Improving performance parameters of combustion engine for racing purposes

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Abstract

Polonec T., Janoško I., 2014. **Improving performance parameters of combustion engine for racing purposes**. Res. Agr. Eng., *60*: 83–91.

Mechanical parts of stock engine have a performance reserve which could be utilized when the engine is used under the race conditions. Especially normal turbocharged engines have their performance parameters designed to drive in traffic, where a good flexibility, reliability, fuel consumption and a long service life is required. It is possible to utilize the whole power of the engine, when changing or modifying some of its external parts and achieve better performance parameters without modifying or changing internal engine components. Performed changes must be realized thoughtfully and on the admittable level, so the engine and other drive train components would not be damaged. In our study we design several changes of external parts of engine which have a significant impact on the improvement of engine performance parameters. Their contribution has been verified in practice by an engine dynamometer.

Keywords: engine; performance parameters; turbocharger; roller dynamometer

It is generally known that the engine performance is substantially dependent on the amount of air (oxygen), which enters to the combustion chamber. Turbocharged engines are using turbochargers or compressors to increase the amount of induced air. The most commonly used charging system turbocharger powered by kinetic energy of exhaust gases (Ferenc 2004; Sloboda et al. 2008; Čupera, Šmerda 2010; Hromádko et al. 2010;).

There are several ways to increase power of turbocharged engine:
– increase of engine displacement,
– increase of turbocharger's boost pressure and airflow,
– decrease of intake air temperature (behind turbocharger),
– reduction of mechanical and airflow losses,
– optimization of intake and exhaust manifolds,
– optimization of combustion processes by sophisticated motor-management.

The purpose is to design the best solutions to improve the performance of normal supercharged engine, to improve acceleration of the vehicle as much as possible, to verify these modifications by measurements on roller dynamometer, to assess their contribution and propose other solutions to achieve even better results.

Supported by the Scientific Grant Agency VEGA of the Ministry of Education of the Slovak Republic and Slovak Academy of Sciences, Grant No. 1/0857/12 and by the Scientific Grant Agency KEGA of the Ministry of Education of the Slovak Republic and Slovak Academy of Sciences, Grant No. 044SPU-4/2014.

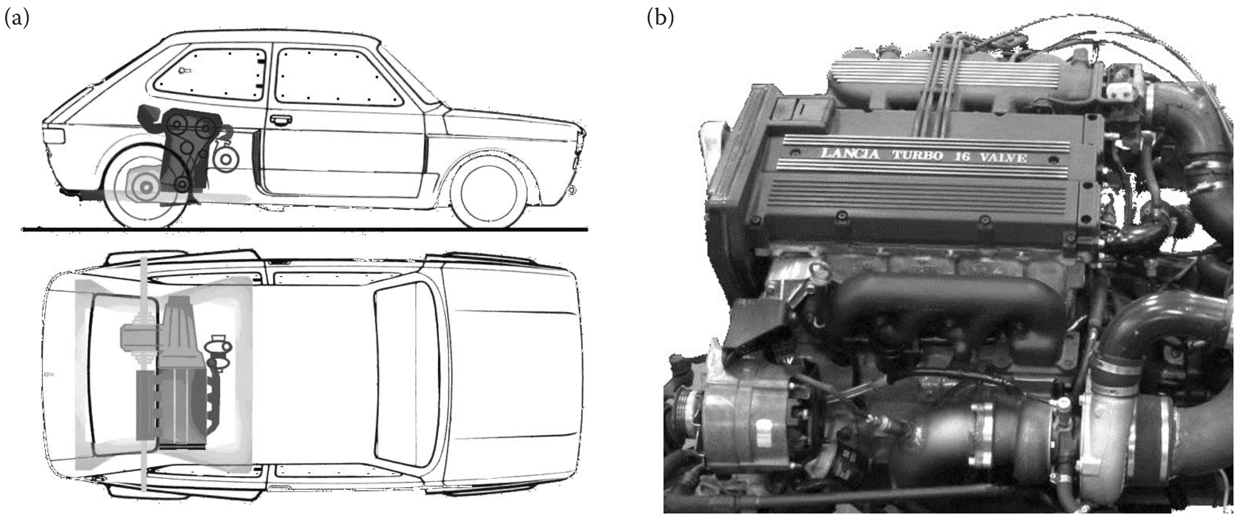


Fig. 1. Measured vehicle (a) Fiat 127A and engine (b) Lancia 2,0 16V Turbo

MATERIAL AND METHODS

Measured vehicle. Performance measurement was done on a special prototype vehicle designed for a drag race (Janoško, Polonec 2011). The base of the vehicle was bodywork of Fiat 127A (Fiat Auto S.p.A., Torino, Italy). As the power unit Lancia 2.0 16V Turbo engine was used, which was placed in the vehicle across, front of rear axle (Fig. 1).

The vehicle was two-door hatchback, with frameless steel body. Total weight of vehicle without driver was 830 kg.

A powerful engine from Lancia Thema, made by Fiat Auto S.p.A., Italy was used. It was petrol engine with charging by turbocharger. Displacement of engine was 1,995 cm^3 (bore: 84 mm, stroke: 90 mm). Engine had 4 cylinders in-line block with 16 valve DOHC head. Compression ratio is 8:1. Max. power of stock engine was 147 kW at 5,500 min^{-1} and torque 298 Nm at 3,750 min^{-1}.

Fuel delivery was provided by simultaneously multi point port fuel injection, controlled by electronic control unit Bosch LE2 – Jetronic (Robert Bosch GmbH, Gerlingen, Germany). Ignition was fully electronic, "wasted-spark" type, controlled by electronic control unit Magneti Marelli MED 601E (Magneti Marelli S.p.A., Corbetta, Italy).

Calculation of suitable turbocharger. In the calculations of suitable turbocharger we took account of future application of the vehicle in races and we defined a max. engine power to 300 kW at 6,000 min^{-1}. The best turbocharger for the intended use of vehicle was calculated using the following relations (Estill 2008). Substituting the results into compressor maps we have chosen the best turbocharger Turbo Tech 103 (Honeywell International Inc., Morris Township, USA).

Airflow needed to achieve the performance target:

$$Q_v = P_m \times \lambda \times Sp_{\mathrm{SB}} \tag{1}$$

where:

Q_v – airflow (kg/min)
P_m – performance target (kW)
λ – air/fuel ratio (–)
Sp_{SB} – brake specific fuel consumption (kg/kW·min)

Required absolute manifold pressure to achieve performance target:

$$p_{\mathrm{ABS}} = \frac{Q_v \times R \times (255.6 + T_p)}{\eta_{\mathrm{VOL}} \times \frac{n}{2} \times V_m} \tag{2}$$

where:

p_{ABS} – required absolute manifold pressure (kPa)
Q_v – airflow (kg/min)
R – gas constant
T_p – intake manifold temperature (°C)
η_{VOL} – volumetric efficiency (–)
n – engine speed (min^{-1})
V_m – engine displacement (cm^3)

Compressor discharge pressure:

$$p_{2C} = p_{\mathrm{ABS}} + \Delta p_{\mathrm{STR}} \tag{3}$$

where:

p_{2C} – compressor discharge pressure (kPa)
p_{ABS} – absolute manifold pressure (kPa)
Δp_{STR} – pressure loss between the compressor and the manifold (determined to 14 kPa)

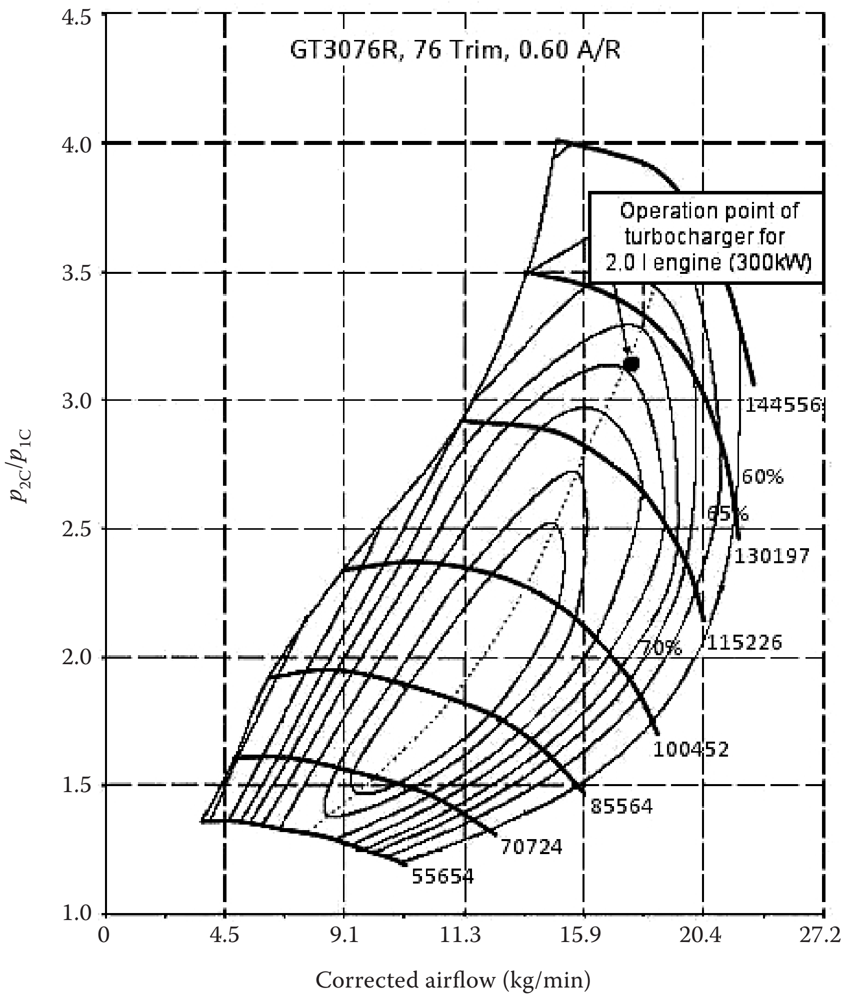


Fig. 2. Compressor map of Garrett GT3076R turbocharger

p_{2C}/p_{1C} – pressure ratio of inlet and outlet of turbocharger

Compressor inlet pressure:

$$p_{1C} = p_{\mathrm{ATM}} - \Delta p_{\mathrm{STR.S}} \tag{4}$$

where:

p_{1C} – compressor inlet pressure (kPa)
p_{ATM} – ambient air pressure (at sea level) (kPa)
$\Delta p_{\mathrm{STR.S}}$ – pressure loss in air filter and piping (determine to 7 kPa)

Pressure ratio:

$$\Pi_{\mathrm{TD}} = \frac{p_{2C}}{p_{1C}} \tag{5}$$

where:

Π_{TD} – pressure ratio
p_{1C} – compressor inlet pressure (kPa)
p_{2C} – compressor discharge pressure (kPa)

Based on calculations of operating parameters and substituting them into various compressor maps we chose a Garrett GT3076R turbocharger (Honeywell International Inc., Morris Township, USA). As seen on the compressor map (Fig. 2), this turbocharger is the most suited to performance requirements and the expected use of the vehicle for racing purpose.

Calculation of theoretical injectors fuel flow:

$$Q_P = \frac{Q_V}{\lambda_N} \times \rho_P \tag{6}$$

where:

Q_P – flow of fuel (kg/min)
Q_v – flow of air (kg/min)
λ_N – numerical value of lambda (–)
ρ_P – fuel density (kg/m^3)

Because of lower heat stress of injectors we calculated with approximately 80% duty cycle. Considering this duty cycle RC Racing injectors with fuel flow 750 cm^3/min were chosen (at 300 kPa fuel pressure).

Calculation of heat ratios in the intercooler (Estill 2008).

$$W = U \times S \times \frac{(\Delta T_1 - \Delta T_2) \div F}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{7}$$

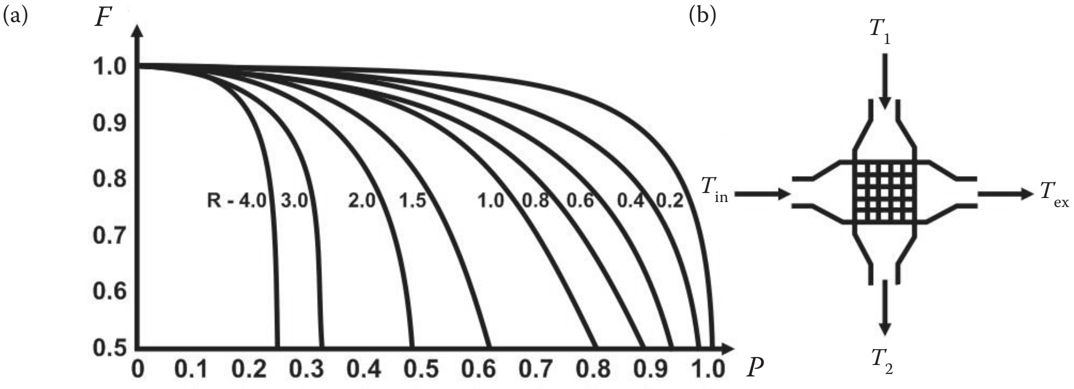


Fig. 3. Heat diagram of correction factor F (a) and intercooler temperature scheme of inlet/outlet flow (b)
F – correction factor (–); P – temperature ratio (–); T_1 – outside (cooling) air temperature on inlet (°C); T_2 – outside (cooling) air temperature on outlet (°C); T_{in} – compressed air temperature on inlet (°C); T_{ex} – compressed air temperature on outlet (°C)

where:
W – total transfer of heat energy (J)
U – heat transfer coefficient (W/m^2·K)
S – heat transfer surface (m^2)
ΔT_1 – difference between intercooler input air temperature and temperature of cooling air behind intercooler (T_{in} – T_2) (°C)
ΔT_2 – difference between output air temperature from intercooler and temperature of cooling air in front of intercooler (T_{ex} – T_1) (°C)
F – correction factor

Determination of the correction factor *F*. Correction factor F, taking into account the unequal distribution of heat at exchanger area, could be read from the diagram according to the calculated values of temperature ratios of P and R (Fig. 4). For calculation of temperature ratios P and R we need to know temperature of compressed air (T_{in}, T_{ex}) and cooling air (T_1, T_2) on the inlets and outlets of the intercooler (Fig. 3).

$$P = \frac{T_{ex} - T_{in}}{T_1 - T_{in}} \qquad (8)$$

$$R = \frac{T_1 - T_2}{T_{ex} - T_{in}} \qquad (9)$$

where:
P, R – temperature ratios
T_{ex} – compressed air temperature on outlet (°C)
T_{in} – compressed air temperature on inlet (°C)
T_1 – outside (cooling) air temperature on inlet (°C)
T_2 – outside (cooling) air temperature on outlet (°C)

Calculating the amount of lost or received heat on one side of exchanger:

$$W = Q_m \times C_P \times \Delta T \qquad (10)$$

where:
W – heat energy transfer (J)
Q_m – mass airflow (kg/min)
C_p – heat capacity of air (J/K·mol)
ΔT – difference of input and output temperatures (K)

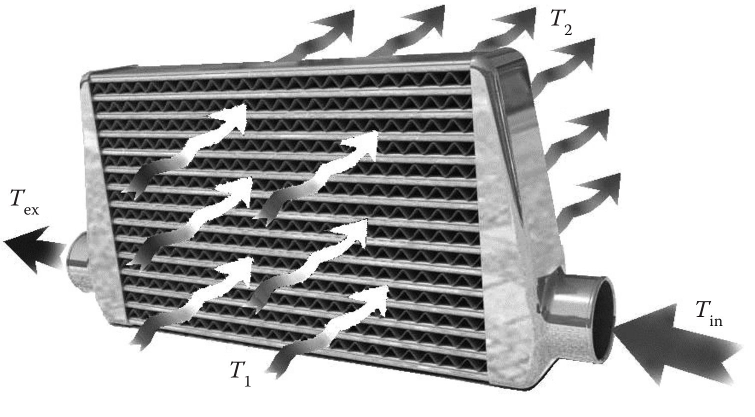


Fig. 4. Airflow through the intercooler
T_1 – outside (cooling) air temperature on inlet; T_2 – outside (cooling) air temperature on outlet; T_{in} – compressed air temperature on inlet; T_{ex} – compressed air temperature on outlet

RESULTS AND DISCUSSION

Performed engine modifications

The engine power can be measured by the dynamometer directly or through the power take-off shaft, or possibly on a roller bench or by the road-board test (Semetko, Janoško 2005). After performance measurement of stock engine the following modifications were made:

Stage 1:
- turbocharger replaced by more powerful type Garrett GT3076R, with rotor on ball bearings,
- injectors replaced by more powerful (RC Racing; RC Engineering, Inc., Higgins Court Torrance, USA) with fuel flow 750 cm^3·min (at fuel pressure 300 kPa),
- shortened exhaust system, removed mufflers,
- modified of sensing airflow by electronic control unit,
- intercooler placed in the box enabling cooling by ice,
- boost pressure controlled by manual boost controller.

Stage 2:
- stock electronic control units replaced by fully programmable unit VEMS ver. 3.6 (Acmv LLC, Wilmington, USA),
- intercooler replaced by "water to air type" with cooling by circulated water,
- intake manifold replaced by shorter type from Lancia Kappa,
- throttle body replaced by bigger one with internal diameter 73 mm,
- stock exhaust manifold replaced by custom steel manifold with pipes with diameter 42 mm,
- boost pressure controlled by electronic control unit with solenoid valve.

Dynamometer

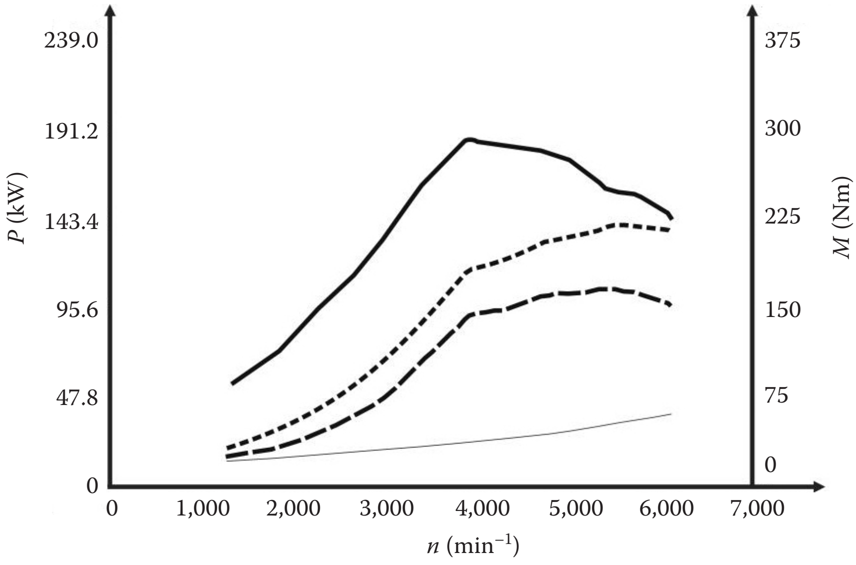
The measurements were performed on the roller dynamometer MAHA LPS 3000 PKW 4 × 4 (MAHA Maschinenbau Haldenwang GmbH & Co. KG, Haldenwang, Germany) (Fig. 5).

Parameters of dynamometer MAHA LPS 3,000 PKW:
- max. measurable output: 520 kW (4 × 4 version),
- max. measurable torque: 1,000 Nm,
- accuracy: +/– 2%,
- conversion of measured parameters according to technical norms.

Procedure of measuring power and torque on dynamometer

The vehicle will start to gradually speed up to 50 km/h, on penultimate gear. Then depress accelerator pedal to maximum and watch the course of performance on monitor up to reach a max. engine

Fig. 5. Positioning vehicle on dynamometer MAHA LPS 3,000 PKW



Performance data				External data	
Corrected power	P_{norm}	– – – – –	142.3 kW	Air temperature	15.5°C
Engine power	P_{mot}	—— —— ——	141.6 kW	Aspirated air temperature	19.1°C
Wheels power	P_{kolo}	————	107.7 kW	Relative air humidity	55.4%
Power losses	$P_{\text{ztráty}}$	▬▬▬▬	34.6 kW	Air pressure	1,008.0 hPa
Torque	M_{norm}		292.5 Nm	Vapour pressure	9.8 hPa
Max. torque at	n		3,905 min^{-1} 126.1 km/h	Oil temperature	16°C
Max. rpm			6,120 min^{-1} 197.6 km/h	Fuel temperature	–

Fig. 6. Measurement of power and torque before engine modifications

speed. When max. speed is reached, the technician pushes the clutch pedal and simultaneously releases accelerator pedal. The wheels are left free to catch up to 0 km/h. Dynamometer now records power loss. The waveform of power, torque and losses are saved to the memory of dynamometer.

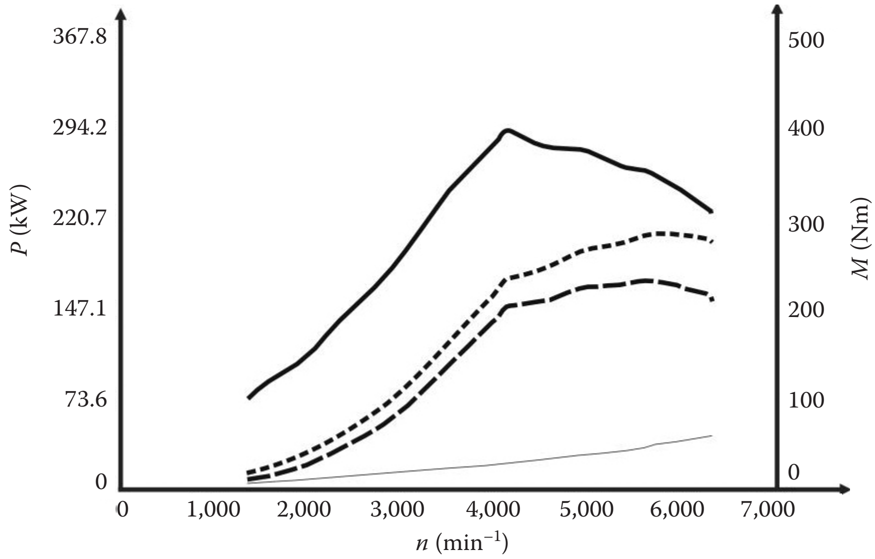
Measurement No. 1 – Stock engine

First measurement was used to determine the initial state. Engine with stock technical specifications without any performance modifications was installed in vehicle. Intake air temperature (in front of air filter) according to the testing laboratory was 19.1°C, relative humidity 55.4% and atmospheric pressure 1,008 hPa.

During the verification measurements slightly lower max. values, than those declared by manufacturer were recorded, but this was attributable to higher mileage and therefore wear of the engine. However, the measured values of power and torque were not significantly different from the manufacturer's values. Max. measured power [adjusted according to DIN 70020 (1993)] was 142.3 kW and torque was 292.5 Nm (Fig. 6).

Measurement No. 2 – First level of modifications

The second measurement was performed after the "Stage 1" of engine modifications (see "Performed engine modifications" section). Conditions of measurement were almost unchanged. According to the testing laboratory intake air temperature (in front of air filter) was 18°C, relative humidity was 56.6% and the atmospheric pressure was 1,007.9 hPa. Intercooler was cooled by ice with the temperature of approx. 0°C. As it is seen on the graph of the second measurement (Fig. 7), the recorded power was 210.2 kW and the torque was 400.3 Nm. An increase



Performance data				External data	
Corrected power	P_{norm}	– – – – –	210.2 kW	Air temperature	15.5°C
Engine power	P_{mot}	—— —— ——	209.5 kW	Aspirated air temperature	19.1°C
Wheels power	P_{kolo}	————	171.1 kW	Relative air humidity	55.4%
Power losses	$P_{ztráty}$	▬▬▬	38.4 kW	Air pressure	1,008.0 hPa
Torque	M_{norm}		400.3 Nm	Vapour pressure	9.8 hPa
Max. torque at	n		4,130 min^{-1} 133.4 km/h^1	Oil temperature	16°C
Max. rpm			6,330 min^{-1} 204.5 km/h	Fuel temperature	–

Fig. 7. Measurement of power and torque after "Stage 1" engine modifications

of performance over the first measurement was 67.9 kW and an increase of torque was 107.8 Nm.

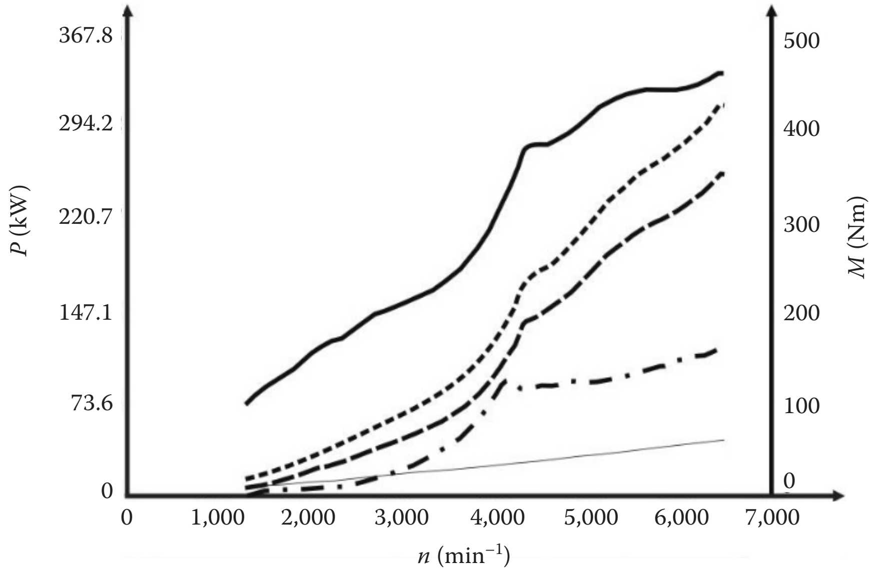
This increase of power and torque was affected mainly by mechanical modifications of external parts of engine, exchange of turbocharger with a more powerful one and an increase of intercooler cooling efficiency.

Measurement No. 3 – Second level of modifications

The third measurement was performed after "Stage 2" engine modifications (see "Performed engine modifications" section). According to the testing laboratory intake air temperature (in front of air filter) was 22.3°C, relative humidity was 33.7% and atmospheric pressure was 982.4 hPa. As it is shown in the graph of the third measurement (Fig. 8), there is one more shape (grey colour), which shows the boost pressure level. Since using new engine electronic control unit, we were able to control the boost level with electronic control unit and solenoid valve. Max. boost pressure was now set to 1.6 bar, but because we wanted to protect transmission system against high torque peaks, boost pressure was increased slowly from 1 bar (at approx. 4,000 rpm) to 1.6 bar at max. engine speed. This allowed driver to utilize engine max. power and torque in high revolutions, which is very useful in race conditions.

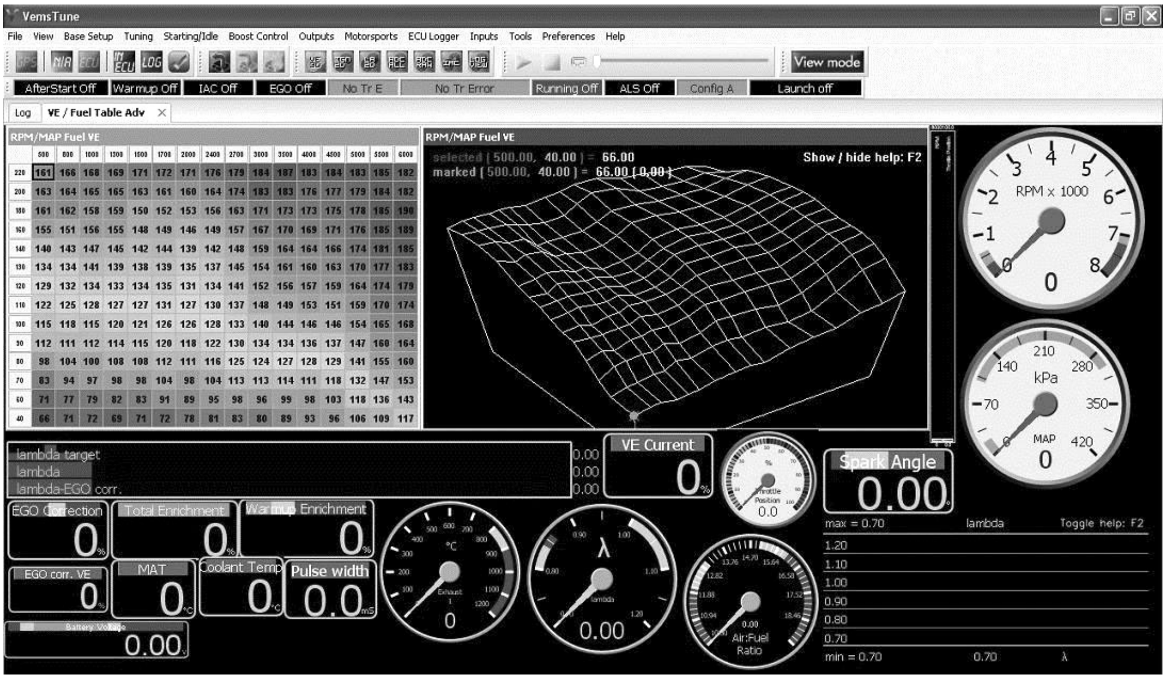
Max. measured power (adjusted according to DIN 70020 (1993)) was 310.7 kW and torque was 457.9 Nm. Increase of performance over the previous (Stage 1) measurement was 100.5 kW and increase of torque was 57.6 Nm. But more interesting is the increase of max. power over stock engine, which was 168.4 kW and for max. torque it was 165.4 Nm. In percentage, it is an increase of more than 118% of max. power and more than 56% of max. torque.

This increase of power and torque mechanical modifications of intake and exhaust system, exchange of intercooler and also exchange of the



Performance data				External data	
Corrected power	P_{norm}	– – – – –	310.7 kW	Air temperature	20.5°C
Engine power	P_{mot}	—— —— ——	300.1 kW	Aspirated air temperature	22.3°C
Wheels power	P_{kolo}	———	256.3 kW	Relative air humidity	33.7%
Power losses	$P_{ztráty}$	━━━	43.8 kW	Air pressure	982.4 hPa
Torque	M_{norm}		457.9 Nm	Vapour pressure	8.1 hPa
Max. torque at	n		6,470 min^{-1} 202.8 km/h[1]	Oil temperature Fuel temperature	19°C –
Max. rpm			6,495 min^{-1} 203.6 km/h	Manifold air pressure (hPa)	-—-—-—-

Fig. 8. Measurement of power and torque after “Stage 2” engine modifications



RPM/MAP Fuel VE

	500	800	1000	1300	1500	1700	2000	2400	2700	3000	3500	4000	4500	5000	5500	6000
220	161	166	168	169	171	172	171	176	179	184	187	183	184	183	185	182
200	163	164	165	165	163	161	160	164	174	183	183	176	177	179	184	182
180	161	162	158	159	150	152	153	156	163	171	173	173	175	178	185	190
160	155	151	156	155	148	149	146	149	157	167	170	169	171	176	185	189
140	140	143	147	145	142	144	139	142	148	159	164	164	166	174	181	185
130	134	134	141	139	138	139	135	137	145	154	161	160	163	170	177	183
120	129	132	134	133	134	135	131	134	141	152	156	157	159	164	174	179
110	122	125	128	127	127	131	127	130	137	148	149	153	151	159	170	174
100	115	118	115	120	121	126	126	128	133	140	144	146	146	154	165	168
90	112	111	112	114	115	120	118	122	130	134	134	136	137	147	160	164
80	98	104	100	108	108	112	111	116	125	124	127	128	129	141	155	160
70	83	94	97	98	98	104	98	104	113	113	114	111	118	132	147	153
60	71	77	79	82	83	91	89	95	98	96	99	98	103	118	136	143
40	66	71	72	69	71	72	78	81	83	80	89	93	96	106	109	117

RPM/MAP Fuel VE

selected (500.00, 40.00) = 66.00
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Fig. 9. Modifying and logging data in programme VemsTune

stock electronic control units by fully programmable unit, allow us to optimise all necessary control parameters of ignition timing and fuel injection.

Tuning of electronic control unit

At the first and second measurements, engine was controlled by stock control units only with modified airflow sensing system for adjusting the correct air/fuel ratio at all driving conditions.

Before measurement No. 3, we installed new, fully programmable electronic control unit VEMS. This unit allowed us to modify all necessary data to tune up main control parameters of the engine. Modification of the parameters is possible in real time, without stopping the engine or disconnecting control unit. Tuning was done in the official computer programme VemsTune (Acmv LLC, Wilmington, USA), which allowed us to change engine management parameters and also to log actual data from all engine sensors (Fig. 9).

CONCLUSION

This paper presents some options and methodology for upgrading the turbocharging and fuel injection system to increase engine power, which can also be used (with some modifications) in other types of turbocharged transport and agriculture vehicles to increase their dynamical parameters or working efficiency.

From the measured values it can be seen what benefits to an increase of the engine performance parameters modifications of engine components performed. During the measurements, we found an increase of the max. engine power, compared to the serial status, up to 168 kW and max. torque up to 165 Nm. This increase is mainly due to an exchange of turbocharger for a more powerful one, optimising thus intake and exhaust manifolds and optimising the fuel and ignition system for new engine setup.

Since we designed the turbocharger to be able to supply the necessary amount of air to the engine power up to 300 kW and engine is currently reaching more than 310 kW, we reached our goal. The manufacturer recommended this type of turbocharger also for engines with power more than 400 kW, but if we want to maintain the reliability at this high performance level, it will be necessary to exchange all internal components of engine by stronger, high quality forged parts in future. Next step must be modification of the transmission system and clutch, because now these parts are on their performance peaks.

To achieve higher performance parameters of the engine in future, it will be necessary to set up boost pressure of turbocharger to higher level, upgrade camshafts and also to upgrade the fuel system.

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Received for publication May 22, 2012
Accepted after corrections September 20, 2012

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