

Development of New Toyota D-Series Turbocharger for GD Diesel Engine

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ABSTRACT

There is increasing demand for highly functional diesel engine turbochargers capable of meeting Euro 6 emissions regulations while improving dynamic performance and fuel economy. However, since these requirements cannot be easily satisfied through refinements of existing technology, Toyota Motor Corporation has developed the new D-series turbocharger for initial installation in its GD diesel engine. The higher efficiency and wider operation range of the new turbocharger enabled the amount of the turbine flow capacity to be reduced by 30%, while helping to improve dynamic response and fuel economy. The mechanism causing the generation of fuel deposits in the fuel injection system upstream of the turbocharger, which was adopted for compliance with emissions regulations, was analyzed and fundamental countermeasures were applied. The result is a new highly functional turbocharger with greatly enhanced reliability.

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INTRODUCTION

As shown in Fig. 1, Toyota Motor Corporation began in-house development of turbochargers in 1982. Notable milestones since then include the development of mass-produced turbocharging systems featuring world-first technologies, such as the ceramic sequential twin turbocharger installed in the Celica and Supra. More recently, Toyota developed and began mass-production of its third generation C-series turbocharger for diesel engines in 2000 (1).

In addition to compliance with emissions regulations, diesel engines are also expected to be fuel efficient and realize fun-to-drive acceleration. To satisfy these requirements, it is necessary to develop high performance turbochargers with higher air flow rate and a lower surge limit flow (Fig.2). However, since this cannot easily be accomplished through refinements of existing technology, Toyota has developed the fourth generation D-series turbocharger to satisfy the demand for such a wider operation range.

This paper describes an outline of this turbocharger for installation on its GD diesel engine and the countermeasures developed for the particular fuel deposits (abbreviated to "deposits" below) that occur in the fuel injection system upstream of the turbocharger.

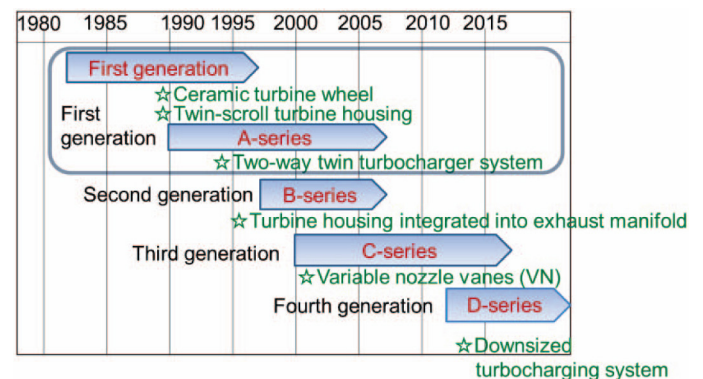


Figure 1. History of Toyota's in-house turbocharger development.

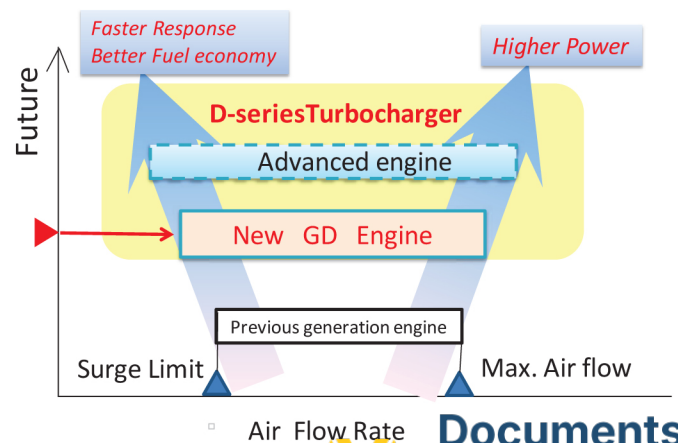


Figure 2. Demand for wider operation range.

OUTLINE OF TURBOCHARGER

Table 1 shows the specification of GD diesel engines(2). A main purpose for the GD engine development is to improve fuel economy and to improve low-end and maximum torque while keeping maximum power compared with previous generation engine. Therefore a demand for the new turbocharger is to achieve class-leading performance with high efficiency, a wide operation range and downsizing.

Table 1. Specification of GD diesel engine

Specification	1GD	2GD
Engine Type	Inline 4 Cylinder	Inline 4 Cylinder
Displacement	cm ³ 2755	2393
Bore x Stroke	mm ϕ 92 x 103.6	ϕ 92 x 90
Compression Ratio	15.6	15.6
Max Power	kW/rpm 130/3400	110/3400
Max Torque	N-m/rpm 450/16-2400	400/16-2000
Turbocharger	Variable Nozzle (CT8DV)	Variable Nozzle (CT7DV)

Fig.3 shows the structure of the CT8DV turbocharger developed for the 1GD engine.

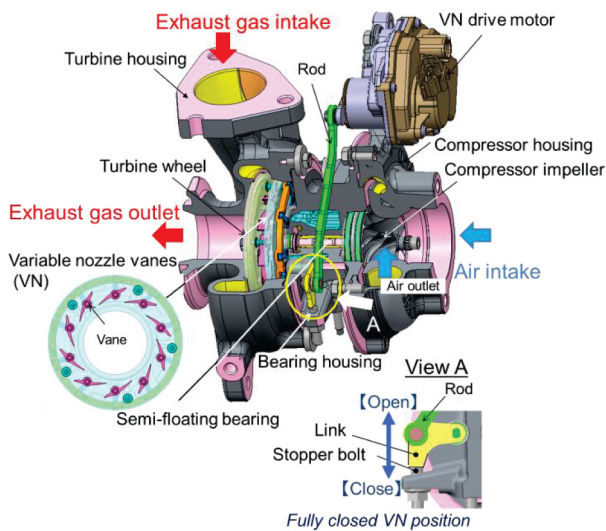


Figure 3. CT8DV turbocharger for 1GD engine

The design concept of compressor and a turbine wheel are crucial to satisfy such a high engine demand.

As for the compressor impeller, a gently bending blade design is newly adopted to avoid a flow separation along the blades by increasing the axial length. This enables to improve surging limit and efficiency optimizing the fluid performance by computational fluid dynamics (CFD). In addition, higher pressure ratio and turbo speed limit were achieved by replacing the conventional precision casting manufacturing method with a method that involves fully machining forged material with a 20% higher specific strength. This enables downsizing the impeller outlet diameter to get higher efficiency at low speed and designing thinner shape blades for reduction pressure loss.

Fig.4. shows an example of the new impeller design. The blade length of the new impeller is longer by 15% and the outlet radius is smaller by 12% compared to the previous design.

As for a turbine wheel, similar design method as for the impeller was adopted. In addition, Full-back disk design was adopted to reduce leak flow from back face with a new thinner shape that reduces stress(Fig.5).

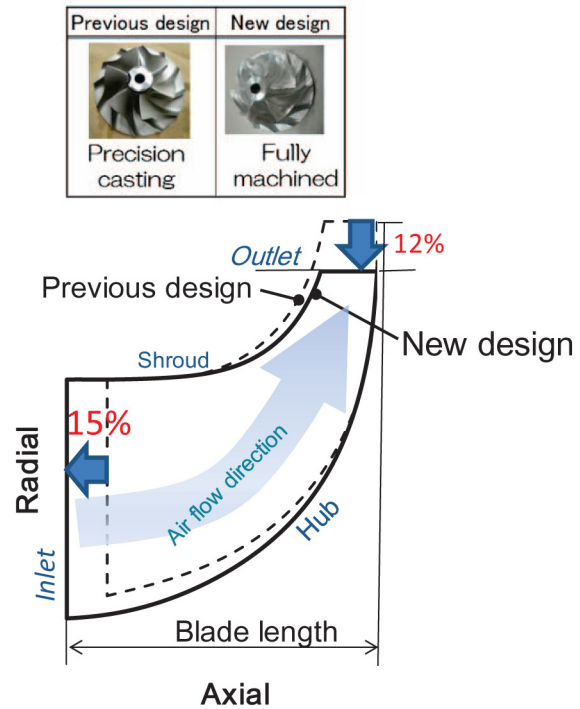


Figure 4. Comparison of compressor impeller design.

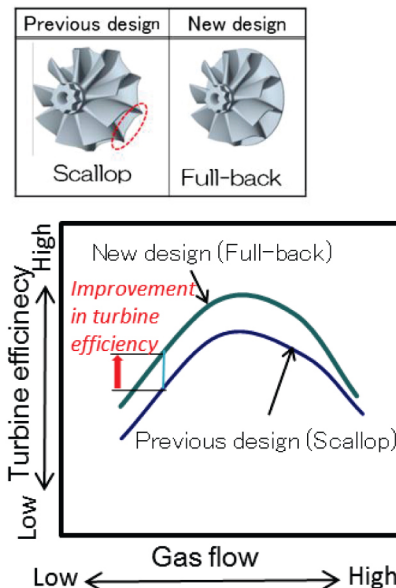


Figure 5. Effect of full-back disk turbine wheel.

Fig.6 shows the result by comparing the turbocharger size as a parameter of compressor outlet diameter with the specific power. It is assumed that the square of the compressor outlet diameter is proportional to both compressor and turbine mass flow rate. The turbocharger for 1GD engine was reduced by 30% compared to the conventional turbocharger with the same specific power. Thereby it achieved a top-class wide operating range and fast response. Furthermore, the consequent reduction in pumping loss helped to improve the fuel economy of the GD engine in the EC test cycle by 2.5%.

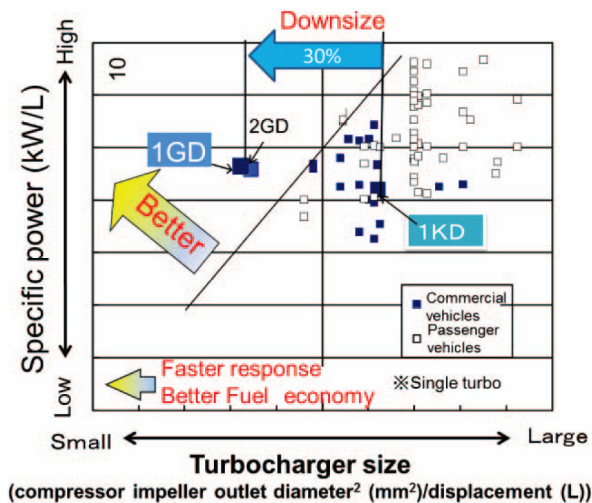


Figure 6. Comparison of Turbocharger size with diesel engine specific power.

REFINEMENTS TO VARIABLE NOZZLE VANE MECHANISM

In a diesel engine, the boost pressure and exhaust gas pressure are controlled by opening and closing the vanes. This control is generally performed using variable nozzle vanes (VN) that can increase the exhaust gas recirculation (EGR) rate to restrict the generation of nitrogen oxides (NO_x), as well as improve both dynamic response and engine torque at low speeds (i.e., low end torque). Since VN are used in high-temperature unlubricated environments, it is important to include a gap between the vanes and sliding surfaces in consideration of thermal deformation of the sliding surfaces due to sliding resistance, as well as tilting of the vanes caused by the fluid force. However, the flow that leaks through this clearance has a large effect on turbine performance. Figure 7 compares turbine efficiency results when the VN side clearance is reduced. The graph indicates that the effect is magnified under low gas flows, because the ratios of the nozzle side flow area becomes bigger relatively when closing the vanes shown in Fig.7. This indicates that reducing the clearance is quite important for improving dynamic response and fuel economy.

It is also effective when fully closed VN control is adopted for the purpose of countermeasure for deterioration in VN sliding performance due to deposit which is mentioned in the following section.

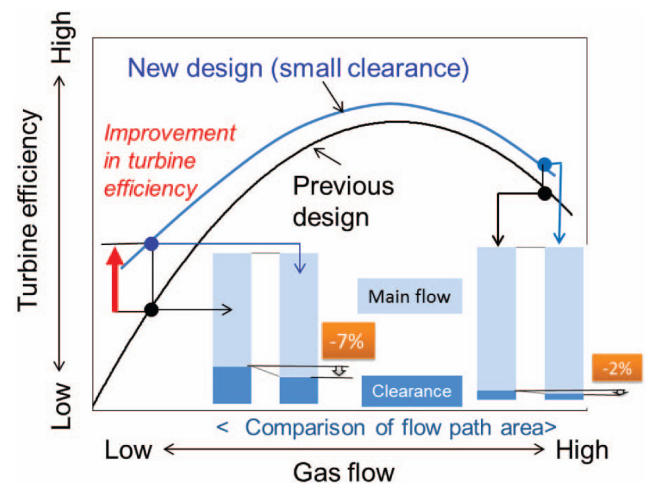


Figure 7. Effect of VN side clearance reduction.

Therefore, this development analyzed the VN geometry to reduce the side clearance. Figure 8 compares the VN geometry of the previous generation and D-series turbochargers.

In the previous design, which supports the sliding surfaces on the turbine side using the turbine housing, the sliding surfaces are susceptible to thermal deformation and oxidation due to the housing. Therefore, a new highly heat resistant plate was placed between the sliding surfaces and the turbine housing that constrains the VN by spring force. This improves the oxidation resistance of the sliding surfaces and enables an increase in the exhaust gas temperature. In addition, by setting the springs so that the force application points are on the spacers (i.e., the support points), this geometry is not subject to bending moment due to the spring force, which is a cause of deformation. The dispersion of deformation distribution in the circumferential direction was reduced by adopting one spacer for every two vanes.

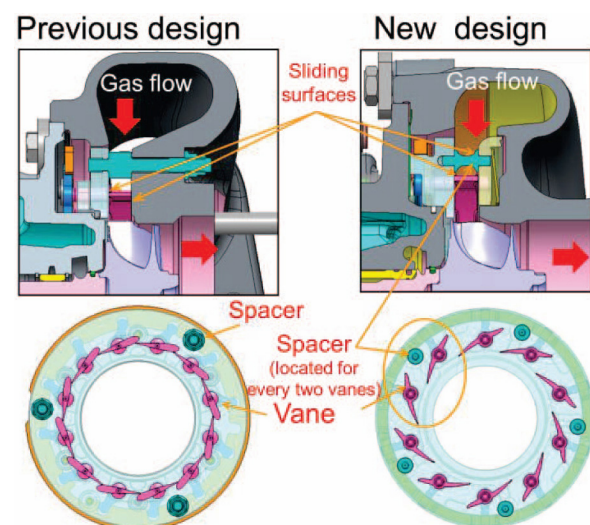


Figure 8. VN geometry comparison.

To reduce the tilting of the vanes, vane shaft supports were adopted on both sides instead of one. These measures enable the VN to operate with less clearance, thereby helping to satisfy emissions regulations.