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Relationships between Common Rail accumulator pressure and vehicle traction properties

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Summary. This study aimed at presenting the relation between Common Rail accumulator pressure and vehicle traction properties (acceleration capacity and ability to achieve maximum velocity be vehicle). Tests were performed using an engine test bench and a CR pressure testing kit. They were conducted in conformity with engine standards (i.e. the requirements being specified in them), creating the engine torque and engine effective power to CR pressure characteristic curves.

When making the final diagrams for vehicle acceleration (at each gearbox ratio) and vehicle velocity (in the last gear) and accumulator pressure, a significant effect of the latter on traction parameters was found.

Key words: Common Rail, vehicle traction properties, pressure in CR accumulator.

INTRODUCTION

The Common Rail is the fuel supply system most frequently used in direct injection compression-ignition engines in passenger cars at present. Its main advantages are the reduction of noise of working engine and amount of harmful substances emitted to the atmosphere as well as power increase [9,11]. The system is more economical to operate and environmentally friendly (less fuel consumption) in comparison to conventional diesel power systems and power supply systems of TDI.

Its major advantage is the possibility to change and adjust injection pressure and injection time owing to separation of pressure-producing element (high pressure pump) from fuel-injecting elements (fuel injectors) by a container (Rail) acting as a pressure accumulator [10,15]. Pressure in the CR is a dynamic value and, together with injection time and fuel temperature, affects the dose size and the whole characteristic of injection, and the same the engine parameters [2,3,4].

In the paper by Gołębiewski [6], a relationship between CR pressure and these parameters has been presented. A growing non-linear relationship between the pressure p_{rail} and engine effective power was found using graphs. The function of engine torque to Common Rail pressure was similar to that in case of engine rotational speed (curves had very similar courses).

Engine operating parameters (working ability and working ability in time unit) are of importance for vehicle movement potential (accelerations and velocities, thus its dynamics). They define motor car traction properties (presented in Fig. 1) [1,5,7,12,13,14,16,17,18,19].

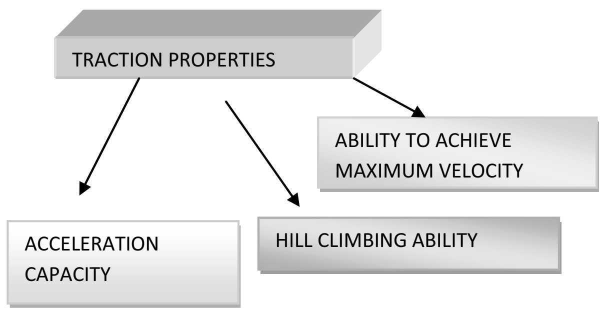

Fig. 1. Division of vehicle traction properties (prepared by the authors)

Acceleration capacity a specified in m/s² has been determined from the following relationship [17,19]:

$$a = \frac{F_N}{m} = \frac{T_{tq\,k}}{mr_k} = \frac{T_{tq} \cdot \eta_{UN} \cdot i_{UN}}{mr_k}, \qquad (1)$$

where :
FN – driving force [N],
m – gross vehicle mass [kg],
T_{tqk} – torque on driving wheels [Nm],
T_{tq} – engine torque [Nm],
η_{UN} – power transmission system efficiency,
i_{UN} – power transmission system ratio,
r_k – wheel kinematic radius [m].

Vehicle linear velocity *v* specified in *m/s* is being defined by the following relation [17,19]:

$$v = \omega_k r_k = \frac{2\pi n_k r_k}{60} = \frac{\pi n r_k}{30 i_{UN}}, \quad (2)$$

where:
ω_k – angular velocity of road wheels [rad/s],
n_k – rotational speed of wheels [min^{-1}],
n – engine rotational speed [min^{-1}].

When changing the velocity units from *m/s* to *km/h*, the result should be multiplied by 3.6.

Article [8] presents the results of traction motor vehicle navigation using techniques designed to determine the characteristics of an external spark-ignition engine is also based on the measurement of the opening of the injector and fuel rail pressure.

The authors of this paper decided to examine a relationship between Common Rail pressure and vehicle traction properties.

OBJECTIVE AND METHODS OF EXPERIMENTS

The objective of this research project was to determine a relation between Common Rail accumulator pressure and vehicle traction properties (acceleration capacity and ability to achieve maximum velocity). In the study, a torque to Common Rail pressure relationship for feeding a direct injection turbocharged compression-ignition engine designed for driving motor cars with a full dose of fuel. Research methods were conducted in conformity with the Polish standard PN-ISO 15550 [21]. Experiments were performed according to the requirements specified in them using an engine test bench.

TEST BED

The test bed consisted of the following components:

a) fuel tank (diesel oil) – a 750 litre fuel tank designed for diesel oil, made of plastic;
b) FIAT Multijet 1.3 JTD engine – a direct injection 4-stroke turbocharged compression-ignition engine with the Common Rail fuel supply system [20];

Table 1. Engine description according to manufacturer's data [20]

FIAT Multijet 1.3 JTD 16 V	unit	value / description
Cylinder diameter	[mm]	69.6
Piston travel	[mm]	82
Compression ratio	-	18.1
Number of cylinders	-	4
Arrangement of cylinders	-	in-line
Injection sequence	-	1-3-2-4
Engine capacity	[cm^3]	1248
Maximum power	[KM/kW]	75/51
Rotational speed at maximum power	[min^{-1}]	4000
Maximum torque	[Nm]	145
Rotational speed at maximum torque	[min^{-1}]	1750

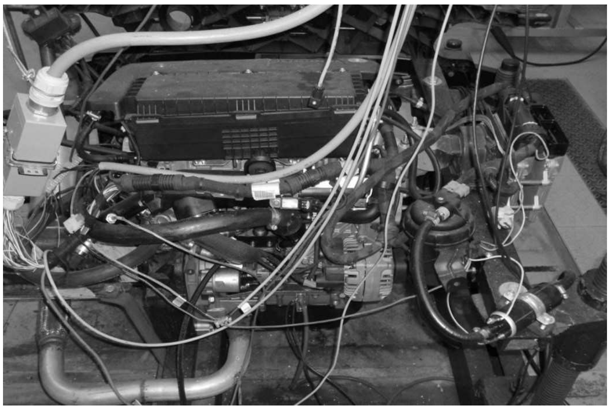

Fig. 2. A view of Fiat Multijet 1.3 JTD 16 V engine

c) EMX 100 eddy current brake manufactured by Elektromex (Poland) – a device for power take off and measurement in combustion engines being placed on test beds;

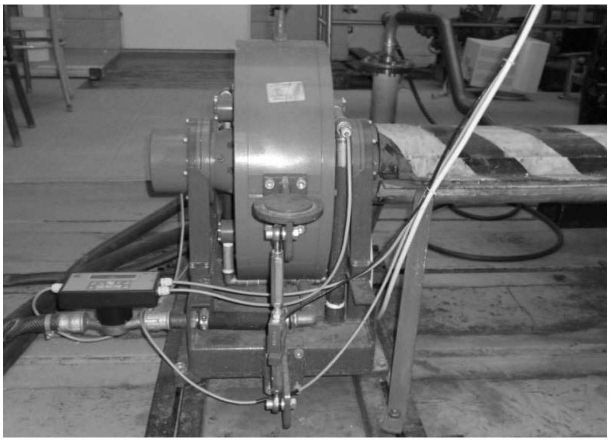

Fig. 3. EMX 100 eddy current brake

d) Common Rail high pressure measurement kit which consisted of: a manometer, a T-connection with nozzles, vent and stopper, a measuring hose, fuel tubbing, a reduction and a venting hose.

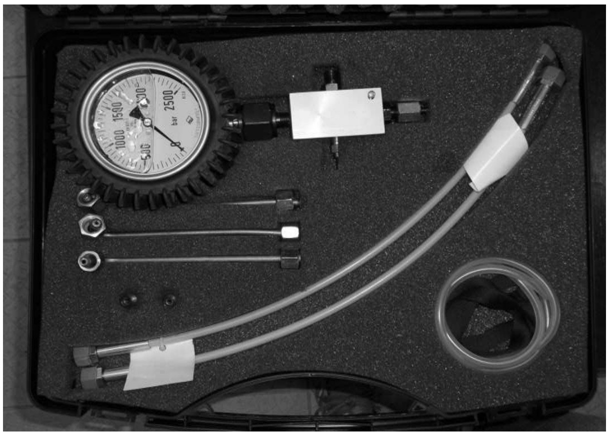

Fig. 4. HP-made Common Rail high pressure measurement kit

COURSE OF TESTS

Results of the measurements being performed are presented in Fig. 5. The engine was fed with a full dose of ON EKODIESEL fuel with the cetane number 51.1. Ambient parameters during the test were as follows:
– ambient temperature T_a = 294 K,
– ambient pressure p_a = 98.5 kPa,
– relative humidity = 40%.

The engine torque and its effective power being measured were corrected according to the relations comprised in the Polish standard PN-ISO 15550 [21]. The use of HP-made high pressure measurement kit allowed creating an engine torque and engine effective power to Common Rail pressure relationship. It is presented below in Fig. 5. The characteristic curves were determined based on measuring points due to very narrow ranges of measurement uncertainties (see section 5). In order to illustrate the relation between CR accumulator pressure and vehicle traction properties, the variables such as vehicle weight, power transmission system efficiency, respective gear ratios and final drive ratio were also necessary.

Table 2. Basic vehicle details

variable	value	unit	where:
m	1932.57	[kg]	gross vehicle mass
η_{UN}	0.9	-	power transmission system efficiency
rk	0.27	[m]	wheel kinematic radius
ibI	3.909	-	first gear ratio
ibII	2.158	-	second gear ratio
ibIII	1.345	-	third gear ratio
ibIV	0.974	-	fourth gear ratio
ibV	0.766	-	fifth gear ratio
ipg	3.438	-	final drive ratio

RANGES OF MEASUREMENT UNCERTAINTIES

The ranges of measurement uncertainties were determined for engine rotational speed, torque and effective power and Common Rail pressure of the engine torque-speed (external) characteristics.

Expanded measurement uncertainty for engine rotational speed was a type B standard uncertainty (depending on the accuracy of measuring device calibration; the experimenter's uncertainty was omitted due to the result self-reading and self-recording measuring system).

Type B standard uncertainty was determined from the following relationship [22]:

$$u_B(n) = \frac{\Delta n}{\sqrt{3}} = \frac{1}{\sqrt{3}} = 0.67[\min^{-1}], \tag{3}$$

where:
Δn – uncertainty of rotational speed sensor calibration = 1 [$\min^{-1}$].

According to the uncertainty propagation law, standard (total) uncertainty was equal to type B standard uncertainty [22]:

$$u_B(n) = u(n) = 0.67[\min^{-1}]. \tag{4}$$

Expanded uncertainty was determined by the following equation [22]:

$$U(n) = k \cdot u(n), \tag{5}$$

where:
k – coverage factor = 2.

$$U(n) = 2 \cdot 0.67 = 1.34[\min^{-1}]. \tag{6}$$

Thus, ultimately, the measurement uncertainty for the whole range of rotational speeds was equal to:

$$U(n) = \pm 2[\min^{-1}]. \tag{7}$$

Expanded uncertainty for corrected engine torque measurement

In the case of type A standard uncertainty, the arithmetic mean of torque was adopted as a measurement result [22]:

$$\overline{T_{tq}} = \frac{1}{n}\sum_{i=1}^{n} T_{tq(i)}, \tag{8}$$

where:
$\overline{T_{tq}}$ – arithmetic mean of engine torque measurement [Nm],
$T_{tq(i)}$ – value of the i-th torque measurement [Nm],
n – number of measurement.

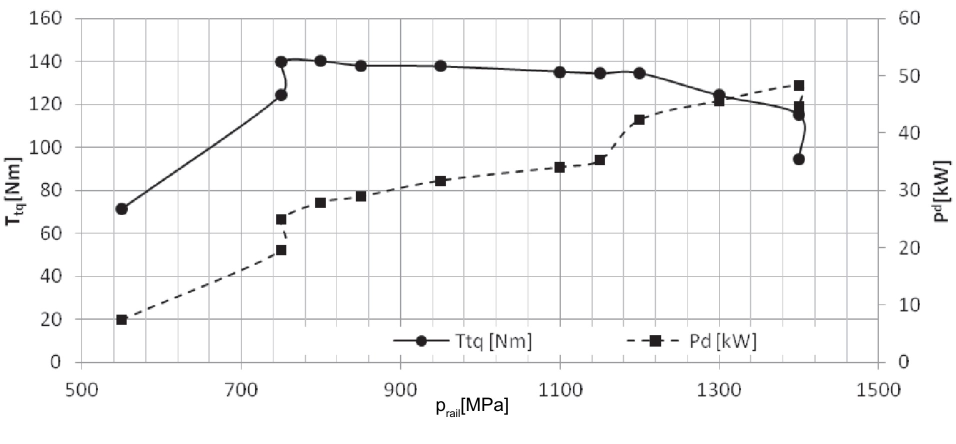

Fig. 5. Engine torque and engine effective power to CR relationship
where: T_{tq} – engine torque, P^d – engine effective power, p_{rail} – Common Rail accumulator pressure

For example for rotational speed n= 1900 obr/min, it was as follows:

Table 3. Calculation table for engine torque measurement

No.	$T_{tq(i)}$	$\overline{T_{tq}}$	$(T_{tq(i)} - \overline{T_{tq}})$	$(T_{tq(i)} - \overline{T_{tq}})^2$
1	140.9	140.1	0.8	0.64
2	140.6		0.5	0.25
3	140.7		0.6	0.36
4	138.3		-1.8	3.24
$\sum (T_{tqi} - \overline{T_{tq}})^2$				**4.49**

Type A standard uncertainty of this result was calculated as a standard deviation [22]:

$$u_A(T_{tq}) = \sqrt{s^2_{\overline{T_{tq}}}} = \sqrt{\frac{1}{n(n-1)}\sum_{i=1}^{n}(T_{tq(i)} - \overline{T_{tq}})^2} = 0.61[Nm]. \quad (9)$$

Type B standard uncertainty was determined from the following relationship [22]:

$$u_B(T_{tq}) = \frac{\Delta T_{tq}}{\sqrt{3}} = \frac{0.1}{\sqrt{3}} = 0.058[Nm], \quad (10)$$

where:
ΔT_{tq} – uncertainty of engine brake calibration = 0.1 [Nm].

According to the uncertainty propagation law, standard (total) uncertainty amounted to [22]:

$$u\ (T_{tq}) = \sqrt{s^2_{\overline{T_{tq}}} + \frac{(\Delta T_{tq})^2}{3}} = 0.62[Nm]. \quad (11)$$

Expanded uncertainty was determined by the following equation [22]:

$$U(T_{tq}) = k \cdot u(T_{tq}), \quad (12)$$

where:
k – coverage factor = 2.

$$U(T_{tq}) = 2 \cdot 0.62 = 1.23[Nm]. \quad (13)$$

Thus, ultimately, the result of engine torque measurement together with measurement uncertainty was written in the following form:

$$T_{tq} = (140.1 \pm 1.3)[Nm]. \quad (14)$$

In the table below, the values of engine torque together with expanded measurement uncertainty are presented for particular rotational speeds. The torque values were calculated as mean values of 4 measurements. The value of coverage factor k was 2, whereas the uncertainty of measurement accuracy for engine brake amounted to 0.1 Nm.

Table 4. Measurement uncertainties for engine torque

	n	$\overline{T_{tq}}$	$u_A(T_{tq})$	$u_B(T_{tq})$	$U(T_{tq})$	$\overline{T_{tq}} \pm U(T_{tq})$
	[min^{-1}]	[Nm]	[Nm]	[Nm]	[Nm]	[Nm]
1	1000	71.4	0.27	0.058	0.56	71.4±0.6
2	1500	124.4	0.38		0.77	124.4±0.8
3	1700	139.6	0.58		1.16	139.6±1.2
4	1900	140.1	0.61		1.23	140.1±1.3
5	2000	138.1	0.32		0.65	138.1±0.7
6	2200	137.7	0.20		0.41	137.7±0.5
7	2400	135.2	0.18		0.38	135.2±0.4
8	2500	134.4	0.06		0.17	134.4±0.2
9	3000	134.6	0.41		0.83	134.6±0.9
10	3500	124.2	0.19		0.40	124.2±0.4
11	4000	115.1	0.13		0.28	115.1±0.3
12	4500	94.6	0.17		0.37	94.6±0.4

Measurement uncertainties for engine effective power were determined in a similar manner.

Table 5. Measurement uncertainties for engine effective power

	n	$\overline{P^d}$	$u_A(P^d)$	$u_B(P^d)$	$U(P^d)$	$\overline{P^d} \pm U(P^d)$
	[min^{-1}]	[kW]	[kW]	[kW]	[kW]	[kW]
1	1000	7.5	0.029	0.058	0.13	7.5±0.2
2	1500	19.5	0.087		0.21	19.5±0.3
3	1700	24.9	0.104		0.24	24.9±0.3
4	1900	27.9	0.112		0.25	27.9±0.3
5	2000	29.0	0.076		0.19	29.0±0.2
6	2200	31.7	0.050		0.15	31.7±0.2
7	2400	34.1	0.029		0.13	34.1±0.2
8	2500	35.2	0.000		0.12	35.2±0.2
9	3000	42.3	0.091		0.22	42.3±0.3
10	3500	45.6	0.065		0.17	45.6±0.2
11	4000	48.3	0.065		0.17	48.3±0.2
12	4500	44.7	0.115		0.26	44.7±0.3

Expanded uncertainty for CR pressure was a type B standard uncertainty (depending on device calibration accuracy).

Type B standard uncertainty was determined from the following relationship [22]:

$$u_B(p_{rail}) = \frac{\Delta p_{rail}}{\sqrt{3}} = 8.34[MPa], \quad (15)$$

where:
Δp_{rail} – uncertainty of CR pressure gauge calibration = 50 [MPa].

According to the uncertainty propagation law, standard (total) uncertainty was equal to type B standard uncertainty [22]:

$$u_B(p_{rail}) = u(p_{rail}) = 8.34[MPa]. \quad (16)$$

Expanded uncertainty was determined by the following equation [19]:

$$U(p_{rail}) = k \cdot u(p_{rail}), \quad (17)$$

where:
k – coverage factor = 2.

$$U(p_{rail}) = 2 \cdot 8.34 = 16.68[MPa]. \quad (18)$$

Thus, ultimately, the measurement uncertainty for the whole range of pressures was equal to:

$$U(p_{rail}) = \pm 50[MPa]. \quad (19)$$

RESULTS

The acceleration values being obtained for each gear were determined using Equation (1) and Table 2. Using the data from Table 2 and Relation (2), the values of vehicle linear velocity in the fifth gear were determined in a similar manner. In Figures 6 and 7 below, the relations between Common Rail accumulator pressure and vehicle traction parameters (acceleration capacity and ability to achieve maximum velocity) are presented. The acceleration values in respective gears themselves depended mainly on the values of selectable ratios, whereas the character of curves showed the vehicle to speed up the best in each gear in the region of Common Rail accumulator pressure being equal to 850 bar. Taking into consideration the range of effective accumulator pressure, it amounted to approximately 40% of the maximum pressure.

The engine torque-speed to CR pressure characteristics pointed to a linear relationship of these parameters. The velocity in the last gear increased together with the increasing value of pressure, which indicates to the fact that the vehicle can be able to achieve maximum velocity at higher values of Common Rail pressure.

CONCLUSIONS

The carried out study allowed drawing the following conclusions:

a) CR pressure affects the course of engine torque characteristic curve and further, which is related to this, the

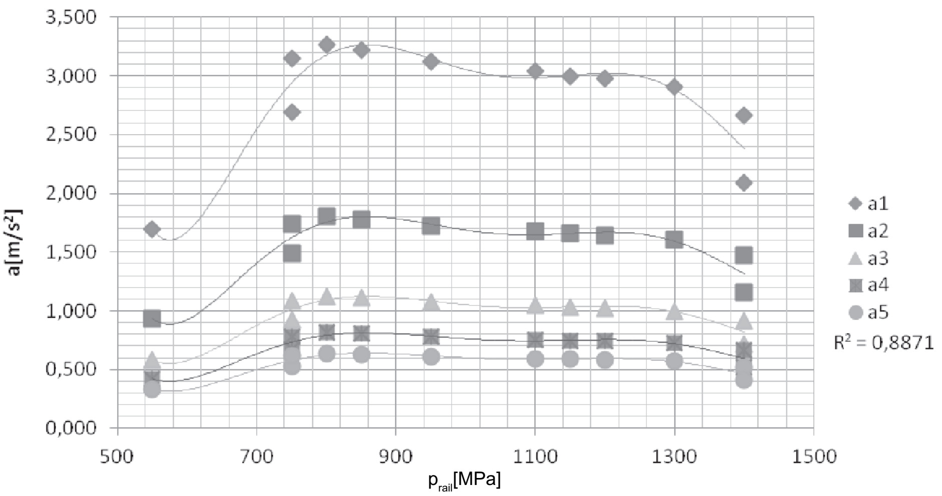

Fig. 6. Relationship between vehicle acceleration and CR pressure:
a – vehicle acceleration, **a1** – vehicle acceleration in first gear, **a2** – vehicle acceleration in the second gear, **a3** – vehicle acceleration in the third gear, **a4** – vehicle acceleration in the fourth gear, **a5** – vehicle acceleration in the fifth gear, $\mathbf{p}_{rail}$ – CR accumulator pressure, $\mathbf{R^2}$ – square of the coefficient of correlation

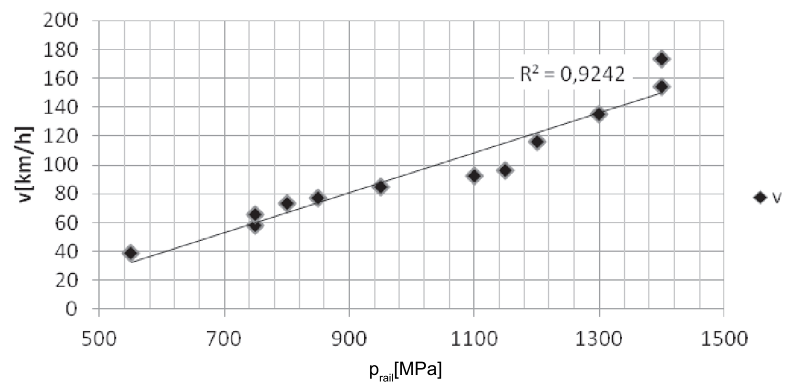

Fig. 7. Vehicle linear velocity (last gear) to CR pressure relationship:
v – vehicle linear velocity (last gearbox ratio), $\mathbf{p}_{rail}$ – CR accumulator pressure

course of vehicle acceleration characteristic curve in respective ratios;

b) it has a significant effect on engine power and thus on the ability of a vehicle to achieve maximum velocity (in the last gear);

c) increase in Common Rail pressure results in better traction parameters but also in higher fuel consumption;

d) it cannot be changed in any range due to durability of engine components (fuel supply system and piston-crank system).

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RELACJE MIĘDZY CIŚNIENIEM W ZASOBNIKU COMMON RAIL A WŁAŚCIWOŚCIAMI TRAKCYJNYMI POJAZDU

Streszczenie. Celem pracy było przedstawienie relacji pomiędzy ciśnieniem w zasobniku Common Rail a właściwościami trakcyjnymi (zdolność do przyspieszania i osiągania prędkości maksymalnej przez pojazd). Badania zostały przeprowadzone przy wykorzystaniu hamowni silnikowej oraz zestawu do sprawdzania ciśnienia w szynie CR. Zostały wykonane zgodnie z normami silnikowymi (wymaganiami w nich określonymi) poprzez utworzenie charakterystyk momentu obrotowego i mocy użytecznej silnika od ciśnienia w szynie CR. Wykonując ostateczne wykresy pomiędzy przyspieszeniem pojazdu (na każdym przełożeniu skrzyni biegów) oraz prędkością pojazdu (na biegu ostatnim) a ciśnieniem w zasobniku stwierdzono o znaczącym jego wpływie na parametry trakcyjne.

Słowa kluczowe: Common Rail, właściwości trakcyjne pojazdu, ciśnienie w zasobniku CR.