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# The New Inline 4 Cylinder 2.5L Gasoline Engine with Toyota New Global Architecture Concept

## Der neue 2.5L Reihen-4-Zylinder-Ottomotor mit Toyota Neue Global Architektur Konzept

## Abstract

In October 2015, Toyota Motor Corporation published “Toyota Environmental Challenge 2050“, which consists of 6 challenges, to contribute towards a sustainable society. Challenge 1 is “New Vehicle Zero $CO_2$ Emissions Challenge“, where the target is to reduce vehicle $CO_2$ emissions by 90% in comparison with 2010 levels by 2050. To accelerate this challenge, Toyota has been developing the new engine series for lower fuel consumption, higher performance and globally better productivity based on “Toyota New Global Architecture (TNGA) concept“.

The new inline 4 cylinder 2.5L gasoline naturally aspirated (NA) engine is the first mass-produced TNGA engine. In order to realize the higher specific power with lower fuel consumption and exhaust gas emissions, a further improvement in combustion was needed. For this purpose, the fundamental structure such as stroke-bore ratio, valve layout has been completely redesigned. As an example, the laser cladded intake valve seat has been adopted to increase both tumble ratio and air flow coeffiency. In addition, the Atkinson cycle using new electrical VVT, D-4S (direct + port injection) system with new multi-hole type DI injector and the high energy ignition coil have been adopted to improve combustion. Furthemore, variable cooling and lubulication systems have been adopted to improve fuel consumption by reducing cooling and friction losses. As a result, over 40% thermal efficiency and over 60kW/L specific power have been achieved. By combining with new TNGA 8-speed Automatic Transmission, over 16% lower fuel consumption and 12% better acceleration performance have been finally realized compared to the previous model.

Im Oktober 2015 veröffentlichte die Toyota Motor Corporation die "Toyota Environmental Challenge 2050", die aus sechs sogenannten Herausforderungen besteht, die zu einer nachhaltigen Gesellschaft beitragen sollen. Die erste Herausforderung lautet “ Neu-Fahrzeug ohne CO2-Emmision”, deren Ziel es ist, die CO2-Emissionen der Fahrzeuge bis 2050 um 90% gegenüber dem Niveau von 2010 zu senken. Um dieses Ziel zu erreichen, entwickelt Toyota eine neue Motorenbaureihe mit geringerem

Kraftstoffverbrauch, höherer Leistung und einer weltweit erhöhten Produktivität basierend auf Toyotas "Neue Globale Architektur Konzept (TNGA)".
Der neue Vierzyylinder-Benzin-Saugmotor 2,5l ist der erste in Serie gefertigte TNGA-Motor. Um die höhere spezifische Leistung bei geringerem Kraftstoffverbrauch und Abgasemissionen zu realisieren, war eine weitere Verbesserung der Verbrennung erforderlich. Hierzu wurde die Grundstruktur wie das Hub-Bohrungsverhältnis und der Ventilaufbau komplett neu gestaltet. Weiterhin wurde ein lasergestützter Einlassventilsitz dazu verwendet, um sowohl das Taumelverhältnis als auch den Luftströmungskoeffizienten zu erhöhen.. Zusätzlich wurde der Atkinson-Zyklus unter Verwendung eines neuen elektrischen VVT-, D-4S-Systems (Direkt- und Porteinspritzung) mit einem neuen Mehrloch-DI-Injektor und der Hochenergiezündspule verwendet, um die Verbrennung zu verbessern. Darüber hinaus wurden variable Kühl- und Schmiersysteme eingesetzt, um den Kraftstoffverbrauch zu reduzieren, indem Kühl- und Reibungsverluste reduziert wurden. Infolgedessen wurden über ein thermischer Wirkungsgrad von über 40% und über 60 kW/l spezifische Leistung erreicht. Durch die Kombination mit dem neuen TNGA-8-Gang-Automatikgetriebe wurden im Vergleich zum Vorgängermodell über um 16% verringerter Kraftstoffverbrauch und eine um 12% höhere Beschleunigungsleistung erzielt.

## 1. Introduction

To cope the energy and environmental issues, it has been becoming more and more important to realize a sustainable society. $CO_2$ reduction must be done and the further improvement of fuel consumption is demanded to the engines. In October 2015, Toyota Motor Corporation published “Toyota Environmental Challenge 2050“, which consists of 6 challenges (Figure 1). Challenge 1 is “New Vehicle Zero $CO_2$ Emissions Challenge“ whose target is to reduce vehicle $CO_2$ emissions by 90% in comparison with 2010 levels by 2050 (Figure 2).


**Figure 1**: 6 challenges of Toyota Environmental Challenge 2050

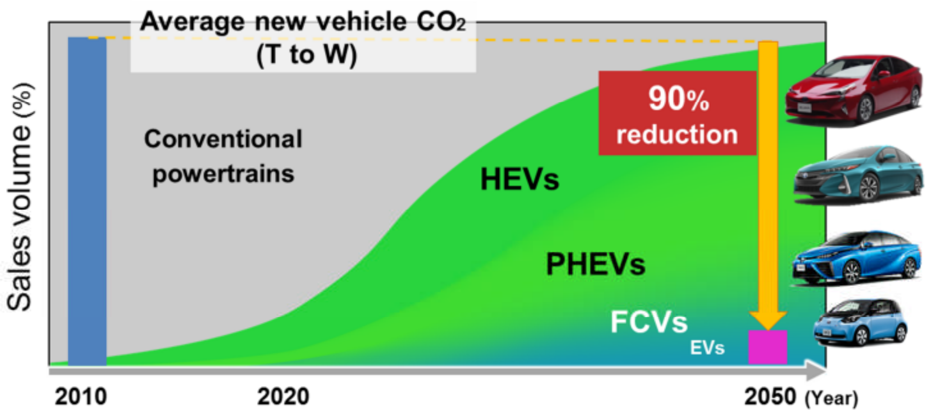



**Figure 2**:Toyota strategy image of next generation vheicles

In addition, it is needed for the future engines to make exhaust gas emissions more cleaner to meet the severer regulations of various countries.
Simultanously, the high performance and the superior drivability are universally important to the expectation of customer for fun-to-drive.
To challenge these requiremets, the conventional small improvements of engine were not enough. It was necessary to redesign the fundamental engine structure such as bore-stroke and the valve layout, hence we have developed the new engine based on Toyota New Global Architecture (TNGA) concept.
In recent years, we have made efforts to improve thermal efficiency and achieved the world top 40% with the engine for the hybrid vehicle (HV).
In order to achieve even higher targets both thermal efficiency and specific power to improve fuel consumption and performance at the same time, we have focused particularly on the high-speed combustion technology.
Figure 3 shows the trend and targets of maximum thermal efficiency and maximum specific power. The targets for new HV engine in TNGA series have been set over 41% and 50kW/L. In addition, the targets for conventional engine have been set over 40% which has been achieved by 2ZR-FXE of the 4th generation Prius, and over 60kW/L to provide driving pleasure for customers.

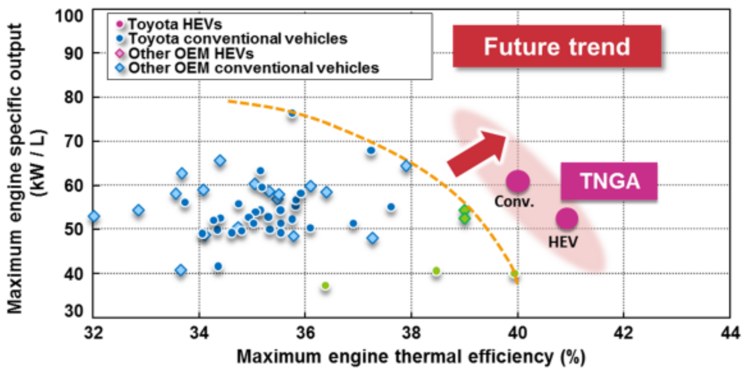



**Figure 3**: Trend and targets of specific power and thermal efficiency

In addition to these targets, the globally better productivity and the dynamic and smooth acceleration have been focused in TNGA concept.
This paper explains the development of the first mass-produced TNGA engine which is inline 4 cylinder 2.5L gasoline NA engine to be coupled with Toyota's new 8-speed automatic transmission (AT) and with driving force demand control systems to achieve high vehicle drivability for fun-to-drive.

## 2. Specifications

In order to achieve 40% of maximum thermal efficiency and 60kW/L of maxmum specific power, high-speed combustion concept has realized by high tumble ratio and high flow coefficient which are commonly adopted in TNGA engines.
Furthermore, motor-driven VVT for Atkinson cycle, new D-4S system (DI & PFI) with new multi hole injectors, cooled EGR system, new cooling system for heat management, Variable oil-pressure pump system for significant mechanical loss reduction and high energy ignition coil for expanding combustion limit are applied .
Figure 4 shows the engine general view and Table 1 shows the main engine specifications.

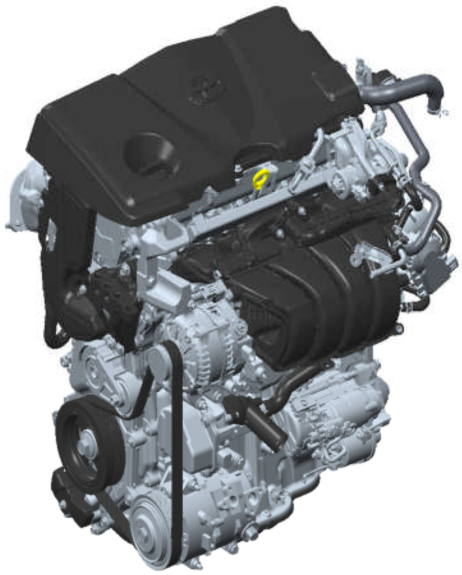


**Figure 4**: TNGA 2.5L engine (conv.)

**Table 1**: Specifications of TNGA 2.5L engine

| Engine | for conventional | | for HV | |
|---|---|---|---|---|
| | New | 2AR-FE | New | 2AR-FXE |
| Engine Type | Inline-4cyl | ← | ← | ← |
| Displacement [cc] | 2487 | 2494 | 2487 | 2494 |
| Bore×Stroke [mm] | Φ87.5×103.4 | Φ90.0×98.0 | Φ87.5×103.4 | Φ90.0×98.0 |
| Compression ratio | 13.0 | 10.4 | 14.0 | 12.5 |
| Max. power [kW/rpm] | 151/6600 | 134/6000 | 131/5700 | 118/5700 |
| Max. torque [Nm/rpm] | 250/5000 | 231/4100 | 221/3600-5200 | 213/4800 |
| Cylinder head/block | Al / Al | ← | ← | ← |
| Valve system | DOHC<br>4 valves per cylinder<br>HLA+Roller rocker | ← | ← | ← |
| | VVT-iE<br>(Motor-driven :Intake)<br>VVT-i (Exhaust) | Dual VVT-i | VVT-iE<br>(Motor-driven :Intake)<br>VVT-i (Exhaust) | Dual VVT-i |
| Lublication system | Variable controlled oil pump | Conventional | Variable controlled oil pump | Conventional |
| Cooling system | Motor-driven W/P | Belt-driven W/P | Motor-driven W/P | Belt-driven W/P |
| Intake system | No | Variable length intake manifold | No | ← |
| EGR system | Cooled EGR | No | Cooled EGR | Cooled EGR |
| Fuel injection system | D-4S<br>(multi hole DI +PFI) | PFI | D-4S<br>(multi hole DI +PFI) | PFI |
| After Treatment | 2CAT, 2A/F | 2CAT, A/F+$O_2$ | 2CAT, 2A/F | 2CAT, A/F+$O_2$ |
| Emissions | SULEV30(LEVⅢ) | PZEV(LEVⅡ) | SULEV30(LEVⅢ) | PZEV(LEVⅡ) |

## 2.1 High speed combustion technologies and engine basic structure

### 2.1.1 Combustion concept and target (Common architecture)

The most suitable base specification of compression ratio and stroke to bore ratio (S/B ratio) were studied at the begining of development to achieve higher thermal efficiency and specific power.

Figure 5 shows effect of S/B ratio on combustion characteristics In terms of maximum thermal efficiency, turbulence intensity for higher combustion speed, lower FMEP and lower surface area to volume (S/V) ratio for lower cooling loss are required. Turbulent intensity improves by increasing S/B ratio, but FMEP contrarily deteriorates due to higher piston speed nor does S/V ratio. In addition volumetric efficiency is important especially for NA engine in terms of maximum specific power, and it deteriorates by increasing S/B ratio due to smaller bore diameter leading to smaller valve diameter.

For these reasons, it is important to select an appropriate S/B ratio to achieve a good balance of thermal efficiency and specific power.

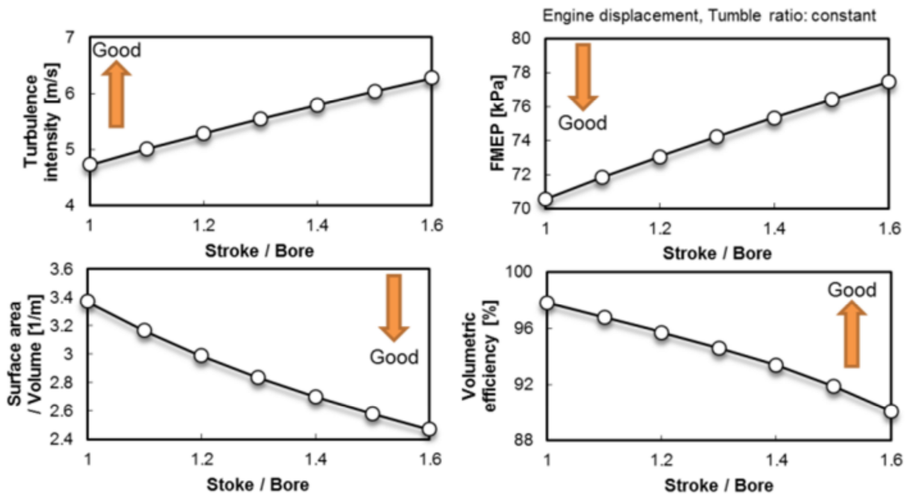



**Figure 5**: Effect of S/B ratio on combustion characteristics

Figure 6 shows the analysis results of the effect of S/B ratio and compression ratio on thermal efficiency and specific power using a 1D model that is validated with an experimental data. At S/B ratio of 1.0, maximum thermal efficiency saturates at around ε13.0 (upper left on Figure 6). This is because of that the amount of EGR is limited by combustion instability, so knocking becomes a limiting factor to improve efficiency by increasing compression ratio. By increasing S/B ratio, EGR limit can be extended due to enhanced combustion stability, so thermal efficiency improves, and the optimal compression ratio for maximum thermal efficiency also changes.

On the other hand in partial load conditions, thermal efficiency improves with higher compression ratio for any S/B ratio (upper right on Figure 6) . However, this trend saturates at around S/B ratio of 1.2 due to relationship with friction, and further S/B ratio increase deteriorates thermal efficiency.

Specific power shows different trend for each compression ratio due to mutual effect of specific power and volumetric efficiency. If compression ratio is increased, power reduces even for smaller S/B ratio with larger valves (lower left on Figure 6).

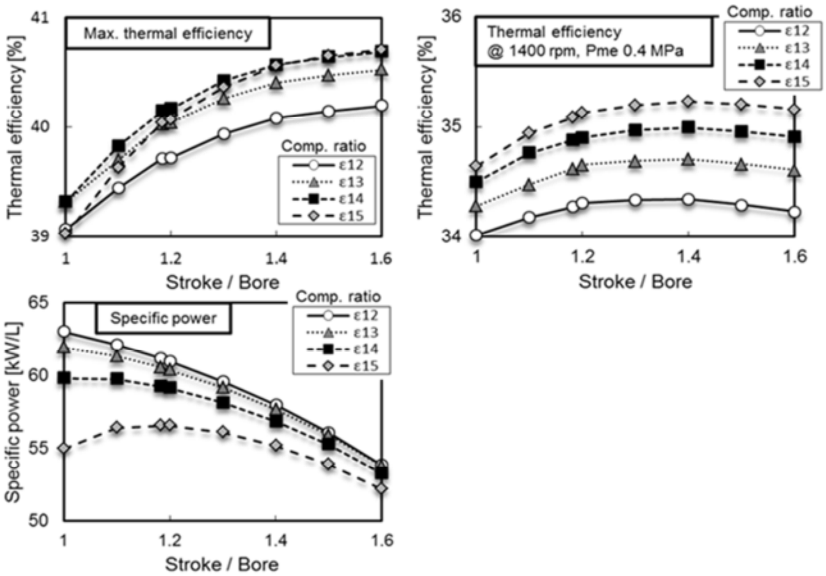



**Figure 6**: Effect of S/B ratio on engine performance

Left graph on Figure 7 shows the engine performance index that takes into account thermal efficiency, specific power, and their frequency of use in each driving mode. Right graph shows the relationship between the maximum thermal efficiency and specific power. Considering driving mode fuel efficiency and power performance, S/B ratio of around 1.2 is optimal, and ε13.0 is estimated to achieve the target maximum thermal efficiency of 40% and specific power of 60kW/L.

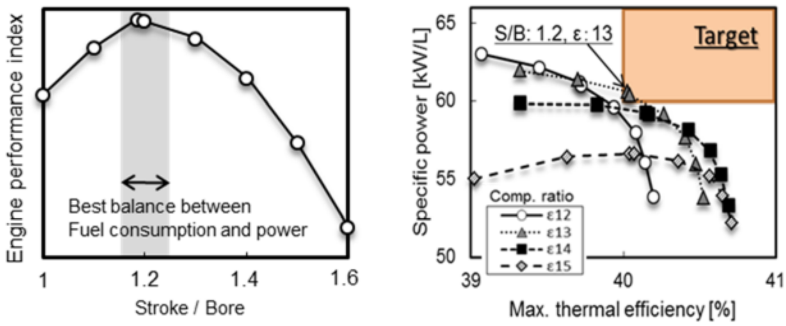



**Figure 7**: Effect of S/B ratio on engine performance and relationship between maximum thermal efficiency and specific power

## 2.1.2 Combustion design

Firstly, in order to realize the basic concept of high speed combustion, the factors affecting in-cylinder turbulence intensity was studied. Figure 8 shows the time-history of transient tumble ratio and turbulence intensity and the combustion specifications.

Transient tumble ratio has the peak A, caused by the incoming flow from the port during intake stroke. After that, it leads to peak B during compression stroke while forming tumble flow, and then starts to decay toward TDC. On the other hand, after transient tumble

reaching the peak B, turbulence intensity reaches peak C, caused by the conversion of tumble flow into turbulence due to compression, and then reaches D toward TDC.
Based on the past experiments and CFD results, it is known that the values of A~C are determined by the engine specifications shown in the boxes in Figure 8. The turbulence intensity and corresponding tumble ratio, and the correlation with the prediction equation deriverd from the engine specifications. Especially intake port design has been most important factor through engine development and for this high speed combustion concept.

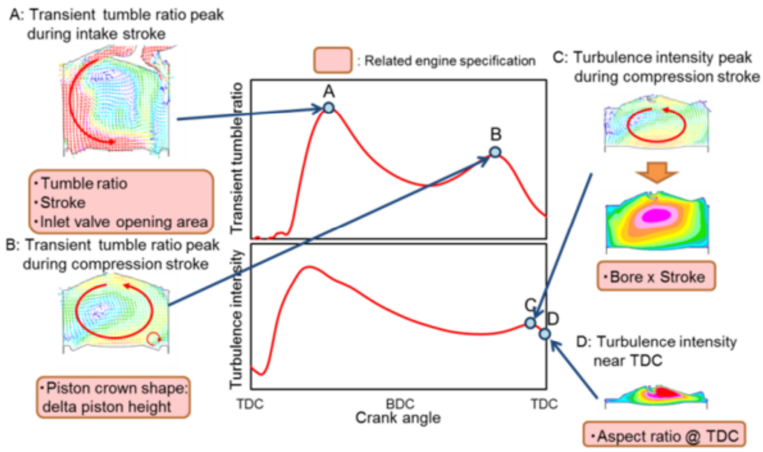



**Figure 8**: In-cylinder flow characteristic and related engine specification

## 2.1.3 Cylinder head (Intake port) and combustion chamber (Piston crown)

To realize maximum thermal efficiency target, turbulence intensity increasing and EGR limit expansion are needed. The result of calculation and experiment, tumble ratio of 2.8 is required. On the other hands, based on the specific power requirement of 60kW/L, volumetric efficiency at the maximum power condition should be 92%, and flow coefficient of 0.48 is required. Figure 9 shows target of intake port performance, which are exceeding current corelations.

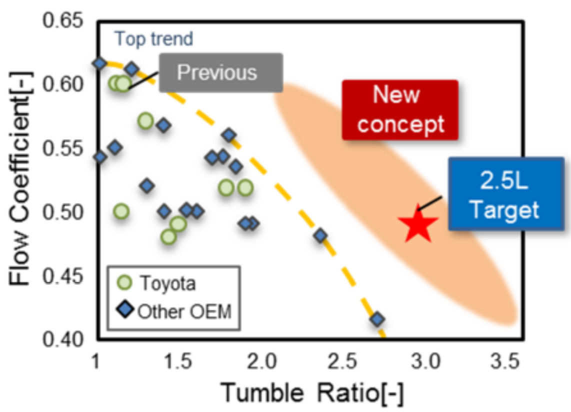



**Figure 9**: Target of intake port performance

The high tumble ratio and the high flow coefficient are contradicting demands. Intake port design such as valve layout has been completely redesigned in order to meet these demands.

Press-fitting valve seat needs thickness between valve seat and water jacket. Therefore, the degree of desgin freedom for valve diameter and side flow is limited. To tackle this issue, Laser-cladded valve seat has been adopted which enables to increase intake valve diameter and to widen valve angle between intake and exhaust. As a result, the linearized intake flow into cylinder has been realized.

Laser-cladded valve seat is molded with cylinder head directly by laser using cladding powder based on copper (Figure 10). In order to fulfill not only productivity but also adaptability to world-wide market, new material with high wear resistance has been developed to cope with alternative fuels.

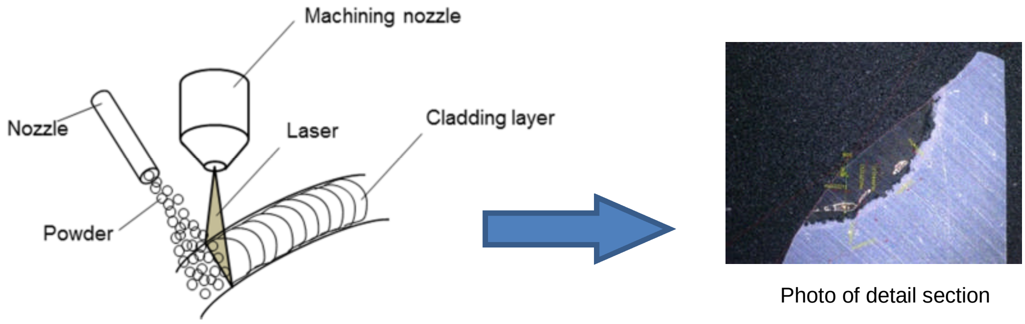



**Figure 10**: Laser cladded valve seat

Figure 11 shows the cross section comparison of intake port between current type and new one. The air flow separation at valve seat has been drastically improved. As a result of this optimization, the tumble ratio of 2.8 and flow coefficient of 0.48 have been achieved.

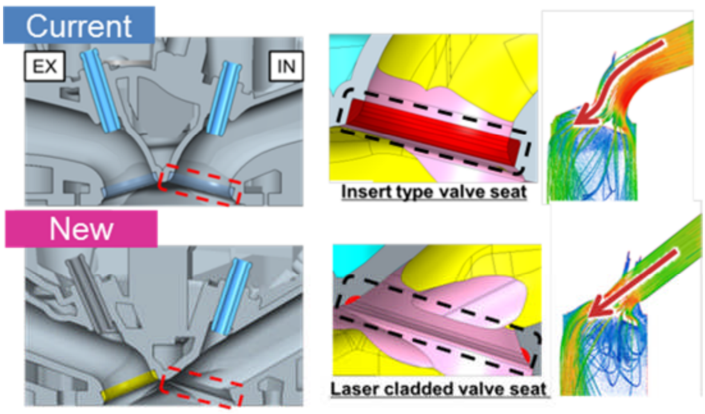



**Figure11**: Comparison of intake port

New D-4S system has been developed in order to improve thermal efficiency and specific power by knock mitigation under high compression ratio.

Considering the future PN regulation, new multi-hole injector with low penetration has been developed to reduce fuel wet on piston and wall surface.

Piston crown has been redesigned to keep a high tumble flow. Figure 12 shows the comparison of the current and the new D-4S piston crown. Spherical-shaped piston crown has been implemented to maintain a high tumble flow provided from highly efficient port.
Figure 13 shows 3D CFD results of these two pistons. It is apparent that the new D-4S piston shape allows incoming flow into cylinder during intake stroke without obstruction and reaches high tumble peak. As a result, it contributes to combustion stability.

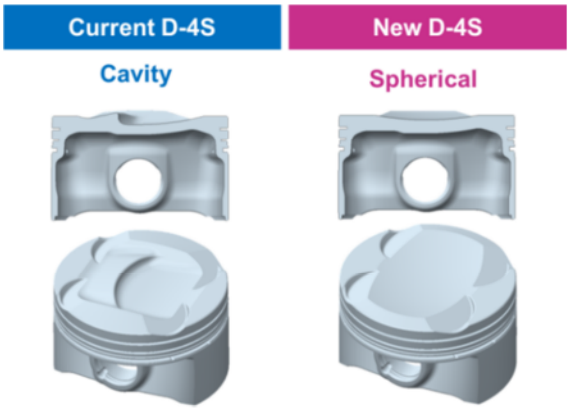



**Figure 12**: Comparison of current and new D-4S piston crown shape

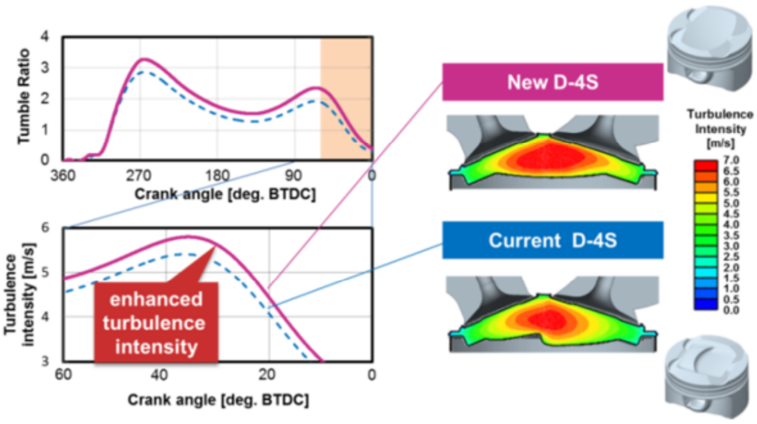



**Figure 13**: Comparison of turbulence intensity between current and new D-4S piston

Furthermore, high energy ignition coil has been implemented for the combustion stability under a high EGR rate and tumble ratio. It has been enabled by making the magnetic circuit gap smaller and enlarged core cross section area larger (Figure 14).
By these items, the stable combustion under 23% EGR ratio has been enabled which has led to maximum thermal efficiency of 40%.

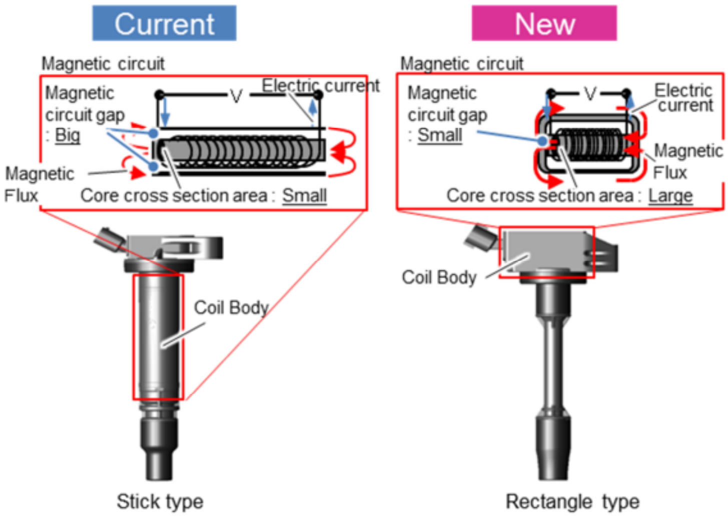



**Figure 14**: Comparison of current and new ignition coil

### 2.1.4 Moving parts

To achieve 40% maximum thermal efficiency, long stroke and high compression ratio has been selected. Furthermore, in order to achieve 60kW/L of specific power, it is needed to realize 6600rpm engine speed with maximum piston speed of 22.8[m/s], so weight reduction of moving parts has been thoroughly carried out to reduce inertia load. This section describes piston and connecting-rod, crankshaft design.

Firstly, significant weight reduction of piston has been realized by redesigning casting mold structure and optimizing casting condition without raw material change. The thickness fo crown top wall, skirt and backside of piston ring groove has been reduced. Figure 15 shows relationship between piston mass including piston-pin and torque per cylinder of NA engines. The world top level light weight piston has been designed realizing more than 14% weight reduction compared to the current 2AR engine.

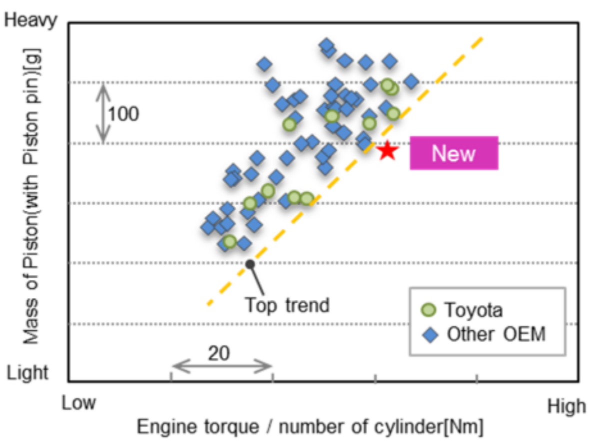



**Figure 15**: Relationship between piston mass and engine torque per cylinder of NA engines

Under high piston speed, it is difficult to balance oil consumption and low friction. To do that, the new technologies such as narrower contact width of piston ring for oil shear resistance reduction, thinner side rail of oil ring for sealing performance improvement against bore deformation and DLC (Diamond-Like Carbon) coating for friction coefficient reduction have been implemented.

High-strength material for connecting rod has been used achieving 30% strength improvement compared to the previous. Furthermore, in order to achieve both connecting rod bearing reliability and weight reduction, for example, big end upper part (Figure16) has been thinned to disperse the surface pressure of bearing, avoiding local high surface pressure. More than 20% weight reduction has been achieved compared to current 2AR engine.

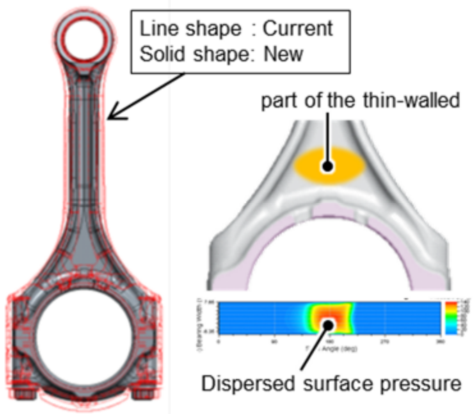



**Figure 16**: Optimized connecting rod

The crankshaft design is very important for reliability under high speed revolution. To design smaller crank pin jounal diameter, the optimization of the arm rigidity of the crankshaft reduced the inertial mass around of #3 crankshaft main journal. As a result of this, the impact load to a block was reduced. Full counter weight structure has been adopted in other Journals and improving balance rate. Even in long stroke specification and a high maximum engine speed, the block structure and size has been equal with current 2AR engine.

### 2.1.5 Optimization of gas exchange (Valve train system)

As well as for performance, appropriate valve timing is important for fuel efficiency in the whole driving condition. This section describes the gas exchange control such as valve timing and so on.

It is important to decide intake valve duration to balance fuel consumption and performance. Intake duration cannot be enlarged too much because of volumetric efficiency decrease due to back flow into intake port especially in low to middle engine speed. As current model, roller follower with Hydraulic Lash Adjuster (HLA) has been implemented. In order to increase time area of intake valve lift, the new concave shape cam rob profile has been extended for valve lift profile extention. Figure 17 shows the comparison of valve lift and operating angle. Time are has been enlarged with duration and lift.

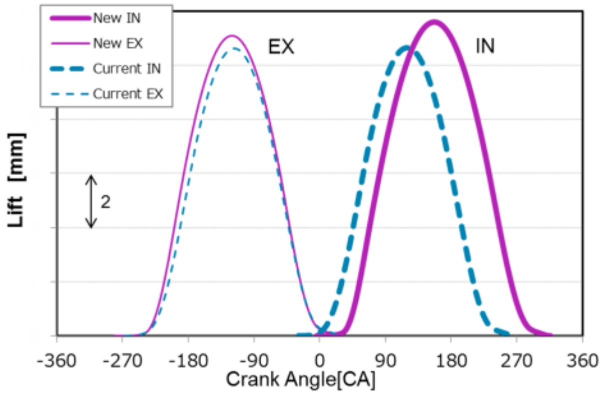



**Figure 17**: Comparison of valve timing

In addition, the Atkinson cycle improves a fuel consumption in the low load engine condition. Figure 18 shows relationship between Intake Valve Closing timing (IVC) and each loss such as pumping loss. As a close timing is retard, in other words increasing IVC, pumping loss decreases. But actual compression ratio and tumble intensity becomes low. As a result, combustion becomes worse and total loss increases. So, these parameters are in trade-off relationship and indicate the optimal valve timing for minimum loss.

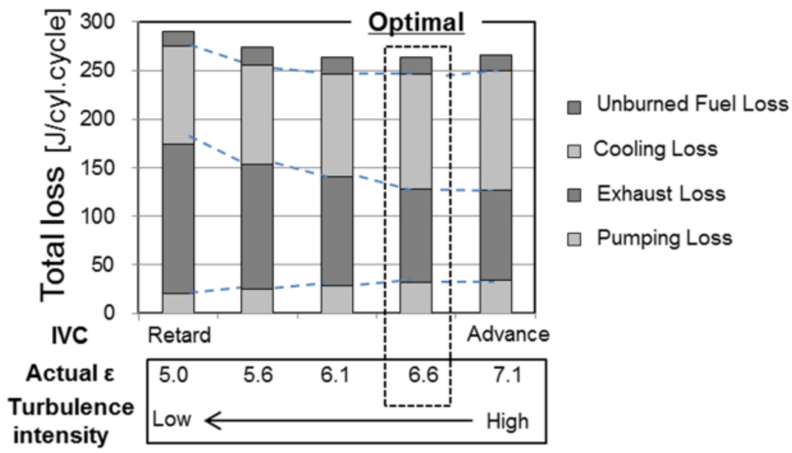



**Figure 18**: Sensitivity of various losses according to IVC

Mid-position lock VVT system (VVT-iW) has been evaluated on the intake side to ensure both engine startability and Atkinson cycle so far. But, motor-driven VVT (VVT-iE) has been finally selected in order to realize the most suitable valve timing in a second from low oil temperature low engine speed which is difficult for VVT-iW to control timing because of low oil pressure.

Center-spooled VVT oil control valvehas been adopted on the exhaust side to improve response speed by shortening the oil passage at low temperature.

Furthermore, in order to contribute to superior drivability and fuel consumption, the demand torque from driver is controlled to an optimal throttle valve and EGR valve position at the same time.

## 2.2 Cooling system

It is important that cooling loss is reduced in order to improve thermal efficiency. In addition, knock mitigation must be done for the performance at the same time.

Not only optimizing cooling circuit but also cooling control system with electric water pump according to the driving condition has been implemented. Furthermore, to improve fuel consumption in cold condition, the heat management system warm-up process improvement has been implemented.

For mitigating knocking, electric water pump that operates free from crank rotation has been implemented not only on HV but also conventional engine.

Water jacket flow system has been redesigned from longitudinal to side flow in order to achieve an optimal balance between cooling performance and pump size (Figure 19).

A Water Jacket Spacer (WJS) is placed in the cylinder block water jacket only in exhaust side to optimize cylinder bore temperature distribution. WJS controls and concentrates the flow to prevent knocking by cooling the upper range of exhaust side cylinder bore. This is the most effective means to cool the tumble flows (Figure 20).

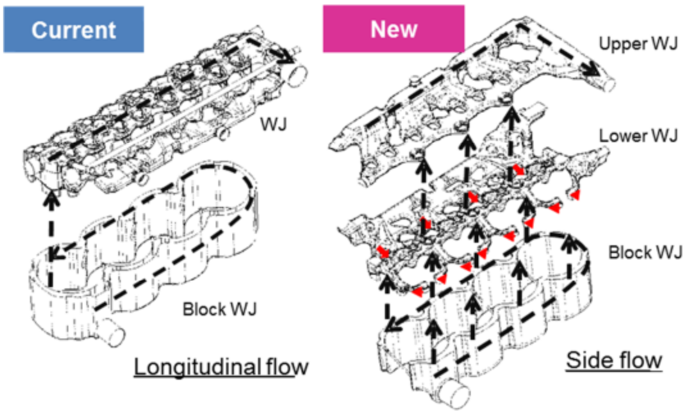



**Figure 19**: Structure of water jackets

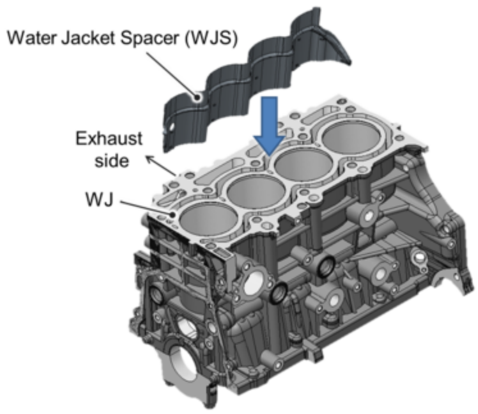



**Figure 20**: Water jacket spacer

This system achieves better cooling performance by high flux even at low engine speed and increases engine torque by knock mitigation.

In addition, heater assisted thermostat contributes to high performance by reducing the water temperature at knocking limit area (Figure 21).

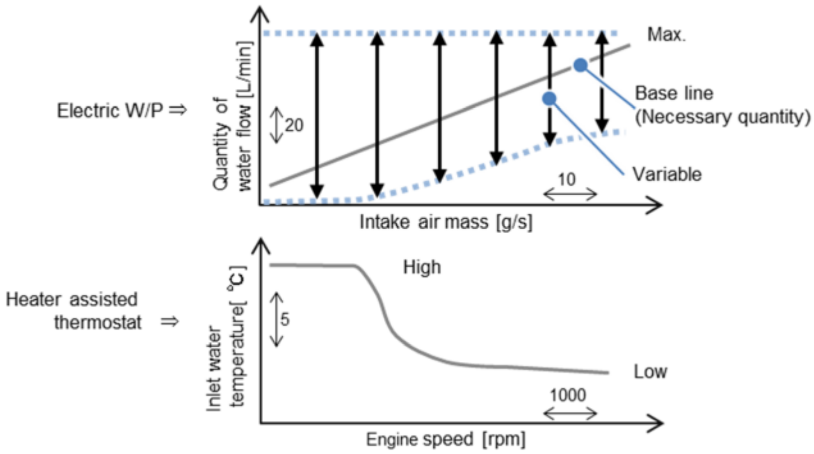
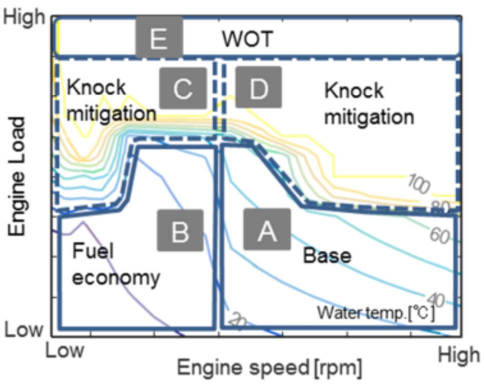



**Figure 21**: Control of electric water pump and heater assisted thermostat

A new logic to control high water temperature has been implemented to reduce cooling heat loss and friction loss in Minimum Spark Advance for Best Torque (MBT) driving condition such as low load. Figure 22 shows strategy for water temperature control.

For improvement of fuel consumption, new heat-management system has been implemented. In order to raise water temperature earlier from cold start condition, the system controls not only the water flow by electric water pump but also water path opening or closing by Flow Shut Valve (FSV).




| Zone | Water temp. | Flow quantity |
|---|---|---|
| A | Low | Base |
| B | High | Decrease |
| C | High | Increase |
| D | Low | Increase |
| E | Low | Max. |

**Figure 22**: Strategy of water flow and temperature control

Figure 23 shows cooling circuit for conventional engine with two FSV units. One is used for heater core flow control and another is for Automatic Transmission Fluid (ATF) warmer-core control.

Opening or closing FSVs depending on ambient temperature or driving condition improves fuel consumption in warm-up process without spoiling heat supply for customer comfortability (Figure 24).

In addition, the most suitable flow control which holds the heat distribution deflection in the engine body in the warm-up process has been carried out and this control prevents noise and vibration (NV) deterioration.

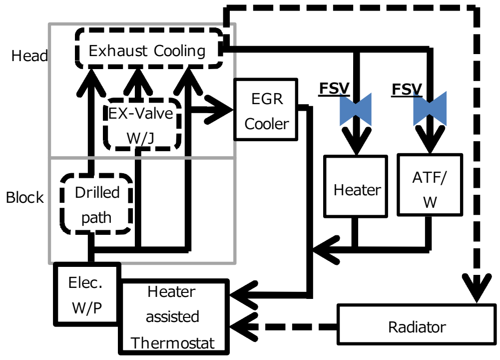



**Figure 23**: Cooling system circuit (for conventional engine)

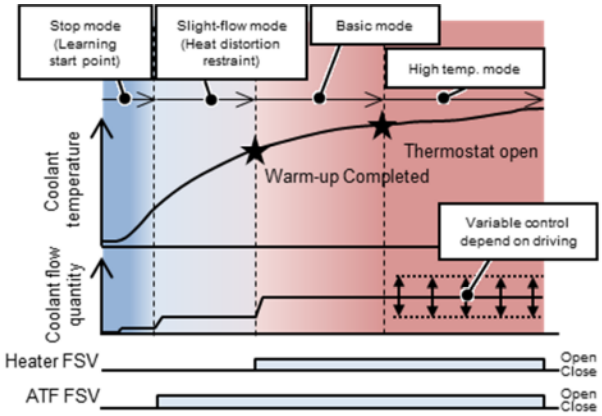



**Figure 24**: Example of control with FSV and coolant flow quantity in warm-up process

## 2.3 Lublication system

Friction loss is also the important factor to improve fuel consumption.

Electronic control-type variable capacity oil pump (Figure 25) has been implemented which can control the oil flow rate with "electrical oil control valve". The rotor is "trochoid type" with less friction than the previous variable type oil pump .

The system can supply the appropriate oil and gurantee the required oil pressures to the several parts of engine according to the main oil channel pressure. In addition, the friction loss has been reduced for fuel consumption.

On the other hand, oil pressure control can stop the oil jet flow to the piston with pressure control. So the temperature of piston rises up earlier in warm-up process and PN emission is reduced. Furthermore, friction loss of piston can be reduced.

The engine oil is supplied by the trochoid oil pump fitted on the front section of the crankcase driven by the chain. The oil circuit is divided into 2 ways, the oil jet channel and the main oil channel (to cam shaft, VVT, HLA channel and Vacuum pump), which make easier to control oil jet flow (Figure 26).

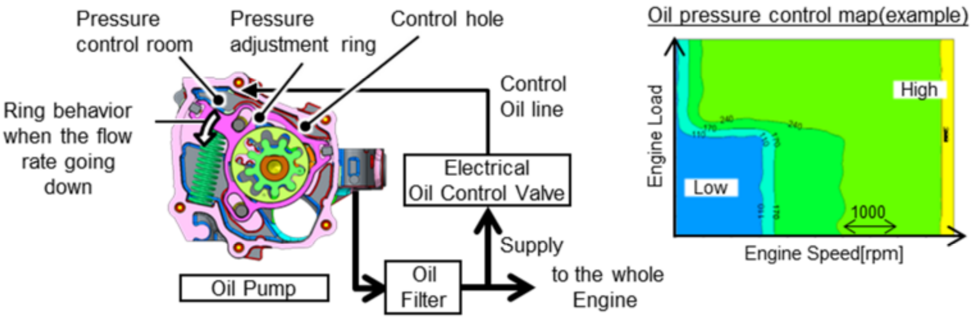



**Figure 25**: Electric control-type variable capacity oil pump

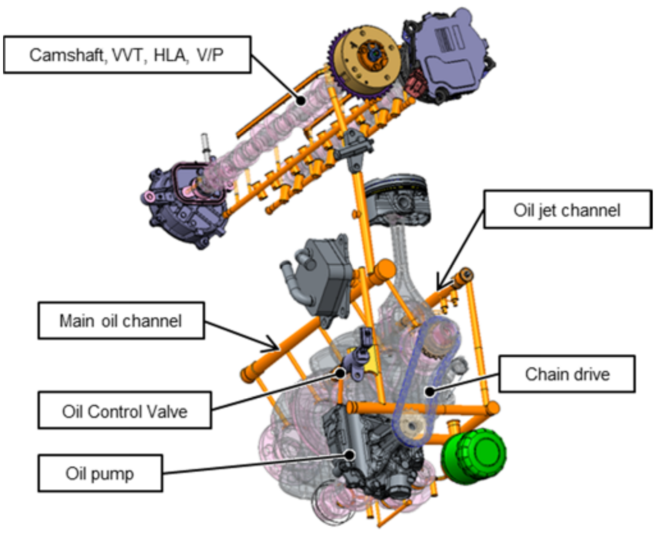



**Figure 26**: Lubrication system

## 2.4 Exhaust system

Catalyst temperature control has become more and more important to adapt to SULEV30 regulation in North America, European Euro6 and so on.

Catalyst position has been decided with warm-up characteristics from a cold start and stoichiometric driving area especially on a high load. The exhaust port layout in the cylinder head, surface area to the catalyst and position of catalyst have been redesigned.

In addition, this engine is installed in vehicle transversely and the exhaust side is designed to be in rear side considering future series development to have design flexibility for the exhaust back pressure reduction and the under-floor catalyst allocation to be as close as possible to the engine.

Furthermore, the 2 A/F sensor control system has been implemented locating sensors in upstream and downstream of 1st catalyst. Therefore the catalyst performance has been improved by better gas detectability leading to the smaller catalyst size and lower precious metal.

## 2.4.1 Exhaust gas cooling in cylider head

Concentrated lateral flow between exhaust valves in cylinder head has been implemented. In addition, water jacket sectional area has been decreased 20% to improve heat transfer coefficient. The temperature of cylinder head between exhaust valves has been reduced about 10°C from the current engine and simultanously water pressure loss and flow quantity have been reduced. Figure 27 shows the layout of the ports and water jackets. This configuration allows to expand the surface area and to lower the temperature by twice of current engine. But since the above-mentioned electric water pump cooling system controls coolant flow rate in warm-up process, the heat loss of exhaust gases can be reduced. On the one hand, as Figure 28 shows, stoichiometric area (λ=1 area : red ) has been greatly enlarged from the current engine.

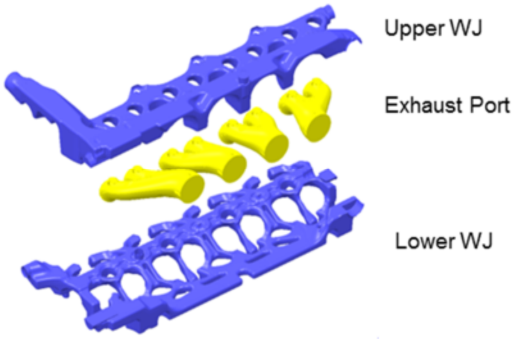



**Figure 27**: Exhaust port and water jackets structure

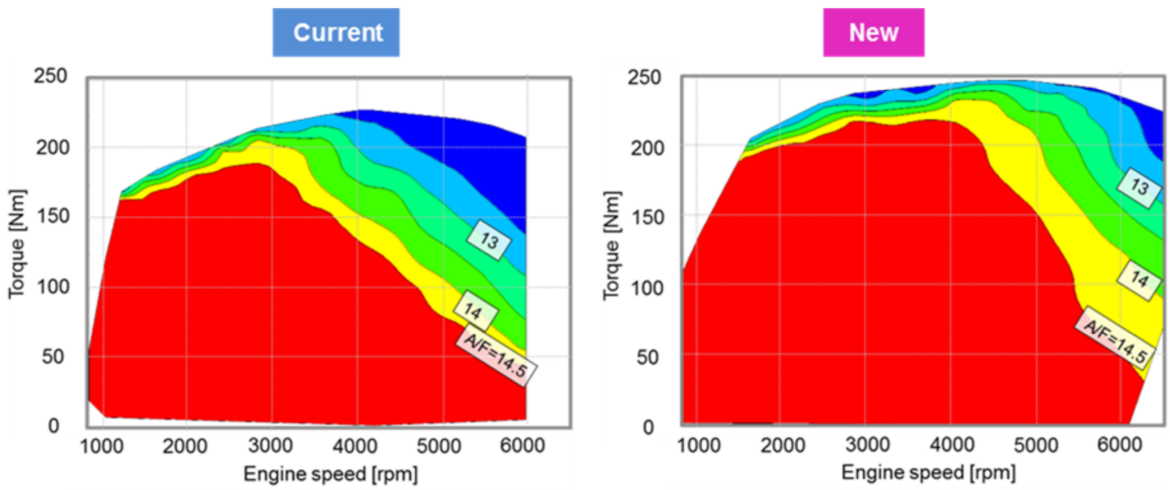



**Figure 28**: Comparison of stoichiometric area

## 2.4.2 Exhaust maniford

As mentioned previously, the exhaust side is rear of vehicle because of the reduction of back pressure for performance (Figure 29). Figure 30 shows the new exhaust manifold. Furthermore, scavenging rate is improved by the optimization of valve timing and the improvement of isometric length adopting 4-1 branch pipe. So, performance is improved by mitigating knocking. In addition, the improvement of gas flow deflection toward catalyst and exhaust sensor contributes to the reduction of precious metal quantity on catalyst.

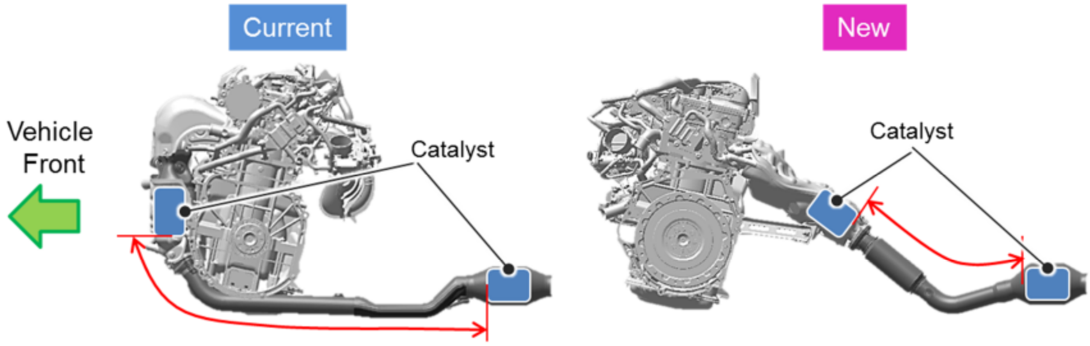



**Figure 29**: Exhaust layout on vehicle

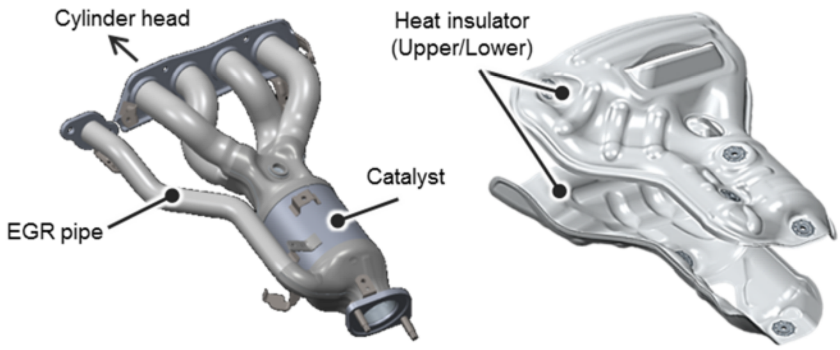



**Figure 30**: Exhaust manifold

The optimization of branch pipe length and exhaust surface area gives not only higher torque performance but also higher exhaust purification performance by earlier warm-up process. And the covering of whole exhaust manifold by aluminum insulator of high radiation rate has improved warm-up process.

The reduction of emissions in cold catalyst warm-up process is very important to adapt to more strict emission regulations. Figure 31 shows the relation and balance between an energy needed for catalyst warm-up and gas quantity from engine at cold condition.

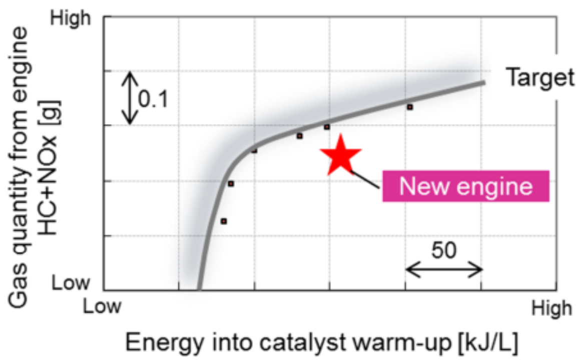



**Figure 31**: Target for necessary energy to catalyst in warm-up process

### 2.4.3 EGR system

The quality of EGR gas which means temperature, passage pressure loss and distribution variation is very important for the cooled EGR system. As mentioned, turbulence intensity has been increased for the high speed combustion and the combustion stability which has led to expand the EGR limit up to 25% in maximum.

The shape of intake manifold has been tuned by CFD to achieve this highest EGR ratio under good combustion and to reduce the variation of EGR distribution between cylinders within less than 1% on the non-Atkinson condition. EGR gas ratio of each cylinder has been equalized by tuning EGR gas passage. And the cooling capacity of gas has been increased not only by EGR cooler but also by EGR cooling passage in the cylinder head (Figure 32).

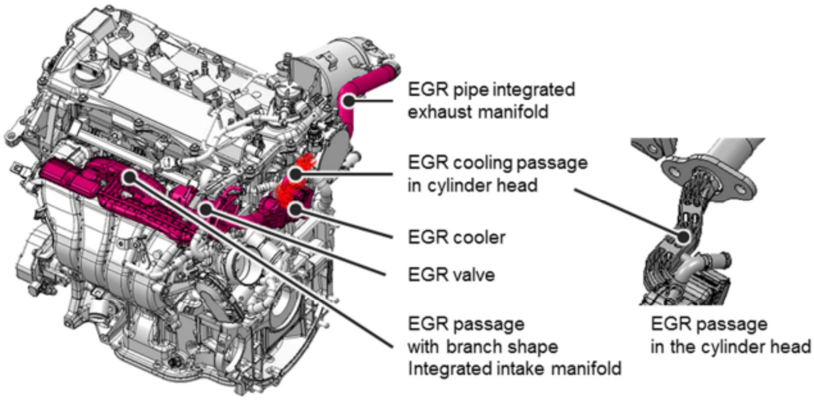



**Figure 32**: EGR system layout

### 2.5 Fuel injection system

To reduce fuel wall wetting is essential technology for PN reduction during the cold condition. Current fan-spray nozzle is an enlarged taper shape, and it has the high capability of atomization and coking robustness. However, fan spray lacks the degree of freedom in the spray layout and injection timing, so the in-cylinder wall wetting needed to be reduced.

And it was difficult to control the small injection quantity because of the wide slit hole shape.

As mentioned previously, the new multi-hole direct injection system with low spray penetration and the spherical-shaped piston have been implemented. Furthermore, instead of the full-lift needle control, the partial-lift needle control has been developed in order to realize a multi-split injection with small quantity for better emission by the optimization of fuel flow rate characteristic.

By optimizing the multiple injection quantities and timings, PN reduction and the stratified combustion for catalyst warm-up with proper air by fuel ratio around spark plug have been carried out (Figure 33).

Therefore, each country emission regulation, for example, SULEV30 regulation in North America has been met.

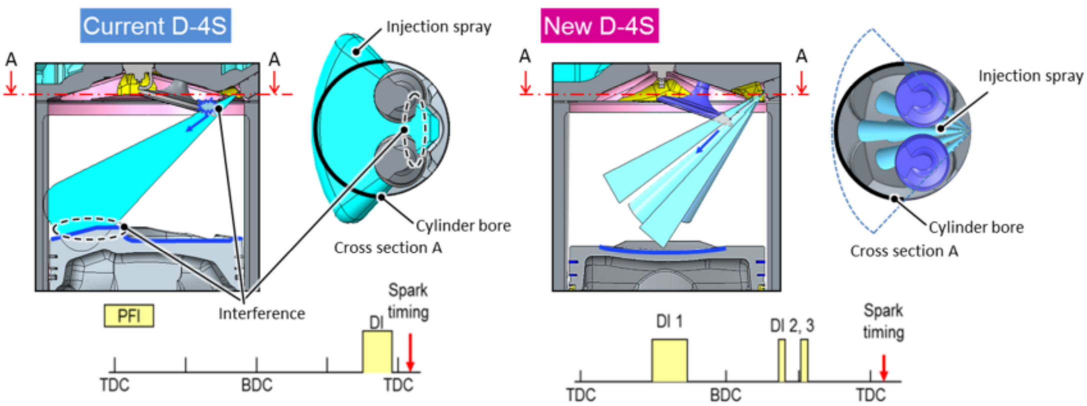



**Figure 33**: Comparison of injection timing and number in catalyst warm-up process

## 2.6 Low mechanical friction design

To reduce the friction between piston and bore, the off-set crank as well as previous engine and the resin coating with surface-smoothing treatment on piston skirt have been adopted. As mentioned previously, DLC coating has been applied on the top ring and the oil ring sliding surfaces.

In the crank system, the resin surface coating has been applied on all connecting rod bearings and main bearings in order to reduce frictions under loaded conditions.

In the lubricating system, the electronically controlled variable capacity trochoid oil pump has been adopted along with low viscosity oil at SAE 0W-16. The combination between the low viscosity engine oil and the variable capacity oil pump has enabled the optimal oil supply to the system. It has resulted in the significant reduction of oil pump driving workload and realized the reduction of overall engine friction.

In the valve-train system, the low friction resin material has been adopted on the sliding surface of chain guide. The beehive valve spring has been adopted for the moving mass reduction and the spring load reduction.

In the accessory system, the water pump has been motorized and the water pump pulley has been removed to reduce the driving workload of accessories. The efficiency of vacuum pump has been improved to reduce the friction.

The combination of the above all improvements has realized the friction reduction more than 20% compared to the current engine (Figure 34).

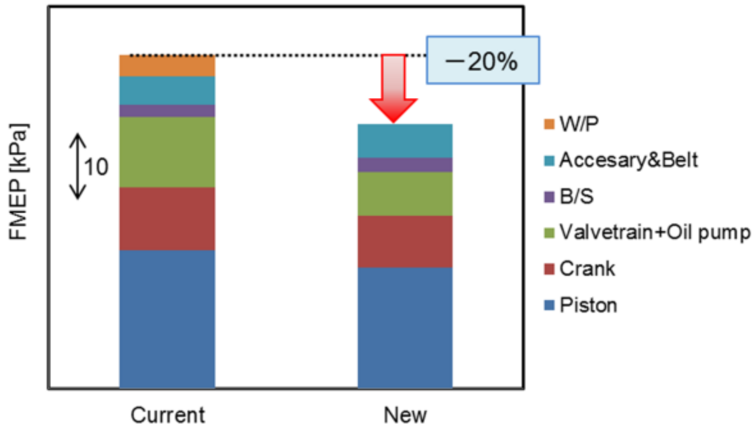



**Figure 34**: FMEP comparison (2000rpm motoring)

## 2.7 Noise and vibration

The engine nouise and vibration (NV) performance as the whole vehicle has been improved by developing the new vehicle platform and the new transmission at the same time.

First of all, regarding the NV performance in low frequency area, the powertrain, e.g. engine and the trasmission, in the engine compartment has been placed ideally by the highly precise prediction of the inertial specifications.

Placing the mount on torque roll axis lowers the level of vibration at the idle speed condition. And placing the center of gravity of the powertrain on the elastic main axis lowers the level in the engine starting also.

Next, regarding the middle frequency area, the powertrain rigidity has been improved by CAE optimizing the combination point with the new transmission. Finally, the mount vibration has been reduced approximately 4[dB] from the current engine.

Figure 35 shows the comparison of contact surface shape between engine and transmission. The surface are has been enlarged to improve powertrain rigidity. As the result, the engine noise has been reduced and a comfortable driving has been brought to customers.

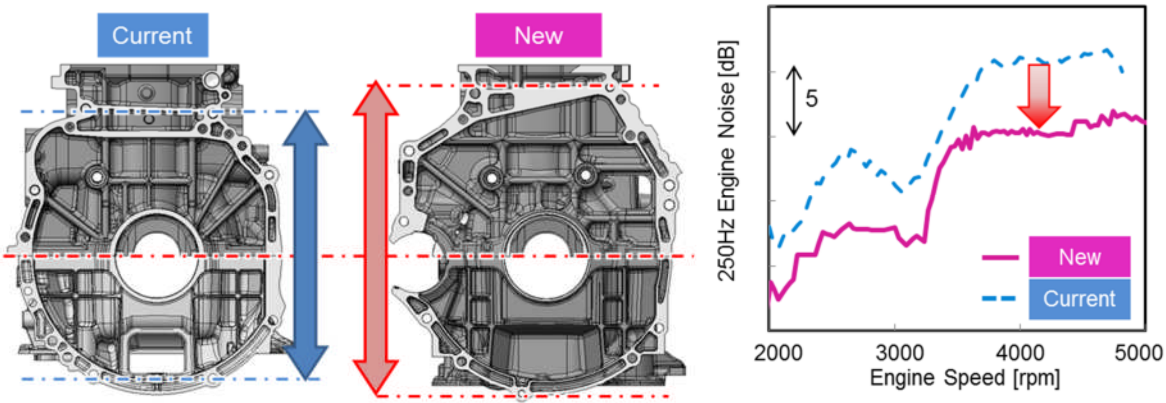



**Figure 35**: Comparison of engine contact surface shape and 250Hz engine noise

As the current engine, balancer which is cassette type and driven directly by crankshaft gear (Figure 36) has been implemented for the customer confortable driving. Some gears are made by resin with aramid fiber reinforcement for preventing gear noise same as current technology.
Generally, high speed combustion increases high frequency band noise. To prevent high frequency noise, the noise insulations made by polyurethane have been assembled onto the right place (Figure 37).

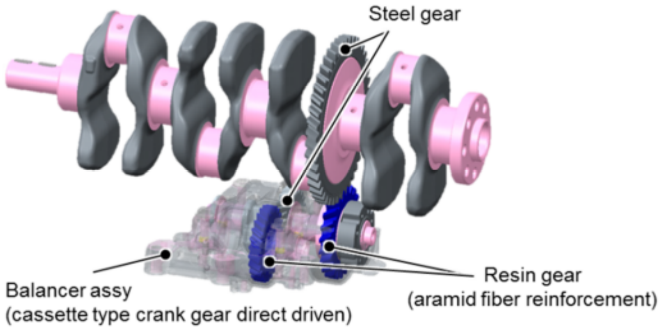



**Figure 36**: Balancer

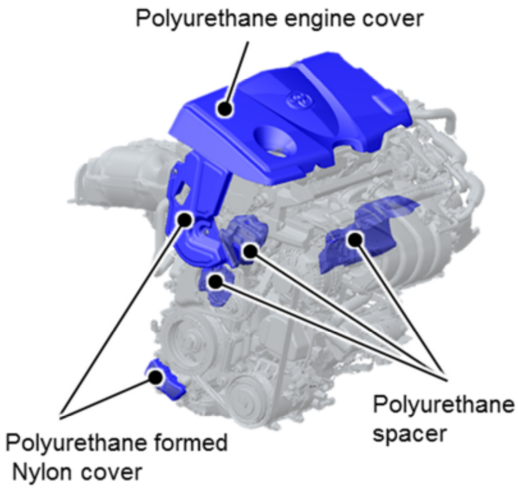



**Figure 37**: NV insulation items

## 3. Engine performance

Figure 38 shows the comparison of the engine power and torque curve of current and new engines. For conventional, 60kW/L specific power has been achieved and torque has been improved in all over the engine speeds. And for HV, 50kW/L has been achieved and torque has been improved all over the engine speed, too.

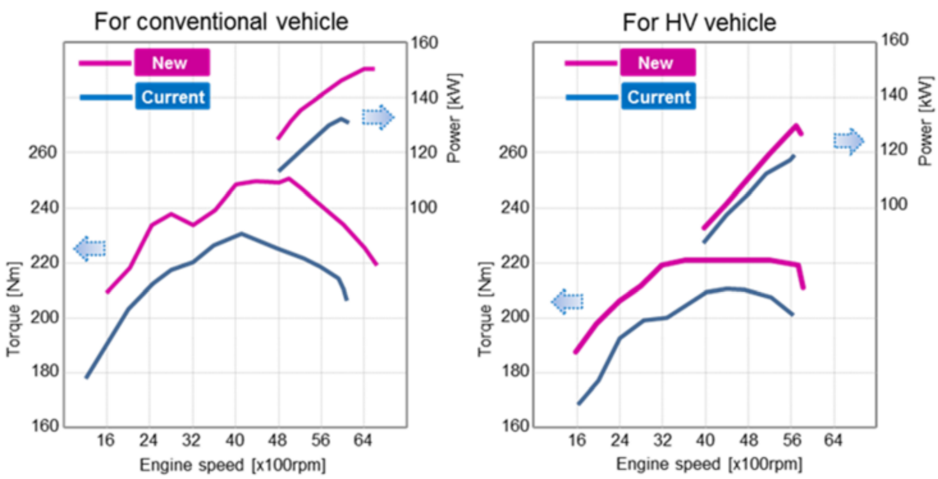



**Figure 38**: The engine power and torque curve

Figure 39 shows the thermal efficiency map of the 2.5L current engine and new engine for conventional vehicle. Figure 40 shows the same for HV. Not only achieving 40% (for HV:41%) of maximum thermal efficiency, but also 35% (for HV:38%) has been realized in wide range of operating conditions. By combining with new 8-speed AT, more than 16% lower fuel consumption has been achieved.

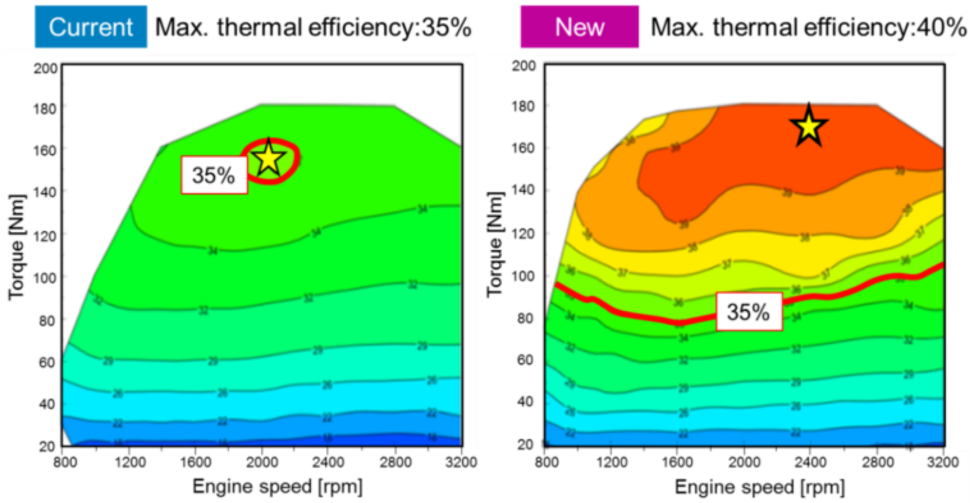



**Figure 39**: Comparison of thermal efficiency map (conventional vehicle)

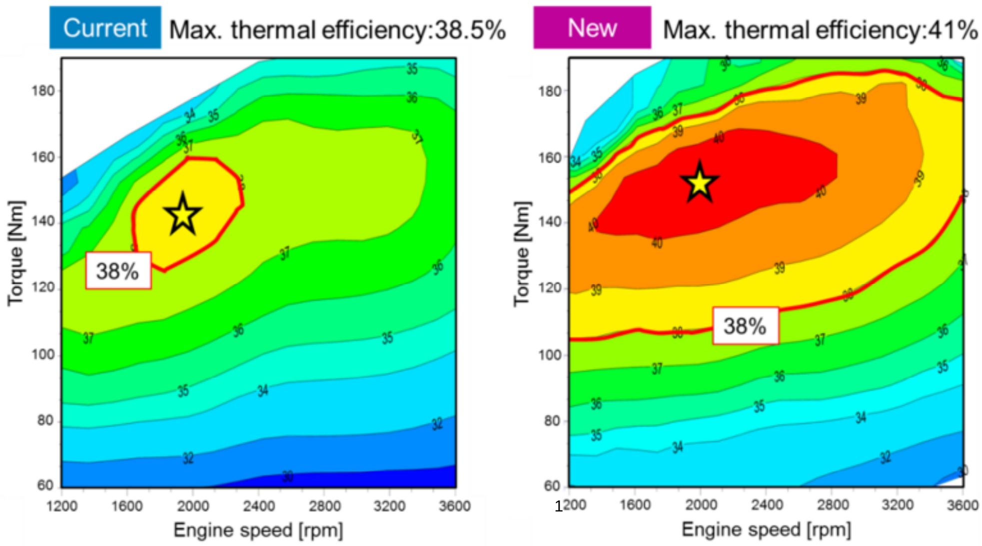



**Figure 40**: Comparison of thermal efficiency map (HV)

## 4. Driving performance

Compared to a vehicle developed focusing on dynamic performance, conventional engine performance targets include steady-state stand-alone engine performance aspects such as maximum power, low-end torque, and maximum torque, as well as wide-open throttle (WOT) vehicle acceleration behavior from a standing start (0 to 100 km/h) and for overtaking (50 to 80 km/h). To develop a vehicle that truly satisfies customer requirements, it is important to define and achieve targets for dynamic performance based on subjective feelings and perceptions, instead of simply achieving certain values.

This development has been carried out from the standpoint of enhancing dynamic performance by combining the new engine with a newly developed 8-speed AT. In this paper, this new developed powertrain based on TNGA concept is called as “TNGA powertrain“. Therefore, in addition to WOT performance, targets has been also defined for fun-to-drive performance under partial operating load conditions and AT shifting performance with respect to longitudinal vehicle acceleration (G).
A new development process hs been adopted with the aim of realizing these targets from the standpoint of the overall powertrain. Figure 41 outlines the concept of the target dynamic performance of the TNGA powertrain. The most important point of this concept is to achieve dynamic performance that matches the driver’s intended operation. The keywords “smooth” and “direct” has been adopted in the development to achieve linear operation with respect to the driver’s accelerator pedal operation.

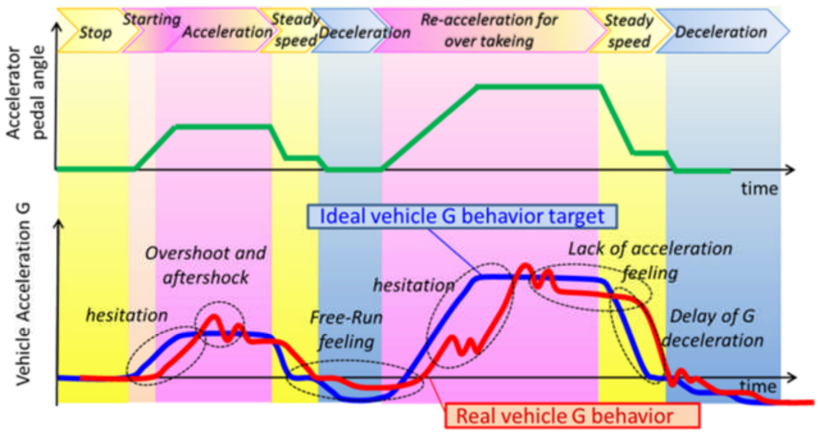



**Figure 41**: Target dynamic performance concept for TNGA powertrain.

## 4.1 Driving force demand control system

As electronic vehicle control systems evolve, control systems over the entire vehicle are becoming larger, more complex and more sophisticated as enhancement of product appeal. Toyota takes a step-by-step approach to the development of rational platforms from the standpoint of the whole vehicle. Under this approach, powertrain systems have also been refined on a step-by-step basis and the driving force demand control has been revised in accordance with the new engine control structure that has been re-developed using the new engine series. Figure 42 illustrates the powertrain driving force control system for the driving force generated in a TNGA powertrain.

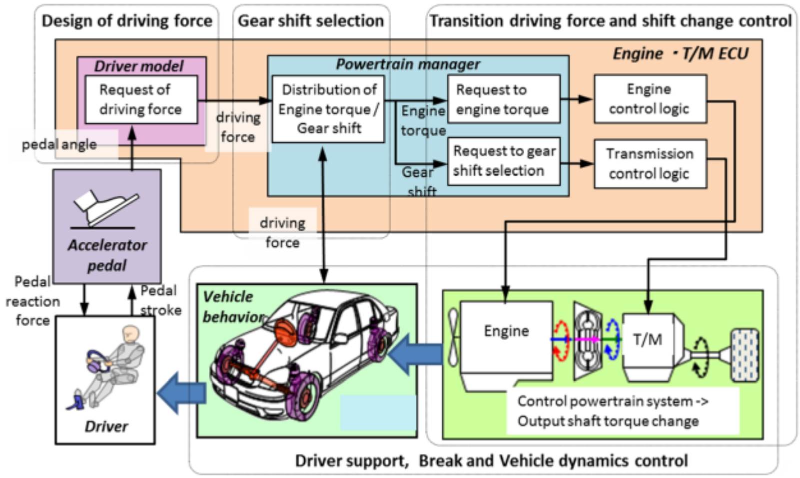



**Figure 42**: Outline of driving force demand control system

## 4.2 WOT acceleration performance

Figure 43(left side) shows comparison of the 0 to 60 mph target G with the actual vehicle G behavior of new and current models. The graph indicates that the driving force and torque response have been improved to meet the targets. Compared to the current model, the new model has a stronger initial G increase and a higher peak G. The speed of G increase declines after reaching its initial peak and the drop in G just after shifting from first to second gears is lower, and smooth and continuous G waveform has been achieved. Figure 43 (right side) shows the target G under WOT acceleration from 50 to 80 km/h. Under this condition, acceleration is performed by kick-down shifting. This creates a sluggish response in the initial phase as the shifting is carried out. It is important to shorten the lag between the initial G response feeling and depressing the accelerator and the sluggish G response as the subsequent shifting is carried out. The new engine is combined with a newly developed AT, which improves the initial G response and achieves the subsequent high peak G and smooth acceleration feeling.

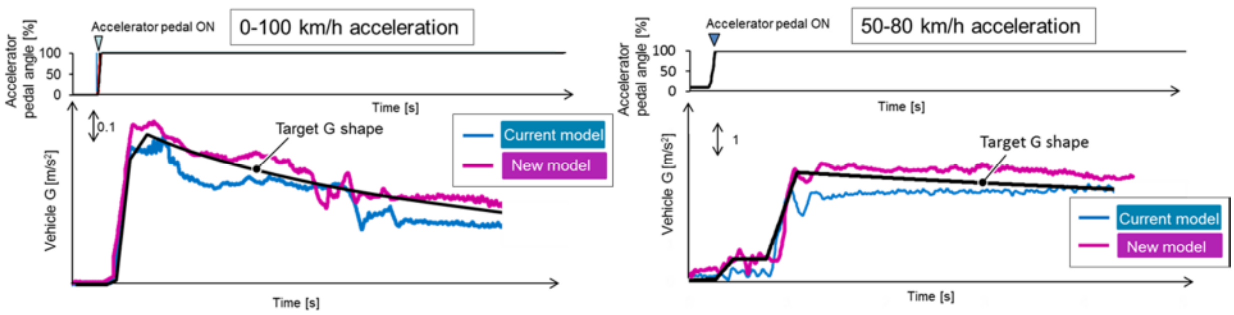



**Figure 43**: Target G waveform under WOT acceleration

## 4.3 Partial accelerator pedal angle acceleration performance

Fun-to-drive acceleration performance with partial accelerator pedal angles has been defined in the development. It is relevant to a wide range of driving situations. The target performance under partial acceleration can be also defined by determining the G behavior. In addition, the G curve after peak G can be also set in accordance with the accelerator pedal angle that enables a feeling of constant acceleration following the Weber Law.

Figure 44 shows the G behavior under sweep acceleration at partial accelerator pedal angle. Setting the driving force based on the Weber Law achieves the target fun-to-drive smooth acceleration. In addition, a control to improve transient throttle response has been adopted using a refined transient air model to improve the continuous G feeling during and after shifting up, and to achieve the target driving force characteristics with high precision.

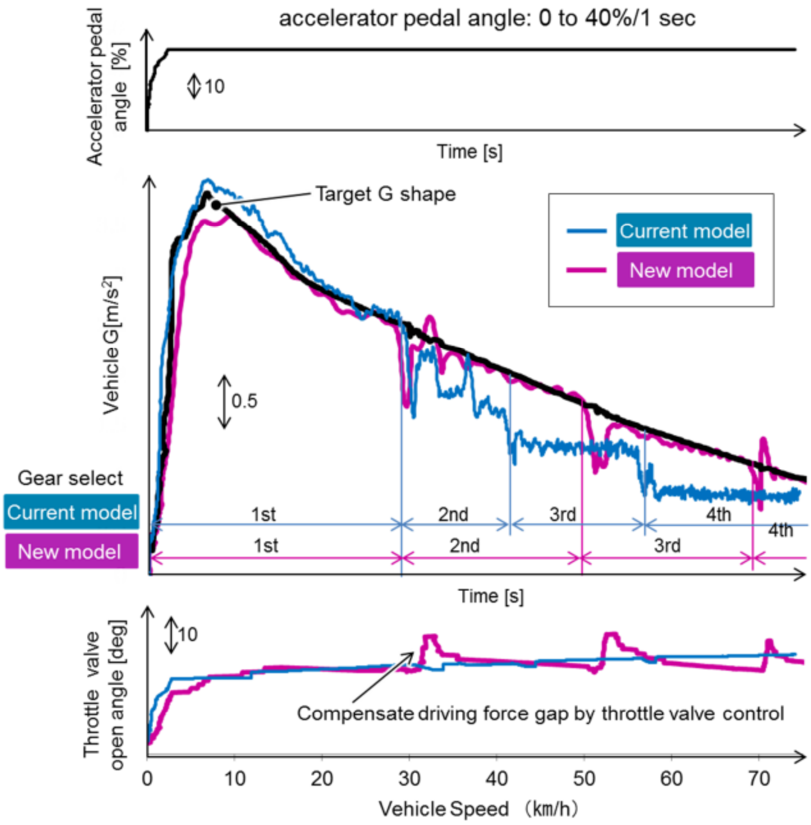



**Figure 44**: Comparison of vehicle G behavior under sweep acceleration

## 4.4 Improvement of accelerator pedal G controllability

In normal driving, the behavior of vehicle in accordance with the accelerator pedal operation in both accelerating from a standing start and accelerating or decelerating from a particular speed (i.e., whether the intended G controllability is achieved) helps to create a fun-to-drive feeling in which the vehicle responds in accordance with the driver's intention. This section explains improvement the G controllability under sweep acceleration from a steady speed of 50 km/h. Figure 45 shows comparison of vehicle G behavior under sweep acceleration. This new engine adopts a control to improve transient throttle response using a transient air model. This control reduces the effect of intake delays caused by the intake volume between the throttle and the cylinders. The top graph shows current model G behavior, the second is same as new model. The new model has realized better followability of G behaivior to acceleration pedal operation. In addition, the expanded lock-up area of the new 8-speed AT restricts engine speed flaring while achieving a linear torque response to accelerator pedal operation.

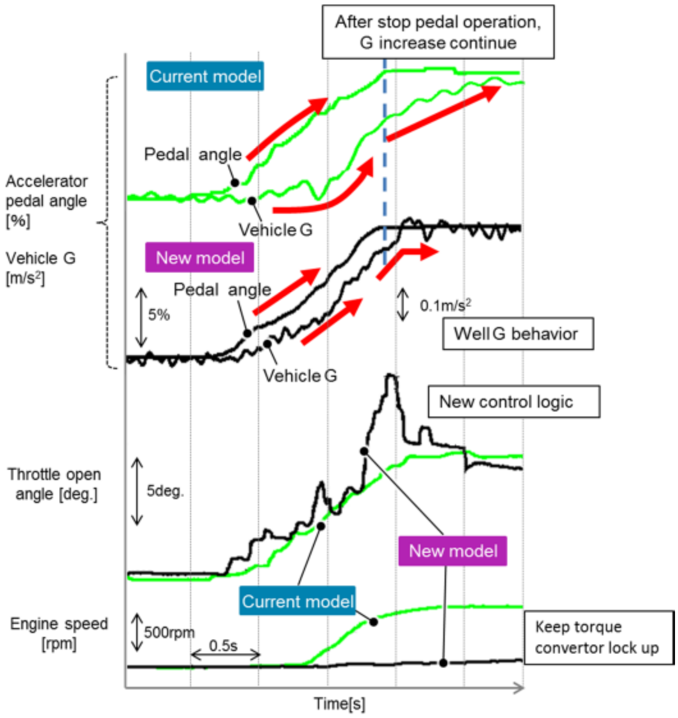



**Figure 45**: Comparison of G controllability under sweep acceleration (50 km/h, accelerator pedal angle: 10 to 20% /1 sec).

## 5. Modular design

Hereafter, the new engines have modular design concept under high speed combustion as common architecture. Firstly, based on this common architecture, Toyota has defined some geometory to realize this combustion and can adopted different cylinder size. For example, some of them are S/B ratio, intake port design for tumble ratio and so on as mentioned previously. Not only design geometory but also implemented techincal items, for example laser cladding valve seat, cooling concept and exhaust direction, have been adopted as modular design.

And the production point of view, production flexibility through the all new engines is very important. The current mass production line has produced just 1 seriese engine for efficiency. But, in case another engine seriese needs to produced, another exclusive line has to be prepared.

To cope with the ever changing costomers‘ requirements, vehicle type (body type, size and powertrain) needs to be continuously updated all over the world. This module concept can

realize to produce different cylinder modules in the same line. The machinig datum is unified over some displacement cylinder modules and engine compornent parts are assembled almost same location, same assembling process from large size to small size engine.
And then, this frexible line will replace to current production lines globally for the customer satisfactions.

## 6. Conclusion

This paper has described the hardware and system characteristics and new driving control of new Toyota inline 4-cylinder 2.5L Gasoline Engine, based on “Toyota New Global Architecture (TNGA) concept“. The main points of this engine can be summarized as follows.
1) High speed combustion package with long stroke, high tumble ratio and flow coefficient has been developed. For conventional, 40% (HV:41%) of maximum thermal efficiency and 60kW/L (HV:over 50kW/L) of specific power have been achieved.
2) As method and devise to realize high speed combustion, laser-cladded valve seat, spherical-shaped piston to keep a high tumble flow have been carried out.
3) The cooling system with electric water pump has been implemented even to conventional engine. High engine performance and low fuel consumption have been balanced with improvement of the pump and FSV (Flow Shut Valve) control in warm-up process and optimization of the most suitable water temperature in each driving condition.
4) Electronic control-type variable capacity oil pump has been developed and the combination of other items has realized the friction reduction more than 20%.
5) Dynamic performance targets have been determined for both WOT acceleration performance and daily driving situations, So, the new model has realized fun-to-drive with direct and smooth acceleration response.
6) Using a new driving force demand control system, a common interface has been established between the engine, transmission and vehicle based on driving force. This interface enables the parallel development and calibration of each system by defining the required driving force to achieve the targeted dynamic performance at the begining of development, thereby the development and calibration process has become more efficient.

References

1. Ohta, Y., Fushiki, S., and Matsuo, S., “The New PRIUS Powertrain: The New 1.8L ESTEC 2ZR-FXE Engine with the New Generation Hybrid System,” presented at the 37th International Vienna Motor Symposium 2016, Austria, 2016.
2. Nakata, K. and Shimizu, R., “Toyota’s New Combustion Technology for High Engine Thermal Efficiency and High Engine Output Performance,” presented at the 37th International Vienna Motor Symposium 2016, Austria, 2016.

3. Chen, S. and Flynn, P., “Development of a Single Cylinder Compression Ignition Research Engine,” SAE Technical Paper 650733, 1965, doi:10.4271/650733.
4. Watanabe, I., Kawai, T., Yonezawa, K., and Ogawa, T., “The New Toyota 2.0-Liter Inline 4-Cylinder ESTEC D-4ST Engine - Turbocharged Direct Injection Gasoline Engine -,” presented at the 23rd Aachen Colloquium Automobile and Engine Technology, Germany, 2014.
5. Yoshihara, Y., Nakata, K., Takahashi, D., Omura, T., and Ota, A., “Development of High Tumble Intake-port for High Thermal Efficiency Engines,” SAE Technical Paper 2016-01-0692, 2016, doi:10.4271/2016-01-0692.
6. Tokuda, S., Kubota, M., and Noguchi, Y., “Development of CFD Shape Optimization Technology using the Adjoint Method and its Application to Engine Intake Port Design,” SAE Technical Paper 2013-01-0969, 2013, doi:10.4271/2013-01-0969.
7. Murase, E., Shimizu, R., “Innovative Gasoline Combustion Concepts for Toyota New Global Architecture,” presented at the 25th Aachen Colloquium Automobile and Engine Technology, Germany, 2016.
8. Kuroda, M., “Nurturing of “Performance Feel,”” MAZDA Technical Review, No. 21, pp158-16, 2003.
9. Tanaka K., “A Study on the Mechanism of Driving Pleasure” Jounal of SAEJ, Vol. 70, No. 4, 2016
10. Matsumoto S,. “ Development of Dynamic Performance for the New Prius”, TOYOTA Technical Review, Vol. 63, No. 232, 2016.