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An engine layout study for common rail systems in large diesel engines

03 Fuel Injection & Gas Admission

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The Congress programme centres around the presentation of Technical papers on engine research and development, application engineering on the original equipment side and engine operation and maintenance on the end-user side. The topics of the 2016 event covered Product Development of gas and diesel engines, Fuel Injection, Turbochargers, Components & Tribology, Controls & Automation, Exhaust Gas Aftertreatment, Basic Research & Advanced Engineering, System Integration & Optimization, Fuels & Lubricants, as well as Users' Aspects for marine and land-based applications.

ABSTRACT

The proper layout of the accumulator function in a common rail system (CRS) is crucial for the competitiveness of the CRS in terms of costs and benefit. The accumulator function can be realized in various ways. It can be built up by a modular system that allows using the same accumulators and connecting parts in engines with different numbers of cylinders or by the integration of a manifold (common rail) that feeds several injectors. Accumulators can be integrated into the injectors. The accumulator function can alternatively be integrated into the fuel pipe line system between the high pressure pump and the injectors.

A systematic approach is presented for the identification of an optimal layout of the accumulator function in diesel engines with a power output in the range between 1 and 5MW by taking into consideration of the engine layout conditions, hydraulic and lifetime requirements, manufacturability and manufacturing economies as well as assembly and maintenance requirements. The effects of various design layouts of the accumulator on the hydraulic performance and fatigue life has been studied by means of hydraulic simulations and fatigue calculations using an in-line six cylinder model engine with 1750kW power output and a nominal operation pressure of 2000bars. Two fundamentally different approaches for the accumulator function have been studied in detail. In one approach the system is built-up in a modular way, whereas an accumulator of 500 or of 80cm3 volume content is integrated into the injectors. In a second approach the CRS was built-up using a one piece accumulator (rail) with 500cm3 volume content. In both approaches the damping effect of orifices on pressure fluctuations was studied. For the rail variant the damping effect of additional, small sized 40cm3 single accumulators, which can be integrated into the injector or in front of the injector, was additionally studied.

Hydraulic performance is discussed in terms of the pressure drop within the injector, injected discharge rate development over one cycle, amount of injected fuel per cycle as well as their reproducibility from one cylinder to the other. It was shown that the best combination with respect to a minimum drop of the injection discharge rate and to a reproducible injection discharge rate function can be achieved with two fundamentally different approaches: (1) by a modular system with 500cm3 accumulator integrated injectors in which pressure pulsations within the jumper lines are dampened by orifices and (2) by a non-modular common rail system with a 500cm3 one piece accumulator, which is stabilized against pressure pulsations within the injector by means of additional 40cm3 single accumulators that are placed in front of the injector or integrated into the injectors. Under given design boundary conditions a slight but visible advantage of variant (1) over variant (2) in terms of the reproducibility of the injection discharge rate function from one cylinder to the other is found.

Damping of pressure pulsations is found to be crucial for the high pressure capability (fatigue life) under nominal operation conditions at the working pressure of 2000bar. The relevance of the hydraulic characteristics diminishes when fatigue lifetime is governed by a high number of start-up/shut-down cycles.

As long as advanced requirements in regard to hydraulic performance and fatigue life can be realized with a modular set-up with large accumulator integrated injectors as well as with an optimized CRS based on an external accumulator system, the decision for either of these approaches must be made dependent upon given engine layout conditions, cost targets, manufacturing opportunities and preferences with respect to assembly and maintenance works. The relative strengths and weaknesses of studied variants are briefly discussed in relation to the above mentioned decision criteria.

INTRODUCTION

Combustion in modern diesel engines is optimised by means of a common-rail system (CRS) [1]. This optimisation of combustion is basically provided by two functions of the CRS: by a high pressure accumulator and by an electronically controlled injection, whereas high pressure injection is enabled by a high pressure pump and the compressibility of diesel fuel within the accumulator volume. This study deals with the optimisation of the accumulator function.

The accumulator volume can be integrated into the injectors or into the fuel pipe line system between the high pressure pump and the injectors. The system can be built up in a modular way that allows using the same components for the integration of the CRS in engines with different numbers of cylinders. The system can also be built up using a manifold (common rail), which supplies multiple injectors with high pressure fuel.

The way the accumulator function is designed affects the hydraulic performance of the CRS, robustness of assembly and maintenance works as well as the manufacturing costs. It may appear that there are no established design guidelines for an optimal balance between the accumulator volume to be integrated into the injector and the volume to be integrated into the pipe line system between pump and injectors. As the demand for the accumulator volume increases designers are faced with the following challenges:

- efficient use of minimal space available for the integration of accumulators, in particular in close proximity to the injection nozzle
- cost efficient manufacturing where the manufacturing effort increases with increasing volume of needed components
- sealing of components with larger sealing diameter and associated higher torques
- robust assembly of the CRS that allow zero fault, manual assembly of more bulky parts

For a straightforward optimisation of the cost-benefit balance of the accumulator function some basic design guidelines need to be developed. This paper is an attempt to establish such guidelines based on a systematic analysis and evaluation of design opportunities, hydraulic simulations, fatigue resistance, manufacturability as well as manufacturing economies of high pressure components.

OBJECTIVES

DESIGN BOUNDARY CONDITIONS

The integration of the accumulator function into the CRS involves three main tasks: (1) the decision on the layout of the accumulator function (modular vs. non-modular CRS, accumulator integrated injectors vs. small injectors), (2) diameter of fuel lines and geometry of sealing connections and (3) the positioning and fixation of all structural members on the engine block. Any pipe line system can be built up using fuel lines, junction or connection blocks, accumulators, pressure sockets and adaptors for the connection of structural members with different bore dimensions. The pipe line system itself provides various opportunities for the integration of edge filters, valves as well as sensors.

The decision on the layout of the CRS is influenced by the opportunities given by the engine layout but is mainly driven by hydraulic requirements, cost targets as well as assembly and maintenance requirements. Sufficient free space within the cylinder head has to be available for the integration of the accumulator function in or near the injector. If the accumulator function is to be placed outside the cylinder head the engine block has to provide opportunities for a safe and reliable fixation of the accumulator and fuel lines.

Figure 1a-f provides an overview of the most relevant variants. In general a minimum number of structural members and high pressure interfaces are strived for reasons of cost and robustness with respect to assembly, bearing in mind that each high pressure interface is a potential source of leakage. The cost of the accumulator function is naturally driven by the accumulator volume of its individual parts and by the total number of high pressure interfaces.

Volume wise fuel lines play a subordinate role as compared to accumulators. Accordingly the connection function of the fuel line is preferably provided by less costly fuel lines with smaller bore sizes that allow easier and safer assembly with lower torques.

The (non-modular) common rail variant shown in Figure 1a and 1b is characterised by the lowest possible number of high pressure components and interfaces. The hydraulic performance of this system can further be improved by the integration of an additional accumulator into the injectors (Fig. 1b) or in front of the injectors (Fig. 1c). A functionally equivalent system can be built up in a modular way (Fig. 1d and 1e). The modular system shown in Figure 1e is characterised by connection blocks (T-pieces) that enables direct connection of fuel lines (jumper lines) without using pressure sockets (quill tubes). It thereby minimises the number of structural parts and interfaces in a modular system.

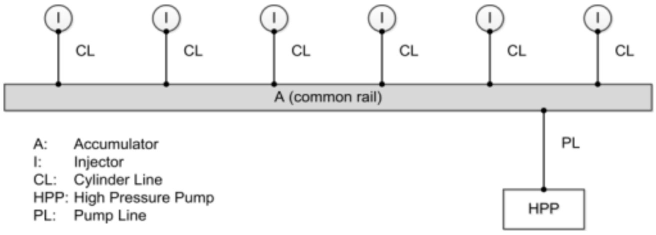

Figure 1a: non-modular CRS with a one piece accumulator

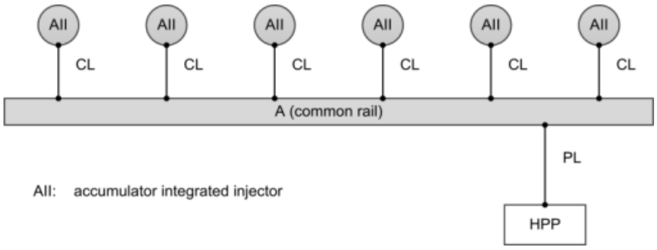

Figure 1b: non-modular CRS with a one piece accumulator and accumulator integrated injectors

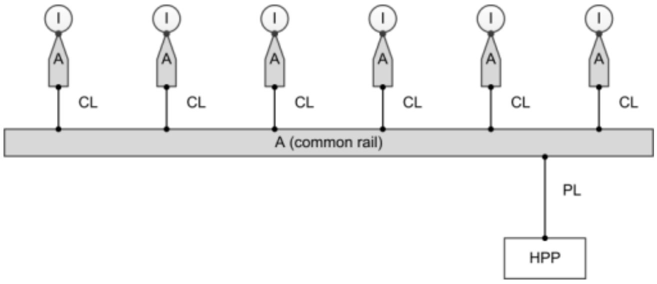

Figure 1c: non-modular CRS with a one piece accumulator and single accumulators

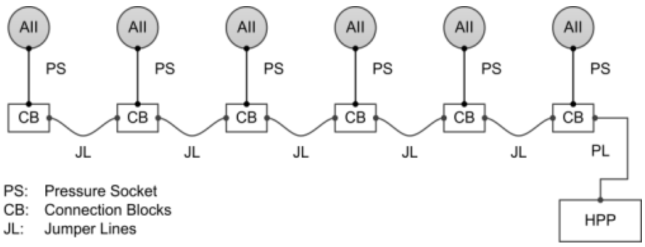

Figure 1d: modular CRS with accumulator integrated injectors connected by jumper lines and pressure sockets

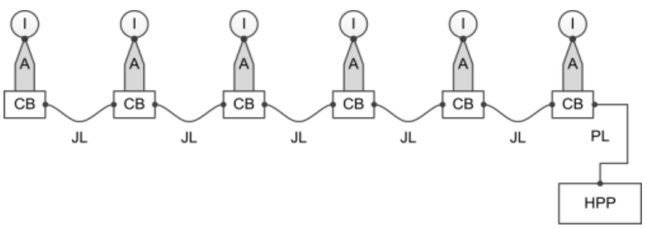

Figure 1e: modular CRS with single accumulators connected by jumper lines and pressure sockets

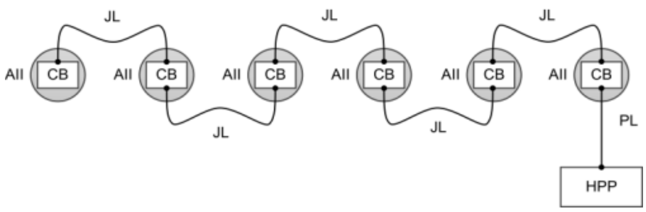

Figure 1f: modular CRS with accumulator integrated injectors connected by jumper lines

HYDRAULIC BOUNDARY CONDITIONS

The basic task in the hydraulic design of the accumulator function of the CRS is to provide a sufficiently large accumulator volume, so that the pressure drop during injection is within an acceptable range and the same injection pressure is supplied to all injectors. A further task in hydraulic design is to prevent and/or to damp high pressure oscillations within the system, since they are the most-likely source of uneven injection discharge rates and therefore unevenly balanced cylinders. In addition to that they reduce lifetime of components from a fatigue point of view.

Pressure drop and high pressure oscillations from the injector have common causes and can efficiently be minimized by adequately large accumulators that are placed within or in front of the injectors. Pressure oscillations can alternatively be minimized by orifices, but their layout has to be done with special care, since they produce additional pressure losses and thereby reducing the hydraulic efficiency of the system.

From a hydraulic efficiency point of view adding more accumulator volume always turns out to be a very potent remedy against both: pressure drop and oscillations problems. In view of the relevance of the accumulator volume on the hydraulic efficiency the question arises how the hydraulic requirements to a CRS can sufficiently be fulfilled with a minimum of total accumulator volume provided. The design opportunities provided by dividing the total accumulator volume in a common-rail volume that supplies fuel and pressure to multiple injectors and in a single accumulator volume supplying fuel and pressure to single injectors appear to be of high practical relevance.

MECHANICAL BOUNDARY CONDITIONS

The basic task of the mechanical design is to ensure the durability of the high pressure components and of the assembly for the entire engine life cycle. The loads on the components originate mainly from pressure pulsations, preloaded connections and mechanical vibrations. The notch effect leads to local stress concentrations at intersections that can be several times higher in magnitude than the surrounding stresses.

The high nominal operating pressure in newer common rail engines introduces high dynamic stresses to the components. The most severe load cycle occurs during the start and the stop of an engine respectively, since the full operating pressure has to be built up or released. The stress caused by it is usually within the high cylce fatigue (HCF) range of the components. The number of the pressure pulsations during transient phases or nominal operating fluctuations is indefinite. These loading blocks must therefore be in the very high cycle fatigue (VHCF) range.

The material used for high pressure applications is predominantly quenched and tempered low alloyed steel with a high tensile strength (≥1000MPa). Despite the high strength of the used steels the introduction of internal compressive stresses by autofrettage is of vital importance for fatigue resistance. The required wall thickness of high pressure components is determined by the material strength and by the applied autofrettage pressure. Accordingly, the required ratio between the outer and the inner diameter is typically in the range between two and three.

MATERIALS AND MANUFACTURING PROCESSES

The realisation of a CRS is essentially challenged by four tasks: (1) the manufacturing of accumulators that can withstand high pulsating stresses; (2) the realisation of high pressure connections; (3) deburring (and cleaning) of internal high pressure sections and (4) the assembly and functional control of common-rail systems before delivery. Solving task (1) and (3) involves the application of special manufacturing and finishing processes that are worth to be briefly outlined here.

Advanced fatigue life requirements of high pressure components are met following a threefold path. It includes the use of high strength steels, the manufacturing of semi-finished, thick-walled products with “fault-free” internal surfaces and the application of a finishing treatment that provides a sufficiently high compressive stress on loaded internal surfaces. Cold drawn tubes can be used for connecting the various parts in the system. In turn gun drilling is needed for the manufacturing of accumulators and pipes with a diameter exceeding 20mm and is currently being applied for common rail manufacturing up to a length of 2800mm.

The manufacturing of large high pressure components is further challenged by finishing of bore intersections, which is difficult to get to by conventional machining technologies. Electro Chemical Machining (ECM) turns out to be the appropriate method for deburring within large accumulators and junction blocks, mainly due to its high material removal rate.

Same as with ECM autofrettage turns out to be the appropriate method for the introduction of high compressive stresses in large components, which is mainly favoured by the achievable compressive stresses along with its high depth of penetration. Alternatively case hardening can be applied for smaller parts with however a reduced effectiveness with respect to fatigue resistance [2,3].

Autofrettage and ECM are generally favoured methods for common rail applications due to their healing effect on any kind of surface imperfections within high pressure components. In both processes manufacturing efficiency can be strongly improved by the application of appropriate tooling.

MANUFACTURING ECONOMIES

Any given layout of a CRS needs to be evaluated with respect to the value it generates for its application relative to the price to be paid for that. Three factors are playing an important role.

The first factor relates to the number of structural parts and particularly the number of sealing interfaces needed to connect the high pressure pump with the injectors. Any design that can be built up with a smaller number of parts and sealing interfaces is likely to provide a cost advantage.

The second factor relates to the total storage capacity that needs to be provided. As the capacity increases material costs and more manufacturing costs of a high pressure component tend to increase as well. However, this is not a general rule. For example, it may be less expensive to drill a large bore diameter by gun drilling than a small bore diameter (Figure 2). Similarly, the processing time in an autofrettage process does not strongly depend on the total storage capacity of a component. Therefore, with respect to the accumulator size one may profit from, it could be economically expressed in terms of storage capacity economies, in which each added cubic millimeter decreases the average costs per unit volume.

The third factor relates to the total quantity of CRS to be produced over the full product life cycle. Manufacturing costs can be reduced by continuous improvement of the process with each produced lot. For large number of units any investment into the manufacturing equipment can further help to reduce unit costs by scale economies. Many structural parts in a common rail system are produced in direct proportion to the number of cylinders. Accordingly enhanced scale economies are to be expected for these parts.

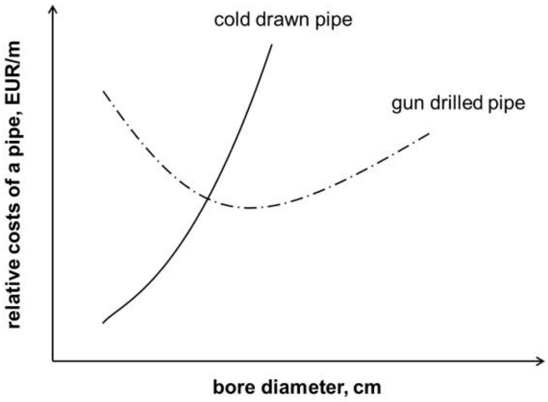

Figure 2: qualitative comparison between total material costs of cold drawn pipes and gun drilled pipes with OD/ID - ratio of 3; gun drilling costs tend to decrease with increasing bore diameter making gun drilled pipes cheaper than cold drawn pipes for bore diameters above 0.5 to 0.8cm

The cost disadvantage of a higher number of individual accumulators in a CRS can be compensated by better scale economies. On the other hand the cost disadvantage of producing less but larger accumulators can be compensated by better storage capacity economies. Table 1 provides an overview for which processes and parts scale and storage capacity economies prevail.

manufacturing / finishing technologies	**scale economies**	**storage capacity economies**
turning & milling	+	0
gun drilling	+	++
autofrettage	+	++
case hardening (carburising)	++	-
electro-chemical machining (ECM)	+	++
reference parts	cylinder lines connection blocks pressure sockets fittings	large accumulators (rail) large junction blocks thick (gun drilled) pipes

Table 1: Assessment of processes and parts with respect to scale and storage capacity economies. The main cost advantage of using a one piece accumulator as a manifold for all injectors in a row is provided by storage capacity economies in manufacturing: gun drilling, ECM and autofrettage.

EXPERIMENTAL METHOD

ENGINE PARAMETERS

A 6 cylinder model engine was designed with the engine parameters listed in Table 2:

engine type	6R
# of cylinders	6
power [kW]	1750
engine speed [rpm]	1000
power per cylinder [kW]	291.7
fuel consumption [g/kWh]	200
injected fuel mass per injection [g/injection]	1.944
injected fuel volume per injection [cm^3/injection]	2.3
rail pressure [bar]	2000
stroke [mm]	320
bore [mm]	250
injection duration (°CA)	25
equivalent diameter of injection holes [mm]	1.13
cylinder distance [cm]	37.5
MEP [bar]	22.3
average piston speed [m/s]	10.67

Table 2: applied engine parameters for hydraulic simulation and fatigue calculations.

ACCUMULATOR LAYOUTS

Two fundamentally different layouts of the accumulator function in the model engine were designed for evaluation. Their double walled layouts are shown in Figure 3. The accumulator volume is filled with red colour. Their geometric parameters are listed in Table 3.

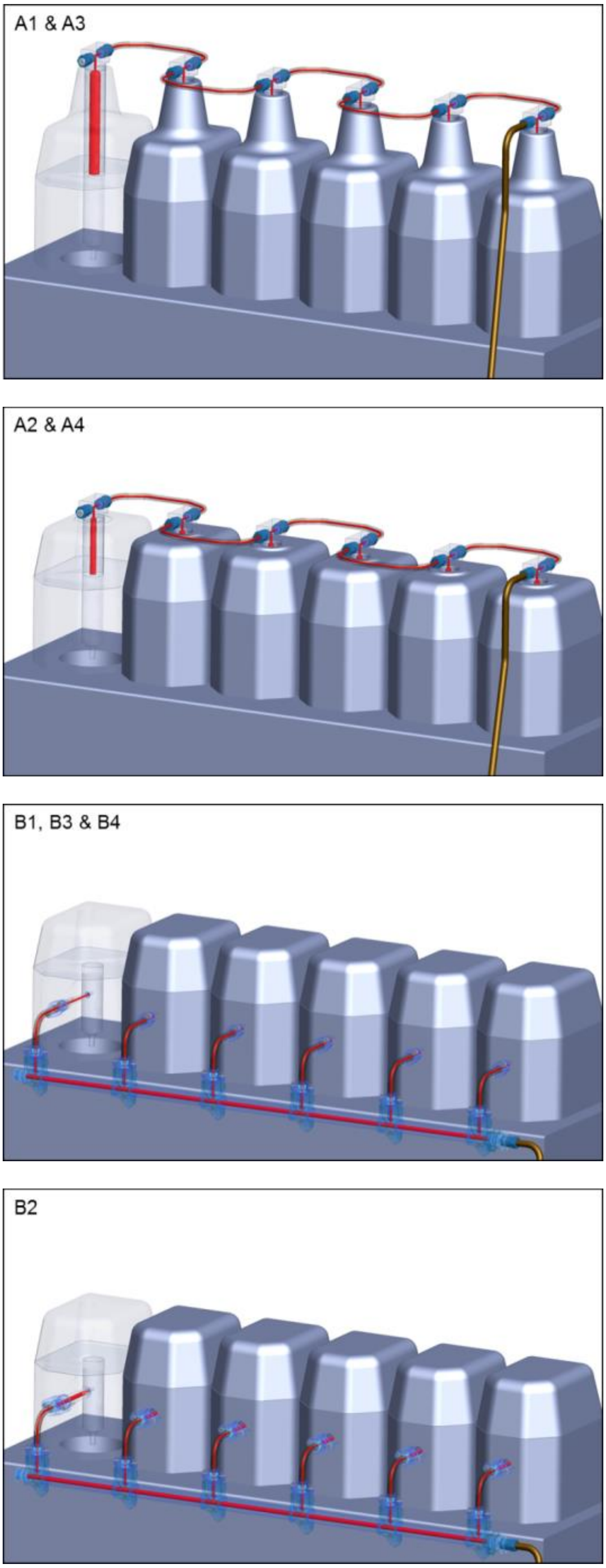

Figure 3: layout of studied common rail systems

designation	parameter	qty.
A1	dampened jumper line system with 500cm^3 accumulators (Figures 1f,3a)	
	bore diameter accumulator [mm] length accumulator [mm] volume accumulator [cm^3] bore diameter jumper line [mm] length jumper line [mm] volume jumper line [cm^3] orifice diameter [mm]	25.0 382.0 481.1 7.0 866.0 33.0 1.5
A2	dampened jumper line system with 80cm^3 accumulators (Figure 1f)	
	bore diameter accumulator [mm] length accumulator [mm] volume accumulator [cm^3] orifice diameter [mm]	22.0 211.0 80.1 0.8
A3	undampened jumper line system with 500cm^3 accumulators (Figures 1f,3a)	
	parameters same as for A1 orifice diameter [mm]	 omitted
A4	undampened jumper line system with 80cm^3 accumulators (Figure 1f)	
	parameters same as for A2 orifice diameter [mm]	 omitted
B1	conventional rail system (Figures 1a, 3b)	
	bore diameter rail [mm] length rail [cm] volume accumulator (rail) [cm^3] bore diameter injection line [mm] length injection line [mm] volume injection line [cm^3]	16.5 225.0 483.2 7.0 543.0 20.9
B2	conventional rail system with additional single accumulators (Figures 1a, 3c)	
	bore diameter single accumulator [mm] length single accumulator [mm] volume single accumulator [cm^3] bore diameter injection line [mm] length injection line [mm] volume injection line [cm^3] orifice diameter [mm]	15.0 220.0 40.0 7.0 277.0 10.6 4.4
B3	conventional rail system with optimised injection rates (Figures 1a, 3b)	
	parameters same as for B1 orifice diameter [mm]	 3.6
B4	conventional rail system with optimised water hammer effect (Figures 1a, 3b)	
	parameters same as for B1 orifice diameter [mm]	 2.45

Table 3: list of the layout parameters of studied common rail systems

Four layouts (A1 to A4) are characterised by an accumulator integrated injector with 500cm^3 large accumulators (A1 and A3) and with a 80cm^3 large accumulator (A2 and A4) respectively. Two of these variants contain an orifice at the injector entry towards the jumper line (A1 and A2).

Four rail variants (B1 to B4) are characterised by a one piece accumulator (rail) with an accumulator volume of roughly 500cm^3, which is about the same as in the accumulator integrated injectors of variant A1 and A3.

Rail variant B2 differ from all other rail variants by an additional accumulator of 40cm^3 that was placed in front of each injector containing no additional accumulator volume.

HYDRAULIC SIMULATION METHOD

Hydraulic simulations of the complete system were carried out based on the parameter listed in Table 2 for an assessment of the hydraulic characteristics of each system described in Table 3. It aimed at an assessment of the rail and injection pressure as well as of the injection discharge rates over a complete injection cycle and for the whole cylinder row. Results from hydraulic simulations were furthermore taken as an input for fatigue life assessments.

A generic model was set up in Matlab/Simulink for the simulation of the different CRS. Since the fluid needs to be considered as compressible, a barotropic equation of state was chosen for the pressure p:

$$p = p(\rho) \tag{1}$$

Since the pressure is only a function of the density ρ the Bulk modulus B is defined as

$$B(p) = \rho \frac{dp}{d\rho} \tag{2}$$

The kinematic viscosity ν, which is needed for the calculation of the friction losses in pipes, is modelled as an exponential function of the pressure and the temperature T. The latter is kept constant for all the simulations.

$$\nu = \nu_0 e^{\frac{aT+bp}{c+T}} \tag{3}$$

The flow rate Q of an orifice was calculated by the pressure difference Δp at the in- and outlet, a flow coefficient α, the flow area A and the density as follows:

$$Q = \alpha A \sqrt{\frac{2\Delta p}{\rho}} \tag{4}$$

Note that the density was always taken for the upstream flow direction.

For concentrated volumes the time derivate of the pressure $\dot{p}$ needs to be calculated. This was done by summing up all the in- and outflows from the considered volume.

$$\dot{p} = -\frac{B(p)}{V} \sum_{i=1}^{n} Q_i \tag{5}$$

Finally the pipes needed to be modelled. This was done by dividing the pipe into several sections with the length Δl, numbering the sections from 1 to *N* and solving a set of two ODEs for each section. The cross section A was kept constant over the entire length of the pipe

$$\dot{p}_{i+1} = \frac{B(p_{i+1})}{A\Delta l}(Q_i - Q_{i+1})$$

$$\dot{Q}_i = \frac{A}{\rho(p_{i+1})\Delta l}(p_i - p_{i+1} - \Delta p_{loss}) \tag{6}$$

Note that with this approach, for the first section the pressure p needs to be given and for the last section the flow rate Q needs to be given as boundary conditions. The pressure loss Δp_{loss} due to friction is modelled by

$$\Delta p_{loss} = \lambda \frac{\Delta l}{d_{hyd}} \frac{\rho(p_{i+1})}{2} \frac{Q_{i+1}^2}{A^2} \tag{7}$$

λ in equation (7) was calculated by the Prandtl-v.Karman-Colebrook relation for Reynolds numbers larger than 3000

$$\frac{1}{\sqrt{\lambda}} = 1.14 - 2\, lg\left(\frac{k_S}{d_{hyd}} + \frac{9.35}{Re\sqrt{\lambda}}\right) \tag{8}$$

and for Reynolds numbers less than 2300 by

$$\lambda = \frac{64}{Re} \tag{9}$$

For all Reynolds numbers in between, interpolation between these two formulas was carried out.

Pressure pulsations, generated by the pump, were not taken into account in this study. Nevertheless, for the sake of convenience, the incoming flow rate is controlled by a PI controller. Thus pressure pulsations for the pump are not absolutely absent in the results, but they are rather small, since every time, more than 20 cycles were simulated and therefore the controller output reached a quite stable state.

For injectors it is assumed that they open and close with infinite large speed. Since the scope of this work is to study the influence of the piping and distribution of the CRS, it can be argued that this is a sufficient assumption.

MECHANICAL CALCULATIONS

The aim of the fatigue calculation is to assess the safety factors of the various components if the part cycles around nominal operation pressure (NOP) of 2000bar during steady operation and between ambient pressure and maximum pressure (MP). The underlying pressure amplitudes were taken from the hydraulic simulation results, which are summarised in Table 4. The fatigue calculation for these components was done by combining analytical and numerical calculations. All fatigue calculations were carried out in accordance with the FKM guideline [4].

The analysis of the bending process and the determination of the autofrettage pressure were done by the use of the maximum distortion strain energy criterion (von Mises) and a non-linear finite element (FE) solver with bilinear material properties. The limit of the beneficial residual stresses from the autofrettage was chosen to be the yield strength of 950MPa of quenched and tempered 42CrMo4 (AISI 4140).

The preload forces of the screwed connections were determined by a table of standard values or by analytical calculations and used as initial loads within the numerical calculation. The friction coefficient was chosen to be 0.12.

The analysis of the stress concentrations caused by the combination of multiple loads was done by a linear FE solver. The scattering of the preload forces was taken into account.

The fatigue theory is determined by a biaxial analysis of the superficial principal stresses over ten steps from peak to peak including a simple filter to suppress stresses lower than 10% of the ultimate tensile strength. The quantification of the interaction of mean and altering stresses was done by the use of the Goodman relation equation (Haigh-diagram). The determination of finite life was done by means of the S-N curve (Wöhler curve).

RESULTS AND DISCUSSION

HYDRAULIC SIMULATION RESULTS

In Figure 4 the injection pressure of one chosen injector and the rail pressure of all considered systems are plotted. The start of injection is set at 250 °CA and the end of injection at 275°CA. Characteristic for the systems with the accumulator integrated injector, which are additionally dampened by an orifice, is a stable and flat rail pressure over the complete injection cycle. The injection pressure shows an initial drop which is continuously approaching the rail pressure over one injection cycle (A1, A2). The maximum pressure drop increases from 85bar to about 495bar, if the volume of the integrated accumulator is reduced from 500cm^3 to 80cm^3 (Table 4). If the orifice is omitted the rail pressure fluctuates with a much higher amplitude whereas the pressure drop in the injector is distinctly reduced.

The conventional rail system with a one piece accumulator (B1) is characterized by moderate pressure fluctuations within the accumulator (rail) but very large pressure fluctuations reaching peak pressures of around 2300bar within the injector. The maximum pressure amplitude within the rail varies between 80 and 170bars. Pressure fluctuations within the injector but also in the rail are strongly dampened by putting a small accumulator in front of each injector (B2) or by adding orifices at the rail exits (B3,B4). The criterion for the choice of the orifice is to avoid any influence from one injector volume to another due to the injection itself. It appears that these fluctuations are most strongly dampened by small orifices that suppress the water hammer effect (B4) however at the cost of an enhanced pressure drop within the injector, which is around 200bar. A small accumulator (B2) is not able to completely suppress injector pressure fluctuations, but can limit the oscillations within a bandwidth of about 200bar within an injection cycle. In variant B3 the orifice size was chosen in a way that the pressure within the injector follows as far as possible the pressure in the rail. Nevertheless remaining pressure fluctuations are still within a bandwidth of around 300 to 400bar.

From the hydraulic perspective, three fundamental requirements exist for the injection equipment: First a good hydraulic efficiency, second the injected fuel amount for all cylinders should be the same and third an equal shape of the injection discharge rate for all cylinders. The 2nd and 3rd requirement guarantee a good load balance between the cylinders. It is clear, if the 3rd requirement is fulfilled, then the 2nd is also fulfilled. In order to compare the hydraulic efficiency of the different systems, we need to compare the injected fuel amount, because a high pressure loss in the injections pressure results in a reduced injection flow rate and therefore in a reduced hydraulic efficiency.

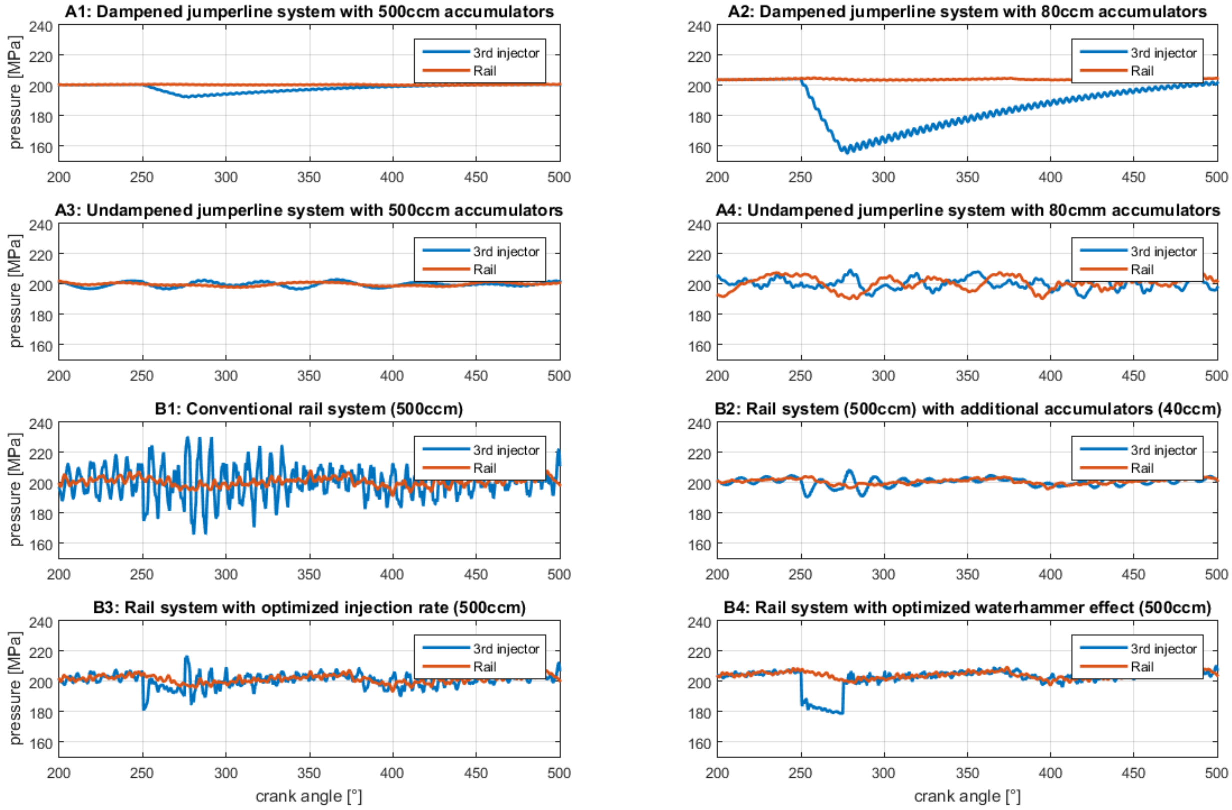

Figure 4: pressure oscillations in the rail (red) and in the injector (blue)

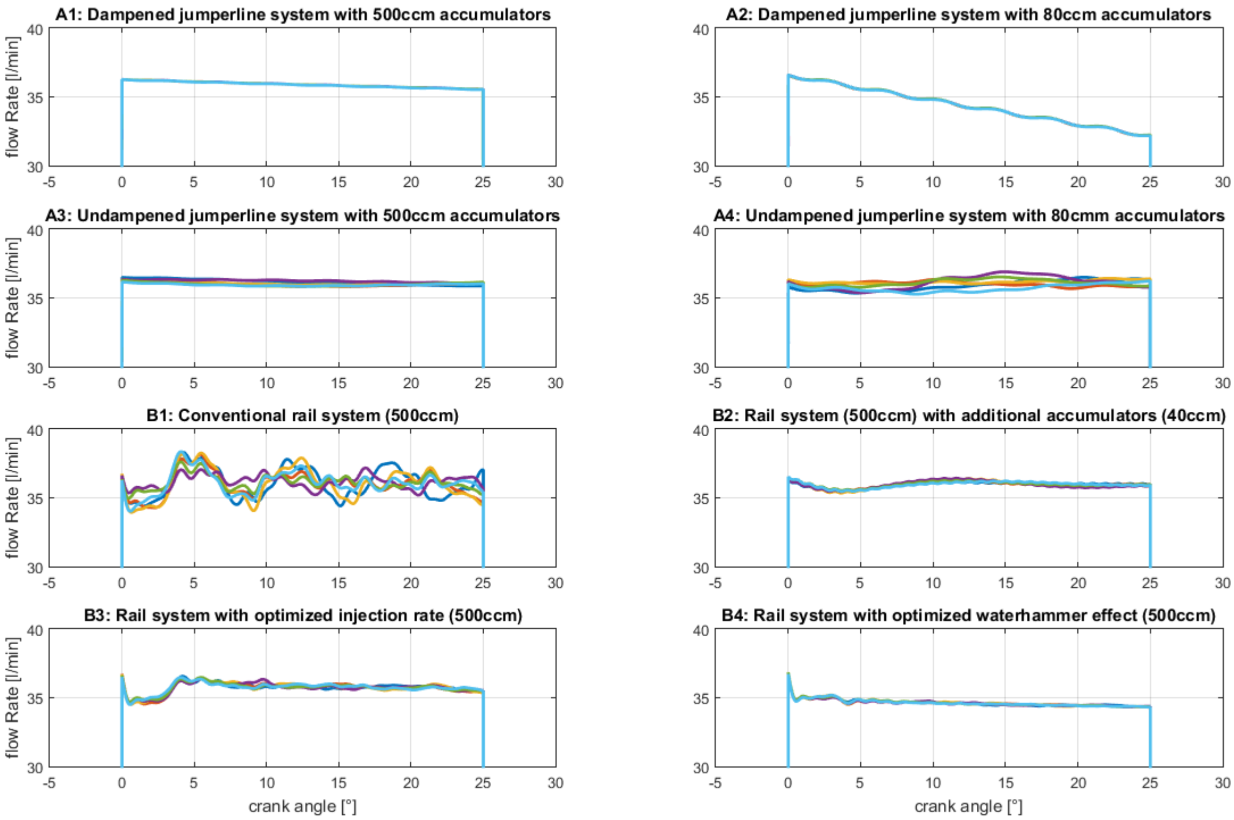

Figure 5: injection discharge rates of all six cylinders

Figure 5 shows the injection discharge rate of all systems and all injectors. The strongest reduction of the injection discharge rate was observed for the dampened jumper line system with small integrated accumulators (A2) and for the common rail system that was optimised towards a minimum water-hammer effect (B4). All other system were able to keep the injection discharge rate on their initial level even though it is fluctuating around a target level of around 36ml/min.

For better comparison of flow rates shown in Figure 5 the data were further processed to get the injected fuel amount per cycle and the standard deviation of the injected fuel amount over one cycle and all cylinders. The result of this analysis is shown in Figure 6.

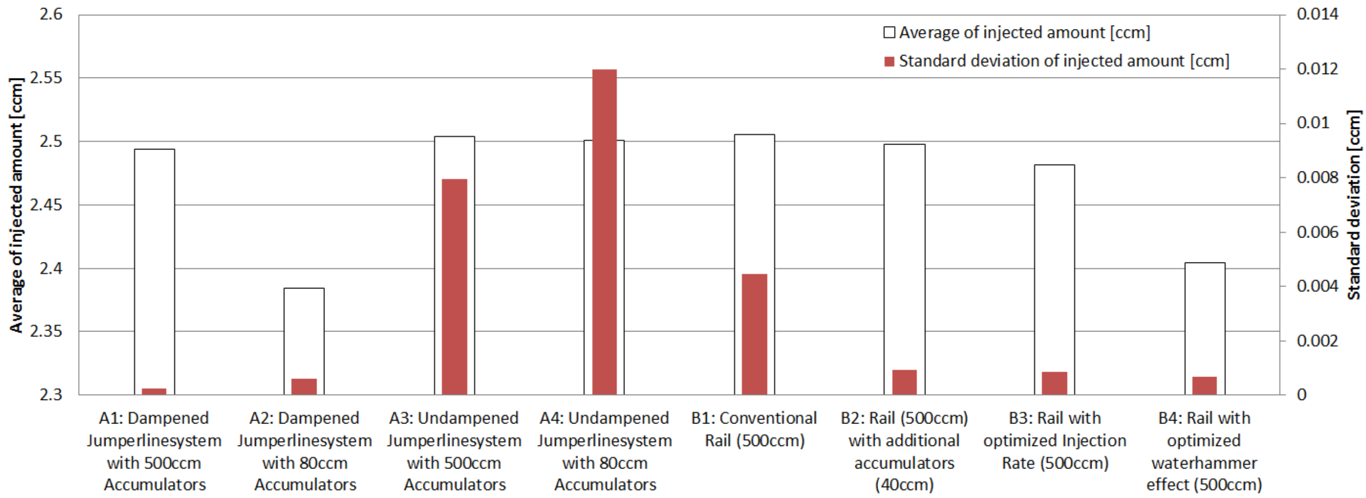

Figure 6: comparison of injected amount of fuel per cl.

As expected from Figure 5 the continuous reduction of the injection discharge rate results in a reduction of the average injected amount. This is strongest for the systems A2 and B4. Thus both mentioned systems A2 and B4 have a bad hydraulic efficiency compared to the other systems. The energy loss is in the range of 4%. The remaining systems mainly differ with respect to the standard deviation of the injected amount of fuel per cycle. In this respect the dampened jumper line system with large accumulator integrated injectors (A1) and the common rail system with additional accumulators in front of small injectors (B2) showed the best characteristics.

While Figure 6 provides an assessment of the reproducibility of the injected fuel amount from one cylinder to the next it does not provide an assessment of the reproducibility of the shape of the discharge rate from one cylinder to the other. Accordingly it is possible that two cylinders have the same injected fuel amount but totally different shapes in the injection discharge rate as depicted in Figure 5. In order to characterize the relative differences of the injection discharge rate between different cylinders a dissimilarity Δ has been defined that is obtained by integrating the difference of the injection discharge rates for all possible combinations of two injectors over one injection cycle as follows:

$$\delta_{ij} := \int_{\varphi_{start}}^{\varphi_{stop}} \sqrt{\left(Q_{inj,i}(\tilde{\varphi}) - Q_{inj,j}(\tilde{\varphi})\right)^2}\, d\tilde{\varphi} \qquad (10)$$

From this integration a dissimilarity Δ can be defined in the following way:

$$\Delta := \frac{1}{2} \sum_{i=1}^{\#cyl} \sum_{j=1}^{\#cyl} \delta_{ij} \qquad (2)$$

In addition to this dissimilarity parameter the drop of the injection rate between start and end of injection was calculated as a percentage of the initial flow rate. Figure 7 shows the comparison with respect to the drop of the injection discharge rate as well as with respect to the reproducibility of the injection discharge rate function during an injection cycle between different cylinders. For best performance both numbers should be as small as possible.

According to the results shown in Figure 5 best reproducibility of the injection discharge rate is in any case achieved with a damped jumper line system with accumulator integrated injectors, independent of the size of the accumulator. However for small accumulators gained equality is bought at the cost of an excessive drop of the injection discharge rate.

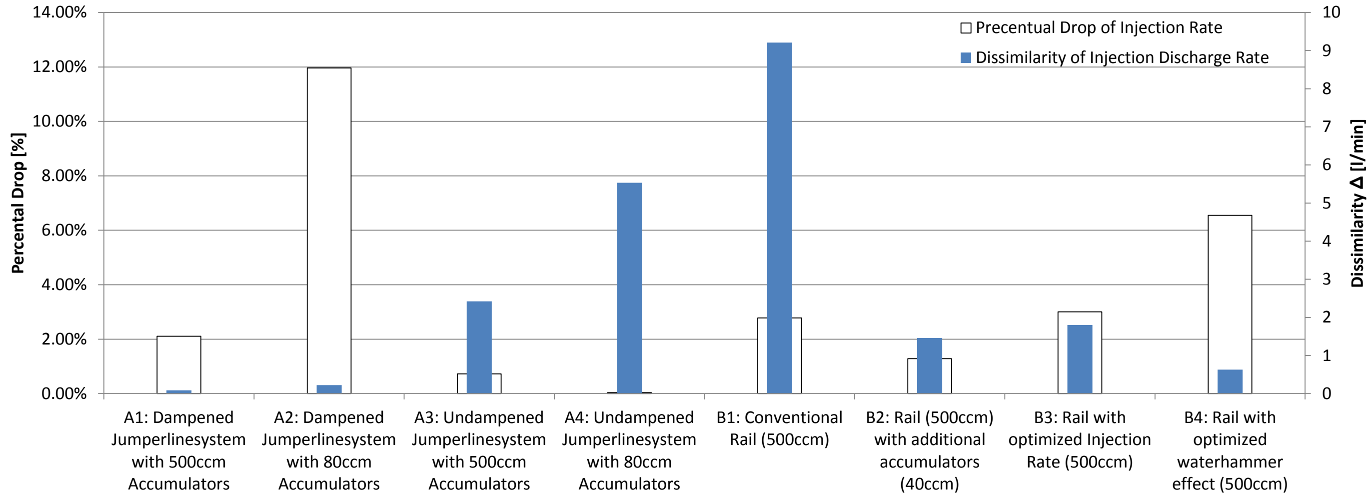

Figure 7: drop of the injection rate during an injection cycle and dissimilarity (non-reproducibility) of injection discharge rate

The rail system with additional accumulators (B2) appears to be the best alternative to a jumper line system with large accumulator integrated injectors. The potential improvement by increasing the volume in the small accumulator as well as by adding orifices is worth to be investigated.

RESULTS OF MECHANICAL CALCULATIONS

For fatigue life assessment two operating conditions are considered and defined as follows:

MPA: cycling between ambient pressure and peak operating pressure

NOP: cycling between minimal and maximal operating pressure at a nominal pressure of 2000bar

Their values are with reference to Figure 4 listed in Table 4.

affected parts	rail (B1-B4), jumper lines & T-pieces (A1-A4)		injection lines & small accumu-lator (B1-B4), accumulator (A1-A4)	
variant	MPA [bar]	NOP [bar]	MPA [bar]	NOP [bar]
A1	2010	5	2010	85
A2	2045	10	2045	495
A3	2015	35	2020	50
A4	2070	170	2100	190
B1	2080	170	2300	630
B2	2040	80	2085	185
B3	2070	130	2160	360
B4	2080	105	2090	300

Table 4: cyclic pressure amplitudes used for fatigue life calculations

Adequate autofrettage parameters, which are listed in Table 5, were defined with regard to a nominal operation pressure of 2000bar.

variant	parameter	qty.
one piece accumulator (B1-B4)	autofrettage pressure [bar]	6000
	wall plastification [%]	11.75
	maximum total strain [%]	3
fuel line (all varaints)	autofrettage pressure [bar]	7000
	wall plastification [%]	14.6
	maximum total strain [%]	0.9
small accumulator body (B2)	autofrettage pressure [bar]	6000
	wall plastification [%]	35.7
	maximum total strain [%]	0.9
T-piece (A1-A4)	autofrettage pressure [bar]	7000
	wall plastification [%]	5
	maximum total strain [%]	2

Table 5: list of applied autofrettage parameters for selected components of all variants

Preload forces listed below were used for the calculation of the stresses, maximum principal stresses and residual stresses of the critical areas, which are based on a yield strength of 950MPa for quenched and tempered 42CrMo4 raw material:

- between rail and fuel line: 56 kN
- between small accumulator and fuel line: 82 kN
- between small accumulator body and cap: 100 kN

Accordingly the following results were obtained for the maximum principal stresses at a reference pressure of 2000bar and residual stresses of the most stressed areas:

- one piece accumulator (B1-B4):
    - max principal stress: 984MPa
    - max residual stress: -950MPa

- fuel line (all variants):
    - max principal stress: 217MPa
    - max residual stress: -294MPa

- small accumulator body (B2):
    - max principal stress: 474MPa
    - max residual stress: -600MPa

- T-pieces (A1-A4):
    - max principal stress: 661MPa
    - max residual stress: -950MPa

The results of biaxial analysis are listed in Table 5 below:

variant	biaxiality factors	qty.
one piece accumulator (B1-B4)	mean biaxiality ratio	0.25
	standard deviation	0.29
	non-proportionality-factor	<0.1
fuel line (all variants)	mean biaxiality ratio	0.03
	standard deviation	0.02
	non-proportionality-factor	<0.1
small accumulator body (B2)	mean biaxiality ratio	0.43
	standard deviation	0.3
	non-proportionality-factor	<0.1
T-piece (A1-A4)	autofrettage pressure [bar]	0.25
	wall plastification [%]	0.21
	maximum total strain [%]	<0.1

Table 5: list of calculated biaxial factors

From the results of biaxial analysis the following conclusions can be drawn:

For the one piece accumulator in B1 to B4 and the small accumulator body in variant B2 the loading is proportional and biaxial. The proper fatigue theory is the maximum shear stress criterion (Tresca).

For the fuel line the loading is proportional and uniaxial. The proper fatigue theory is the maximum principal stress criterion (Rankine). The final results for the lifetime assessments are summarised in Table 6.

Basically all investigated systems are fatigue resistant for cyclic loadings around the nominal operating pressure (NOP) with an adequate safety factor (SF). The safety factors for components of the jumper line system (A1 to A4) are found to have distinctly higher safety factors. These higher safety factors are provided by the circumstance that all parts except the accumulator integrated injector are loaded under the flat pressure conditions within the pipe line system.

The safety factors of the injectors where higher pressure pulsations appear under nominal operation conditions (Figure 4) remains to be investigated by taking into consideration of the specific features of the injectors.

		rail		fuel line		small accumulator		T-piece (connection block)	
		SF	Nf	SF	Nf	SF	Nf	SF	Nf
A1	NOP	-	-	1651.3	∞	-	-	418.56	∞
	MPA	-	-	4.37	∞	-	-	1.31	∞
A2	NOP	-	-	823.91	∞	-	-	207.54	∞
	MPA	-	-	4.3	∞	-	-	1.28	∞
A3	NOP	-	-	236.05	∞	-	-	59.95	∞
	MPA	-	-	4.36	∞	-	-	1.3	∞
A4	NOP	-	-	48.64	∞	-	-	12.38	∞
	MPA	-	-	4.24	∞	-	-	1.26	∞
B1	NOP	7.91	∞	13.12	∞	-	-	-	-
	MPA	0.89	563k	3.79	∞	-	-	-	-
B2	NOP	16.77	∞	44.67	∞	17.38	∞	-	-
	MPA	0.91	639k	4.21	∞	1.79	∞	-	-
B3	NOP	10.3	∞	22.98	∞	-	-	-	-
	MPA	0.9	581k	4.05	∞	-	-	-	-
B4	NOP	12.63	∞	27.64	∞	-	-	-	-
	MPA	0.89	563k	4.2	∞	-	-	-	-

Table 6: summary of lifetime assessment calculations

Abbreviations:

NOP: nominal operation pressure
MPA: maximum pressure amplitude
Nf : number of cycles to failure
SF: safety factor
A1: damped jumper line system with 500cm^3 accumulators
A2: damped jumper line system with 80cm^3 accumulators
A3: undamped jumper line system with 500cm^3 accumulators
A4: undamped jumper line system with 80cm^3 accumulators
B1: conventional rail system with one piece accumulator
B2: conventional rail system with one piece accumulator and additional accumulators
B3: conventional rail with optimised injection rates
B4: conventional rail with optimised water hammer effect

It is noteworthy that the safety factor falls by a factor of three to four hundred if the maximum pressure amplitude (MPA) is taken instead of the nominal operation pressure (NOP) for the calculation of the safety factor. It suggests that fatigue life that the relevance of the hydraulic characteristics diminishes when large numbers of start-up/shut-down cycles prevail. Under MPA conditions the relative difference in the safety factors between similar parts is much smaller.

The relative differences for the components in different variants are resulting from the different design boundary conditions applied. In particular the given design boundary conditions made it possible to design the T-piece in the jumper line system with a lower notch factor than the rail.

SUMMARY AND DISCUSSION

A systematic approach was presented for the identification of various opportunities for the design of the accumulator function of a CRS and for evaluating these design opportunities regarding space availability, assembly robustness and manufacturing costs, hydraulic performance and the mechanical high pressure capabilities. The evaluation has been demonstrated using an in-line model engine with 6 cylinders and a power output of 1750kW.

Two fundamentally different approaches for the design of the accumulator function have been assessed concerning the hydraulic characteristics and their impact upon fatigue life using a hydraulic simulation of the complete CRS under conditions of a nominal operating pressure of 2000bar. In a first approach the accumulator function has been almost completely integrated into the injectors of a modular common rail system. In a second approach the accumulator function has been integrated into the pipe line system between the high pressure pump and the injectors whereas the accumulator function is primarily provided by a manifold (common rail) that supplies all six cylinders with fuel and pressure. Further optimization of these layout variants, which is provided by the introduction of orifices or by the introduction of an additional small accumulator in front of the injectors in the second approach, has been simulated.

The hydraulic characteristics have been discussed with respect to pressure fluctuations within the injector as well as within the pipeline system, pressure drop within the injector, development of the injection discharge rate from one cylinder to the other, the variations of injected fuel amount as well as with respect to the variation of the injection discharge rate function between different cylinders. It was shown that the best combination with regards to a minimum drop of the injection discharge rate and a reproducible injection discharge rate function can be achieved by way of two fundamentally different approaches: (1) by a modular system with large integrated injectors, in which pressure pulsations within the jumper lines are dampened by orifices at the injector inlet and (2) by a non-modular common rail system, which is stabilized against pressure pulsations within the injector by means of additional single accumulators that are placed in front of the injector or integrated into the injectors.

These two most promising approaches fundamentally differ in terms of the damping role of orifices and of small accumulators. When large accumulator integrated injectors are used the introduction of orifices is necessary for a competitive reproducibility of the injection discharge rate function. In addition to that a comparatively large volume has to be integrated in each injector to avoid a disadvantageous drop of the injection discharge rate over one cycle. If the volume in the injector in the first approach reaches the volume of a common-rail in the second variant a slight but visible advantage of variant (1) over variant (2) with respect to the reproducibility of the injection discharge rate function from one cylinder to the other is found.

The achievable durability of a pipe line (common-rail) system under nominal (steady) operation conditions (NOP) sensitively depends upon the pressure fluctuations and the stress concentrations at bore intersections. Accordingly all measures that improve the hydraulic performance of the CRS tend to increase the high pressure capability of the pipe line (common-rail) system. This can be regarded as a potential advantage of a modular system with large integrated accumulators, which have shown lowest pressure pulsations in the pipe line system. However, the volume (capacity) of such accumulator integrated injectors has to be comparatively large to avoid large pressure drops during injection that are driving the fatigue problem within the injector. Finally, the relevance of the hydraulic characteristics for the achievable high pressure capability starts to diminish as soon as the fatigue life is governed by a high number of start-up/shut-down cycles.

When it comes to a decision between the above mentioned variants (1) and (2) it is worth mentioning that variant (2) can also be built up in a modular way according to Figure 1e. From a hydraulic point of view there is no fundamental difference between a one piece accumulator and an accumulator that is built up with several jumper lines, which are connected by connection blocks. From an economic point of view this decision is to be made dependent upon how efficient the individual parts can be manufactured. The modular system is competitive if the higher number of parts can be efficiently manufactured using good scale economies. The cost competitiveness of the one piece accumulator depends on the availability of appropriate equipment and tooling for gun drilling, autofrettage and electrochemical machining (ECM).

The modular CRS that can be assembled on the engine is characterized by a higher number of sealing connections. Care has to be taken that sealing surfaces are not damaged during assembly as they are the most likely source for leakages that are not recognized until the complete assembly is pressurized. In view of this assembly risk, thin fuel pipes that can be mounted with more flexibility and low applied torques are predominantly preferred for modular systems.

The advantage of the non-modular (double-walled) system based on a one piece accumulator is that it can be more efficiently pre-assembled under controlled conditions with appropriate assembly aids and tested with respect to the tightness of the high and low pressure system before mounting on the engine.

Robust and safe disassembly and assembly is particularly required for maintenance works. In addition to that enhanced contamination risks appear outside isolated assembly rooms during field applications. Here too, a minimum effort for disassembly / assembly during maintenance works is preferred. In a common rail design based on a one piece accumulator it is sufficient to disassemble only the parts belonging to one and the same injector when this particular injector needs to be removed.

The modular CRS with a moderately large volume in the injector may turn to become disadvantageous compared to a non-modular CRS when the volume in the pipe line system is to be further increased. The thicker pipe lines that include multiple bends are more costly to manufacture, have to be sealed at higher torques and bear additional risks for damaging of the sealing surfaces during assembly.

It may appear that at this point of time there is no fundamental reason to rule out modular or non-modular common rail systems, systems based on accumulator integrated injector, and systems in which the accumulator function is optimised within the pipe line system between high pressure pump and injectors. The modular as well as the non-modular approach bear their own optimisation possibilities with respect to a competitive cost-benefit balance. Each variant needs to be critically evaluated regarding to design opportunities provided by the engine layout conditions, desired hydraulic characteristics, manufacturing capabilities, assembly and maintenance requirements as well as concerning the product life cycle costs before any rational decision can be made.

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