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EXPERIMENTAL INVESTIGATIONS FOR MEETING STANDARDS OF DIESEL ENGINE VISIBLE EMISSIONS

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Abstract: *The paper presents research work done on a turbocharged direct injection diesel engine for the meeting of visible emissions according to ECE 24 Regulation. The engine 392-L4 DT was tested on the dynamometric bench being measured specific performances on total speed characteristics (power, torque, specific fuel consumption, charge air pressure and exhaust gas temperature) and visible emissions - in terms of smoke number (N_s) expressed in Hartridge smoke units (HSU). Several experimental settings were done in order to reduce smoke emissions, especially at lower speeds, by variation of the injector nozzle hole diameter and variation of turbocharger configurations.*

Key words: *diesel engine, visible emissions, experimental settings.*

1. Introduction

Diesel engines are used predominantly to power heavy trucks, buses and tractors. Their high thermal efficiencies resulting from lean combustion and high compression ratios lead to low fuel consumption and carbon dioxide emissions, being also attractive for lower power applications such as light trucks, vans and passenger cars. However, the major concern over diesel engines is to control the more stringent pollutant limits imposed by legislation along with improving specific performance indicators. Among the most spread European emission standards, regulation ECE R 24 [3] defines provisions for approval with regard to "visible pollutants" which means smoke emission.

Thus in the present study are presented engine dynamometric experiments which were carried out on a light truck diesel engine to investigate the effectiveness of several methods to reduce smoke emission under the limit of compliance with EU regulation.

2. Experimental Set-Up

The tested engine was manufactured at Roman Truck Company being provided to power 6.5 tonne light trucks. The engine codification is 392-L4-DT, the first group of digits representing the engine displacement, 3.92 liters, cylinder configuration and number, namely - in line cylinders and 4 cylinders and, finally, the group of letters DT representing the type of injection (direct injection) and the type of supercharging (turbocharged).

The series production engine having the cylinder block no. 011 was tested on the 220 kW eddy-current type dynamometric

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test bench (Ono-Sokki) at Road Vehicle Institute (INAR). The main engine specifications are presented in Table 1 [10].

Engine technical data Table 1

Engine type	Diesel, 4 stroke, direct injection
Cylinder configuration	4-cylinder, in line
Bore x Stroke [mm]	102 x120
Displacement [l]	3.92
Compression ratio	17.5:1
Rated power [kW]	77
Rated speed [rpm]	2800
Max. torque [N·m]	309
Max. torque speed [rpm]	1800

The engine performance depends on atmospheric conditions and auxiliary equipment. During the tests the atmospheric pressure ranged within 700-716 mm Hg and ambient air temperature 15-25 ºC [8]. The tests were performed according to Romanian engine testing standard [5] which is equivalent to ISO 1585 [4], being corrected the measured values. The engine performance is corrected according to pressure and temperature with correction coefficient α, with f_a atmospheric factor and f_m engine factor, with the formula:

$$\alpha = f_a^{fm}, \tag{1}$$

$$f_a = \left(\frac{99}{p_s}\right)^{0.7} \left(\frac{T}{298}\right)^{1.5}, \tag{2}$$

p_s - atmospheric dry pressure expressed in kilopascals, T - atmospheric temperature in Kelvin.

The air temperature in absolute Kelvin scale is measured at the engine inlet at 0.15 m upstream the air filter:

$$f_m = 0.036 \cdot q_c - 1.14, \tag{3}$$

$$q_c = \frac{q}{r}, \tag{4}$$

with q - fuel flow in milligram per cycle and per liter of total swept volume (mg/(L·cycle) and r - air pressure ratio of compressor outlet and compressor inlet. The calculated values of α ranged in 1.00-1.01, the interval being included in the standard requirement of 0.9-1.1.

The measured values of effective power, torque and fuel consumption were corrected using the following formula, written for effective power:

$$P_{corr,ef} = \alpha \cdot P_{m,ef}, \tag{5}$$

with $P_{corr,ef}$ - corrected effective power and $P_{m,ef}$ - measured effective power.

During the testing the engine was equipped with:

- A type in-line injection pump;
- fuel correction device;
- nozzle hole diameters -0.36 and 0.30 mm;
- Holset turbochargers;
- no fan;
- no supplied alternator.

The engine was instrumented with temperature sensors (cooling liquid, oil and exhaust gas), pressure sensors (oil, air charge, exhaust gas) and flowmeters (air and fuel).

As cooling agent was used the distilled water from the cooling system of the test bench, the temperature being kept in the range of 75-80 °C. For the engine operation it was used diesel fuel according to standard EN 590 and lubricant with viscosity class SAE 15W40.

The smoke emission was measured with Hartridge opacimeter, type (MK3), which has the effective length of measurement tube of 430 mm and readings in HSU (Hartridge Smoke Units) or m^{-1}. The exhaust gas opacity was carried out with the engine running under full-load and at steady speed, between 1200-2800 rpm.

For each of the engine speeds at which the absorption coefficient was measured, the nominal gas flow was calculated by means of the following formula [3] for four-stroke engines:

$$G = \frac{V \cdot n}{120}, \tag{6}$$

where G - rated exhaust gas flow, in liters per second (L/s), V - cylinder displacement of the engine, in liters (L), n - engine speed, in revolutions per minute (rpm).

The series production engine is equipped with injection pump having 4 injectors, with 4 hole nozzle with 0.36 mm diameter and 0.7 mm length each. The hole axes are positioned on the lateral surface of a cone with the cone edge angle of 150°.

3. Design of Experiments

Among the techniques of lowering emissions in engine design, two were considered here, one acting on injection process and the other on air flow rate supplied to the engine.

The first technique investigates the behaviour of injection pump with smaller nozzle diameters which improves the quality of the pulverisation, typically expressed in Sauter mean diameter of the droplet (SMD). A smaller SMD lowers vaporisation period and auto-ignition delay so the injection timing can be reduced too [1], [2], [6]. As a result the maximum in - cylinder pressure is diminished leading to smooth engine running.

Preliminary research work was performed on this engine type in form of numerical simulations. In the study [7] a semi-empirical mathematical model was conceived by applying the similitude and flow equations. Based on the input data (geometry of the injection pump elements, fuel rate per cycle and opening injector pressure) there were computed the injection period, maximum injection pressure, spray penetration, SMD, total droplet surface, spray time variation of spray penetration and angular cone dispersion, Reynolds numbers of the flow through the nozzles.

The simulations predicted enhancement of fuel spray quality by reducing the nozzle diameter from 0.36 mm (series nozzles) to 0.32 mm (calculation), as presented in Table 2. The calculations were performed for rated speed (2800 rpm) and for maximum torque speed (1800 rpm).The reduction of the nozzle diameter predicts a reduction of SMD of 12% and an increase of the droplet area of 14% for rated speed, having similar values for maximum torque speed. So the first solution for increasing engine performance was to use smaller hole diameter nozzles of 0.30 mm, even smaller than 0.32 mm used in simulation and to compare with series ones of 0.36 mm.

The second technique refers to the optimisation of the turbocharger operation.

So there were identified two experiments of the tested engine in which all the equipment and parameters were the same:

A - engine test with nozzle hole diameters of 0.36 mm (series nozzles);

Output data on numerical simulation for 392-L4-DT Table 2

Nozzle diameter [mm]	**Engine speed [rpm]**	**Real injection time [°CR]**	**Penetration [mm]**	**SMD [μm]**	**Droplet area [mm^2]**	**Reynolds number**
0.36	2800	13.6	56.5	57.25	50985	19667
	1800	11.9	59.6	70.20	47352	14691
0.32	2800	15.2	54.3	50.12	58244	19667
	1800	13.1	57.7	61.15	54359	14691

B - engine test with nozzle hole diameters of 0.30 mm.

The rated speed characteristics for A and B experiments showed that power (Figure 1), torque (Figure 2) and specific fuel consumption (Figure 3) have fair values, but smoke emissions (Figure 4) are higher than the threshold value for A experiment, being at the limit for B experiment. As a consequence the next experiment required a modified turbocharger with a higher charge air pressure and a higher air flow rate to limit the smoke emissions. The solution, entitled experiment C, provided the turbine with a new housing having around 9% smaller exhaust outlet section (Φ 61 mm) than turbine section in experiments A and B (Φ 64 mm), keeping also the gain of 0.30 mm injector nozzle.

The explanation of the air charge pressure rise in the compressor is the thermodynamic coupling between compressor and turbine which are mounted on the same shaft. When the turbine exhaust outlet area is reduced then velocity of exhaust gas is increased resulting higher speeds of the turbocharger; the induced higher speed of the compressor leads to higher air charge pressures.

It can be noticed on Figures 1-5 a better behaviour of solution C than A and B. In terms of exhaust gas temperature measured downstream of the turbocharger (Figure 6), C experiment revealed a lower profile of temperatures showing that thermal energy of the fuel was more efficiently turned into mechanic work in the combustion chamber and turbine. The results of C experiment were not considered good enough and a new experiment D was performed reducing further the turbocharger sections with 21% from Φ 61 mm to Φ 54 mm.

4. Interpretation of Results

An overview of the set of A, B, C and D experiments shows that the engine power curve from Figure 1 kept its profile with increase of maximum 6% from rated power for D experiment reported to A (series) experiment. The torque curves illustrated in Figure 2 evolved to a more stable profile, which means improved dynamics of the vehicle operation, even that A, B and C experiments had higher values than D experiment with 1.5%. Both power and torque were in the limits of ±5% of declared standard values [10].

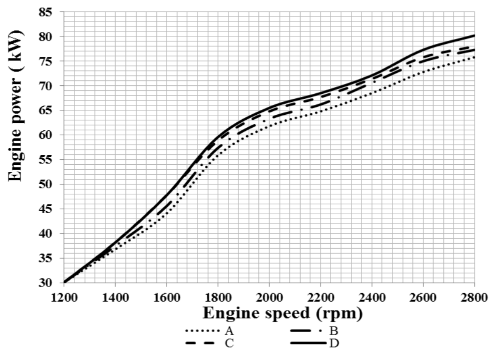

Fig. 1. *Power versus speed at full load*

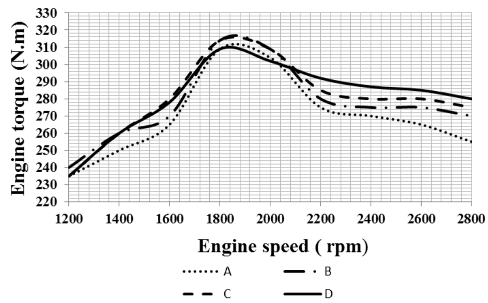

Fig. 2. *Torque versus speed at full load*

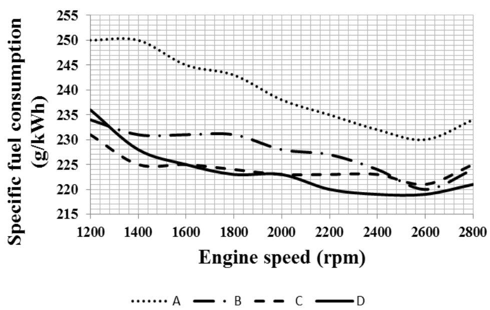

Fig. 3. *Specific fuel consumption versus speed at full load*

Fuel economy was constantly improved as specific fuel consumption decreased from A to D especially in the range of speeds of interest between maximum torque speed and rated power speed.

The most relevant image for engine behaviour is Figure 4 which represents engine smoke emissions. The red line indicates which are the limits imposed by ECE Regulation.

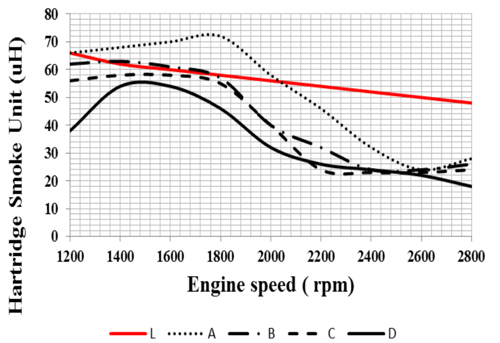

Fig. 4. *Smoke emission versus speed at full load*

If the red line was exceeded by A and B experiments, the modifications brought to turbochargers in C and D experiments succeded to gradually reduce smoke level and to comply with the limits [9].

The reason of failure for A and B experiment was the too low air charge pressure produced by compressor (480 mm Hg than rated value imposed in product standard [10] of 600 mm Hg), as can be seen in Figure 5.

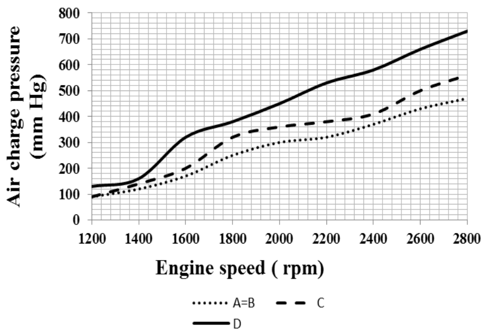

Fig. 5. *Air charge pressure versus speed*

The D experiment exceeded standard value up to 730 mm Hg due to modified turbine housing outlet with a smaller section than C.

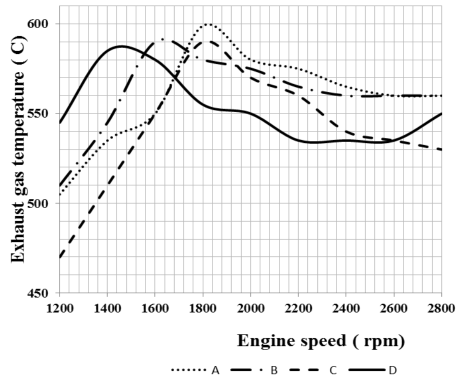

Fig. 6. *Exhaust temperature versus speed*

The confirmation of the correctitude of experiments is the profile of exhaust gas temperatures downstream of the turbine illustrated in Figure 6.

5. Conclusions

The paper summarizes research work performed in four steps in order to obtain the approval on the 392-L4-DT engine on visible emissions.

The test results confirmed that the engine complies with the values of smoke limits according to ECE 24 Regulation in the conditions of D experiment conditions:

- the nozzle hole diameter of injectors was 0.3 mm;
- the turbine section was reduced until the compressor charge air pressure reached the value of 730 mm Hg.

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