

Characteristics of Turbocharger with TiAl Turbine Wheel in a Downsizing GDI Engine

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ABSTRACT

Steady and transient tests in a downsizing Gasoline Direct Injection (GDI) in-line 4 cylinders 2.0 liter engine were carried out to investigate characteristics of turbocharger with Titanium aluminide (TiAl) turbine wheel. The density of TiAl material is lower than Inconel 718 (Inconel) which is raw material for conventional turbine wheel. The objective of this study was to investigate the effect of light rotational inertia of turbine wheel on engine performance. Performance of TiAl turbine wheel turbocharger itself was also compared to that of Inconel turbine wheel turbocharger. Except for the turbine wheels, all experimental conditions were matched to be the same load and engine speed conditions. The compressor total-to-total pressure ratio of TiAl turbocharger was higher under part load condition due to higher turbocharger speed of TiAl turbocharger, which was led by lower rotational inertia of TiAl turbine wheel, while the engine performance was not much improved. In addition, brake specific fuel consumption (BSFC) was little deteriorated due to lower swallowing capacity of TiAl turbine. Under wide open throttle (WOT) conditions, the engine torque was slightly lower compared to Inconel turbocharger case. This was due to the same reason, which was lower swallowing capacity. However, difference of BSFC between both turbochargers was not big because of other affecting factors such as ignition timing and over fueling. In the transient test, the time for boost pressure build-up of TiAl turbocharger was shorter by up to 13.04 % than that of Inconel turbocharger. Turbocharger speed of TiAl turbocharger was also higher at the same time after onset of acceleration. In addition, the time for the reaching peak engine torque was shorter by at most 10.24% in case of TiAl turbocharger compared to that of Inconel turbocharger.

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INTRODUCTION

In present, improvement of fuel economy and reduction in exhaust gas emissions are trends of automotive industry due to depletion of fossil fuels and emission regulations. Automakers cannot sell their vehicles if they are not able to meet the stringent regulations in the near future. Due to the two challenges, the makers have developed technologies to improve fuel economy and to reduce the emissions. Downsizing technologies including direct injection and boosting are promising technologies to meet the regulations and better fuel economy. Downsizing engines are regarded to be superior to natural aspirated larger displace volume engine from heat, pumping, and mechanical frictional losses of perspectives [1]. Boosting technologies determine not only engine performance and emissions, so ways of boosting have to be carefully chosen; there are several boosting ways: supercharging, turbocharging, and electrical boosting. Among

types of boosting technologies, turbocharging is considered as the best way to improve fuel economy and emissions due to that it uses exhaust waste energy while supercharger uses energy from the engine.

Turbocharging and direct injection were technologies only for diesel engines before a couple of decades. Direct injection technology started to be adopted into gasoline engines in early nineties by Mitsubishi [2]. Turbocharging technologies were only for high performance cars in gasoline engine vehicles in the past. Even though turbocharging technologies are recently common technologies for gasoline engines, the level of turbocharger technologies of gasoline engines are lower than that of diesel engines. However, there have been efforts to increase level of turbocharging technologies for gasoline engines. The key requirements for advanced turbochargers are increased engine steady performance over a wide operation range for fun to driving and improved fuel economy. Recently, not only

improvements of steady, but transient performances, reduction in turbo lag, are general research trends of turbocharger. Adoption of a variable turbine geometry (VTG) turbocharger was one of the first attempts to increase both performances. VTG turbine changes vane angle fit in turbine flow condition, which leads to increase low end torque and reduce turbo lag. However, due to high exhaust gas temperature of gasoline engine, VTG turbochargers have been rarely installed into gasoline engines. Mahle developed integrated water-cooled exhaust manifold to reduce exhaust gas temperature and applied conventional VTG turbocharger of diesel engine [3]. VTG with water-cooled exhaust manifold improved transient response and performance at high engine speeds, but the engine torque was deteriorate at low engine speeds due to reduced exhaust gas temperature due to heat loss by coolant compared to conventional wastegate type turbocharger. In addition, Honeywell adopted an axial turbocharger, called as "Dual boost," instead of common radial type turbocharger to increase transient performance [4]. Axial type turbocharger operates with higher turbine efficiency below 0.7 of blade speed ratio [5]. In general, automotive turbine flow condition range of blade speed ratio is below 0.7, which means that the turbine efficiency of axial turbocharger can be higher than that of radial type turbocharger at most of operation points. Furthermore, the rotational inertia of axial turbocharger itself was also light. Due to these reasons, the time for boost pressure build up was shortened. However, steady performance was not improved compared to conventional radial type turbocharger. Adoptions of electric booster or two stage turbocharger were also developed to improve performances [6, 7]. The electric booster increased transient performance significantly, but consumed electric energy which worsened fuel economy. Sequential turbocharger including two-stage turbocharger consists of high pressure turbine for fast response and low pressure turbine for higher power density of the engine. The two-stage turbocharger increased both steady and transient performances. However, above four boosting configurations increased cost of vehicle and complexity of the system. Mixed flow turbine wheel was also a good attempt to increase transient performance [8]. The peak efficiency was obtained at lower blade speed ratio compared to conventional automotive radial type turbocharger. Above those attempts were related to rotational inertia of turbocharger. Rotational inertia is regarded as one of turbocharger parameters to determine engine and turbocharger performances especially for transient operation. Use of light material such as TiAl material for turbine wheel is also a way to improve turbocharger performance.

A TiAl turbine wheel turbocharger has lower rotational inertia compared to conventional Inconel turbocharger due to lower density of TiAl. In a previous research, Mitsubishi conducted vehicle test and found that low inertia TiAl turbine wheel turbocharger was superior compared to conventional Inconel turbocharger [9]. However, they focused on durability of TiAl material and introduced simple vehicle test

result. In addition, TiAl turbine wheel had better transient characteristics due to lower inertia even though TiAl turbine wheel shape was not optimized due to its low castability in a numerical study [8].

In this study, steady and transient tests in a turbocharged multi-cylinder engine were carried out to investigate the characteristics of low rotational inertia TiAl turbine wheel turbocharger. At first, turbocharger characteristics of both turbochargers were compared under steady state conditions. Thereafter, the effect of turbocharger characteristics on engine performance including fuel consumption, combustion phasing, and engine torque under various steady stage conditions was analyzed. Finally, effect of TiAl turbine wheel on transient performance of turbochargers and engine were revealed.

EXPERIMENTAL APPARATUS

Test Engine and Turbochargers

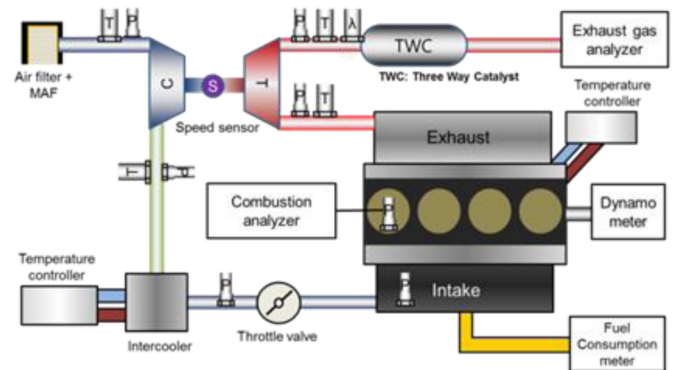


Figure 1. Experimental apparatus and sensor locations

A downsizing turbocharged gasoline direct injection (GDI) in-line 4 cylinder in-line engine from Hyundai motor company was tested on the steady and transient operation conditions. The tests were conducted on an engine test bench with an eddy current (EC) dynamometer. The engine was developed to replace conventional 2.4 L naturally aspirate GDI engine. The specification of the test engine is shown in table 1. The base engine has a twin-scroll type turbocharger with an Inconel turbine wheel and an electric wastegate actuator. In this study, to evaluate the effects of light inertia of turbine wheel, a TiAl turbine wheel was substituted for an Inconel turbine wheel. Table 2 shows the characteristics of both turbine wheel materials. Both materials are widely used as turbine blades and disks of jet engines and generators due to high thermal strength. However, density of TiAl material is lower than that of Inconel material. Due to this, the weight and inertia of rotational part of TiAl turbocharger are lower than those of conventional Inconel turbocharger. Furthermore, TiAl material has higher strength under high temperature condition [9]. However, due to low castability of TiAl material, the thickness of TiAl turbine wheel was thicker compared to Inconel one. To compensate this

disadvantage, the throat area of the wheel should be readjusted for the same flow capacity. Therefore, the shape of TiAl turbine wheel was little different from that of Inconel turbine wheel. However, turbocharger performances including flow capacity of both turbochargers were the same on a turbocharger test bench from result of turbocharger manufacturer.

Table 1. Specification of test engine

Engine type	In-line 4 cylinder
Displacement volume	1998 cc
Valve train	DOHC (Pneumatic CVVT)
Bore X stroke	86 X 86 mm
Compression ratio	9.5
Max power	278 PS @ 6,000 rpm
Max torque	37.2 kgfm @ 1750 ~ 4500 rpm

Table 2. Characteristics of turbine wheels

	Inconel	TiAl
Density	8190 kg/m ³	4430 kg/m ³
Total weight of rotational part	0.202 kg	0.149 kg
Total inertia of rotational part	261 x 10 ⁻⁷ kg m ²	198 x 10 ⁻⁷ kg m ²

Measurements and Data Acquisition

The sensors and their locations are shown in [figure 1](#). A Piezo electric pressure transducer (Kistler, 6052C) was installed into the 1st cylinder to measure in-cylinder pressure. A Piezo resistive pressure transducer (Kistler, 4045A) was used to measure intake pressure. The two pressure transducers were to analyze combustion characteristics of the engine. For the turbocharger measurements, pressures at the turbine inlet and outlet were measure by Piezo resistive pressure transducers (Kistler, 4045A and 4007B) with cooling adopters due to high exhaust gas temperature of the engine. Pressures at the compressor inlets and outlets were measured with Piezo resistive pressure transducers (Sensys, PSHH0003BAAG). K-type thermocouples were used to measure temperatures at inlet and outlet of both turbine and compressor. For air mass flow rate measurement, a hot wire type sensor (Bosch) was chosen for entire operation conditions of the engine. A turbocharger speed sensor generated one Transistor to Transistor Logic (TTL) signal per one turbocharger shaft revolution. Data acquisition board (IOtech, personal DAQ 3005) has 1 MHz of data acquisition frequency and 16-bit resolution. The acquisition frequency of each measurement was at least 50 kHz at steady state operation conditions. Signal from each sensor was treated and saved with lab-made program in DASYlab platform of

National Instruments. Measurement data stated above were saved from 101st cycle to 200th cycle after onset of measurement under steady state conditions. Representative performance index value for each operation condition was 100 cycles averaged under steady state conditions.

Engine and Turbocharger Performance Indexes

Engine performance parameters - the heat release rate was calculated by [equation \(1\)](#) during combustion stroke from Heywood [1]. With this equation, heat release rate per crank angle degree (CAD) was calculated. It was assumed that the heat losses from crevice and cylinder wall as well as blow-by effect were negligible, which meant the last three terms in [equation \(1\)](#) were not used to calculate heat release rate.

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma-1} P_c \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dP_c}{d\theta} - C_p T \frac{dm}{d\theta} + \frac{dQ_{ht}}{d\theta} + h_{cr} \frac{dm_{cr}}{d\theta} \quad (1)$$

Turbocharger performance parameters - total to total compressor pressure ratio was defined in [equation \(2\)](#). Because the turbocharger in this study was twin-scroll type, single averaged turbine inlet pressure in one scroll was not suitable to evaluate the turbocharger performance. Total to static turbine pressure ratio was calculated in [equation \(3\)](#) [10].

$$PR_{t-t_c} = \frac{P_{02}}{P_{01}} \quad (2)$$

$$PR_{t-s,t} = \frac{1}{2} (PR_{in} + PR_{out}) \quad (3)$$

$$\text{where } PR_{in} = \frac{P_{03-in}}{P_4}, PR_{out} = \frac{P_{03-out}}{P_4}$$

For the turbine swallowing capacity, sum of intake mass flow and amount of fuel injected was used instead of mass flow parameter because turbine inlet and outlet temperatures were similar in cases of both turbochargers at the same engine operation condition. Due to slow response of thermocouples, averaged temperature values were used. However, instantaneous temperatures were calculated with assuming adiabatic compression and expansion. This was shown to provide a good approximation of the measured dynamic temperature [11]. Calculated instantaneous performance parameters were used and post processed by ensemble averaging as representative value (Cycle averaged parameter). This cycle averaged parameter was more appropriate rather than equivalent quasi-steady parameters to represent turbocharger performance [12].