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IMPROVING DIESEL ENGINE PERFORMANCE BY AIR-TO-AIR INTERCOOLING

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Abstract: *The paper studies the increase of performance of a heavy duty turbocharged diesel engine when the intake air is cooled in an air-to-air heat exchanger, being turned into a turbocharged and intercooled engine. There were presented dynamometric tests, being evaluated engine performance parameters such as rated power, torque, hourly and specific consumptions and smoke level on the speed characteristic at total load. A stress is laid on the influence of air charge thermodynamic parameters on cooler effectiveness and pressure loss.*

Key words: *diesel engine, intercooling, air-to-air cooler.*

1. Introduction

The engine manufacturers are always in search of methods to increase the power output at the same volume displacement. By applying the thermodynamic laws, the quantity of air - fuel mixture inserted in combustion chamber can be raised primarily by turbocharging, which leads to the increase of air pressure and temperature in the combustion chamber meaning that engine operates at a higher thermal load. Secondly, the charge air can be cooled allowing higher mass of air to be introduced in the combustion chamber, every cycle, thus leading to higher power output and lower fuel consumption. Also, the charge air cooling reduces the temperature in cylinders and exhaust system, improving engine durability. On emissions, intercooling produces lower smoke (particulate) emissions as effect of higher density and air-fuel ratio, as well as lower NO_x emissions [1-3].

The most practical solution for air charge cooling is the use of an air-to-air heat exchanger, which is typically placed in front of the coolant heat exchanger in vehicle configurations.

The effectiveness of the air-to-air heat exchanger (ε) is defined as the ratio of the real charge air temperature drop across the cooler core to the temperature differential available for cooling [1]:

$$\varepsilon = \frac{T_2 - T_3}{T_2 - T_1}, \tag{1}$$

where: ε - heat exchanger effectiveness; T_1 - temperature of the ambient cooling air; T_2 - temperature of charge air entering the cooler (approximately equal to compressor discharge temperature); T_3 - temperature of charge air exiting the cooler (approximately equal to intake manifold temperature).

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The air flow through the cooler induces pressure loss which can be calculated using Equation:

$$\Delta p = p_u - p_d , \quad (2)$$

where: Δp - pressure drop across the cooler; p_u - air pressure upstream the cooler; p_d - air pressure downstream the cooler.

The processes in the cooler should be a trade-off between cooling effectiveness, involving ε values as higher as possible and flow restrictions, involving Δp values as lower as possible.

The present paper describes a stage in the R&D work aiming to increase diesel engine performance, applied to a heavy duty direct injection engine used for commercial vehicles, by implementing the air charge intercooling on a turbocharged engine. The main results of research work were quantified reported to the performance of the turbocharged engine version; the newly developed engine-turbocharged and intercooled should have higher rated power, maximum torque and reliability as well as lower specific fuel consumption, at the same or lower smoke emission. Also the parameters of cooling are assessed in terms of effectiveness (Eq. 1) and pressure loss (Eq. 2).

2. Engine Test Procedure

The tested engine was manufactured at Roman Truck Company (Brașov, Romania), being a turbocharged direct injection engine 798-05, having the series number 4538.

The newly developed engine, abbreviated 798-05R, derives from 798-05, having applied charged air intercooling and derating. The engine parameters before intercooling are described in Table 1 [4].

The 798-05R engine was instrumented with temperature sensors measuring cooling liquid, oil and exhausts gas temperatures, as well as pressure sensors to measure oil and air charge pressure.

Engine technical data Table 1

Engine type	**Diesel, 4 stroke, direct injection**
Cylinder configuration	6-cylinder, in line
Bore x Stroke [mm]	102 x112
Displacement [L]	5.5
Compression ratio	17:1
Rated power [kW]	98
Rated speed [rpm]	2800
Max. torque [N·m]	412
Max. torque speed [rpm]	1800

The tests were performed according to Romanian engine testing standard [6] which is equivalent to ISO 1585 [7]. During engine testing the ambient temperature was 15 °C and atmospheric pressure was 719 mm Hg.

The engine was equipped as follows:
- 6 blade cooling fan Φ 530 x 80 mm;
- no compressor and unloaded alternator.

The opacimeter used was Hartridge MK3 which has the effective length of measurement tube of 430 mm and readings in HSU (Hartridge Smoke Units) or m^{-1}.

The engine performance (power, torque, hourly and specific fuel consumption) has been measured on the 300 kW MEZ-VSETIN dynamometric test bench at Road Vehicle Institute Brașov (INAR) [8], as can be seen in Figure 1. The air cooler was placed in front of the engine coolant radiator, similar to the positioning on the vehicle.

In order to compensate the effect of increased power on the engine reliability, the engine was derated, the rated speed being reduced from 2800 rpm to 2500 rpm.

The engine was previously running in for 20 hours. The air intake system provided a pressure loss of 270 mm column H_2O and the exhaust system 370 mm column H_2O, meeting the product specification.

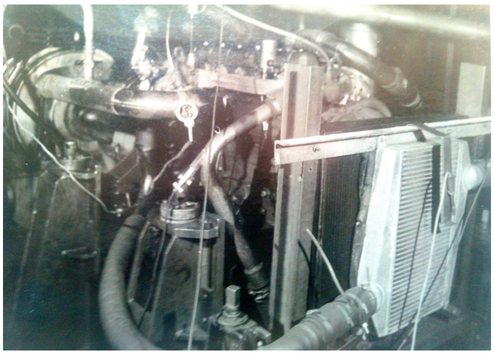

Fig. 1. *Intercooled engine during testing*

The charge air cooler was made of aluminum being placed in front of the liquid cooler, its main parameters being presented in Table 2.

The engine tests included two types of measurements:

A. Thermodynamic parameters of charge air and exhaust gas (air temperature at cooler intake and engine intake, exhaust gas temperature upstream and downstream the turbine, as well as pressure loss on the cooler);

Heat exchanger parameters Table 2

Cooler material	Aluminum
Reference no.	89.08101.6007
Fin shape	undulated, 0.5 mm thick
Tube shape	rectangular
Frontal area [m^2]	0.203
Mass [kg]	18
Dimensions [mm]	676 x 300 x 60

B. Engine speed characteristics at total load (power, torque, hourly fuel consumption, specific fuel consumption, smoke emission).

Engine configuration is presented in Figure 2. The engine was provided with M30 Super 2 lubricant.

3. Interpretation of Results

A. The thermodynamic parameters of charge air and exhaust gas have been measured in order to find out the effectiveness of the charge cooler. There were measured the air temperature at cooler intake, T_2, and at engine intake, T_3, with the engine running at full load, over the whole range of speeds.

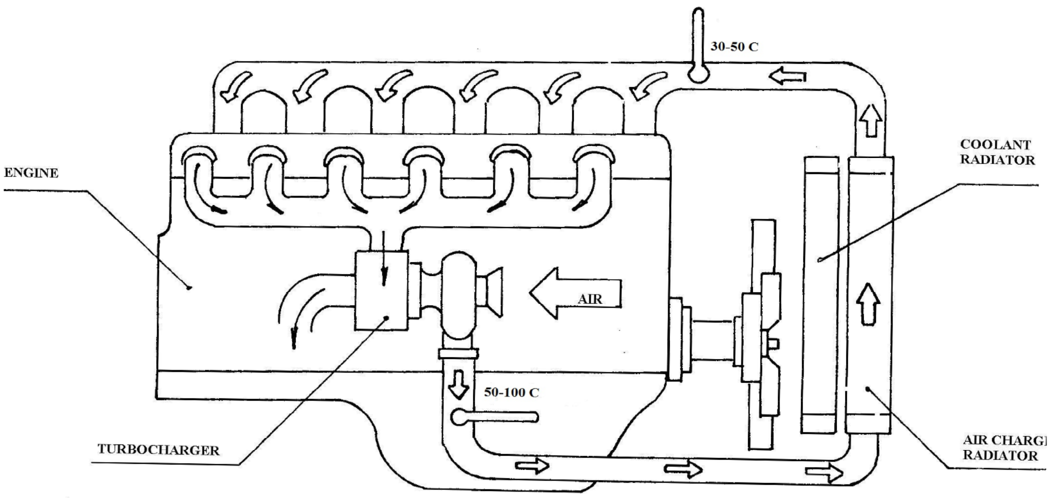

Fig. 2. *Configuration of turbocharged-intercooled diesel engine*

Figure 3 illustrates the evolution of temperatures, being obviously that the difference of temperatures, (T_2 – T_3), is significant, the air cooler lowering the air charge with a mean difference of 39 °C.

The typical values of cooler effectiveness are 0.6-0.7, the higher values meaning that air charge is more intensely cooled [5].

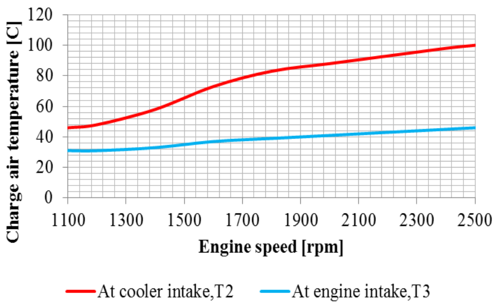

Fig. 3. *Air temperatures versus speed*

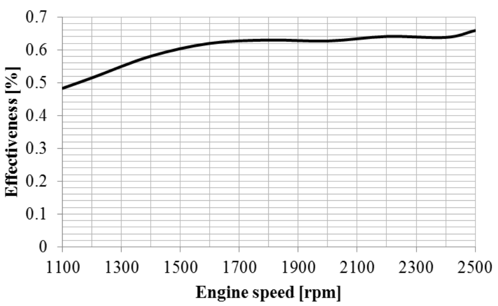

Fig. 4. *Cooler effectiveness versus speed*

Figure 4 illustrates the values of cooler effectiveness which meets the recommended range for speeds higher than 1500 rpm.

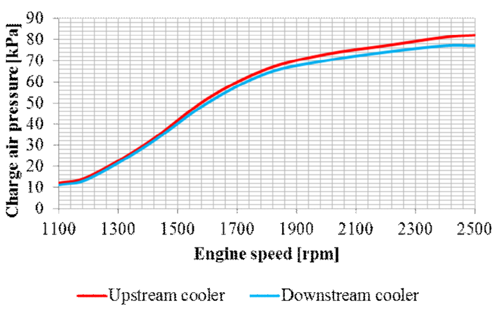

Fig. 5. *Air pressures versus speed*

The air charge pressure loss varied in the range of 0.007-0.049 bar, as it can be seen in Figure 5, the mean value being 0.025 bar.

According to literature [5] the maximum pressure loss for motor vehicle engine is 0.1 bar, confirming the fact that the air cooler does not introduce too much pressure loss. The measured exhaust temperature downstream turbine plotted in Figure 6 confirms the usefulness of turbocharging, being lowered in average with 63 °C.

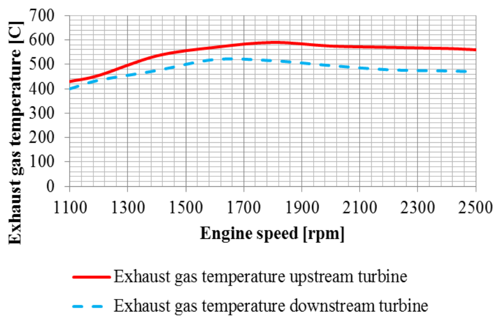

Fig. 6. *Exhaust temperature versus speed*

The reduction of air charge temperature in combustion chamber implies the lowering of exhaust gas temperature, as Figures 3 and 6 show, predicting also lower NO_x emission in exhaust gas.

B. Engine speed characteristics at total load required the measurement of torque, speed and smoke and the calculation of power, hourly and specific fuel consumption. The results of the tests are presented having as reference the performances of the basic engine 798-05, in Figures 7-11.

The engine power is represented in Figure 7, showing significant increase of rated power.

The engine torque gain is evident and its profile shown in Figure 8 indicates a better torque rise, the engine accomplishing its work more rapidly.

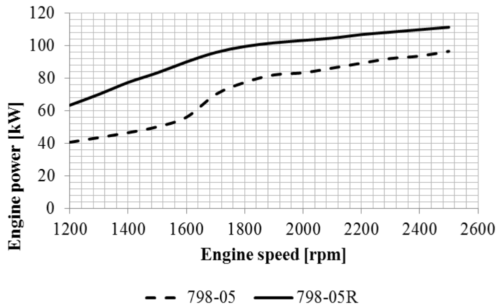

Fig. 7. *Engine power outputs at full load*

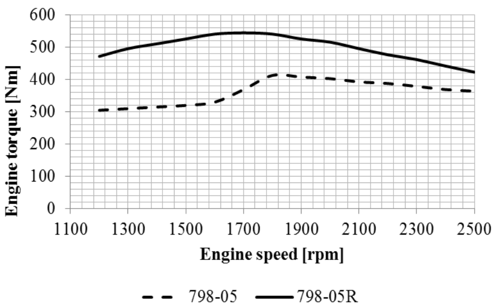

Fig. 8. *Engine torque outputs at full load*

The increase of hourly fuel consumption seen in Figure 9 is natural as the gain of power is higher.

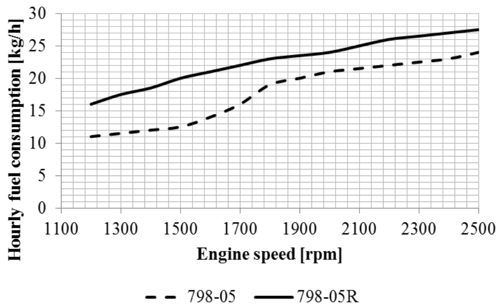

Fig. 9. *Hourly fuel consumptions*

The real indicator of engine efficiency is the specific fuel consumption, plotted in Figure 10, which indicates for the intercooled engine 9 g/kWh reduction and a shift towards lower speeds corresponding to maximum torques.

The smoke emission was measured according to prescription of ECE R24 Regulation [9] which indicates limit values

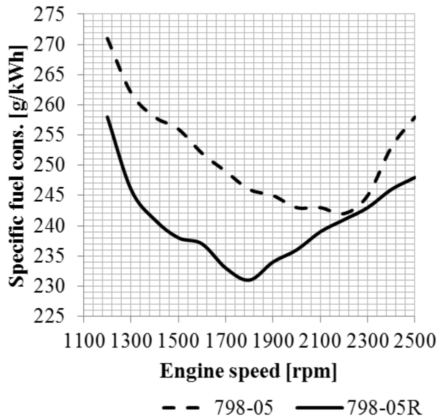

Fig. 10. *Specific fuel consumptions*

in function of engine exhaust gas flow. Figure 11 illustrates a fair reduction of smoke emission.

As smoke emission at 1400 rpm is pretty closer to the threshold value, further research work is required for optimisation of engine operation at low speeds, introducing more air by means of modified turbocharger housing or waste gate.

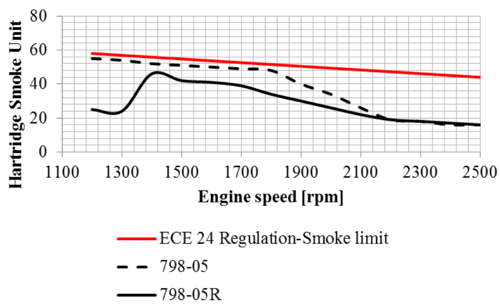

Fig. 11. *Engine smoke emissions*

4. Conclusions

The testing of the newly designed air-to-air cooler proved to be appropriate for engine intercooling in terms of effectiveness and pressure loss.

By mild intercooling the newly developed engine - turbocharged and intercooled (798-05R) demonstrated 15 kW higher rated power (15%), 100 N·m higher maximum torque (30%) and 9 g/kWh

(5.7%) specific fuel consumption lower than the turbocharged version (798-05), at lower smoke emissions.

Acknowledgements

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