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New Common Rail Systems suited for Diesel Engines from 1 to 5 Megawatt: modeling simulations and hardware results

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Abstract: The compressibility of the fuel and the effects of pressure wave dynamics govern the injection in a Common Rail System (CRS). The layout of the system volume and its geometrical distribution within the system therefore strongly affect the injection events at every injector. Based on the simulation results achieved when modeling the CRS by means of the hydraulics simulation tool AMESim, new common rail injectors and high pressure pumps were designed, built and tested. They represent the key to novel systems which are best suited for Diesel Engines with an overall power output from 1 to 5 MW, as used in locomotives, ships, power generation and heavy earth-moving machinery. The injectors have an accumulator chamber within an injector body which easily fits into the engine's cylinder head. The actuating components

are all placed in the front part, leading to a compact design. The injection rates of the injector demonstrate its good performance with or without multiple injections. The injection law is compared to that of a former system equipped with rails without an accumulator in the injector. A high pressure pump with a sinusoidal eccentric shaft and a sliding shoe between the shaft and the base of each plunger delivers the pressurized fuel. To obtain a long operating life, special emphasis has been given to the lubrication between the plunger base and the shoe. The pressure in the working chamber of each plunger is transmitted to its base by means of a small piston. The new pump for the above mentioned Common Rail Systems is of very simple design. The performance results demonstrate its excellent hydraulic efficiency.

INTRODUCTION

Ten years ago, in fall 1997, the Common Rail System (CRS) was introduced on the passenger car (PC) diesel engines market [1]. At the same time the CRS was introduced by MTU Motoren- und Turbinen Union Friedrichshafen on their 4000 Series engines, a family with a power range from 800 to 2700 kW [2]. The manufacturer of the MTU system is its subsidiary L'Orange [3]. In this decade, the CRS has overtaken all other fuel injection systems in the passenger car domains and has replaced the former inline or rotary jerk type systems, and now also the unit injector. In the On- and Off-Road commercial engine domains a similar trend is taking place, even if many systems are still of the jerk type. The main reason for this slower trend is the fact that many of these engines, especially with a power output above 1 MW, have a lifespan of two decades or more. Thus, the CRS must cope with the longer life and higher reliability required from the fuel system. The technology of PC-CRS cannot just be scaled up to make a good system for engines with higher power and longer life. Substantial development work is needed.

The CRS mentioned above have one or more fuel rail(s). The rail represents a quiet pressurized region to create a condition of almost equal pressure for all injectors and injections. For engines of 1 MW or more, the rail is bulky and a big cost factor.

A Deutz 628 8 cylinder inline engine with a cylinder volume of 12.5 Litres, a power of 2 MW and a max speed of 1050 RPM has been equipped with a CRS with two rails, each one serving 4 cylinders, Figure 1 [4]. The bulkiness of the two rails is also due to the fact that they must have a double wall for safety reasons.



Figure 1: Common Rail System for the Deutz 628 8 cylinder inline engine [4]

The first and second generation passenger car CRS had and in part still have a static injector leakage due to a low pressure region in the injector between the nozzle needle and control piston guides. The two guides consist of a tight sliding fit with a clearance of 1-3 microns. The leakage occurs across the guides from the control chamber and from the nozzle chamber into the low pressure region [3, 5]. This condition is causing heat and friction and is one main life-limiting factor of the injector.

Work has been done in the PC and in the Off-Road domains to eliminate the static leakage [6, 7]. In many instances this is a difficult task due to the limited space available for the injector in the engine cylinder head.

Common Rail high pressure pumps for engines from 1 to 5 MW require new developments as well. The available pumps are of complex design. The slow, sinusoidal pumping characteristic of CRS pumps poses the challenge of ensuring reliable lubrication between the elements which transform the rotation of the eccentric shaft in a linear movement of the pumping piston.

The aim of this paper is to present new technological solutions which eliminate the above mentioned shortcomings of CRS in the Off-Road engine power domain between 1 and 5 MW.

INJECTOR WITH ACCUMULATOR

To obtain a CRS without rails, it is necessary to place an accumulator volume within the injector body.

Figure 2 shows the initial injector design for the engine of reference [4].

This injector has zero static leakage. The one piece injector needle is long. Its top end extends above the high pressure inlet connector, which connects sideways by means of a threaded inlet stud. At the upper end of the needle the hydraulic control device with intermediate valve [3] and the control solenoid are placed. The intermediate valve ensures a very rapid and clean injection end. Only a very small quantity of fuel is released to control the needle movement. A threaded screw plug closes off the high pressure region and separates this from the solenoid actuator and its valve. A relatively long upper injector part has the function of bridging the space under the cylinder head cover and allows for the solenoid wires to exit on top of the upper injector part outside the cylinder head cover.

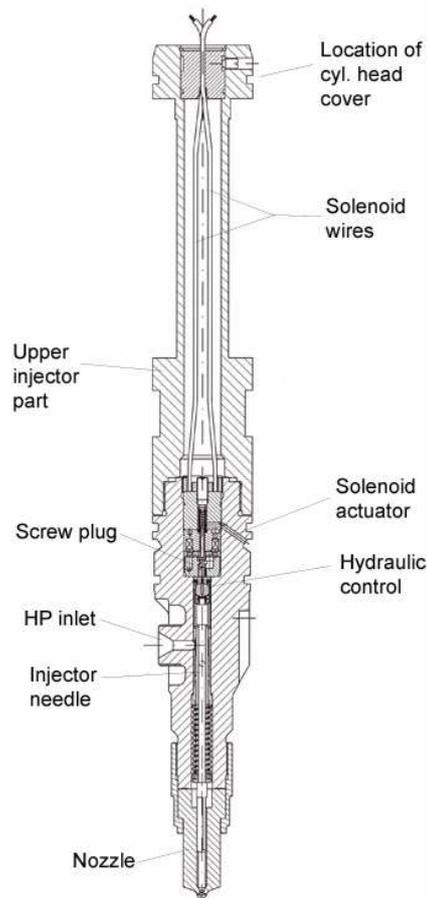


Figure 2: Initial Common Rail Injector design for the Deutz 628 engine [4].

Due to the sidewise high pressure inlet stud and the long needle, there is no space to place an injector's accumulator. This design is not practical for a system without rails.

The following was undertaken to resolve this situation:

- reduce the needle length to a minimum, so that the hydraulic control device and the control solenoid can be placed underneath the sidewise high pressure connector, and
- utilize the space of the upper injector part to place an accumulator chamber within the injector body.

The new injector design is shown in Figure 3.

The needle valve is very short and can be placed, together with a needle guide sleeve and the hydraulic control device, within the injector's nozzle body. The end piece of the hydraulic control device rests at its upper face against a delimiting spacer plate. The needle spring transmits its upper force via the guide sleeve to the end piece.

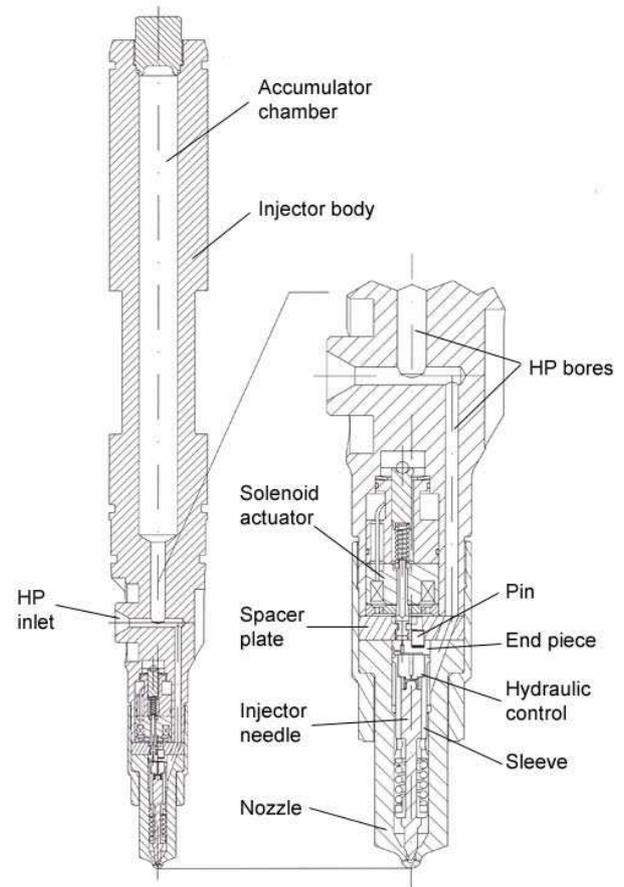


Figure 3: New Common Rail Injector with accumulator.

Due to this and the fuel pressure, the end piece seals together with the spacer plate. Therefore also this injector has zero static leakage. All functional components of the hydraulic control device can be easily assembled.

The solenoid valve is placed sidewise in a bore of the injector body which is one work piece including the accumulator. The end piece must be oriented in order to align its discharge bore with the axis of the solenoid valve. This is achieved by a positioning pin resting in an inner recess of the end piece and in a corresponding recess of the spacer plate.

A high pressure bore connects the nozzle across the spacer plate with the injector's fuel inlet. A bore on the body axis connects the fuel inlet with the injector's accumulator.

THE SYSTEM WITHOUT RAILS

Figure 4 is a 3D-view of a CRS without rails incorporating injectors as shown in Figure 3 and suitable for engines with a power of 1 to 5 MW.

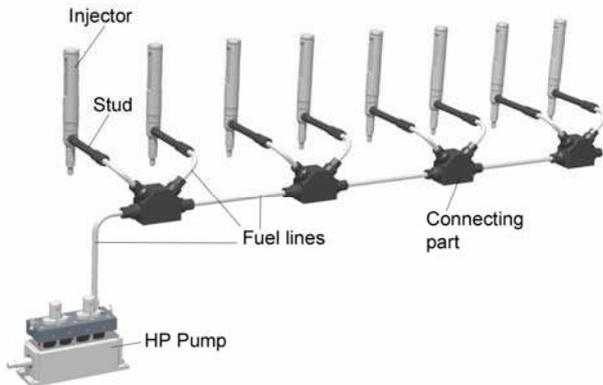


Figure 4: Common Rail System without rails for Diesel Engines with a power of 1 to 5 MW.

The injectors with individual accumulators are interconnected by means of inlet studs to each injector and relatively thin fuel lines, where two fuel lines departing from the studs end into a connecting part. The connecting part totally has four connecting locations. The other two are used to interconnect the groups of two injectors, one end location is receiving the pump delivery, while the other has a blind plug.

A CRS as shown in Figure 4 is simpler than a system with rails like Figure 1. If the outline of the existing conventional injector used on a corresponding engine can be maintained, it becomes of very flexible use on new or used engines (as a retrofit), because the installation does not require modifications to the engine's cylinder head. In this case, for most applications, the injector's accumulator must have a volume of only 20 to about 50 times the maximum quantity injected per stroke.

With this small accumulator volume the pressure drop in the accumulator during the injection event is big and is the cause of large transient pressure waves in the fuel lines, which normally impair the correct function of the CRS.

However, when the HP inlet of every injector is connected to the system fuel lines by means of a wave dynamics and dampening system (WDD) as presented in Reference [8], the above mentioned shortcomings can be overcome. The WDD consists of a check valve opening in the direction

of every injector's accumulator, paralleled by a throttling orifice. It provides a unidirectional, almost unhindered flow in the direction of the actually injecting injector and prevents at the same time the detrimental propagation of the pressure wave due to the hydraulic hammer at the needle valve closing of any injection into the remaining part of the system. The WDD is a very effective means to optimize the function of CRS with a very small overall HP volume.

The location to build-in the wave dynamics and dampening feature can be selected. For example this can be right at the injector HP inlet. Alternatively anywhere within each inlet stud or even within each connecting part. However, one WDD per injector must be provided.

The CRS with 2 rails of Figure 1 has a total HP volume of 1820 cm³ while the CRS without rails of Figure 4 has a HP volume of 820 cm³ [8]. The individual accumulators of this system have a volume of 70 cm³, which is 35 times the maximum quantity injected (2000 mm³ per stroke).

Figure 5 shows the pressure histories in the injector's accumulators and in the nozzle sacs of the system without rails, obtained with the same AMESim simulation model as described in Reference [8].

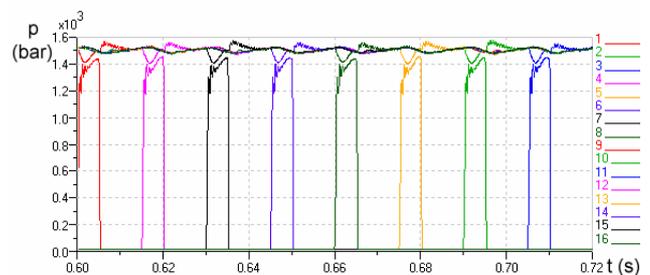


Figure 5: Pressure histories in the accumulator and in the nozzle sac of the 8 injectors of the system without rails obtained with the AMESim simulation model [8] for the simulation time 0.60 to 0.72 seconds, corresponding to 2 engine revolutions with 8 injection events.

The tracings 1 to 8 of Figure 5 are the pressure histories in the individual injector's accumulators, while the tracings 9 thru 16 represent the injection pressure in the nozzle sac of each injector.

During each injection, the pressure in the corresponding injector's accumulator drops about 100 bar from 1500 to 1400 bar and subsequently recovers during the injection event and reaches the initial 1500 bar pressure right before the end

of injection. This pressure pattern in each nozzle sac will generate a beneficial, continuously increasing injection rate shape from the beginning up to the end of the injection event.

Since the hydraulic hammer at the end of every injection is contained within the injector's accumulators and released to the system's fuel lines only by means of the injector's throttle, it does not harm and all injectors of the system have the same conditions of operating pressure at the right point in time.

MEASURED INJECTOR PERFORMANCE

The performance of realized injectors is measured on an injector test bench with calibration fluid. The measuring apparatus is an EMI3000 [9], which measures the injection rate shape and the integral thereof, the injected quantity per stroke.

Figure 6 shows the injection rate shapes of an injector according to the design of Figure 2 at the system pressure 1500 bar. Five different energize pulses to the injector's solenoid actuator with a duration of 1, 2, 3, 4 and 5 ms are used. The injection events are superimposed and their duration rises according to the increasing energize times.

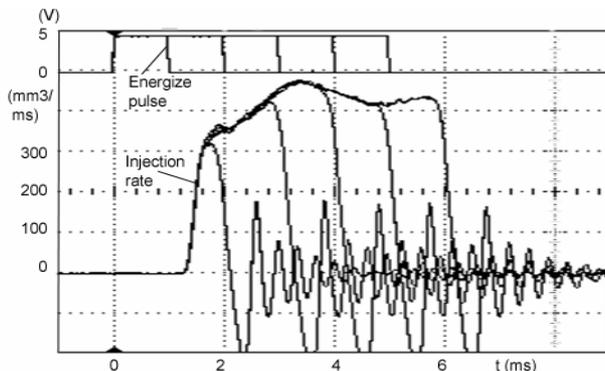


Figure 6: Injection rate shapes of an injector according to the design of Figure 2 at the system pressure of 1500 bar for 5 different energize times of the solenoid actuator.

The rate shape has a maximum 2 ms after injection begin, then drops and recovers to some extent for the longest duration of 5 ms, corresponding to a full load injection duration. This kind of rate shape is known [3] from heavy duty CRS with rails. It is due to the pressure pulsation generated in the initial part of every injection event, which propagates from the

injector's nozzle upstream in the fuel pipe until it reaches the rail. From the rail the inflow in the fuel pipe at recovered pressure reaches the nozzle before the injection event is completed, because for practical reasons the length of the fuel pipe is short.

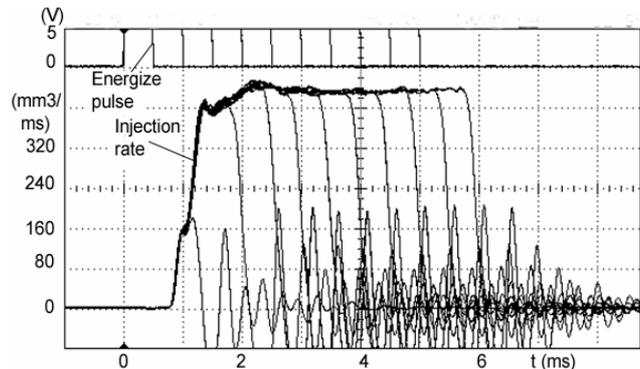


Figure 7: Injection rate shapes of an injector according to the design of Figure 3 with an injector's accumulator at the systems pressure of 1500 bar for 10 different energize times of the solenoid actuator.

Figure 7 shows the measured, superimposed injection rate shapes of an injector according to Figure 3 with an injector's accumulator at the systems pressure of 1500 bar for 10 different energize times of the solenoid actuator, ranging from 0.5 ms up to 5,0 ms.

The injection rate shape is almost square and does not show the oscillation of Figure 6. Due to the intermediate valve, the end of injection is very fast.

Figures 6 and 7 show an oscillation of the injection rate below the state of zero flow after the end of each injection event, which swings for some time. This is due to the EMI3000 apparatus, which uses a piston displacement to determine the injected volume. The piston swings a few times after the end of an injection because it cannot be abruptly stopped.

For the injector according to Figure 3, Figure 8 shows the injected volumes V_I in mm³ per stroke as a function of the solenoid energize time and for the selected pressures of 500, 700, 1000, 1300 and 1500 bar. At every pressure and energize time a good control of the delivered fuel is realized, even for small injected volumes under 100 mm³ per stroke, which are used during pilot and/or post injections.

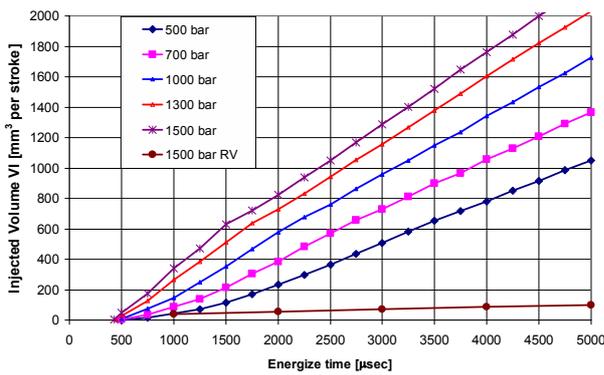


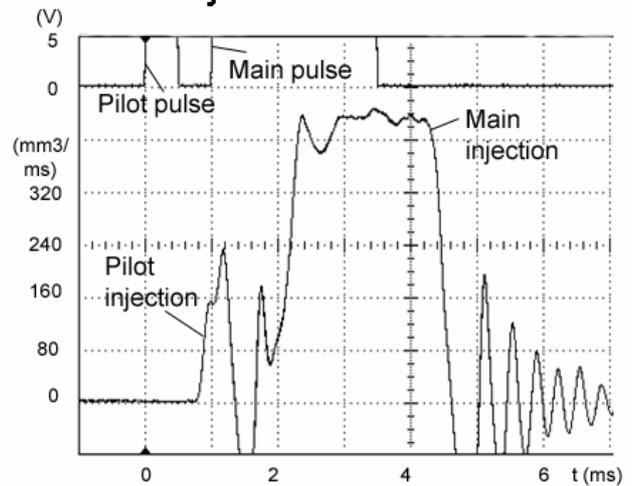
Figure 8: Map of injected volumes V_I in mm^3 per stroke as a function of the solenoid energize time and the system pressure (parameter) for the injector with accumulator, as well as returned volume RV in mm^3 per stroke.

In Figure 8, for the pressure of 1500 bar is also plotted the return volume RV . This is the volume spilled back from the injector to the return line at no pressure during every injection event. Since the injector has zero static leakage, this is the only parasitic loss of the injector. For a full load injection of 2000 mm^3 per stroke it amounts to 100 mm^3 or 5% of the injected fuel. This means that 95% of the fuel delivered from the HP pump into the system is injected into the engine's combustion chambers.

Figure 9 shows tracings of the injection rate with a pilot and a main injection on the upper picture as well as with a pilot-, a main and a post-injection on the lower picture. The time scale of these oscilloscope tracings is 1 ms per division.

Due to the fast end of the single injection events, the EMI3000 measuring apparatus shows again oscillations which do not correspond to the actual delivery rate at the injector's nozzle and thus the measured signals are not fully representative. Even so it can be seen that the separation time between the pilot and the main, as well as between the main and the post injection is very short and amounts to 0.2-0.3 ms. Longer separation times can be realized as well by using a longer separation between the solenoid energize pulses.

Pilot injection



Pilot and post-injection

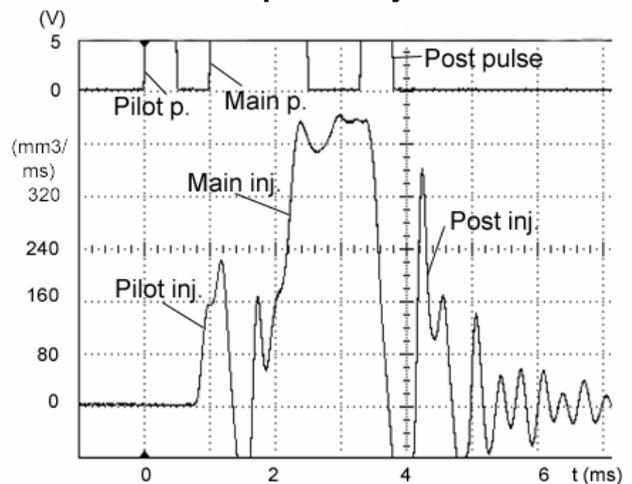


Figure 9: Rate shapes of the injector according to the design of Figure 3 for the system pressure 1500 bar; pilot and main injection as well as pilot-, main- and post-injection.

COMMON RAIL HP-PUMP

The HP-Pump is a four cylinder pump with a harmonic, sinusoidal movement of the plungers, each spaced apart 90 degrees pump shaft rotation in order to obtain an equal distribution of the individual pumping strokes.

Figure 10 shows the CRS pump and its main functional data. The effective fuel delivery is controlled at the inlet by means of an inlet throttle valve. Therefore the HP-pump only brings the fuel quantity needed from the injectors at a given engine running condition from low to high pressure, minimizing the power demand.

The design of the plunger drive mechanism is shown in Figure 11. This mechanism consists of a sliding shoe placed between the shaft eccentric and the base of each plunger, similarly to passenger car CRS pumps. It converts the rotation of the pump shaft into a linear movement of the plungers. To achieve a long life, oil lubrication has been selected in connection with plain bearings.

The rotation direction of the eccentric shaft is clockwise. Figure 11 shows an eccentric position where the shoe and the plunger are in the middle between the bottom and the top dead centers. The shoe is at its maximum left position.



Fuel delivery	12 Liters/Min.@2500 RPM
Pressure	0-1600 bar
Dimensions	L40 x B15 x H28 cm
Weight	35 Kg

Figure 10: Picture of four cylinder CRS HP-Pump and main functional data.

In the position shown, the lubrication of the contact surface between the plunger base and the sliding surface of the shoe is critical, because the relative horizontal movement component is zero. In most operating cases in this position the pump is delivering fuel at high pressure and the pumping chamber pressure is high. The high force resulting from the pressure in the pumping chamber is transmitted to the plunger base.

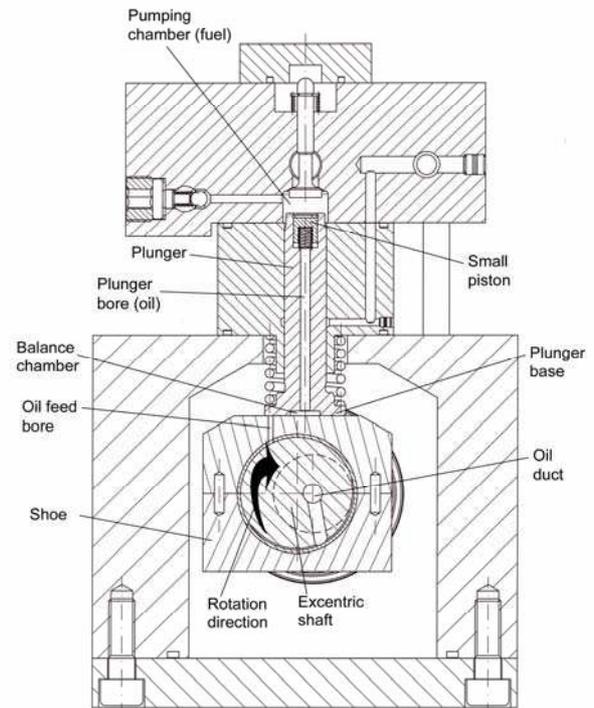


Figure 11: Schematic sectional drawing of the CRS pump with eccentric drive shaft and shoe between the plunger base and the eccentric.

When rotation of the shaft continues, the shoe moves and a relative speed arises between the plunger base and the shoe. With previous contact of the parts with no oil there in between, the parts are prone to seizures which impair the function of the pump.

To overcome this, the plunger has been equipped with a small internal piston. A plunger bore connects the underside of the small piston with a balance chamber between the plunger base and the shoe. During the entire pumping stroke, the balance chamber and the plunger bore form a closed volume filled with oil. The small piston thus transmits the fuel pumping chamber pressure to the oil trapped in the plunger bore and in the balance chamber, creating a pressure balance. The oil in the balance chamber unloads the contact surface and at the same time is a source of high pressure oil which lubricates this surface. Critical tribologic situations are therewith avoided. During the suction stroke an oil feed bore connects to the balance chamber and refills this from any oil losses during the pumping stroke [10].

The effect of the fuelling characteristic of the presented HP Pump in a CRS with a reduced overall HP volume such as the system of Figure 4

has been investigated using the AMESim simulation model of Reference [8]. If dynamic pressure waves arise due to the pumping strokes of the pump, the most critical location is the inlet at the beginning of the individual fuel lines leading to each injector, see Fig. 4, because a fluctuating pressure at this location will influence the injected quantity at the injectors.

To isolate the effects caused by the injections from those caused by the HP Pump, the injections were switched off in the model, so that no fuel is injected. The results are shown in Figure 12.

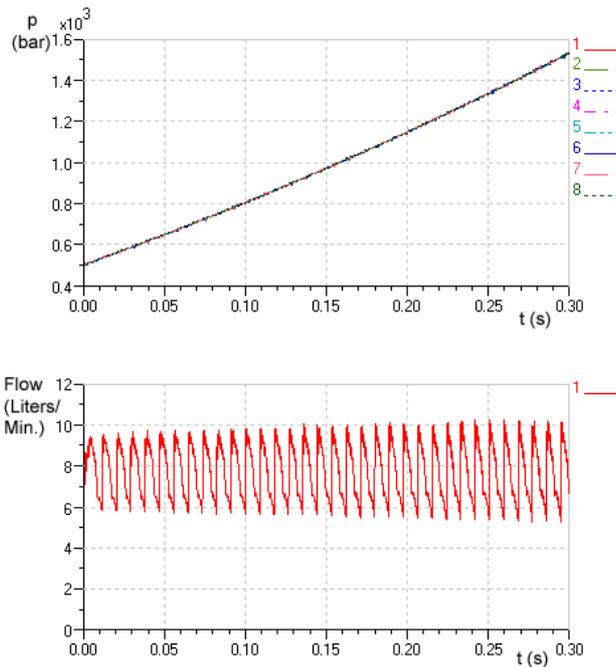


Figure 12: Pressure histories with injectors switched off at the beginning of the 8 fuel lines of the system according to Figure 4, obtained with the AMESim simulation model of Reference [8], together with the history of the HP fuelling at the pump outlet.

The high pressure fuelling into the system (bottom picture of Figure 12) fluctuates between 5 and 10 Liters/Min., this means up to 50%. The higher fluctuations in the time between 0.20 and 0.30 s is due to the higher outlet pressure and the resulting stiffer bulk modulus of the fuel.

These relatively high fluctuations of the pump fuel delivery do not cause noticeable pressure waves in the overall HP volume, as shown on the top picture of Figure 12. At the beginning of simulation, the pressure in the system has been set to 500 bar. This pressure gradually increases at the inlet to the individual injector's fuel lines due to the delivery of the pump and the absence of

injection events. The 8 tracings are practically super imposed; the effects of pressure waves are negligible.

Figure 13 shows the measured fuel delivery of the pump as a function of speed and pump HP outlet pressure with the fully open inlet throttle. The theoretical delivery is plotted as well.

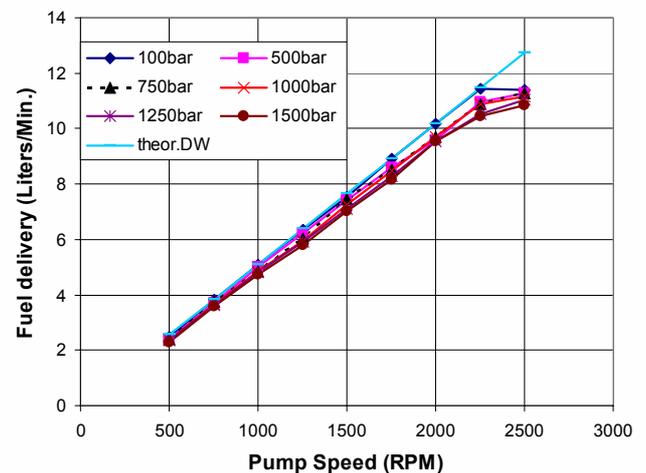


Figure 13: Maximum fuel delivery of the HP pump as a function of speed and delivery pressure.

The effective fuel deliveries are almost independent from the HP outlet condition. Even at low speed between 500 and 1000 RPM, there is only a small reduction between the theoretical fuel delivery and the effective delivery at 1500 bar.

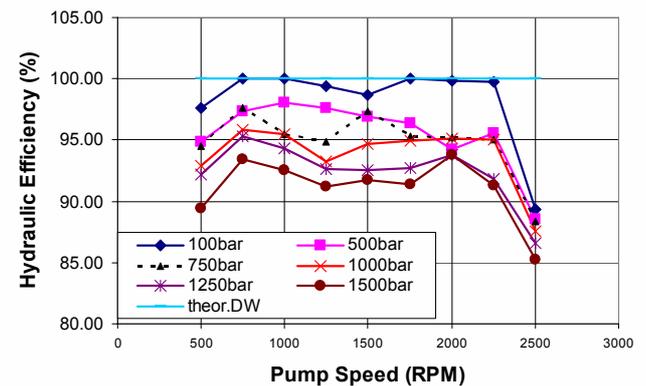


Figure 14: Hydraulic efficiency of the HP pump.

In Figure 14 the hydraulic pump efficiency is plotted as a function of speed and pressure. At pressures between 500 and 1500 bar, the efficiency is between 90% and 95%, independently of the selected outlet pressure.

Between 2250 and 2500 RPM, independently from the outlet pressure, the delivery stops

increasing due to the inlet throttle which limits the inlet fuel to about 11 Liters per minute. The pump was laid out for the engine of Reference [4], requiring 8.8 Liters per minute of fuel injected at full load.

CONCLUSIONS

A Common Rail System for Heavy Duty Diesel engines in the power range of 1 to 5 MW has been developed.

The injectors of this system have zero static leakage and individual accumulators. The control elements including the solenoid, the hydraulic needle control and the needle valve are all placed in the front injector part, leading to a very compact design.

The use of a new device providing a one-way flow in the direction of any injector that is injecting and dampening pressure waves in the opposite direction enables the use of a system without rails and with simple, relatively thin connection fuel lines between the injectors and the HP Pump.

A continuously rising injection pressure is achieved during the injection event. The injector performance leads to properly injecting 95% of the fuel supplied to the injectors. Pilot and post injections with very short intervals can be realized.

The 4 plunger Common Rail Pump delivers fuel at high pressure with an efficiency of 90 to 95%. A pressure balance piston allows unloading the contacting surface between the plunger base and the shoe when a high pressure is present in the pumping chamber. The critical lubricating condition is therewith avoided. The pump design, similarly to passenger car pumps, is very simple.

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