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Original scientific paper

Application of VNT Turbocharger in Spark Ignition Engine with Additional Expansion of Exhaust Gases

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Abstract: The paper provides an analysis of a VNT-type turbocharger implementation influence on operating parameters of the spark-ignition engine with additional expansion of exhaust gases. Previously, the research of the engine was performed with a charging system composed of a turbocharger with a simple waste-gate valve. The research was done mainly in order to improve the engine's performance for lower rotational speed range. The second reason of the described modification was to obtain lower fuel consumption. The four series of tests with boost pressure rising gradually from 0 to 0.8 bar were conducted for rotational speed of range 2000 - 3200 rpm. The described analysis of the obtained results showed that application of the VNT-type turbocharger caused significantly improved performances of the engine, especially in a lower rotational speed region. Also, decrease in the brake specific fuel consumption of the spark ignition engine with additional expansion of exhaust gases has been achieved for the rotational speed of value 2400 rpm.

Keywords: additional expansion of exhaust; efficiency; five-stroke spark ignition engine; turbocharger; variable nozzle turbine

1 INTRODUCTION

Certainly, it can be stated that from the very beginning of the development of combustion engines, the improvement of their total (fuel conversion) efficiency was one of the main goals of the inventors and designers [1].

One of the ways to improve total efficiency of internal combustion engine is increasing of expansion ratio of exhaust gases in a relation to compression ratio of air-fuel mixture [2]. Theoretically, there is a possibility to apply such an expansion ratio, which gives the pressure of exhaust gases equal to the ambient pressure at the end of the power stroke – Fig. 1. It causes increasing of the energy recovery ratio from the exhaust gases leaving the engine to the atmosphere.

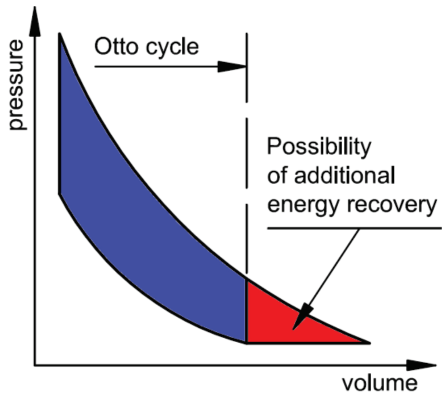

Figure 1 Possibility of energy recovery in a process of additional expansion of exhaust gases

There are several ways to implement this concept. In the first place, the solution of English engineer James Atkinson should be mentioned here [3]. In the second half of the 19^{th} century, he constructed at least two different internal combustion engines with specific crank mechanisms. Using of this complex crank mechanism caused that piston stroke during power and exhaust was significantly higher than in intake and compression. Thanks to this solution, Atkinson achieved significantly lower fuel consumption in its engines in comparison with the engines of other inventors from the end of the 19^{th} century.

A different way to apply increased expansion ratio in the four-stroke engine was developed by Ralph Miller [4] in a half of the 20^{th} century. He proposed to decrease an effective compression ratio through a significantly delayed closing of the intake valves. Caused by that, a lower value of volumetric efficiency was compensated by using a supercharger driven from the crankshaft of the engine.

Currently, the expansion ratio increased above the effective compression ratio is used in an increasing number of internal combustion engines, mostly to improve fuel conversion efficiency [5].

2 ENGINE WITH ADDITIONAL EXPANSION PROCESS

A completely different idea has been developed by a Belgian inventor Gerhard Schmitz [6]. In the year 2003 his invention of a so called five-stroke engine was patented in the United States. In this three cylinder in-line engine the outer cylinders perform classic 4-stroke cycle. Exhaust gases do not leave to the atmosphere but through the internal channels in a cylinder head go alternately to a middle cylinder in which they are further expanded. It gives additional work on the crankshaft. The scheme of the concept described above is presented in Fig. 2.

The middle cylinder in which additional expansion of exhaust gases occurs has approximately two times higher displacement in comparison with each of the fired cylinders. This fact gives the correct process of additional expansion. An engine, which is the subject of invention, was developed and tested by the English company Ilmor Engineering in the year 2007. This engine was equipped with a turbocharger, what allows reaching high performance together with low fuel consumption.

It should be mentioned here that the design of five-stroke engine proposed by G. Schmitz was not the first engine ever in which additional expansion of exhaust gases was applied. The double expansion of exhaust gases was used in several engines designed in the late nineteenth. Among others Verbundmotor from Deutz Company and Compound-Motor designed by Rudolf Diesel are relatively well-known and described [8]. These designers took a pattern of the solution previously known in steam engines. The prototype internal combustion engines with additional expansion were built, but many problems occurred during their exploitation and generally they did not meet the expectations placed in them. Most of the problems were related to the high thermal load of the valves and cylinders of additional expansion, which, taking into account contemporary materials and production technology, did not give any chance for trouble-free and efficient operation of such engines. Due to these facts, the concept has not found use for about one hundred years.

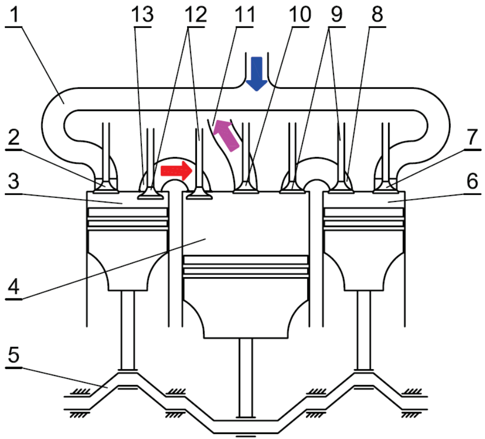

Figure 2 Scheme of the five-stroke engine [7]:
1 – Intake manifold, 2 and 7 – Intake valves, 3 and 6 – Fired cylinders, 4 – Additional expansion cylinder, 5 – Crankshaft, 8 and 13 – Channels connecting fired cylinders with additional expansion cylinder, 9 and 12 – Valves opening passage between fired and additional expansion cylinders, 10 – Exhaust valve, 11 – Exhaust channel

An engine designed according to a concept similar to the project of Ilmor Engineering, but constructed based on an existing in-line 4 cylinder engine was developed at the Cracow University of Technology in years 2012-2014 [9].

A general view of this engine mounted on the test stand is shown in Fig. 3.

Figure 3 General view of the engine with additional expansion of exhaust gases developed at the Cracow University of Technology

As a basis of this construction, a turbocharged EA113 VW engine of displacement 1984 cm^3 was used. In the engine presented above 1st and 4th cylinder perform classic 4 stroke cycle and after that exhaust gases are provided to internally connected cylinders 2nd and 3rd to perform the additional expansion process. Only after that exhaust gases get into the turbocharger. It makes that displacement of this engine equal to a half of the primary value what gives 992 cm^3. Technical data of the tested engine are presented in Tab. 1.

Table 1 Technical data of the tested engine [10, 11]

Parameter	Value/Description
No. of cylinders	2 fired / 2 for additional expansion
Displacement of fired cylinders, cm^3	992 (both)
Displacement of add. exp. cyl., cm^3	992 (both)
Bore, mm	82.5
Stroke,	92.8
No. of valves	4 per cylinder
Compression ratio, -	10.5
Overall expansion ratio, -	21
Fuel system	Direct Injection
Injection pressure, MPa	7
Engine Management System	AEM EMS 30-1010
Injector driver	Denso 131000-1041
Ignition system	individual coils
Fuel	Petrol, RON 98
IVO/IVC fired cylinders	7° ATDC / 17° ABDC
EVO/EVC fired cylinders	28° BBDC / 8° BTDC
IVO/IVC additional exp. cylinders	28° BTDC / 8° BBDC
EVO/EVC additional exp. cylinders	28° BBDC / 8° BTDC

Various tests of the engine developed at the Cracow University of Technology were made. Some results of this research can be found in the papers [7, 9, 10, 12, 13] and dissertation [14]. In the recent years simulations of operation of the similar engine are also conducted at the Shanghai Jiao Tong University by Li et al. [15]. Gerhard Schmitz also continues development of his engine in a new research team. In the work [16] promising results of experiments of the new five-stroke engine were presented. Authors plan to optimise their engine to operation in a range-extender for electric vehicle.

3 MOTIVATION TO USE A VNT-TYPE TURBOCHARGER IN RESEARCH-ENGINE

The results analysis from the previous research has shown that used turbocharger was not selected fully properly. The applied turbocharger came from a SI engine with displacement of 1600 cm^3. On the one hand, larger turbocharger gives lower exhaust backpressure, what is beneficial in terms of energy recovery in an additional-expansion process. On the other hand, boost pressure may be too low at low engine rotational speed. Generally, turbocharger selection for an engine, in which exhaust gases are after double expansion, is not obvious. Classical approach to this matter may not give expected result. A set of technical data of the turbocharger applied primarily in the engine with additional expansion process is presented in Tab. 2. Boost pressure generated by the compressor is controlled with an integrated turbine bypass valve (waste-gate).

During the test at WOT (wide-open throttle) a problem with obtaining the assumed value of the boost pressure has occurred. Below the rotational speed of 2800 rpms, the boost pressure could not reach 0.3 bar with fully closed waste-gate valve, what can be seen in Fig. 4. At the rotational speed of 2800 rpms and above the energy of exhaust gases after the additional expansion was already sufficient to hold the boost pressure at the value of 0.3 bar.

Table 2 Technical data of the primarily used turbocharger

Parameter	Value/Description
Model/No.	KP 39, 54399700122
Manufacturer	BorgWarner
Boost pressure control method	waste-gate valve, normally closed
Rotational speed, rpm	up to 210 000
Turbine inlet diameter, mm	34.5
Turbine outlet diameter, mm	33
Turbine rotor outer diameter, mm	38.5
No. of blades of turbine rotor	9
Compressor inlet diameter, mm	35.7
Compressor outlet diameter, mm	34
Compressor rotor outer diameter, mm	47.3
No. of blades of compressor rotor	5 short / 5 long
Waste-gate seat diameter, mm	25

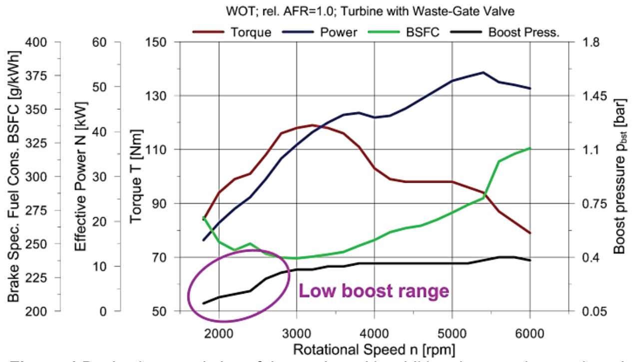

Figure 4 Basic characteristics of the engine with additional expansion equipped with a turbine with Waste-Gate Valve

Table 3 Technical data of the VNT-type turbocharger

Parameter	Value/Description
Model	BV35-54359710015
Manufacturer	BorgWarner
Boost pressure control method	variable geometry of turbine nozzle
Rotational speed, rpm	up to 260 000
Turbine inlet diameter, mm	29.5
Turbine outlet diameter, mm	32
Turbine rotor outer diameter, mm	34.3
No. of blades of turbine rotor	9
Compressor inlet diameter, mm	27.5
Compressor outlet diameter, mm	25.6
Compressor rotor outer diameter, mm	38.9
No. of blades of compressor rotor	5 short / 5 long

As it is clearly visible in the figure above, there is an area near the value of the rotational speed of 2400 rpm, where brake specific fuel consumption reaches the increased value in relation to minimal value obtained at the rotational speed of about 3000 rpm. It was assumed that there is a possibility of reduction of the brake specific fuel consumption for lower values of the rotational speed through the use of another turbocharger, which can give increased boost pressure. It can also give increase in engine's output. Due to relatively low exhaust gas temperature after additional expansion process [12], it is possible to use a turbocharger with Variable Nozzle Turbine.

This kind of turbocharger gives good potential in control of engine output. BV35-type BorgWarner turbocharger used in 1250 cm^3-class compression-ignition engine was chosen. Specifications of the BV-35 turbocharger are presented in Tab. 3.

In Fig. 5 a view of the VNT turbocharger mounted in the research engine is shown.

Figure 5 BV35 VNT-type turbocharger mounted to the engine with additional expansion of exhaust gases

An assembly of the VNT turbocharger required development of a re-designed exhaust manifold and exhaust pipe. Intake pipes of the research engine were also replaced. The control system of variable nozzle system uses vacuum as a working medium instead of boost pressure in the previous charging device. For this reason a PWM-controlled 3-way electromagnetic valve was replaced by another one.

4 RESULTS OF EXPERIMENTAL RESEARCH

The experimental tests were conducted for the four values of the engine rotational speed: 2000, 2400, 2800 and 3200 rpm. During each test, the throttle was fully open. Relative Air-Fuel Ratio was held on the value 1.0 using wide-band oxygen sensor with an appropriate controller cooperating with engine management system.

Main specifications of the used measuring equipment are presented in Tab. 4.

Table 4 Main specifications of the used measuring equipment

Parameter	Value/Description
Dynamometer type	Eddy-current
Maximum absorbed power, kW	100
Maximum rotational speed, rpm	10 000
Maximum torque, Nm	200
Force sensor type	strain-gauge
Rotational speed sensor type,	variable reluctance
Boost pressure sensor type	piezoresistive
Boost pressure sensor rage, bar	0 - 2.5
Fuel consumption meter type	gravimetric
Method of fuel consumption metering	mass of fuel consumed in a defined time

First measurement was for fully open turbine nozzle system. Then boost pressure was increased for a value 0.1 bar up to the value of 0.8 bar. The spark advance angle was set at about 2 degrees of crank angle before the knock limit for each of the measurement points.

In Fig. 6 the results of brake specific fuel consumption (BSFC) and boost pressure as a function of engine torque obtained at a constant rotational speed 2000 rpm are shown. Red and white points mark values of BSFC and related boost pressure obtained with the turbocharger with waste-gate valve.

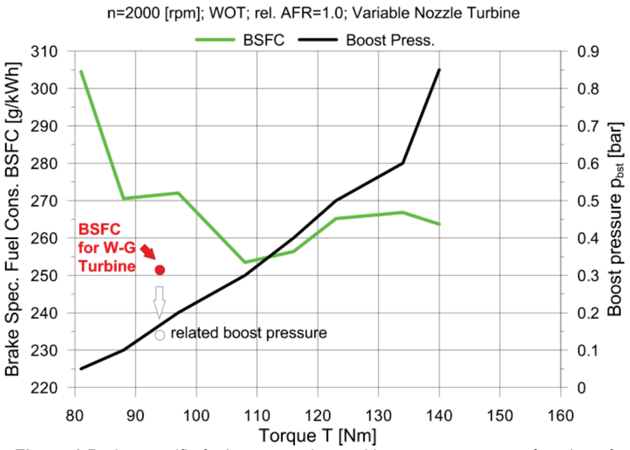

Figure 6 Brake specific fuel consumption and boost pressure as a function of engine torque obtained at rotational speed of 2000 rpm

Maximum torque reaches 140 Nm obtained at boost pressure equal to 0.83 bar. Minimal BSFC equals 253 g/kWh is only slightly higher than it was with turbocharger with waste-gate valve (250 g/kWh) for the same rotational speed (2000 rpm).

In Fig. 7 results obtained at rotational speed 2400 rpm are presented.

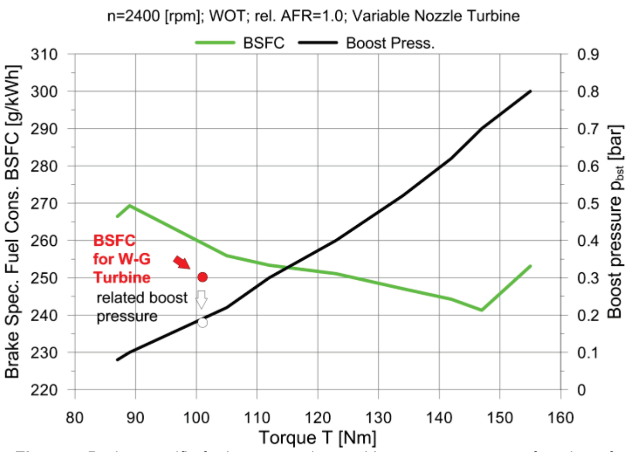

Figure 7 Brake specific fuel consumption and boost pressure as a function of engine torque obtained at rotational speed of 2400 rpm

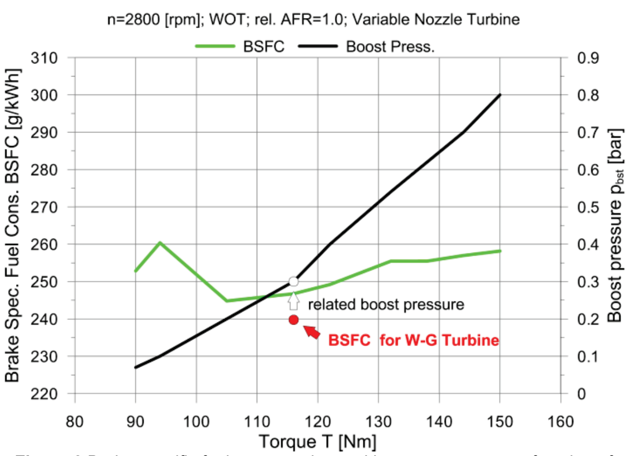

Figure 8 Brake specific fuel consumption and boost pressure as a function of engine torque obtained at rotational speed of 2800 rpm

At this operating point decreasing of BSFC was obtained. Minimal value equals about 241 g/kWh and was obtained at a load of 147 Nm. It means 9 g/kWh decrease of BSFC obtained at this rotational speed in comparison with the result for the engine equipped with the turbocharger with waste-gate valve. Maximum torque reaches at this rotational speed a value of 155 Nm obtained at boost pressure equal 0.8 bar. It was the maximum obtained torque in all observed measuring points.

In the next Fig. 8 the results of BSFC and related boost pressure in a function of torque obtained at rotational speed 2800 rpm are presented.

In the case presented above a minimal value of BSFC is again slightly higher in comparison to the engine with turbocharger without variable nozzle turbine. The difference equals about 5 g/kWh. The torque with VNT-type turbocharger increased from 116 Nm to 150 Nm.

Results recorded for rotational speed of 3200 rpm are shown in Fig. 9.

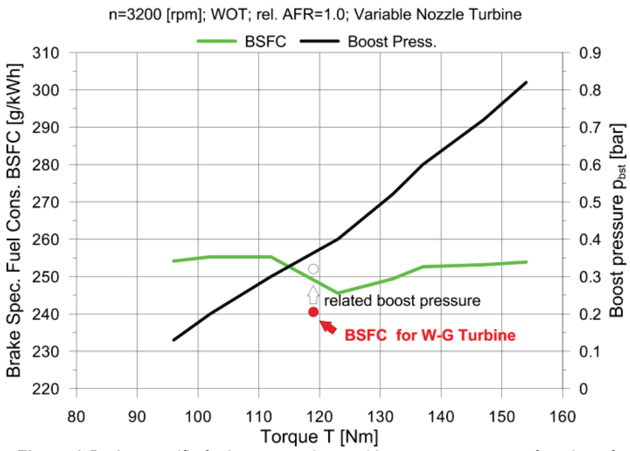

Figure 9 Brake specific fuel consumption and boost pressure as a function of engine torque obtained at rotational speed of 3200 rpm

At rotational speed of 3200 rpm engine generates a value of torque equal to 154 Nm at boost pressure 0.82 bar. The minimal value of BSFC is higher by about 5 g/kWh what is similar to the above mentioned case.

5 ANALYSIS OF THE OBTAINED RESULTS

Basing on the research results shown above, it was possible to make two comparisons for the engine with additional expansion equipped with two types of turbocharger:

- in relation to generated torque,
- in relation to obtained brake specific fuel consumption.

A comparison of torque generated by the research engine equipped with turbocharger with waste-gate valve and variable nozzle turbine as a function of rotational speed is presented in Fig. 10.

The maximal increase of the torque has been recorded at rotational speed 2400 rpm. The torque increased from 101 Nm to 155 Nm what means a relative increase for over 50% of the initial value. For the other values of the rotational speed significant increase in torque values was also obtained.

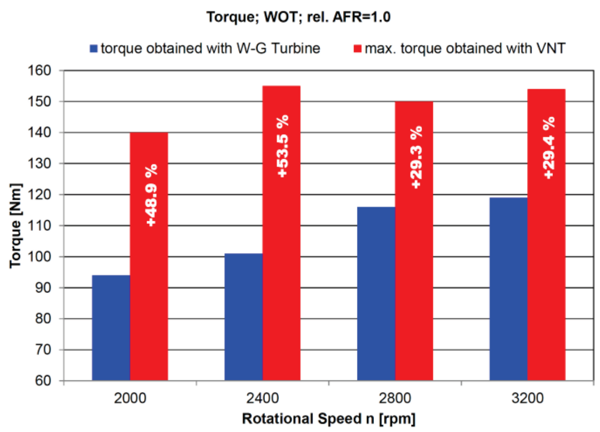

Figure 10 Comparison of torque generated by the research engine equipped with turbocharger with waste-gate valve and variable nozzle turbine as a function of rotational speed

In Fig. 11 a comparison of value of brake specific fuel consumption obtained by the research engine equipped with turbocharger with waste-gate valve and variable nozzle turbine as a function of rotational speed is shown.

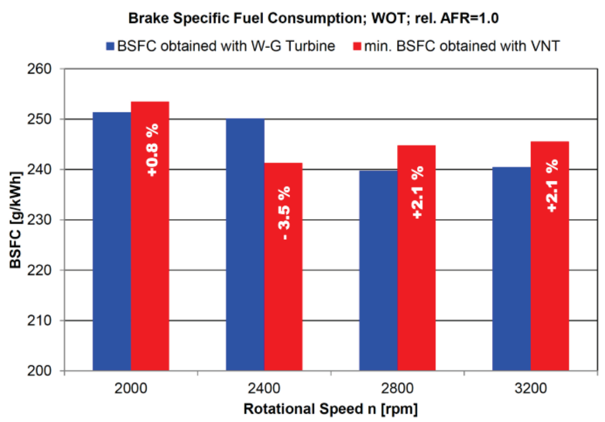

Figure 11 Comparison of brake specific fuel consumption obtained by the research engine equipped with turbocharger with waste-gate valve and variable nozzle turbine as a function of rotational speed

At rotational speed 2400 rpm a 3.5% decrease of brake specific fuel consumption was obtained. At the other values of rotational speed a relatively low increase in the minimal value of brake specific fuel consumption was noted.

6 CONCLUSIONS

Detailed analysis of the obtained results allowed formulating the following conclusions:

- application of the turbocharger with variable nozzle turbine caused an increase of the boost pressure obtained in the range of engine rotational speed of 2000 – 3200 rpm, for each selected operating point the boost pressure value of 0.8 bar was reached, some problems with stability of the engine operation occurred above this value which is caused by compressor operation close to a surge line,
- the increase in boost pressure caused an increase in the engine torque for each of the considered values of rotational speed, at rotational speed 2800 rpm torque increase reached over 50% in comparison with the previous value,
- at the engine rotation of 2400 rpm a 3.5% decrease in brake specific fuel consumption in relation to the engine equipped with turbocharger with waste-gate valve was obtained, for other rotational speed values the minimum brake specific fuel consumptions were higher by about 0.8-2.1% in comparison with the values obtained by the engine with the turbocharger with waste-gate valve, what may indicate lower efficiency of the implemented turbocharger operating at this points of the engine operation map; generally, it should be pointed out here that the lowest BSFC value obtained with VNT turbocharger was not lower than the lowest value recorded previously with W-G turbine,
- brake specific fuel consumption lines have minimal values obtained at one of the middle values of boost pressure. This fact is caused by the characteristics of VNT turbocharger and phenomena occurring in the engine with additional expansion whilst boost pressure increases.

To summarize, there is a possibility for the next research into the presented scientific problem. It concerns especially determining the exhaust emissions of the engine with the new turbocharging system. This will be the next step of development of the presented engine. The other possible ways of development of the engine are reducing of heat losses between fired- and additional-expansion cylinders and application of gaseous fuels, what may give here beneficial results [17], [18].

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Abbreviations and symbols

ABDC	–	after bottom dead centre,
ATDC	–	after top dead centre,
BBDC	–	before bottom dead centre,
BTDC	–	before top dead centre,
BSFC	–	brake specific fuel consumption, g/kWh
max.	–	maximal,
min.	–	minimal,
n	–	rotational speed, rpm
N	–	engine effective power, kW
p_{bst}	–	boost pressure, bar
PWM	–	pulse width modulation, %
rel.AFR	–	relative air-to-fuel ratio, -
RON	–	research octane number
T	–	engine torque, Nm
VNT	–	variable nozzle turbine
W-G	–	waste-gate valve
WOT	–	wide open throttle

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