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Investigating the Fuel Type Influence on Diesel CR Pump Performance

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Abstract

The paper presents an experimental investigation on a three-plunger radial pump for diesel common rail injection systems.
The effort is aimed at highlighting the influence on the pump torque demand due to the use of an unconventional fluid, focusing the attention on a pump model typically adopted in the field of light duty diesel engines.
Pump performances have been measured on a fuel injection system test-bed. A relatively wide operation range of the pump has been considered, in terms of fuel delivery pressure and pump shaft regime, providing torque demand at each operating point. Once the experimental characterization has been completed for the standard fuel (ISO 4113 reference fuel), a Waste Cooking Oil (WCO) biodiesel fuel has been tested, at the same pump operation conditions. In the present study, the influence of such alternative fuel on pump operation has been investigated. After the description of the experimental set-up, results have been presented and discussed, highlighting the influence of the fuel type in terms of torque demand.

Keywords: Common-Rail high-pressure pump, fuel type, WCO biodiesel, torque demand, experimental characterization

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1. Introduction

Several factors influencing the engine performance are tied to the features of the fluids used as fuel or as lubricant. Several research topics consider the interaction among the fluid properties (fuels and lubricants) and the engine performance, such as the pollutant formation during combustion, the chemical-physical compatibility among fluids and engine materials, and the influence on the performance on ancillary components.

In the field of diesel engines, high pressure CR pumps are the most power-demanding units among the ancillary components; it means that even the small percentage-alterations of a CR pump performance are reflected on the engine efficiency, appreciably. Although many research activities have been devoted to the development of CR injectors and nozzles [1-6], very few contributions in the literature are devoted to the research on the high-pressure CR pump performances [7,8]; a fundamental contribution is reported in [9], consisting in a thorough experimental-theoretical study on radial piston CR pumps; here the two most common types of CR pump layouts have been analyzed and compared, also highlighting the impact of the electro-mechanical devices used to control the rail pressure level, namely the Pressure Control Valve (PCV) and the Fuel Metering Valve (FMV). The pump head-capacity curves have been determined at different revolution speeds and they have been compared with the injector flow requirements, evaluating also the efficiency of the pressure control strategy. Ageing effects, dynamic pump behavior and mechanical-hydraulic efficiency have been assessed as a function of head and speed. All the tests have been referred to the ISO 4113 standard calibration fluid.

Concerning the influence of fuel type or properties, some effort has been devoted investigating the injection phasing [10]. Such a topic is particularly important when fully mechanical injection systems are used (e.g. inline pump systems for industrial applications) [11]; indeed, injection phase directly depends on the pressure wave propagation within the high pressure pipe and on the pressure dynamics within the drift chambers of the system components, namely pump, delivery valve and injector.

In the case of CR injection systems, the actual impact of fuel type or properties on injection phasing still exists, but it influences in a minor way the injection strategies, as the timing of injection events does not depend on the mechanical phasing (e.g. cam and follower).

Research activities on fuel type effects on diesel CR pump performance still have to be reported in the literature; in the present paper, the results of experimental activities on such a topic are presented and discussed. The investigations have been oriented to assess the influence of a biodiesel fuel on pump mechanical performance, in terms of torque demand for different load conditions.

As it is described in the following section, two fluids have been considered; the ISO 4113 calibration fluid and a biodiesel fluid, belonging to the Waste Cooking Oil (WCO) category [12-14].

Nomenclature

CR	Common Rail
PWM	Pulse Width Modulation
RPM	Revolution Per Minute
WCO	Waste Cooking Oil
η_{mh}	Mechanical-hydraulic efficiency
ξ	Pumping work reduction factor
Δp	Pressure difference
iV_{displ}	Pump capacity
λ_V	Induction coefficient
C	Measured torque
C_{th}	Theoretical torque

ΔC	Torque difference
$\bar{\rho}$	Average fluid density
ρ_{lp}	Fluid density at low pressure port of the pump

2. Materials and Methods

2.1. High pressure pump under test

Experiments have been carried out on a commercial three-plunger radial pump. In this unit, each plunger is provided with automatic spring-loaded intake and delivery valves. As a volumetric machine, the delivered flow rate ideally depends on its capacity and shaft speed. In the adopted setup, delivery pressure level, which is actually the pressure level in the rail, is imposed by a Pressure Control Valve (PCV) located in the rail, only. The valve is used to throttle a calibrated discharge orifice, whose passage section is variable according to the pressure target. The PCV solenoid is driven under PWM logic in a closed-loop control strategy; when the PCV is not energized, the orifice throttling depends just on a closure spring preload, whose effect is set to be relatively weak, so that rail pressure level is in the range of a hundred bar.

The pump body is provided with three hydraulic connections, namely the feeding, the recirculation and the high pressure ports; a low pressure feeding system impels the fluid from the reservoir to the feeding port at regulated pressure and temperature. The high pressure port is connected to the rail. In the current set-up, injectors have been disconnected and their ports on the rail have been plugged. The main pump specifications are reported in Table 1.

Table 1. High pressure pump specification.

Pump specification	
Pump model	Bosch CP1
Pump displacement	0,69 cc/rev
Architecture	3-plunger
Nominal max pressure	1400 bar
Typical speed range	500-1900 RPM

2.2. Test bed description and control strategy

The pump test bed belongs to a complete injection system test bed. The pump is operated by an electrical drive embedded in a dynamometer. The high pressure fuel injection pump is clamped to the test bed frame and the pump shaft is connected to the dynamometer rotor of an electrical drive, whose frame is suspended on a couple of high precision bearings. The torque exerted at pump shaft is obtained by the axial force exerted on the load cell at the torque-arm. The force signal is conditioned, amplified and averaged on several pump cycles.

The actual shaft speed is optically encoded and controlled under a closed-loop algorithm, so that it is independent on pump load.

A two-stage electronic unit is used to generate the PWM duty cycle used in the control of PCV solenoid; a logic electronic unit generates the PWM signal at 1 kHz frequency, according to the target pressure in the rail volume. An electric power unit provides the necessary current to drive the solenoid. The rail pressure is measured by means of a piezo-resistive transducer; the signal is conditioned, amplified and averaged on several pump cycles and used as the actual value in the pressure-control closed-loop algorithm.

A low pressure feeding system is used to provide the fuel supply to the high pressure pump, according to the recommendations of the pump producer. The same system is also used to control the fluid temperature at constant level.

2.3. Fluid properties

As mentioned in the introduction section, the adopted calibration fluid satisfies the ISO 4113 standard. The alternative tested fluid is represented by a Waste Cooking Oil (WCO) biodiesel. The adopted biodiesel is compliant to the EN-14214 regulation and it is produced by trans-esterification and distillation processes of waste cooking oil. Table 2 reports the available characteristics of the two considered fluids.

Table 2. Specification of fluids.

WCO biodiesel specification at 40°C	
Kinematic viscosity	4,1 (cSt)
Density	843 (kg/m^3)
ISO 4113 calibration fluid at 40°C	
Kinematic viscosity	2,8
Density	773 (kg/m^3)

2.4. Test cases

Different operating regimes of the pump have been considered, namely 600, 1300 and 1800 RPM; pressure levels have been swiped from 600 to 1400 bar with a hundred bar steps. The following Table 3 resumes the adopted test condition in a matrix, aimed at the assessment of the pump torque request. Fluid temperature at pump inlet has been stabilized at 40°C.

Table 3. Test matrix.

Pump speed (rpm)	Rail Pressure (bar)								
600	600	700	800	900	1000	1100	1200	1300	1400
1300	600	700	800	900	1000	1100	1200	1300	1400
1800	600	700	800	900	1000	1100	1200	1300	1400

3. Results

The results related to the experimental campaigns are organized in two paragraphs; the first of them is dedicated to the experimental set-up assessment using the ISO4113 standard fluid, while the second one shows the comparisons, in terms of torque demand, between the cases related to the standard fluid and the cases related to the alternative fluid.

3.1. Experimental set-up assessment

In the first instance, the pump behavior operating with the calibration fluid is compared with what is reported in the literature. Indeed, in [9] the mechanical-hydraulic performance trends related to the same pump model have been reported, operating with the ISO 4113 calibration fluid under comparable test conditions. According to [9], the hydraulic efficiency of the pump is defined as

$$\eta_{mh} = \frac{\xi \lambda_V \Delta p i V_{displ}}{2\pi C} \tag{1}$$

in which $\lambda_V = \dfrac{m_{suc}}{\rho_{lp} i V_{displ}}$ and $\xi = \rho_{lp} / \bar{\rho}$. This coefficient takes the reduction in the pumping work caused by the fuel compressibility into account. The theoretical pump-required torque is equal to

$$C_{th} = \frac{\xi \lambda_V \Delta p i V_{displ}}{2\pi} \tag{2}$$

On the basis of Eq. (2), the mechanical-hydraulic efficiency given by Eq. (1) can be expressed as a torque ratio

$$\eta_{mh} = \frac{C_{th}}{C} \tag{3}$$

Expressing C as $C_{th} + \Delta C$ in Eq. (3) and taking Eq. (2) into account, the mechanical-hydraulic efficiency can be rewritten as

$$\eta_{mh} = \frac{1}{1 + \dfrac{\Delta C}{\left(\dfrac{\xi \lambda_V}{2\pi}\right) \Delta p i V_{displ}}} , \tag{4}$$

where the term ΔC is due to the friction forces between the mobile and stationary parts and to the fluid viscosity. Once the torque demand at pump shaft is measured, to express the mechanical-hydraulic efficiency of the pump it is necessary to determine the theoretical torque required in each of the operating points considered. In other words, it is necessary to determine the coefficients ξ and λ_V . The first coefficient is determined on the basis of the characteristics of the working fluid; indeed, in the case of the well-known ISO 4113 calibration fluid, it is possible to refer to the literature or to a valid data base; in the current case the coefficient has been evaluated on the basis of the libraries and the models available within the Amesim software [15]. As concerns the induction coefficient λ_V , it is possible to refer to [9], where its trend versus rail pressure is available. After the theoretical torque has been determined at the investigated working points of the pump, it is possible to trace the trends of mechanical-hydraulic efficiency. If the these trends are found in accordance with those reported in [9], they allow to conclude that the measured torque values in the case of the ISO 4113 fluid are reliable.

Figure 1 (Left) shows the trends of the induction coefficient vs. delivery pressure as reported in [9], for three different rotation speeds of the pump shaft. These trends have been here used in the calculation of mechanical-hydraulic performance; concerning the availability of the values relating to the cases at 1300 and 600 rpm, it should be noted that these have been extrapolated from [9] and then reported in Figure 1 (Right). In Figure 2 (Left), the trends of mechanical-hydraulic efficiency vs. delivery pressure have been reported for different pump shaft speeds; these values are originally shown [9] in a normalized form with respect to the maximum measured efficiency (0,93). Here, they have been reported to the standard form. The trends of experimental mechanical-hydraulic efficiency are shown in Figure 2 (Right); from a qualitative point of view, they show an evident dependence on delivery pressure, with a relatively marked downward concavity; with reference to the Eq.4 and to the trends reported in the literature, it is possible to assess the capability of the current experimental set-up in catching the mechanical-hydraulic behavior of the pump, namely the terms C_{th} and ΔC; concerning the dependence on the rotation speed, the trends found in the literature highlight the fact that at intermediate speeds (e.g. 1000 rpm) a higher mechanical efficiency is obtained compared to the cases with minimum and maximum speed. The current experimentation has evidenced this feature, even if the registered dependence is less evident below 900 bar. Referring to the literature data closest to the cases of interest for the current experimentation, Figure 3 (Left) highlights the good agreement between the low-speed cases (600 rpm and 500 rpm in the literature), whereas Figure 3 (Right) reports the agreement obtained at high speed (1800 rpm and 1900 rpm in literature). On the basis of these comparisons, the current experimental set-up is validated.

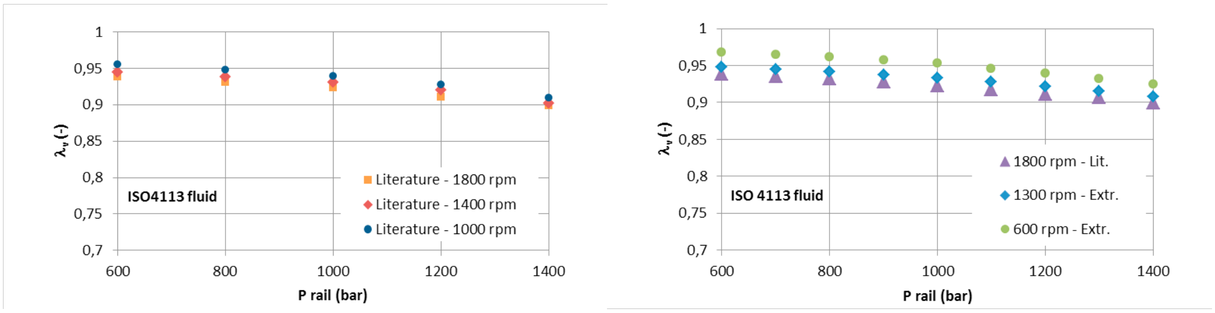

Fig. 1. (Left) Pump induction coefficient λ_v; (Right) Pump induction coefficient λ_v - trends found in the literature [9].

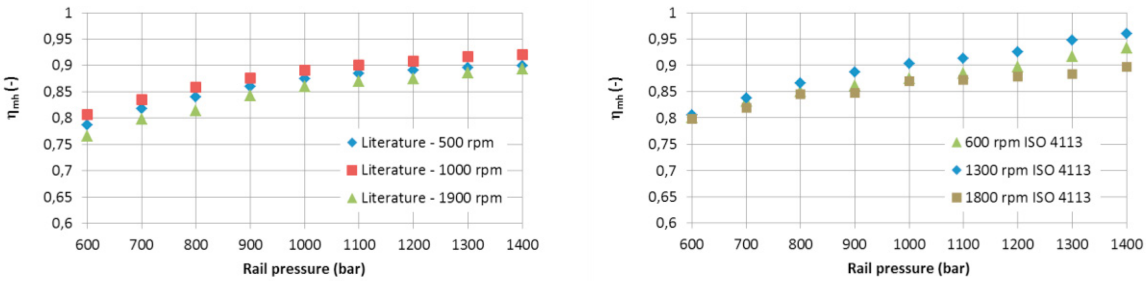

Fig. 2. (Left) Mechanical-hydraulic efficiency of the pump under investigation; (Right) Pump mechanical-hydraulic efficiency – elaboration of the trends found in the literature [9].

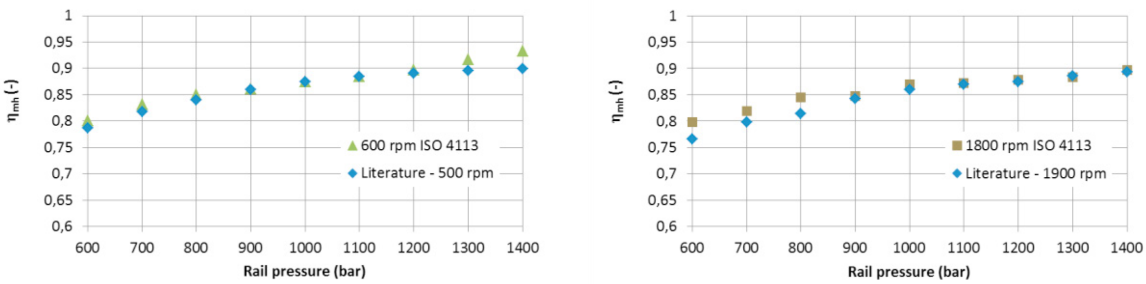

Fig. 3. Pump mechanical-hydraulic efficiency – comparison towards the trends found in the literature [9] (Left) 600 rpm; (Right) 1800 rpm.

Once the assessment phase of the experimental set-up is completed, the investigation is further conducted by passing to the analysis of the pump torque request.

3.2. Torque demand of the pump

The results relating to the pump torque demand at the various operating points taken into consideration are reported. Figure 4 (Left) shows the trends of the torque at pump shaft versus the pressure of the rail, at fixed speed (600 rpm). By comparing the values obtained with the two different working fluids, it is possible to observe very similar behaviors, for each level of pressure considered. Figure 4 (Right) shows the torque values obtained at increased speed (1300 rpm). In this case the trends are qualitatively similar to what has been seen previously, but this time the torque values related to the WCO biodiesel fluid deviate from those relating to the reference fluid, showing an increase in the torque demand; this tendency increases with increasing pressure in the first half of the curve, while it appears decreasing in the second half of the curve. Turning to the trends related to the third level of speed here considered, Figure 5, the behavior is similar to the previous case; the dependence of the torque demand from the pressure is very similar in the cases 1300 rpm and 1800 rpm, where substantial differences are not appreciated at the same pressure at the different speeds considered.

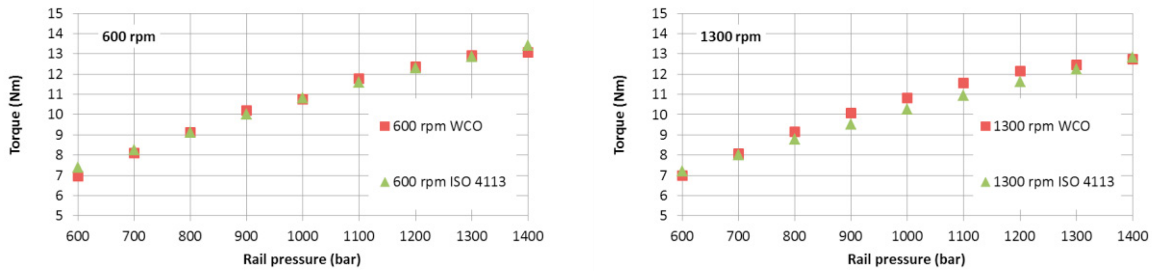

Fig. 4. (Left) Torque at pump shaft – 600 rpm; (Right) Torque at pump shaft – 1300 rpm.

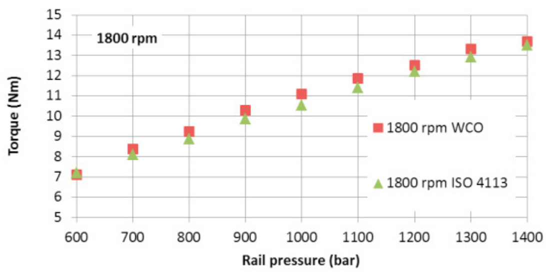

Fig. 5. Torque at pump shaft – 1800 rpm.

4. Conclusion

The high pressure pump of common rail systems operates according to a logic in which the delivered fuel flow rate largely exceeds the engine fuel demand; therefore it can be considered that the operating conditions of the pump in terms of delivery pressure and number of revolutions are independent of any alterations in the flow rate induced by the type of fuel used. On the other hand, the alteration of the torque demand due to the use of the WCO biodiesel was found significant. Therefore, it is expected that an injection system already built and operating on an engine will be sensitive to the WCO biodiesel in terms of torque demand, and will remain substantially insensitive to the possible alteration of the delivery flow rate. From this viewpoint, the investigation was conducted to assess the fuel type influence on the behavior of the pump in terms of torque-request; the focus has been pointed on a pump model typically adopted in the field of light duty diesel engines. As a first step, the pump was characterized by using the ISO 4113 calibration fluid. Taking the opportunity to compare the results with the only data available in the literature [9], it was intended to verify the reliability of the experimental test procedure. Once the characterization phase with the calibration fluid was completed, the attention has been switched to the alternative fluid (WCO biodiesel) and the differences in terms of torque-request have been highlighted. The alternative fluid proved its capability to influence the torque demand and a significant increase was observed in tests at speeds above 600 rpm; the increases have shown sensitivity to the pressure level (in the range 700-1300 bar) and the differences have been found in the order of 5%.

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References

[1] Wloka J. et al. (2018) Continuous Development of Common Rail Injection Technology for Medium-speed 4-stroke Diesel Engines. In: Siebenpfeiffer W. (eds) Heavy-Duty-, On- und Off-Highway-Motoren 2015. Proceedings. Springer Vieweg, Wiesbaden, ISBN 978-3-658-21582-8, DOI https://doi.org/10.1007/978-3-658-21583-5_9

[2] Payri, R., Viera, J.P., Gopalakrishnan, V., Szymkowicz, P.G. (2017) The effect of nozzle geometry over ignition delay and flame lift-off of reacting direct-injection sprays for three different fuels, Fuel, 199, pp. 76-90.

[3] Mohammadi, H., Jabbarzadeh, P., Jabbarzadeh, M., Shrevani-Tabar, M.T. (2017) Numerical investigation on the hydrodynamics of the internal flow and spray behavior of diesel fuel in a conical nozzle orifice with the spiral rifling like guides, Fuel, 196, pp. 419-430.

[4] Ganippa, L.C., Andersson, S., Chomiak, J., Matsson, A., (2003) Combustion characteristics of diesel sprays from equivalent nozzles with sharp and rounded inlet geometries, Combustion Science and Technology, 175 (6), pp.1015-1032, DOI: 10.1080/00102200302350.

[5] Payri, R., García, J.M., Salvador, F.J., Gimeno, J., Using spray momentum flux measurements to understand the influence of diesel nozzle geometry on spray characteristics, (2005) Fuel, 84 (5), pp. 551-561, DOI: 10.1016/j.fuel.2004.10.009.

[6] Som, S., Ramirez, A.I., Longman, D.E., Aggarwal, S.K., Effect of nozzle orifice geometry on spray, combustion, and emission characteristics under diesel engine conditions, (2011) Fuel, 90 (3), pp. 1267-1276, DOI: 10.1016/j.fuel.2010.10.048.

[7] Qian D, Liao R, Xiang J, Sun B, Wang S. (2016) Fluid-Structure Interaction Analysis on the Performance of the High-Pressure Fuel Pump for Diesel Engines. ASME. ASME International Mechanical Engineering Congress and Exposition, Volume 4A: Dynamics, Vibration, and Control ():V04AT05A040. doi:10.1115/IMECE2016-66766.

[8] Qian D, Liao R, (2015) Theoretical analysis and mathematical modelling of a high-pressure pump in the common rail injection system for diesel engines. Proc IMechE Part A: J Power and Energy 2015, Vol. 229(1) 60–72, DOI: 10.1177/0957650914539677

[9] Catania A.E., Ferrari A., (2011) Experimental Analysis, Modeling and Control of Volumetric Radial-Piston Pumps. J. Fluids Eng 133(8), 081103 (Aug 19, 2011) (12 pages) doi:10.1115/1.4004443.

[10] Chiatti, G., Chiavola, O., and Palmieri, F., Injector Dynamic and Nozzle Flow Features in Multiple Injection Modeling, SAE Technical Paper 2007-24-0038, 2007, https://doi.org/10.4271/2007-24-0038.

[11] Caresana, F., (2010) Impact of Biodiesel Bulk Modulus on Injection Pressure and Injection Timing. The effect of residual pressure. Fuel, Volume 90, Issue 2, 2011, Pages 477-485, ISSN 0016-2361, DOI https://doi.org/10.1016/j.fuel.2010.10.005.

[12] Chiatti G, Chiavola O, Palmieri F (2018) Impact of Waste Cooking Oil in Biodiesel Blends on Particle Size Distributions from a City-Car Engine. Journal of the Energy Institute. 2018, 91(2):262-269. doi: 10.1016/j.joei.2016.11.009.

[13] Chiatti G, Chiavola O, Palmieri F (2017) Vibration and Acoustic Characteristics of a City-Car Engine Fueled with Biodiesel Blends. Applied Energy. 2017, 185(1):664-670. doi: 10.1016/j.apenergy.2016.10.119.

[14] Chiatti G, Chiavola O, Palmieri F, Albertini S (2014) Combustion and Emissions Characterization of Biodiesel Blends in a City-Car Engine. Energy & Fuels, 2014, 28(8):5076–5085. doi: 10.1021/ef501023q.

[15] Simcenter Amesim, 2018 Siemens Product Lifecycle Management Software Inc. Amesim Manuals, Technical Bulletins and Libraries; Release 16.