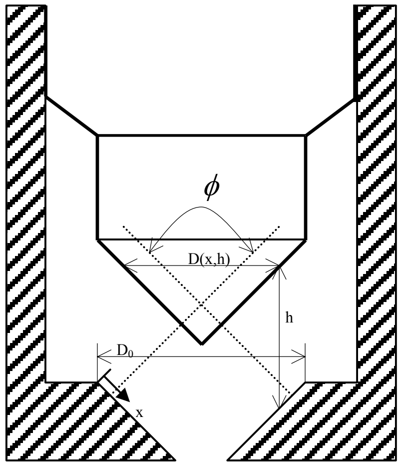
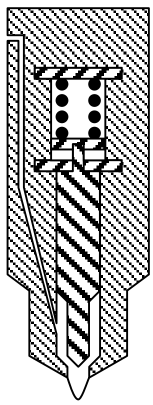


Fig.9 Injector needle

$$A(x) = 2 \cdot h \cdot \cos\left(\frac{\pi - \phi}{2}\right) \cdot \sin\left(\frac{\pi - \phi}{2}\right) \cdot \left(\frac{\pi}{4}\right) \cdot \left[\frac{D_0}{\cos\left(\frac{\phi}{2}\right)} - 2 \cdot x \cdot \tan\left(\frac{\phi}{2}\right) - h \cdot \cos^2\left(\frac{\pi - \phi}{2}\right)\right] \tag{14}$$

The mechanical part of the injector model will now be connected to the hydraulic part. The fuel flows trough the bores (Pipe) and into the variable volume surrounding the valve needle (C-field). The pressure in this volume acts on the injector needle. The fuel then flows through the flow area given by the position of the needle (R-element) and in to the sac volume (C-element). The pressure differential between the sac volume and the engine cylinder (Se-element) determines the flow into the cylinder through the nozzle bores (R-element).



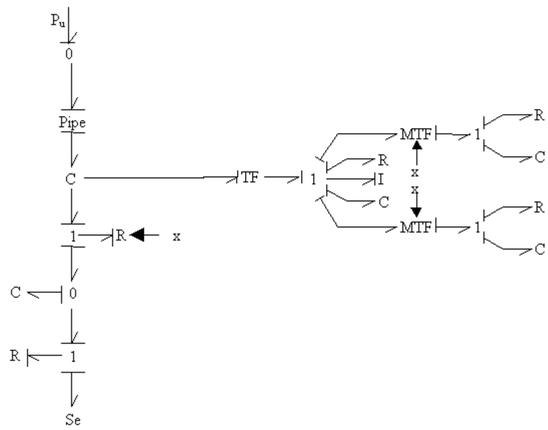


Fig. 10 Bond graph model of conventional injector

As we can see to the right, the bond graph model of the common rail injector is a bit more comprehensive than the model of the traditional injector. The model of the mechanical part of the solenoid valve is quite similar to the model of the mechanical part of the valve needle. The major difference is the force from the electromagnet (Se-element).

At the top of the internal bores (Pipe), the fuel flows through a small orifice (R-element) into a variable volume on top of the pressure rod (C-field). The pressure in this volume acts on both the valve needle and the valve shutter. From this volume, the fuel flows through a valve (R-element) where the flow area is determined by the position of the valve shutter (Solenoid controlled) and into an another variable volume (C-field). This volume is connected through an orifice (R-element) to the low-pressure fuel system (Se-element).

Input to the injection valve models is the upstream pressure.

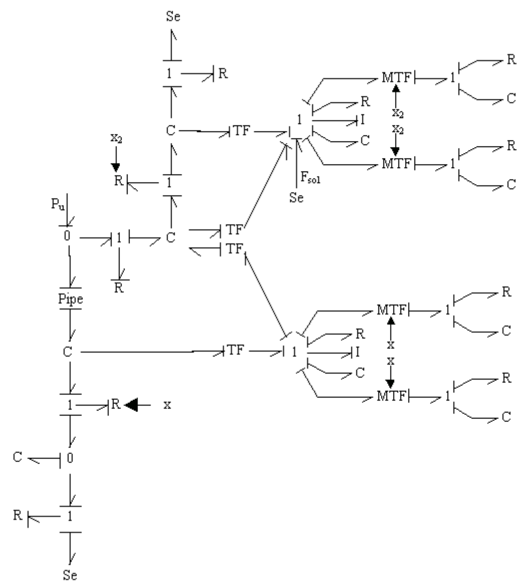


Fig. 11 Bond graph model of common rail injector

Complete system model
The sub models presented above are modelled using the modelling tool MS1. The sub-models are used to assemble total models of a traditional injection system and a common rail injection system much in the same manner as building the physical system. Using bond graphs in combination with matching software enables the engineer to make changes and modifications to a model in a straightforward manner without reformulation of equations. The figure below shows the two models as they appear in MS1.

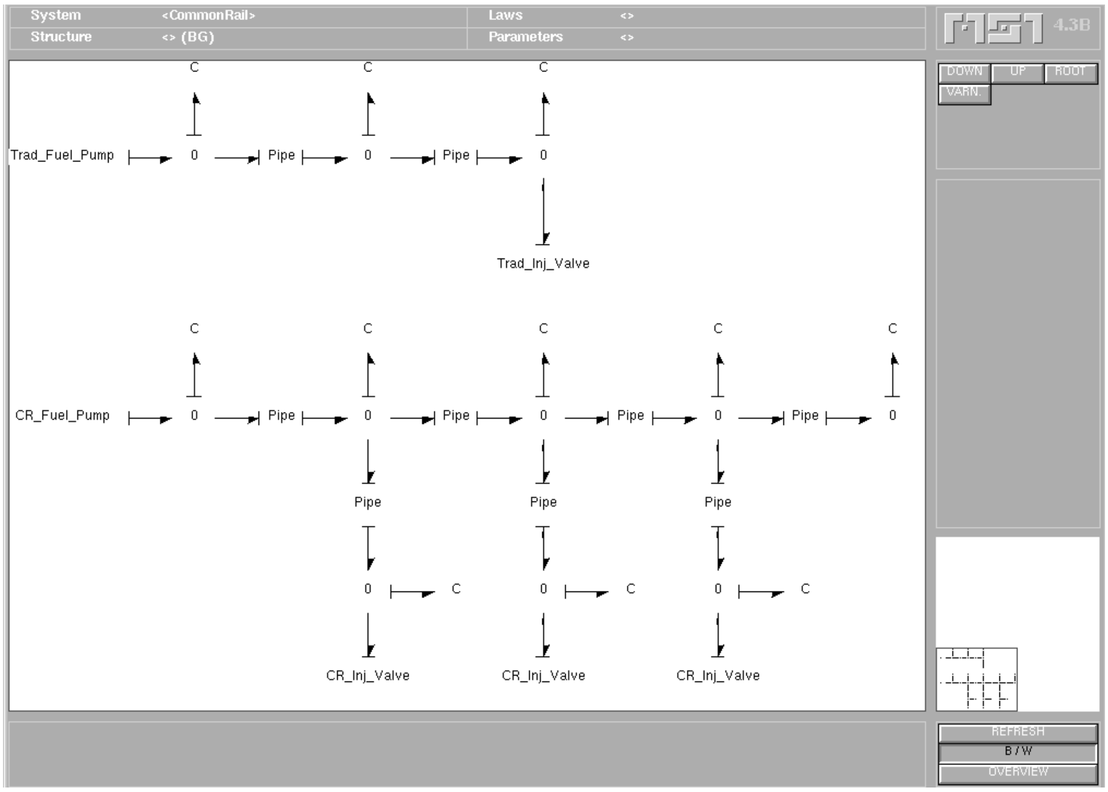


Fig. 12 The fuel injection system models as they appear in MS1

MS1 is a modelling environment aimed at the study of dynamic systems that can be represented by ordinary differential-algebraic equations. Its multi input/output possibilities provide a consistent handling of several model representations in addition to bond graphs. MS1 generates source code used for simulation in the designated solver, in our case ACSL.

SIMULATION

The plot shows the pressure in the sac volume for both the traditional injection system and the common rail injection system. The pressure in the rail is 1000 bar.

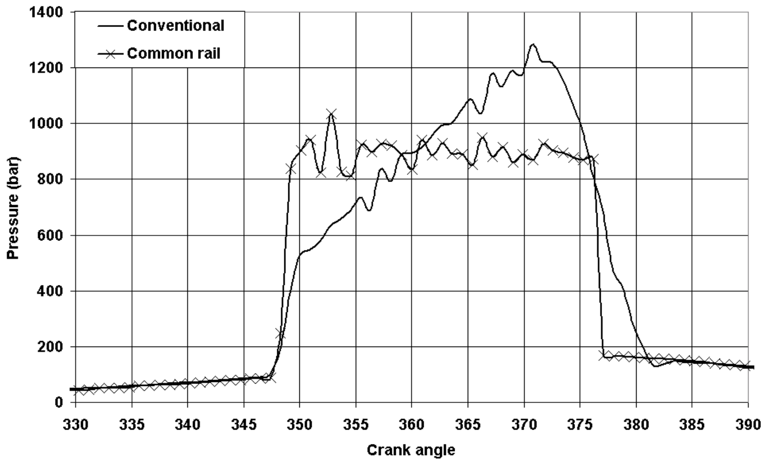


Fig. 13 Pressure in sac volume for the traditional and common rail injecteion system

Characteristically, the common rail system gives a distinct opening and closing of the injection valve. This secures rapid atomisation and mixing between fuel and air in the initial phase, and minimises the end-phase of the injection period with rapidly dropping pressure. The pressure in the sac volume is fairly constant throughout the injection. For the traditional injection system, the pressure in the sac volume rises more slowly in the early stages of injection. However, throughout the injection the pressure in the sac volume continues to increase due to the cam profile and is exceeding the pressure obtained by the common rail system. But in the final stage, the pressure in the sac volume decreases slower as the pressure is relieved at the pump side of the system.

Now, what does this mean? A constant or slightly decreasing pressure in the sac volume gives a too intensive rate of injection at the start causing a steep combustion pressure rate, limited by the maximum firing pressure. The next phase of combustion, the rate of energy conversion is mixing controlled meaning that the injection pressure should be as high as possible. The injection pressure supplied by the common rail system is at it's highest during the initial stages of the combustion, and may even decrease throughout the injection. All together this means that the rate of injection is not optimal for a medium speed combustion process.

For the traditional cam driven system the rate of injection is progressive which is beneficial for the diesel process. The start of the injection is depending upon the spring force of the injector and the injection process is load dependent. The end of the injection is also a weak part of a mechanical system. Thus, the ideal system should be a combination of the two.

SUMMARY AND CONCLUSIONS

Bond graph models are developed for the most common components found in injection systems. These sub-models are used to assemble models of a traditional injection system and a common rail injection system much in the same manner as assembling the physical system.

Earlier, the most time consuming part of modelling and simulation was the formulation of proper source code for simulation. Less time was left to the actual modelling of the system, and perhaps most important of all, analysis of simulation results. By using bond graphs in combination with matching software like MS1, this is no longer the case. The generation of computer code is done by merely pressing a button. This allows for the modeller to get source code from the updated model in just a matter of seconds. Simulations are performed in ACSL allowing for fast and accurate results.

The rate of energy conversion in the cylinder, commonly called rate of heat release, is closely related to the rate of injection, implying that the rate of injection is the key to an effective combustion process in the diesel engine. For common rail systems, the pressure differential over the nozzle bores is constant or declining throughout the injection. In the authors' opinion, the rate of heat injection should be progressive throughout the injection in order to maintain high cylinder pressure in the expanding cylinder volume where the concentration and availability of oxygen is declining. The traditional injection system will provide an inclining pressure pattern in the cylinder, but is not able to provide high injection pressure at all load conditions. The end of the injection is also a weak part of a mechanical system and variable timing and duration of the fuel injection is not easy to obtain. A more ideal injection system should be a combination of the two and MARINTEK is currently conducting internal activities based upon traditional injection system where a progressive pressure differential throughout the injection is obtainable along with a more distinct opening and closing of the injector.

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