CHARGING SYSTEM OF SPARK IGNITION ENGINE WITH TWO TURBOCHARGERS

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Abstract

The purpose of the investigation was the application of two turbochargers system in spark ignition engine and determining turbochargers' work parameters depending on throttle opening and engine's rotation speed. System with small turbocharger and larger variable geometry turbocharger in parallel connection (three-stage turbocharging) was examined. The engine used during the investigation was 1300 cm displacement SI engine with modified intake and exhaust manifolds. Intake and exhaust manifold modification including only implementation of turbochargers and sensors was done for experimental purposes. Specific values of maximum boost pressure were obtained by introducing a waste gate valve system with appropriate characteristic.

Proper choice concerning work parameters of the charging system allows to improve torque characteristic in wide range of engine's rotation speed. The system with additional small turbocharger allowed to increase torque value in low engine's speed range as well as to increase boost pressure in high engine's speed range for throttle opening angle values above 50 %. In medium engine's speed range the best results were given by variable geometry turbocharger. The two turbochargers system and values of maximum boost pressure were controlled by the system with two waste gate valves. Improving total efficiency was obtained in medium engine's speed range. The application of two turbochargers system as modification of naturally aspirated spark ignition engine allows to improve torque flexibility rate. There is a possibility to apply the charging system with two turbochargers, with boost pressure control system, in already existing, naturally aspirated engine without decreasing compression ratio and modifying engine's control system.

Keywords: *spark ignition engine, charging system, turbocharger*

1. Introduction

The increasing amount of car transportation units and shortcomings in energy resources as well as threatening environmental pollution require the reduction of energy consumption and generation of pollutants in transport. The development of a new environmentally friendly generation of internal combustion engines becomes a necessity.

Supercharging and downsizing is one of the most promising approaches to reduce the fuel consumption of spark ignition (SI) engines [1]. The power output of an internal combustion engine is proportional to the mean effective pressure, the speed, and the total piston displacement. An increase in the piston displacement results in a significant increase in engine weight, installation space and a deterioration in efficiency due to the increased friction loss. The mean effective pressure is proportional to the density of air, the effective efficiency, the volumetric efficiency and it is inversely proportional to the excess air factor. The density of air depends on the charge pressure and charge air temperature. The effective power output of the engine is significantly increased with the increase in the air density.

With the turbocharged SI engines, the higher charge pressure results in higher ultimate compression temperatures. This increases the risk of auto ignition and of knocking. For this reason, sometimes it is necessary to lower the compression ratio. High exhaust gas recirculation rates increase the risk of knocking, particularly in small engines, when exhaust pipe is in front of the turbine inlet. In part-load operation, the mass flow of turbocharged SI engines is throttled, and a bypass (open-air circulation plate) must be used around the compressor.

Quite important question seems to be as well the adaptation of fuel injection and ignition system. In case of race car engines lack of exhaust gases mass flow in low engine speed range is compensated by a considerable increment of torque and power in high speed range. Furthermore, high power of this type of engine allows the implementation of a system with one turbocharger. When taking into consideration low and medium displacement engine, that type of technical solution is not suitable and leaves a space for different supercharging systems. Due to sensitivity of SI engine to knocking, there is a need to limit boost pressure or modify engine control unit (ECU) with specific approach to engine's load, boost pressure, in-cylinder mixture temperature and quality of fuel.

In turbocharged engines the delay appears and this is the reaction to changing torque demands. Turbochargers produce substantial boost pressure only after having reached high rotation speeds. One of several approaches which have been suggested to improve work parameters of the engine at low and medium rotation speed is a supercharging system with two turbochargers in parallel configuration with a fast small and efficient large device.

2. Technical features of the engine and experimental setup

The object of the experimental tests was an engine with modified intake end exhaust manifolds. The intake and exhaust manifold modification including only implementation of turbochargers and sensors was done for experimental purposes. The characteristics of the used engine have been described in Tab. 1. The naturally aspirated engine without decreasing compression ratio was used.

Tab. 1. Engine basic characteristics (naturally aspirated engine)

Type of engine	Spark ignition engine
Number of cylinder	4 in line
Bore	72 mm
Stroke	79.6 mm
Maximum power	64 kW at 6000 rpm
Maximum torque	121 Nm at 4000 rpm
Compression ratio	11.0 : 1
Number of valves	4 per cylinder

Two turbochargers in parallel connection were used in a charging system: small and larger variable geometry turbocharger. The two turbochargers system and values of maximum boost pressure were controlled by the system with two waste gate valves and a control valve (Fig. 1).

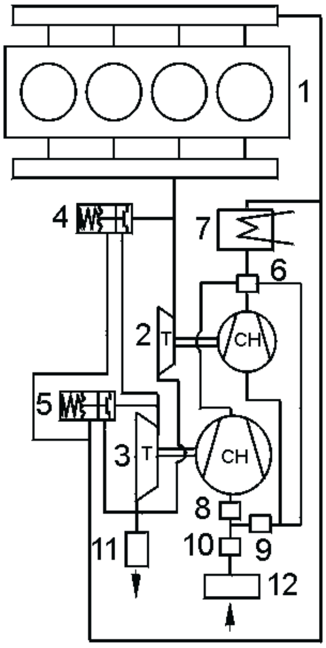

Fig. 1. Scheme of system with two turbochargers, 1 - spark ignition engine, 2 - small turbocharger, 3 - variable geometry turbocharger, 4, 5 - waste gate valve, 6 - control valve, 7 - intercooler, 8, 9 - check valve, 10 - air mass sensor, 11 - catalytic converter, 12 - air filter

In order to achieve a sufficient charge pressure at low engine speed, a small turbine was chosen with small neck cross section.

The work of turbochargers depended on pressure in the intake manifold. In low and medium engine speed range, at low load of engine, only the small turbocharger (2) was used. At medium load of engine, in medium and higher engine speed range the waste gate (4) directed exhaust gas to the larger turbine (3). Interaction of both compressors was controlled by three-way valve (6). The air flow in the intake manifold was controlled by plate check valves (8, 9). To limit the associate component load, the charge pressure was controlled to a constant value by allowing the excess exhaust gas enthalpy stream to bypass the turbine (waste gate 5). At higher load of engine in high engine speed range, the work of turbocharger with the variable turbine geometry (3) was supported by a small turbocharger (2). The additional air cooler was used in the intake system (7).

All tests were carried out at similar environmental conditions. The room with a test stand was air-conditioned which provided the right ambient temperature in the range of 293-298 K. The application of cooling air system enabled to maintain the work temperature of the examined charging system below 973 K. The temperature was measured by pyrometer which was mounted on the housing of the turbine. It was possible to keep the temperature of the cooling liquid within the range of 340-365 K because water-cooled heat exchanger was applied in the cooling system of the engine.

Scheme of the turbocharged engine test apparatus and measuring system is shown in Fig. 2. The test was equipped with a 100 kW eddy current dynamometer, controlled by the electronic system. The turbochargers of the test engine consisted of a radial turbine and centrifugal compressor. As it was shown in Fig. 2, the experimental apparatus was composed of the test engine, a dynamometer, control system of fuel, intake air and exhaust gas. Values of torque (M_d), power (P_e), admitted mass of fuel per unit of time (m_B), air-flow mass (m_1), air temperature (T_1), intake manifold air pressure (p_2), exhaust gas temperature (T_3), emission of CO, CO_2, HC, NO_x, and air-fuel ratio (λ) were measured during engine's test. Engine's coolant and turbine temperature, as well as ignition advance angle and fuel iniection time were constantly monitored during all tests. Apart from main work parameters of the engine were measured up - and downstream parameters of each turbocharger.

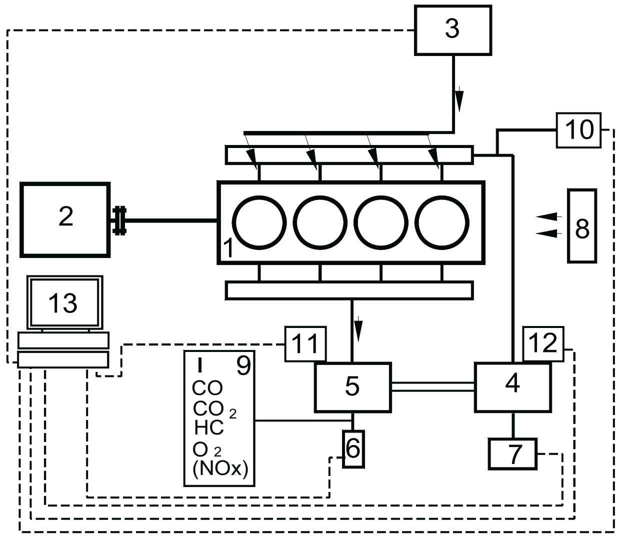

Fig. 2. Scheme of experimental apparatus, 1 - spark ignition engine with turbocharger system, 2 - dynamometer, 3 - fuel distribution system with measuring equipment, 4 - intake manifold with compressors of turbochargers, 5 - exhaust manifold with turbines of turbochargers, 6 - exhaust gas temperature sensor, 7 - air mass and temperature measurement equipment, 8 - cooling fan, 9 - exhaust emission measuring equipment, 10 - intake manifold pressure sensor, 11 - exhaust manifold temperature and pressure sensors, 12 - intake manifold temperature sensor, 13 - signal controller and data analyzer

3. The basic data for turbocharger

The basic data for turbochargers can be determined from the mathematical model. There are already models of different complexity in the literature [2, 3, 4, 5]. The turbocharger speed is set depending on the power balance between compressor and turbine:

$$\frac{d\omega_{TL}}{dt} \cdot J_{TL} \cdot \omega_{TL} = P_V \cdot P_T \,. \tag{1}$$

For the static state:

$$P_V + P_T = 0 \,, \tag{2}$$

$$\dot{m}_V + \dot{m}_B = \dot{m}_T \,, \tag{3}$$

and power is defined by equations:

$$P_V = \dot{m}_V \cdot \Delta h_{sV} \cdot \frac{1}{\eta_{sV} \cdot \eta_{mV}} \,, \tag{4}$$

$$P_T = \dot{m}_T \cdot \Delta h_{sT} \cdot \eta_{mT} \cdot \eta_{sT} \,. \tag{5}$$

Isentropic enthalpy gradient in compressor (Δh_{sV}) can be defined as:

$$\Delta h_{sV} = R_1 \cdot T_1 \cdot \frac{k_1}{k_1 - 1} \cdot \left[\left(\frac{p_2}{p_1} \right)^{\frac{k_1 - 1}{k_1}} - 1 \right], \tag{6}$$

and isentropic enthalpy gradient in turbine ((Δh_{sT}) can be defined as:

$$\Delta h_{sT} = R_3 \cdot T_3 \cdot \frac{k_3}{k_3 - 1}\left[1 - \left(\frac{p_4}{p_3}\right)^{\frac{k_3-1}{k_3}}\right]. \tag{7}$$

The group efficiency (η_{TL}) is defined as the overall efficiency of the charge system:

$$\eta_{TL} = \eta_{mV} \cdot \eta_{sV} \cdot \eta_{mT} \cdot \eta_{sT}, \tag{8}$$

The turbocharger main equation (with k_1 = 1.4) for compressor pressure ratio (p_2/p_1) can be defined by the equation:

$$\frac{p_2}{p_1} = \left[1 + \frac{\dot{m}_T}{\dot{m}_V} \cdot C_1 \cdot \frac{T_3}{T_1} \cdot \eta_{TL} \cdot \left(1 - \frac{p_4}{p_3}\right)^{\frac{k_3-1}{k_3}}\right]^{3.5}, \tag{9}$$

where:
C_1 - is constant.

If is assumed $\dot{m}_T / \dot{m}_V$ as known, compressor pressure ratio is a function of the group efficiency, exhaust gas counterpressure, downstream turbine pressure, turbine upstream temperature and compressor upstream temperature.

The turbine mass flow can be defined by the equation:

$$\dot{m}_T = A_T \cdot \psi_T \sqrt{2 p_3 \cdot \rho_3}, \tag{10}$$

where:
A_T - is a turbine equivalent cross section and flow function (ψ_T) can be defined as:

$$\psi_T = \sqrt{\frac{k_3}{k_3 - 1}} \cdot \sqrt{\left(\frac{p_4}{p_3}\right)^{\frac{2}{k_3}} - \left(\frac{p_4}{p_3}\right)^{\frac{k_3-1}{k_3}}}. \tag{11}$$

The pressure p_3 is obtained with a given turbine as a function of the mass throughput and gas state and it depends on engine speed, piston displacement, density downstream of the turbocharger system and turbine measurements.

4. Results and discussions

A test grid covering from 1000 to 6000 rpm, and from 25%, 50%, 75% and 100% throttle opening values was designed. Boost pressure had to be reduced to 0,35 bar in order to provide stable engine's run in all conditions including variable engine speed and the whole range of throttle opening angle with restricted fuel consumption. Results of the tests are showed in Fig. 3.

The results of the tests of engine with two turbochargers was compared with the results of tests of naturally aspirated engine and the engine with one turbocharger with variable geometry turbine [6].

Proper choice concerning work parameters of the charging system allows to improve torque characteristic in wide range of engine rotation speed. The boost pressure in low engine speed range was increased by 25 % throttle opening value (Fig. 4).

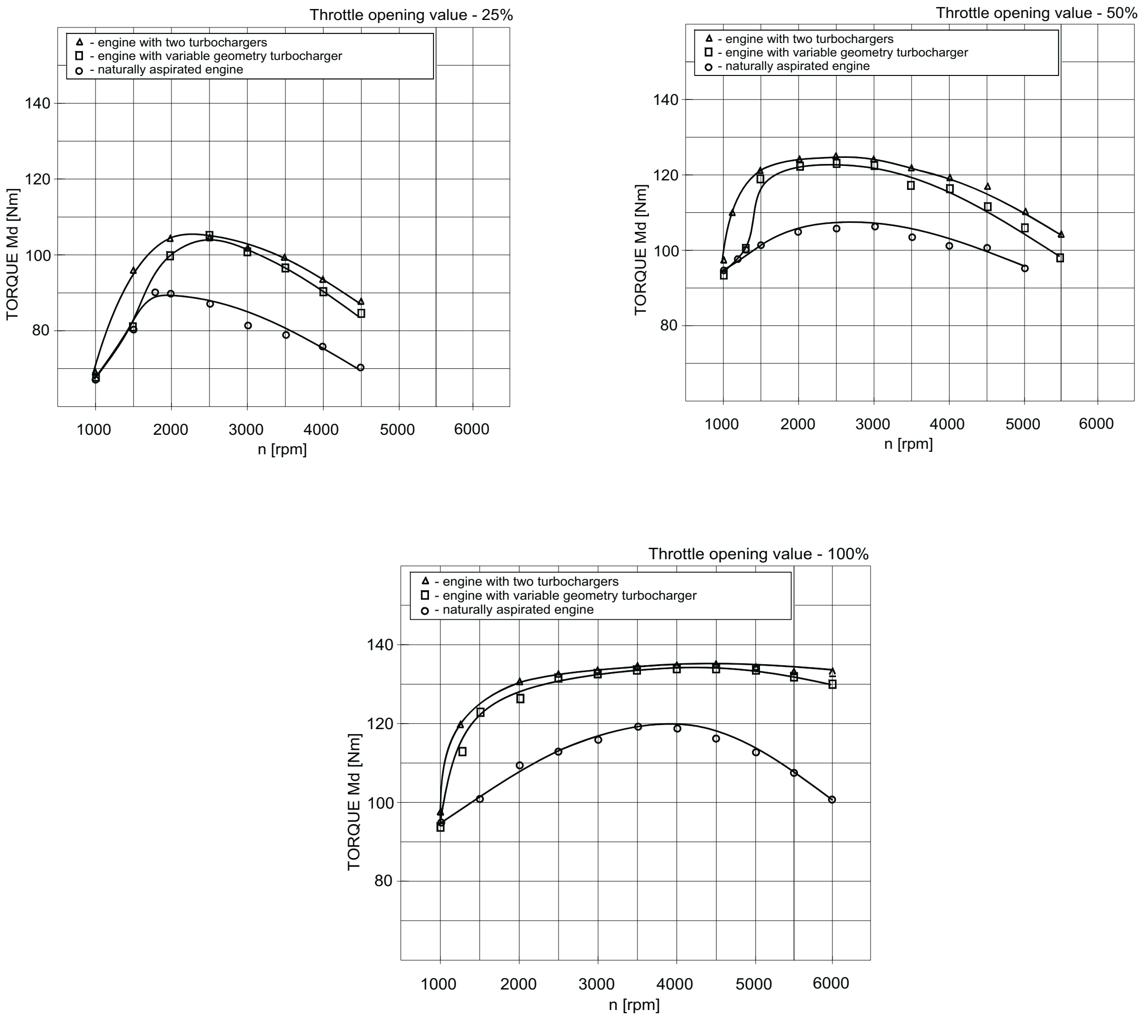

Fig. 3. Results of tests of engine with two turbochargers system, torque curve

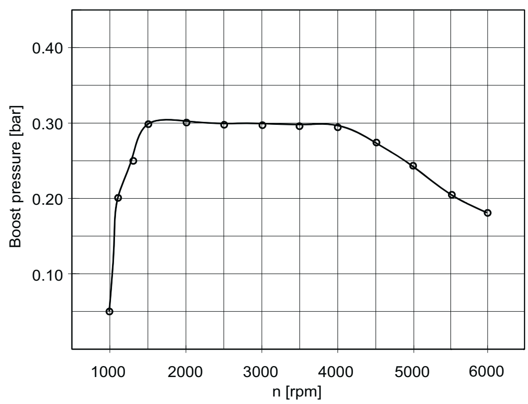

Fig. 4. Charge pressure curve

The system with additional small turbocharger allowed to increase significantly the torque value in low engine speed range as well as to increase boost pressure in high engine speed range for throttle opening values above 50^o.

Effective efficiency (η_e) of engine was calculated by formula:

$$\eta_e = \frac{P_e}{\dot{m}_B \cdot H_B}, \tag{12}$$

The effective efficiency in medium engine's speed range was much better for medium throttle opening values (50-75%). The curve of effective efficiency is shown in Fig. 5. In higher engine's speed range and for throttle opening values above 75%, the aim of increasing efficiency was not achieved due to higher fuel consumption.

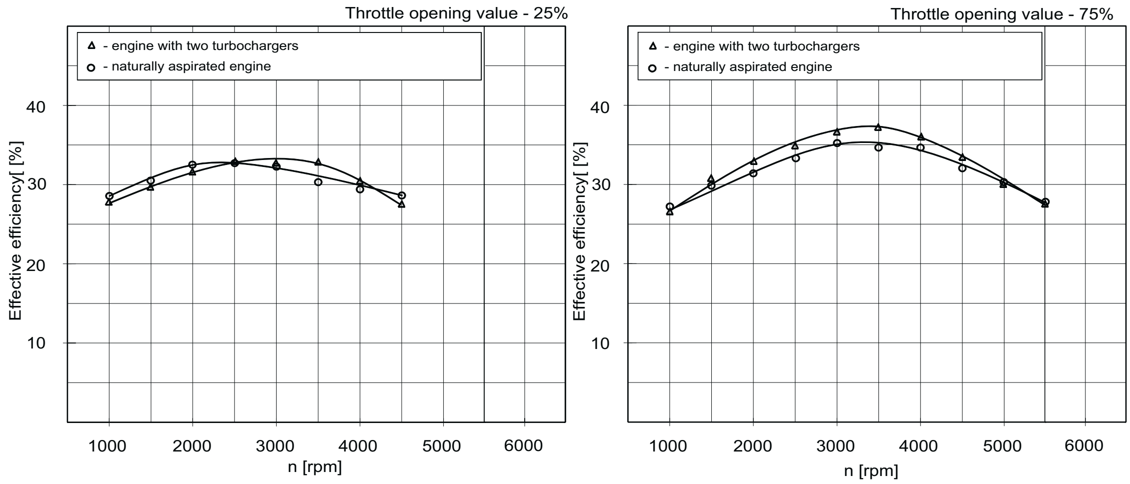

Fig. 5. Effective efficiency characteristics

The application of two turbochargers system as modification of naturally aspirated spark ignition engine allows to improve torque flexibility rate. There is a possibility to apply the charging system with two turbochargers, with boost pressure control system, in already existing, naturally aspirated engine without decreasing compression ratio and modifying engine's control system.

The application of changeable characteristics in the waste gate valve which reduces the charging pressure by controlling characteristics variable of the valve depending on the engine work parameters allows to use higher values of boost pressure within the range of mean values of the engine load. Pneumatic-mechanical controlling of the air streams from both compressors can be replaced by an electronically controlled system which makes it possible to reach better parameters of engine performance in transient states.

The work was the first approach to apply charging system with two turbochargers in small spark ignition engines without decreasing the compression ratio.

5. Nomenclature

J: Polar inertia moment
ω: Angular velocity
P: Power
p: Pressure
$\dot{m}$: Mass flow
T: Temperature
ρ: Density
R: Gas constant
η: Efficiency

Δh: Isentropic enthalpy gradient
ψ : Flow function
k: Isentropic exponent
H: net calorific value
Subscript
V: Compressor
T: Turbine
TL: Turbocharger
1: Compressor upstream
2: Compressor downstream
3: Turbine upstream
4: Turbine downstream
s: Isentropic
m: Mechanical
B: Fuel

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