

Honda R&D

The background of the cover features a close-up, artistic photograph of several Honda turbine compressor wheels. These wheels are arranged in a circular pattern, with some in sharp focus and others blurred in the background, creating a sense of depth. The wheels have a complex, multi-bladed design. The entire image is overlaid with a semi-transparent light blue filter, which makes the text stand out while still showing the intricate details of the turbine parts.

Technical Review

April 2017
Vol.29 No.1



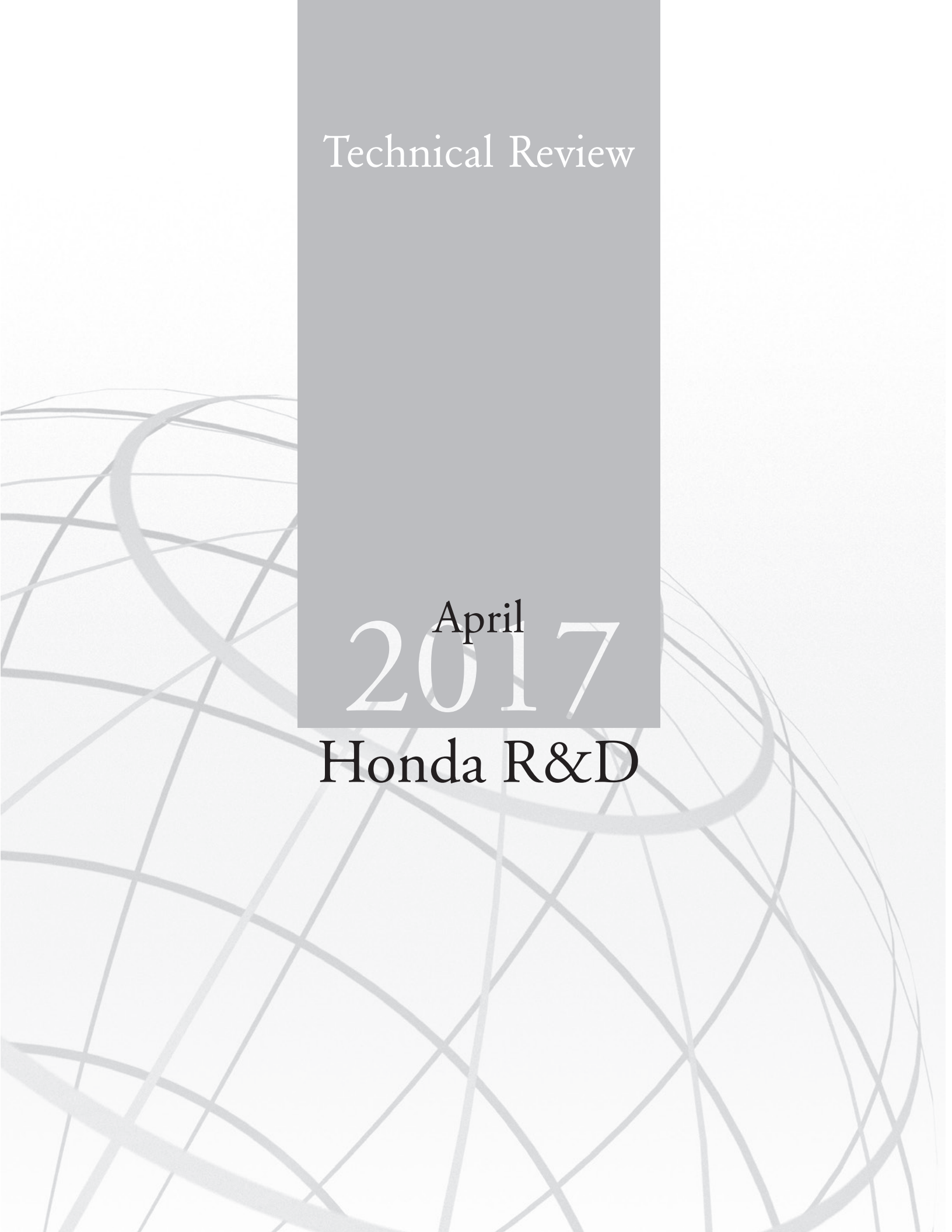
Comment on the Cover

The motif used in the cover design for this issue of Technical Review is based on fans employed in the cooling of small Honda power product engines. While simply a component when fitted in an engine, their form is born out of the natural phenomenon of wind. Depending on the viewer's perspective, they might resemble flowers, or the shells of sea creatures.

Considering that Honda's engineers grapple with natural laws on a daily basis, the outcomes that they create may be thought of as "natural objects" in themselves.



Power Products R&D Center
Michihito OKADO

The background features a light gray wireframe sphere on the left side, composed of intersecting curved lines. A solid gray vertical bar runs down the center of the page, serving as a backdrop for the main text.

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April
2017

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Development of New 1.0 L I3 Gasoline Direct Injection Turbocharged Downsizing Engine

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ABSTRACT

A new 1.0 L inline 3-cylinder VTEC TURBO engine that achieves enhanced fuel economy and driving performance has been developed as part of the cluster of new-generation power train technologies known as EARTH DREAMS TECHNOLOGY, which realizes both environmental and driving performance.

This engine has the same combustion concept as the previously released 1.5 L and 2.0 L 4-cylinder VTEC TURBO engines, and also applies common technologies such as high-tumble-flow intake ports, multi-hole injectors, and valve timing control of both the intake and exhaust camshafts. In addition, technologies such as a variable displacement oil pump, timing belt in oil system, and narrow crankshaft have been applied to comprehensively reduce friction. A compact class vehicle with this engine installed achieved 26% better fuel economy in European driving mode compared to the previous engine, and a newly developed compact turbocharger helped improve driving performance with 20% higher low- and medium-speed torque. Furthermore, both vibration and noise were kept in competitive level.

1. Introduction

New and stricter target values for fuel economy regulations have been set for compliance in developed countries in Europe, North America, and elsewhere from 2020 to 2025. Developing countries are also moving to implement those targets, so OEM manufacturers in Europe, in particular, are developing turbocharged downsizing engines as a technology that both complies with those fuel economy regulations and achieves product competitiveness.

Honda has released 1.5 L^{(1), (2)} and 2.0 L^{(2), (3)} 4-cylinder VTEC TURBO engines as part of the cluster of new-generation power train technologies known as EARTH DREAMS TECHNOLOGY, which increases both environmental and driving performance. These VTEC TURBO engines achieve equivalent dynamic performance with a smaller displacement than conventional naturally aspirated (NA) engines, and friction has also been reduced to enhance fuel economy.

These VTEC TURBO engines are positioned as world strategy engines intended to help meet fuel economy

regulations and increase product competitiveness in the European, North American, Chinese, and other markets. The newly developed 1.0 L 3-cylinder VTEC TURBO engine is also being marketed as part of this engine series. This paper introduces the construction, main technologies, and performance achieved by the developed engine.

2. Development Goals

Turbocharged downsizing engines have smaller displacements compared to an NA engine of equivalent power, so the friction is less, and use of the high-load range at low engine speeds makes it possible to reduce the pumping loss and enhance fuel economy. In addition, turbocharging provides sufficient driving performance even with smaller displacement when high power is needed during acceleration and high-speed driving. A 1.0 L 3-cylinder engine was newly developed that uses downsizing and various friction-reducing technologies to enhance fuel economy. The development goals were as follows.

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- (1) 20% or greater enhancement of same vehicle class fuel economy compared to the previous 1.8 L NA engine
- (2) 20% enhancement of low- and medium-speed torque
- (3) Reduction of 3-cylinder characteristic vibration produced by reciprocating mass

The displacement was selected in consideration of the friction reduction rate relative to the 1.8 L NA engine, which meets the above-noted fuel economy target, and the acceleration response during starting when turbocharging does not function, which is an issue for downsizing. Figure 1 shows an overview of the developed engine.



Fig. 1 Overview of developed engine

3. Main Engine Specifications

Table 1 compares the specifications of the developed engine with those of the current 1.5 and 1.8 L NA engines. In order to maximize the utilization of existing production facilities, the main dimensions, such as bore diameter, are the same as those of the current 1.5 L NA engine. Approximately 50% of the parts were newly developed for this engine.

The main feature of the developed engine is the use of friction-reducing technologies such as a narrow crankshaft, timing belt in oil system, variable displacement oil pump, and electronically controlled thermostat. In addition, the heat mass of the exhaust system upstream to the catalyst is reduced by mounting the turbocharger directly to the cylinder head integrated with the exhaust manifold. This promotes early catalyst warm-up, reduces emissions, and suppresses the drop in fuel economy performance due to the air-fuel ratio enrichment needed during the warm-up process.

Table 1 Comparison of engine specifications

Engine specifications	Developed 1.0 L I3 turbocharged	Previous 1.5 L I4 NA	Previous 1.8 L I4 NA
Displacement [cm ³]	988	1496	1798
Bore [mm]	73	73	81
Stroke [mm]	78.7	89.4	87.3
Stroke / Bore [-]	1.08	1.22	1.08
Compression ratio [-]	10.0	11.5	10.6
Valve number In/Ex [-]	2/2	2/2	2/2
Valve diameter In/Ex [mm]	28/23	29/25	33/26
Crankshaft main/pin journal diameter [mm]	38/35	46/40	55/45
Piston with oil gallery	Applied	Not applied	Not applied
VTC	Dual VTC	In-VTC	None
VTEC	In	In	In
Variable displacement oil pump	Applied	Not applied	Not applied
Electrically controlled thermostat	Applied	Not applied	Not applied
External EGR system	None	Hot	Hot
Fuel injection system	Direct	Direct	Port
Camshaft drive	Timing belt	Timing chain	Timing chain
Fuel	RON 95	RON 95	RON 95
Power [kW/rpm]	95/5500	97/6600	106/6500
Torque [Nm/rpm]	200/2250	155/4600	174/4300

4. Construction and Achieved Technology

4.1. Cylinder Block and Cylinder Head

The cylinder block is made of die-cast aluminum alloy with an iron liner cast in. A dual cooling system structure that uses a sub-jacket formed in front of the cylinder block to split the coolant flows into two streams was used to enhance the cooling performance to compensate for the increased power density. One stream flows directly to the cylinder head, with the other flowing to the cylinder head via the cylinder block water jacket (Fig. 2).

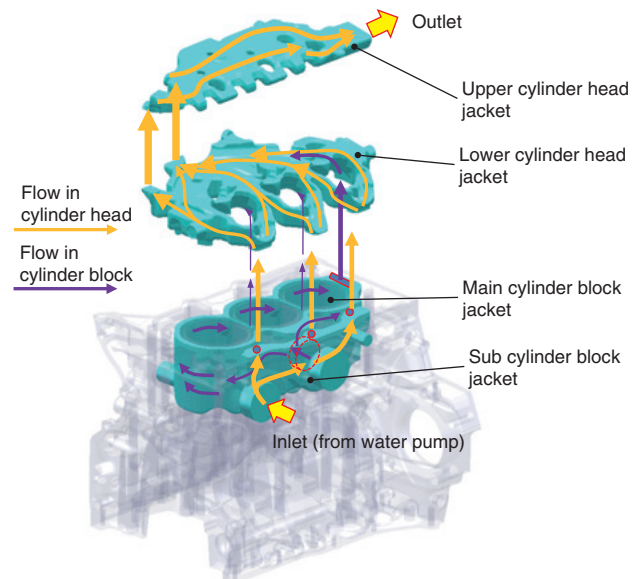


Fig. 2 Coolant flow in engine

The cylinder head uses a vertically split two-piece cooling jacket construction, and the coolant traverses the combustion chamber, enabling to evenly cool the combustion chamber of each cylinder (Fig. 3).

4.2. Cooling System

The electronically controlled thermostat shown in Fig. 4 was installed at the coolant channel outlet of the engine to both enhance fuel economy by reducing friction and to secure strength. The wax-element opening temperature of the electronically controlled thermostat is set to 103°C, which is 20°C higher than that of the conventional system. Increasing the coolant temperature under low-load conditions reduced friction by maintaining a high oil temperature, and enhanced mode fuel economy by approximately 0.6%. In addition, the wax-element piston has a built-in ceramic heater that enables opening at a lower coolant temperature than the set temperature by directly heating the wax. Under high-load conditions, power is supplied to the ceramic heater to lower the opening temperature and control the coolant temperature like in the conventional system, in order to secure strength.

4.3. Reciprocating Parts

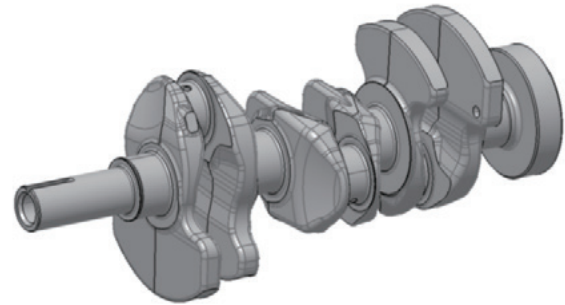
The crankshaft uses high-strength nitrided steel, and the main journal and crankpin journal diameters were reduced

in order to reduce friction. Figure 5 shows an overview of the crankshaft and compares the dimensions with those of the 1.5 L NA engine.

In addition, even vibration rooted in the first-order coupled inertia moment is a phenomenon that is a challenge to avoid in a 3-cylinder engine, a balancer-less system was used to avoid increases in friction and weight.

Countermeasures for vibration produced by the first-order coupled inertia moment included reducing the weight of reciprocating parts and setting a crankshaft balance ratio that reduces vertical vibration at the engine mount points, which greatly contributes to body vibration. Balance ratios of 0% and 75% were compared when developing this engine, and 75% was selected.

Figure 6 shows the results of comparing the vibration at powertrain mount points due to the first-order coupled inertia moment with that of competing vehicles. This



	1.0 L I3 turbocharged	1.5 L I4 NA
Pin bearing diameter [mm]	35	40
Main bearing diameter [mm]	38	46
Crank radius [mm]	39.16	44.5

Fig. 5 Crankshaft overview

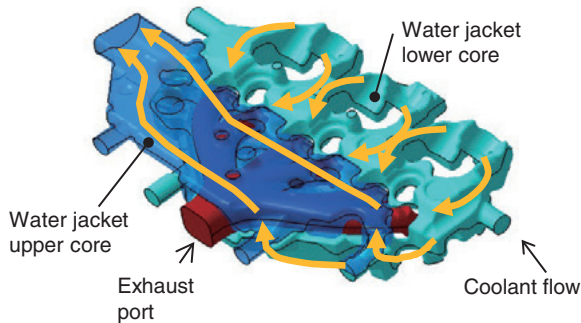


Fig. 3 Configuration of two-piece water jacket and coolant flow in cylinder head

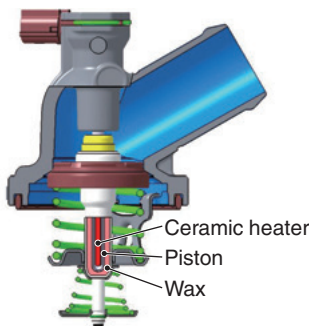


Fig. 4 Sectional view of electrically controlled thermostat

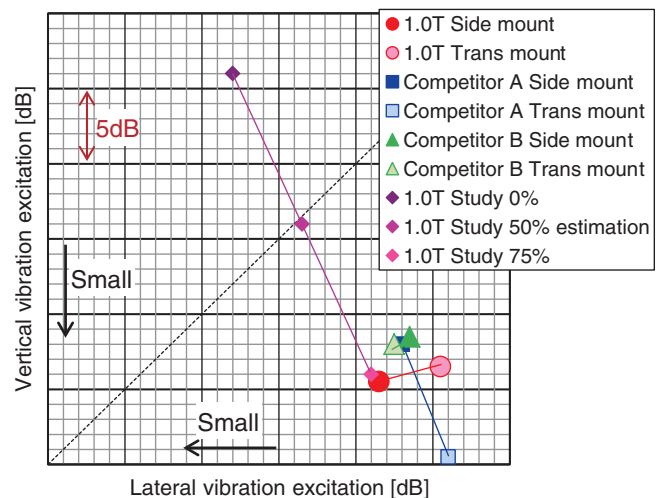


Fig. 6 1st order vibration excitation of I3 turbo charged engine at 1000 rpm

confirmed that the developed engine is sufficiently competitive with rival vehicles.

The pistons use a cooling channel to compensate for the increased thermal load due to turbocharging (Fig. 7). This reduced the piston crown temperature by approximately 20°C, enhancing both strength and the knocking characteristic, leading to a higher compression ratio, thereby enhancing fuel economy by 0.5%.

Like the 1.5 and 2.0 L VTEC TURBO engines, the connecting rod used the controlled forging method to compensate for the high compression load due to turbocharging. The controlled forging method uses a two-step forging process to reinforce the I-beam area. Increasing the buckling strength of the I-beam area made it possible to reduce the weight by 18% compared to conventional forging technology.

4.4. Valve Train

The valve train uses valve timing control (VTC) for both intake and exhaust, with a VTEC rocker arm on the intake side that can switch between high and low lift. Optimal VTC and VTEC control is performed according to the operating conditions such as the load and engine speed. This enhances fuel economy under low-load conditions while still delivering torque under high-load conditions. Figure 8 shows the valve train external view.

3-cylinder engines with a short overall length have limited layout space especially when installing a turbocharger, so an efficient layout for the variable valve control system is needed. The developed engine uses a VTC structure that positions the VTC control solenoid and oil control valve (OCV) over the camshaft as shown in Fig. 9.

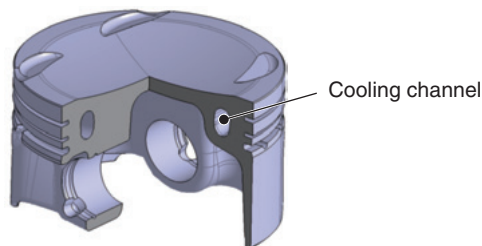


Fig. 7 Cut-off view of piston

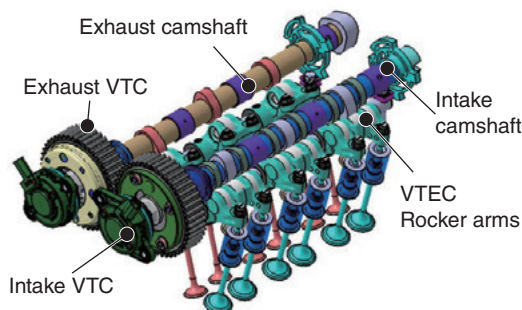


Fig. 8 Schematic view of valve train

This simplified the oil circuit, and the effects of the built-in check valve helped deliver high responsiveness relative to the VTC size.

A VTEC mechanism that switches between low and high lift is used on the intake side. In low-lift mode, fuel economy is enhanced by an early valve-closing Atkinson cycle described hereafter, which enhanced brake specific fuel consumption (BSFC) by up to 5% and mode fuel economy by 2%. Figure 10 shows the intake and exhaust valve timings and lifts, and Fig. 11 shows the VTEC and VTC switching strategy map. Regarding the application of VTEC to this engine, an early-closing Atkinson cycle valve

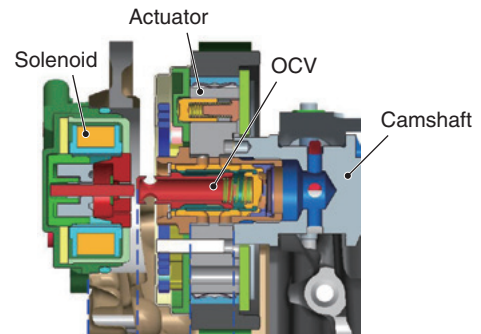


Fig. 9 Structural drawing of VTC

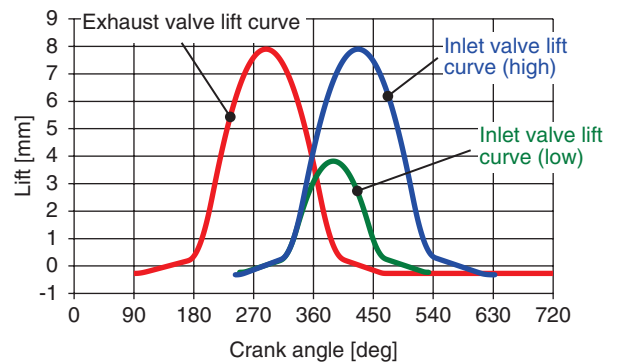


Fig. 10 Valve timing and lift

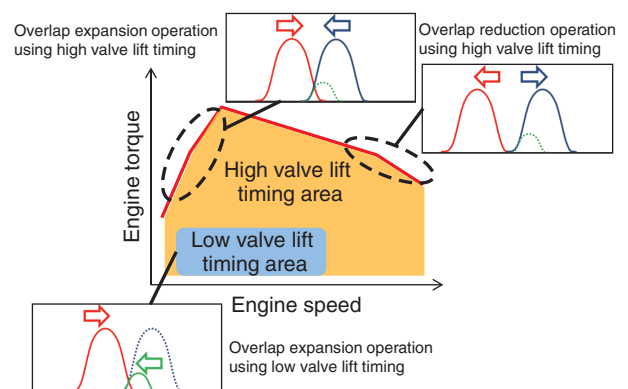


Fig. 11 VTEC and VTC switching strategy map

timing was used to enhance fuel economy under low-load conditions in consideration of the vehicle class in which the engine is mounted. While the Atkinson cycle reduces pumping loss under low-load conditions, it also restricts the valve timing and lift, which reduces the torque under high-load conditions, making it a challenge to simultaneously achieve both fuel economy and power.

Proprietary VTEC technology was used to address this issue, providing enhanced fuel economy under low-load conditions as well as sufficient torque under high-load conditions by switching the cam according to the load. Figure 12 shows the indicated diagram at each valve timing, and demonstrates that pumping loss is reduced by selecting early valve-close timing under low-load conditions.

The developed engine drives the camshaft and oil pump using timing belts located inside the engine (Fig. 13). The

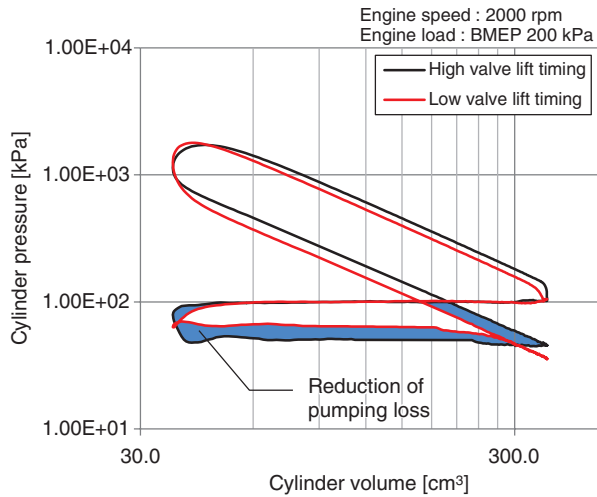


Fig. 12 Indicated diagram

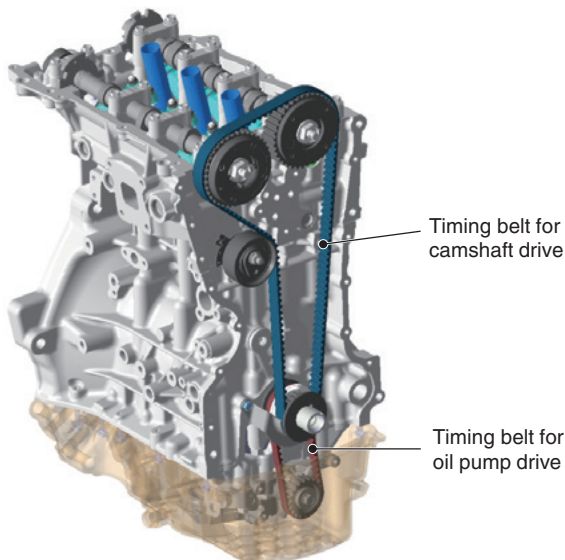


Fig. 13 Overview of timing belt system

Table 2 Comparison of timing belt materials

		Belt in oil	Conventional Belt
Tooth fabric	Material	PA66+Aramid	
	Surface treatment	Epoxy resin	H-NBR
Rubber	Material	H-NBR +Aramid fiber	
	Back face	Back face fabric	H-NBR

timing belts are located inside the engine, so the belt noise is reduced and engine friction is also reduced by 1.8% compared to a conventional timing chain. Timing belt materials were developed with specifications that provide sufficient durability for use in the oil system environment. Figure 14 shows the timing belt construction, and Table 2 compares the material specifications with those of a conventional timing belt in air.

4.5. Lubrication System

A variable displacement oil pump was used to reduce engine friction. The pump is located in the lower interior of the engine, and is driven by the crankshaft via a timing belt. An electronically controlled solenoid is used to switch the oil pressure between two levels in accordance with the engine load and speed, thereby maintaining the target oil pressure or less. This helps prevent the oil pressure from rising excessively even during the warm-up conditions of low oil temperature and high viscosity, further reducing friction and enhancing mode fuel economy by approximately 1%.

Variable oil pump capacity is achieved by use of a structure that changes the eccentricity by swinging the cam ring forming the outer ring of the vane pump (Fig. 15). Oil pressure applied to the chambers above and below the cam ring controls this eccentricity.

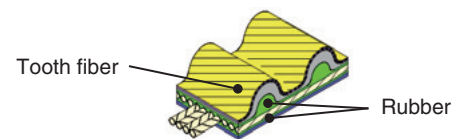


Fig. 14 Construction of timing belt

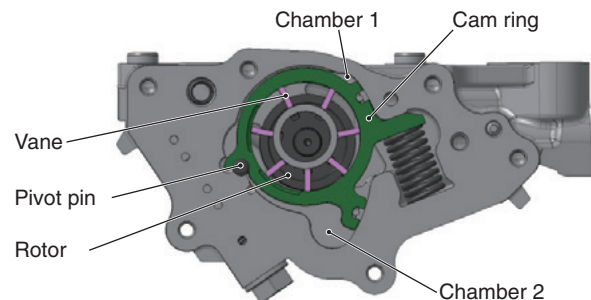


Fig. 15 Sectional view of oil pump

4.6. Turbocharger

A single scroll-type turbocharger was used (Fig. 16). Use of a compact turbine wheel reduced the inertia compared to a conventional turbocharger, and in combination with an electric waste gate actuator achieved high torque and high response from low engine speeds with enhanced fuel economy under low-load conditions. A turbine housing made of cast stainless steel offers high heat resistance, and the specifications do not rely on enriching the air-fuel ratio to reduce the exhaust temperature, even when exhaust gases are at high temperatures up to 950°C. And the higher permissible turbine speed enabled to support higher power (Table 3).

4.7. Combustion Concept

The combustion concept of the developed engine is rapid combustion like the previously released 1.5 L and 2.0 L 4-cylinder VTEC TURBO engines. This was achieved by the use of high-tumble flow ports, VTC on both the intake and exhaust camshafts, and multi-hole direct side injectors, which creates high-tumble flow with the optimal air-fuel mixture, and enhances fuel economy by reducing the time loss in the high-load range. Figure 17 compares the port and combustion chamber shapes of the developed engine and the 1.5 L NA engine. The developed engine uses a

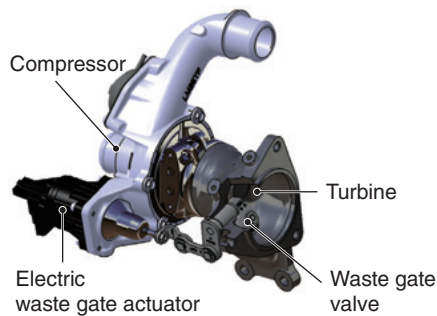


Fig. 16 Cut-off view of turbocharger

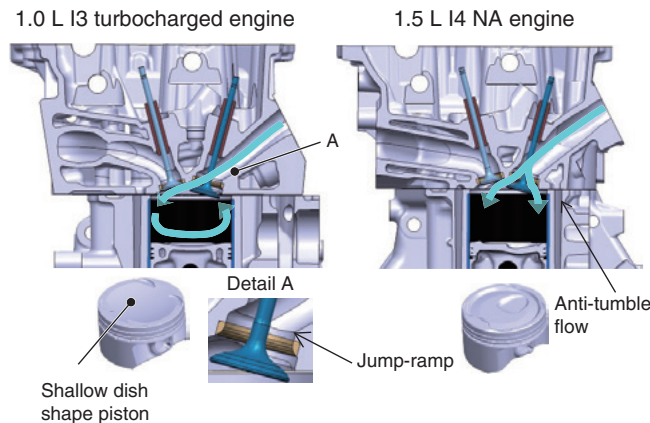


Fig. 17 Comparison of intake port and piston top shape

Table 3 Comparison of turbocharger specifications

Turbocharger specification		1.0 L	1.5 L
Max. power [kW/rpm]		95/5500	110/5500
Max. torque [Nm/rpm]		200/2250	203/1600-5000
Turbocharger supplier		Borg Warner	Mitsubishi
Turbine	Scroll	Mono	Mono
	Wheel diameter/vane	$\phi 31/10$	$\phi 37/11$
	Permissible speed	285000 rpm	245000 rpm
	Housing material	DIN1.4837+Nb [Austenitic cast steel]	A3K [Austenitic cast steel]
Compressor	Wheel diameter/vane	$\phi 37/5+5$	$\phi 46/6+6$
	Permissible speed	285000 rpm	241000 rpm
Waste gate	Actuator	Electric	Electric
	Valve diameter	$\phi 20$	$\phi 28$

flatter intake port angle compared to the NA engine, which realizes an intake flow along the pentroof wall surface on the exhaust side (Fig. 18). In addition, by providing a small raised edge at the bottom of the intake port (Fig. 17 detail A), the main intake airflow is guided to the upper part of the combustion chamber, strengthening the tumble flow.

The direct injector hole layout and the size of each injector hole were selected to reduce fuel adherence to the piston and liner in order to suppress soot generation and dilution of the lubricating oil, and to increase the homogeneity of the air-fuel mixture to ensure rapid combustion (Table 4).

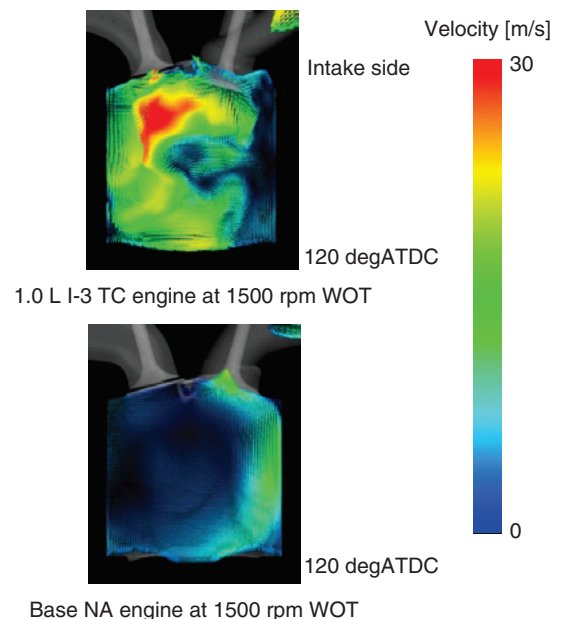
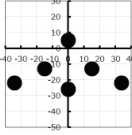
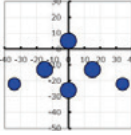
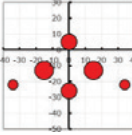


Fig. 18 Comparison of in-cylinder flow (CFD analysis)

Table 4 Injector hole configuration

	Specification at early stage	Specification at middle stage	Specification at final stage
Spray targeting and hole size arrangement			
Fuel flow distribution	Even	Uneven (small)	Uneven (large)
Piston impingement	+	++	++
Liner impingement	-	+	++
Homogeneity	+	+	+
Ranking	3	2	1
++: Good +: Average -: Poor			

5. Achieved Performance

5.1. Power

This engine was developed with the goal of achieving torque characteristics equal to or better than those of the 1.8 L NA engine, to enable use in passenger car categories that include small to medium-sized vehicles. A compact turbocharger with an electric waste gate increases torque, especially in low- and medium-speed ranges. In addition, setting the optimal valve timing in each engine speed range in consideration of scavenging by VTC and internal EGR made it possible to generate a maximum torque of 200 Nm at 2250 rpm. Figure 19 compares the power characteristics with those of the 1.5 and 1.8 L NA engines. The torque at 2250 rpm is 20% higher than that of the 1.8 L NA engine, helping ensure stress-free driving. In addition, maximum power of 95 kW generated even with a displacement of only 1.0 L realizes a good balance between torque and power that can be utilized in various vehicle classes.

5.2. Fuel Economy

Figure 20 shows the position of the developed engine in terms of mechanical friction versus engine speed and mechanical friction of different displacements. Figure 21 shows the position of the developed engine in terms of BSFC at typical operating points. The mechanical friction level of the developed engine can be confirmed as near the lower limit of both engine speed and displacement trends, with BSFC positioned lowest among turbocharged engines of the same displacement^{(4), (5)}.

Figure 22 compares the BSFC of the developed engine with that of the 1.8 L NA engine. Optimal VTC and VTEC control achieved a BSFC of 240 g/kWh or less over a wide range, which enhanced both mode fuel economy and actual driving fuel economy by approximately 20%. Figure 23 compares the acceleration performance from 80 km/h to 120 km/h and mode fuel economy for the developed engine and the 1.8 L NA engine in the same vehicle class. This shows that mode fuel economy was enhanced by 26% while still delivering the same or better acceleration performance.

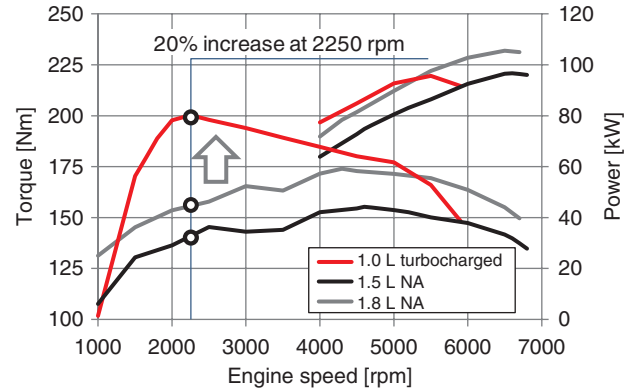


Fig. 19 Comparison of output curves

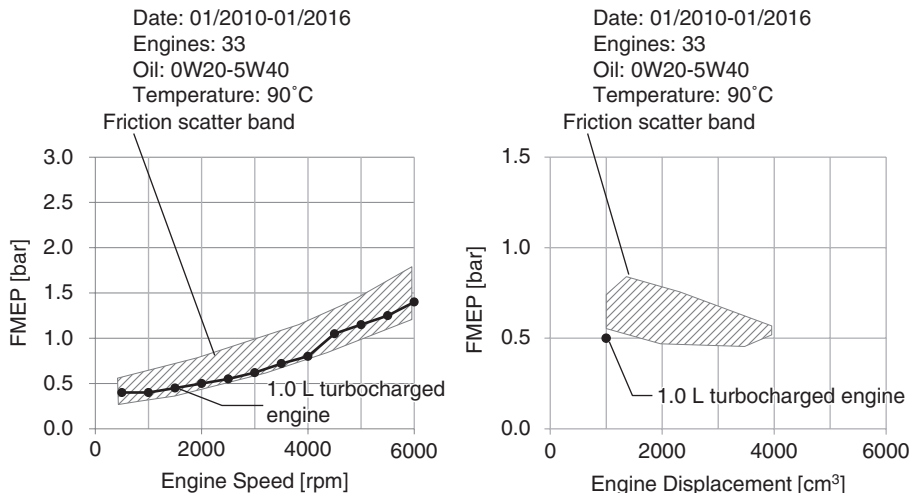


Fig. 20 Comparison of engine mechanical friction (Source of scatter band: FEV)

5.3. Emissions

In addition to hardware technology such as the abovementioned high-tumble-flow ports, the software was also modified to perform split fuel injection (up to three times) and multiple ignition (up to five times). This enhanced combustion in the cold engine condition and achieved EURO 6c level potential.

Enhanced combustion promoted catalyst warm-up by retarding the ignition timing, and decreasing the duration of inefficient catalyst warm-up also leads to enhancement of fuel economy.

5.4. NV

5.4.1. Engine radiation noise

Figure 24 shows the engine radiation noise characteristics. Turbocharged downsizing engines are known for typically increased noise at low engine speeds compared to NA engines. While the developed engine also exhibits this tendency, it is largely due to rapid combustion for the purpose of higher efficiency and increased low-speed torque.

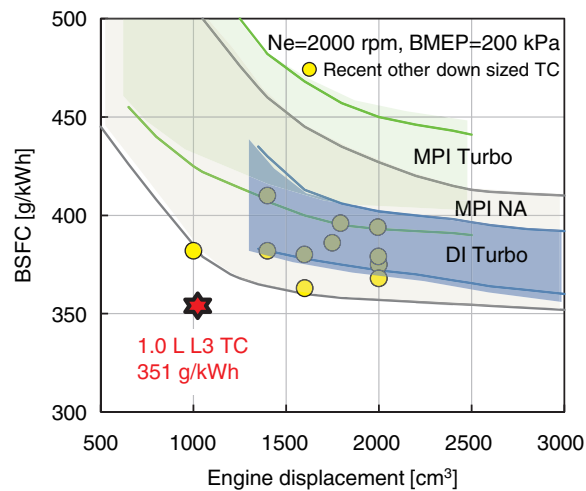


Fig. 21 BSFC (Source of scatter band: AVL)

Noise reduction measures were implemented for the main components, and also for noise radiated from the turbocharger and related piping, which does not occur in a conventional NA engine. As a result, compared to engines in the same torque and power bands, noise levels are positioned near the lower level of the turbocharged

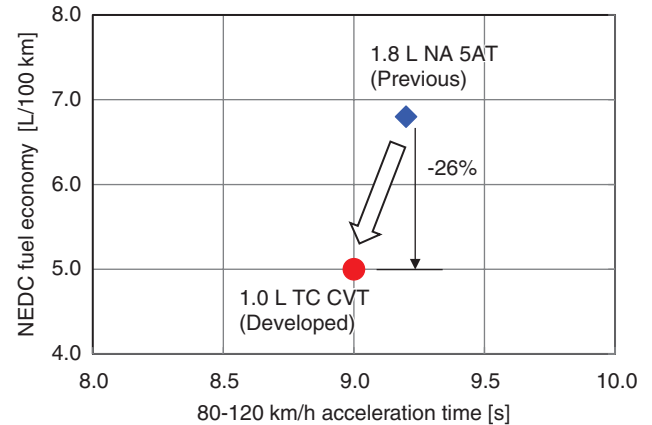


Fig. 23 Acceleration performance and fuel economy

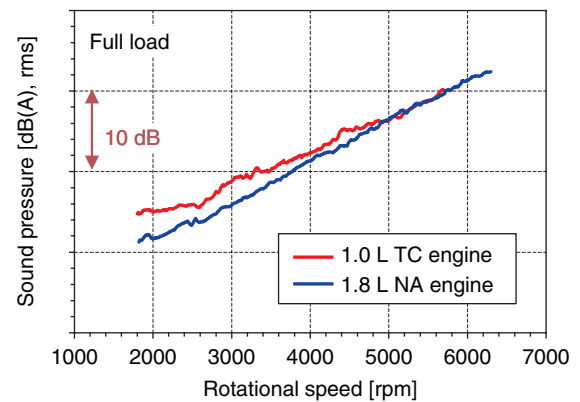


Fig. 24 Engine radiation noise

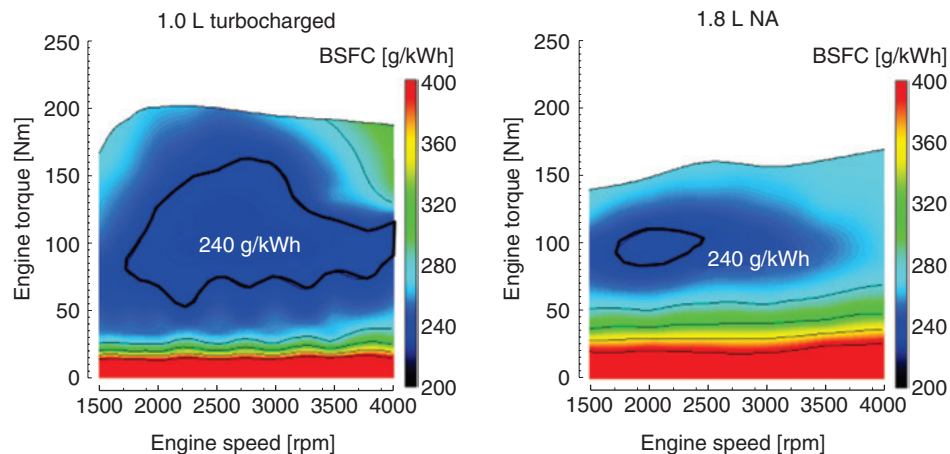


Fig. 22 Comparison of BSFC

engine group as shown in Fig. 25, demonstrating sufficient competitiveness of the developed engine.

5.4.2. Engine mount point vibration

Vibration levels are approximately in the middle compared to turbocharged engines in the same torque band, further demonstrating competitiveness (Fig. 26).

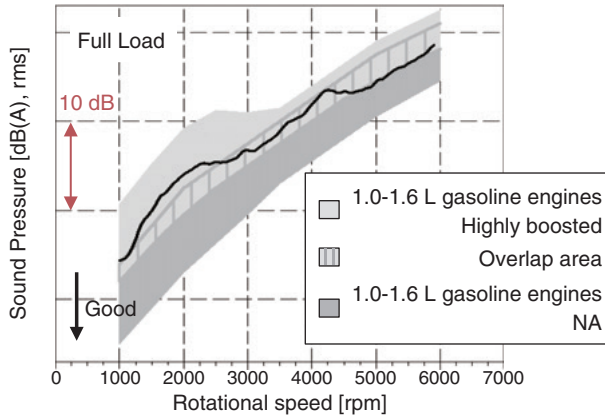


Fig. 25 Comparison of radiation noise
(Source of scatter band: FEV)

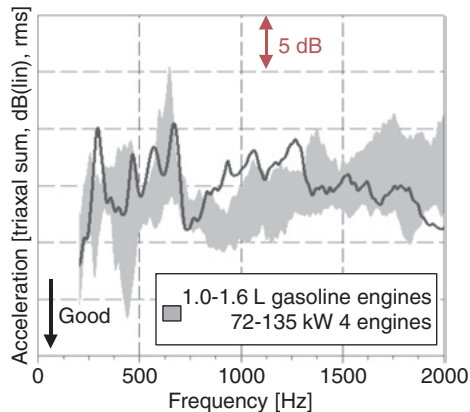


Fig. 26 Comparison of engine vibration
(Source of scatter band: FEV)

6. Conclusion

The use of downsizing and friction reducing technologies obtained the following high performance potential compared to the current 1.8 L NA engine.

- (1) Mode fuel economy enhanced by approximately 26% in the same vehicle class.
- (2) Low- and medium-speed torque enhanced by 20% while delivering comparable power.
- (3) Noise and vibration performance relative to low- and medium-speed torque help ensure sufficient product competitiveness.

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Vehicle Dynamics Performance Enhancing Technology for New CIVIC

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ABSTRACT

Technology was developed for the new CIVIC with the goal of achieving the top vehicle dynamic performance in the segment.

A new rear suspension with high rear equivalent cornering power was developed to achieve a balance between handling performance and ride comfort. Road-holding cornering performance and flat ride performance were simultaneously achieved by reducing the friction and hysteresis of the suspension moving parts to reduce the force transferred from the road surface to the body, and by non-coupling various vehicle behaviors. A dual-pinion variable gear ratio electric power steering system was used and the steering support stiffness was increased to simultaneously achieve both a smooth and linear steering feel and good handling over wide steering maneuver and vehicle speed ranges. Linear handling performance and good ride comfort from initial start to long-distance driving were realized through the use of body bulkheads with a ring structure that increases torsional stiffness and a lattice frame layout that increases floor stiffness. These technologies achieved top-level vehicle dynamic performance among same-segment vehicles around the world.

1. Introduction

The 2016 year model CIVIC (new CIVIC) was developed with the grand concept of “The CIVIC” based on the three pillars of Charismatic, Soulful, and Comfortable. Charismatic means achieving an advanced, high-quality design and a package with good interior comfort. Soulful means a new power train and platform that realize impressive driving which exceeds that of European vehicles with good dynamic performance. Comfortable means providing good ride comfort, high NVH performance, and the latest comfortable and reassuring equipment. Particularly with regards to Soulful, it was decided to provide remarkable performance that exceeds customer expectations, return to the sportiness that is the CIVIC mainstream, and create a new trend for the compact segment.

Based on the above grand concept, the value of the vehicle dynamic performance provided to customers was investigated, and the goal was set as a secure feeling that exceeds customer expectations and provides an impressive agile feeling.

Secure feeling means a high suspension response that enables tires to hold the road even on uneven surfaces, and performance that reduces passenger body and head shaking

and stabilized eye level.

Agile feeling means a quick body response with good turn-in ability and a low profile with a low center of gravity that holds the intended driving line.

Specific driving scenarios to be provided to customers and the performance items in those scenarios were derived from these goals.

Directional stability that enables a secure feeling even during high speed driving, limit handling that stabilizes vehicle behavior even during sudden steering, a smooth and accurate steering feel from small steering maneuvers, and good ride comfort from the start of driving to long-distance cruising were set as representative secure feeling items.

Good response to steering input, linear handling with little lag, and cornering performance that holds the vehicle low to the road were set as typical agile feeling items. Predicted advances by competitors were derived from comparative analysis and also taken into account to set competitive physical target values.

2. Performance Building Concepts

Development was performed in accordance with the following concepts to achieve the target vehicle dynamic performance for the new CIVIC.

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- (1) Provision of high rear equivalent cornering power (CP) as the basis for simultaneously achieving both handling performance and ride comfort.
- (2) Reduction of the force transferred from the road surface to the body and non-coupling of various vehicle behaviors to achieve both cornering performance that holds the vehicle to the road and flat ride performance.
- (3) Increased steering system support stiffness and separation of assist systems to achieve both a smooth and linear steering feel and good handling from small steering maneuvers to large steering angles and from low to high speeds.
- (4) Increased body stiffness to realize linear handling performance with little lag relative to steering input and good ride comfort from the start of driving to long-distance cruising.

3. Achieved Technology

Previously developed performance design technology^{(1),(2)} was used and various new technologies were applied to achieve the target performances in accordance with the performance building concepts. The main contents are described below.

3.1. Rear Suspension

A new multi-link rear suspension was used to simultaneously achieve both handling performance and ride comfort. Figure 1 shows an overview of the new suspension.

In order to realize the high lateral stiffness needed to enhance handling performance, the suspension geometry was set so that the lateral load is mainly shared by the lower and upper arms and the longitudinal load that has a high contribution to ride comfort is mainly borne by the trailing arm (Fig. 2).

The use of 6-point rigid connection between the rear subframe and the body increased the lateral coupling stiffness, which has a high contribution to enhancement of handling performance, and a high spring rate was set for the

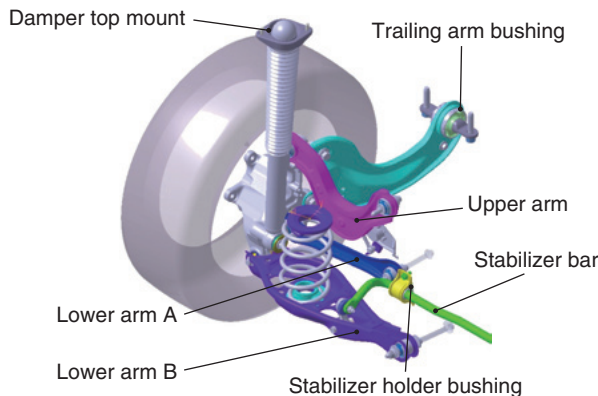


Fig. 1 Overview of new rear multi-link suspension

radial direction of the bushing that bears the lateral load.

As a result, in consideration of the balance with damping of disturbance, the equivalent CP gain was increased by reducing changes in alignment due to rolling and increasing changes in alignment due to lateral force compared to the previous model (Fig. 3).

Addition of a drawing bushing and intermediate plate to each bushing that supports the lateral force simultaneously achieved a high radial direction spring rate and a low torsional spring rate, and reducing the friction and the hysteresis element of the suspension moving parts achieved a smooth initiation of motion. In addition, a low spring rate was set for the trailing arm bushing, which has a high contribution to enhancement of ride comfort. This enabled simultaneous achievement of handling performance and ride comfort (Fig. 4).

Use of an input force dividing-type damper top mount enabled use of low-density, low-compression materials while still maintaining durability. Furthermore, changing from rubber materials to urethane materials enhanced the damping characteristic in the large amplitude range and

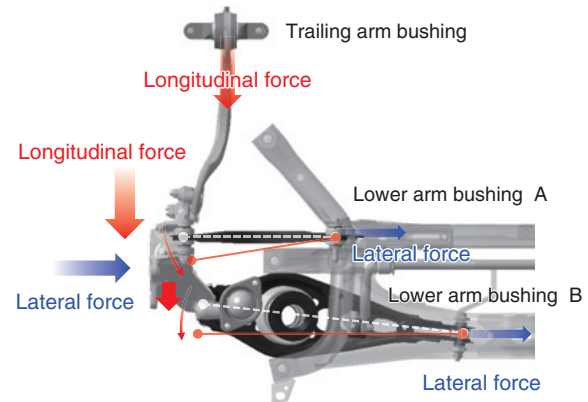


Fig. 2 Features of new rear suspension

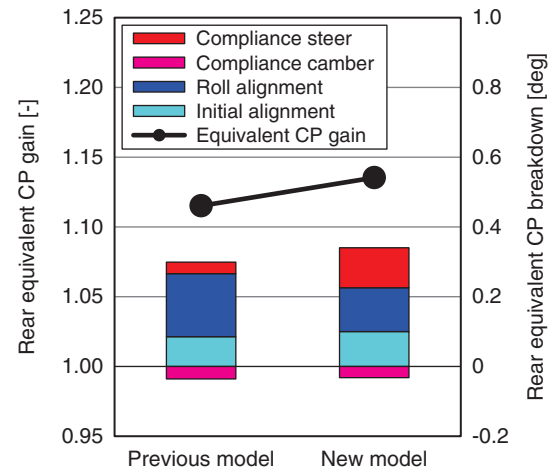


Fig. 3 Rear equivalent CP of new rear suspension

enhanced the vibration-damping characteristic to near the natural frequency of sprung mass.

In addition, the input to the body increases for grades equipped with a tire size of 17 inches or more, so a hydraulic trailing arm bushing was applied. This reduced body vibration enhanced road-holding ability when driving at high speeds, and enhanced damping of harshness (Fig. 5).

3.2. Front Suspension

A strut-type front suspension was used as in the previous model, but the suspension geometry was reviewed to achieve the target linear handling performance with little lag relative to steering input (Fig. 6).

The new model used an L-shaped lower arm to separate each bushing function.

Figure 7 compares the previous model and the new model.

In the previous model [Fig. 7(a)], it was not easy to reduce the spring rate of the compliance bushing, which has

a high contribution to ride comfort, because the lower arm bushing and the compliance bushing needed to have high lateral stiffness to support the load with respect to the lateral direction that is required for handling performance.

In contrast, the new model [Fig. 7(b)] uses an L-shaped lower arm, so the lateral force is borne by the high-stiffness

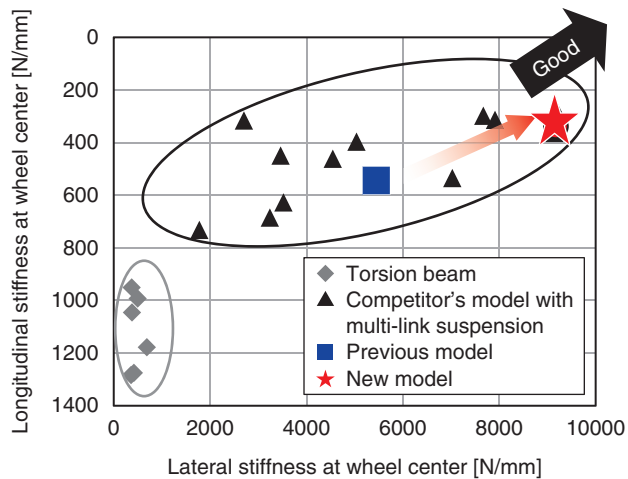


Fig. 4 Rear suspension lateral and longitudinal stiffness at wheel center

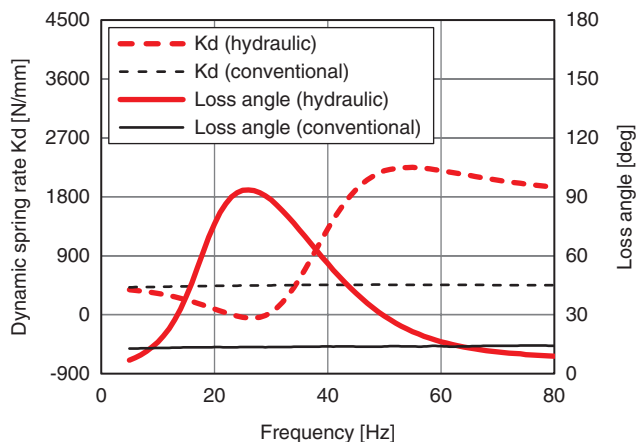


Fig. 5 Characteristics of rear hydraulic trailing arm bushing

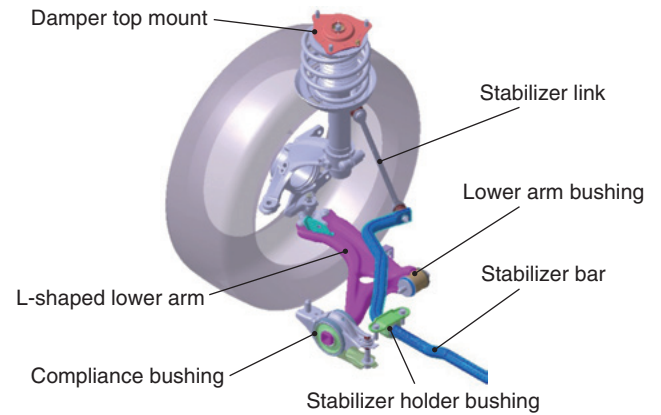


Fig. 6 Overview of front strut suspension

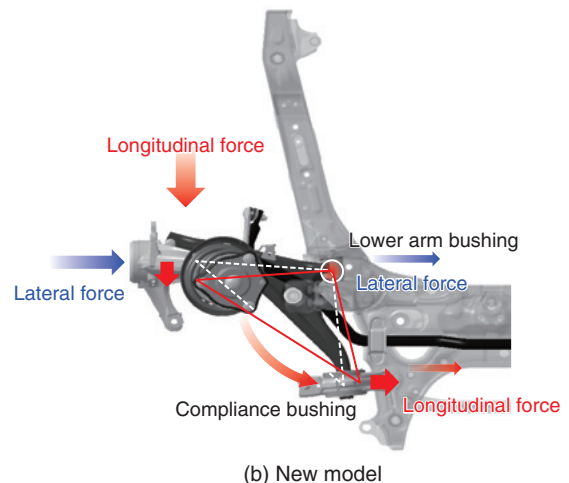
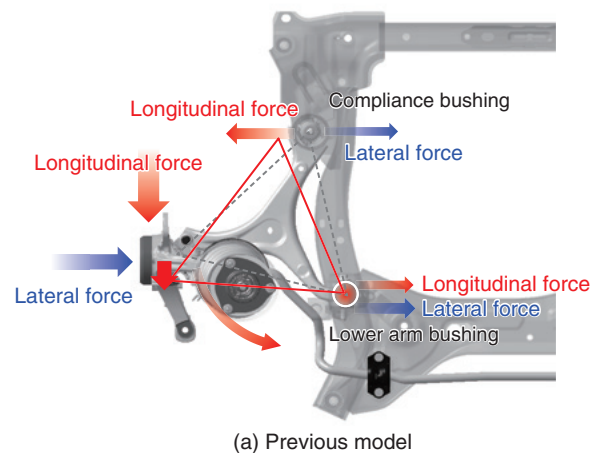


Fig. 7 Features of new front suspension

lower arm bushing and the longitudinal force is borne by the low-stiffness compliance bushing. This made it possible to simultaneously achieve handling performance and ride comfort.

Figure 8 shows the relationship between the lateral stiffness and longitudinal stiffness of the suspension at the wheel center position. The new suspension made it possible to achieve both a high lateral stiffness and low longitudinal stiffness that are the top level in the segment.

The axis of the compliance bushing was changed to fit its torsional direction to the suspension vertical stroke, and the torsional spring rate was reduced. Furthermore, use of a hydraulic type enhanced the damping characteristic in the wheel center longitudinal resonance range (10 to 20 Hz), which reduced steering shimmy and body vibration when driving at high speeds and enhanced harshness damping (Fig. 9).

In addition, the stabilizer spring rate was increased and the spring rate of the main spring was reduced compared

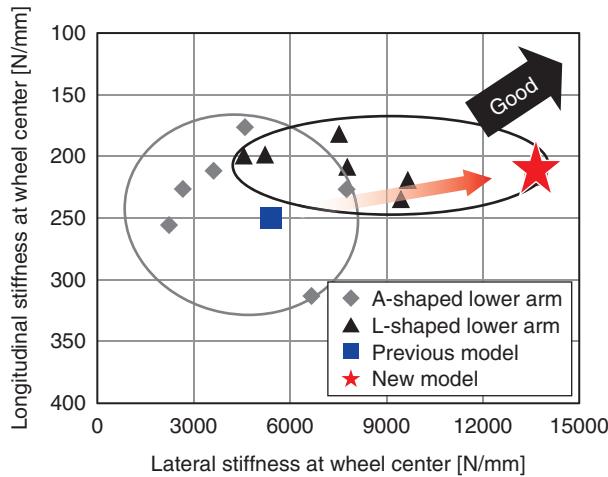


Fig. 8 Front suspension lateral and longitudinal stiffness at wheel center

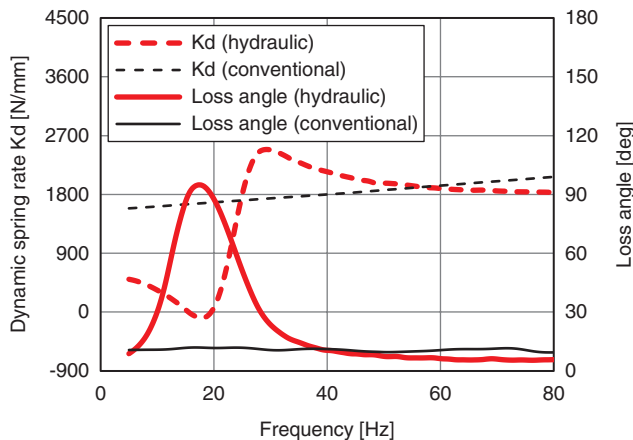


Fig. 9 Characteristics of front hydraulic compliance bushing

to the previous model. This lowered the bounce and pitch natural frequency while increasing the roll natural frequency of the body, which achieved non-coupling.

As a result, the roll response and road-holding ability were enhanced and the ground load fluctuation and roll amount were reduced.

The front roll center height was also reduced to half or less than that of the previous model to realize a cornering feel that holds the road with a low position.

3.3. Steering

A dual-pinion electric power steering system in which the motor assist mechanism is separate from the steering gear was used to achieve the goal of smooth and accurate steering feel from small steering maneuvers (Fig. 10).

Stays were added and the gearbox connection stiffness was increased in order to increase the lateral support stiffness that contributes to the steering feel (Fig. 11).

In addition, the steering shaft and steering hanger pipe diameters were increased, the column support stiffness was increased, and the vertical support stiffness was increased to reduce the vertical and lateral vibration on the steering wheel (Fig. 12).

Furthermore, the use of a variable steering gear ratio achieved both good turn-in ability at low speeds and stability at high speeds (Fig. 13).

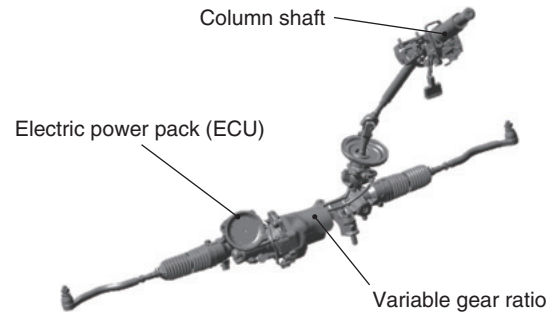


Fig. 10 Overview of dual pinion variable gear ratio electric power steering system

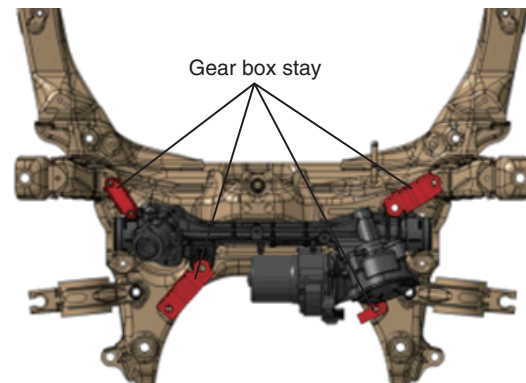


Fig. 11 Steering gear box stays

3.4. Body Stiffness

To achieve both linear handling performance with little lag relative to steering input and good ride comfort from the start of driving to long-distance cruising, body structures were reviewed for the new model to transfer the load input from the suspension to the body smoothly based on the concept of continuity of the structure and contour lines from the suspension input points to the cabin.

To achieve linear handling with little lag relative to steering input, the front and rear bulkheads were changed to ring structures to reinforce the torsional stiffness, centering on increased stiffness around the suspension connecting points that transmit the input from the road surface (Fig. 14).

To reduce vibration at the floor surface that contacts passengers' feet, a lattice frame was applied to the floor

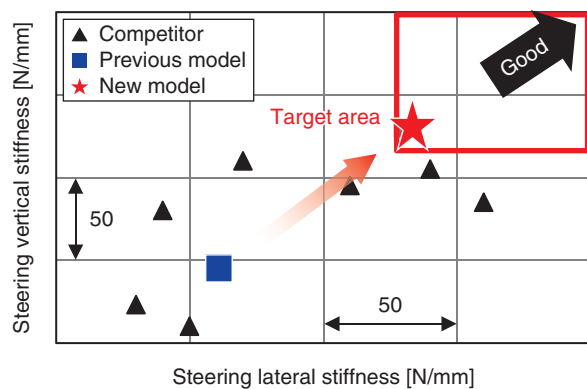


Fig. 12 Steering lateral and vertical stiffness

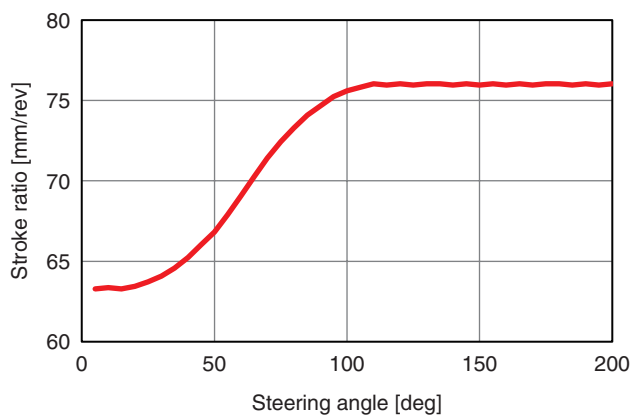


Fig. 13 Steering ratio characteristics

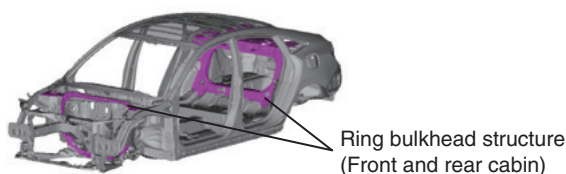


Fig. 14 Front and rear bulkhead structure

(Fig. 15) and the panel thickness was increased by approximately 60% compared to the previous model to increase the cabin bending stiffness and the floor stiffness.

In addition, the tunnel mode level was reduced by adding floor braces as shown in Fig. 16⁽³⁾.

Furthermore, locations with high sensitivity to input from the suspension were extracted using CAE, and countermeasures were taken to reduce stress concentration. However, regarding dynamic performance, there are some locations with high sensitivity for handling performance and ride comfort that are a challenge to express using CAE. The details for these locations, such as reviewing shapes and adding welding spots, were determined based on the test driver sensibilities (Fig. 17).

These countermeasures increased the static torsional stiffness by 25% and the floor bending stiffness by 57% compared to the previous model.

The results of bench tests using a 4-pillar shaker indicated that the transfer sensitivity from the front damper connecting point to the front seat connecting point could be reduced compared to the previous model as shown in Fig. 18.

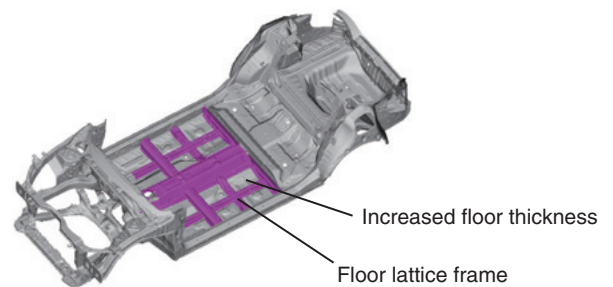


Fig. 15 Floor stiffness reinforcement items

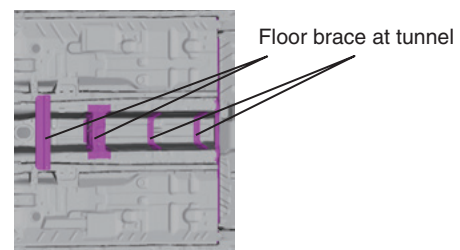


Fig. 16 Floor braces at tunnel

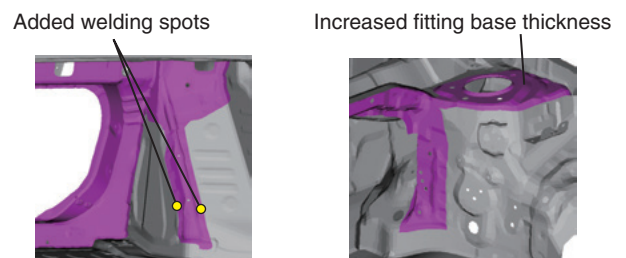


Fig. 17 Body stiffness enhancement points

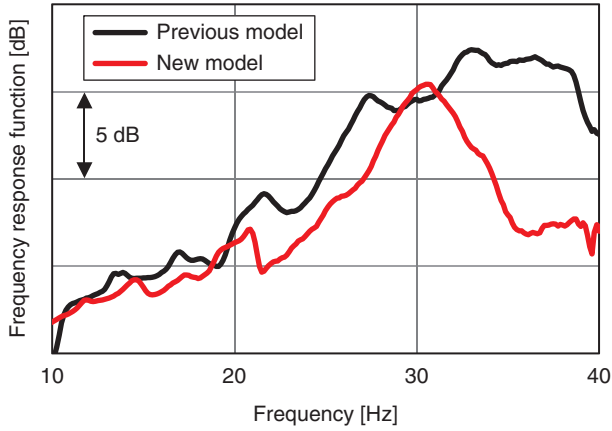


Fig. 18 Frequency response function of front input

4. Verification of Degree of Performance Achievement in Completed Vehicle

The degree of target performance achievement was confirmed by vehicle tests. Representative results are described below.

The yaw rate frequency was set as the performance for directional stability even during high speed driving, and was reduced compared to the previous model, achieving the top level in the segment (Fig. 19).

The yaw rate frequency means the average amplitude of the yaw rate fluctuation component produced when driving in a straight line.

A smooth and accurate steering feel from small steering maneuvers was realized by achieving steering torque characteristics with the target smooth and linear steering torque rise relative to the yaw rate during turn-in from 100 km/h (Fig. 20).

Flat ride in the low-frequency range (0.5 to 3 Hz) and flat ride in the mid-frequency range (3 to 8 Hz) were plotted to express good ride comfort from the start of driving to long-distance cruising, and the top level in the segment was

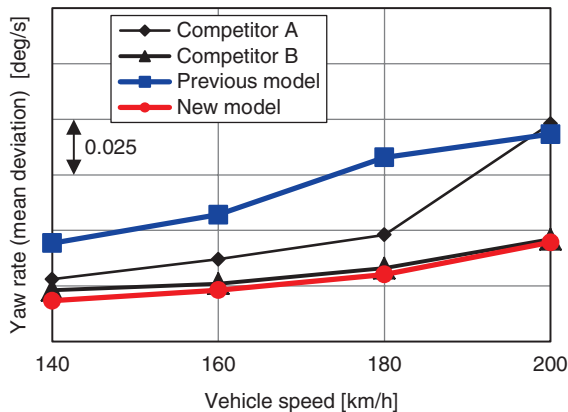


Fig. 19 High speed directional stability

achieved in terms of the in-house index (Fig. 21).

In addition, a harshness performance index was derived from the impact shock, shock sound, and damping, and the top level in the segment was achieved (Fig. 22).

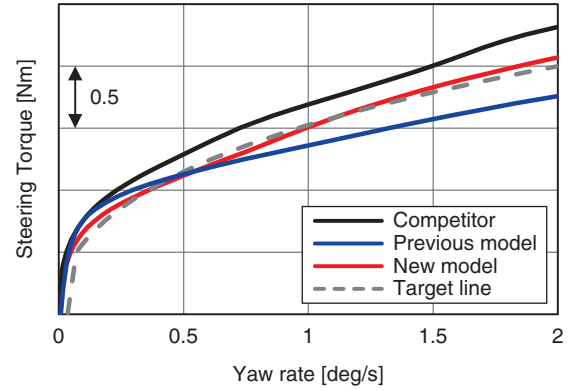


Fig. 20 Steering torque characteristics

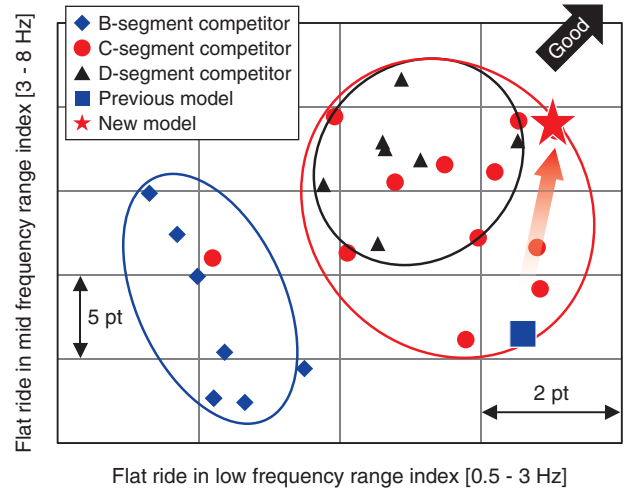


Fig. 21 Flat ride performance

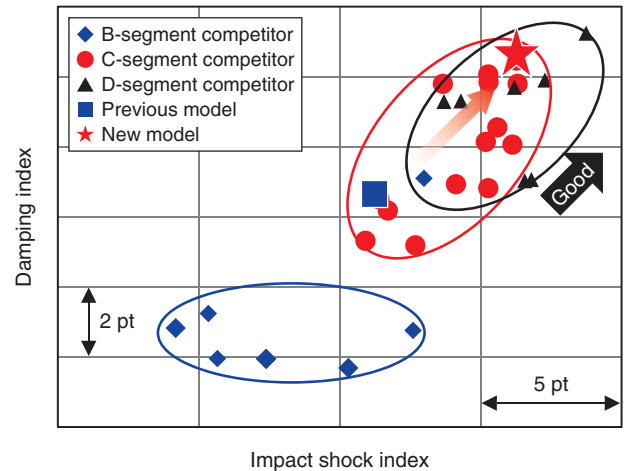


Fig. 22 Harshness performance

Linear handling performance with good response and little lag relative to steering input was expressed by the yaw rate gain and the frequency dependency of the yaw rate gain. As a result, high road-holding ability that is not dependent on the steering frequency was achieved (Fig. 23).

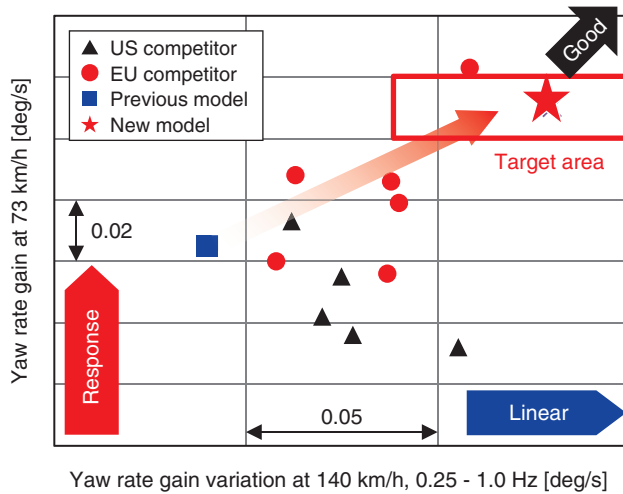


Fig. 23 Response and linearity

5. Conclusion

Technology was developed for the new CIVIC based on a new platform concept to achieve the top-level vehicle dynamic performance in the segment, including European vehicles. The achieved performance contents are as follows.

- (1) Use of a new rear suspension made it possible to separate the input force, achieve a high rear equivalent CP, and reduce the spring rate in the longitudinal and vertical directions, and together with use of a hydraulic compliance bushing enabled simultaneous achievement of both handling performance and ride comfort.
- (2) Road-holding cornering performance and flat ride performance were simultaneously achieved by reducing the friction and hysteresis of the suspension moving parts to reduce the force transferred from the road surface to the body, and by non-coupling of body roll, bounce and pitch.
- (3) Use of a dual-pinion variable gear ratio electric power steering system and increasing the system support stiffness enabled simultaneous achievement of both smooth and linear steering feel and good handling from small steering maneuvers to large steering angles and from low to high speeds.
- (4) Transfer sensitivity was reduced by the use of body bulkheads with a ring structure and by enhancing the floor stiffness, which realized both linear handling performance and good ride comfort from the start of driving to long-distance cruising.

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Aerodynamic Development for High-speed Stability in New CIVIC

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ABSTRACT

Aerodynamic design for the new model CIVIC sought to reduce aerodynamic drag and help ensure vehicle stability at high speeds. Equivalent lift was employed as an evaluation index expressing the effect of aerodynamic forces on vehicle dynamic performance, and the importance of reducing rear equivalent lift in helping to ensure stability was verified in driving test measurements. In order to reconcile the reduction of rear equivalent lift with low drag, the control of longitudinal vortices at the rear of the vehicle was set as a goal, with measures adopted in both the platform and the exterior. The realization of increased underfloor flow velocity through the use of underfloor devices and the aerodynamic engineering of the exterior side-view profile and the area around the rear pillars reduced rear equivalent lift by approximately 50% against the previous model. This has limited changes in rear tire performance with changes in vehicle speed, and helped to ensure vehicle stability in the high-speed range.

1. Introduction

It is well known that aerodynamic performance fulfills important functions for racing vehicles. At the same time, there is a tendency to believe that because the effect of aerodynamic performance on mass-production vehicles is much lower than on racing vehicles, that aerodynamics does not have a particularly strong effect on vehicle stability in these cases. In fact, however, the aerodynamic forces acting on mass-production vehicles, including the effect of cross winds, are not insignificant enough to be ignored.

In the case of racing vehicles, aerodynamic devices are attached to the vehicles to generate downforce and increase the vertical load on the tires, thus increasing the lateral force, in order to increase cornering speed.

However, in mass-production vehicles, it is generally the case that shapes that generate lift are employed in order to make it possible to realize functional demands in the normal use range and to satisfy styling demands. Because this results in a decline in the vertical load and a consequent decline in the lateral force on the tires when vehicle speed increases, it tends to result in reduced stability.

In particular in the area of vehicle stability at high speed, differing aerodynamic specifications will produce differences able to be felt by the driver; aerodynamic force control represents an important element in vehicle

dynamics. Many questions remain to be answered in the field of high-speed vehicle stability in response to aerodynamics in the high-speed range. The research focus is expanding from steady-state flows to include non-steady-state flows, and a good deal of research is being conducted on flow fields resulting from disturbances produced by natural wind⁽¹⁾.

The necessity to realize low drag (low CD) has increased within the development process for mass-production vehicles in order to satisfy today's intense demand for competitiveness in the area of fuel efficiency, and it is not easy to reconcile this with high-speed vehicle stability. In addition, the inputs resulting from aerodynamic forces represent an element in the system characteristics (including the tires and suspension), making it challenging to set separate target values for aerodynamics.

The development of the new model CIVIC was therefore conducted with consideration for reconciling high-speed vehicle stability with the realization of a low CD in order to achieve a high level of fuel efficiency from the initial stages of the platform development process. This paper will discuss the aerodynamic methods used to reconcile target high-speed vehicle stability with low CD in the development of the new model CIVIC from the perspectives of platform development and exterior development.

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2. Aerodynamic Vehicle Stability Goals for New Model CIVIC

The dynamic performance concept for the new model CIVIC was the realization of outstandingly agile linear performance and a feeling of stability. Aerodynamic force control in the high-speed range would be an important element in balancing agile handling in the low-speed range with a feeling of stability in the high-speed range.

In order to realize a feeling of stability at high speeds, it would be necessary to achieve improvements in basic indices of dynamic performance, for example steady-state yaw rate gain and lateral acceleration phase delay at high speeds.

The concept of equivalent CL , which encompasses the effects of lift and side force, was introduced as an evaluation index that links these indices of dynamic performance and steady-state aerodynamic forces^{(2), (3)}.

The goal during the development of the new model CIVIC was to clarify the contribution of front and rear equivalent CL to dynamic performance, to substantiate the effectiveness of the index on vehicle dynamics through actual vehicle measurements, and to balance the target equivalent CL with low CD .

3. Contribution of Equivalent CL to Dynamic Performance

3.1. Formularization of High-speed Vehicle Stability and Aerodynamic Forces

Aerodynamic forces act on the aerodynamic center of the vehicle body as drag, lift, and side force. These forces are supported by the contact patches of the front and rear wheels, via the suspension. Understanding the contribution to the tires is an important factor when considering the effect of aerodynamic forces on vehicle stability. Cornering power (CP) is the lateral force generated on the tire per unit slip angle, and is employed as an index of basic tire performance. Figure 1 shows the relationship between CP and the vertical load on the tires. Here, K is tire CP, and

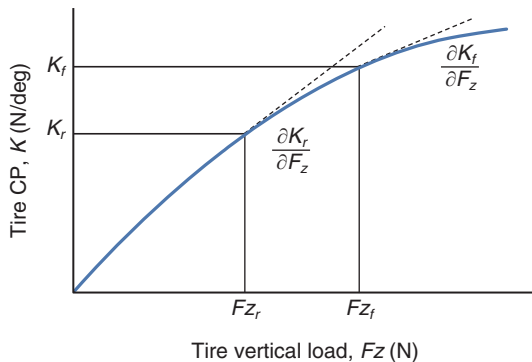


Fig. 1 Tire CP and vertical load for FF car

F_z is the vertical load on the tires. The subscripts f and r respectively refer to front and rear. Tire CP displays a nonlinear characteristic in relation to load. In an FF vehicle like the CIVIC, the allocation of load is further towards the front than the rear, and because of this the vertical load on the rear tires is lower than that on the front tires. Rear CP is therefore prone to change in response to changes in vertical load.

Assuming steady-state cornering, the effect of aerodynamic forces including side force can be understood as the fluctuation in the vertical load that produces change in the lateral force on the tires. Eq. (1) shows the change in tire CP due to aerodynamic forces. Here, ρ is air density, V is vehicle speed, A is the frontal projected area of the vehicle, CL is the lift coefficient, CS is the side force coefficient, and CYM is the yaw moment coefficient.

$$\Delta K_i = -\frac{1}{2} \cdot \frac{1}{2} \rho V^2 A \cdot CL_i^* \cdot \frac{\partial K_i}{\partial F_z} \quad (i=f \text{ or } r) \quad (1)$$

In this equation, $*$ represents equivalent CL , a hypothetical lift coefficient that expresses the effect on the tires, including CS and CYM .

$$\begin{aligned} CL_f^* &= CL_f + (\Delta CL)_f^{CS, CYM} \\ &\cong CL_f - \frac{\partial CS_f}{\partial \beta} \frac{l_f K_f}{l_r K_r} \bigg/ \frac{\partial K_f}{\partial F_z} \end{aligned} \quad (2)$$

$$\begin{aligned} CL_r^* &= CL_r + (\Delta CL)_r^{CS, CYM} \\ &\cong CL_r - \frac{\partial CS_r}{\partial \beta} \bigg/ \frac{\partial K_r}{\partial F_z} \end{aligned} \quad (3)$$

$$CS_f = \frac{CS}{2} + CYM, \quad CS_r = \frac{CS}{2} - CYM \quad (4)$$

Here, β represents the slip angle of the vehicle body. CS_f and CS_r respectively represent the front and rear side force coefficient. These coefficients express the allocation of side force to the front and rear tires by the yaw moment.

The contribution of CS and CYM to equivalent CL is around at least 10%. The improved correlation between the magnitude of equivalent CL and sensory evaluations has been verified by study of CS and CYM based on past evaluations offered by drivers in tests using actual vehicles⁽³⁾.

The effect of equivalent CL on vehicle dynamics will be considered using the stability factor, an indicator of vehicle stability. The stability factor, K_s , is defined by Eq. (5).

$$K_s = -\frac{m}{2l^2} \frac{l_f K_f - l_r K_r}{K_f K_r} \quad (5)$$

Here, m is the vehicle mass, l is the wheelbase, and l_f and l_r are the respective distances from the center of gravity

to the front and rear axles.

Generally, the higher the stability factor, the more the vehicle displays an understeer characteristic, and the greater the stability of the vehicle. Figure 2 shows an example of the relationship between tire CP and the stability factor in relation to vehicle speed when equivalent CL is acting uniformly on the front and rear tires. Taking into consideration the load-dependency of tire CP discussed above, in an FF vehicle rear tire CP tends to be particularly affected by equivalent CL_r . Therefore, despite the fact that the same equivalent CL is acting on the front and rear tires, i.e. the reduction of vertical load is the same for the front and rear tires, the amount of change in tire CP will differ for the front and rear tires. Because of this, the stability factor declines with increasing vehicle speed, and the vehicle tends to display an increasing tendency towards oversteer as vehicle speed increases. This can also be considered as aerodynamic forces acting to make tire CP dependent on velocity. Reducing the effect of this characteristic was the target for high-speed vehicle stability in this development.

3.2. Equivalent CL and Handling Indices

As in the case of the stability factor, the basic characteristics of the vehicle dynamic system change with changes in front and rear tire CP produced by equivalent CL . These changes appear as changes in handling indices, which express the dynamic maneuverability and stability of the vehicle in relation to yaw. Standard handling indices include steady-state yaw rate gain, yawing resonance frequency, and handling capacity. These are used, respectively, as indices to express the magnitude of yaw angle velocity, responsiveness, and convergence of transient response in relation to steering, and can be expressed using front and rear tire $CP^{(4)}$.

When a vehicle is driving at high speeds, it is desirable

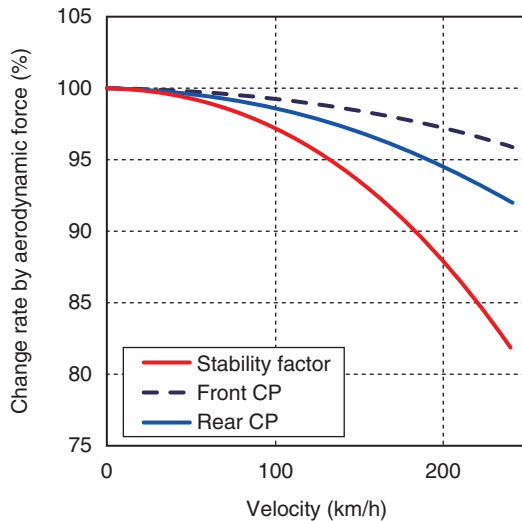


Fig. 2 Velocity dependency of stability factor and tire CP in relation to aerodynamic forces

that its yaw dynamics should be constrained in relation to slight steering input, that the vehicle should respond instantaneously to corrective steering, and that its dynamic behavior should converge rapidly following corrective steering. It is desirable therefore that the yaw rate velocity should be low, that responsiveness should be high, and that the convergence of transient response should be rapid.

Figure 3 shows isolines of the handling indices in relation to equivalent CL (which provides a standard) and desirable directions for equivalent CL during high-speed driving. When the realization of a balance between stability and a good transient response characteristic is set as the target, the desirable direction for equivalent CL_f differs for each of the indices, and cannot be uniquely determined. By contrast, a reduction in CL_r is the desirable direction in the case of all the indices. This is because rear tire CP increases with a reduction in equivalent CL_r , indicating the importance of maintaining high rear CP for vehicle dynamics. An increase in the rear tire CP in the high-speed range alone would not be manifested as a change in the suspension characteristic. It was therefore considered that reducing equivalent CL_r would represent an effective means of boosting straight-line stability in the high-speed range.

3.3. Actual Driving Tests Using Different Equivalent CL_r

Actual driving tests were conducted in order to study the effectiveness of reducing equivalent CL_r in increasing high-speed vehicle stability.

Aerodynamic devices were fitted to vehicles in order to produce two specifications with differing equivalent CL_r (Aero Configuration A and Aero Configuration B). Measurements were conducted in order to determine how these different aerodynamic configurations would change the steering response of an actual vehicle. Figure 4 shows wind tunnel measurements of equivalent CL and the change in tire CP at 180 km/h when switching from Aero Configuration A to Aero Configuration B. Equivalent CL_r is lower for Aero Configuration B than for Aero Configuration

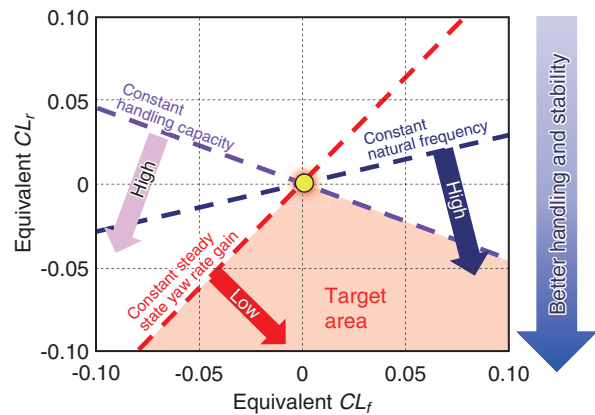


Fig. 3 Preferable directions for equivalent CL_f and CL_r in terms of handling indices

A [Fig. 4(a)]. Because of this, rear tire CP at 180 km/h is more than 3% higher for Aero Configuration B than for Aero Configuration A. By contrast, the change in front tire CP is less than 1% [Fig. 4(b)]. It may therefore be considered that the difference in measurement results with the change in the aerodynamic specifications originates in change in rear tire CP, in other words change in equivalent CL_r .

Figure 5 shows measurement results for yaw rate gain and lateral acceleration phase delay for the two aerodynamic configurations. These measurement results, taken in an actual vehicle, show that in the high-speed range, the change from Aero Configuration A to Aero Configuration B reduces steady-state yaw rate gain by approximately 10% and lateral acceleration phase delay by approximately 14%. This meant that the hypothesis that reducing equivalent CL_r would be effective in maintaining high-speed vehicle stability was supported by the results of actual vehicle measurements. Equivalent CL_r was therefore focused on in the development of the new model CIVIC.

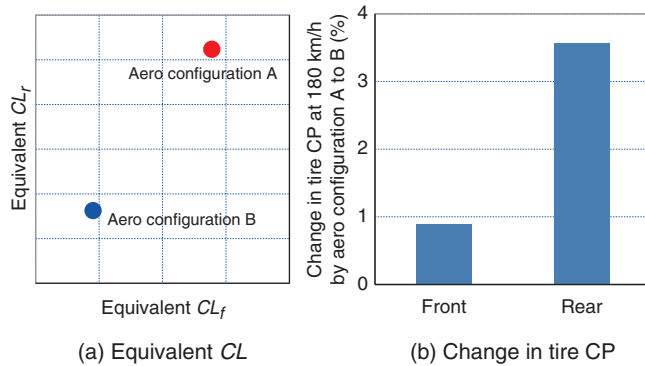


Fig. 4 Difference in equivalent CL between two aerodynamic configurations and its effect on tire CP

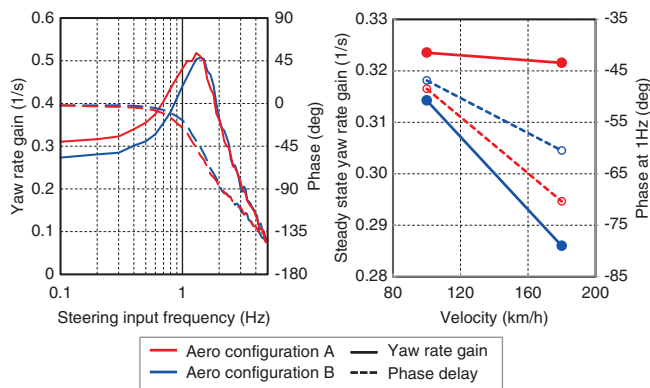


Fig. 5 Results of steering response tests using two aerodynamic configurations

4. Balancing Low Equivalent CL_r and Low CD

Based on Eq. (3), increasing CS_r/β , reducing CL_r , and controlling the degree of reduction in rear tire CP when the tire load was reduced could be considered as methods of reducing equivalent CL_r . It was therefore necessary to increase CS_r/β and reduce CL_r by modifying the vehicle's aerodynamic characteristics.

The greatest advantage of the introduction of the concept of equivalent CL was that multiple objective functions (front and rear CL , CS , and CYM) could be aggregated into front and rear equivalent CL . At the stage of optimization of the actual shape of the vehicle, there would thus be no necessity to consider a breakdown of these functions; it would be sufficient to focus on reducing equivalent CL .

At the same time, fixing the concept for the flow field in order to realize the various aerodynamic characteristics set as aims for the development is an important factor in aerodynamic development. This section will introduce the aerodynamic concepts and approaches employed to realize an increase in CS_r/β and a reduction in CL_r .

4.1. Increased CS_r/β

Increased CS_r/β would mean reduction of the aerodynamic yawing moment, and could be realized by increasing CS and reducing CYM . However, because increasing CS might reduce stability in cross winds, it would be desirable to achieve a reduction in CYM without increasing CS .

Figure 6 shows CFD results for changes in CS and CYM produced by different exterior changes and underfloor devices. Underfloor devices produce only small changes in CS and CYM , of 5% or less. It was therefore considered that the effect of the platform and underfloor devices on CS and CYM could be ignored.

In the case of exterior changes, changes to the basic

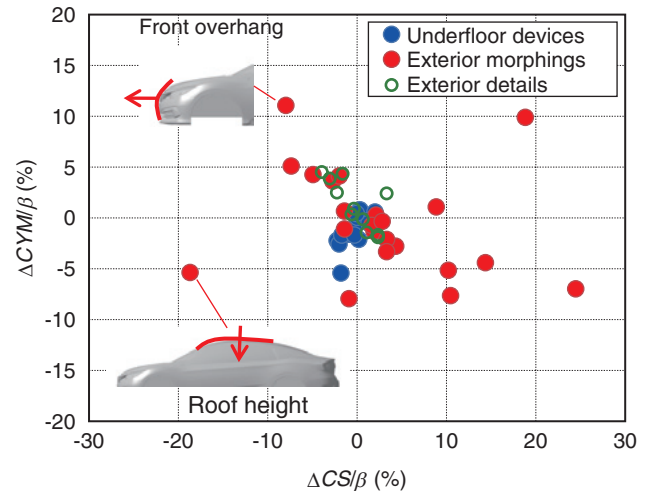


Fig. 6 CFD prediction of changes in CS and CYM by various aerodynamic features of prototype car

dimensions that determine the framework, including overhang and total height, i.e. major changes to the basic package, produce major changes in CS and CYM . However differences in the shape of the vehicle introduced at the stage of optimization of the detailed aerodynamic shape produce only small changes.

4.2. Reduced CL_r

4.2.1. Reduction of CL_r in platform development

Figure 7 shows the aerodynamic concept for the new model CIVIC, formulated in order to balance reduced equivalent CL_r and low CD . This concept involved accelerating the underfloor flow in order to create an upward flow (upwash) and controlling the downward flow (downwash) on the vehicle exterior in order to control longitudinal vortices at the rear of the vehicle. Control of rear longitudinal vortices was expected to correct the growth of uneven left-right longitudinal vortices when a yaw angle is produced, in addition to improving the aerodynamic yaw characteristic.

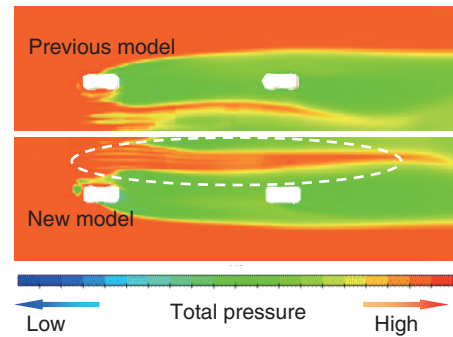
From the stage of platform development, the area for application of aerodynamic undercovers was expanded across the entire vehicle, and the parts on which aerodynamic covers would not be used, including the rear suspension, the heat baffles, and the silencer, were comprehensively smoothed in order to accelerate the underfloor flow. Figure 8 shows the underfloor shape of the new model CIVIC.

Figure 9(a) shows differences in the total pressure distribution under the floor between the previous and the new models, and 9(b) shows a comparative diagram of the velocity fields around the underfloor sections of the two models. The smoothing of the parts from the engine undercover and floor cover to the rear bumper has increased the ratio of the high-pressure areas in the flow that travels from between the front tires and exits at the rear of the vehicle in the new model against the previous model [Fig. 9(a)]. In other words, flow velocity does not decline before the rear of the vehicle, indicating that it has been possible

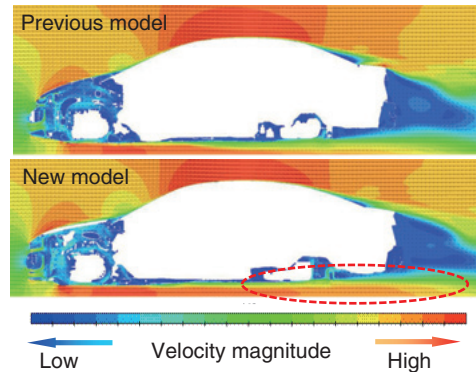
to accelerate the underfloor flow [Fig. 9(b)]. These results demonstrate that reduced CD has been reconciled with reduced CL_r under the floor without antagonistic results.



Fig. 8 Aerodynamic treatment on new CIVIC underfloor



(a) Total pressure distribution at $Z=100$ mm



(b) Velocity distribution throughout underfloor

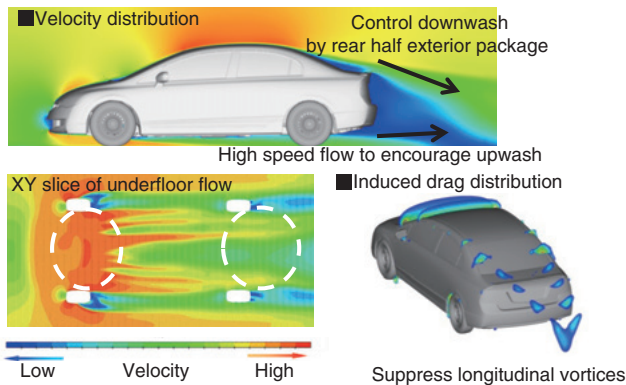


Fig. 7 Aerodynamic concept developed to balance low equivalent CL_r with low CD

Fig. 9 Difference in flow under floor between new and previous model platform

4.2.2. Exterior aerodynamic concept

Compared to the underfloor flow, which is affected by factors including the tires, differences in level between parts, and the hot air emitted by the engine housing, the flow around the exterior of the vehicle encounters few shapes that would produce loss, and flow velocity is high. The flow around the exterior therefore has the dominant effect on the flow field for the vehicle as a whole.

Three measures were applied in the exterior development process in order to realize reductions in both CL_r and CD .

The first of these was to pinch up the front edge of the hood, emphasizing the edge only to the degree that separation would not be generated, and inducing suction in this section.

The second approach, in line with the pinching up of the front edge of the hood, was to move the roof peak forward, in order to move the area of suction on the roof forward.

The third measure was to reduce the depth of the angle of the side view by shifting the base of the C-pillar to the rear in order to control downwash at the rear of the vehicle.

The first two approaches will be considered first. Pinching up the front edge of the hood and moving the roof peak forwards in order to shift the area of suction generated in the upper half of the vehicle towards the front and produce a pitch-up moment resulted in a comparative reduction in CL_r .

Because it is positioned at the front extremity of the vehicle, changes in the shape of the front edge of the hood in particular promote change in the pressure peak and longitudinal shift in the center of pressure. It is therefore one area that will tend to produce change in the front-rear balance of CL .

Figure 10 shows an example of change in the pressure coefficient and change in front-rear CL when the front edge of the hood is pinched up. Changing the shape of the front edge of the hood to the shape shown by the white line changes the pressure distribution around the hood, and the area of low pressure shifts towards the front by comparison with the original shape. Thus, although there has been no change in CL itself, the center of pressure, shown by the red arrow, has shifted to the position shown by the blue arrow, and the pitch-up moment acting on the center of the vehicle has increased. This shifts the front-rear distribution of CL towards the front, with the result that the pinching up of the

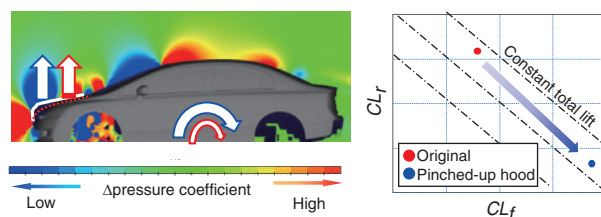


Fig. 10 Pressure difference and CL balance shift by pinching up front end of hood

front edge of the hood reduces CL_r .

While the degree will differ, moving the roof peak forwards can also be expected to have a similar effect.

Figure 11 shows the difference between the shape of the new and the previous models. Figure 12 shows the difference in the pressure coefficient. The forward shift in the pressure distribution shown in Fig. 10 can be seen in the change in pressure shown by the solid line in Fig. 12, and it was therefore considered that CL_r had been reduced.

In addition to this, moving the roof peak forwards reduced the change in the rate of curvature of the roof from the roof peak to the rear window, facilitating pressure recovery. This has controlled the generation of boundary layers and the occurrence of separation due to strong adverse pressure gradients at the rear edge of the roof, as shown by the broken lines in Fig. 12. The maintenance of a high-velocity flow up to the rear of the trunk is considered to have increased control of downwash (Fig. 13).

The shifting of the C-pillar to the rear will now be



Fig. 11 Side view comparison of new and previous models

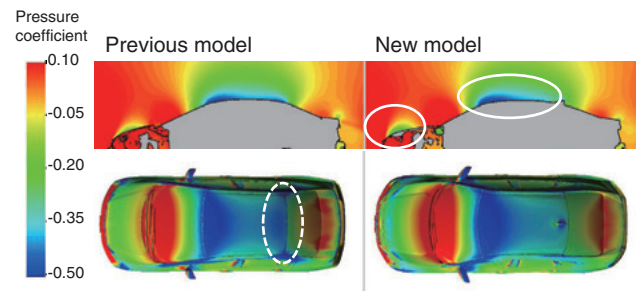


Fig. 12 Difference in pressure coefficient distribution by hood and roof profiles

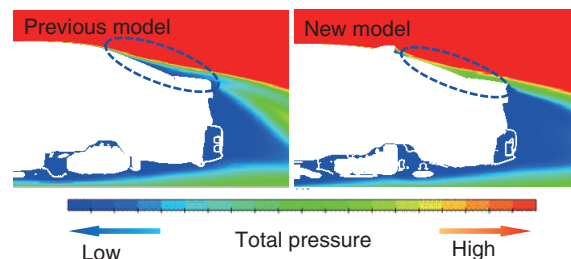


Fig. 13 Change in flow separation around rear window

considered. Figure 14 shows the difference in the shape around the C-pillar between the new and previous models.

Figure 15 shows the pressure distribution around the C-pillar for the new and previous models.

As Fig. 15(a) shows, in the previous model not only was suction strong around the C-pillar, but, as noted above, it was also strong at the rear edge of the roof, producing variations in pressure distribution around these areas. By contrast, in the new model, suction is weak around both the C-pillar and the rear edge of the roof, and there is a minimal pressure differential with the surrounding area. This indicates that shifting the base of the C-pillar to the rear made the surface pressure distribution around the C-pillar uniform.

Figure 15(b) shows the pressure distribution on the surface of the vehicle for a cross-section at the rear edge

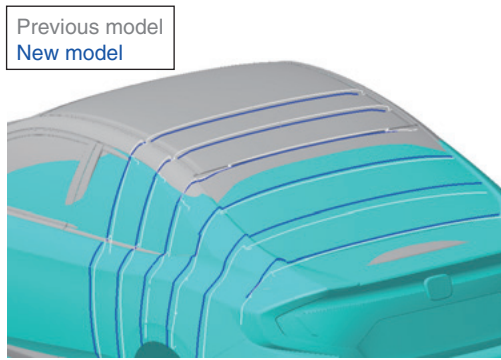
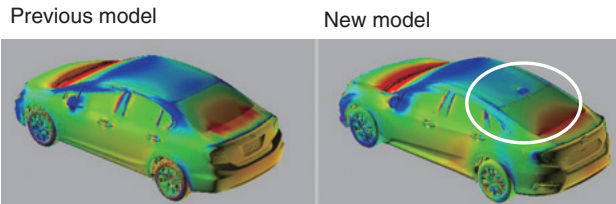
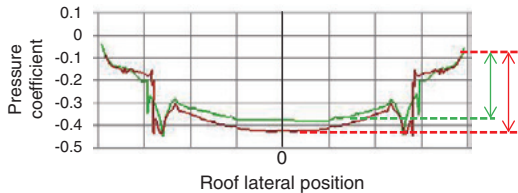


Fig. 14 Shallower inclination of C-pillar on new model



(a) Uniformity of pressure coefficient in rear end of new model



(b) Pressure difference between C-pillar and roof

Fig. 15 Difference in pressure coefficient around C-pillar

of the roof. In the new model, there is a minimal pressure difference between the side face of the pillar and the rear edge of the roof. If this pressure difference were high, it would represent the point of origin of a flow from the side face of the vehicle to the rear edge of the roof, and the downwash would wrap around the C-pillar and form a longitudinal vortex. By creating a uniform pressure distribution around the pillar, the C-pillar shape in the new model controls the rolling-in from the side face of the vehicle to the center that might wrap around the C-pillar and form the point of origin for longitudinal vortices at the rear of the vehicle. The velocity component from the side face of the C-pillar towards the vehicle center has been reduced in the new model (Fig. 16).

The shape from the C-pillar to the trunk in particular has become a design icon of the new model CIVIC, presenting the shape of a sedan but offering the image of a coupe depending on the angle from which it is viewed. In this section of the vehicle, design demands and aerodynamic performance reached an even more sophisticated level of balance.

These modifications of the form of the vehicle and the realization of increased underfloor flow velocity have controlled downwash and the rolling-in from the side of the vehicle, suppressed the formation of longitudinal vortices at the rear of the vehicle, and balanced reduced CD with reduced CL_r .

Figure 17 shows the difference in longitudinal vortices at the rear of the vehicle between the new and previous models. Figure 17(a) shows the distribution of vorticity in the direction of the main flow at the rear of the vehicles, demonstrating that the vortices formed at the rear of the new model have decreased. As a result, the distribution of induced drag, which expresses the drag component produced by longitudinal vortices, has also been reduced [Fig. 17(b)].

The measures discussed above have made it possible to realize a reduction of approximately 50% in equivalent CL_r in the new model CIVIC against the previous model. Figure 18 shows a comparison of equivalent CL_r in the new and previous models and a breakdown of the components of equivalent CL_r . The CL_r and CS_r terms in this breakdown are the first and second terms respectively on the right-hand side of Eq. (3). Pursuing development with a focus on

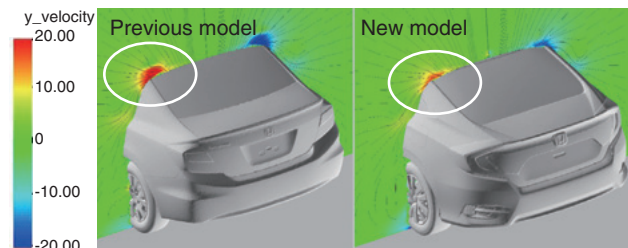


Fig. 16 Y-velocity difference around C-pillar

reducing equivalent CL_r , not only made it possible to reduce CL_r , but also to increase CS_r . This has reduced the decline in rear tire CP due to speed by approximately 50% against the previous model (Fig. 19).

CD has also been reduced by 14% against the previous model. Aerodynamic specifications were developed that provided the potential for the realization of a balance between the target, reduced equivalent CL_r , and low drag.

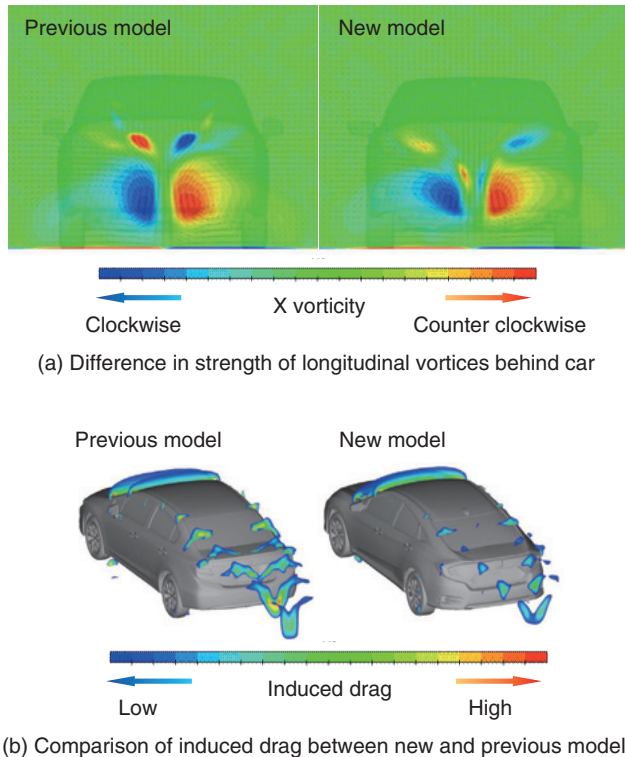


Fig. 17 Suppression of growth of longitudinal vortices for new CIVIC

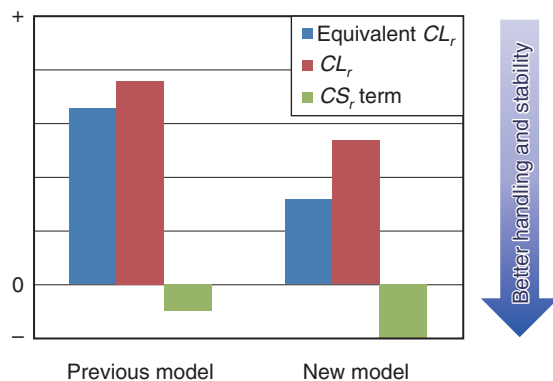


Fig. 18 Comparison of equivalent CL_r between new and previous models

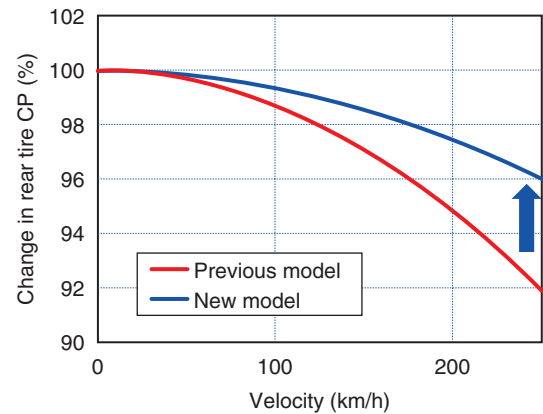


Fig. 19 Comparison of velocity dependency of rear tire CP between new and previous models

5. Conclusion

The development process seeking to realize high-speed stability in the new model CIVIC can be summarized as follows:

- (1) Equivalent CL was employed as an index during the development process, and development focused on reducing equivalent CL_r in order to improve high-speed handling indices.
- (2) The hypothesis regarding the effect of equivalent CL_r on straight-line stability at high speeds was supported by quantitative evidence obtained from measurements conducted in actual driving tests.
- (3) The control of the formation of longitudinal vortices at the rear of the vehicle was addressed as a method of balancing a reduction in equivalent CL_r and a reduction in CD . Aerodynamic specifications capable of mass production were realized in both the new platform and the vehicle exterior.

In future, it will be necessary not only to further elucidate the mechanisms by which aerodynamic forces, including non-steady-state forces, affect vehicle behavior, but also to propose solutions via forms able to be mass-produced. It will also be essential to reconcile any measures with the realization of low drag. The authors therefore wish to comprehensively enhance aerodynamic design capability, including approaches to the feasibility of designs.

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Masayuki MINAGAWA

Advanced Technology for Interior Vibration Reduction for New CIVIC

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ABSTRACT

Steering vibration while idling and floor and accelerator pedal vibration while accelerating were reduced for the new CIVIC. Steering vibration while idling was reduced by positioning the engine mounts, which are the main transfer path, in locations with a small engine vibration amplitude. This reduced the engine vibration transferred to the body without lowering the spring constants of the engine mounts, and provided power train support stiffness while at the same time reducing the vibration transfer amount. Floor vibration while accelerating was reduced by positioning the exhaust mounts in locations with a small exhaust vibration amplitude to reduce exhaust vibration transfer, and by increasing the floor panel thickness and adding cross members to increase the floor stiffness and reduce the floor vibration sensitivity. Accelerator pedal vibration while accelerating was reduced by offsetting the position of the fuel pipe bracket connected to the dashboard lower where the accelerator pedal module mounting point is located.

1. Introduction

Customer interest in NVH performance is increasing. In addition to interior quietness, there are also high expectations for reduction of steering, floor, and accelerator pedal vibration, which are locations that passengers contact.

Vibration while idling and accelerating is a phenomenon that occurs when engine vibration due to engine torque fluctuation and inertia force is transferred to the body. Effective countermeasures include reducing the spring constants of bushings to reduce the vibration transfer amount, and body countermeasures involving weight increase to reduce vibration sensitivity. However, these countermeasures present issues for also achieving good handling, ride comfort, and fuel economy performance.

Therefore, interior vibration while idling and accelerating was reduced using methods that make it possible to also achieve good handling, ride comfort, and fuel economy performance.

2. Interior Vibration Reduction Goals

- (1) Reduction of steering vibration while idling.
- (2) Reduction of floor vibration and accelerator pedal vibration while accelerating.

This paper describes the technology applied to achieve the above goals.

3. Technology for Reduction of Steering Vibration while Idling

3.1. Mechanism

Steering vibration while idling occurs when engine vibration produced by engine torque fluctuation is transferred to the body via the main transfer path of the engine mounts that elastically support the power train.

This vibration transfer amount can be quantified using the engine vibration at the engine mount points, the spring constants of the engine mounts, and the vibration sensitivity from the engine mount connection points to the steering (hereafter, steering vibration sensitivity), which is a body side vibration characteristic.

Figure 1 shows the engine mount system configuration for the 1.5 L VTEC TURBO engine. The power train support load is shared by the two mounts (hereafter, main mounts) of the engine side mount located on the upper right side of the cylinder block and the transmission mount located on the upper left side of the transmission. The drive reaction force is supported by a torque rod that connects the lower part of the oil pan and the subframe. This torque-rod-type engine mount system employs a structure based on technology applied to the 2006 model year CIVIC⁽¹⁾.

3.2. Issues and Countermeasures

The idling engine speed for an inline 4-cylinder engine

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is approximately 600 to 900 rpm, and the frequency band of the 2nd-order component that is the main issue is approximately 20 to 30 Hz.

The vibration transfer amount can be reduced by reducing the spring constants of the engine mounts. However, this reduces the power train support stiffness and increases the vibration amount, so also achieving good handling and reducing the jerking feeling while accelerating is an issue.

In addition, a method of using the Supplemental Restraint System (SRS) module of the steering system as a dynamic damper⁽²⁾ has been developed as a method of directly reducing steering vibration. However, the resonance frequency of the steering system is approximately 40 Hz, which is higher than the idling frequency band, so reduction of the 2nd-order component of the steering vibration while idling has little effect. Furthermore, enhancement of vibration sensitivity by changes to the body and steering structure that do not involve weight increase are quite challenging from a technical perspective.

Investigation therefore focused on the power train vibration mode while idling, and investigated a method of reducing engine vibration at the main mount points without reducing the spring constants of the main mounts, by positioning the main mounts in locations with a small vibration amplitude.

3.3. Investigation of Optimal Main Mount Layout

The power train is forcibly excited by torque fluctuation while idling, and vibrates in the rotation direction with the torque roll axis⁽³⁾ (the axis determined by the inertia main axis and the crankshaft angle) shown in Fig. 2 as the rotation axis (hereafter, power train rolling vibration). The power train rolling vibration amplitude decreases as the power train rolling vibration approaches the torque roll axis. The engine vibration amplitude at the main mount points while idling can be reduced by positioning the main mounts in locations with a small power train rolling vibration amplitude.

Figure 3 shows the steering vibration sensitivity at the engine side-mount point with respect to vehicle longitudinal direction and vertical direction input. This shows that the vibration sensitivity in the vehicle longitudinal direction is approximately 10 dB lower than that in the vehicle vertical direction. Therefore, positioning the main mounts

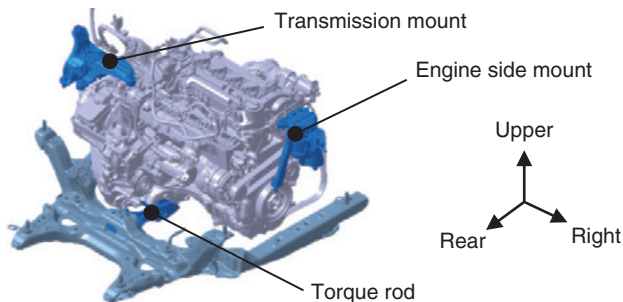


Fig. 1 Torque rod type engine mount system

in locations that minimize engine vibration in the vehicle vertical direction, even if vibration in the body longitudinal direction increases, was investigated.

3.4. Investigation Results of Optimal Main Mount Layout

The structure provides the two main mounts on the side frame at both ends in the crankshaft direction of the power train, leaving little freedom of movement in the vehicle lateral direction. In addition, the main mounts need sufficient support stiffness with respect to the drive reaction force, so lowering the position in the vehicle vertical direction reduces the support stiffness, which makes simultaneously reducing the jerking feeling while accelerating an issue. CAE was therefore used to investigate the optimal position in the vehicle longitudinal direction that minimizes the engine vibration in the vehicle vertical direction at the main mount points while idling.

Figure 4 shows the engine side-mount position and corresponding change in engine side-mount vibration in the vehicle vertical direction for the three cases when the engine side-mount position is located on the torque roll axis (zero longitudinal offset) (Case-1), moved 40 mm toward the front of the vehicle (Case-2), and moved 40 mm toward the rear of the vehicle (Case-3). This shows that the engine vibration in the vehicle vertical direction at the mount

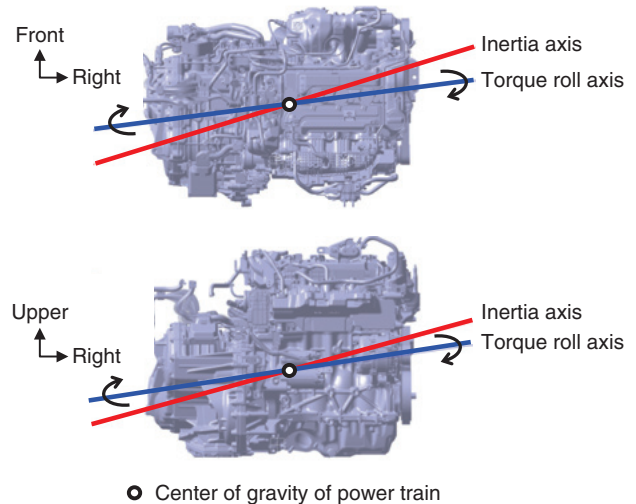


Fig. 2 Vibration mode while idling

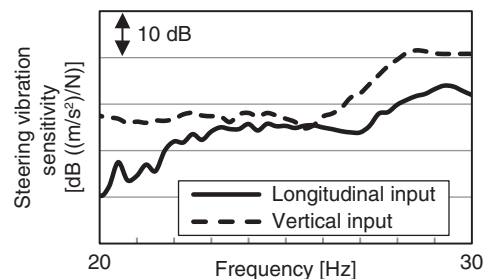


Fig. 3 Comparison of vibration sensitivity

position is smallest when the engine side-mount position in the vehicle longitudinal direction is positioned directly on the torque roll axis. These results were used to determine the positions of both main mounts.

Figure 5 shows the engine 2nd-order vibration in the vehicle vertical direction at the engine side-mount point of a vehicle equipped with the 1.5 L VTEC TURBO engine at the idling engine speed. Enhancement of the mount layout based on the above analysis results reduced the input vibration by approximately 3 dB compared to the previous model.

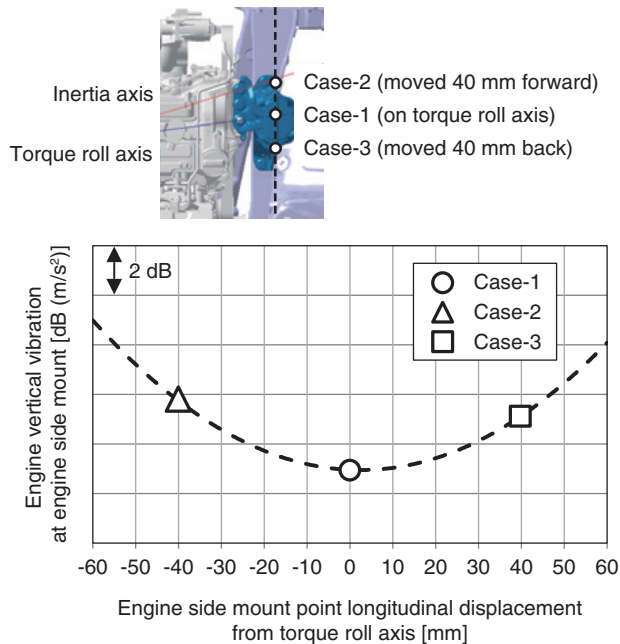


Fig. 4 Engine side mount point investigation result

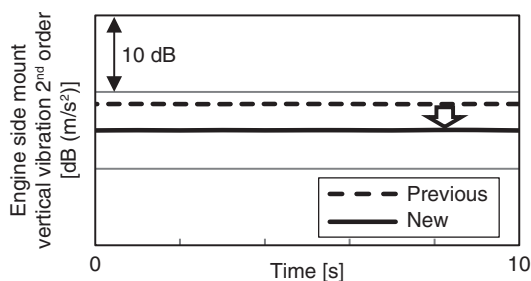


Fig. 5 2nd order engine vibration while idling

4. Technology for Reducing Floor and Accelerator Pedal Vibration

4.1. Mechanism

Floor and accelerator pedal vibration while accelerating occurs when vibration originating in the engine is transferred to the body via the engine mounts, the exhaust system, and hoses and pipes such as for the climate control and fuel

systems. For an inline 4-cylinder engine with a maximum engine speed of approximately 7,000 rpm, the frequency range of this vibration is mainly the frequency band of the 2nd-order component, or approximately 30 to 230 Hz.

Floor vibration is greatly influenced by the vibration of the exhaust system, which has hanger points on the underside of the body. Figure 6(b) shows the exhaust system configuration of the 1.5 L VTEC TURBO engine.

The exhaust system is supported elastically by a total of six mounts on the front subframe, the floor tunnel, and at the front and rear of the two main silencers. The number of main silencers was increased from one for the 1.8 L naturally aspirated engine of the previous model to two main silencers to support high power output.

In addition, the exhaust vibration at the floor tunnel exhaust hanger point shown by point B in Fig. 6 has a large influence. This means that floor vibration can be reduced by reducing the exhaust vibration amplitude at point B, and by reducing the vibration sensitivity from point B to the floor (hereafter, floor vibration sensitivity).

The accelerator pedal module (APM) is the accelerator pedal link part and is connected to the dashboard lower, so dashboard lower vibration in the vehicle longitudinal direction has a major influence on accelerator pedal vibration.

Figures 7 and 8 show the APM, heater hose, and fuel pipe bracket connection structures to the dashboard lower. The APM is connected to the interior side, and the fuel pipe bracket is connected to the engine compartment side. The heater hose passes through the dashboard lower and is connected to the climate control unit, but the fuel pipe

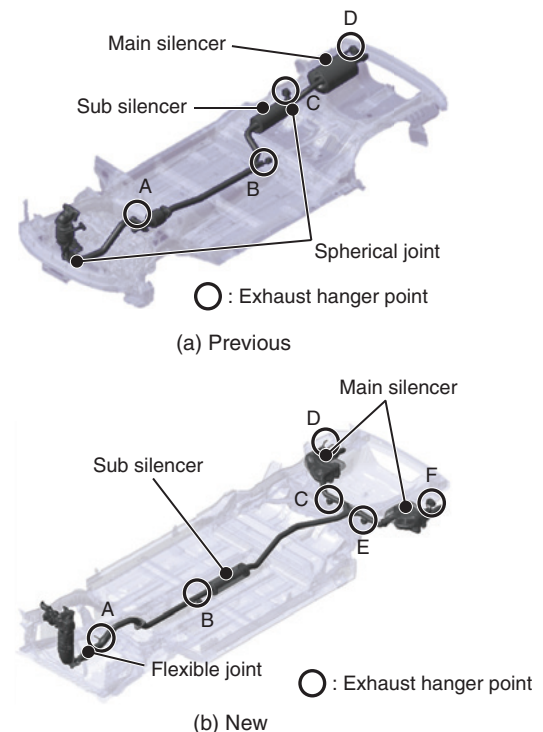


Fig. 6 Exhaust system

bracket has a clamp point to regulate heater hose movement. Engine vibration is transferred to the fuel pipe bracket via these pipes and hoses, and vibrates the dashboard lower. Therefore, one method for enhancement is to reduce the vibration sensitivity from the fuel pipe bracket to the APM (hereafter, APM vibration sensitivity) in order to reduce the accelerator pedal vibration.

4.2. Issues and Countermeasures

Engine vibration is transferred to the exhaust system via the joint with the catalyst, and amplified by the natural elastic resonance mode of the exhaust system (hereafter, exhaust elastic mode). Reducing the spring constants of the exhaust mounts is an effective means of reducing vibration transfer from the exhaust system. However, this reduces the exhaust system support stiffness and increases the vibration amount, so also achieving good handling becomes an issue in the same manner as when reducing the spring constants of the engine mounts.

In addition, while enhancement of body stiffness to reduce the vibration amplitude is an effective body-side countermeasure for floor and accelerator pedal vibration, this leads to an increase in vehicle weight, so also achieving fuel economy performance is an issue.

An effort was therefore made to reduce vibration transfer from the exhaust system to the floor by changing the exhaust joint structure to reduce the vibration transferred from the engine to the exhaust system, and by focusing on the exhaust elastic mode and positioning the exhaust hanger points in locations with a small vibration amplitude.

In addition, the floor stiffness was increased to enhance the body-side vibration sensitivity. Meanwhile, the floor system specifications, including sound isolation parts such as the floor panel and floor carpet, were also investigated to avoid increasing the vehicle weight. Regarding accelerator pedal vibration, the positions where the heater hose and fuel pipe bracket are connected to the dashboard lower were

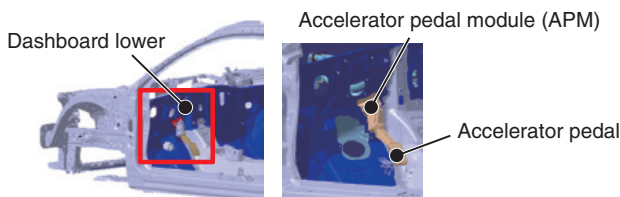


Fig. 7 Accelerator pedal module (APM)

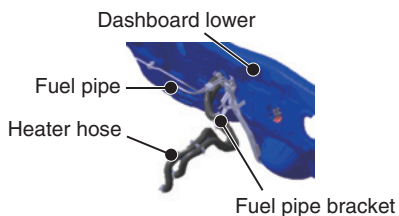


Fig. 8 Heater hose and fuel pipe bracket connection

changed to reduce the APM vibration sensitivity from the dashboard lower piping connections.

4.3. Reduction of Exhaust Vibration

4.3.1. Application of flexible joint

The spherical joints used in the previous model were changed to a flexible joint to reduce the amplitude of the engine vibration transferred to the exhaust system. This flexible joint has a bellows structure that provides elasticity and flexibility, effectively reducing vibration transfer. In order to maximize the vibration transfer reduction effect of the flexible joint, the layout positions the axial direction of the flexible joint on the engine rotation roll axis tangent to reduce vibration transfer in the axial direction (Fig. 9).

4.3.2. Exhaust elastic mode and exhaust mount layout

As a result of analyzing the exhaust elastic mode, investigation focused on the vertical bending elastic mode around 100 Hz as an elastic mode that has a major influence on floor vibration. It was found that in the exhaust system of the 1.5 L VTEC TURBO engine, the position around the floor tunnel part that has a small exhaust vibration amplitude is approximately 600 mm toward the front of the vehicle compared to the exhaust mount point B position set for the previous model (Fig. 10). So the position of exhaust mount point B was determined from these results.

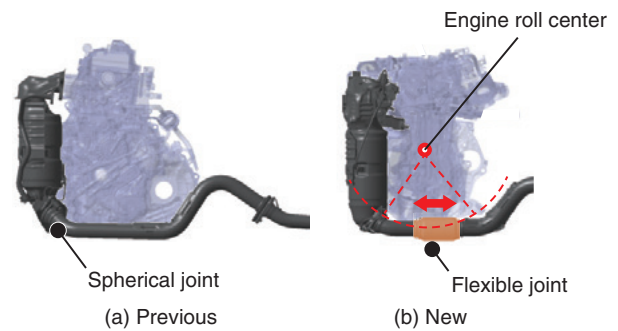


Fig. 9 Flexible joint layout

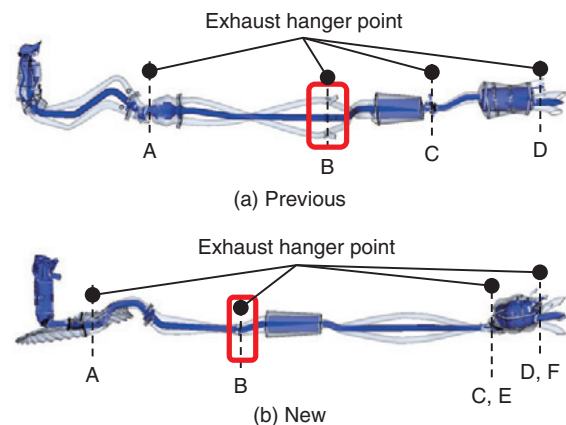


Fig. 10 Exhaust vertical bending mode around 100 Hz

Figure 11 shows the exhaust vibration in the vehicle vertical direction at exhaust mount point B. Application of the flexible joint reduced vibration by approximately 10 dB, and optimization of the exhaust mount point B position also reduced vibration by approximately 10 dB, for a combined exhaust vibration reduction effect at exhaust mount point B of approximately 20 dB compared to the previous model.

4.4. Reduction of Floor Vibration Sensitivity

Figure 12 shows the changes to the floor panel structure. The floor panel thickness was increased compared to the previous model and cross members were added to reduce the floor vibration sensitivity.

Excitation point inductance in the vehicle vertical direction is an index used to evaluate floor stiffness, and was reduced by approximately 20 dB compared to the previous model (Fig. 13) at the evaluation position shown in Fig. 12. As a result, the floor vibration sensitivity from exhaust mount point B was reduced by approximately 17 dB compared to the previous model (Fig. 14). Increasing the floor system weight was suppressed by reducing the weight of the floor carpet, which is a sound proof part.

4.5. Reduction of Accelerator Pedal Vibration Sensitivity

Investigation focused on the relative positions of the fuel pipe bracket and the APM on the dashboard lower as a means of reducing the APM vibration sensitivity from the fuel pipe bracket connected to the engine compartment side

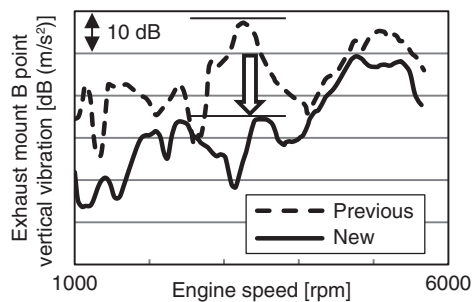


Fig. 11 Exhaust vibration reduction

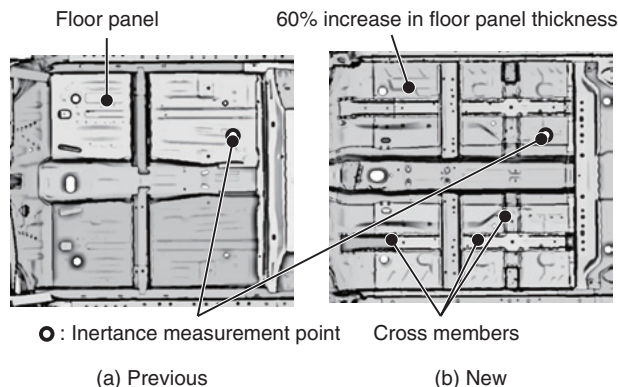


Fig. 12 Comparison of floor structure

of the dashboard lower. The new layout offsets the piping connection parts by 120 mm as shown in Fig. 15 compared to the previous model. As a result, the piping vibration transfer sensitivity to the accelerator pedal was reduced by approximately 10 dB (Fig. 16).

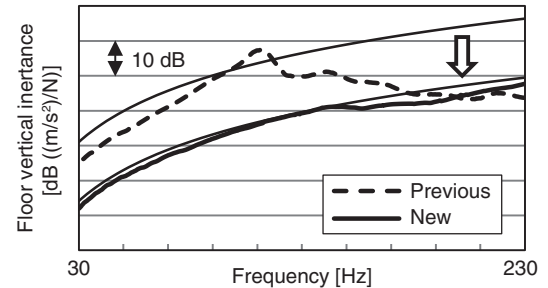


Fig. 13 Floor panel vertical stiffness

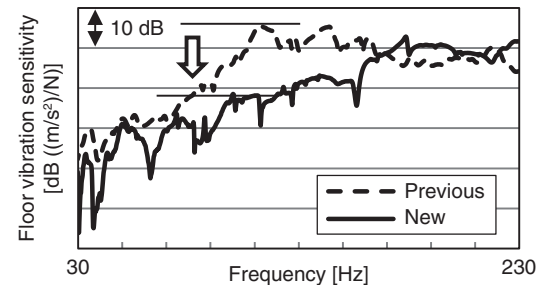


Fig. 14 Floor vibration sensitivity

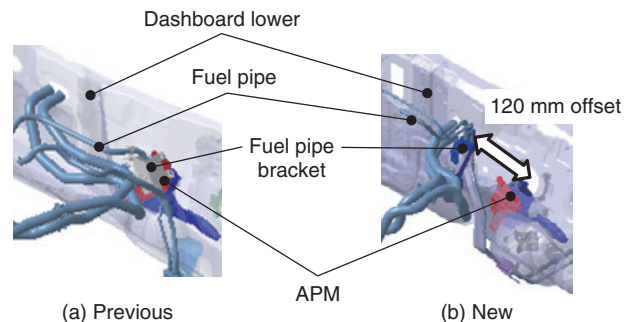


Fig. 15 Heater hose and fuel pipe bracket connection

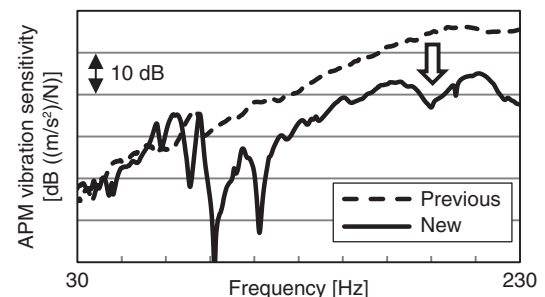


Fig. 16 Vibration sensitivity from fuel pipe bracket on dashboard lower to APM

5. Verification of Results in an Actual Vehicle

Application of the above technology achieved the target interior vibration reduction.

Figure 17 shows the steering 2nd-order vibration in the vehicle vertical direction while idling in the D range engine speed. Vibration was reduced by 8 dB compared to the previous model. Figure 18 shows the 2nd-order vibration of the floor in the vehicle vertical direction at the driver's foot position while accelerating, and Fig. 19 shows the 2nd-order vibration in the plane direction of the accelerator pedal head while accelerating. Vibration was respectively reduced by 10 and 7 dB, compared to the previous model.

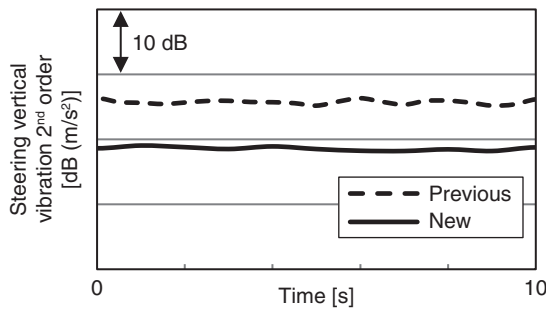


Fig. 17 Steering vibration while idling

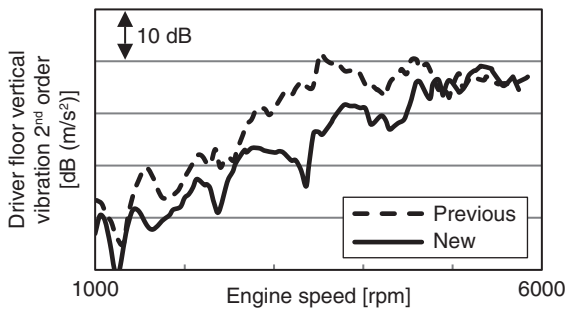


Fig. 18 Floor vibration while accelerating

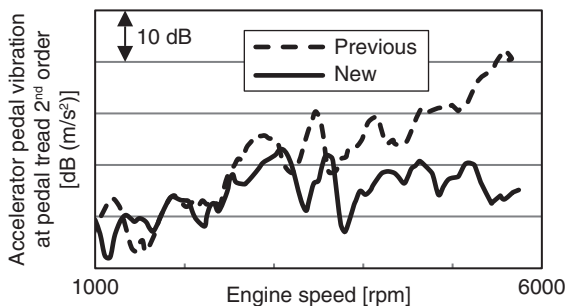


Fig. 19 Accelerator pedal vibration while accelerating

6. Conclusion

- (1) Steering wheel vibration while idling was reduced by analyzing power train rolling vibration while idling and positioning the main mounts in locations with the minimum engine vibration to reduce the vehicle vertical direction component of the engine vibration input from the main mounts.
- (2) Exhaust vibration transferred to the body was reduced by applying a flexible joint to the exhaust system and by positioning the floor part exhaust mounts in locations with a small vibration amplitude based on the exhaust elastic mode analysis results. In addition, vibration sensitivity was reduced by increasing the floor stiffness and by offsetting the APM mounting point and fuel pipe bracket position. These changes succeeded in reducing floor and accelerator pedal vibration while accelerating.

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Advanced Technology for Road Noise Reduction for New CIVIC

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ABSTRACT

Road noise performance was enhanced when developing the new CIVIC. To reduce low-frequency road noise when driving on rough road surfaces, the suspension main vibration mode and sub-frame vibration mode frequencies were controlled so that they do not overlap with the acoustic mode frequencies that are determined by the cabin shape and volume. In addition, to reduce high-frequency road noise when driving on flat road surfaces, soundproof specifications were applied to locations with a high contribution to cabin noise, and the sealing performance of the doors was enhanced. These measures achieved road noise performance that is competitive as a global model.

1. Introduction

Customer interest in NVH performance continues to increase. In particular, the quietness of competing vehicles centering on the European premium class has evolved to a high level of quality that surpasses mere quietness and makes an impression on customers. Efforts are being made to reduce road noise over a wide frequency band from low-frequency road noise when driving on a rough road surface to high-frequency road noise when driving at high speeds. This paper describes the technology applied to reduce both low- and high-frequency road noise, which have different mechanisms.

2. Road Noise Reduction Goal

Figure 1 shows the cabin noise spectrum when driving on a rough road surface. As a result of comparison with the previous model, competing vehicle A, and competing vehicle B, enhancement of low-frequency road noise from approximately 100 to 200 Hz and high-frequency road noise in the frequency band above 800 Hz were set as enhancement goals for the new CIVIC.

The following two technologies were applied to achieve the above goals.

- (1) To reduce low-frequency road noise, the suspension and subframe vibration modes were analyzed, modes with a

high contribution were identified, and these frequencies were separated from the cabin acoustic mode frequency to reduce the sound pressure.

- (2) To reduce high-frequency road noise, the soundproof package layout was optimized based on panel contribution analysis, and the size of the holes and gaps (hereafter, body leakage area) was reduced.

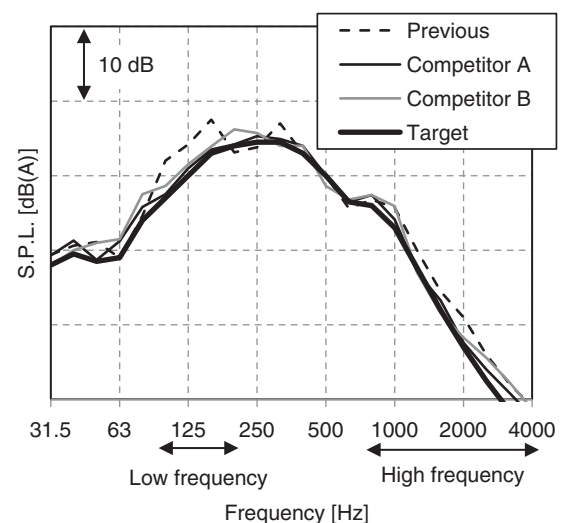


Fig. 1 Road noise comparison to competitors

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3. Low-frequency Road Noise

Low-frequency road noise produced when driving on a rough road surface is a phenomenon wherein tire and suspension vibration excited by the road surface roughness is transmitted to the body via the suspension bushings, and the body panels vibrate the air, producing noise. Therefore, an effective method for reducing low-frequency road noise is to reduce the spring constants of the suspension bushings to isolate vibration. In addition, adjusting the main vibration mode frequencies of the suspension in consideration of the combination with the cabin acoustic modes is also a known method for reducing low-frequency road noise⁽¹⁾.

However, reducing the spring constants of the suspension bushings reduces the suspension support stiffness, which has a significant influence on handling. Therefore, to realize the target chassis dynamic performance of the new CIVIC, it is necessary to select a method that can simultaneously achieve both handling and road noise performance.

3.1. Low-frequency Road Noise Reduction Goals

The new CIVIC uses a strut-type front suspension and a multi-link-type rear suspension. The rigid camber mode and elastic camber mode of each suspension are known to be the main vibration modes that have a major influence on low-frequency road noise⁽¹⁾. The suspension vibration at these vibration mode frequencies has a resonance peak, so it is necessary to adjust the suspension vibration mode frequencies so that there is no overlap with the body response-side characteristic peak.

The body characteristics are evaluated by the cabin noise produced when the suspension connecting point is excited (hereafter, body acoustic sensitivity). The cabin acoustic modes are one factor that determines the body acoustic sensitivity. Figure 2 shows the cabin acoustic modes that strongly influence the body acoustic sensitivity. Low-frequency road noise was reduced by separating these frequencies and the front and rear suspension mode frequencies.

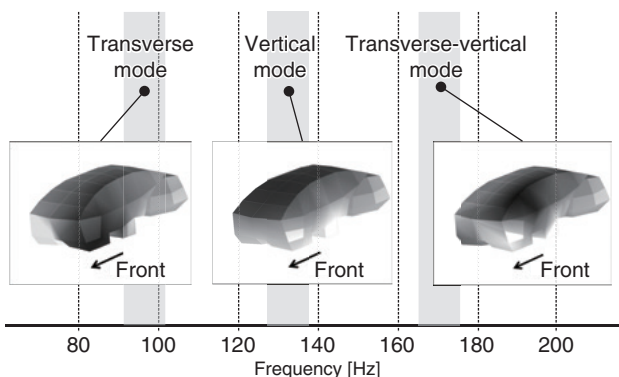


Fig. 2 Cabin acoustic mode

3.2. Front Suspension

The new CIVIC uses a strut-type front suspension. This suspension has an elastic camber mode at approximately 180 Hz where the wheel rotates in the camber direction while the wheel disk surface, hub bearing, knuckle, and other parts undergo elastic deformation (Fig. 3).

It is known that cabin noise can be reduced the more the cabin acoustic mode frequency (approximately 170 Hz) and the elastic camber mode frequency are separated⁽¹⁾. The cabin acoustic mode frequency is determined by the cabin shape and volume, so it is not easily changed. Therefore, increasing the elastic camber mode frequency was investigated, and the elastic camber mode target frequency range was set from 195 to 205 Hz based on the cabin noise reduction target.

The strain energy density of the suspension parts in elastic camber mode was analyzed using CAE, and the results showed that the area from the hub to the damper connection part of the front knuckle has a high strain energy density and a high contribution to the elastic camber mode frequency (Fig. 4).

The elastic camber mode frequency could be increased to the target range by increasing the thickness by approximately 9 mm at the position indicated by the arrows in Fig. 5(a) (original) and (b) (modified). Figure 6 shows the damper vibration and Fig. 7 shows the cabin noise during wheel excitation. These figures show that the elastic camber mode frequency was separated from the acoustic

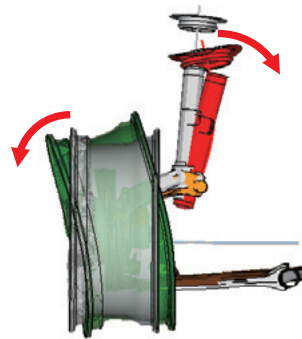


Fig. 3 Front strut suspension in elastic camber mode

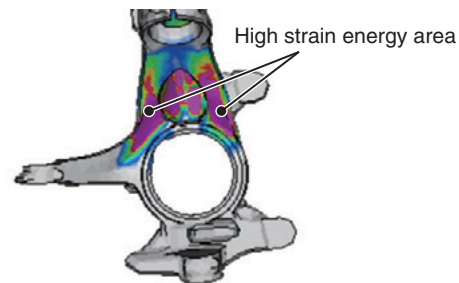


Fig. 4 Strain energy density of front knuckle in elastic camber mode

mode frequency, which enabled reduction of the sound pressure in the 170 to 180 Hz range by 3 dB (approximately 30% reduction).

3.3. Rear Suspension

The multi-link-type rear suspension has a rigid camber mode at approximately 125 Hz where the knuckle and wheel rotate together in the camber direction (Fig. 8).

In addition, the suspension has a rear subframe part

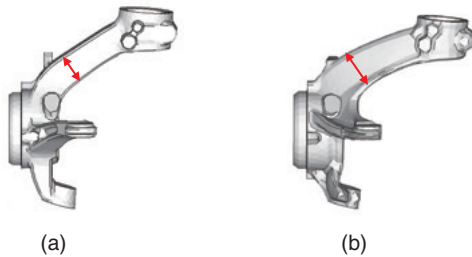


Fig. 5 Front knuckle modification

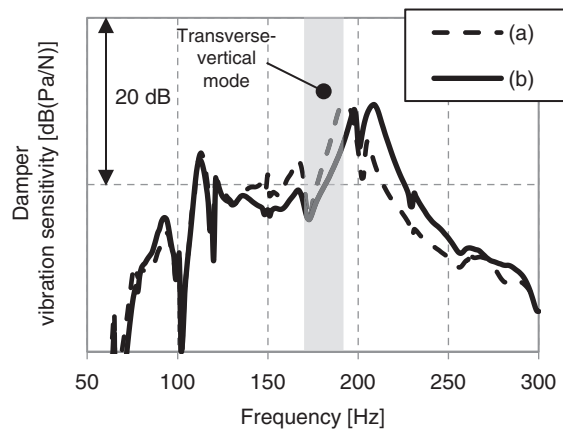


Fig. 6 Front suspension vibration sensitivity

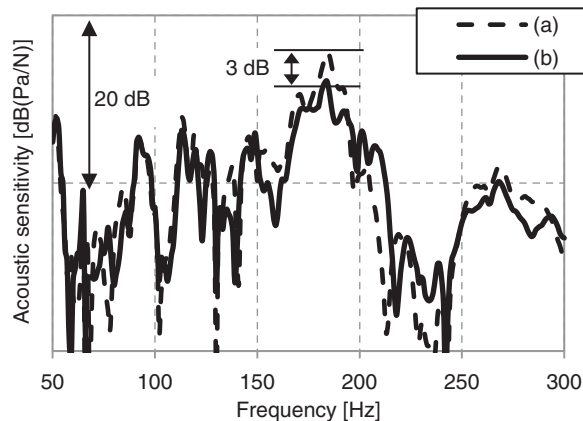


Fig. 7 Reduction of road noise by increasing stiffness of front knuckle

that connects the rear suspension and the body. This rear member has a longitudinal resonance mode at approximately 130 Hz, which is close to the rigid camber mode of the suspension (Fig. 9).

There is also a cabin acoustic mode at approximately 130 Hz, so to reduce road noise at this frequency, it is necessary to control the rear member resonance of the rear subframe in addition to the rigid camber mode of the rear suspension.

Unlike the elastic camber mode, the rigid camber mode does not involve elastic deformation, so it is not easy to change the vibration mode frequency by changing the stiffness of the arm, knuckle, or other parts. It was therefore decided to separate the rear subframe vibration mode frequency from the cabin acoustic mode frequency. The target frequency was set to the range from 185 to 195 Hz from the viewpoint of also achieving handling stiffness performance. Figure 10 shows this frequency arrangement.

The vibration mode of the rear member of the rear subframe produces deformation in the longitudinal direction as shown in Fig. 9. In order to increase the vibration mode frequency, it is necessary to increase the stiffness of the connection between the rear and side members. Figure 11 shows the rear subframe shape. Stiffness was increased by modifying the shape of the rear member to increase the size and widen the connection with the side member as shown in Fig. 11(a) (original) and (b) (modified). This made it

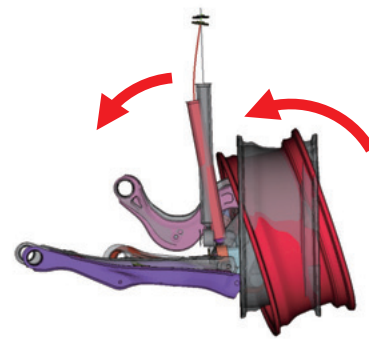


Fig. 8 Rear multi-link suspension in rigid camber mode

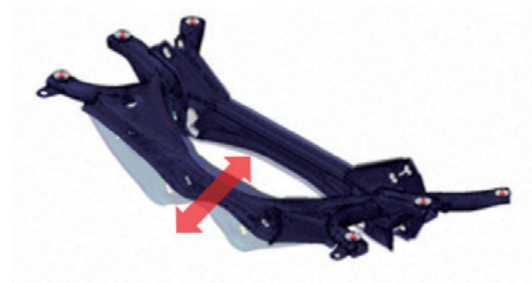


Fig. 9 Longitudinal resonance mode of rear member of rear subframe

possible to achieve the target frequency.

Figure 12 shows the rear subframe longitudinal vibration and Fig. 13 shows the cabin noise during rear wheel excitation. Subframe vibration at the cabin acoustic mode frequency in the vehicle longitudinal direction was reduced by 7 dB, and cabin noise was reduced by 5 dB.

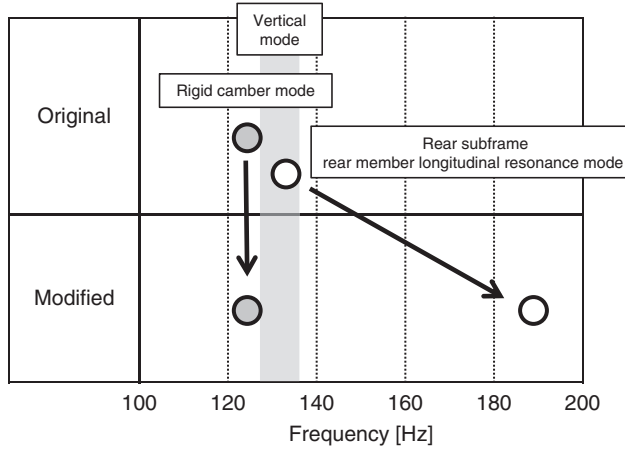


Fig. 10 Rear suspension resonance map

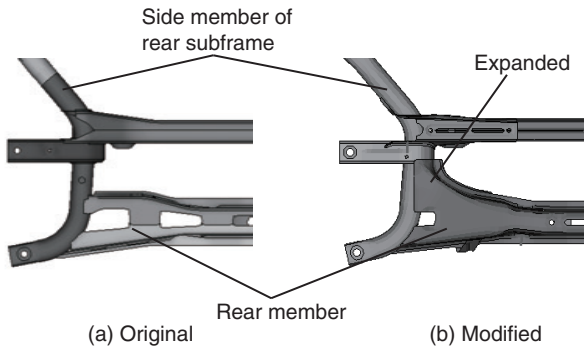


Fig. 11 Rear subframe design change

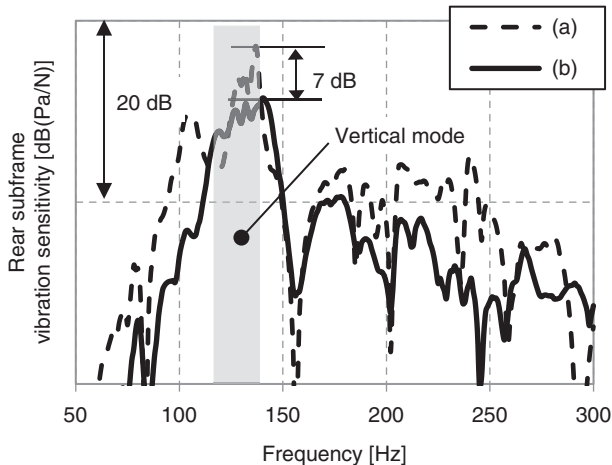


Fig. 12 Rear subframe vibration reduction

4. High-frequency Road Noise

High-frequency road noise includes phenomena such as tire radiation noise produced when road surface roughness vibrates the tires and the resulting tire surface vibration vibrates the air, and pumping noise produced when the air in the tire grooves is trapped by tire contact with the ground and then released. These noises enter the cabin by penetrating the panels and through gaps in the body, so the application of soundproof materials is one effective measure. However, the addition of soundproof materials increases the weight, so simultaneously achieving vehicle weight reduction to enhance fuel economy performance is an issue. For this reason it is necessary to investigate countermeasures, including the body side structure to which the soundproof materials are applied, to minimize the increase in soundproof system weight accompanying the enhanced performance.

4.1. High-frequency Road Noise Goals

First, body locations with a high contribution to road noise were identified, and the soundproof package layout was optimized. Next, countermeasures were taken to block sound entry paths by reducing the body leakage area.

4.2. Optimal Soundproof Package Layout Based on Contribution Analysis

Contribution analysis was performed and the soundproof specifications were investigated as an index for evaluating the energy transmission loss of each acoustic input path to the cabin⁽²⁾. Figure 14 shows the contribution analysis results at 1 kHz.

These results showed that the highest contribution rate was from the wheel house and floor around the driver and the rear-seat foot areas, and accounted for approximately 60% of the total.

Enhancement of the acoustic absorption performance of the under-floor space, the noise insulating performance of the floor surface, and the acoustic absorption performance

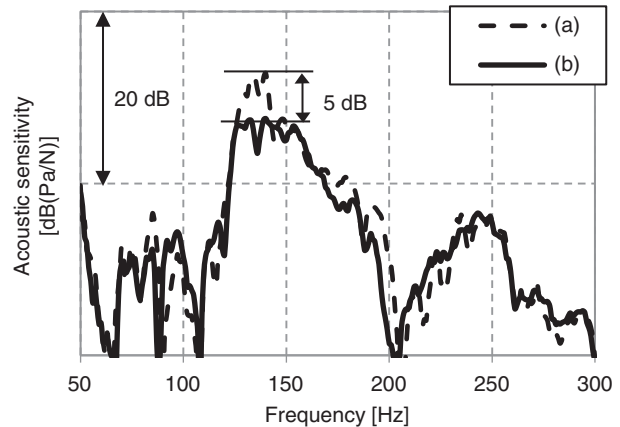


Fig. 13 Reduction of road noise by increasing stiffness of rear member of rear subframe

inside the cabin are effective means of enhancing the energy transmission loss at the floor. Acoustic absorbent materials were applied to the floor undercover and the tunnel heat baffle to enhance the acoustic absorption performance of the under-floor space. In addition, a structure that arranges carpet formed with an integrated insulator and carpet skin was used to increase the noise insulating performance of the floor (Fig. 15).

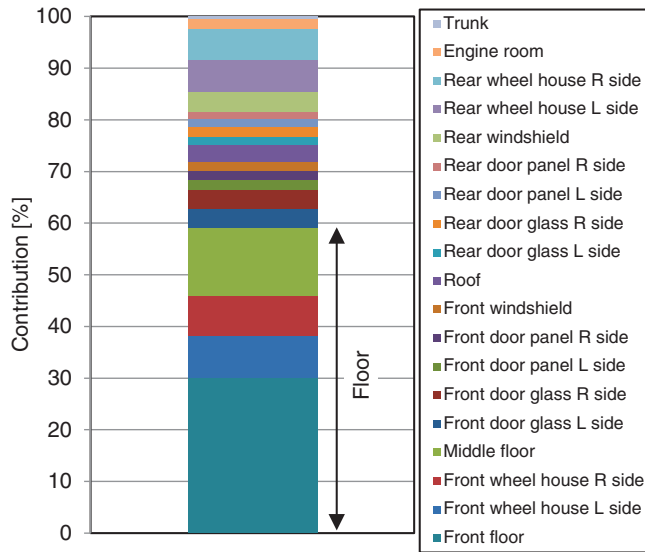


Fig. 14 Area contribution to 1 kHz road noise

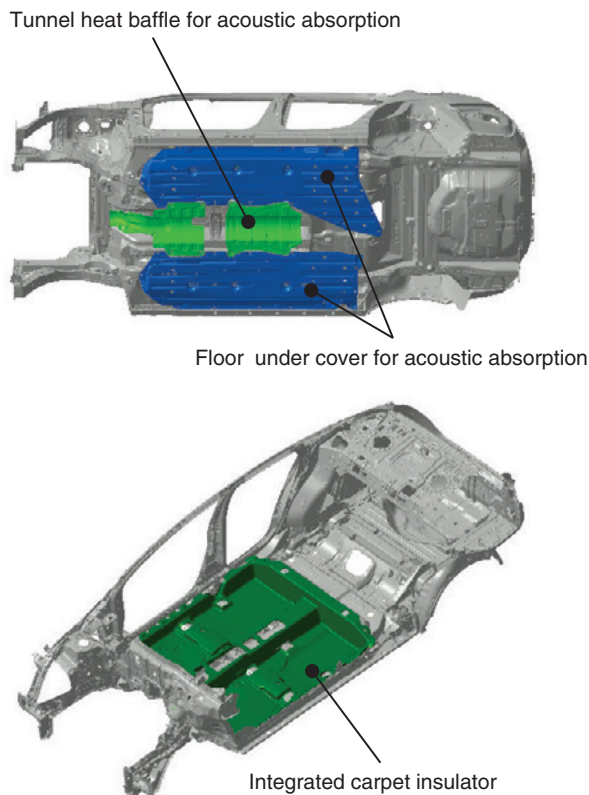


Fig. 15 Sound proof system

These countermeasures enhanced reduced the energy transmission loss of the floor by 3.5 dB compared to the previous model (Fig. 16).

4.3. Reduction of Body Leakage Area to Help Prevent Noise Entry

Noise entry paths can be blocked by reducing the body leakage area, representing an effective means of reducing high-frequency road noise.

Contribution analysis was performed for the body leakage area in the previous model, and the results showed that the doors make a large contribution, accounting for approximately 40% of the overall leakage area. The results of more detailed analysis showed that the main factors were the drain holes located at the bottom of the doors, the gaps at the ends of the inner weather strips, and the gaps in the hole seal that covers the door work holes (Fig. 17).

A lip-shaped door lower seal that covers the drain holes was applied to prevent noise entry from the drain holes (Fig. 18). Molded rubber was applied to both ends of the inner weather strip to reduce the gaps with the door inner panel (Fig. 19). A structure that covers the door harness clip with the hole seal was used to reduce the gap between the hole seal that covers the door work holes and the door harness (Fig. 20).

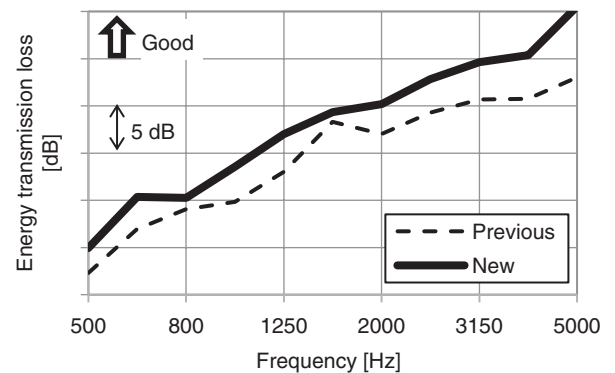


Fig. 16 Energy transmission loss at floor

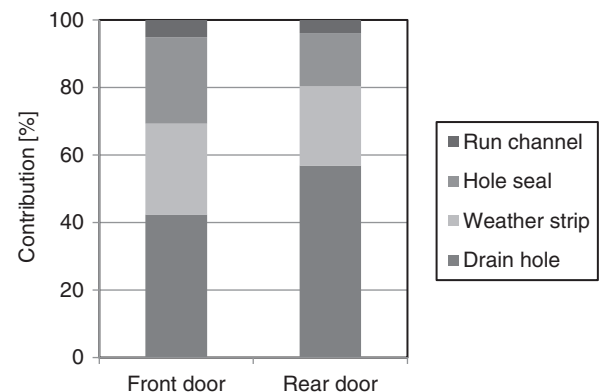


Fig. 17 Body leak area contribution of door

As a result of these measures, the body leakage area was reduced by approximately 30% compared to the previous model.

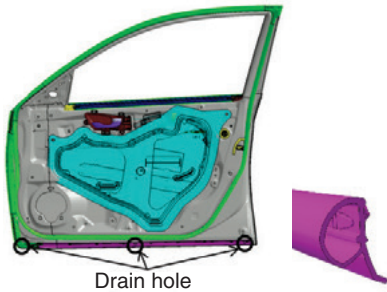


Fig. 18 Door lower seal

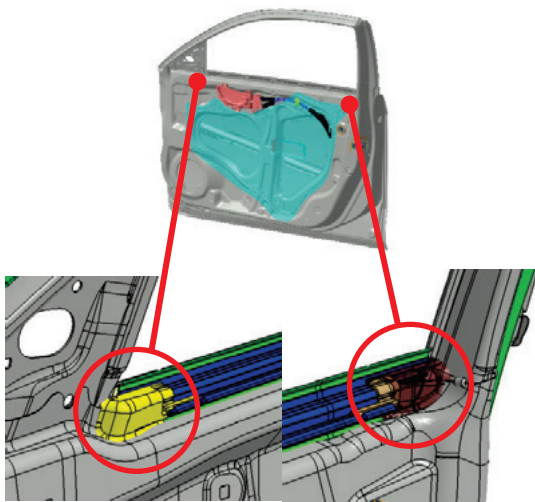


Fig. 19 Door inner weather strip with mold rubber

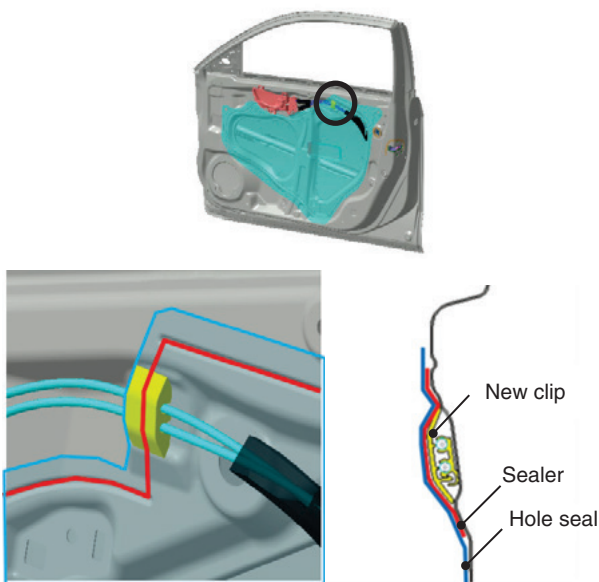


Fig. 20 Door harness clip

5. Results of Verification in an Actual Vehicle

Incorporation of the above technology made it possible to achieve the target quietness while cruising.

Figure 21 shows the cabin noise spectrum at the front seat when driving on a rough road surface. The developed technology achieved road noise reduction of 5 dB at the low frequency of 125 Hz and 3 dB at high frequencies of 1000 Hz and above compared to the previous model.

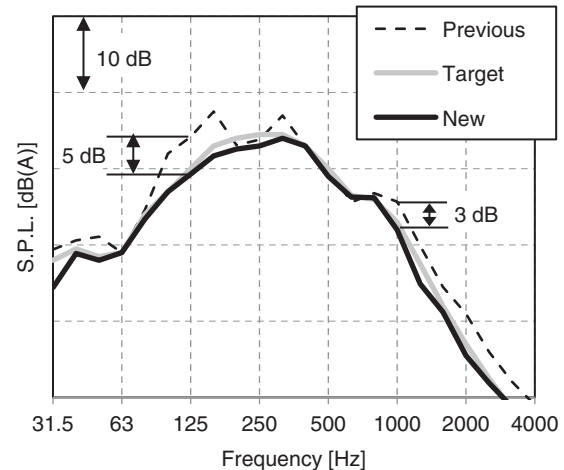


Fig. 21 Road noise on rough surface

6. Conclusion

- (1) The suspension and subframe were analyzed with regards to low-frequency road noise, and modes with a high contribution to cabin noise were identified. Low-frequency road noise was reduced by 5 dB by separating these frequencies from the cabin acoustic mode frequencies.
- (2) Panel contribution analysis was performed with regards to high-frequency road noise, the floor was identified as making a large contribution, and the soundproof package layout was optimized. In addition, the doors were identified as making a major contribution to body leakage, and countermeasures were taken to enhance the sealing performance. These measures reduced high-frequency road noise by 3 dB.

The above technology achieved road noise performance that is competitive as a global model.

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Vehicle Dynamics Performance of CLARITY FUEL CELL

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ABSTRACT

The dynamic performance targeted with the CLARITY FUEL CELL has the unique new appeal of a fuel cell vehicle with a full complement of universal values of a passenger car, such as ease of driving. The Accord PHEV, which is a vehicle in the same class as the CLARITY FUEL CELL and has a high level of performance, was chosen as the benchmark for performance objectives. Specifically, this means that the CLARITY FUEL CELL provides a solid driving experience, making full use of one of the distinctive advantages of electric vehicles, that they allow drivers to freely control the car's speed using only the accelerator pedal. It provides handling performance that demonstrates how easily course-holding can be managed when driving, a smooth, comfortable ride from the moment the car starts moving, and a positive brake feel that makes the car easy to control in all the various driving scenarios encountered from day to day. It is also extremely quiet. The developed vehicle satisfied the objectives and successfully realized the image that was desired.

1. Introduction

Automobiles are on a course of evolution toward electrification and a shift toward next-generation alternative fuels as ways to address issues such as the growing demand for fuel-saving technology to deal with the depletion of fossil fuel resources, as well as the issue of global warming caused by CO₂ and other such environmental issues. Fuel cell vehicles (FCV) involve the use of high-pressure hydrogen tanks and complex auxiliary devices, as well as modifications to vehicle body construction and other such measures to protect these systems from the shock of collisions. The resulting increase in weight has become an issue. While lower rolling resistance coefficients (RRC) for tires have been promoted in order to increase fuel efficiency, the adoption of tires with lower RRC also introduces the important issue of assuring grip performance, which is reduced by a lower RRC, and providing satisfactory dynamic performance with a heavy vehicle body.

The CLARITY FUEL CELL has a major role to play in FCV dissemination and expansion. In order to provide the joy of free mobility to every occupant, including passengers, this vehicle was developed using the most up-to-date technologies.

The vehicle had a distinctive layout⁽¹⁾ with the hydrogen tank mounted behind the rear seat and tire covers added to the rear wheel houses to reduce air resistance. Particular attention was therefore focused on dealing with the handling performance issues that arose as a result of this layout. The longitudinal distribution of weight was shifted more toward the rear than the common front wheel drive (FF) vehicle with an internal combustion engine (ICE). In addition, the placement of tire covers on the outside of the wheel houses added to the constraints on the size of tire. These made it necessary to examine performance with a high degree of accuracy.

This paper takes dynamic performance as a comprehensive term for the handling performance, ride comfort, steering feel, brake feel, dynamic performance, and noise and vibration performance that drivers sense subjectively as their impressions of the vehicle. The technology applied to realize the target objectives will be introduced below.

2. Development Concept and Aims

The development concept was a “Red Carpet to a New Era.”⁽¹⁾ The meaning packed into this expression refers to the novel appeal, unique to the FCV, in addition to the full

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complement of passenger car performance, ease of use, and other such universal values. In this way, the vehicle is intended to be a special vehicle that will propel the customer into a new era (Fig. 1).

The Accord PHEV, which was launched on the market before the CLARITY FUEL CELL as an environmental vehicle in the same class, was used as the benchmark based on this concept. Goals for dynamic performance were set to surpass that benchmark, as follows.

- (1) Electric-power appeal: The speed control performance that is unique to the electric vehicle is communicated to the customer as a point of appeal.
- (2) Car appeal: The various types of dynamic performance that provide an impression of the vehicle are communicated to the customer as a point of appeal.

Concerning electric-power appeal, control of acceleration and deceleration are well-known characteristics of electric vehicles, and excellence in this matter is a point of appeal. Therefore, importance was placed on achieving a balance between powerful acceleration, in which there is no delay from the instant the accelerator is pressed down in a stationary car until maximum torque is generated, and sustained smooth acceleration without changing gears after a certain point. Noise originating in the power train was also drastically reduced.

Alternative fuel cars, hybrid cars, and other such cars in categories considered to have a high level of environmental performance are equipped with power train systems that add weight, and with low-RRC tires. Therefore, they require more advanced technology to secure performance that compares with ICE vehicles of the same class in areas such as handling performance, ride comfort, steering feel, brake feel, and noise and vibration. This vehicle was developed so that the kind of dynamic performance that is felt by the driver and passengers as their impression of the vehicle would be equal to or greater than the performance of an ICE vehicle.



Fig. 1 Honda CLARITY FUEL CELL

3. Realizing Electric-power Appeal

3.1. Smooth and Direct Acceleration Sensation

In most ICE vehicles, it is not possible in principle to avoid a time lag from the time the accelerator is pressed until the combustion of fuel is converted into kinetic

energy, or to avoid fluctuations in acceleration immediately after acceleration due to the interposition of a torque converter. However, electric vehicles do not experience that kind of loss, nor do they need torque converters. Consequently, they can provide a greater sensation of smooth and direct acceleration than can ICE vehicles. This advantage was used effectively to enliven the driving force characteristics. The default setting that is in effect when the power train is started up is called normal mode. It is configured with the aim of realizing high-quality driving, so that it assures full responsiveness while allowing for smooth driving without any jerking, even when accelerator operation involves some roughness. A sport mode was also configured with greater responsiveness of driving force than normal mode, to allow the experience of driving with a greater range of variation⁽¹⁾.

Figure 2 shows the acceleration in each mode when the accelerator pedal (AP) is pressed to go rapidly from fully closed to 50% open. In an ICE vehicle, the torque amplification effect of the torque converter is lost immediately following acceleration, and a drop in acceleration can be observed when the coupling area is reached. With the CLARITY FUEL CELL, however, acceleration is activated with good response even in normal mode, following which acceleration is maintained without any drop in speed. In sport mode, a balance is achieved between even greater responsiveness and sustained acceleration.

3.2. Operability Capable of Freely Accelerating and Decelerating with the AP

In an electric vehicle, the kinetic energy of the vehicle can be recovered when the AP is off, which means that deceleration characteristics can be configured over a broad range. Therefore, the system was configured to provide acceleration performance with regard to the AP opening of the kind shown in Fig. 3, so that jerking does not occur even with AP operation that is somewhat rough in normal mode, such as during the acceleration described in section 3.1, and so that sport mode provides acceleration and deceleration control that can be felt over a range of

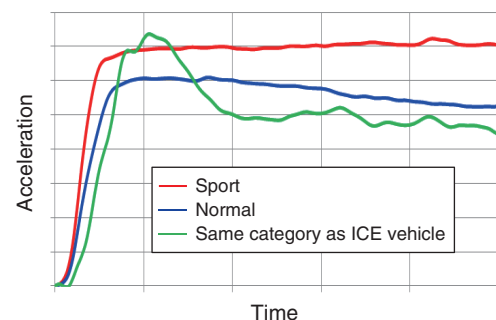


Fig. 2 Acceleration in normal and sport modes with AP (50%)

operation. Figure 4 shows deceleration with respect to vehicle velocity when the AP is fully closed. Sport mode obtains deceleration that is stronger at vehicle speeds that are higher than in normal mode. When the AP is fully closed and the pedal is again pressed down, the acceleration performance shown in Fig. 3 can be achieved, indicating that the range of possible acceleration and deceleration control has been expanded relative to normal mode.

3.3. High Degree of Power Train Quietness

When the FCV is generating electric power, it is

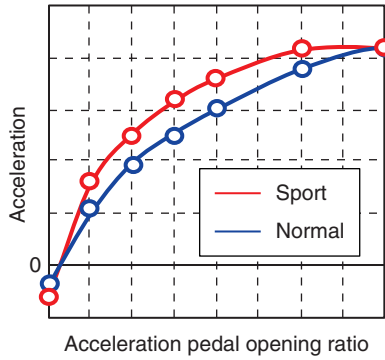


Fig. 3 Acceleration characteristics

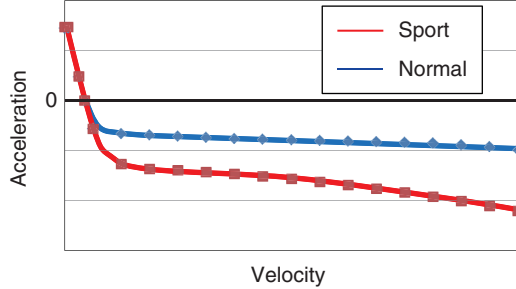


Fig. 4 Deceleration characteristics

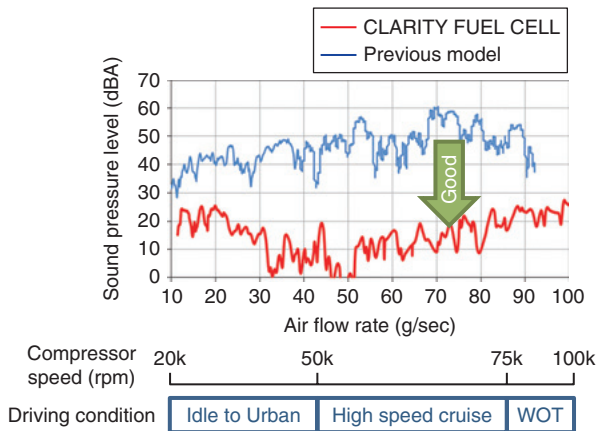


Fig. 5 Reduction of compressor noise in cabin

supplied with oxygen in the form of air to cause a chemical reaction with hydrogen. Therefore, the CLARITY FUEL CELL is equipped with a motorized turbo compressor⁽²⁾. The adoption of this compressor has reduced operating noise by approximately 40% over the previous Lysholm compressors, enhancing cabin quietness from starting up the power train to stopping it (Fig. 5).

4. Realizing Car Appeal

4.1. Ease of Line Tracing when Driving

Compared to ordinary ICE vehicles with FF, this vehicle faces two issues: the longitudinal distribution of weight, which tends to be rear-heavy, and the adoption of low-RRC tires. In order to deal with these issues, an investigation of handling performance was carried out through simulations during the initial stage of development. The vehicle specifications, including the height of the center of gravity and the tire size, were incorporated into the initial layout so as to achieve the dynamic performance defined in terms of suspension geometry and other such objectives. The rigidity required in each part of the vehicle body for that purpose was also assured by adopting a unique construction.

Several tire sizes were investigated, from the largest mountable size to the smallest allowable size according to the load factor with respect to wheel load. The characteristics of the tires alone were estimated using statistical tire models, and given the cornering power (CP) characteristics relative to axle load, the choice was narrowed down to two types, 235/45R18 and 225/50R17⁽³⁾⁻⁽⁴⁾ (Fig. 6).

Simulations were conducted of the dynamic performance of various types, typified by frequency response, USOS, and double lane changes with the CP characteristics of the various tires, and a comparison was

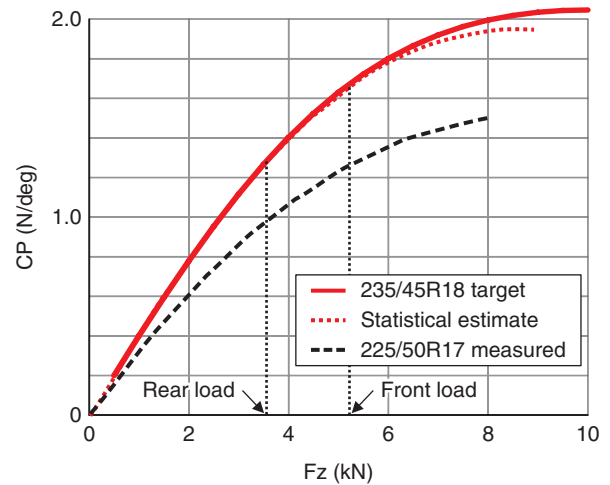


Fig. 6 Measured tire CP (Fz) characteristics of 225/50R17 and target CP (Fz) of 235/45R18

made with the benchmark Accord PHEV. Figure 7 shows the frequency response calculation results as representative of the investigated content.

With the 225/50R17, even when there was a high response gain in a low-frequency band near a steady state condition, it dropped at 1 Hz or higher. With the 235/45R18, however, it was found that response could be enhanced across the entire range. Given these results, the 235/45R18 size tire was selected.

It is well known that in addition to tire grip performance, the rigidity of the rear body and suspension attachment points are important in assuring line tracing performance. However, in order to provide space in the trunk and lower the center of gravity, the hydrogen tank was structured to be

cradled in the rear subframe was devised, as shown in Fig. 8. Since this limited the design space, assuring rigidity became an issue. Therefore the rear bulkhead structure shown in Fig. 9 and the structure shown in Fig. 10 that places stiffeners under the rear subframe were devised. In this way, a rear body and suspension attachment structure that is lightweight and highly rigid were successfully realized, so that ease of line tracing is achieved even with the use of low-RRC tires.

4.2. Smooth Ride Feeling

The objective set for ride comfort was to realize a smoothness that is appropriate for the seamless and smooth acceleration performance that is unique to electric vehicles. Therefore, sensitive frequency response dampers (SFRD)⁽⁵⁾ were adopted. These were also adopted in the

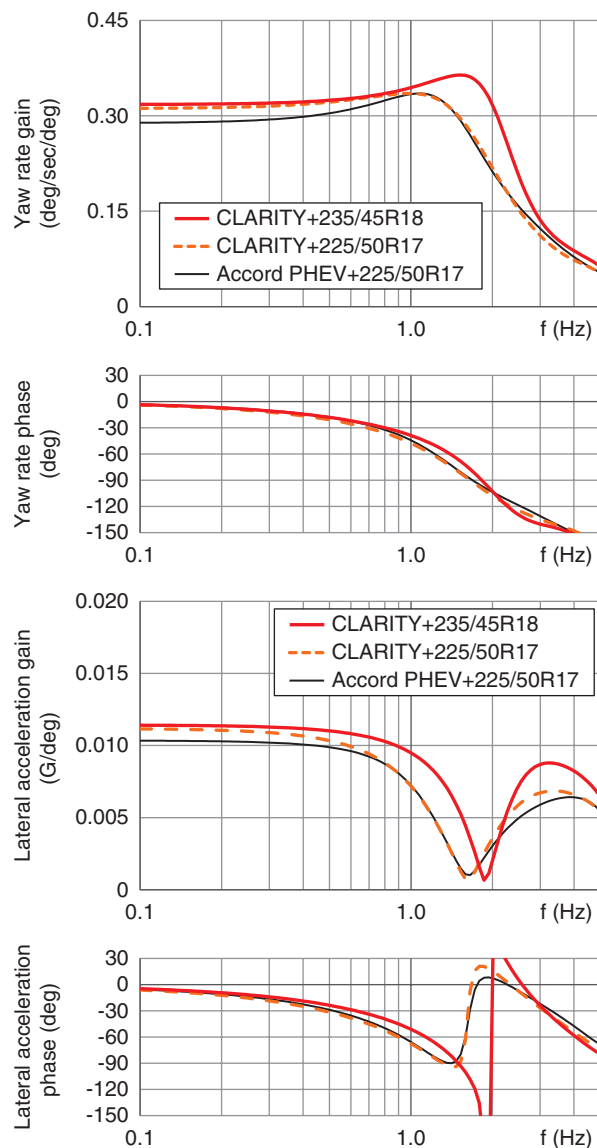


Fig. 7 Bode diagrams of estimated frequency response of yaw rate (upper) and lateral acceleration (lower) with various tire characteristics

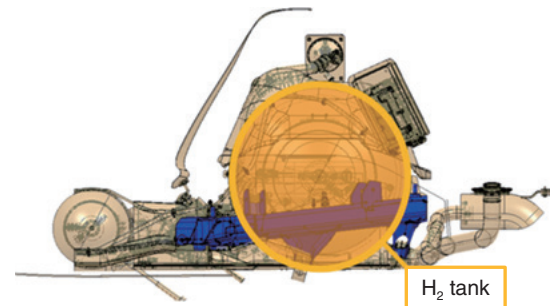


Fig. 8 Side view of rear subframe with H₂ tank

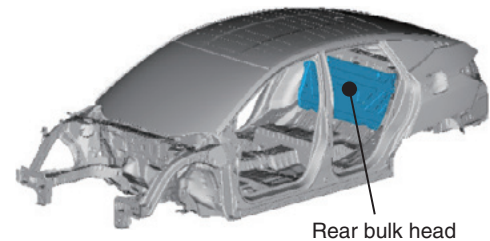


Fig. 9 Rear bulk head

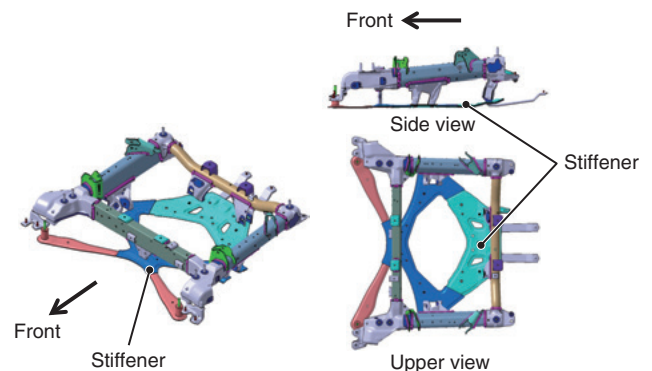


Fig. 10 Rear subframe with stiffener

Accord PHEV, which was launched on the market before this vehicle, and the characteristics of these devices were matched to those of the body described in section 4.1. Running tests were conducted repeatedly under various conditions, and sensory evaluations performed by experts in running tests were used to configure comprehensive settings that included springs, stabilizers, and rubber bushings. The system was raised to a higher level of maturity and the aimed-for dynamic performance was achieved.

4.3. Natural Motion when Changing Course

Vehicles with a low level of handling performance do not readily maintain stable behavior, so they cannot readily be driven smoothly, and passengers also experience discomfort from the motion of the vehicle while it is being driven. Attention was therefore focused on the relationship between the yaw rate and roll with respect to steering wheel operation. By reducing hysteresis and heightening linearity, the predictability of vehicle behavior relative to steering was increased.

Suspension specification values and steering control settings were developed with reference not only to sensory evaluation, but also to actual vehicle behavior based on this conceptual approach. The result was that the vehicle motion when steering was felt to be as natural as or more natural than that of the Accord PHEV that was chosen as the benchmark (Fig. 11).

4.4. Easily Controllable Brakes

For the brakes, the same electric servo brake system⁽⁶⁾ used in the Accord PHEV was adopted in order to provide the required performance in a variety of driving scenarios. The objective was defined as realization of an easily controlled brake feel. The method for achieving this was to set the reaction force to the brake pedal stroke. When the degree of deceleration is small, the slope of the pedal reaction force with respect to the stroke is increased in order

to increase the reaction force with respect to the stroke. It is set so that the reaction force would increase gradually from that point if the brake pedal is pressed down even further from that point, depending on how far it is pressed. As a result of these measures, the pedal reaction force was made smoothly continuous from the normal operating range to the range where deceleration increases, and brake performance that can be controlled as easily as or more easily than in the benchmark Accord PHEV was realized.

4.5. Quietness with Regard to Road Noise and Wind Noise

The FCV power train is quieter than that of an ICE vehicle. It was reasoned that FCV occupants would therefore be bothered by the intrusion of road noise and wind noise into the cabin as much as or more than the occupants of ICE vehicles, and the target performance for interior quietness was set at a higher level than for an ICE vehicle of the Accord class. As shown in Fig. 13, the absorption of sound in areas close to the noise input source and the insulation of sound along the path of its intrusion were accomplished efficiently. Sound insulating glass was

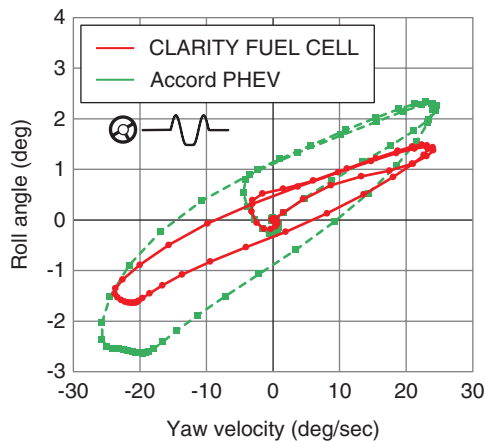


Fig. 11 Relationship between roll and yaw velocity during sinusoidal steer

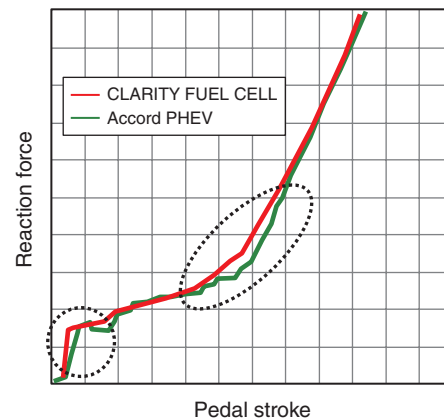


Fig. 12 Relationship between brake pedal stroke and reaction force

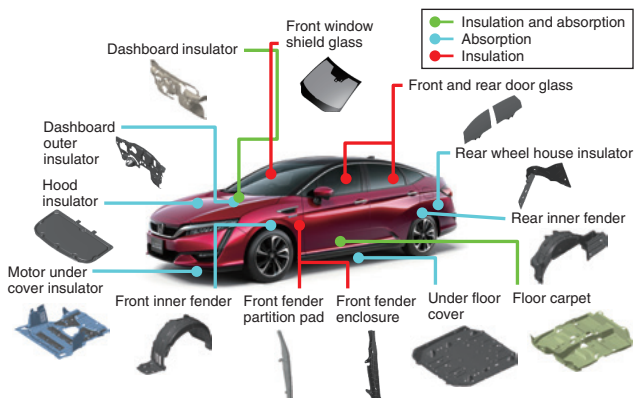


Fig. 13 Sound proof item

also used at many points to reduce noise intrusion.

The floor carpet was made using a multilayered structure that is highly effective in sound insulation and sound absorption. Attachment parts were also produced to reduce the areas of gaps in order to enhance the sound insulation and sound absorption efficiency (Fig. 14).

The wheels were also equipped with resonators in order to limit road noise from tire cavity resonance and shock absorber noise when driving over different levels^{(7), (8)} (Fig. 15).

The effects of these measures and the adoption of an active noise control (ANC) system⁽⁹⁾ achieved road noise performance in the E-segment class, as shown in Fig. 16. Favorable performance was also achieved in wind noise, as shown in Fig. 17.



Fig. 14 Floor carpet layout



Fig. 15 Wheel resonator

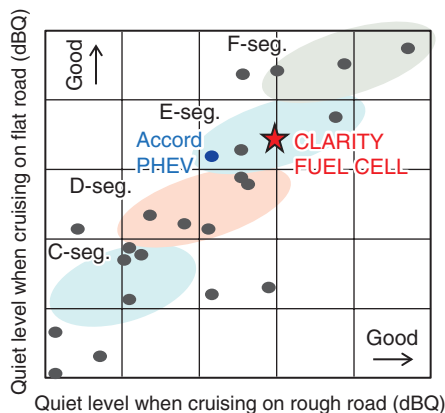


Fig. 16 Quiet feel index (cruising)

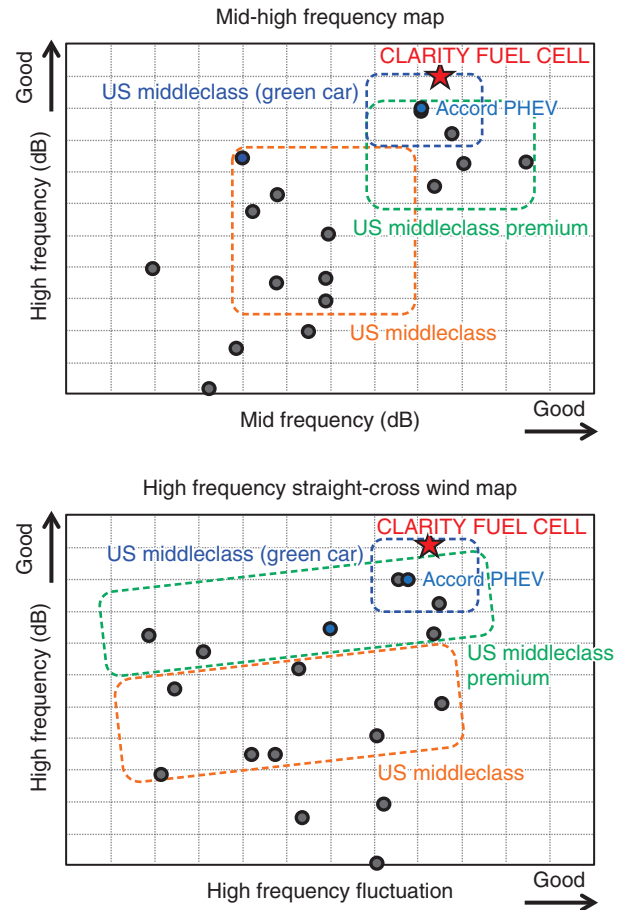


Fig. 17 Wind noise index

5. Conclusion

Development was conducted with the aim of increasing dynamic performance for both electric-power and car appeal. The Accord PHEV was used as the benchmark for the objectives for each, and ways for advancing performance from that level were considered thoroughly from the customer's perspective.

The various aims were achieved using a variety of technologies. As a result of achieving those individual aims, development is considered to have successfully produced a vehicle endowed both with the unique new appeal of an electric vehicle and with the appeal of a car. The hope is that this will become a special car that moves forward with the customer into a new era.

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Large-sized and Lightweight Aluminum Hollow Diecast Subframe Using High Pressure Diecasting

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ABSTRACT

A large-sized, lightweight aluminum, hollow diecast subframe with increased strength was developed. Technology for baking the entire interior of the core, which is the key to realizing a closed section structure, and materials with enhanced castability were applied to the subframe for the first time known to date, and realized a thin wall structure even with high-pressure diecasting. In addition, the use of a seamless closed-section structure increased the package efficiency and achieved weight reduction of approximately 46% while securing the same stiffness as that of a welded steel plate subframe. These developments realized a top-class lightweight subframe.

The developed subframe was mounted in the CLARITY FUEL CELL, and both reduced the vehicle weight and facilitated fuel cell power train installation under the front hood, achieving high package efficiency equivalent to that of a vehicle with an internal combustion engine.

1. Introduction

There is an increasing need to reduce automobile weights in order to reduce carbon and save energy to protect the global environment and to enhance vehicle maneuverability. Technology that enhances both fuel economy performance and maneuverability while at the same time achieving good environmental performance and marketability will become even more necessary in the future. The development of technology that further reduces the vehicle body weight including chassis parts is called for as fuel cell vehicles and electric vehicles start to become popular⁽¹⁾.

The front subframe (Fig. 1) that joins the suspension parts and power plant to the vehicle body is one of the larger-sized chassis parts, and optimizing the front subframe structure to reduce the weight contributes greatly to reducing the overall vehicle weight.

Honda has performed material substitution to aluminum to reduce the subframe weight. For example, technology that uses friction stirring to weld together expanded material and steel plates in automobiles has been developed for diecast parts with an open-section structure that formerly used aluminum high-pressure diecasting (HPDC) and squeeze diecasting⁽²⁾, and this technology has been applied to the ACCORD HYBRID and FIT EV subframes. However, welding parts are needed when creating a closed-section

structure by welding, and securing an efficient section shape given the limited layout space inside the engine and motor compartments becomes an issue. Conventional technology employs an aluminum hollow integrated structure made using gravity diecasting (GDC) and low-pressure diecasting (LPDC), but casting by GDC and LPDC calls for a certain thickness, and weight increase is an issue. Technology was developed that uses a core created by the lamination manufacturing method together with LPDC to reduce the wall thickness in order to reduce weight⁽³⁾, but it was a

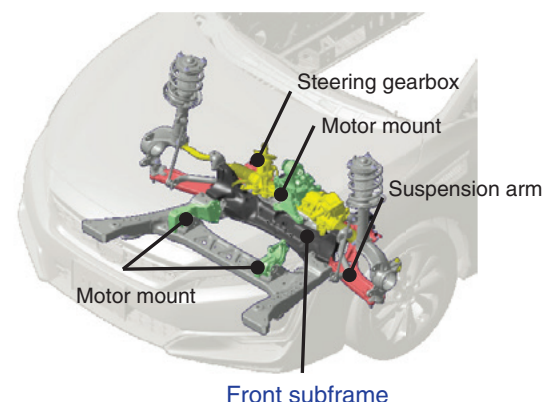


Fig. 1 Front subframe position in vehicle

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Table 1 Overview of subframe manufacturing methods

	GDC	LPDC	HPDC	
Joint	None	None	FSW	None
Thickness	△	△	○	○
Weight	△	△	○	◎
Layout efficiency	○	○	△	○
Productivity	△	○	◎	◎

◎ : Excellent
○ : Good
△ : Acceptable

challenge to secure mass productivity. Table 1 gives the characteristics of closed-section structures created using different manufacturing methods.

Honda developed closed-section technology using aluminum HPDC and applied it to the CBR600RR motorcycle frame. However, this technology welds together small-sized diecast parts with a closed-section structure and diecast parts with an open-section structure manufactured using GDC, so hollow integrated casting was not achieved⁽⁴⁾, and there is room for further weight reduction.

In order to address these issues, a new subframe was developed by using existing facilities with HPDC, which is advantageous for weight reduction, in order to realize large-sized parts with a thin wall closed-section structure⁽⁵⁾.

2. Development Goals

This study set the following goals and developed new subframe manufacturing technology.

- (1) Realization of a large-sized, thin wall closed-section structure using HPDC

Establishment of core manufacturing conditions that can be applied to HPDC and use general-purpose HPDC facilities, and increased core strength to help realize larger products with thinner walls.

- (2) Realization of mass productivity including materials that simultaneously achieve both castability and good mechanical properties

Making use of the advantages of hollow diecasting using a core, which enables to secure fluidity due to lower thermal conductivity, to realize mass productivity using materials suited to diecasting.

- (3) Realization of a top-class lightweight subframe

Weight reduction due to thinner walls, a seamless structure that reduces fatigue starting points, elimination of welding parts in order to increase freedom of layout, and stabilization of product wall thickness.

3. Realization of Large-sized Thin Wall Closed-Section Structure by HPDC

If the HPDC gate speed is too fast, the core will shift or split, resulting in shape defects. If the casting pressure is too high, the core will be crushed and shrink, resulting in thicker product walls. Therefore, the casting conditions were changed from the typical HPDC conditions to reduce the gate speed and casting pressure to within the ranges where the sand core does not break during diecasting (Fig. 2).

3.1. Core Manufacturing Conditions

Figure 3 shows the diecasting process flow. The core

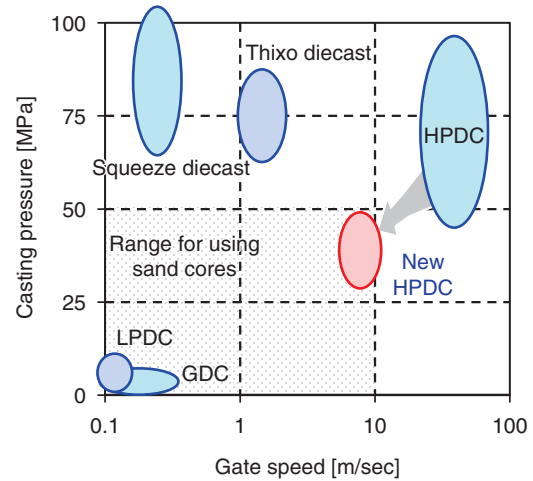


Fig. 2 Target casting pressure and gate speed target

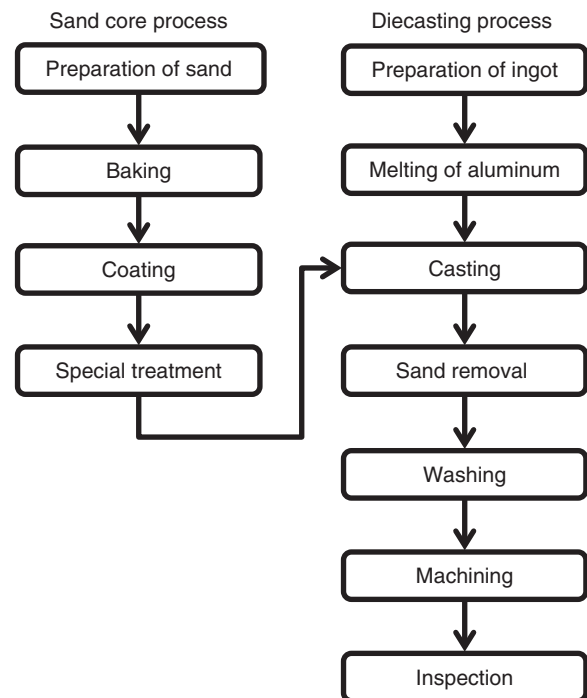


Fig. 3 Manufacturing process

is applied to a large-sized subframe, so it is larger and thicker than a conventional core, which calls for a longer baking process time. In addition, a surface coating material is applied to the core to enhance the melt flow rate and the sand removal performance. The completed core is then fixed in the die and aluminum diecasting is performed. After diecasting is complete, the gate area is struck to vibrate the product in order to break down and discharge the core.

3.2. Hollow Casting Technology

The core was cast by die baking with phenol-type resin-coated sand (RCS) as the core material. Figure 4 shows the cross-section of a core baked using conventional baking conditions. This shows that only the surface parts that contacted the die are baked, and most of the interior was not heated, so the core strength is low. In this condition the core will split or be crushed during diecasting with the result that the product wall thickness will increase, so it is necessary to bake the entire interior of the core. The phenol resin used in RCS starts to harden at a temperature of 160°C or more and degrades rapidly at 400°C or more, so the baking conditions should be set within the range from 160°C to 400°C.

In addition, increasing the baking temperature bakes thick areas of the core more quickly, while thin areas become over-baked, resulting in strength differences by area. Furthermore, over-baked areas are brittle and break easily, so the ideal baking conditions were found that enable evenly baking the entire core (Fig. 5).

3.3. Materials that Achieve both Castability and Mechanical Properties and Establishment of Mass Productivity

Aluminum diecast materials used in chassis parts need high strength, high toughness, and corrosion resistance. Therefore, aluminum-silicon (Al-Si) and aluminum-magnesium (Al-Mg) materials are used that have smaller loading amounts of elements such as copper (Cu) and iron (Fe), which adversely affect corrosion resistance. Regarding the material selection, Al-Mg material was selected that exhibits stable strength at a gate speed of approximately 10 m/s, which is one of the casting conditions (Fig. 6). Al-Mg material has lower strength than conventional Al-Si material

Area that looks white is not baked.

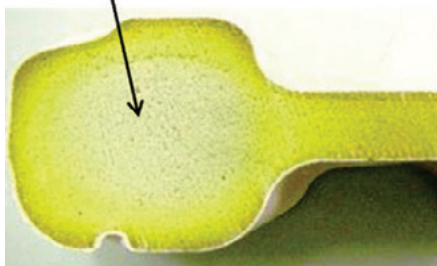


Fig. 4 Sand core cross section after baking

that has been heat-treated, and shrinkage cracks easily occur due to shrinkage at the time of solidification, so it has lower castability and is not often used as a casting material for large-sized thin wall parts. To address these issues, this study added Si to Al-5Mg material, which precipitated Mg_2Si , enhancing the mechanical properties. In addition, this also lowers the liquid phase temperature, so pressure can be transferred until immediately before solidification, which enhances molten metal feeding.

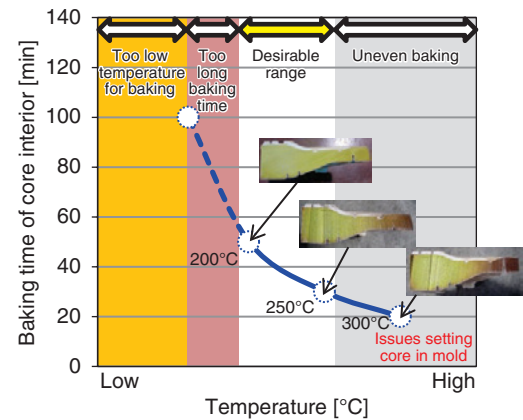
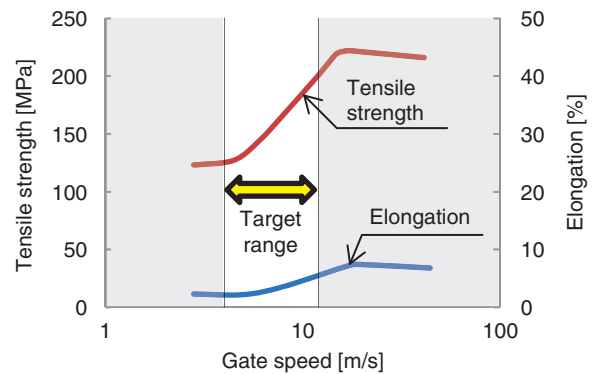
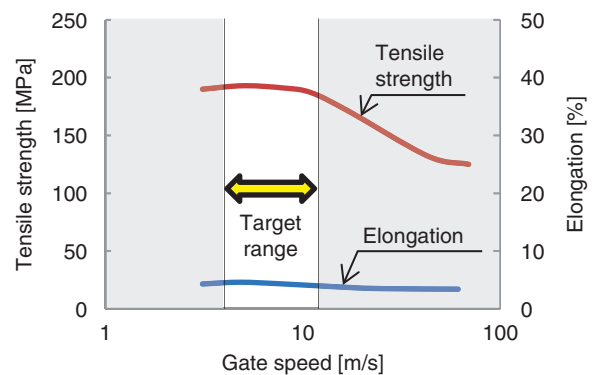


Fig. 5 Relationship between baking time and temperature



(a) Al-Si material



(b) Al-Mg material

Fig. 6 Relationship between gate speed and strength

The Si loading amount was determined by performing casting tests using a sheet-shaped test piece die (length 300 mm × width 100 mm × thickness 6 mm) as a pre-test, and comparing the mechanical properties. Figure 7 shows the relationships between the Si loading amount and the tensile strength (TS), 0.2% yield strength (YS), and elongation (EL). This study determined that a Si loading amount of approximately 2.0% provided the material with the appropriate balance between strength and elongation.

Materials used in subframes need both high elongation to satisfy the vehicle crash performance and high strength to reduce the vehicle weight. Strength and elongation typically have a trade-off relationship, so the optimum material needs to be selected in accordance with the purpose of use. A part die was therefore used to confirm the change in mechanical properties in more detail in the Si loading amount range of 1.8% to 2.2%. Figure 8 shows the tensile test piece sampling positions. As in the pre-test, the tensile strength

peaked at 2.0%, but the Si loading amount was set to 1.8% in consideration of toughness, which is a part requirement characteristic (Fig. 9).

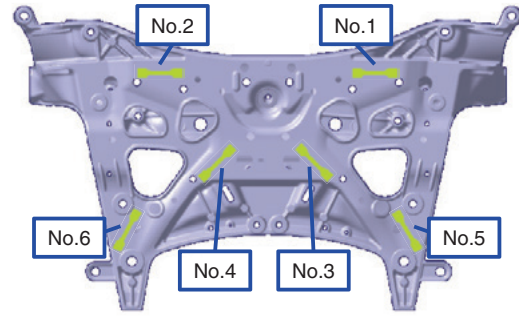


Fig. 8 Tensile test piece sampling positions

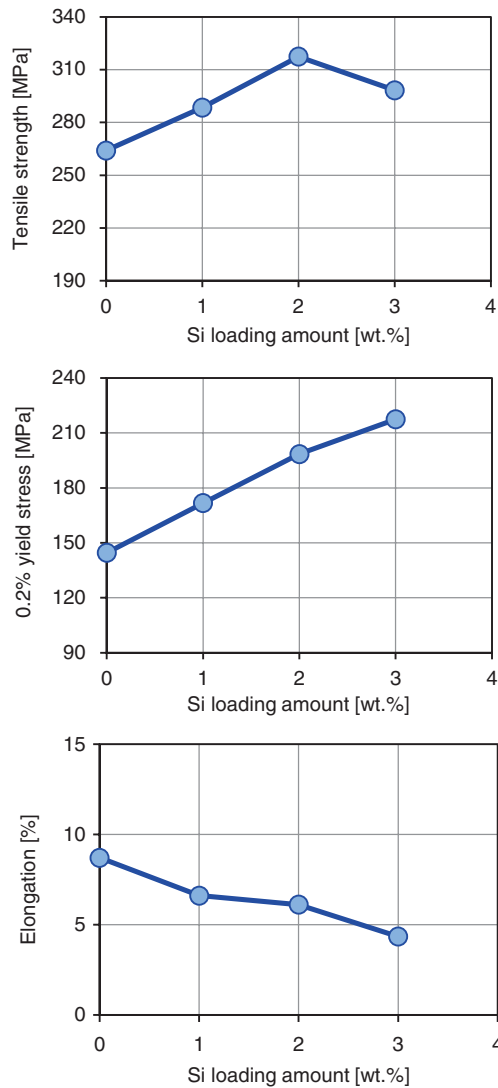


Fig. 7 Change in mechanical properties of test piece by Si loading amount to Al-5Mg material

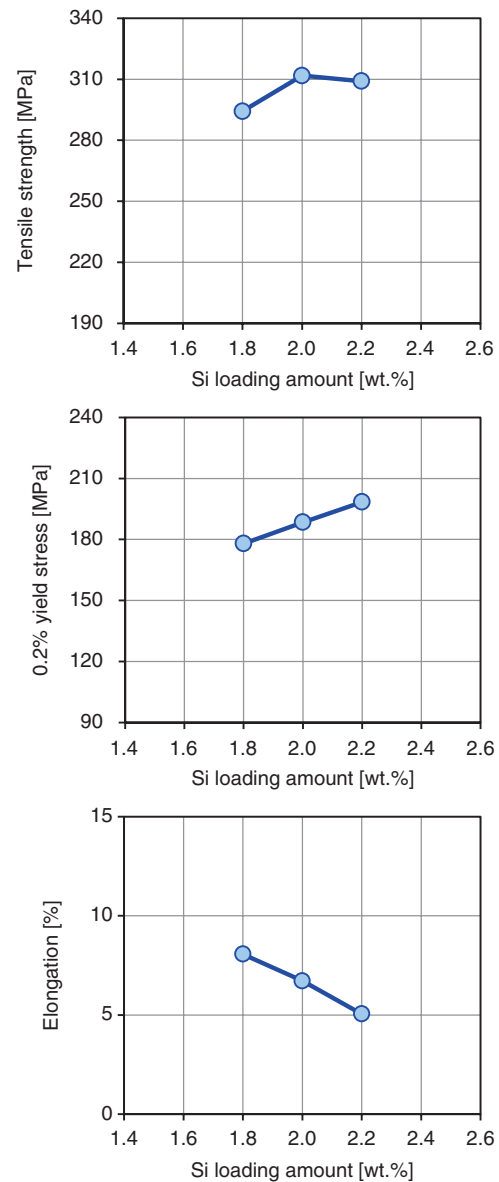


Fig. 9 Change in mechanical properties of subframe by Si loading amount to Al-5Mg material

4. Realization of a Top-class Lightweight Subframe

Development of this new subframe used aluminum hollow diecast technology, so the closed-section structure is seamless instead of the conventional welded structure. However, depending on the subframe shape, the molten metal that flowed into the die during casting deforms and damages the core due to the casting pressure, so a structure that prevents this damage is needed. This study therefore set the core holding positions during casting at opposing positions on the top and bottom of the subframe. This suppressed core deformation during subframe casting and stabilized the product wall thickness (Fig. 10). In addition, core strength analysis was performed for different casting pressures during aluminum filling, the stress generated on the core was quantified, and a product shape was found that helps prevent stress concentration in the core (Fig. 11). Furthermore, a uniform cast wall thickness is usually set to enhance castability, but casting analysis and product

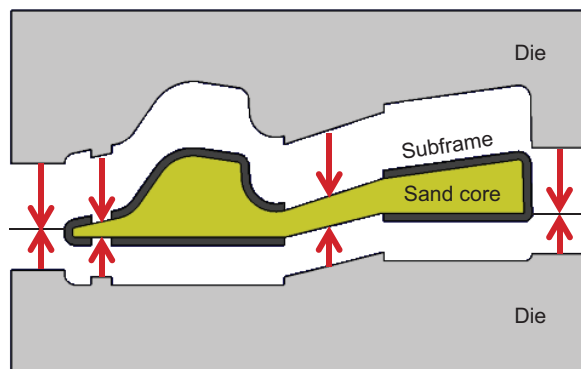


Fig. 10 Sand core holding positions

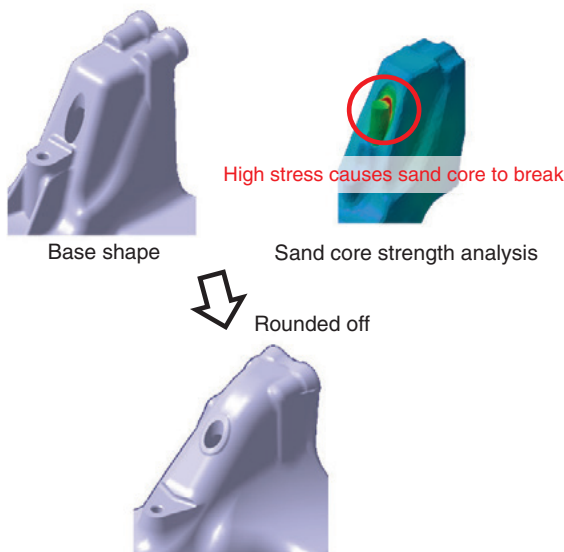


Fig. 11 Design using sand core strength analysis

design were performed in parallel. This enabled the use of a variable wall thickness that positions the necessary material in the necessary locations, achieving a structure that is both lightweight and has high stiffness. Specifically, the wall thickness of the suspension lower arm and steering gearbox mounting area that contributes greatly to maneuverability was set to 3.5 mm to ensure support stiffness, the wall thickness of the suspension mounting areas was set to 2.5 mm to ensure bolt tension, and the wall thickness of the lower surface area was set to 3.0 mm to enable absorption of crash energy without separating. This enabled to realize both the ideal stiffness distribution and high strength level demanded of the subframe (Fig. 12).

In addition, welded parts are unnecessary and seamless casting can be performed, which ensured high stiffness with a space-saving design compared to conventional weld joining (Fig. 13). As a result, a lightweight subframe layout with good stiffness that also ensures crash performance could be achieved even within the restrictions of the limited motor compartment space of the CLARITY FUEL CELL.

The mechanical properties were confirmed by cutting a test piece from the product and measuring the tensile strength, 0.2% yield strength, and elongation. Judging from the performance requirements of the applied part, the results confirmed that yield strength and elongation that stably meet or exceed the target values could be obtained. In addition, the performance was also confirmed by static destruction tests and durability tests under actual vehicle input conditions.

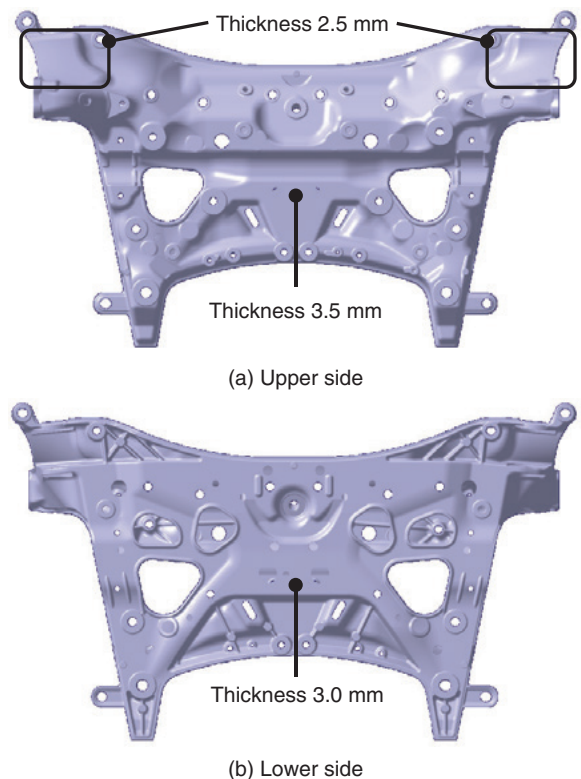
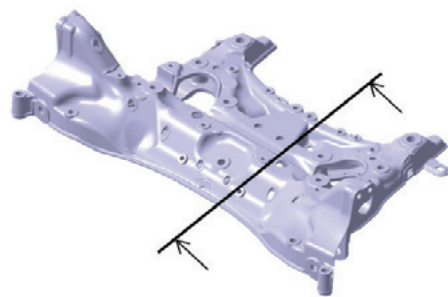
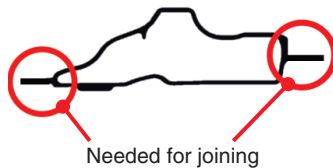


Fig. 12 Subframe thickness



(a) Section cut area



(b) Conventional subframe



Seamless closed section structure

(c) CLARITY FUEL CELL subframe

Fig. 13 Structure of closed section

The shape of the newly developed aluminum hollow diecast subframe was optimized by measures such as material substitution from steel plate to aluminum and elimination of welding parts. This achieved weight reduction of approximately 46% compared to the conventional part, and realized a top-class lightweight subframe (Fig. 14).

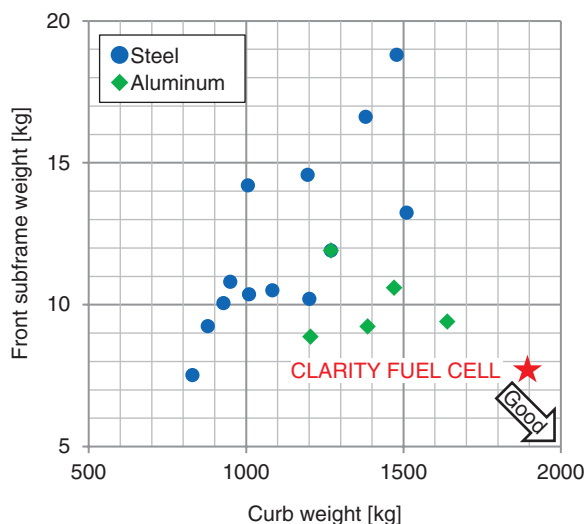


Fig. 14 Subframe weight

5. Conclusion

This study reduced the weight of the front subframe, which is one of the largest chassis parts, and established an HPDC production method that can be applied to mass production.

- (1) Core casting technology was developed that can be applied to HPDC, and conditions were established that enable even baking of the entire interior of a large-sized core.
- (2) A chemical composition that satisfies both castability and mechanical properties was created by adding 1.8% Si to non-heat treatment-type Al-5Mg materials
- (3) The developed aluminum hollow diecast subframe achieved weight reduction of approximately 46% compared to a conventional subframe formed by pressing and welding steel plate. This realized a top-class lightweight subframe.

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Development of Chassis Performance Design Technologies for MBD of Mass-production Vehicles

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ABSTRACT

Technologies have been developed that enable practical application of MBD in function design and performance design stages using the V-model, in order to realize efficient development of mass-production vehicles incorporating complex systems. These technologies, all essential for the application of MBD in mass-production vehicle development, can be broadly divided into three categories: a technology that reduces the time necessary for the formulation of simulation models; a technology that reduces the time necessary for the definition of simulation conditions; and a technology that reduces the time necessary for the sharing of simulation results. These technologies have reduced the simulation output time for MBD in function design and performance design stages to approximately 1/50th of the previously required time. As a result, it has become possible to apply MBD in function design and performance design during the development of mass-production vehicles.

1. Introduction

With the ongoing development of today's mobile society, the automotive market is no longer concentrated in developed nations and is expanding in emerging nations, resulting in considerable automotive diversification. As Fig. 1 shows, an increased focus on environmental issues is propelling diversification of engines, motors, and hybrid powertrains. Changes in lifestyles are diversifying categories, resulting in the development of sedans, minivans, SUV, and other vehicle types in order to reflect customers' needs. With increasing safety requirements, a variety of driving support systems such as SH-AWD and Active Yaw Control are being fitted in vehicles, and devices and electronic systems are diversifying. As a result, automotive systems have rapidly become more complex in recent years⁽¹⁾⁻⁽³⁾.

When developing complex systems, prior to entering into consideration of fine details such as the shapes and characteristics of the components comprising the system, designers examine whether or not the system adequately fulfills its intended functions. In doing so, an effective method is to consider the system mechanisms based on the components making up the general framework, and to study the functions of the system overall^{(4), (5)}. A development process that moves on to detailed study of the constituent components when directions for potential specifications have been obtained at this stage enables efficient creation

of complex systems⁽⁶⁾. This type of process can be generally expressed as the V-model shown in Fig. 2⁽⁷⁾. In this model, the design stages running in sequence from the upstream of the design process are called conceptual design; function design and performance design; and structural design and parts design.

The function design and performance design stages positioned between the upstream conceptual design and the downstream structural design and parts design stages

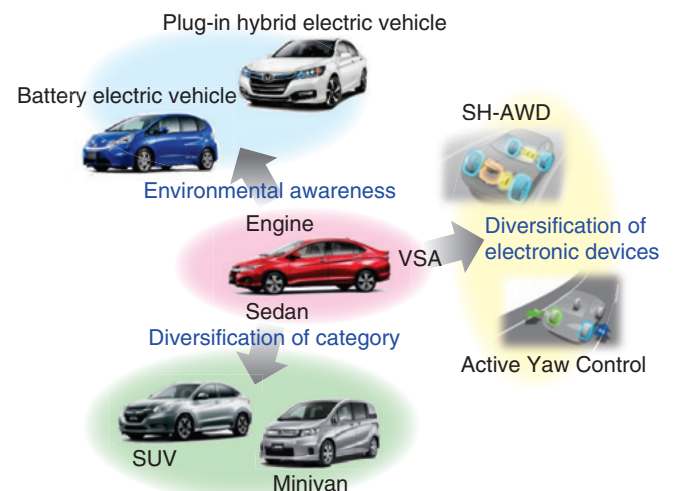


Fig. 1 Automotive diversification

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are important to the efficient and optimal development of complex systems. Model-based development (MBD) methods, involving verification in actual vehicles only after outlooks for potential specifications have been obtained through consideration of specifications at the desk using simulation models, are available for studies at the function design and performance design stages⁽⁸⁾.⁽⁹⁾ An alternative development approach, in which no simulations are conducted and studies commence with the creation of prototypes, and specifications are determined experimentally through repeated trial and error until targets are satisfied, will be termed “model-free development” (MFD) in this paper. Figure 3 expresses each of these development processes as flow charts⁽¹⁰⁾.

The introduction of MBD necessitates the formulation of simulation models (simply termed “models” below) of the vehicle that is the subject of the development. The formulation of the initial model takes time, but when this model has been completed, it is possible to proceed with the study of specifications at the desk, doing away with the necessity of repeating the process of trial and error numerous times using a real vehicle. In addition, because the system mechanisms are clearly incorporated in the model, the effects of solutions employed to realize target

performance are also clear. By contrast, because MFD commences the process of study of the specifications with verification tests in actual vehicles, without using models, the process of trial and error is repeated with the system mechanisms still uncertain, making it challenging to understand the effects of the solutions employed to realize target performance. When engaged in the development of comparatively simple systems, MFD may allow the goal to be reached within a shorter period, but as the complexity of the systems increases, the level of difficulty involved in searching for solutions to realize target performance also increases, and there are cases in which it may not be possible to determine an outlook towards the realization of the targets when MFD is employed.

While the application of MBD using the V-model to the development of vehicles incorporating complex systems is essential, it will be necessary to reduce the time required for function design and performance design in order to apply MBD in the development of mass-production vehicles. To date, there has been no consideration of concrete methods in relation to the time necessary for computational studies using simulations and the degree of reduction in the necessary time for the application of MBD in development for mass production. This paper will discuss research conducted to clarify the time restrictions on function design and performance design, and develop technologies that would make the application of MBD practical, in order to enable application of MBD using the V-model to chassis development for all mass-production models in development.

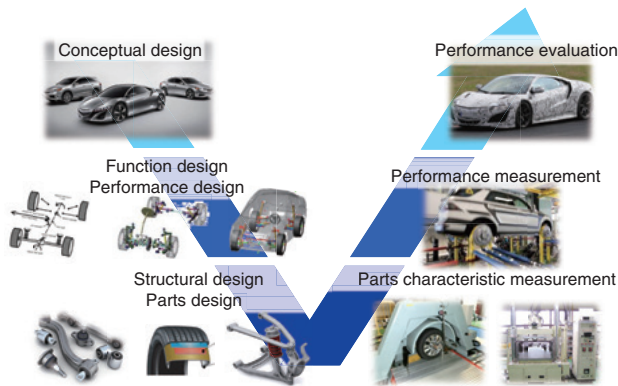


Fig. 2 V-Model for chassis development

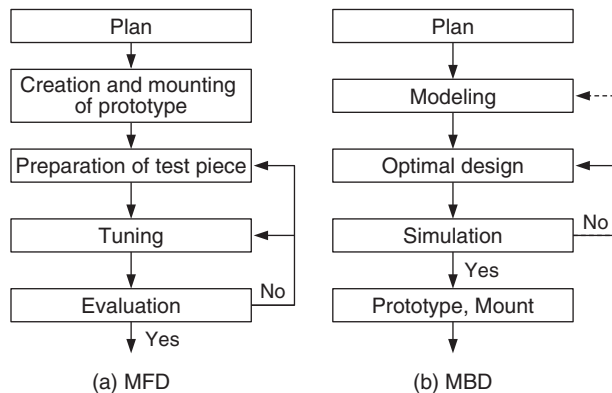


Fig. 3 Concept of MFD and MBD

2. Function Design and Performance Design

1D simulations⁽⁸⁾ play an important role in implementing MBD in automotive function design and performance design. In 1D simulations, the characteristics of the constituent components of the vehicle are represented as blocks, and the reciprocal effects of components on each other are expressed as connections shown by arrows. MATLAB/Simulink is a standard 1D simulation tool. The main focus in terms of function design when formulating models is considering how the vehicle systems should be understood and what is the structural makeup of the mechanisms in the model. When the preliminary model that is initially formulated based on function design is completed, designers determine and consider the optimum combination of component characteristics. This is the main focus in terms of performance design. Even if the details of the dimensions and characteristics of the actual individual components have not been decided, if the overall characteristics of the components have been defined, it is possible to proceed with computation-based studies using 1D simulations⁽¹¹⁾.

When studying whether a specific configuration offers a potential outlook for the achievement of the target functions and performance using 1D simulations, if no viable

direction can be determined at a specific time, then it is necessary to change the configuration of the system. There is a mutually complementary relationship between function design and performance design, and both are essential to the implementation of MBD. Figure 4 diagrammatically illustrates function design and performance design concepts in relation to the roll motion of a vehicle. The top image in Fig. 4 shows a study of an adequate model for the computation of roll motion. The middle and bottom images show studies of the characteristics of the components in this model for the computation of roll motion. Here, the middle image corresponds to function design and the bottom image to performance design. Broadly speaking, function design considers the composition of the system, and performance design considers the combination of component characteristics^{(12), (13)}.

When the component characteristics that will be reflected in the final design drawings are determined, the number of items reaches several hundred. Figure 5 shows part of a table of these items. Here, this will be termed a performance design item list. The vertical columns show items for which system target values exist (simulation output), and the horizontal rows show items representing characteristics of the individual components making up the system (items for parameter studies). Performance design using 1D simulations seeks the optimum solution

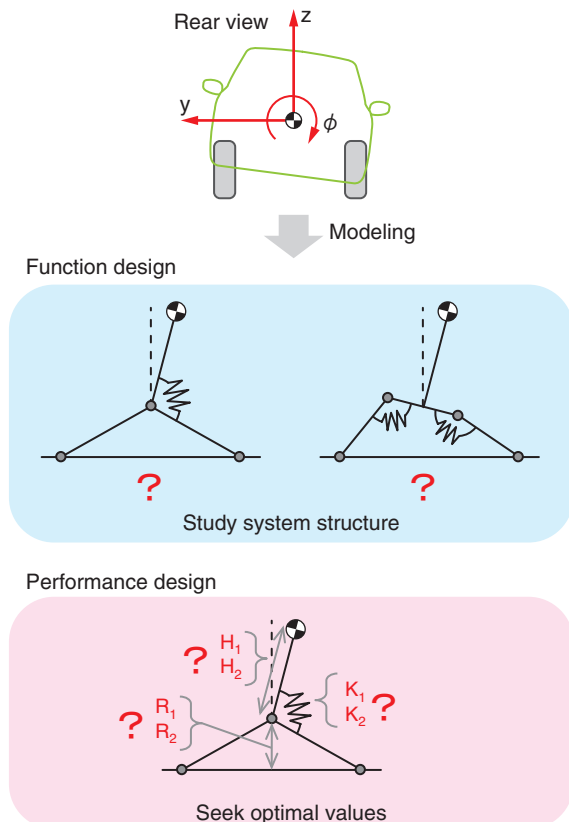


Fig. 4 Function design and performance

from the combination of these several hundred component characteristics.

The time necessary for function design and performance design is the time from the formulation of an adequate model to the sharing of the results of calculation studies with structural designers. In concrete terms, this includes the time necessary for searching for the optimum combination of component characteristics, deciding all of the values to be incorporated in the performance design item list, and outputting a report of calculation results, up to the sharing of data in report briefings and other meetings.

	車両										タイヤ									
	車両諸元(寸法)					車両諸元(重量)					操安関連					乗				
	全長	全幅	ホイールベース	トレッド	車高(含定機込込)	エンジン出力	トランスミッション位置	ピッチ慣性モーメント	ロール慣性モーメント	ヨー慣性モーメント	定軸下重量	ばね下重量	回転部相当重量	SLR	荷重-CAT特性	荷重-スリップ特性	スリップ角-CAT特性	キャンバール角-SAT特性	キャンバール角-SAT特性	タイヤねじりばね
内輪タイヤ中心最小回転半径																				
車体最外側最小回転半径																				
ヨーレート定常ゲイン																				
ヨーレートピークゲイン																				
ヨーレート共振周波数																				
ヨーレート減衰レベル																				
ヨーレート位相遅れ@0.5Hz																				
ヨーレート位相遅れ@1.0Hz																				
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縦Gピークゲイン																				
縦G共振周波数																				
縦G減衰レベル																				
縦G位相遅れ@0.5Hz																				
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ロールレートピークゲイン																				
ロールレート共振周波数																				
ロールレート減衰レベル																				
ロールレート位相遅れ																				
ロールレート位相遅れ																				
相対ヨーレート																				

Fig. 5 Performance design item list

3. Technological Issues

In order to determine specifications in mass-production vehicle developments, various prototypes are formulated for a specific range of characteristics, and these prototypes are combined in a process of trial and error, with the optimum specifications determined experimentally. Based on a survey of the time expended on the process of trial and error in MFD conducted between 2010 and 2011, it was estimated that the average time per trial run until a judgment was made was around six hours.

Generating results from simulation computations in a shorter time than the time per trial run in MFD would be an effective means of realizing the application of MBD to the development of mass-production vehicles. This would mean producing results from the computations conducted in MBD in less than six hours.

However, as of 2010, function design and performance design for certain specific mass-production vehicles took

a combined total of approximately 300 hours. In order to practically apply function design and performance design using MBD to mass-production vehicles, it would therefore be necessary to reduce the time required to $1/50^{\text{th}}$ of the normal amount. Figure 6 compares the time taken by repeated trial and error in MFD with that until computation results are obtained in MBD as the height of bar graphs.

What should be borne in mind here is that the process of trial and error in MFD is in fact not concluded with one trial; the process is repeated numerous times, and the more complex the system in development is, the greater the increase in the number of trials.

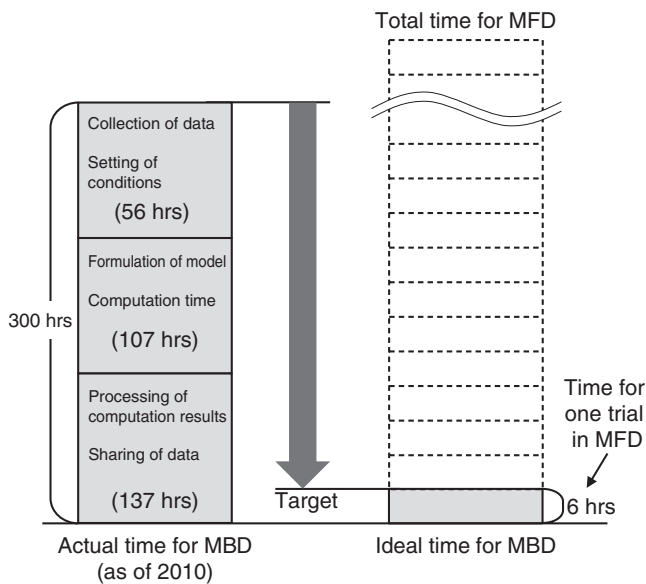


Fig. 6 Target for time reduction

4. Approaches to realizing Practical Application of MBD

4.1. Reducing Time Required for Function Design

The major issue in transitioning from MFD to MBD is reducing the time required for both function design and performance design. The discussion here will first consider the approach to reducing the time required for function design.

When studying a diverse range of characteristic values while seeking the optimum specifications to realize the target values for a complex system, the approach is to begin by studying the most important characteristic values for the system using a simplified model, steadily increasing the detail of the model in order to gradually proceed with the study of the range of characteristic values. Simulations are conducted in a number of stages in this process, but there has been no clear template for the separation of the process into stages, and conventionally individual models have been

formulated for each model in development, with component characteristics individually determined as well. Because of this, every study requires time for the formulation of models. If it were possible to determine the necessary details for studies in each stage and define the model to be used at each stage in advance, it would basically no longer be necessary to expend the time currently employed in formulating models. The research under discussion divided this process into three stages, and established standard models that would allow proceeding with function design for almost any vehicle model in development.

For the first stage, a model was defined that would realize the necessary accuracy in a short period of time. The simulation in this first stage will be termed Level 1 here. As the graph in Fig. 7 shows, the items that can be treated in the simulation increase as the model becomes more detailed, but the labor involved in seeking solutions that realize the target performance also increases. The broken arrow in Fig. 7 shows Level 1. While calculation proceeds in a short period of time, a relatively large number of items can be treated in this model. Because the Level 1 model is a simple block diagram of the type shown in Fig. 8, it can also be easily expressed as a numerical formula. Calculations can therefore be conducted using spreadsheet software such as Microsoft Excel, making this model ideal for rapidly proceeding with parameter studies⁽¹⁴⁾⁻⁽¹⁶⁾.

For the second stage, a model was defined incorporating all of the minimum number of necessary elements for the calculation of all the items in the performance design item list. Here, this simulation will be termed Level 2. The Level 2 model is a block diagram of the type shown in Fig. 9. Because it is more challenging to express this model

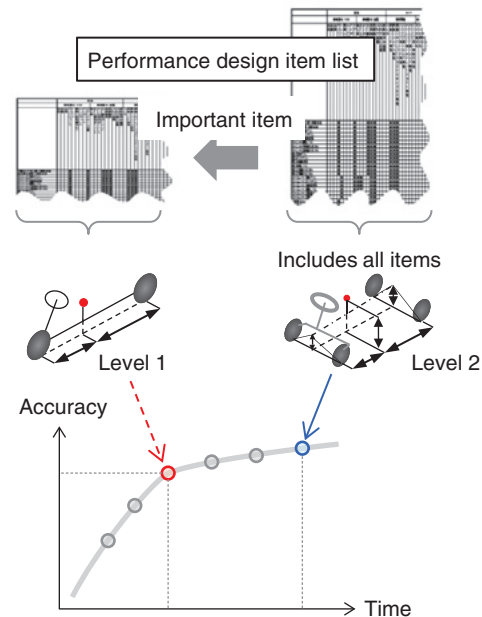


Fig. 7 Relationship between detail of model and output speed

exclusively in mathematical formulas, it is not suited to studies using tools like spreadsheet software. The use of specialized 1D simulation software such as MATLAB/Simulink would be more effective⁽¹⁷⁾.

The model in the third stage is a multibody dynamics analysis model that models all of the components making up the chassis. This type of model is generally called a full vehicle simulation. Here, this simulation will be termed Level 3. Figure 10 shows an image of an animation of a simulation model conducted using ADAMS. The Level 3

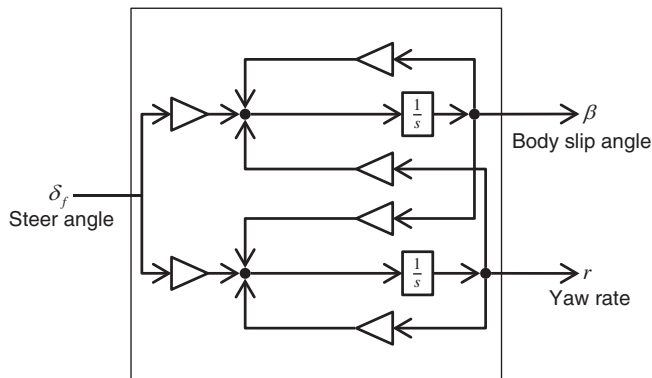


Fig. 8 Model for Level 1

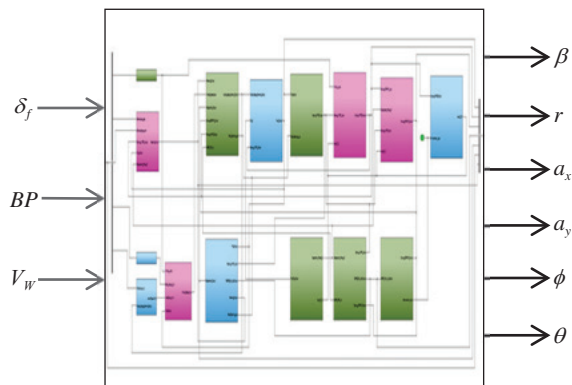


Fig. 9 Model for Level 2

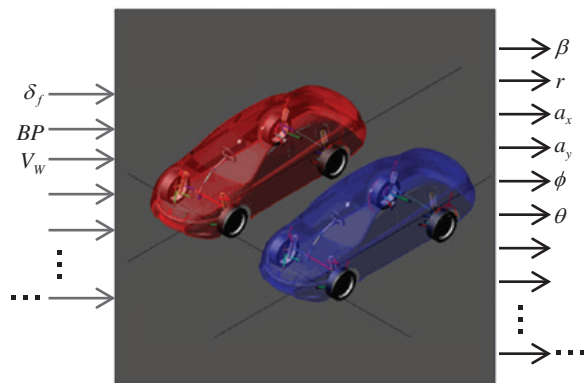


Fig. 10 Model for Level 3

simulation is a 3D simulation rather than a 1D simulation⁽¹⁸⁾.

As of 2016, if designers have models from Level 1 to Level 3, it is possible to study dynamic performance for almost all vehicle models in development. It may be considered that it would be possible to apply MBD to the development of mass-production vehicles using Level 3 (the level at which the most detailed computations can be conducted) alone, but in fact the use of Level 3 only would not allow the achievement of the intended goals. This is because conducting performance design at Level 3 without Levels 1 and 2 would take more time; simply setting the computation conditions for Level 3 would require more time than the target time⁽¹⁹⁾.

4.2. Reducing Time Required for Performance Design

The approach to reducing the time necessary for performance design will next be discussed. In order to reduce the time expended on performance design, the research focused on the time from the setting of specific conditions to the sharing of simulation results with relevant parties, known as output time. In addition to the formulation of the computation model, time is expended in MBD chassis performance design on collecting data and setting computation conditions, and on processing computation results and sharing reports. Figure 11 shows a breakdown of output time.

Even if computation time were to be reduced, the time required for setting computation conditions and defining input items would weaken the effect in reducing output time. In addition, when results are output, if the compilation of those results into reports and the sharing of those reports with other parties takes time, the effect is the same. It was apparent that a great deal of time other than computation time was expended in the entire process, and that this was a factor making the reduction of the time necessary for performance design challenging. The time required for the development of a specific model was measured, and the results are presented as a graph in Fig. 6. A survey of this graph shows that even if the time necessary for the formulation of the model were to be reduced, it would not go far towards the realization of the target reduction.

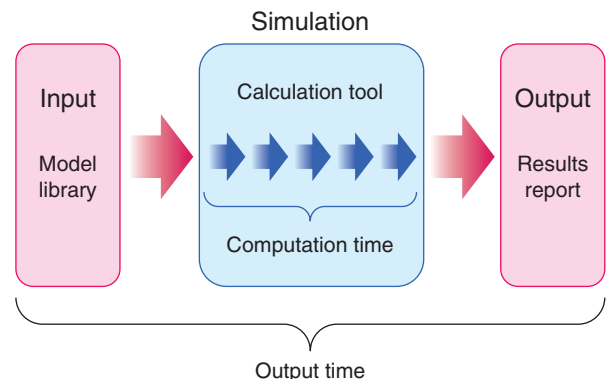


Fig. 11 Computation time and output time

Performance design can be broadly divided into a first half and a second half. The reduction of the time expended in the first half of the process will initially be discussed. The previous section discussed the definition of three models in order to reduce the time expended on function design; it was possible to unambiguously determine the input and output of the simulations using these models. Because input and output would always be the same if the same models were always used, it would be a simple matter to automate the processes related to input and output if the same data format were employed. It would be possible to program the simulation tool in advance to help ensure that input and output data matched the format of the performance design item list, and that Level 1 and Level 2 data were formulated. The creation of a model library is an example of this type of process. Here, “model library” is a general term for a database that stores simulation calculation conditions and a conversion system that enables the simulation tool to directly import the data. The development of a technology that automatically selects simulation conditions from the model library and automatically inputs them to the simulation tool reduced the time necessary for collecting data and setting calculation conditions, which had previously been performed by human operators. Figure 12 compares the time required for input procedures in 2010 and 2014 as bar graphs. These graphs show that the developed technology has reduced the time necessary for the first half of the performance design process to 1/30th of that previously required.

A further issue in reducing the time necessary for performance design was the reduction of the time needed for the compilation of the simulation results into reports and the sharing of the data, corresponding to the second half of the procedure. Conventionally, results have been obtained from computations using a different model each time. These results have then been compiled into a variety of materials and individual reports have been made in order to share data.

If it were possible to output computation results from the Level 1 and Level 2 models in accordance with the items on the performance design item list, it would be a simple matter to automatically formulate the data as graphs or in another desired format. This would reduce the time required for the compilation of reports. The authors defined these automatically formulated reports as part of the performance specifications, and developed an environment that would make it possible to share them in a database. Figure 13 shows an extract of part of the performance specifications for a vehicle. Shared management of performance specifications in a database has reduced the labor involved in sharing the details of studies and the results of calculations, reducing the time needed for these procedures. Figure 14 shows a breakdown of these reductions. The time required for the processing of the calculation results and the sharing of the report, corresponding to the second half of the performance design process, has been reduced to around 1/70th of the previous time.

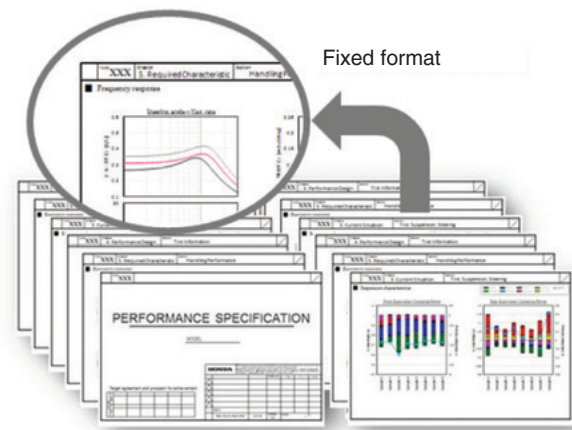


Fig. 13 Performance specifications

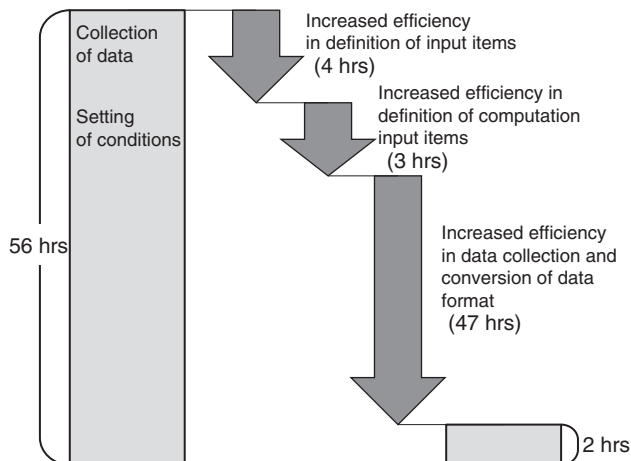


Fig. 12 Enhanced input efficiency

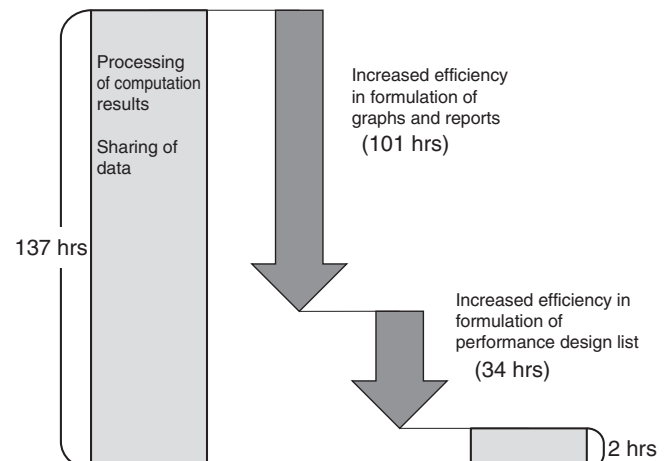


Fig. 14 Enhanced output efficiency

5. Examples of Implementation of Performance Design

5.1. Studies towards Increased Dynamic Performance in S660

The concept for the S660 was to create a vehicle displaying agile handling, making it possible for users to enjoy full-fledged sports performance in a lightweight car. In order to underline this concept, the main points of focus were responsiveness to initial steering input and turning performance when the wheel was turned, and dynamic performance targets were set for handling. At the same time, it was also necessary to realize adequate stability for a sports car. Targets for stability at high speeds were therefore also set. It was necessary to realize all of these targets simultaneously. Specific performance design target values were set as follows:

- (1) Stability at high speeds: Realization of a stability factor of at least $0.001 \text{ s}^2/\text{m}^2$ and a handling capacity of at least 220 m/s^2 at 100 km/h ⁽²⁰⁾⁻⁽²⁴⁾.
- (2) Agile handling: Realization of a maximum lateral acceleration of at least 10.4 m/s^2 and a maximum rear axle slip angle of 10 deg or less during a 100 km/h J-turn^{(20), (22)-(24)}.

The major issue in the case of performance design for the S660 was efficiently determining the specifications that would simultaneously realize both of these targets from among an enormous number of potential combinations.

It was necessary to develop the S660 in a short period of time, but because a large number of components necessitated specialized design, the degree of freedom in design was high, and an enormous number of combinations of individual components was therefore available in seeking to realize target performance. MBD was employed from the initial stages of the development process. Simulation studies of some tens of combinations of specifications were conducted in a short period of time, and specifications that were considered optimal from the performance design perspective were finally proposed. The two graphs in Fig. 15 show an example of the allocation of specifications in order to realize target performance. The graphs show only the most representative indicators⁽²⁴⁾. Like the S660, in the case of the development of vehicles that differ considerably from other mass-production vehicles in terms of factors such as basic specifications and drive type, it is challenging to obtain basic values because little data exists to provide a reference for characteristic values or property values of detailed sections. Given this, if the process of seeking optimal specifications from among countless combinations of component characteristics while simultaneously studying large numbers of target values had been undertaken experimentally, using trial and error, it is probably unlikely that specifications making it possible to realize the targets would have been determined within the development period for this model. The developed method made it possible to proceed with the study of different specifications while

simultaneously verifying large numbers of performance demands from the beginning of the development process for the S660⁽²⁵⁾.

5.2. Balancing Emergency Handling Performance and Ride Comfort in the CLARITY FUEL CELL

A vehicle provided with advanced technologies, the CLARITY FUEL CELL uses fuel cells to generate electricity and drive a motor. Another target was to realize an extremely quiet and comfortable vehicle. In addition to this, the goal was to create a package that would reconcile a practical layout for each component with design features. This represented an unfavorable condition in relation to reducing the vehicle's moment of inertia. It is also necessary to employ a range of devices in FCV, increasing the weight of the vehicles against gasoline vehicles of the

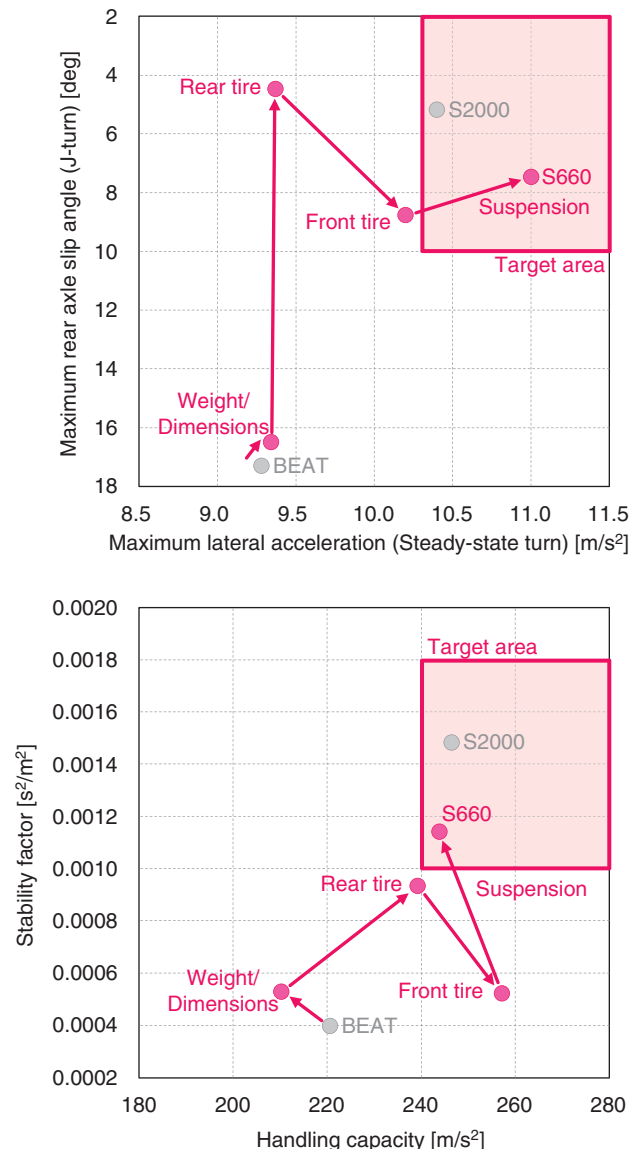


Fig. 15 Performance design for S660

same class. In other words, developers were presented with the task of realizing equivalent dynamic performance to a normal vehicle of the same class in a heavy FCV with a high moment of inertia. Because the battery stack that supplements the fuel cell stack and the drive motor were particularly heavy, the layout of these components would have a particularly great effect on dynamic performance, and would be the key to balancing ride comfort with quietness. The performance design target values for dynamic performance set under these various restrictions were a delay time in lateral acceleration response during a double lane change (DLC) of no more than 0.15 sec in 1st and 0.175 sec in 2nd. Figure 16 shows delay time in lateral acceleration response during a DLC.

A countless number of combinations of tire, suspension, and other component characteristics could be considered towards the realization of the targets discussed above, and seeking specifications that balance ride comfort and quietness while also satisfying the target packaging through trial and error would be no easy task. Because the layout of the stack and the motor and the stiffness of the mounts would result in major changes in ride comfort and quietness, it was essential to conduct studies of dynamic performance while also taking these factors into consideration. Using the developed method, it was possible to conduct computations for some tens of combinations in a short period of time while considering the balance with performance demands other than dynamic performance. Figure 17 shows an example of the results of one study, demonstrating that the developed method is effective even in developments of this type, in which the development proceeds in tandem with studies attempting to realize a balance with performance demands other than dynamic performance and with the demanded specifications⁽²⁵⁾.

5.3. Studies towards the Realization of Increased Stable Handling Potential in the Linear Region for the 2014 model year FIT

As a global model, the 2014 model year FIT (2014 FIT) would be marketed throughout the world, and the aim for

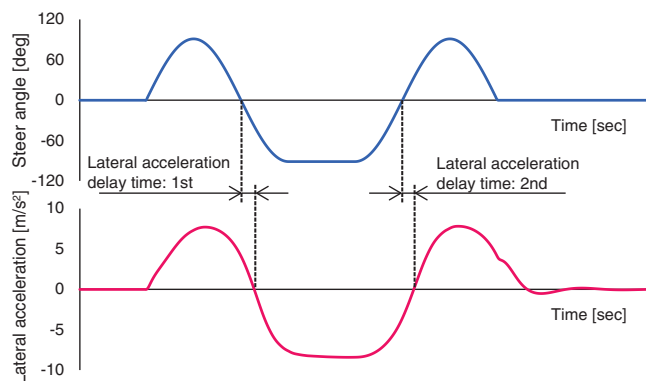


Fig. 16 Lateral acceleration during DLC

the development was therefore to simultaneously realize specifications for all global variations. It was therefore necessary to seek combinations within a short period of time that would satisfy target performance demands from a diverse range of specifications that differed for each region in which the vehicle would be sold. It had also been decided that a wide range of derivative models, including sedans, wagons, SUV, and minivans, would also be developed using the platform developed for the 2014 FIT. It was thus essential to boost the performance potential of the vehicle from the initial stages of development in order to respond to derivative models with a wide variety of shapes. MBD using the developed method was applied to the development, and because it was possible to refer to data for the 2008 model year FIT (2008 FIT), calculation conditions could be formulated and target performance set rapidly for the 2014 FIT. The 2014 FIT was the first model to be developed with full-scale application of performance design using MBD to the study of stable handling in the linear region.

Because MBD was employed in studies for the 2014 FIT, it was possible to conduct calculations for all specifications for all global models, and to verify the potential of all specifications to realize target values using simulations. In other words, it was possible to verify the potential of the base vehicle in relation to derivative models without tests in actual vehicles.

The ability to practically apply MBD in the development of a mass-production vehicle made it possible to also simultaneously establish potential outlooks in relation to the performance of derivative models being developed in parallel. For example, studies were conducted of the performance of the suspension in a state of rebound in order to help ensure high road clearance for the SUV model; studies were conducted in order to further increase ride

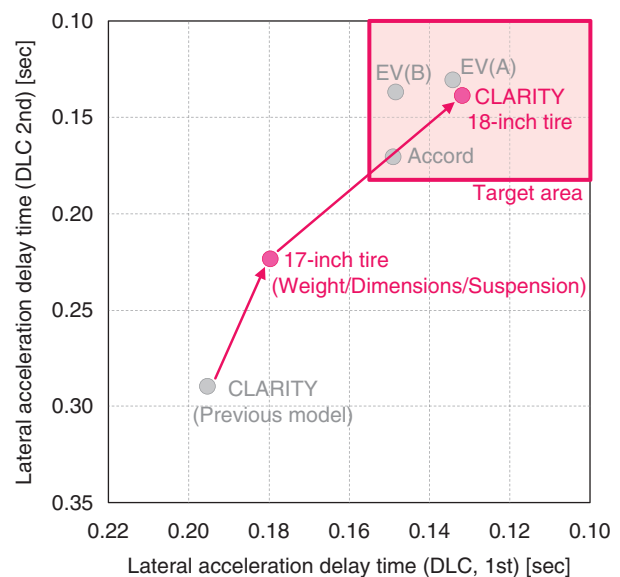


Fig. 17 Performance design of 2016 model year FCV

comfort in the sedan model; and, during the development of the minivan model, studies were conducted of the dimensional specifications providing the maximum overhang for the FIT platform, seeking the optimal specifications in relation to target values for dynamic performance. Figure 18 shows a diagram of computation studies conducted for the 2014 FIT, including derivative models. The use of the developed technologies made it possible to conduct the large number of studies shown here within a short period of time.

5.4. Studies towards the Realization of Increased Handling Potential in the Limit Region for the 2017 model year CR-V

The major issue in terms of dynamic performance in the development of the 2017 model year CR-V (2017 CR-V) was to realize stable handling in the limit region for all specifications⁽²⁶⁾. A wide range of variations were set for the 2017 CR-V, encompassing a range of powerplants (including turbo, diesel, and hybrid), 4WD and FWD drive types, two- and three-seat seating layouts, 17-inch, 18-inch and other tire sizes, and a panorama roof option. A large number of potential specifications therefore existed, all of which were predicted to have a major effect on dynamic performance. Using MFD to conduct studies towards the realization of target performance in relation to all of these specifications would not have been easy. For the performance design of the 2017 CR-V, it was necessary to conduct studies not only towards the realization of stable handling in the linear region, but also stable handling in the limit region, within a short period of time.

Because data for the preceding 2012 model year CR-V (2012 CR-V) were available in the case of the 2017 CR-

V, it was possible to rapidly set calculation conditions and performance design target values. By contrast with the development of the 2012 CR-V, simulation studies proceeded using models defined for the three levels at basically the initial stage of development, allowing developers to proceed with sufficient consideration not only of efficiency, but also of accuracy. Figures 19⁽²¹⁾ and 20⁽²⁸⁾ show simulation results obtained at the time. Here, steer angle amplitude expresses the rate of multiplication of the steering wheel angle (defined as 1[A]) when turning

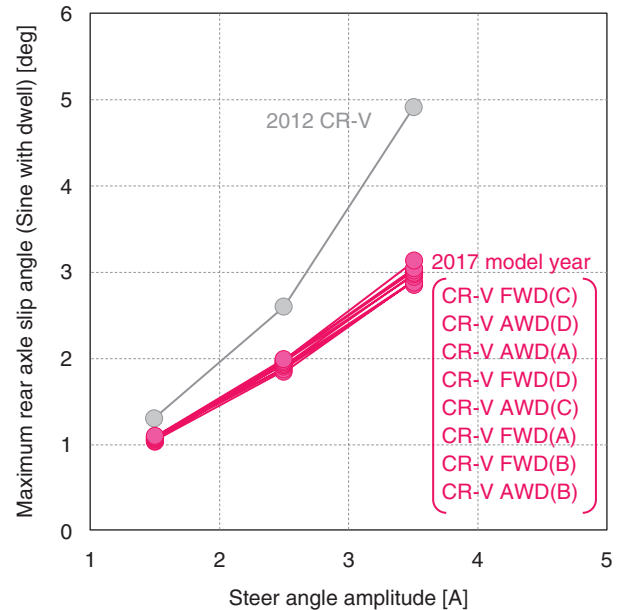


Fig. 19 Study of emergency handling

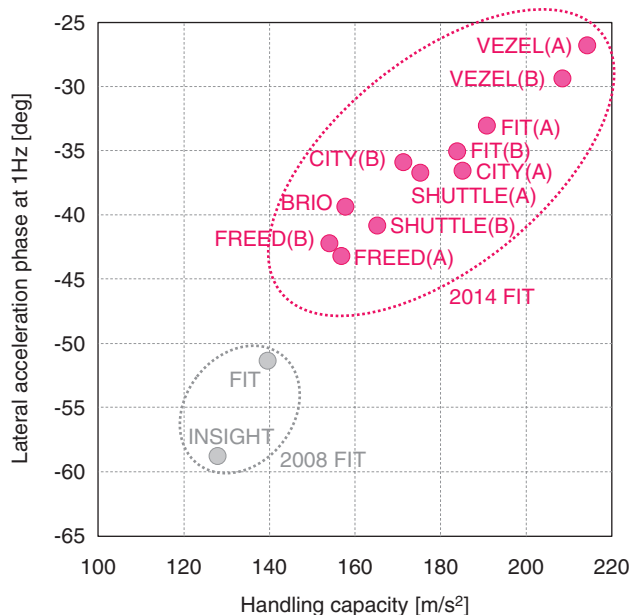


Fig. 18 Performance design of 2014 FIT

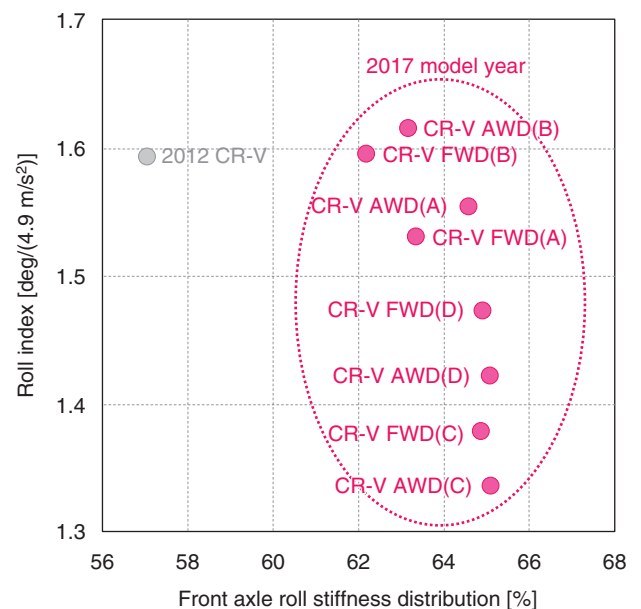


Fig. 20 Performance design for 2017 CR-V

in a fixed circle at a lateral acceleration of 2.9 m/s^2 ⁽⁵⁾. By contrast with the previous model, when only the results of studies of representative models were available for theoretical study, for the 2017 CR-V, simulation studies were conducted for all of the large number of specifications. During the development of the 2017 CR-V, it was demonstrated that performance design using MBD could be applied to a wide range of studies, extending even to nonlinear elements, in this case the tires and bump rubber bushes⁽²⁷⁾.

The four examples discussed above demonstrate, based on actual results, that the developed technologies can be applied to a wide range of elements within the chassis development process for mass-production vehicles.

6. Conclusion

The simulation output time for MBD in function design and performance design stages was reduced to around 1/50th of the previously required time, making it possible to apply MBD to the development of mass-production vehicles. The research process can be summarized as follows:

- (1) Standard vehicle simulation models for performance design were defined for three levels.
- (2) Defining the necessary input items for simulations increased the efficiency of procedures, reducing the time required for the first half of the performance design process to approximately 1/30th of that previously needed.
- (3) Defining performance specifications as the format for simulation results and creating a database of performance specifications reduced the time expended up to the sharing of results, corresponding to the second half of the performance design process, to approximately 1/70th of that previously needed.
- (4) The time required for function design and performance design using MBD has been reduced to approximately 1/50th of that previously needed.

Afterword

The development of the method discussed in this paper has enabled practical application of MBD in the chassis development process for mass-production vehicles. The authors hope in future to apply MBD to the full range of models using this method, and to expand the scope of function design and performance design methods using MBD to areas including dynamic performance in the driving/braking region and passenger comfort, seeking to realize a highly efficient automotive development process.

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Broadly-applicable Adaptive Damper System without Use of Dedicated Sensors

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ABSTRACT

An adaptive damper system that does not require the use of dedicated sensors has been developed. Employing the proportional relationship between front wheel speed and contact load, which change with input from the road surface, the system estimates the vehicle's vertical vibration state quantity using only the wheel speed, longitudinal acceleration, and yaw rate signals conventionally employed for systems such as Vehicle Stability Assist and airbag systems in Supplemental Restraint System. This does away with the necessity for the use of sensors for the measurement of the vertical acceleration of sprung mass and suspension stroke sensors. Using the estimated values, the system boosts ride comfort through the application of damping force control by adaptive dampers in order to control vertical vibration. In addition, the system also offers unsprung mass control and pitch and roll control in order to boost road-holding performance and handling and stability performance. The adaptive damper units feature a triple-tube configuration with externally mounted pressure-control solenoid valves, making it possible to set the damping force characteristic and the range of variation in damping force to match the weight, size, and concept of different vehicles. The developed system is capable of application to a wide range of vehicle categories.

1. Introduction

A variety of electronically controlled adaptive damper systems have been proposed in efforts to reconcile handling and stability and ride comfort. More recently, control systems incorporating real-time control logics that feed back vehicle state data have been proposed^{(1), (2)}. Honda employs an electronically controlled adaptive damper system that it terms the Adaptive Damper System, or ADS, in mass-production vehicles. The ADS uses a control logic based on skyhook control in order to balance the isolation from road surface input obtained by minimizing the damping force of the dampers with vibration suppression performance in relation to sprung mass resonance. This control logic is widely known as a method of controlling vertical vibration of the vehicle body in order to increase ride comfort. It is necessary for the purposes of this control to know the vertical velocity of the sprung mass and the damper stroke velocity. To enable calculation of these parameters, the conventional ADS system configuration has featured three dedicated sensors to detect the vertical acceleration of the sprung mass and suspension stroke sensors positioned on each wheel, as shown in Fig. 1. Because this large number

of sensors and the required wiring has increased the overall cost of the system, the application of ADS has been limited to luxury vehicles. In order to deliver the luxurious ride comfort realized by ADS to a greater number of users, it will be necessary to reduce the cost of the system and promote its application in a wider range of vehicle categories.

In order to realize this goal, methods of applying skyhook control without the use of sensors for the measurement of the vertical acceleration of sprung mass and suspension stroke sensors, the dedicated sensors employed in the ADS, have been proposed^{(3), (4)}. These methods make active use of wheel speed signals⁽⁵⁾⁻⁽⁷⁾, which fluctuate with road surface input and the vehicle's dynamic state during operation. Obtaining signals from wheel speed sensors already fitted as part of the Vehicle Stability Assist (VSA) system via a Controller Area Network (CAN), the methods estimate the vehicle's vertical vibration from changes in wheel speed and damp said vibration through skyhook control.

In order to apply ADS in mass-production vehicles, in addition to increasing ride comfort by damping the sprung mass, it would also be necessary to boost

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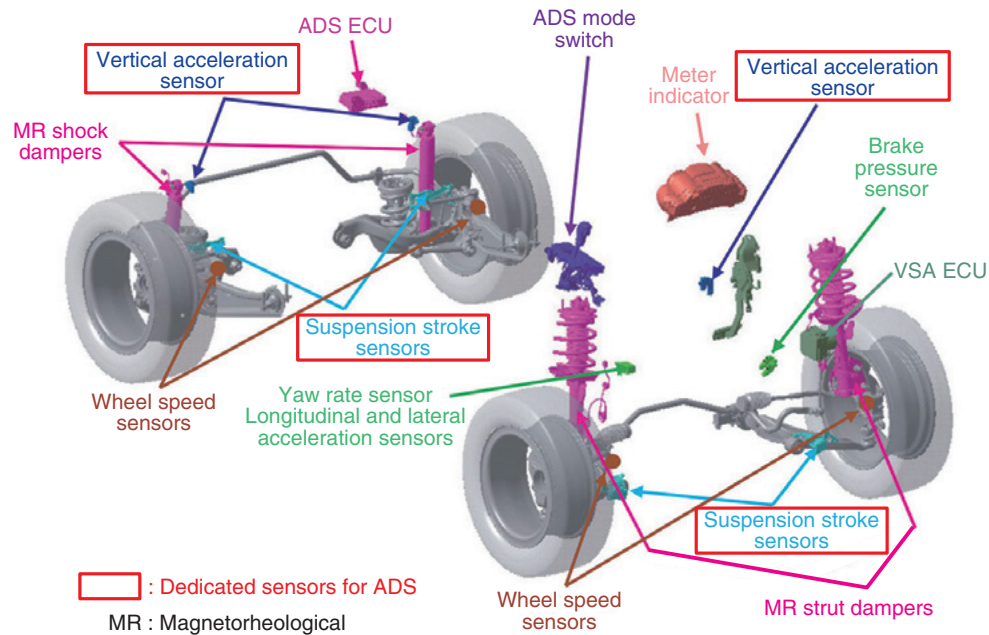


Fig. 1 Components of conventional ADS

road-holding performance and handling and stability performance by controlling posture changes in the pitch and roll directions while damping unsprung mass resonance. It would additionally be necessary to adjust the range of variation in damping force and the damping force characteristic of the adaptive damper units in accordance with the weight, size, and concept of a wide range of vehicle categories, from small cars to large vehicles such as SUV. The development process discussed in this paper created a system based on these considerations.

2. Development Aims

Looking toward the application of ADS in a wide range of mass-production vehicles, the following development aims were set in order to realize a system displaying a similar level of body damping performance to the conventional system while controlling system cost, and to enable adjustment of the damping force characteristic of the adaptive damper units in accordance with the weight, size, and concept of a range of vehicles.

- (1) In addition to a skyhook control using only signals already employed by other systems, the realization of a control that suppresses unsprung mass resonance and changes in pitch and roll behavior in order to increase road-holding and handling and stability performance, while also maintaining ride comfort.
- (2) Development of adaptive damper units in which the range of variation in damping force and the damping force characteristic can be adjusted for application in a wide range of vehicle categories.

3. Control Logic

3.1. Overview of Control Logic

This section will discuss the control logic employed in realizing the vehicle fitting and mass production of the system. ADS employs a control logic based on skyhook control in order to balance isolation from road surface input with the damping of sprung mass resonance. In addition, road-holding performance is increased by suppressing unsprung mass resonance, and handling and stability performance is increased by controlling changes in pitch and roll behavior resulting from acceleration and deceleration by the driver and steering input such as lane changes.

Figure 2 shows a block diagram of the control logic incorporated in an actual vehicle. As input signals to the control logic, vehicle speed at each wheel, longitudinal acceleration, lateral acceleration, yaw rate, and steering angle signals are supplied to the system by other systems [VSA, Supplemental Restraint System (SRS), and Electric Power Steering (EPS)] via a CAN.

The state quantity estimation part calculates the vertical velocity of the sprung mass and the damper stroke velocity necessary for the application of skyhook control based on changes in wheel speed^{(3), (4)}. However, because in an actual driving environment changes in wheel speed during vehicle operation can be caused by factors including changes in vehicle speed and changes in the turning radius in addition to changes in the contact load, it is necessary to conduct pre-processing in order to remove these factors from the wheel speed signal in advance. In the wheel-speed-signal preprocessing block, processing is applied using the

longitudinal acceleration signal and the yaw rate signal in order to remove changes in wheel speed due to acceleration, deceleration, and operation of the steering wheel from the front wheel speed signal. Following this, changes in the contact load are calculated from the preprocessed wheel speed signal and input to a quarter-car model for estimation (to be explained in the next section), making it possible to estimate the sprung mass velocity and damper stroke velocity for each wheel, which are essential parameters for the application of skyhook control.

The following four independent control blocks, each possessing a different function, are provided in the body control logic section; these control blocks use the estimated state quantities and other input signals in order to control changes in vehicle posture involving three degrees of freedom of the sprung mass of the vehicle body (vertical, pitch, and roll) and to control unsprung mass resonance.

- (1) Skyhook control using estimated state quantity values controls sprung mass resonance while isolating the vehicle body from vibration, boosting ride comfort
- (2) Pitch control using the longitudinal acceleration signal controls pitch behavior during acceleration and deceleration
- (3) Roll control using the lateral acceleration signal and the steering angle signal controls roll behavior during operation of the steering wheel
- (4) Unsprung mass control using the wheel speed signal controls unsprung mass resonance and boosts the road-holding performance of the tires

Pitch and roll control are determined on the basis of changes in the longitudinal and lateral acceleration signals, and unsprung mass control is determined on the basis of changes in the wheel speed signal. Output is restricted to a frequency range in which any effect on ride comfort will be minimized. The system's control output is set such that the level of control applied by each control block is adjusted in relation to vehicle behavior, and the maximum value for the

output of each control block that will help ensure that there is no shortfall in damping force is selected.

3.2. Method of Estimation of Front Axle State Quantity

In order to apply skyhook control without the use of dedicated sensors, the system estimates the vertical velocity of the sprung mass and the damper stroke velocity, the essential parameters for control, from the wheel speed signal and other existing CAN signals. This section will discuss the method of estimation of the state quantity of vertical vibration using the proportional relationship between changes in wheel speed and changes in contact load.

In order to consider the effect of deflection of the tires due to fluctuation in the contact load as a result of road surface input, the relationship between vehicle speed and wheel speed in a vehicle traveling at a constant speed will be considered. With the vehicle speed as V and the tire radius as R , as shown in Fig. 3, and the standard tire radius during conversion from the angular velocity of the wheel to vehicle speed as R_0 ($= \text{constant}$), the vehicle speed calculated from the wheel speed sensor, V_0 , is determined by Eq. (1)^{(4), (7)}.

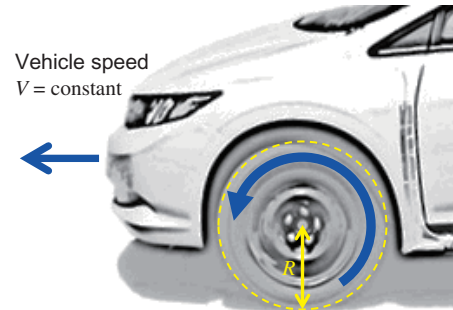


Fig. 3 Vehicle speed and wheel speed

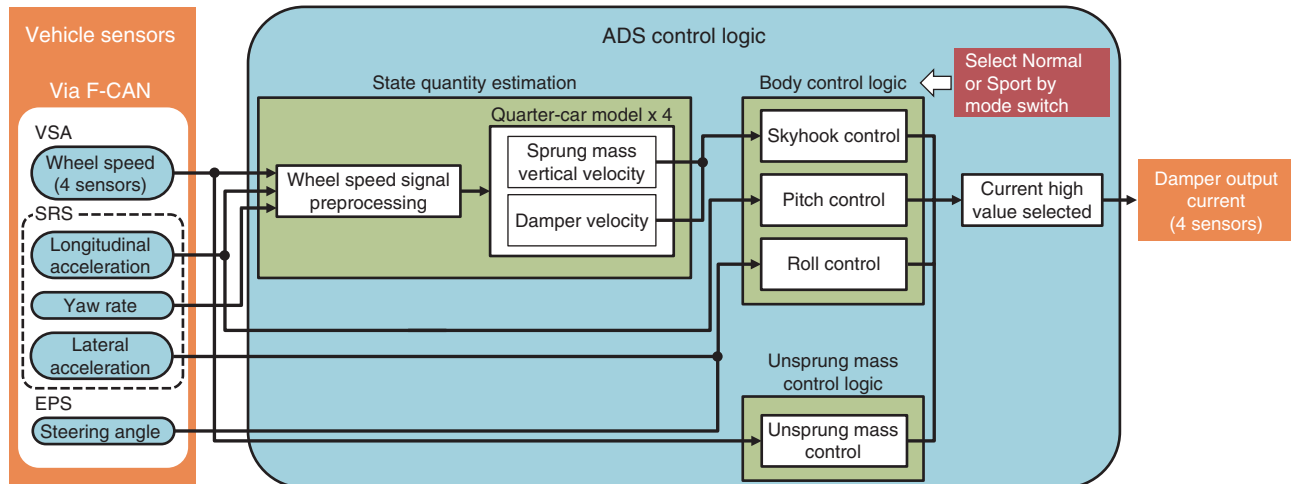


Fig. 2 Control logic block diagram

$$V_0 = \frac{R_0}{R} V \quad (1)$$

Considering the case in which, due to road surface input, R becomes ΔR lower than R_0 and V_0 becomes ΔV greater than V , Eq. (1) can be transformed into Eq. (2).

$$V + \Delta V = \frac{R_0}{R_0 - \Delta R} V \quad (2)$$

If, following Maclaurin expansion with consideration that $R_0 \gg \Delta R$, the terms from the second term onwards are ignored as infinitesimal in order to simplify the expression of the relationship between ΔV and ΔR , Eq. (3) is generated.

$$\Delta V = \frac{\Delta R}{R_0} V \quad (3)$$

Expressing the change in contact load, ΔF , in terms of ΔR and the vertical spring constant of the tires, K_1 , gives Eq. (4).

$$\Delta F = K_1 \Delta R \quad (4)$$

Here, the effect of the tire damping term is sufficiently low to be ignored. Substitution of the relationship shown in Eq. (3) into Eq. (4) gives Eq. (5).

$$\Delta F = \frac{R_0 K_1}{V} \Delta V \quad (5)$$

Because R_0 , K_1 , and V in Eq. (5) are constants, the change in the contact load, ΔF , is proportional to the change in wheel speed, ΔV .

Figure 4 shows the relationship between change in wheel speed and change in contact load for a front wheel, measured using a six-component force meter on the wheel of a vehicle proceeding straight ahead at 100 km/h on an undulating road. Signal processing was applied using a band pass filter to isolate the respective sprung mass resonance frequency bands in the wheel speed signal and



Fig. 4 Relationship between change in wheel speed and change in tire contact load for front wheel

the contact load signal. As Fig. 4 shows, change in the contact load displays a linear relationship with change in wheel speed in specific frequency bands, demonstrating that the relationship shown in Eq. (5) holds true under actual driving conditions⁽⁴⁾.

Having demonstrated that change in the tire contact load can be determined from change in wheel speed, the determination of the vertical vibration state quantity, essential for the application of control, from change in the contact load will next be considered. Inputting the change in the contact load determined from the change in wheel speed, u_{1F} , to the unsprung mass, M_{1F} , in the three-degree-of-freedom single-wheel model shown in Fig. 5, makes it possible to calculate the vertical vibration state quantity, and to determine the sprung mass velocity, $\dot{x}_{2F}$, and the damper stroke velocity, $\dot{x}_{2F} - \dot{x}_{1F}$, at the front wheel.

3.3. Method of Estimation of Rear Axle State Quantity

Figure 6 shows the relationship between change in wheel speed and change in contact load for a rear wheel during driving, with the same signal processing applied under the same conditions as for the front wheel in the preceding section. Because the relationship between the two parameters is not linear, as shown in Fig. 6, it would

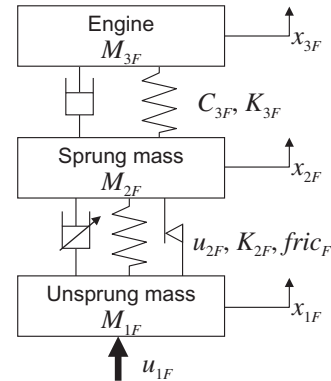


Fig. 5 Quarter-car model for estimation of state quantities for front axle

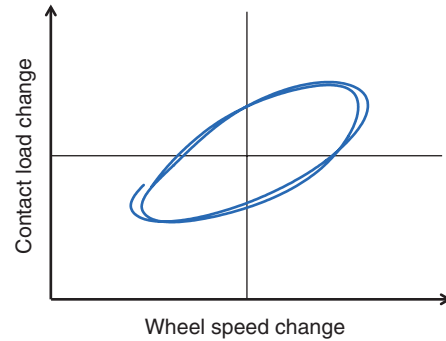


Fig. 6 Relationship between change in wheel speed and change in tire contact load for rear wheel

be challenging to estimate the vertical vibration state quantity for the rear wheels using the method discussed in the preceding section. This phenomenon is observed in suspension systems with a configuration that displaces the tire contact patch longitudinally with vertical displacement of the suspension. This configuration is designed to have an anti-lift effect that mechanically controls lift behavior in the rear wheels when the brakes are applied. For example, as the tire contact patch is displaced to the rear with displacement of the suspension towards the direction of compression with road surface input, the wheels move towards the rear of the vehicle body, and wheel speed is reduced. In the case of suspension systems employing an anti-lift geometry, change in wheel speed due to the longitudinal displacement of the wheels cannot be ignored, and because of this it is not possible to realize sufficient linearity between change in wheel speed and change in contact load⁽⁴⁾.

As indicated above, it is challenging to estimate the contact load from wheel speed for the rear wheels, which often employ anti-lift geometry. Given this, a method that uses the front wheel speed and the estimated front wheel state quantity was proposed⁽⁴⁾. This method determines the vertical vibration state quantity for the rear wheels by employing the road surface displacement estimated at the position of the front wheels for the road surface displacement at the position of the rear wheels, with consideration of the time lag determined by the vehicle's speed and its wheelbase.

Figure 7 shows a block diagram of the principle used in estimation. The change in the contact load calculated from the change in wheel speed, u_{1F} , is input into the front wheel estimation model discussed above (Fig. 5), and the unsprung mass displacement, x_{1F} , is calculated when estimates of the state quantity for front wheel control are conducted. Replacing the tire vertical spring constant with K_{1F} and ΔR with $x_{0F} - x_{1F}$ in Eq. (4), the contact load for the front wheels, u_{1F} ($=\Delta F$), is determined by Eq. (6).

$$u_{1F} = K_{1F} (x_{0F} - x_{1F}) \quad (6)$$

If this equation is solved for road surface displacement at the front wheels, x_{0F} , road surface displacement at the contact patch for a front wheel can be calculated using Eq. (7).

$$x_{0F} = \frac{u_{1F}}{K_{1F}} + x_{1F} \quad (7)$$

If the wheelbase is termed WB , the time for the rear wheels to reach a point of the road surface passed by the front wheels, T_{delay} , can be expressed as Eq. (8).

$$T_{delay} = \frac{WB}{V} \quad (8)$$

Road surface displacement at the rear wheels estimated from the front wheels, x_{0R} , can therefore be expressed as Eq. (9).

$$x_{0R}(t) = x_{0F}(t - T_{delay}) \quad (9)$$

Inputting the estimated road surface displacement when the rear wheel passes, x_{0R} , to a two-degree-of-freedom rear-wheel estimation model makes it possible to calculate the vertical vibration state quantities for the rear wheels, and to determine the sprung mass velocity, $\dot{x}_{2R}$, and the damper stroke velocity, $\dot{x}_{2R} - \dot{x}_{1R}$.

3.4. Estimation Results for Vehicle State Quantity

Because the estimated vehicle state quantities are intended for use in real-time control, the state quantities for the front wheels during actual driving are estimated using real-time processing. Figure 8 shows an example of the results. The graphs show the time series waveforms of the estimated values of the vertical vibration state quantities for a vehicle traveling straight ahead at a speed of 100 km/h on an undulating road. The upper graph shows the front sprung mass velocity, and the lower graph shows the front damper stroke velocity. Results for sprung mass velocity calculated

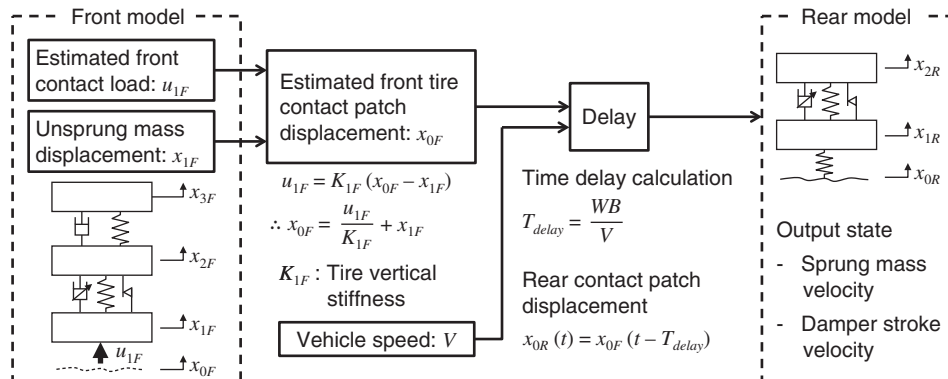


Fig. 7 Conceptual diagram of state quantity estimation for vertical vibration

from sprung mass acceleration sensor output and results for damper stroke velocity calculated from suspension stroke sensor output are superimposed on the estimated results. The results demonstrate that it is possible to estimate state quantities for the front wheel axle using the front wheel speed signal.

Similarly, Fig. 9 shows estimation results for the rear wheel axle. The graphs show that phase lag is reduced in results for the rear state quantities estimated from front wheel speed and estimated values in comparison to results estimated using the rear wheel speed signal.

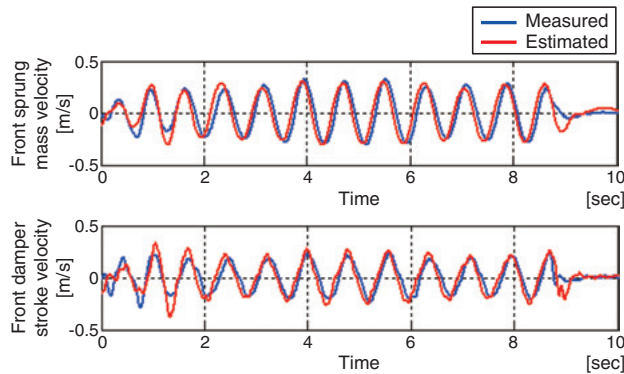


Fig. 8 Comparison of estimated waveforms of vertical vibration state quantity for front axle

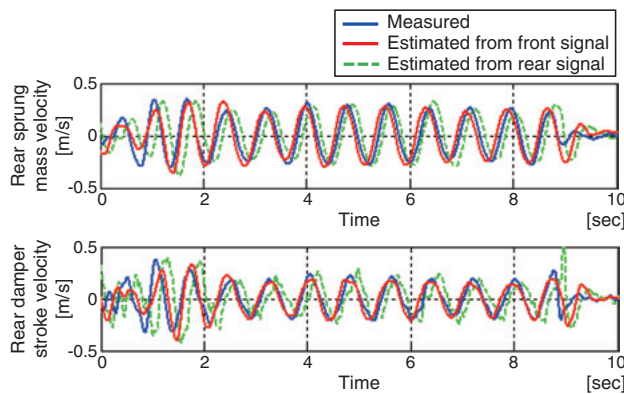


Fig. 9 Comparison of estimated waveforms of vertical vibration state quantity for rear axle

4. System Configuration

4.1. Sensors

Figure 10 shows the configuration of the developed system. Because the system does away with the three sensors for the measurement of the vertical acceleration of sprung mass and the suspension stroke sensors at each wheel employed in the conventional system, it has a simple configuration, with the only parts exclusively for ADS use being the control ECU and the adaptive damper units

positioned at each wheel. This has done away with the weight and the cost of the sensors, sensor brackets, couplers, and wiring. Input signals are provided via the CAN. The wheel speed signal is obtained from the VSA system, the longitudinal and lateral acceleration signals and the yaw rate signal are obtained from the SRS airbag system, and the steering angle signal is obtained from the electric power steering (EPS) system.

4.2. Adaptive Damper Unit

Because the damping force changes with the weight, size, and concept of the vehicle, it was necessary to adjust the damping force characteristic of the adaptive damper units accordingly. In addition, a wide range of variation in damping force would also be necessary in order to make it possible for the effect of damping force control by the adaptive dampers to be displayed in a greater range of scenarios, and it was therefore desirable for damping force to be variable for both the tension and compression strokes of the dampers.

Based on these considerations, adaptive damper units employing replaceable, externally mounted pressure control solenoid valves were employed in the developed system. Because solenoid valves are able to adjust the pressure characteristic in relation to the oil flow rate, replacing the valves makes it possible to change the damping force characteristic of the adaptive dampers.

Figure 11 shows schematic diagrams of the configuration of the triple-tube adaptive dampers. The diagrams show schematic images of the respective oil flow during the tension and compression strokes. As the piston rod moves during the tension stroke, the piston valve closes and oil flows from the upper damper chamber to the solenoid valve via the inner tube. At this time, negative pressure in the lower damper chamber opens the bottom valve, and oil is supplied from the reservoir chamber. As the piston rod strokes during the compression stroke, the bottom valve closes, and increased pressure in the lower damper chamber opens the piston valve, and oil flows into the upper damper chamber. Following this, the flow becomes the same as that during the tension stroke, and oil is supplied to the solenoid valve. This means that oil flows to the solenoid valve from the same direction during both strokes, making it possible to control the damping force during both strokes using a single solenoid valve.

Figure 12 shows an example of a damping force characteristic. The graph shows the load characteristic in relation to piston velocity when currents that produce the minimum and maximum damping force are impressed in the coil. An example of the damping force characteristic of a conventional damper is included for comparison. Figure 12 shows that the current impressed in the coil is able to change the damping force characteristic during both tension and compression strokes. In addition, the adaptive damper damping force characteristic shown in the graph encompasses the damping force characteristic of the conventional damper.

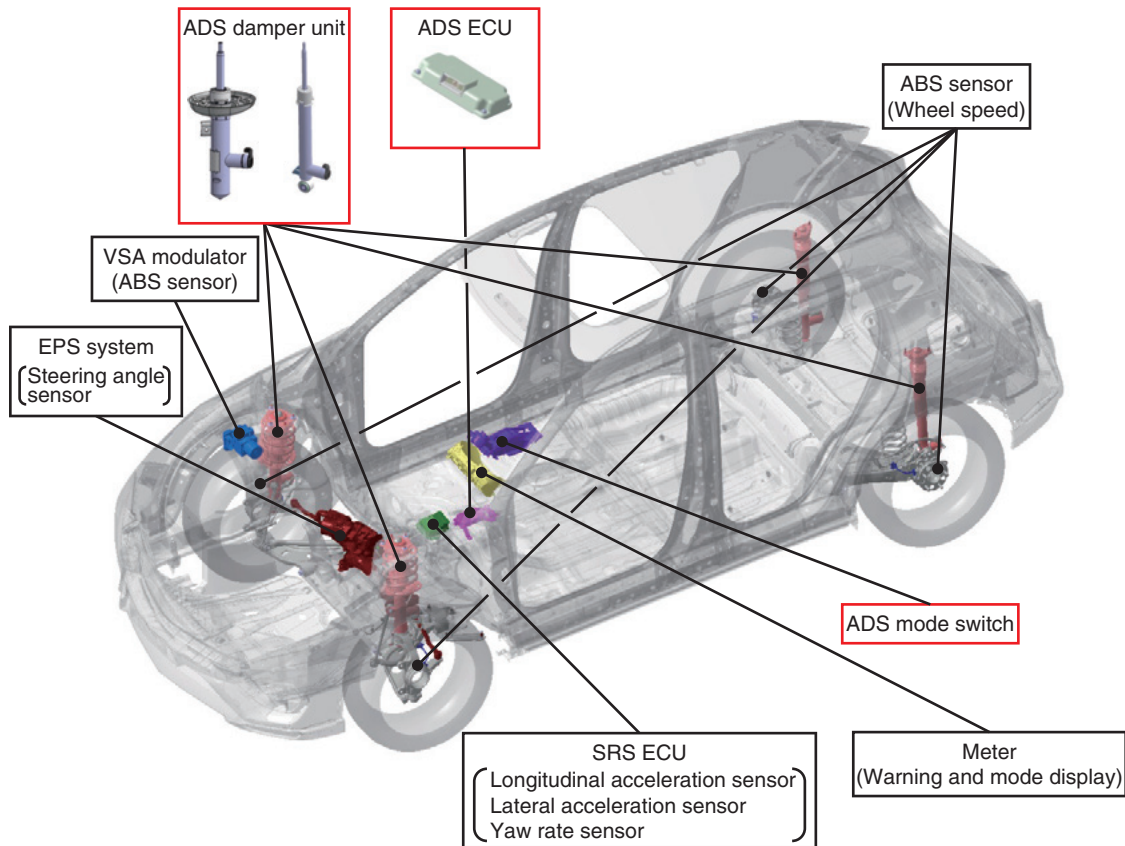


Fig. 10 Components of ADS without dedicated sensors

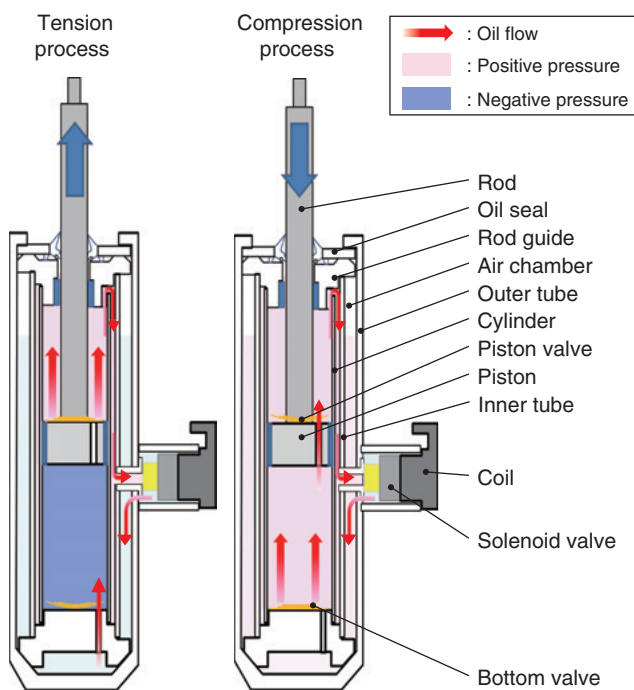


Fig. 11 Triple tube damper configuration

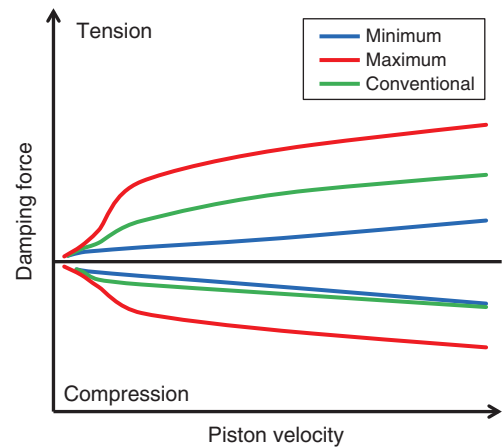


Fig. 12 Typical damper force characteristic

Figure 13 shows an example of the damping force characteristic of an adaptive damper unit adjusted through replacement of the solenoid valve. As the graph shows, it was possible to set the maximum damping force higher and the minimum damping force lower during both the tension and compression strokes.

The damping force characteristic of the adaptive damper unit can be changed by replacing the solenoid valve, enabling adjustment of the characteristic in accordance with the weight, size, and concept of vehicles in a wide range of categories.

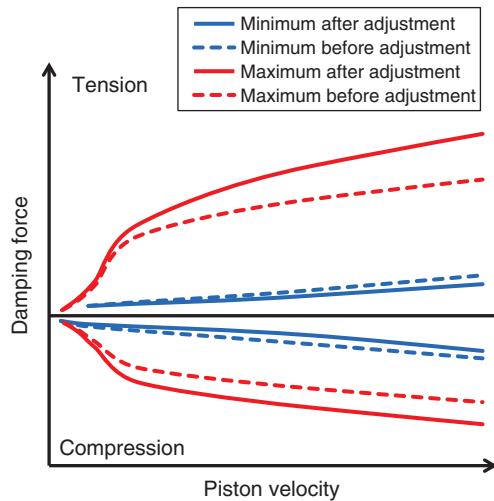


Fig. 13 Example of adjusted damper force characteristic

5. Actual Vehicle Trials

In order to verify the effectiveness of the developed system in an actual vehicle, acceleration sensors were fitted to the damper mounting points of a test vehicle and sprung mass acceleration signals were measured during actual driving. The developed system was fitted in the test vehicle, and the damping force of the adaptive dampers was controlled without the use of dedicated sensors. To make it possible to compare results for the developed system with conventional ADS using dedicated sensors, sensors for the measurement of the vertical acceleration of sprung mass and suspension stroke sensors for control purposes were also fitted in the vehicle, and a switch was installed to make it possible to switch between the developed system and the conventional system. Measurements focused on comparing vibration suppression performance in the 1 Hz to 2 Hz range of sprung mass resonance and isolation performance in the 3 Hz to 8 Hz range. The tests were conducted on a test course modeling an urban road with frequent input from the road surface in these frequency bands.

To begin, tests were conducted to determine the effect of conventional ADS using dedicated sensors in controlling vertical vibration of the vehicle body. Figure 14 shows an example of a comparison of the power spectral density (PSD) of the vertical acceleration of sprung mass at the rear wheels for control by conventional ADS, the use of minimum damping force without the application of

control, and the damping force provided by a conventional damper. The results show that the application of control by the conventional ADS resulted in greater suppression of vibration close to the 1 Hz to 2 Hz range of sprung mass resonance than the use of minimum damping force. Control by the conventional ADS also reduced the level of vibration close to the 3 Hz to 8 Hz range for isolation to a lower level than the damping force provided by the conventional damper. These results indicate that skyhook control is able to control sprung mass vibration irrespective of the frequency range, and increases ride comfort against the use of conventional dampers. Unsprung mass control, which boosts road-holding performance, and pitch and roll control, which boost handling and stability, were also applied, and results showed no increase in the vibration level in the range for isolation or other effects that would result in a decline in ride comfort.

Next, PSD for the vertical acceleration of sprung mass was compared for conventional ADS and the developed system in order to compare the effectiveness of the systems in controlling vehicle vibration. Figure 15 shows the

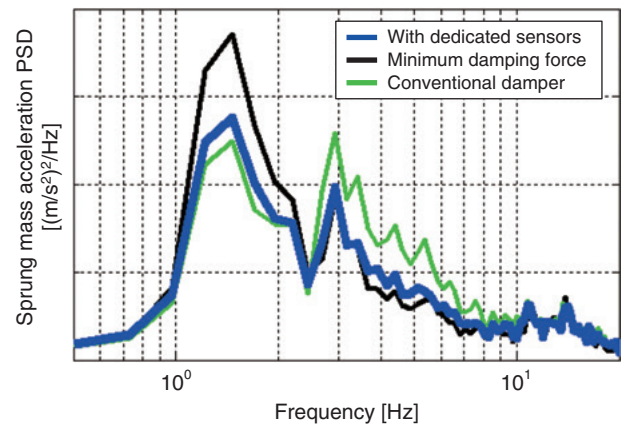


Fig. 14 Example of PSD of sprung mass acceleration for ADS using dedicated sensors

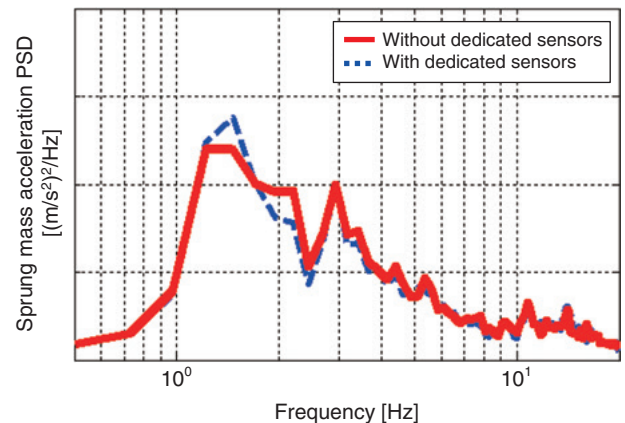


Fig. 15 Comparison of PSD of sprung mass acceleration for ADS using dedicated sensors and ADS using estimation

results of this comparison. The level of vertical vibration of sprung mass for the developed system is similar to that for conventional ADS in the range of sprung mass resonance and the range for isolation.

These results demonstrate that control using the developed system produces a similar vibration characteristic to control using the conventional system, for which sensors for the measurement of the vertical vibration of sprung mass and suspension stroke sensors are necessary, and that the developed system is able to realize the same effect in controlling vehicle vibration as the conventional system.

6. Conclusion

An ADS that reduces system costs while offering a wide range of damping force settings and applicability to diverse vehicle categories has been developed.

- (1) The development of a control logic that estimates the state quantities of vertical vibration has made it possible to control vertical vehicle vibration using skyhook control without the need for dedicated sensors. In addition, the system also realizes unsprung mass control and pitch and roll control that maintain ride comfort.
- (2) The system employs adaptive damper units with a triple-tube configuration that enables control of damping force during both the tension and contraction strokes using a single solenoid valve. The damping force characteristic can be adjusted for different vehicle types through replacement of the externally mounted solenoid valve.

The developed ADS has been employed in the Acura AVANCIER SUV, the UR-V, and the CDX compact SUV, and is scheduled for future application in other models.

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Longitudinal 9-speed Dual Clutch Transmission for 2nd-generation NSX

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ABSTRACT

The world's first longitudinal nine-speed dual clutch transmission was developed for the second-generation NSX. Its purpose is to realize the design, acceleration performance, and dynamic performance that are appropriate in a supercar for the new era, as well as to provide smooth, comfortable driving on city streets and on expressways. By eliminating the shift mechanism for first gear and sharing the transmission gears, the total length was reduced 120 mm relative to these measures not being applied, achieving a rear overhang that is 35 mm shorter than other manufacturers' supercars with seven-speed dual clutch transmission. In order to deal with the higher power output, the optimal oil was respectively used for the clutch and gear rooms, while the adoption of a high-capacity dual wet clutch, high-strength gear material, and high-strength case material increased capacity to 550 Nm. In order to deal with high-load driving on the circuit and under other such conditions, the respective cooling and lubrication systems for the clutch and gear rooms were optimized by the use of four heat exchangers and a baffle cover that completely encloses the clutch. The gear ratio was optimized for each shifting stage, the rigidity of the shift fork was increased, and a double-cone synchronizer was adopted for second to fifth gears, thereby realizing high gear change response while retaining quiet cruising at high speeds.

1. Introduction

With the aim of giving form to drivers' universal desire to pursue the joy of driving control still further, the second-generation NSX was developed as a supercar for a new era with the keywords "New Sports eXperience" to realize unprecedented levels of driving control.

The second-generation NSX has a longitudinally mounted midship V6 3.5-L twin-turbocharged engine and a direct drive motor directly connected to the crankshaft together with a nine-speed dual clutch transmission (DCT) to drive the rear wheels. The twin motor unit (TMU) mounted on the front axle is combined with the SPORT HYBRID SH-AWD four-wheel drive system to deliver highly responsive acceleration performance that utilizes the high power produced by the system. Table 1 shows the specifications for the power train system mounted in the second-generation NSX. Figure 1 shows the layout of the components⁽¹⁾.

Conventional supercars have been equipped with a six- to seven-speed longitudinal DCT that realized overwhelming acceleration in circuit driving. The transmission for the second-generation NSX was required to produce a balance of further enhanced supercar performance with smooth, comfortable driving on ordinary public roads.

Table 1 Specifications of power train system

Power train system	SPORT HYBRID SH-AWD Maximum power 427 kW Maximum torque 646 Nm
Engine	Twin-turbocharged DOHC V6 3493 cm ³ 373 kW at 6500-7500 rpm 550 Nm at 2000-6000 rpm
direct drive motor	Water-cooled motor and generator directly connected to crankshaft 35 kW at 3000 rpm 148 Nm at 500-2000 rpm
Transmission	9-speed DCT
TMU	Twin traction motor and generator unit 27 kW at 4000 rpm × 2 73 Nm at 0-2000 rpm × 2

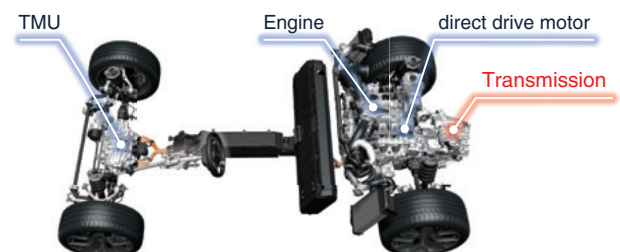


Fig. 1 Power train system components

* Automobile R&D Center

Realizing these aims necessitated a longitudinal nine-speed DCT that achieved a balance between greater compactness and added gear stages for optimal gear ratios at each shifting stage.

2. Development Aims

In developing the longitudinal nine-speed DCT for the second-generation NSX, the following items were defined as development aims.

- (1) Achieve a balance between a configuration with nine shifting stages and greater compactness.
- (2) Achieve the increased capacity to enable transmission of the supercar's high power output.
- (3) Achieve thermal feasibility during circuit driving and other such high-load driving.
- (4) Achieve a high level of instant launch, linear acceleration performance, and smooth, quiet cruising.

3. Basic Construction and Main Specifications

Table 2 shows the main specifications of the longitudinal nine-speed DCT for the second-generation NSX. Figure 2 shows a main sectional view of the DCT.

3.1. Parallel Multi-shaft Longitudinal Structure

The structure was designed so that the output from the engine and direct drive motor would work through the dual wet clutch (DWC) to transmit torque to the various shafts. A parallel multi-shaft structure was adopted for the transmission gears in which the drive gears of odd-numbered stages are located on the main shaft and those of even-numbered stages are located on the secondary shaft, with the input shaft and idle shaft placed in between, while the driven gear is located on the counter shaft and is shared⁽²⁾. The use of hypoid gears also made it possible to replace the output shaft at a right angle to give the transmission a longitudinal structure.

3.2. Close Ratio Configuration

First gear was configured with a low ratio to provide instant launch as SPORT HYBRID SH-AWD. Maximum shift range was configured with a high ratio to achieve quiet cruising at high speeds. The gear ratios of the seven stages between them were given a close ratio configuration that would retain acceleration in a power peak zone wherein the engine is kept in the high-speed range (6000 rpm) up to the maximum vehicle speed (Fig. 3).

3.3. Two Types of Oil

Two types of oil are used in order to obtain maximum performance from the nine-speed DCT. Two oil seals were positioned between the clutch case and the input shaft and another two between the input shaft and the main shaft. This structure was adopted to separate the oil in the clutch room from that in the gear room (Fig. 4).

Table 2 Transmission specifications

Transmission type		9-speed DCT
Torque capacity [Nm]		550
Maximum engine speed [rpm]		7500
Power capacity [kW]		373
Overall gear ratio	1st	13.754
	2nd	8.719
	3rd	6.370
	4th	5.116
	5th	4.341
	6th	3.720
	7th	3.156
	8th	2.677
	9th	2.272
Reverse		8.580
Ratio range		6.055
Overall length [mm]		633
Oil	Clutch room	ATF
	Gear room	Gear oil

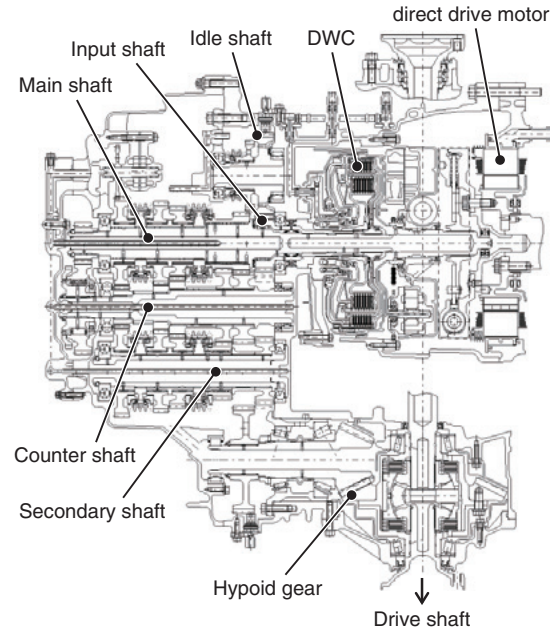


Fig. 2 Main sectional view of developed DCT

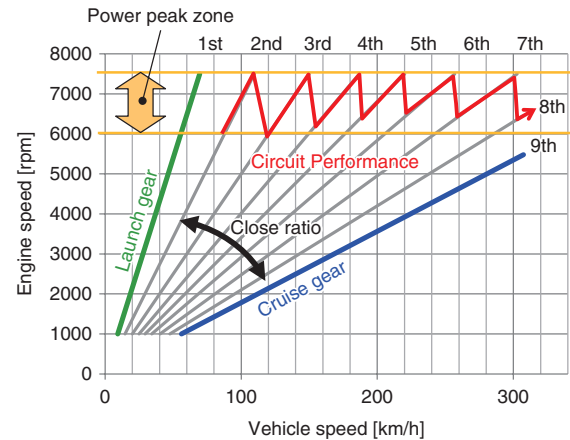


Fig. 3 Speed diagram of 9-speed DCT

A newly developed gear oil was adopted for the gear side to provide high viscosity and high oil film retention under conditions of high temperature and high surface pressure in order to support high power output. For the clutch side, however, an automatic transmission fluid (ATF) with an optimal blend of additives for superior oil properties was adopted in order to achieve a balance between DWC torque transmission performance and anti-judder characteristics (Fig. 5).

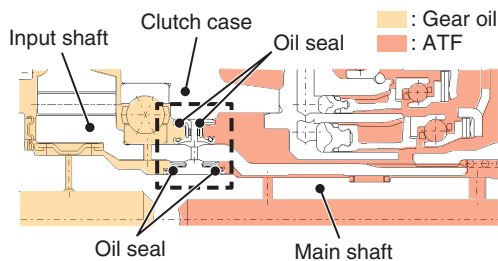


Fig. 4 Oil seal for oil separation

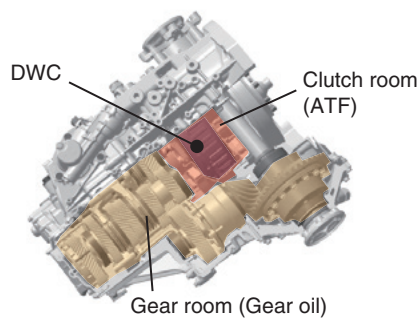


Fig. 5 Separation of clutch and gear room oil

4. Techniques Addressing Requirements

4.1. Added Gear Stages and Greater Compactness

It was necessary to reduce the total length of the longitudinal nine-speed DCT so that the limited overhang length would reduce yaw moment when turning as well as increase design freedom for the rear of the vehicle.

In order to limit the increase in total length caused by the addition of gear stages, the shift mechanism used in other shifting stages was eliminated from first gear. The adopted structure has a one-way clutch (OWC) shared by the first driven gear and parking gear. The fourth and fifth, sixth and seventh, and eighth and ninth gears were aligned and configured to share the driven gear on the counter shaft. The output drive gear and the reverse driven gear were configured to share the third driven gear (Fig. 6). These measures for a compact layout reduced the total length 120 mm relative to when the measures are not taken, while the adoption of a nine-speed system simultaneously

achieved a rear overhang that is 35 mm shorter than other manufacturers' supercars with a seven-speed DCT (Fig. 7).

4.2. Support for High Power

The nine-speed DCT was required to transfer the high power output from the engine and direct drive motor.

For the clutch, the DWC with torque capacity of 370 Nm used in the 2015 model year Acura RLX was used as the base and given greater capacity. The thickness was respectively optimized for the steel plates in the odd- and even-numbered stages, and the number of friction discs was increased. Combining this with optimized lubrication achieved a torque capacity of 550 Nm while retaining the former dimensions (Fig. 8).

In order to achieve high capacity without increasing total transmission length, high-strength steel was adopted for all the transmission gears and the module specifications were optimized. The durability of the hypoid gears was enhanced

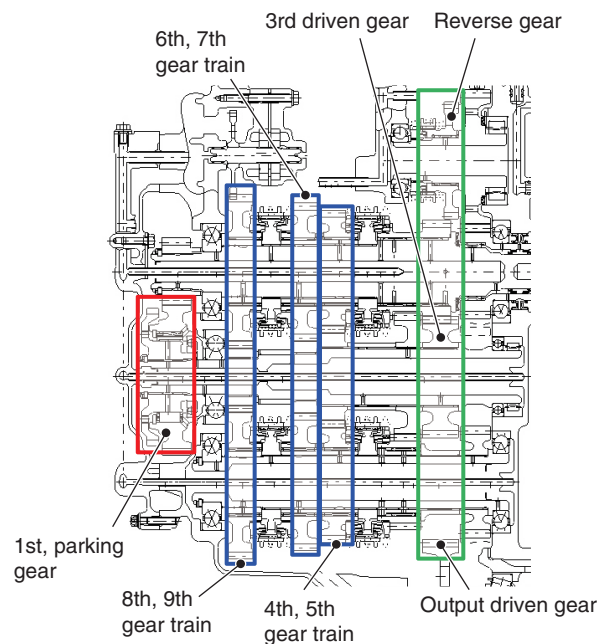


Fig. 6 Structure of gear train

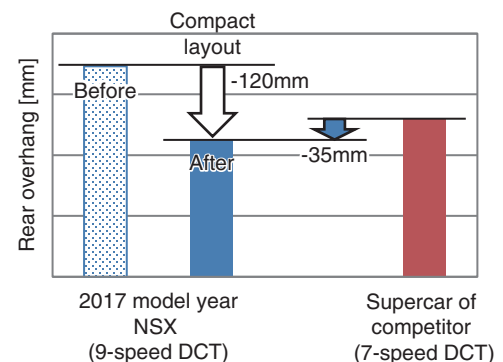


Fig. 7 Rear overhang

by polishing the tooth surfaces and applying precision surface processing. For the transmission case, a high-silicon aluminum material with superior abrasion resistance was adopted. These measures increased the fatigue strength of the transmission case by 20% relative to the 2015 model year Acura RLX.

The application of these techniques and the two types of oil described earlier together achieved the increased torque capacity of 550 Nm, capable of transmitting the high power output of a supercar.

4.3. Thermal Feasibility in High-load Driving

4.3.1. Cooling and lubrication systems

The clutches and gears were equipped with their respective cooling and lubrication systems in order to realize cooling performance matched to the severe use conditions that involve constant driving at high speeds as well as continuously alternating between acceleration and braking. Four heat exchangers were installed (Fig. 9), and the respective systems were optimized to provide cooling and lubrication appropriate to the level of heat generation in each part (Fig. 10).

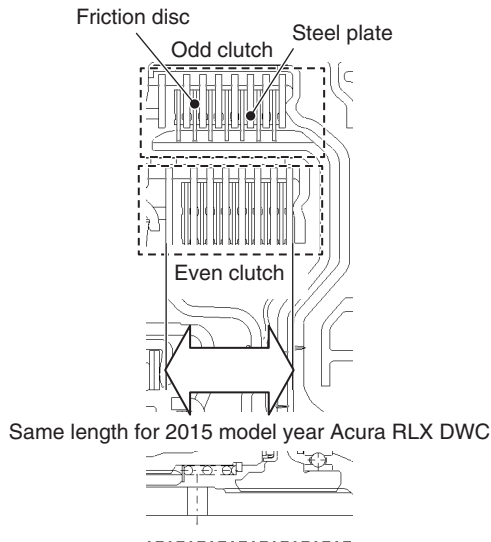


Fig. 8 High capacity DWC

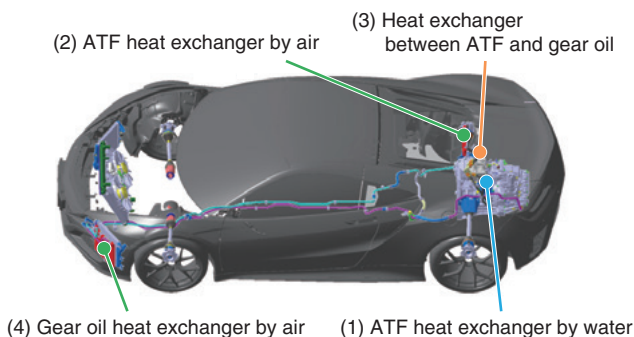


Fig. 9 Heat exchanger layout

On the clutch side, since higher heat is generated in the DWC when absorbing energy at high engine speeds, the system was structured to drive the oil pump in proportion to engine speed. A flow-regulating mechanism was adopted to optimize lubrication to the DWC temperature and operating state. In addition to conventional water-cooled (1) and air-cooled (2) heat exchangers, the system was also equipped with a heat exchanger (3) that exchanges heat between the ATF and gear oil to increase cooling performance. The water-cooled heat exchanger (1) and the heat exchanger (3) were given parallel circuits to reduce pressure loss. On the gear side, since more heat is generated when the vehicle is traveling at higher speeds, the system was configured to drive the oil pump according to engine speed. The air-cooled heat exchanger (4) was positioned at the left front of the vehicle to heighten cooling efficiency, and the system was configured to directly provide the meshing portion of the hypoid gears, where heat generation is especially intense, with oil that has been passed through the air-cooled heat exchanger (4).

4.3.2. Lubricating oil management

Aeration of the lubricating oil occurs due to the rocking that occurs during acceleration, deceleration, and turning, and due to agitation by the DWC and gears. It was necessary to limit this aeration because it reduces the discharge volume of the oil pump, which tends to reduce cooling efficiency. The breather tube, which limits the rise in pressure inside the transmission, also had to be kept from blowing out oil. In order to manage the lubricating oil to address these needs, the respective lubricating structures of the clutch and gear rooms were optimized.

For the clutch room, a baffle cover that completely covers the DWC was adopted in order to limit oil agitation due to DWC rotation and to facilitate the discharge flow of oil supplied from the DWC shaft core (Fig. 11). The shape of the baffle cover and the location of the oil drain were optimized in order to limit the rise of the fluid surface due to wobbling of the oil within a compact space. For the gear room, a forced lubrication structure that blows oil directly

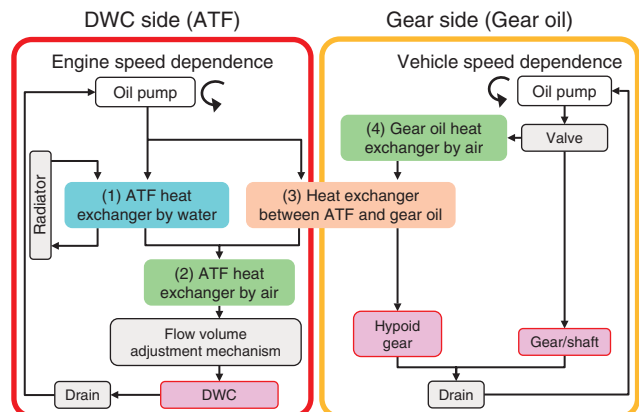


Fig. 10 Lubrication and cooling system

onto the meshing portion of the transmission gears was adopted (Fig. 12) in addition to the provision of oil to the gears through lubrication pipes inside each of the shafts, thereby ensuring a stable supply of lubricant to the gear tooth surfaces.

The use of these techniques achieved a high level of cooling performance and stable lubrication management, realizing thermal feasibility during high-load driving. As one example of the results, Fig. 13 shows the cooling effect during one lap at the Nürburgring Circuit. This confirmed that there is sufficient cooling performance to enable the nine-speed DCT to demonstrate its performance capabilities.

4.4. Heightening Shift Mechanism Response

As the shift mechanism realizes enhancement of acceleration performance as well as smooth driving, it demands high gear change response. In order to shorten the shifting time when changing gears, a high-response gear-shifting system was adopted that combines a high-capacity double-cone synchronizer, a high-rigidity shift fork, and a gear actuator (GA) (Fig. 14). These elements were also optimized.

A double-cone synchronizer was adopted to increase the capacity of second through fifth gears, because the energy absorbed during synchronized gear shifts is larger with

those shifts. With the goal of setting the gear shifting load at a higher level and shortening synchronization time, the material used for all the blocking rings from second to ninth gear was changed from the cold-rolled steel used in the 2015 model year Acura RLX to higher-hardness chromium molybdenum steel to enhance durability.

High-strength cast steel, which allows superior shaping flexibility, was used as the material for the shift fork. The rail and the shift fork portions were cast as a single piece (Fig. 15), and the resulting higher rigidity reduced the displacement of the shift fork tip when under load input from the GA. Optimizing the shape also enhanced the

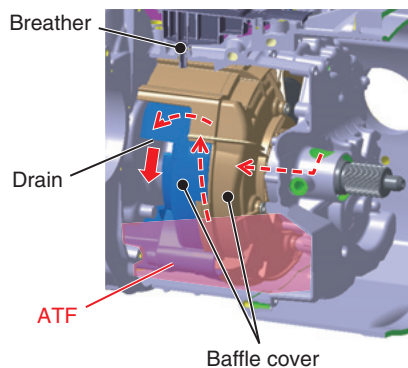


Fig. 11 Oil flow in baffle cover

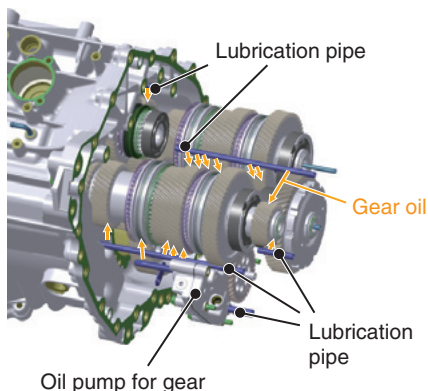


Fig. 12 Direct lubrication of gears

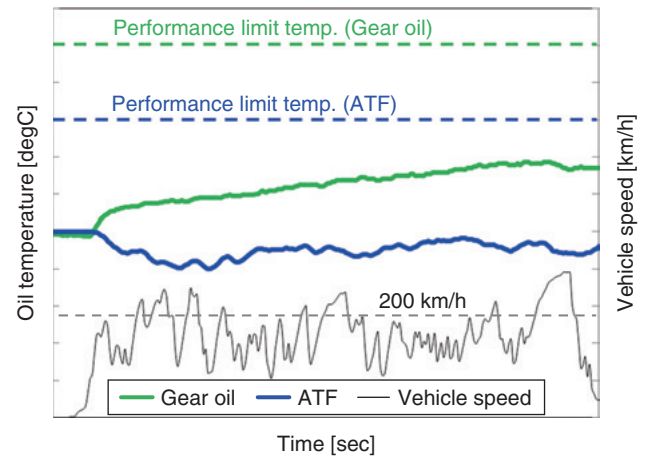


Fig. 13 Oil temperature behavior on circuit drive

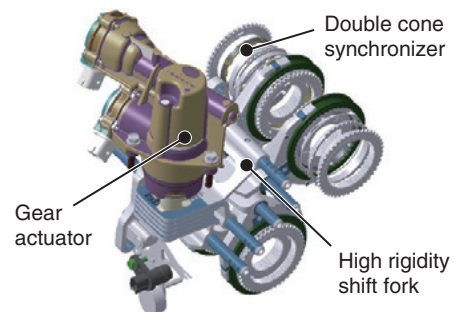


Fig. 14 High response gear shifting system

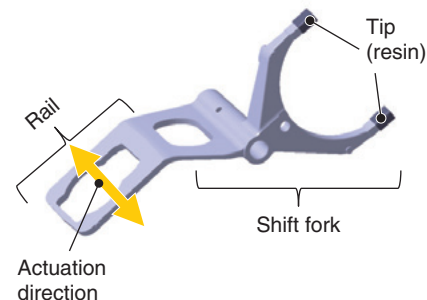


Fig. 15 High rigidity shift fork

balance of the load on the left and right tips of the shift fork during synchronization. Shortening the GA shaft length heightened rigidity, increasing gear shift system rigidity by 40% or more over the 2015 model year Acura RLX, and reducing load power loss at the shift fork tips with respect to the load input from the GA.

These techniques enabled the gear shift load during synchronization to be set even higher than in previous Honda models with a GA. In addition to increasing the synchronizer capacity, synchronization time was shortened by 65% (Fig. 16).

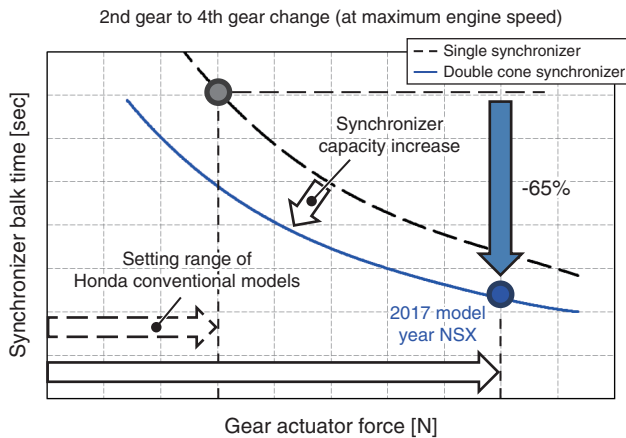


Fig. 16 High response gear shifting

5. Balancing Acceleration Performance and Quietness

5.1. Enhancing Acceleration Performance

The optimization of gear ratios described above realized a close ratio with a common ratio in second through eighth gears that is even lower than in the seven-speed DCT of competitors' supercars (Fig. 17), contributing to enhanced acceleration performance.

The combination of the high system power output of the SPORT HYBRID SH-AWD with the optimized gear ratios and high gear change response of the nine-speed DCT achieved instant launch and linear acceleration performance. Figure 18 shows acceleration at launch control. This reduced gear change time by up to 50% over that of the seven-speed DCT in rival supercars. These measures contribute to acceleration from 0 to 100 km/h in the three-second range, while also enhancing mid-range acceleration performance.

5.2. Quiet Cruising

Setting the nine-shift range at a high ratio reduces engine speed during high-speed cruising. Even when cruising at 100 km/h, engine speed is just 1800 rpm or less, which is on a par with the 2015 model year Acura RLX. Smooth and quiet cruising even at high speeds is ensured.

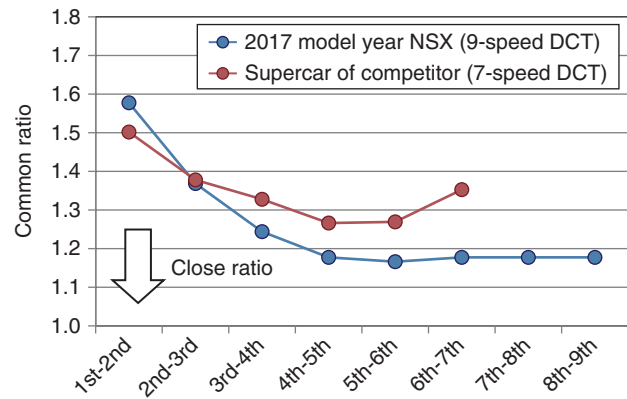


Fig. 17 Sport close ratio

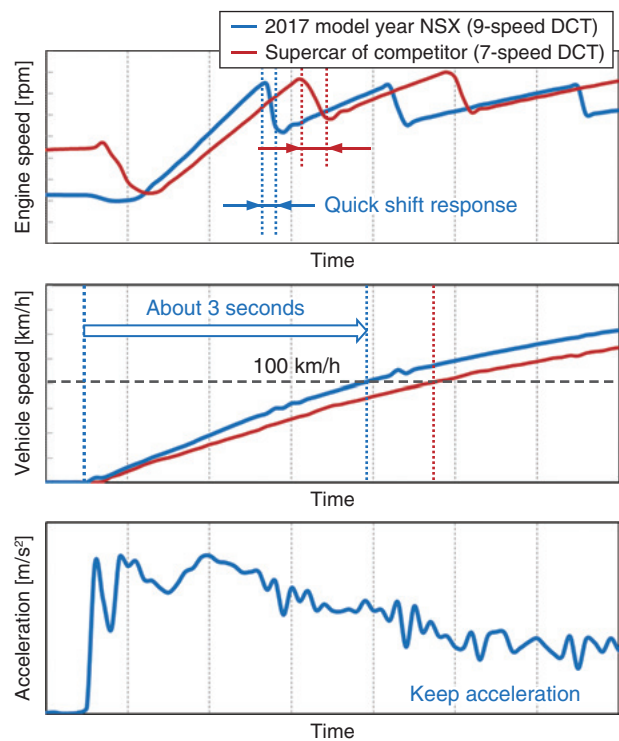


Fig. 18 Acceleration at launch control

6. Conclusion

A longitudinal nine-speed DCT for use in the second-generation NSX was developed and achieved the following results.

- (1) Elimination of the first-gear shift mechanism and sharing of transmission gears shortened the total length by 120 mm, and at the same time a nine-speed configuration was adopted, a rear overhang that is 35 mm shorter than the seven-speed DCT of other manufacturer's supercars was achieved.
- (2) Oil was optimized for the clutch and gear rooms, and

the adoption of a high-capacity DWC, high-strength gear material, and high-strength case increased torque capacity to 550 Nm.

- (3) The respective cooling and lubrication systems for the clutch and gear rooms were optimized by the use of four heat exchangers and a baffle cover that completely encloses the DWC, increasing thermal efficiency during high-load driving on the circuit and under other such conditions.
- (4) Optimized gear ratios for each shifting range, a high-rigidity shift fork, and a double-cone synchronizer for second to fifth gears achieved high gear change response as well as a balance between acceleration performance from 0 to 100 km/h in the three-second range and quietness with engine speeds at 1800 rpm or lower when cruising at 100 km/h.

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Development of Sports Car Sound for Hybrid Super Sports Cars

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ABSTRACT

Developing the sports car sound for the next-generation Acura NSX Sports Hybrid required a unique approach to vehicle noise and vibration development. Starting with a holistic acoustic strategy for the vehicle platform design, insulation package and chassis isolation specifications were constructed to accentuate cabin quietness during hybrid driving. Employing multiple acoustic technologies, devices and control strategies, specific frequency and order content was enhanced to maintain the original NSX heritage sound while adding functional purpose to audibly communicate instantaneous power unit feedback to the driver. Execution of this strategy achieved the vehicle sound concept and the hybrid super sports car segment competitiveness was redefined.

1. Introduction

The fundamental goal for the second-generation NSX is to be a leader in today's cutting edge supercar market, yet pay tribute to the heritage of the original NSX, which was considered revolutionary and helped to build the Acura brand nearly 25 years before. The primary focus for the sound concept was to maintain the heritage sound of the original naturally-aspirated (NA) V6 engine. The challenge was to achieve this with the new hybrid electric twin turbo V6 engine. A secondary goal was to celebrate the electric vehicle (EV) driving capability of this next-generation supercar. This was achieved by building on the foundation of a quiet interior by using sealing, noise, vibration, and harshness (NVH) treatments, and road and engine isolation strategies. Next, power unit breathing systems were designed to celebrate the character inside the unique 75 deg V6 twin turbo engine. Finally, five specific actively controlled systems were developed and tuned to deliver the desired sound including audible feedback to the driver indicating transmission gear changes and engine speed. This enables the driver to focus on the road while minimizing distractions from driving, such as continuously looking down at the gauges for feedback. The final product is a super sports car that can transform from a quiet, luxury grand touring car, to a raucous, roaring, track monster, with just the twist of its Integrated Dynamic System (IDS) knob.

The original Acura NSX, first introduced in 1991, played

Table 1 Sound characteristics of the original Acura NSX

Hardware	Sound quality contribution	Sound content
Exhaust system	Low frequency, powerful sound	3 rd order
	High RPM 'Racing' sound	6 th order
Intake system	Mid frequency 'Guttural' sound	450 Hz resonance

a large role in the development of the next-generation Acura NSX with the intent to pay homage to the characteristics which made the original car so unforgettable and revered in the sports car market. The original car was powered by a 60 degree, naturally-aspirated V6 engine, mounted mid-ship, immediately behind the driver. This power unit provided excellent sound quality and driver feedback through several hardware systems listed in Table 1:

Figure 1 shows the cabin sound of the original NSX during full throttle acceleration in third gear. As highlighted below, each rpm range has a unique sound character.

The finely tuned specifications of the intake and exhaust systems' sound quality contributed to the driver feeling connected to the car. Pressing the accelerator pedal resulted in an increased acoustic response from the vehicle. The sound was directly linked to the acceleration feeling, a quality desired to carry forward to the next-generation NSX.

With the NA engine of the original NSX, the authentic

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sound quality of the engine's breathing systems is a key aspect of its character. This authenticity is also a quality desired to carry forward to the next-generation NSX.

Automotive technology, construction and performance have changed considerably since the original NSX was launched, and the next-generation NSX needs to satisfy the expectations of customers in the current super sports car market. Traditional super sports cars are often defined by particularly sporty NVH characteristics, including loud combustion noise, powertrain mechanical noises, lack of road isolation, and strong vibrations. These vehicles tend to be raw and unrefined, making for a very authentic sports sound experience; however, they are also very tiring to drive, tending to wear on a driver and making anything more than a short trip uncomfortable. This segment positioning is shown in Fig. 2, along with the typical luxury sports sedan vehicle and Acura product placement.

Over the past 25 years, the introduction of electronic controlled powertrain and sound devices enabled manufacturers to better balance the general driving NVH refinement and sports car sound. These technologies permit the driver to select their desired driving setting, within some performance window, as shown in Fig. 3.

Market analysis shows the customer expectation in this segment has been raised significantly since the release

of the original NSX. Most buyers expect a vehicle that is reasonably luxurious and refined, suitable for longer drives, but that can also deliver a super sports car, visceral and raw engine sound experience.

This paper describes the target setting and development of the NSX sports car sound, including hardware selection, specification setting, tuning, and competitive performance results.

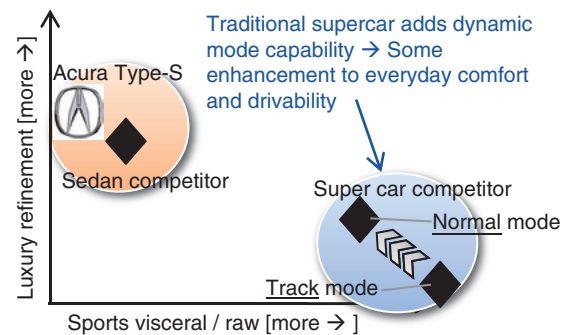


Fig. 3 2015's sports car market landscape

2. Target Setting

With an understanding of the current sports car market landscape and customer expectations as indicated in the previous chapter, an engine sound concept was defined to deliver a New Sports eXperience (NSX). Every driving situation encountered by the customer was considered, such as acceleration, cruising, and gear changes. Then, the desired sound performance in each of these driving situations and IDS modes was defined, as shown in Fig. 4.

The sound concept established several 'key words' to describe critical noise performance characteristics, as listed and defined in Table 2.

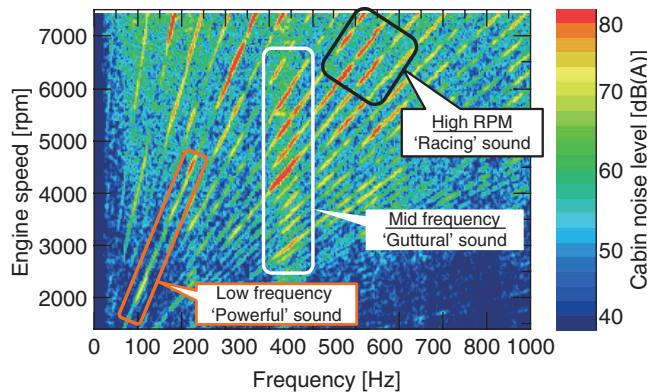


Fig. 1 Original Acura NSX cabin sound during acceleration

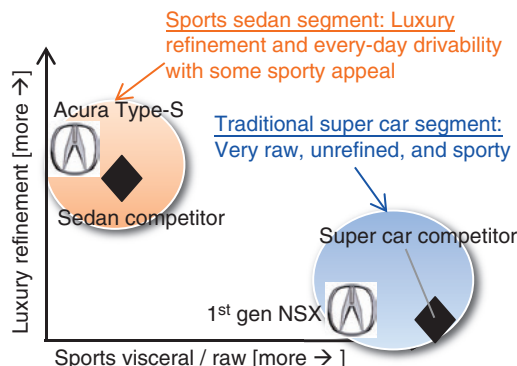


Fig. 2 1990's sports car market landscape

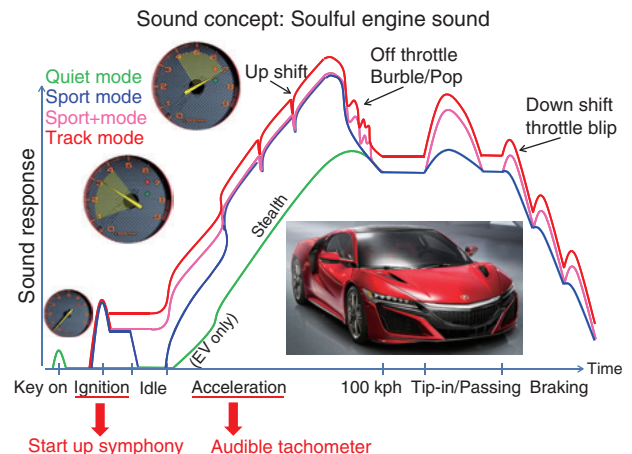


Fig. 4 Vehicle sound concept

Table 2 Target key word definitions

Target key word	Target character
Start-up symphony	Engine start-up sound, which provides audible feedback based on the selected Dynamic Mode – Quiet mode → Subdued startup maximizing quietness Sport mode → Exciting, high revving engine flare with fast, race-car like response
Audible tachometer	Sound that indicates to the driver the approximate speed of the engine, without visual reference to the tachometer. To be used as an aid, which permits the driver to keep their eyes on the road or track, but still keep exact upshift/downshift points for maximum performance

IDS modes are key to this concept by creating a fully adjustable driving experience, where each mode provides the driver a different vehicle character including a tailored acoustic performance impression. Each IDS mode has a unique intended use, as summarized in Fig. 5.

Quiet mode:

The Quiet mode concept is to enable the driver to minimize detection by the outside world as much as possible, abating noise and creating a comfortable, quiet cabin environment.

Sport mode:

The Sport mode concept is to enable the driver to experience an NSX, integrating hybrid vehicle driving with that of a super sports car with exceptional sound response to driver input, while maintaining everyday drivability. Sport mode is the default start-up mode, although Quiet mode may also be selected.

Sport+ mode:

The Sport+ mode concept is focused on winding or mountain road driving, eliminating internal combustion engine stop and giving the driver increased on-road audible feedback and performance.

Track mode:

The Track mode concept is for circuit use only, to achieve fast, repeatable lap times and give the driver



Fig. 5 IDS mode concept and intended uses

maximum acoustic feedback that is even audible while wearing a helmet.

Quantitative targets are based on a metric which compares cabin sound level during full throttle acceleration to the number of orders that can be heard by the driver, known as audible orders. Audible orders are defined as orders within 7 dB(A) of the overall A-weight cabin sound. This was experimentally determined to be the threshold of order contribution to sound quality. Acceleration sound with fewer audible orders has harmonic sound, while acceleration sound with higher numbers of audible orders has modulation or a more guttural sound. Figure 6 shows the sound performance of the previous NSX as well as several key competitors. The trend shows that louder vehicles have more audible engine orders, so the NSX target range, highlighted in blue, defines the appropriate number of orders required to achieve competitiveness in the 2015's sports car market landscape as indicated in Fig. 3, and maintain the heritage performance of the original NSX.

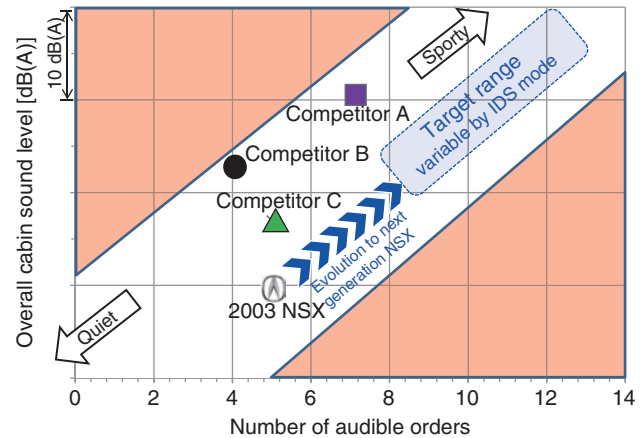


Fig. 6 Cabin sports sound target during full throttle acceleration

3. Approach

In order to achieve all aspects of the vehicle sound concept, several different hardware and software control systems were investigated⁽³⁾, as shown in Table 3.

3.1. Insulation Material Package

Traditional super sports cars are more analogous to racecars when it comes to noise and vibration cabin quietness. Of course weight is the enemy of acceleration and handling performance, so it's easy to choose lightweight or low coverage insulators in the interest of preserving maximum performance. The challenge with the NSX development is to achieve the concept of Quiet mode driving, while minimizing added weight from the material package.

Table 3 Concept achievement method

Sound concept component	Hardware / Software / Control system
Quiet mode, overall cabin comfort	Insulation material package
Startup symphony	Engine startup control
Balance of quietness and low frequency powerful sound	Exhaust system and active exhaust valve
Guttural sound	Intake sound control
Audible tachometer	Active sound control
Off-throttle Burble/Pop	Combustion stability control
IDS mode differentiation	Combination of all systems

This was addressed by specifying the package based on the noise source hierarchy. Figure 7 shows a representation of the input intensity of each noise source.

As shown in Fig. 7, the primary noise sources are behind the driver. Focus is then concentrated on treatment in this area, centralizing the weight in the rear bulkhead panel insulator, rear panel carpet, rear partition glass, and many other smaller parts to create high noise transmission loss from the engine room to the cabin. For the noise sources in the front of the vehicle, light weight absorptive materials are applied to the front bulkhead panel and wheel arch regions, where the heavier weight, barrier materials are not required.

The result of this strategic isolation is dramatic and benefits the vehicle concept in several ways:

- (1) Cabin quietness: In Quiet and Sport modes, the isolated cabin creates a smooth transition between IC engine running to full EV driving and is often imperceptible by the driver. EV driving is very quiet in the cabin and the highway cruising noise level is more like a sports sedan than a typical super sports car. Cabin quietness competitiveness is shown in Fig. 8, where NSX demonstrates best in class road noise performance.

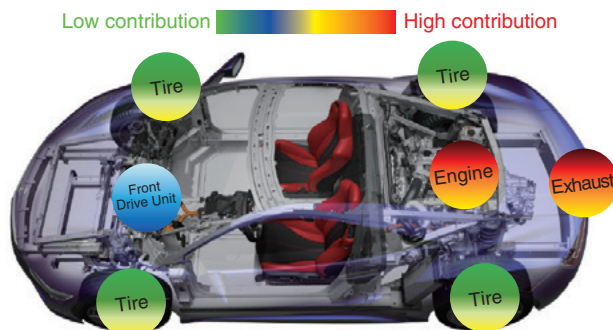


Fig. 7 Air-Bourne source noise contribution

- (2) Low noise floor: With a low noise floor, undesirable noises are suppressed. This provides a clean palette for the addition of combustion and power unit related sports sounds. For example, exhaust noise is not competing with timing chain noise to reach the driver's ear. With the timing chain noise suppressed, the exhaust noise can be more clearly heard and enjoyed. Engine mechanical noise isolation during acceleration is shown in Fig. 9, where NSX realizes best in class performance.

3.2. Engine Startup Control

Engine startup is an important part of the emotional experience with any super sports car. This is when the power unit comes to life and announces its presence to the world. It should impress upon the driver, as well as spectators outside of the car, the sportiness and high capability of the vehicle. Often, this is achieved by a loud, high revving startup, and the NSX concept is shown in Fig. 10. The challenges for the development of the NSX were twofold:

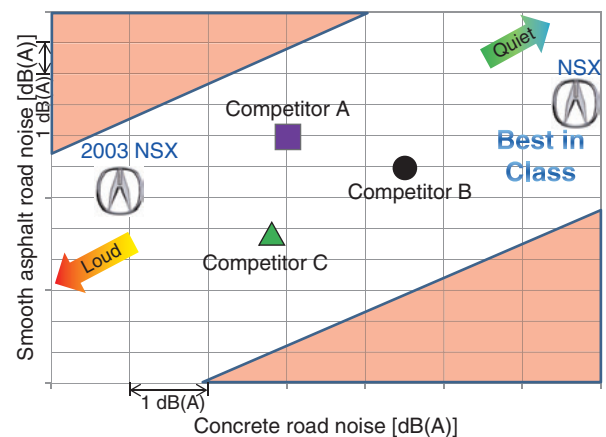


Fig. 8 Insulation material package - road noise performance

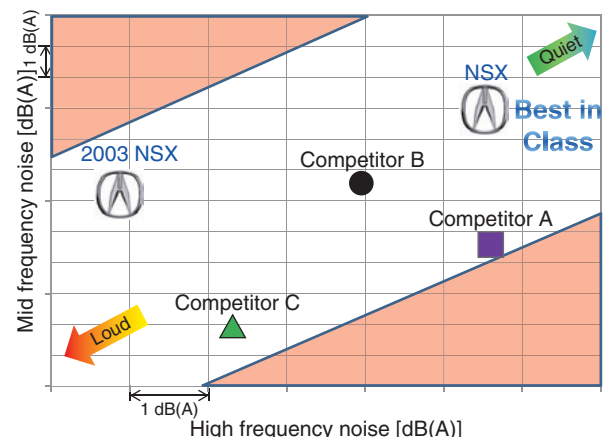


Fig. 9 Engine noise during part throttle acceleration

- (1) To achieve a sporty startup engine flare and corresponding sporty exhaust sound with a turbocharged engine.
- (2) To differentiate the startup sound based on the IDS mode selection of the driver. For example, keeping a subdued startup in Quiet mode, but an exciting startup in Sport mode.

To overcome these challenges, a new approach was required, and an idea was conceived to use the rear drive unit electric motor to control the engine flare speed and amplitude based on IDS mode. This is accomplished by using the motor torque to speed up [during RPM rise] engine speed during cranking, as shown in Fig. 11.

Additionally, engine control software was developed to consider the IDS mode at engine startup. If the driver starts the vehicle in Quiet mode, the engine performs a subdued, quiet start with minimal rev-up over idle speed. If the driver starts the vehicle in any other IDS mode, the engine executes a high revving, quick rev startup, similar to a lightweight racing engine.

3.3. Exhaust System and Active Exhaust Valve

The exhaust system of a super sports car has always been a vitally important component, and many times it is the primary sports sound device. At the start of the project,

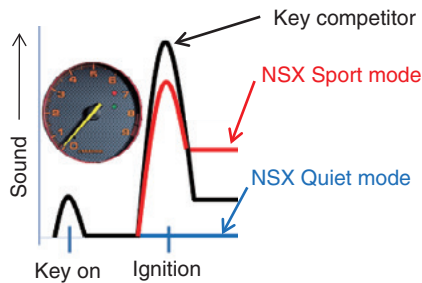


Fig. 10 Engine startup control concept

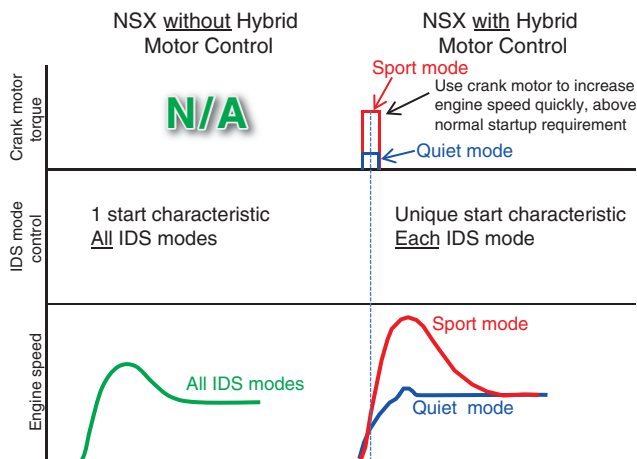


Fig. 11 Hybrid startup control concept

Table 4 Exhaust system specification detail

Specification	Valve closed	Valve open
Silencer volume and type	18 L Pass-through	18 L Side branch
Flow rate at 600 mmH ₂ O	134 L/s	444 L/s
Noise attenuation at 100 Hz	48 dB	34 dB

it was clear that packaging space was small and thermal requirements were severe in order to achieve Track mode performance requirements. The technical challenge was realizing the packaging and thermal requirements, while still achieving both the Quiet and Sport mode concepts, which required radically different noise levels.

To balance those requirements and minimize weight and packaging space, an Active Exhaust Valve (AEV) was introduced, creating a dual mode design that could achieve IDS mode concepts as well as all the aforementioned functional requirements.

The functional design target is to minimize weight as much as possible, while still achieving tailpipe noise targets. The traditional approach to achieve the tailpipe noise target is to use an appropriate silencer volume and then tune the attenuation characteristic and frequencies accordingly. With limited packaging space in the NSX, the approach was changed to utilize the AEVs for flow restriction. Flow restriction has a broadband noise attenuation effect, and this approach achieved the Quiet mode sound targets with the valves closed. With the valves open, the system delivers the desired sports sound and driver feedback from the tailpipe sound.

Silencer volume, flow rate, construction, and valve operation were thoroughly tuned to establish the final, post catalyst, specifications, listed in Table 4. An illustration of the final exhaust system with AEV is shown in Fig. 12.

Once the hardware specification was fixed, the focus shifted to the electronic control of the valves to achieve the vehicle sound concept. Each IDS mode has a different sound concept, so during extensive proving ground, on-road and dynamometer testing, the valve tuning was programmed to achieve the concept shown in Table 5.

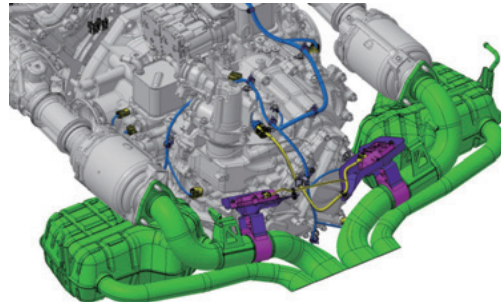


Fig. 12 Exhaust system with AEV

Table 5 Exhaust system sound IDS concept

IDS mode	Quiet	Sport	Sport+	Track
Control logic	Closed	Control by map	Open	Open
Sound concept	Maximize quietness	Balance comfort and sportiness	Max audible feedback	Max audible feedback

For Sport mode, more extensive mapping was developed to ensure cruising comfort and spirited driving needs could all be met with the same setting. Maps were developed with accelerator pedal position, engine speed, and transmission gear as inputs to determine when the valve should be open or closed. Hysteresis was also included to prevent ‘busyness’ during driving around the map line boundary conditions as shown in Fig. 13. In other words, to prevent frequent valve operation during steady state cruising conditions.

The AEV is also used to accentuate the startup sound differences between Quiet and Sport modes. The AEV stays closed at Quiet mode startup to minimize exhaust noise and maintain the quiet character. While in Sport mode, the AEV is open during the engine startup flare to emphasize the startup sound, but closes after several seconds, to achieve the Sport mode concept of comfortable idle and cruising.

Exhaust noise performance of the final design specification has large differentiation between valve closed and valve open operation as shown in Fig. 14. The main order sound pressure level change is around 25 dB(A) and permits the customer to easily appreciate an audible sound change when switching between IDS modes.

The result of this exhaust system development is a versatile, single world specification system, which satisfies the vehicle sport sound concept, with the lowest post-catalyst system weight in its class at 12.3 kg.

3.4. Intake System and Intake Sound Control

The intake system of the original NSX is the single strongest contributor to the sports sound of that vehicle. The

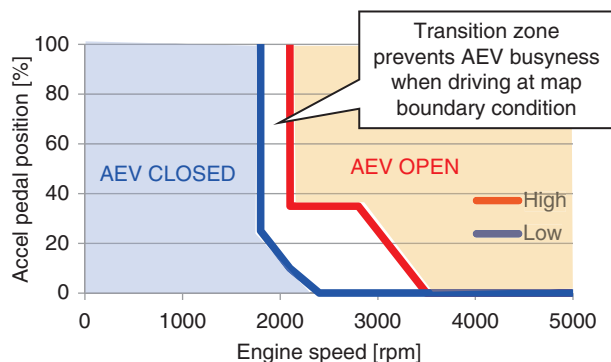


Fig. 13 AEV sport mode control logic map

guttural, mid-frequency modulated intake sound is clearly audible in the cabin and very responsive to driver throttle input. This sound is created by the movement of the intake valve, and is accentuated by the VTEC™ system when valve lift increases dramatically and then becomes airborne through the air-cleaner orifice. Once the sound becomes airborne, it can easily be heard by the driver, because the orifice is directly behind the driver’s head.

On turbocharged engines, the intake valve source noise must travel the path highlighted in Fig. 15 before it can become airborne. The intake noise is significantly reduced by the increased intake pipe length, intercooler, and the turbocharger. The noise attenuation of the turbocharger varies by operating load, as highlighted in Fig. 16⁽¹⁾, and attenuates approximately 15 dB(A) in the combustion noise frequency range during acceleration. This high noise attenuation creates a significant challenge to achieve the vehicle’s emotional sound target; therefore, a technical solution was devised.

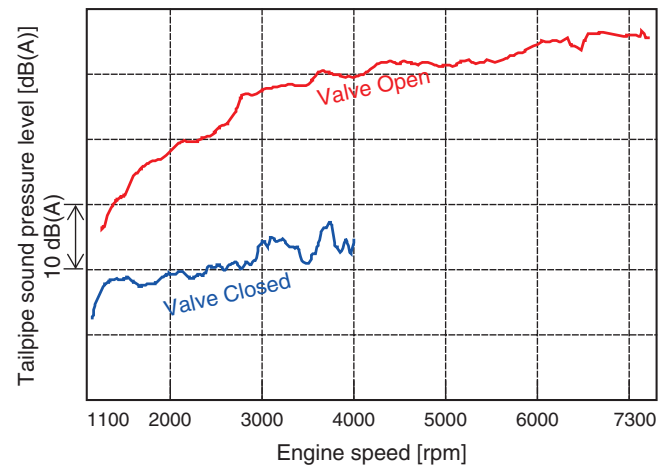


Fig. 14 Exhaust system tailpipe noise at full throttle acceleration

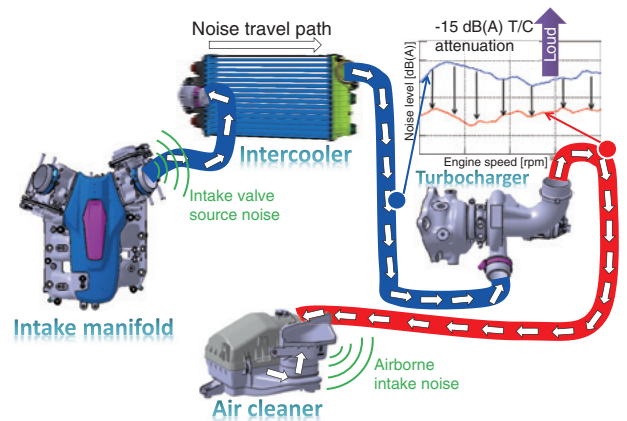


Fig. 15 Intake sound transfer path of turbocharged engine

After investigating multiple technologies, in an effort to provide the most authentic, guttural intake sound inside the cabin, a sound transmitting diaphragm was selected⁽⁴⁾. This technology connects to the high pressure intake duct, where the natural intake noise is present and has not been attenuated by the turbocharger. The diaphragm transfers the high pressure pulsations or source sound, to atmospheric pressure ducting, which can then be routed to the cabin and heard by the passengers, otherwise known as the output sound, as shown in Fig. 17.

Several technical challenges existed to implement this technology and achieve the vehicle concept for cabin sound and IDS mode differentiation, as shown in Table 6. The critical goal was to achieve a natural sound that has a strong link to acceleration response.

When considering the resonant frequency of the Intake Sound Control (ISC) System, there are four primary components as shown in Fig. 18.

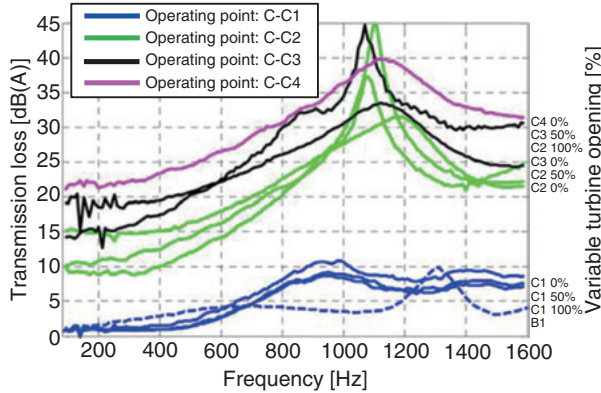


Fig. 16 Noise attenuation due to turbocharger

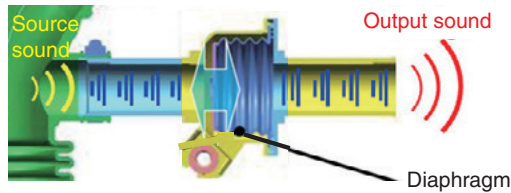


Fig. 17 Sound diaphragm functional layout

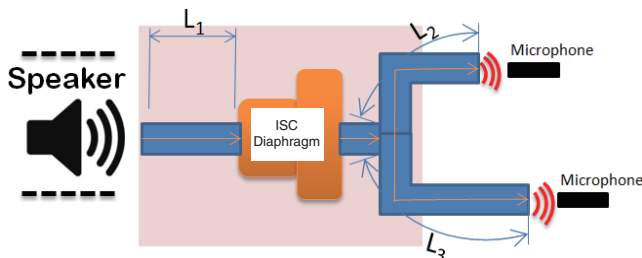


Fig. 18 Bench speaker system attenuation setup

Table 6 Intake sound control technical challenges

Technical Challenge	Solution
1. Create guttural intake sound matching NSX heritage sound	a. Tune diaphragm resonance to best match source noise input frequency and desired heritage intake frequency b. Design pre- and post-diaphragm pipe lengths of $\frac{1}{4}$ wave resonances to coincide with target frequency
2. Create IDS mode variable sound	a. Design a variable control valve with a vehicle integrated control system

- (1) L_1 pipe: High pressure inlet
- (2) ISC diaphragm
- (3) L_2 pipe: Ambient pressure left side duct
- (4) L_3 pipe: Ambient pressure right side duct

To achieve the desired cabin sound, these components have to work together in harmony. The diaphragm itself has some limitations, because it is an existing design. Retuning would require significant investment and R&D cost, which is not desirable on a low volume super sports car like the NSX. The challenge is to use the existing diaphragm design to match the target 450 Hz frequency range to achieve the heritage intake sound. The solution is to tune the $\frac{1}{4}$ wave resonances of the L_1 , L_2 , and L_3 pipes to coincide with the target frequency, and work with the good broadband response of the ISC diaphragm. The resonance calculation method is shown in Fig. 19. The duct resonant frequency is equal to the speed of sound in the medium (air) at the operating temperature of the engine, divided by the wavelength. The total system bench attenuation result is shown in Fig. 20.

Vehicle package studies were performed to determine feasible pipe lengths. A variety of pipe lengths were rapid prototyped and actual sound performance was measured by bench speaker frequency response testing. Using this process, the system performance and specifications were determined, and an illustration of the final ISC system construction is shown in Fig. 23.

Once component specifications were completed, feasible systems were installed in the vehicle and confirmed by chassis dynamometer and on-road testing. The cabin sound was evaluated by microphone data and subjective judgment. As shown in Fig. 21, the ISC system achieved the targeted

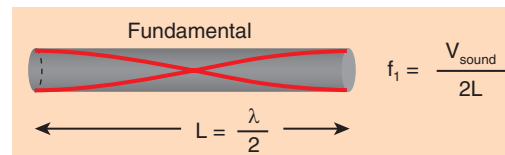


Fig. 19 $\frac{1}{4}$ pipe resonance calculation

frequency of the original NSX, and by subjective driving evaluation, the ‘guttural intake sound’ character is easy to appreciate.

In addition to the tuned pipes and diaphragm, the ISC system is equipped with an electronically controlled valve between the high pressure pipe and the diaphragm. This valve enables control of the sound output of the system to achieve the engine sound targets for each IDS mode. In addition to separation of the IDS modes, a control map for the valve angle was developed to create a linear sound increase with RPM and avoid resonance peaks and valleys. The ISC control valve can quickly adjust position based on driver control and engine operation inputs. The inputs to this system are accelerator pedal position, engine speed, transmission gear, and IDS mode. To this end, map tuning was performed on the chassis dyno, and then confirmed on-road, to achieve the desired sound progression at various throttle angles, engine speeds, shift points, and IDS modes, as shown in Fig. 22.

Additionally, the ISC system and corresponding control valve is used to accentuate the transient sound response, as a driver aid function. For example, when the transmission downshifts, the engine throttle opens to rev-match the engine and transmission speed, and correspondingly, the ISC valve opens, and creates a short burst of sound. This

sound alerts the driver that a downshift has occurred, so the driver doesn’t need to look down at the gauge cluster, and can focus on driving. The blue line in Fig. 22 shows the ISC valve open angle mapping during a downshift, getting more aggressive with each IDS mode.

3.5. Active Sound Control

Active Sound Control (ASC) is a Honda-exclusive engine sound enhancement system, that uses the vehicle’s audio system to add engine order sound to the cabin⁽²⁾. It uses several inputs, such as engine speed, accelerator pedal position, IDS mode⁽⁵⁾, and acceleration rate. Several engine orders can be added, and in the case of NSX, four orders were chosen, with the purpose of filling and smoothing the

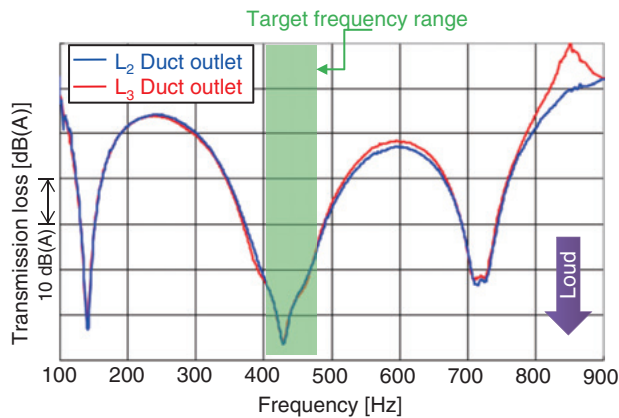


Fig. 20 Bench speaker attenuation test result

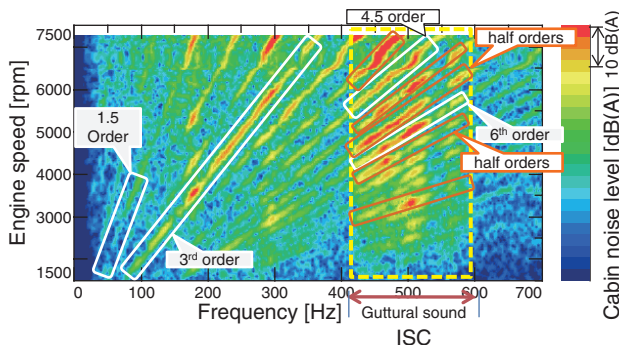


Fig. 21 Final ISC cabin sound at full throttle acceleration

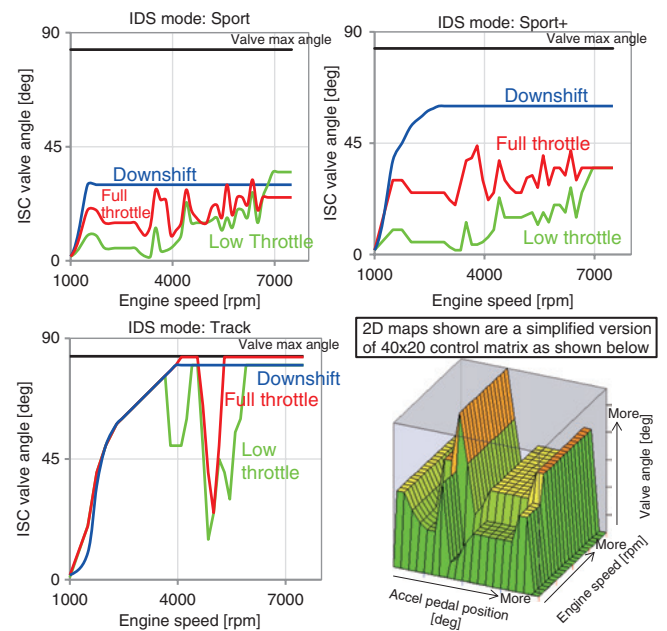


Fig. 22 ISC valve mapping by IDS mode

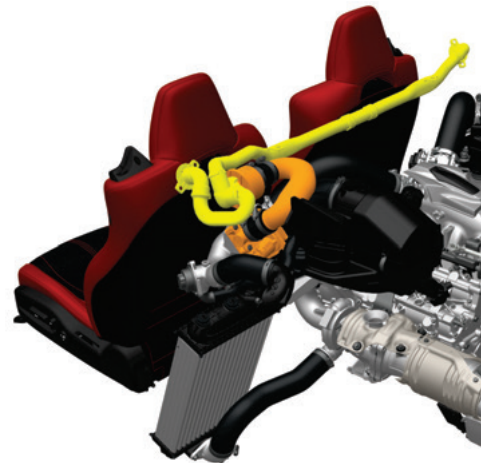


Fig. 23 Final ISC system construction

sound around the other sports sound devices. The orders are separated into two separate strategies:

1. Low RPM Powerful Sound: 3rd and 1.5 orders are introduced to add to the low frequency noise floor and accentuate powerful sound during low rpm driving. This sound is appreciated at tip-in or low throttle accelerations.
2. High RPM Shrill Sound: 3.5 and 4.5 orders are introduced at high accelerator pedal position and high RPM to heighten the audible tachometer sound, helping the driver distinguish audibly when the engine is approaching redline.

The gain level of each ASC order with respect to engine speed is shown in Fig. 24

3.6. Combustion Stability Control for Emotional Exhaust Sound Feedback

One exciting aspect of super sports cars is the sound the driver experiences at transient conditions. Analogous to racecars, sports cars also tend to ‘snap, burble, and pop’ at shift timing or over-run situations and the NSX sound concept incorporated this spirit. This sound was achieved by creating unstable combustion at transient circumstances, by adding ignition retard, based on engine load and engine speed, as shown in Fig. 25. When the combustion

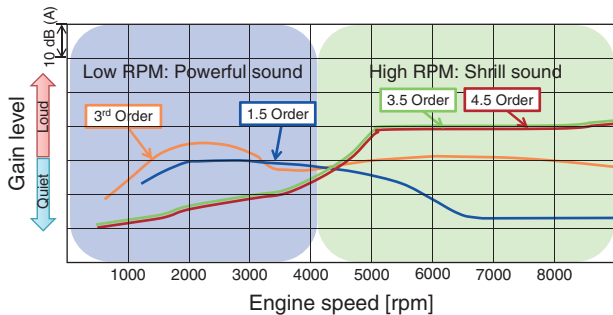


Fig. 24 ASC order gain map

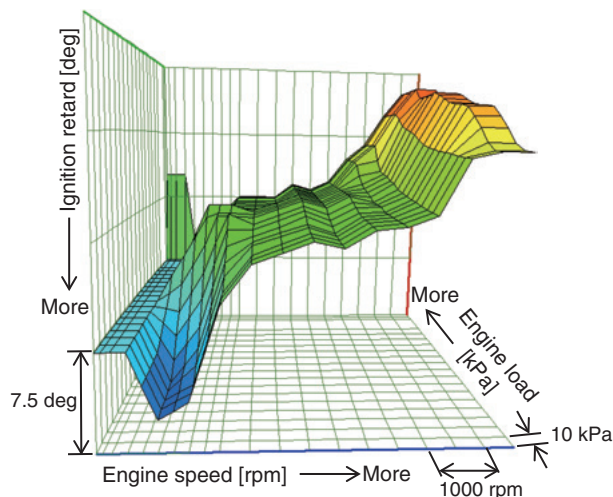


Fig. 25 Ignition retard map

is delayed, the exhaust valve begins to open before the combustion process is completed, allowing complete combustion to finish in the exhaust system, providing an audible ‘snap, burble, and pop’.

This sound also has a functional purpose. The sound can be an indicator to the driver that a change has occurred. Examples are gear shifts, lift throttle, or light throttle application, where the driver might normally need to look down and check gauges to confirm gear position or engine speed; the driver can instead focus on driving with eyes on the road.

4. Validation

Combining all of these sound technologies into a single package required extensive proving ground and public road driving to achieve the vehicle sound concept for each system and confirm the systems work together in harmony. Prototype vehicles were tested and tuned in the numerous situations a customer will drive their NSX. This included everything from race tracks and highways to the winding canyon roads of Malibu, the Autobahn of Germany and winding mountain roads of northern Italy and Austria.

The electronic controls for each sub-system are tuned individually to match the IDS mode intended driving conditions shown in Fig. 5. The quiet platform of the NSX enables the sport sound devices to create a wide dynamic range of sound, which can be appreciated by the driver as the sound responds directly to his or her inputs. The addition of each sound device contributes a unique flavor, combining to make the NSX character sound. Figure 26 shows the contribution of each system to the cabin sound during acceleration compared to the original NSX. Engine noise in the cabin was reduced compared to the

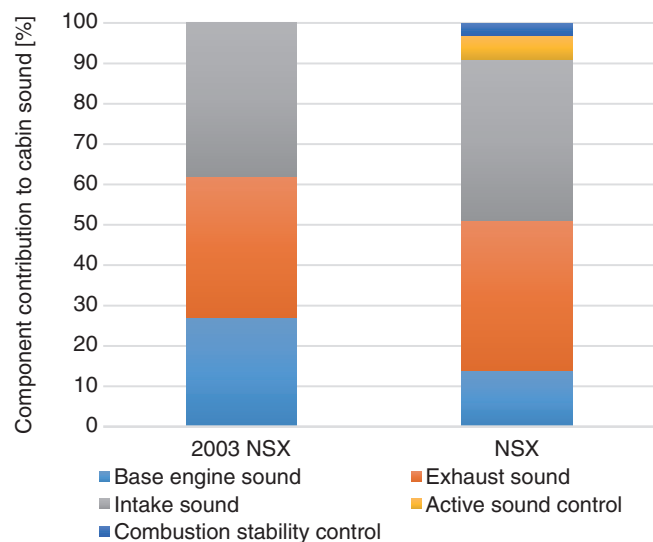


Fig. 26 Component contribution to engine sound during acceleration

original NSX in order to meet the Quiet mode concept, and additional technologies were added to achieve 2015's sports car market competitiveness. The contribution of intake and exhaust systems to the overall cabin sound during acceleration remains similar to the original NSX, realizing a balance that celebrates the legacy of NSX sound.

Figure 27 shows the final sound performance of the mass production spec vehicle during full throttle acceleration in third gear. The sound performance matches the original sound concept, with sound levels that are distinctly different between IDS modes. The frequency contribution of each device is shown in Fig. 28. Each of the systems contribute to a different frequency range and combine to create a unique aural spectrum which achieves the sound concept components explained in Table 3.

Figure 29 shows the final sound quality achievement of the mass production spec vehicle during full throttle acceleration in third gear. The high number of orders is achieved by the combination of ISC, AEV, and ASC systems, combining harmonic and guttural modulation sound into a single package.

Figure 30 shows how the NSX bridges the gap between sedan and super-car, accomplished through multiple personalities, available for the driver to access with the flick

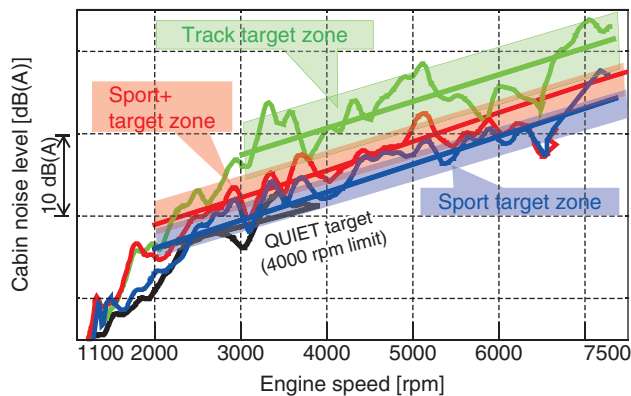


Fig. 27 Final acceleration sound in each IDS mode

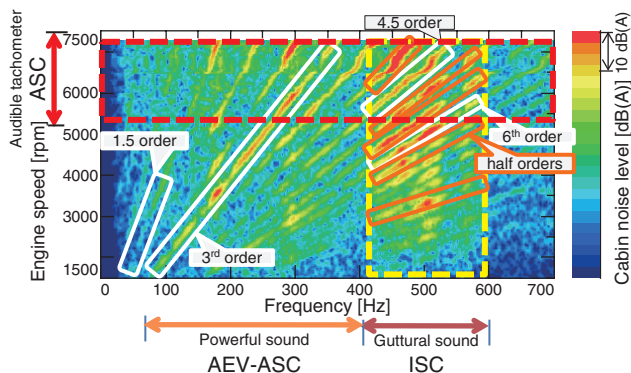


Fig. 28 Final sound achievement frequency content

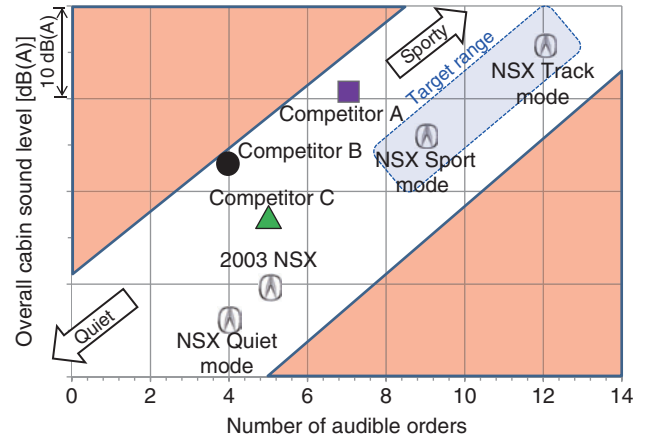


Fig. 29 Final cabin sports sound achievement during full throttle acceleration

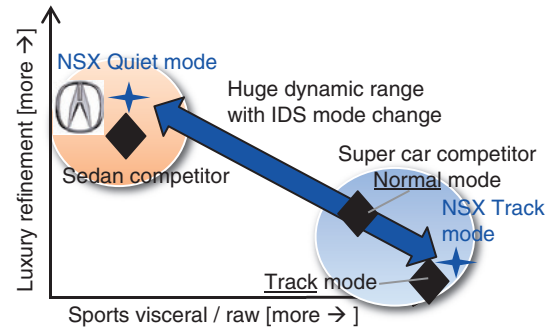


Fig. 30 Final sound achievement competitiveness

of a knob. This brings a truly New Sports eXperience, a no compromise super sports car, one that can be comfortably driven every day, over short or long distances, spirited canyon road driving, or even on the racetrack.

5. Summary

The system discussed in this paper was developed to realize the potential of the next-generation sports car with a New Sports eXperience. The system integrations create strong emotional sound connected with acceleration feeling, along with functional sound feedback to the driver. Additionally, the system supports the IDS concept to permit the driver to have direct control of the cabin sound experience.

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Materials Research Method using Smart Materials Informatics

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ABSTRACT

The achievement of outcomes through the use of informatics techniques employing big data is today being reported. However, because of the lack of opportunities to obtain high volumes of data (upwards of several thousands or several tens of thousands of data), these techniques still present issues in terms of practical applicability. In the case of materials in particular, because it is necessary to evaluate characteristics following the creation of test pieces, obtaining upwards of several thousands or several tens of thousands of data is unrealistic. A Smart Materials Informatics method that would make it possible to make predictions based on even a limited amount of data was therefore defined. Study of the amount of data necessary for predictions showed that predictions could be made using only several tens to hundreds of data. The use of the new method in studying functional materials (catalyst, magnet, and thermoelectric materials) made it possible to identify materials possessing characteristics suited to their respective purposes in a short period of time, demonstrating the practical applicability of the method.

1. Introduction

As the process of economic globalization continues and international competition between automakers intensifies, there is increasing demand for manufacturers to enhance their research and development capabilities in the area of materials⁽¹⁾. Given that they form the components used in automobiles, the characteristics of the materials employed have a tremendous effect on various automotive specifications and design elements. The creation of new high-performance materials is therefore highly significant, producing fundamental changes in automotive performance. The approach to seeking new materials, i.e. the methodology employed in materials research, is an important element in this creation of new materials. Empirical methods (study of the correlation between basic elements such as electronegativity and ion radius and material characteristics, theoretical consideration of the principles by which specific functions are realized, formulation and testing of hypotheses) have up to the present contributed to the creation of large numbers of new materials. However, in the case of functional materials like catalyst, magnet, and thermoelectric materials, elements that are challenging to observe, such as crystal structure and electrons, affect material characteristics in complex ways. As a result, even if the principle by which a characteristic is realized is understood, it may not be possible to easily

propose a method of manifesting this principle. Because of this, the search for functional materials has often involved a repeated process of creation and testing of materials. This method of seeking new materials entails repeated procedures including preparation of starting materials, formulation of materials, manufacture of test pieces, and evaluation of characteristics. This process is inefficient, in addition to involving considerable time and expense.

At the same time, computers have undergone a marked evolution. The evolution has been exponential; over the course of several decades, the power of CPUs has increased approximately 300,000-fold, and the storage capacity of computer memories by approximately one million times. Software technologies such as machine learning have also evolved with these evolutions in computer capacity. Machine-learning technologies are evolving that make it possible to predict unknown data with a high degree of accuracy by learning known data. Against this background, the prediction of material characteristics using materials informatics (MI) methods representing a fusion of materials science and information science is attracting attention as a means of reducing the necessity for the creation of test materials and the evaluation of material characteristics⁽²⁾. Between several hundreds and several tens of thousands of data are normally used as the training data for these methods. In the search for new materials for automotive applications, however, data is initially obtained by

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formulating materials and evaluating their characteristics. Given this, increasing the number of data is inherently not an easy task. While MI methods may be effective, in cases such as these there is almost no opportunity to put them to practical use. Realizing the ability to make predictions based on a limited number of data is therefore an important focus.

Designating a method of making it possible to predict the performance of materials based on limited data Smart Materials Informatics (SMI), the research discussed in this paper therefore attempted to determine the minimum essential number of data necessary to help enable the making of predictions.

2. Study of Essential Number of Data for Prediction

As the response of target variables to explanatory variables becomes increasingly complex, the number of data necessary for accurate predictions necessarily increases. In studying the essential number of data for prediction, it is therefore important to understand the nature of data concerning functional materials.

Figure 1 shows an example of a response surface for performance P of a functional material in relation to explanatory variables x_1 and x_2 .

In the case of the nonlinear response surface shown in Fig. 1, when the explanatory variable x_1 is small, the increase of x_2 in relation to P is positive, as shown by the curve L1; when the explanatory variable x_1 is large, the increase of x_2 in relation to P is negative, as shown by the curve L2. In the case of systems in which target variables respond nonlinearly in relation to explanatory variables in this way, it is not easy to establish guidelines for the process of seeking new materials. In addition, the level of challenge involved increases when the number of explanatory variables increases. Catalyst, magnet, and thermoelectric functional materials, in which nonlinearity is comparatively strong, were therefore selected to provide case studies of the use of SMI methods in seeking new materials. It would

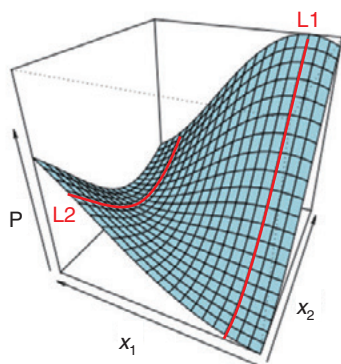


Fig. 1 Example of response surface

therefore be important to utilize data correlating with the nonlinearity displayed by these functional materials in studying the necessary number of data for prediction.

2.1. Relationship between Number of Data and Prediction Error

A specific number of data would be necessary in studying the practical applicability of the SMI method. It would not be practical to actually manufacture materials and evaluate their characteristics in order to acquire these data for the purposes of the study. Formulas making it possible to calculate target variables from arbitrary explanatory variables were therefore employed to obtain large numbers of data, and the non-experimental data generated by these formulae were utilized in studying the practical applicability of the SMI method. A non-parametric machine learning method able to comparatively freely generate regression equations in response to data was selected as the prediction model. The hyper-parameters of the selected machine learning method were tuned to help ensure a high level of predictive accuracy even when a limited number of data were employed. In order to further increase predictive accuracy, the data were subjected to preprocessing using principal component analysis to help ensure the orthogonality of the explanatory variables. In addition, functions compatible with nonlinearity were selected as the functions for use in the machine-learning program.

Internal validation was employed as an indicator of the predictive accuracy of the prediction model, making it possible to determine predictive accuracy for data other than training data, even when there were a limited number of data. Leave-one-out cross-validation (LOOCV) was selected as the form of internal validation. In addition, in order to make it possible to determine the relationship between the number of data and predictive error, a six-dimensional data set composed of five explanatory variables and target variables was formulated using the formulas for the generation of non-experimental data mentioned above. Jackknife sampling, a random sampling method that does not allow duplication, was applied to this data set for the various numbers of data, and average values for LOOCV when multiple repetitions were conducted were calculated. The number of jackknife samples taken was restricted to 100 from considerations of computation time and accuracy. Figure 2 shows the results.

As Fig. 2 indicates, the trend of the results is for LOOCV to become lower as the number of data increases. This is considered to be due to the fact that as the number of data increases, the volume of information also increases, increasing the accuracy of the prediction model. When the number of data increases over 50, LOOCV becomes less prone to reduction, and the amplitude of the decline in LOOCV narrows. This is considered to be because the representational ability of the model is reaching its limit, and the effect of the increase in the number of data in relation to LOOCV is becoming reduced. The relationship

of LOOCV to the number of data shown in Fig. 2 will be termed the LOOCV profile below.

Mathematically speaking, the LOOCV profile is known to be inversely proportional to the number of data. However, while the LOOCV profile shown in Fig. 2 is basically inversely proportional to the number of data, during the initial period when the number of data is low, the LOOCV fluctuates wildly, and its behavior in this domain is not theoretically explained. It was therefore considered that clarifying the causes of this fluctuation would be important in helping to enable the use of a valid SMI method.

2.2. Study of Minimum Necessary Number of Data

The causes of the extreme fluctuation in LOOCV were studied to help enable the use of a valid SMI method. In order to realize a visual understanding of the behavior of the prediction model in the low-data-number domain, the number of dimensions of the formulas for the generation of non-experimental data mentioned in Section 2.1. was reduced, and a three-dimensional non-experimental data set consisting of target variables (TV) in relation to two explanatory variables (x_1 and x_2) was formulated. The lengths of the maximum and minimum values of the explanatory variables x_1 and x_2 in this non-experimental data set were each divided into 100 uniform intervals, and TV were calculated in relation to 100,000 grid points representing all the combinations of x_1 and x_2 , creating a 100,000-point data set. Figure 3 shows a three-dimensional response surface generated for this data set.

The response surface shown in Fig. 3 is a theoretical surface produced by the formulas for the generation of non-experimental data. The response surface shown in Fig. 3 is calculated when the error of the machine learning prediction model is zero.

Next, the lengths of the maximum and minimum values of x_1 and x_2 were divided into three, four, five, and six uniform intervals, and non-experimental data consisting of explanatory variables and target variables for 9, 16,

25, and 36 points, representing all the combinations of x_1 and x_2 , were learned, and prediction models for each data set were formulated. The explanatory variables from the 100,000-point data set were input to the prediction models, and predicted values for target variables were calculated for each model. Figures 4(a), (b), (c), and (d) show the resulting response surfaces. As these results show, as the number of data increases, the predicted surfaces more closely approach the theoretical surface shown in Fig. 3. The change in the response surfaces with the increase in the number of data is particularly great in the low-data-number domain. It is considered that because the number of training data in this domain is low, the prediction model is unable to represent the entire shape, and a slight increase in the number of data results in a major change in the response surface. It was therefore considered that, despite the fact that LOOCV values are calculated low in the domain in which there is a major change in the shape of the predicted response surface, there is a strong possibility of randomness that negatively affects the reliability of the values, and they should not be employed as LOOCV values.

By contrast, it was considered that, because there was no major change in the shape of the response surfaces in the domain in which the number of training data is high despite the increase in the number of training data, and the prediction model was able to represent the entire surface, the reliability of the calculated LOOCV values was high.

Based on the above, the cause of the extreme fluctuations in LOOCV in the initial domain in which the number of data was low in the results of calculations of the LOOCV profile was the fact that the basic shape of the prediction model was not fixed in this domain. It was considered that if the number of data were increased up to the domain in which the fluctuations in LOOCV became low, the prediction model would capture the basic shape of the response surface, and the LOOCV values would be valid.

In addition, within the range of the non-experimental data employed, the number of data at which it becomes possible to apply the prediction method is between several

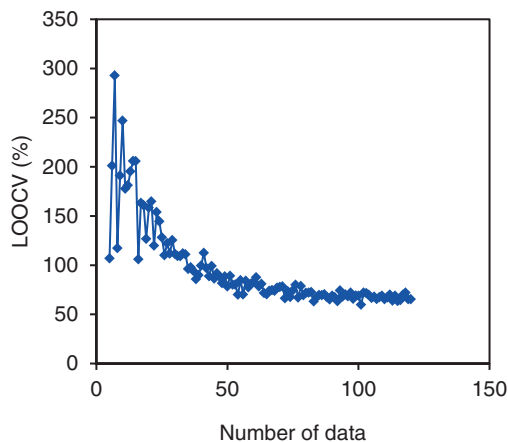


Fig. 2 Relationship between number of data and LOOCV

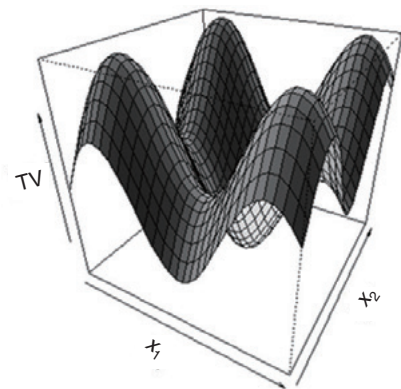
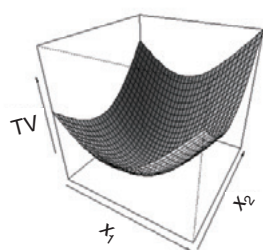


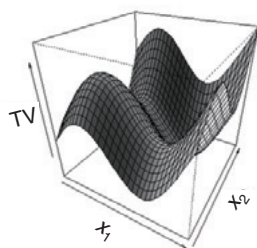
Fig. 3 Theoretical surface

tens and hundreds of data. With this quantity of data, it becomes possible to actually produce a material and evaluate its characteristics.

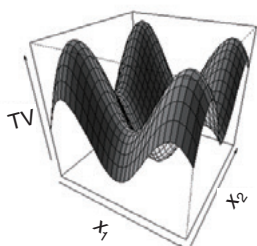
The studies discussed above verified that it would be possible to employ SMI methods to search for new materials. In order to determine whether SMI methods would be practically applicable, searches for new materials were conducted using the method in relation to the catalyst, magnet, and thermoelectric materials selected as case studies.



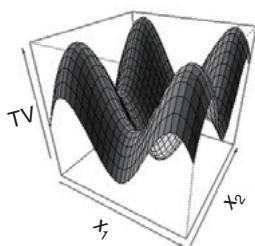
(a) Predicted surface: 9 training data



(b) Predicted surface: 16 training data



(c) Predicted surface: 25 training data



(d) Predicted surface: 36 training data

Fig. 4 Predicted surfaces for different numbers of training data

3. Study of Practical Applicability of SMI Method through Creation of New Materials

3.1. Catalyst Material

Exhaust gas conversion performance at the low temperatures at engine start is a particular demand on automotive three-way catalysts⁽³⁾. Three-way catalysts are made up of precious metals and a material that supports them. These precious metals initially exist in the form of particles of around several nm, but the particles aggregate in the high-temperature automotive exhaust gas environment, and these aggregates grow to between several tens and several hundreds of nm.

The support material was focused on in an attempt to control this aggregation of precious metal particles in high-temperature environments. With element 1 termed A, element 2 termed B, and oxygen termed O, a comprehensive search for the elements A and B to be employed in a composite oxide ABO was attempted. Only a 1:1 ratio of A to B was considered. Figure 5 shows the prediction procedure of this catalyst material.

Element-specific property data for elements A and B when they exist separately, including primary ionization energy and melting point, were considered to be explanatory variables, and the specific surface area of palladium (Pd) particles when supported by the material composed of elements A and B was the target variable. Given that it was necessary that elements A and B could be employed in an automotive catalyst, elements that are gaseous at ordinary temperatures and radioactive elements were excluded. When elements that could not be employed were excluded, the number of potential elements was 64 for both elements A and B. This produced 2080 possible combinations of A and B, including the combination A=B. A and B were varied to produce 52 ABO support materials. The specific surface area of Pd particles supported on these materials was measured, generating 52 training data. LOOCV was calculated for this number of data to determine whether it fell within the scope for application of the prediction model using the method discussed in Chapter 2. Figure 6 shows the results.

Based on Fig. 6, it was judged that at 45 data and above, the fluctuations in LOOCV began to abate, and at 52 data the fluctuations tapered off. The number of data was

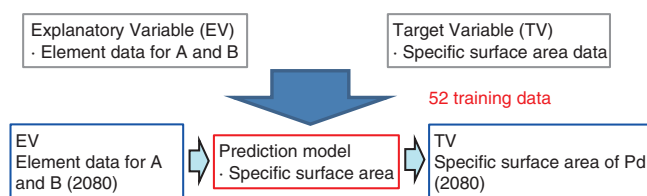


Fig. 5 Procedure for prediction of specific surface area of Pd

therefore in the domain in which LOOCV in the prediction model was reliable, and it was judged that the prediction results could be utilized in selecting a catalyst support material. The explanatory variables for the abovementioned 2080 combinations of elements A and B were input to the prediction model formulated using these 52 data, and the specific surface area of Pd particles supported by materials composed of elements A and B was predicted. Combinations of elements A and B for which high specific surface areas were predicted were given priority for selection. Catalysts were created using these combinations, and the specific surface area of Pd particles supported by each support material was measured using CO gas absorption measurements. Table 1 shows the measurement results.

As a result, it was possible to create four support materials (I, II, III, and IV) on which the specific surface area of supported Pd particles was higher than the specific surface area when a mass-production support material was utilized. A supported Pd catalyst was created using support material I, the material producing the highest specific surface area in supported Pd particles, and its conversion characteristic following aging treatment was evaluated using model gases. Figure 7 shows the results.

As Fig. 7 indicates, the catalyst employing the developed support material displayed a superior conversion characteristic to a catalyst employing the mass-production support material in relation to CO, HC, and NO_x. This was considered to be because the supported Pd particles were small and dispersed and a high specific surface area had been maintained, even following aging. Pd support material I was therefore analyzed using a scanning electron microscope (SEM), with the results shown in Fig. 8.

The Pd particles supported by support material I were around several nm. Energy-dispersive X-ray spectroscopy showed that the particles were elemental Pd. These results indicate that a catalyst material had been developed that supported Pd particles with a high specific surface area

Table 1 Summary of catalysts with higher specific surface area of Pd particles than conventional support material

Support material	Element A	Element B	Specific surface area of Pd particles
I	$\alpha 2$	$\beta 2$	1.39
II	$\alpha 1$	$\beta 1$	0.21
III	$\alpha 2$	$\beta 3$	0.13
IV	$\alpha 3$	$\beta 3$	0.34
Reference (Conventional support material)	Al	Al	0.11

even following aging, and that the SMI method employed in predicting the specific surface area of the Pd particles was valid. Support material I was selected based on these results, making it possible to create a supported Pd catalyst displaying superior performance to a catalyst using the mass-production support material.

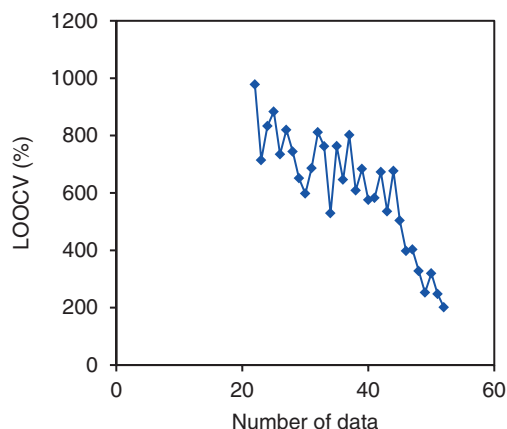


Fig. 6 Relationship between number of data and LOOCV when studying support materials

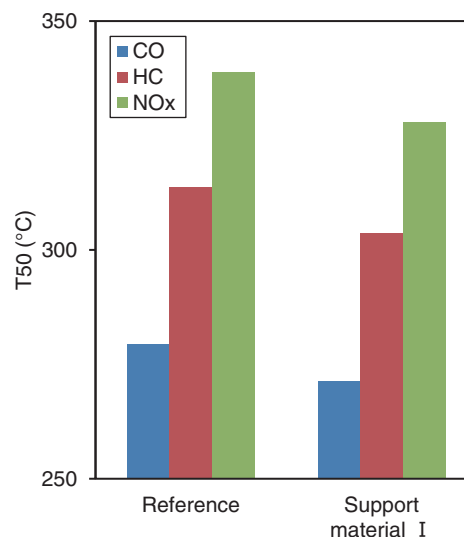


Fig. 7 Temperature for 50% conversion of individual gases

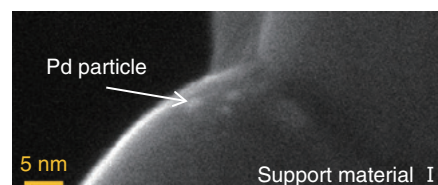


Fig. 8 SEM image of support material- I-with Pd particle

3.2. Magnet Material

Today, as automotive electrification proceeds, a higher level of performance is demanded from the magnets employed in drive motors. It is necessary for magnets employed in automotive use environments to maintain coercivity at high temperatures, and the realization of a fine structure in the magnet material is an important factor in achieving this goal⁽⁴⁾. Past studies have indicated that the two-stage low- and high-temperature processing undergone by hot-deformed magnets refines the structure of the material and renders it uniform⁽⁵⁾. The SMI method was therefore applied in relation to the parameters of the hot deformation process shown in Fig. 9, and a magnet material displaying a high level of performance was sought.

Figure 10 shows the procedure employed in applying the abovementioned SMI method.

The temperature (T1), amount of pressing (AP1), and velocity of pressing (VP1) employed in the first procedure and the temperature (T2) and amount of pressing (AP2) employed in the second procedure of the hot deformation process were considered explanatory variables, and the products of the magnetic flux density, Br, and coercivity, iHc, of magnets created while varying these explanatory variables were considered target variables. Twenty-eight types of training data were obtained. LOOCV profiles were calculated for these 28 types of training data, and it was verified that the extreme fluctuations in LOOCV had tapered off, and the data were within the domain in which results for LOOCV were reliable. The 28 types of training data were learned using a machine learning method compatible with nonlinearity, and a prediction model was formulated.

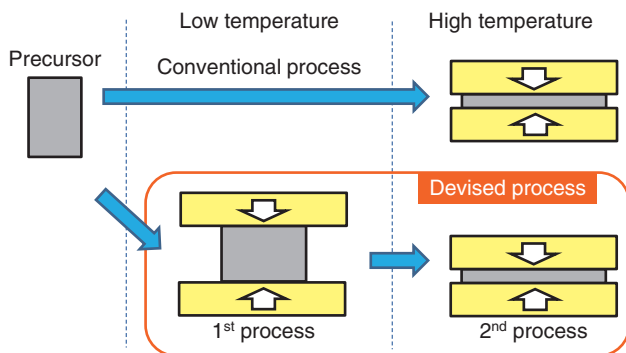


Fig. 9 Schematic image of hot deformation process

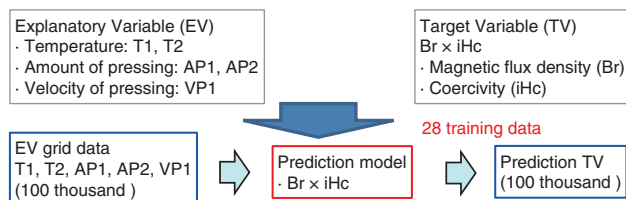


Fig. 10 Prediction procedure for magnet material

Table 2 Prediction result yielding highest level of magnet performance

TV (Br × iHc)	T1 (°C)	AP1 (%)	VP1 (-)	T2 (°C)	AP2 (%)
252	494	27.8	2.0E-05	700	20

100,000 hypothetical explanatory variables, representing all possible combinations when each explanatory variable was divided into 10 increments, were input into the prediction model, and 100,000 target variables were predicted. Table 2 shows the prediction results that yielded the highest level of magnet performance.

A magnet was created using the hot deformation process conditions shown in Table 2, and its performance was evaluated. Figure 11 shows the results, together with past results.

“Base” in Fig. 11 is the material employed when research on magnets produced using the hot deformation process commenced. “Y” is the magnet material obtained following approximately two years of research. When material Y was obtained, 28 data concerning the performance of magnets produced using different hot deformation conditions were obtained. A prediction model was formulated based on these 28 data, and the optimum conditions shown in Table 2 were predicted. The calculations for this prediction took around five minutes. A magnet was created using the conditions shown in Table 2, and magnet material X, which displayed superior performance to magnet material Y, was obtained.

The use of the SMI method made it possible to formulate a prediction model based on a limited number of data and determine the optimum conditions for the manufacture of a magnet material in a short period of time.

3.3. Thermoelectric Material

Half or more of the chemical energy of gasoline is exhausted as heat energy in automobiles. There are

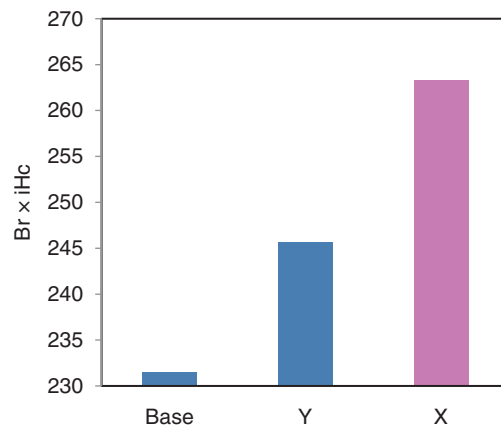


Fig. 11 Results of evaluation of magnet performance together with past data

expectations on thermoelectric materials to function as elements positioned between high- and low-temperature sections, able to recover heat energy by statically converting it into electrical energy⁽⁶⁾. Theoretically, the efficiency of thermoelectric materials in recovering heat energy increases in proportion to the temperature difference, but the temperature resistance of the materials represents an issue. The realization of increased temperature resistance through modification of the elemental composition of a thermoelectric material was therefore investigated. Figure 12 shows the crystal structure of skutterudite, the thermoelectric material selected for study.

The thermoelectric material shown in Fig. 12 is made up of the elements lanthanum (La), iron (Fe), cobalt (Co), and antimony (Sb). The thermoelectric characteristic, η , and the level of temperature resistance of the material are known to change with changes in these constituent elements and the concentration of the carrier.

A prediction model was formulated using the elemental composition shown above, and a study was conducted to seek a thermoelectric material displaying heat resistance while maintaining the same thermoelectric characteristic. Figure 13 shows the prediction procedure.

The quantities of La, Fe, Co, and Sb in the thermoelectric material and the carrier concentration were considered to be explanatory variables, and the thermoelectric characteristic, η , and the level of temperature resistance of the material were considered to be the target variables. The thermoelectric characteristic, η , and the level of temperature resistance of the material were measured when the explanatory variables, the material's elemental composition and carrier concentration, were changed as considered appropriate, and 19 types of training data were

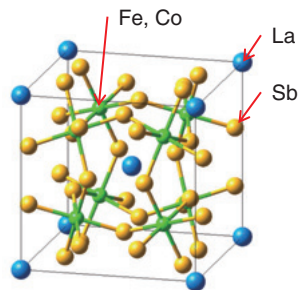


Fig. 12 Illustration of Skutterudite crystal structure

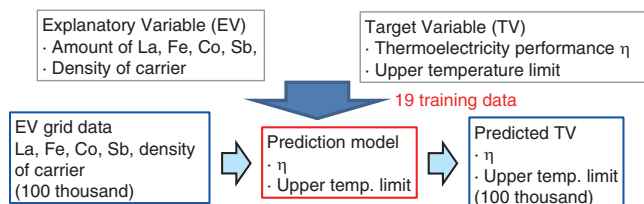


Fig. 13 Prediction procedure for thermoelectric material

obtained. The LOOCV profiles were calculated, and it was verified that the data were within the domain in which the LOOCV values could be considered reliable. The training data were employed in the formulation of two prediction models, for the material's thermoelectric characteristic, η , and its level of temperature resistance. 100,000 grid data, representing all possible combinations when each explanatory variable for each of the two prediction models was divided into 10 increments, were input into the prediction models, and 100,000 target variables were predicted. Figure 14 shows a Pareto diagram of the results.

Two plots for which the temperature resistance of the material was high at a specific thermoelectric characteristic, η , or above were selected from the prediction results, and a thermoelectric material was manufactured under conditions that made it possible to reproduce the selected elemental composition and carrier concentration. The thermoelectric characteristic, η , and level of temperature resistance of the material were then measured. Results for thermoelectric characteristic, η , and level of temperature resistance matched within the error bar, indicating that it had been possible to create a thermoelectric material possessing a higher level of temperature resistance than conventional materials.

These results indicate that the use of the SMI method in relation to characteristic values that are in a trade-off relationship, such as the thermoelectric characteristic, η , and the level of temperature resistance in this study, makes it possible to develop materials that respond to specific needs with pinpoint accuracy.

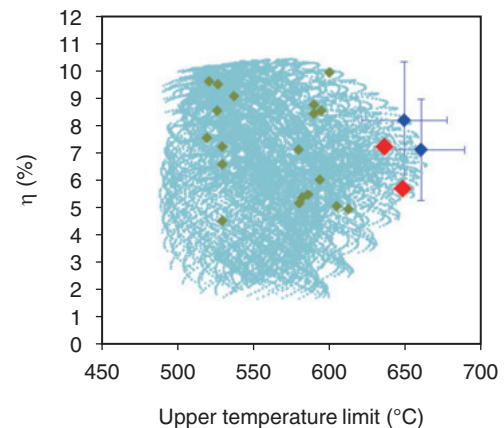


Fig. 14 Pareto diagram of η and upper temperature limit

4. Conclusion

A study of the behavior of LOOCV in relation to the number of data when applying the SMI method to material data demonstrated that the number of data that would be

effective in making predictions was between several tens of data and 100 data, indicating that the method could be employed in seeking new materials. The application of the SMI method to catalyst, magnetic, and thermoelectric materials demonstrated that the method made it possible to develop new materials in line with the aim for the material, for example boosting characteristics or increasing temperature resistance, within a short period.

While predictions were made only for functional materials in this study, the scope of applicability of the method is not restricted to these materials. If training data made up of target variables and explanatory variables that adequately explain the target variables is available, the method makes it possible to predict the results of tests not yet performed.

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Efficient Vehicle-level Coordination in Functional Safety Process for a Whole Vehicle

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ABSTRACT

On-board logical electrical and/or electronic systems are required to comply with functional safety standard ISO 26262. Functional safety activities involve implementing a standard-compliant process as a measure to help prevent any physical injury or damage to human health that may be caused by system malfunctions. In order to efficiently comply with ISO 26262 in automotive development relating multiple systems intricately, the personnel working on each system need to rationally coordinate and implement the functional safety activities for a whole vehicle. Honda has therefore added to the functional safety process its own process of coordinating functional safety requirements. The process involves investigating how the systems of the whole vehicle relate at the vehicle level, formulating methods to allow necessary and sufficient boundaries of functional safety activities to be set, conducting impact analyses cooperatively by using the system interfaces as guides, and coordinating the functional safety requirements. This eliminates the overlap in impact analysis scope, improving accuracy for system internal impact analyses. Further, adding the proprietary process to link systems has resulted in exhaustive implementation of high-precision impact analyses at the vehicle level, in turn forming a vehicle-level functional safety process for the whole vehicle capable of appropriately deploying functional safety requirements.

1. Introduction

With the number and complexity of electronic control systems on the rise in modern vehicles, functional safety standard ISO 26262 (Fig. 1) was formulated, officially going into effect in November 2011⁽¹⁾. Generally, on-board electrical and/or electronic (E/E) systems use equipment such as switches, sensors, and communication devices to digitize and input driver operation status and vehicle status, and then use the input information to control one or more systems and perform functions. If a system malfunctions, the driver or system is unable to control the mechanisms as intended, resulting in hazards (Fig. 2).

With ISO 26262, in addition to intrinsic safety, development is required to determine the causes for any failure in an E/E system causing hazards resulting in physical injury or damage to human health, and to apply measures to reduce hazard risks to acceptable levels. These activities are required for each item, consisting of a system or array of systems that implement a function at the vehicle level. Before ISO 26262 went into effect, systems to be developed were mainly managed by electronic control units (ECU) and specific functional units. Thus, hazard analysis

and risk assessment (H&R) for items has been performed for each system, with functional safety requirements assigned to each system before performing the activities in ISO 26262 from Part 4, “Product development at the system level,” to the end (Fig. 1). As ISO 26262 (Fig. 1) is a waterfall-type V-model development process, any

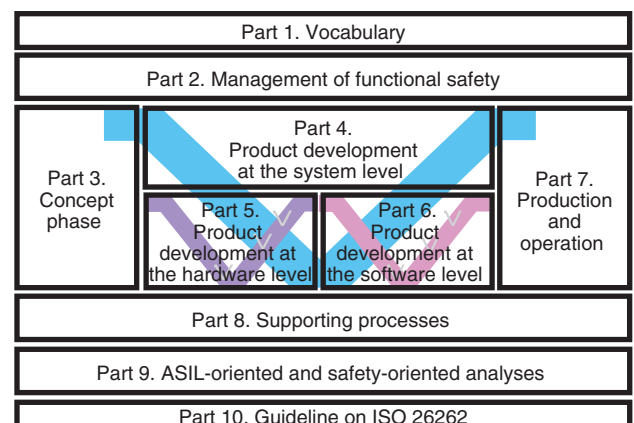


Fig. 1 Overview of ISO 26262

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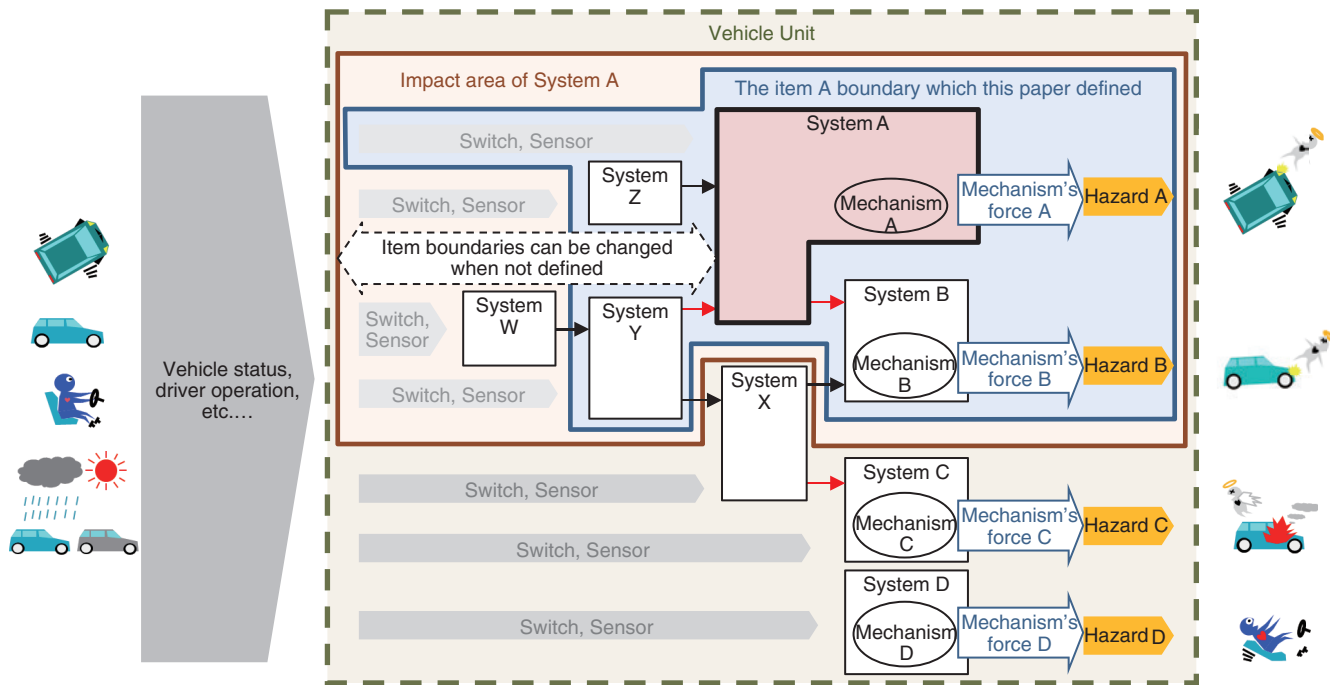


Fig. 2 Relationship among systems, item boundaries, and vehicle level

omissions or oversights in design and analysis performed in the upstream part of the process will require many man-hours to rework, in turn increasing work time in the downstream parts of the process. Thus, how precisely the impact analysis can be completed in the upstream part of the process with well-organized item boundaries is the key to efficiently achieving functional safety at the vehicle level. However, while the functional safety process is defined in ISO 26262 for individual items, it is not clearly defined for coordinated development of multiple items at the vehicle level. Honda has therefore developed a functional safety process for a whole vehicle. The process allows functional safety requirements to be deployed appropriately for the whole vehicle in order to resolve the aforementioned issues in ISO 26262 Part 3, “Concept phase,” corresponding to the upstream part of the functional safety process.

2. Development Goals

In ISO 26262-compliant development, functional safety activities are required for the items. However, given that different system personnel may define item boundaries differently, rework due to omissions and oversights as well as waste due to duplicate work will occur at the vehicle level unless item boundaries are set according to fixed rules. The completeness and accuracy of impact analysis in the upstream part of the process will impact efficiency downstream. If items are not properly coordinated upstream in the process, extra time will be required to verify that the analysis scope is complete even if complementary functional

safety activities are performed in the other systems. Further, having personnel for the individual systems set the scope for analysis any larger than their own systems is inefficient as it will result in personnel for multiple systems analyzing duplicated scope. Moreover, given how carry-over has been integrated into the development scheme to help reduce man-hours, the process presented in this exercise must also allow reuse of development resources.

In light of the above points, in order to perform highly precise functional safety activities efficiently at the vehicle level, the goals for the functional safety process for the whole vehicle presented in this exercise were determined to be as follows:

- (1) Clearly specifies the paths of transmission (interface) between systems for hazard impacts and functional safety requirements.
- (2) Creates a scheme for deciding item boundaries so that unnecessary overlap in impact analysis scope is minimized.
- (3) Creates a functional safety process for a whole vehicle so that impact analyses for hazards can be coordinated between systems, and that functional safety requirements can be assigned and coordinated.

3. Functional Safety Process for the Whole Vehicle

A functional safety process for the whole vehicle was developed by adding the processes linking the various systems to the existing functional safety process based upon

the schemes for interface classification and hazard impact transmission described below.

3.1. Interfaces Clarified

3.1.1. Interface classifications

This process is characterized by using interfaces as guides for activity. These interfaces are classified into three categories: mechanism's force, operation requests, and information. Mechanism's force interfaces communicate physical effects, and operation request interfaces manipulate the mechanism's force of other systems. Meanwhile, interfaces are classified as information interfaces when it is unclear whether it has any direct impact upon a hazard.

3.1.2. Scheme for hazard impact transmission

Examples of how hazard impacts are transmitted are given in Figs. 3(a)-(d). In the figures, systems are represented as squares, mechanisms as ellipses, information interfaces as black arrows, operation request interfaces as red arrows, mechanism's force interfaces as blue-gray arrows, and hazards as orange arrows. For example, in Fig. 3(a), a hazard occurs when System E erroneously controls the mechanism due to an internal malfunction in System E, with the mechanism generating an abnormal force. Meanwhile, in Fig. 3(b), a hazard occurs despite System F not having a mechanism when System F outputs an operation request to System G, impacting System G's mechanism. Next, in Fig. 3(c), the impact of a hazard caused by the mechanism of System J is transmitted all the way to the switches, sensors and other vehicle-level input devices via information, System I and System H. Possible means of reducing the impact of such a hazard include implementing a safety mechanism in the system located between the hazard and vehicle-level input devices [Fig. 3(d)].

The standard architecture shown in Fig. 5, with the interfaces clearly specified between the systems, was created based on the above interface classifications and schemes for hazard impact transmission. Using this standard architecture with the interfaces as guides, a process was developed in which system personnel perform their impact analyses cooperatively and coordinate functional safety requirements. This helps system personnel to easily identify what risks exist in their systems, as well as the risk source and scale, facilitating appropriate assignment of functional safety requirements.

3.2. Overlap Reduced in Impact Analysis Scopes

3.2.1. Item boundary conditions

As illustrated in Section 3.1.2., systems are thought to output mechanism's force and operation requests that directly result in hazards, or to indirectly cause hazards through information. As safety measures can be implemented across all paths resulting in the hazards if functional safety requirements are properly assigned between adjacent systems, item boundary conditions have

been set as follows.

When interfaces are input, the systems and their mechanisms linked via those interfaces are included in the item scope. Meanwhile, when interfaces are output, the systems and their mechanisms linked via those interfaces,

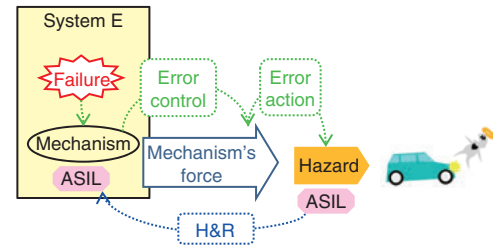


Fig. 3(a) Hazard occurrence

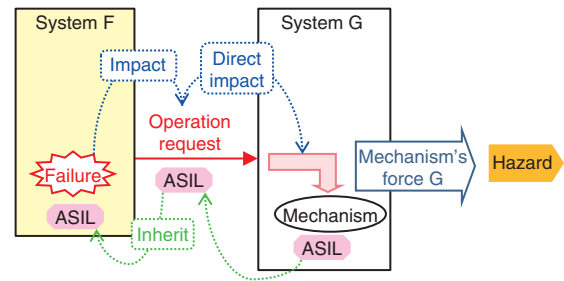


Fig. 3(b) Hazard due to operation request

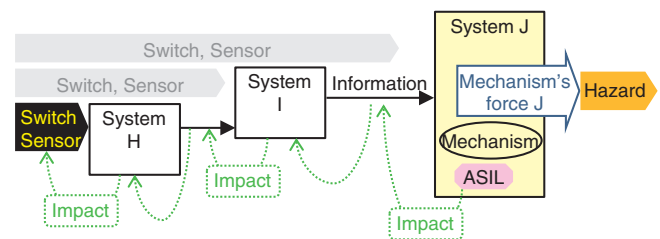


Fig. 3(c) Transmission path of hazard impact

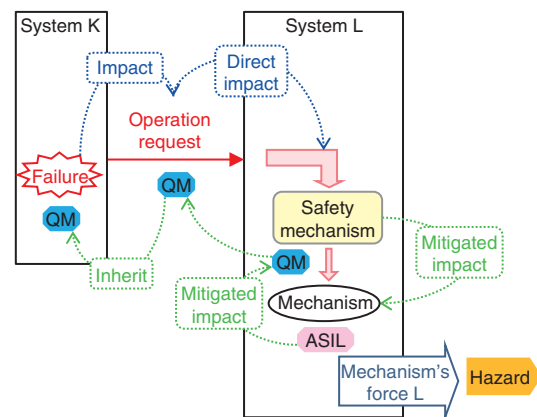


Fig. 3(d) Mitigation of hazard impact by safety mechanism

mechanism's force, and related hazards are included in the item scope.

3.2.2. Preconditions

Preconditions for the process are as follows. The scale of hazard impact as transmitted via each interface is expressed in terms of ASIL. In addition, personnel are posted for each system, with the personnel implementing safety measures for the functional safety requirements assigned to their systems, working as described in ISO 26262 from Part 4, "Product development at the system level," to the end (Fig. 1).

The standard architectures were created based on the above item boundary conditions and preconditions, with item boundaries organized for a whole vehicle. With unneeded overlap in impact analysis scope reduced by developing a process for system personnel to perform their impact analyses cooperatively using these standard architectures, system personnel can concentrate on analyzing the impacts within their own systems.

3.3. Functional Safety Process for the Whole Vehicle

System personnel and Process Manager work together to implement Processes (1)–(6) as given below. System personnel perform Processes (2)–(4) and (6) for the item of each system as the scope for their functional safety activities. Meanwhile, the Process Manager manages the progress for each system and implements the functional safety activities for the whole vehicle (Fig. 4).

Process 1: Process Manager creates and distributes basic architectures for each item in the vehicle under development

Process 2: System personnel create preliminary architectures

Process 3: System personnel perform hazard impact analyses

Process 4: Functional safety requirements coordinated between system personnel

Process 5: The Process Manager updates standard architectures

Process 6: System personnel perform functional safety activities in ISO 26262 from Part 4, "Product development at the system level," to the end (Fig. 1)

Processes (1)–(6) are described in detail below.

3.3.1. Process Manager creates and distributes basic architectures for each item in vehicle under development

The Process Manager investigates on-board vehicle systems and their related systems and applies the item boundary conditions given in Section 3.2.1. to create the standard architecture (Fig. 5). At the onset of vehicle development, the Process Manager checks the on-board systems for the vehicle under development, creates basic architectures from the standard architecture, restricting them to the necessary systems, and distributes the basic

architectures to personnel for each system (Fig. 6).

3.3.2. System PIC creates preliminary architecture

Based on the basic architecture, a person in charge of the system (system PIC) coordinates with the personnel from related systems to specify which interfaces will be used (or not used) for the vehicle under development and creates a preliminary architecture for his/her own system (Fig. 7). When creating the preliminary architecture, the system PIC coordinates with the personnel from related systems if the interfaces and related systems are insufficient, and requests the Process Manager to change the preliminary architecture.

The Process Manager will then update the preliminary architecture for the system to be changed.

3.3.3. System PIC performs hazard impact analysis

For systems outputting their mechanism's force, the system PIC performs H&R for his/her own system and evaluates the ASIL(s) for the hazard(s) of the mechanism's force. For the system with an operation request that acts upon the mechanism's force with the ASIL being input, the system PIC will assign the ASIL to the operation request input to his/her own system and inform the PIC of the system outputting the operation request. Next, the PIC of the system outputting the operation request confirms the reported ASIL and assigns the reported ASIL to the operation request being output (Fig. 8).

The system PIC then produces a concept design for his/her own system with the assumed allocation of safety mechanism elements and performs an impact analysis within the system. The PIC assigns ASILs to the interfaces implemented in the system and informs personnel for related systems (Fig. 9). Finally, the PIC will check the ASILs assigned to the interfaces in his/her own system to confirm the validity of the safety mechanisms as assumed

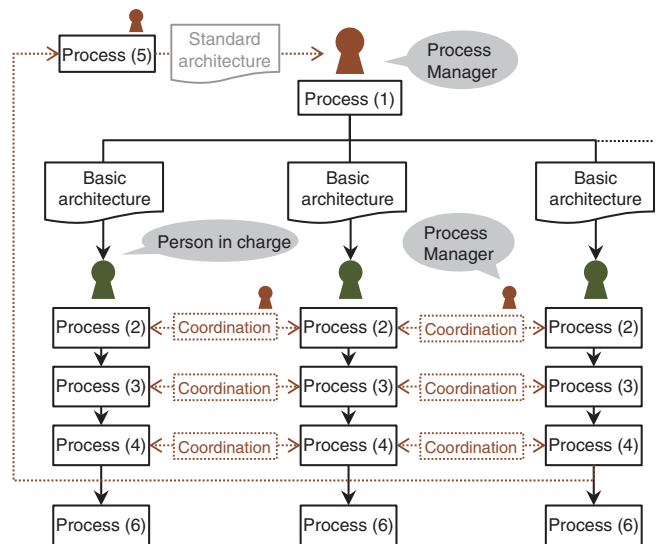


Fig. 4 Functional safety processes at vehicle level

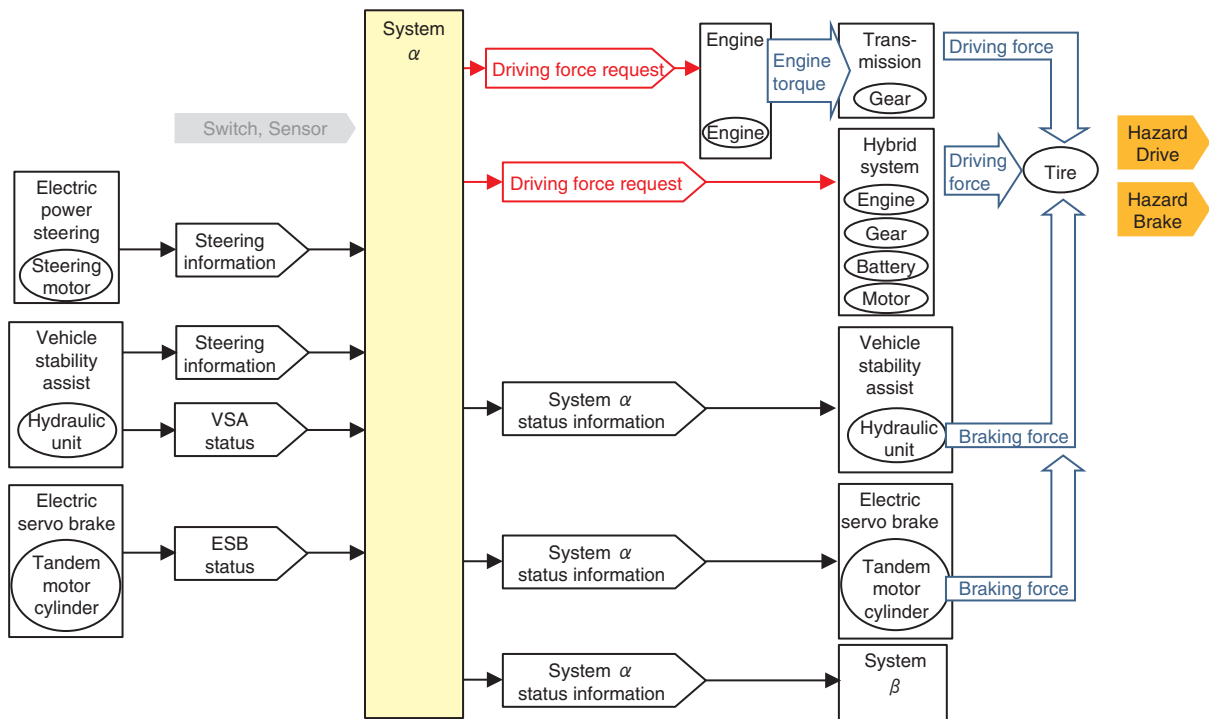


Fig. 5 Standard architecture

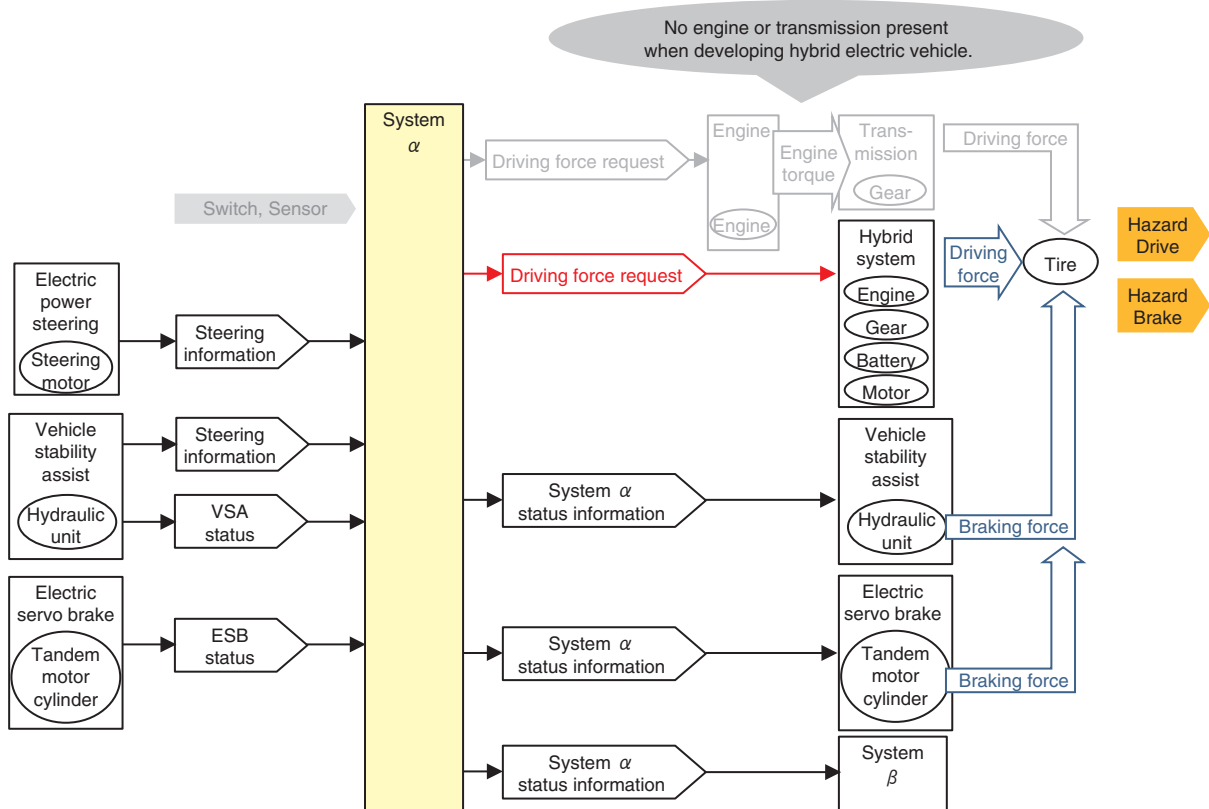


Fig. 6 Basic architecture

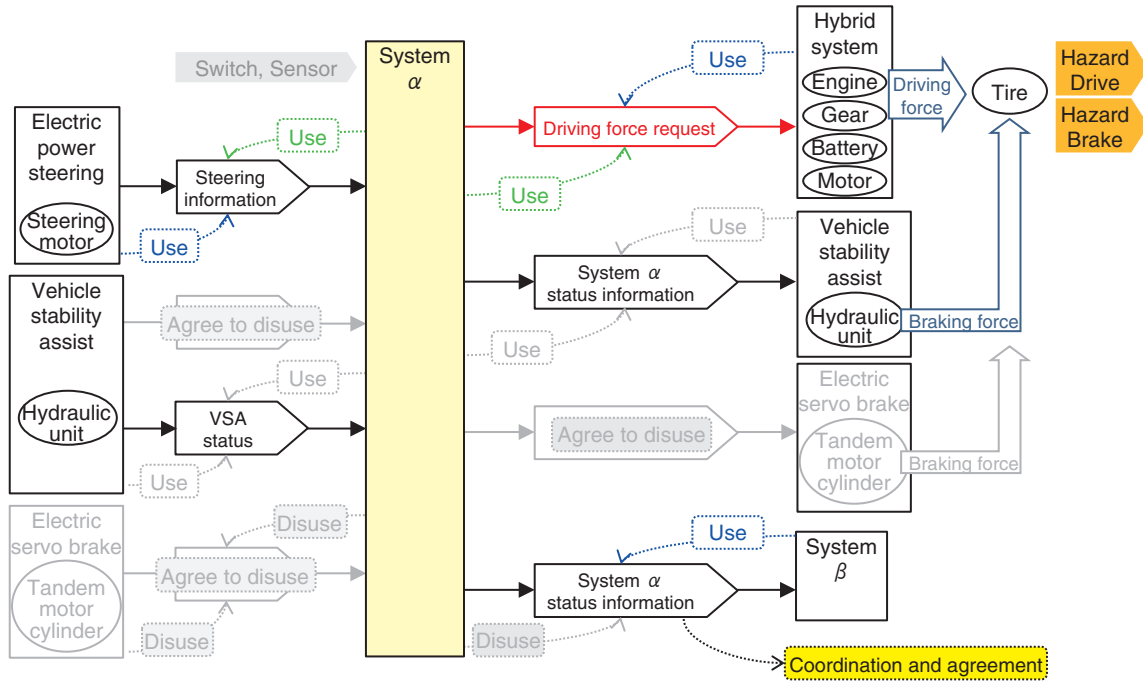


Fig. 7 Process of creating preliminary architecture

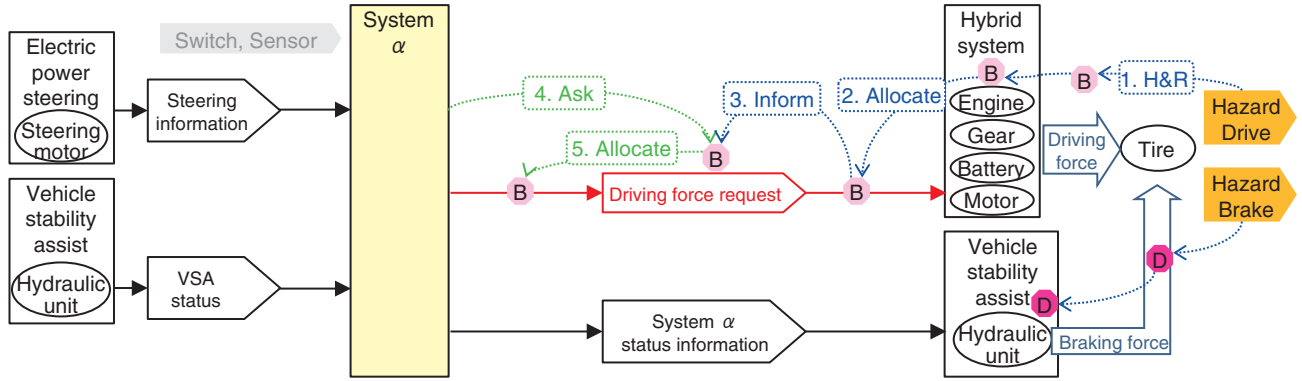


Fig. 8 Hazard analysis and risk assessment of mechanism's force and impact analysis of operation request

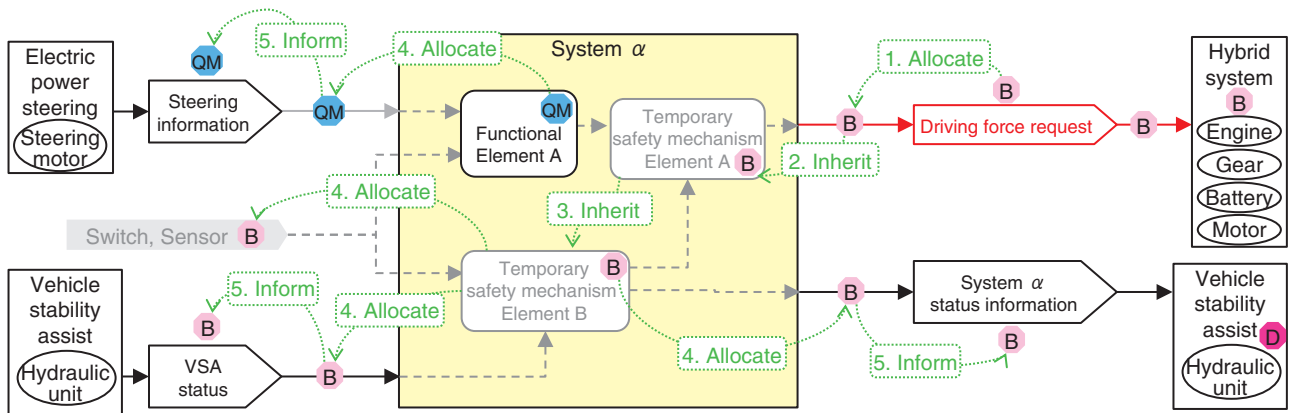


Fig. 9 Concept design assuming allocation of safety mechanisms, and impact analysis between systems

in the concept design. If, for example, a higher ASIL than expected has been assigned, concept design and impact analysis is redone (Fig. 10).

3.3.4. Functional safety requirements coordinated between system personnel

The system PIC assigns the functional safety requirements to the system interfaces with ASILs assigned. When safety mechanisms for those interfaces are to be

implemented by related systems, the PIC assigns and coordinates the functional safety requirements and agrees upon their details with personnel for the related systems (Fig. 11). When confirming and coordinating functional safety requirements assigned from related systems, the system PIC is to agree upon functional safety requirements with personnel from the related systems when such requirements are assigned to the system interfaces from the related systems (Fig. 12).

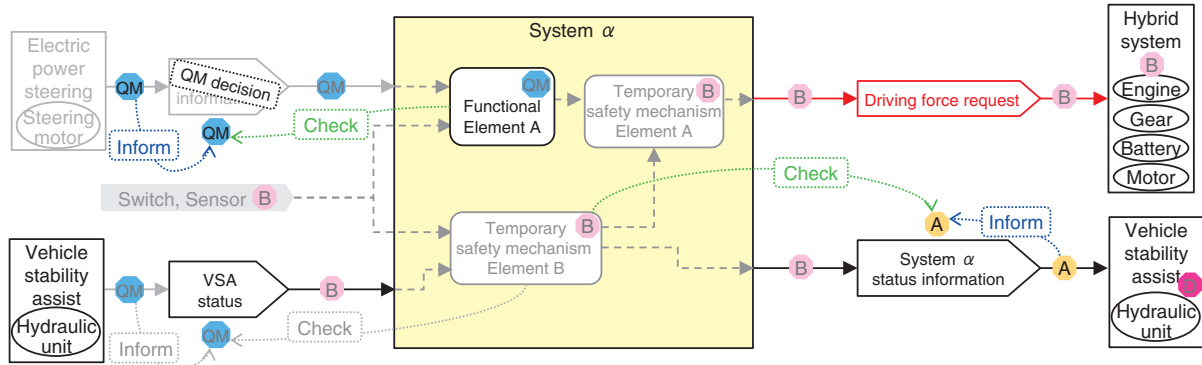


Fig. 10 Check for allocated ASILs

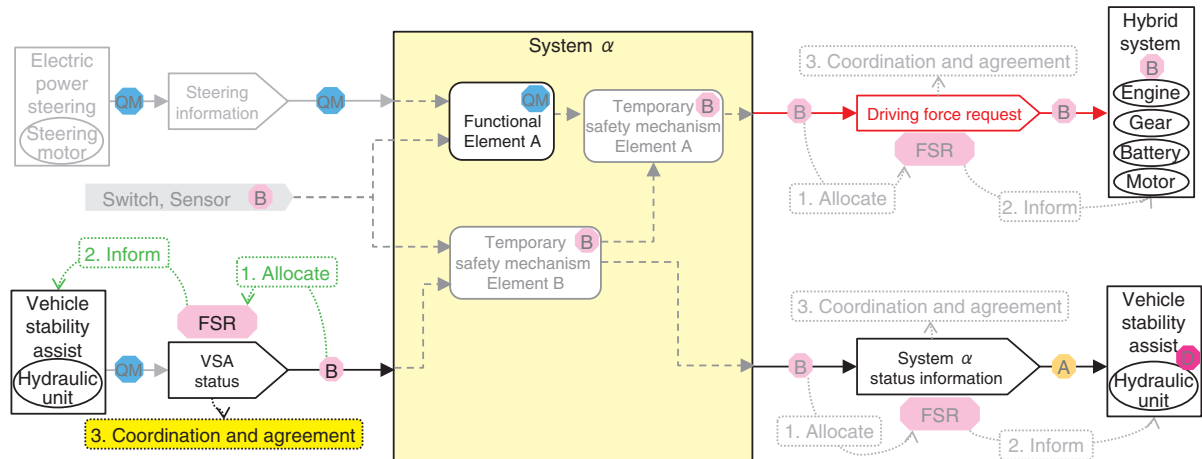


Fig. 11 Coordination and agreement of functional safety requirements assigned to other systems

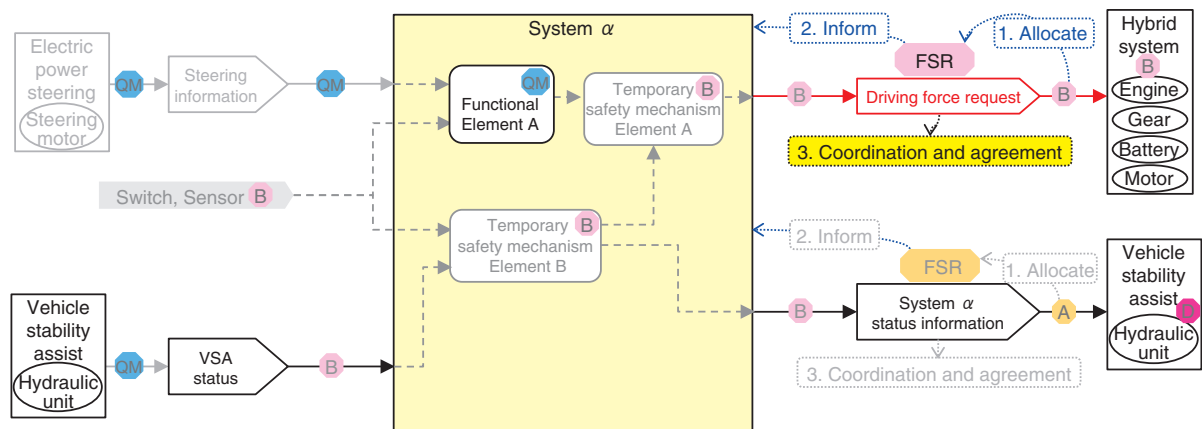


Fig. 12 Coordination and agreement of functional safety requirements assigned from other systems

3.3.5. Process Manager updates standard architectures

If new interfaces are added to the completed architectures, the Process Manager is to add the additional interfaces to the standard architectures.

The updated standard architectures will then be used as a development resource for subsequent vehicles.

3.3.6. System personnel perform functional safety activities in ISO 26262 from Part 4, “Product development at the system level,” to the end

Functional safety requirements assigned to a system in the activities described in Section 3.3.1.-3.3.5. are added to the requirements originally assigned within the system before functional safety activities from Part 4 to the end are implemented⁽²⁾.

Organizing item boundaries for the whole vehicle as given above and developing a functional safety process for the whole vehicle to be implemented through coordination of system personnel using standard architectures with inter-system interfaces clearly specified has made it easier to spot omissions and oversights in the assignment of functional safety requirements in vehicle-level functional safety activities. Further, highly polished standard architectures created after the coordination of functional safety requirements can be incorporated in development of subsequent vehicles, which has made the development with carry-over easier. Thus, an efficient functional safety process has been developed for the whole vehicle.

4. Conclusion

The development of this process has produced the following outcomes:

- (1) System interfaces have been classified into the mechanism’s force, operation requests, and information, and how the systems interrelate have been clearly specified. This helps system personnel to easily see what risks exist in their systems, as well as the source and scale, facilitating appropriate assignment of functional safety requirements.
- (2) Item boundary conditions are set with the systems and their mechanisms linked via interfaces, with the mechanism’s force and related hazards included in the item scope. This has reduced unneeded overlap in impact analysis scope, allowing system personnel to concentrate on analyzing the impacts within their own systems.
- (3) System personnel perform their impact analyses cooperatively with the Process Manager managing progress for each system, creating a functional safety process for a whole vehicle with coordinated functional safety requirements. This has made it easier to identify omissions and oversights in the assignment of functional safety requirements.

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APPENDIX

An explanation of the terms used in this paper as taken from functional safety standard ISO 26262 is given below:

- (1) Item
System or array of systems to implement a function at the vehicle level
- (2) Hazard
Potential source of physical injury or damage to human health caused by malfunctioning behavior of the item
- (3) ASIL
Automotive Safety Integrity Level. The levels are A, B, C, and D, with ASIL A representing the least stringent and ASIL D, the most stringent level.
- (4) QM
QM means the safety integrity level below ASIL A, and it denotes no requirement to comply with ISO 26262. The process for the items determined as QM is put under conventional quality management.

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Proposal of Development Process Using DSM and Simulation

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ABSTRACT

A development process was created to perform request-type development without reworking by determining the target system performance and the means for achieving that performance from the initial development stage. This process uses a Design Structure Matrix to determine the most reasonable development order, performs simulations to predict whether the target system performance can be achieved, and comprehensively calculates alternate solutions in case the target performance is not achieved. Data mining technology is then applied to these results to quickly extract the most reasonable solution from among multiple solution candidates. By way of illustration, using power train system development as an example, performance was investigated for the areas of fuel economy performance, acceleration from standstill, and low-engine-speed booming noise, and it was verified that the system designers could narrow down the optimal balance solution.

1. Introduction

In vehicle development, power trains continue to diversify as destinations expand due to globalization. Furthermore, rising safety needs are resulting in the increasing inclusion of Active Safety systems, so both devices and control are becoming more complex at ever-faster rates. However, development periods and costs are limited, so there are demands for technology that enables efficient development of these types of multifarious and complex automobile systems.

The V-model is an approach that stratifies the design and test processes of the system to be developed according to the respective grades, and matches the verification levels⁽¹⁾. In addition, model-based development (MBD) utilizes models from the initial design process and front-loads the verification to be performed in the test processes⁽²⁾. Use of a MBD V-model that combines this V-model with MBD (Fig. 1) makes it possible to reduce concerns in initial development processes and perform development without reworking in later processes. Figure 1 shows the MBD V-model for an overall vehicle system. Requirements include customer requests, market trends, laws and regulations, and other overall vehicle demands. The system designer determines the target performances in accordance with these requirements, and designs subsystems and parts to meet those target performances. This approach is spreading to each automobile maker, and the MBD V-model has also been used to promote efficient device development

for Honda powertrain systems^{(3), (4)}.

To apply the MBD V-model to an overall vehicle system, it is necessary to coordinate multiple differing target performances arising from various demands, and to design subsystems and parts. That is to say, even if the design staffs for each subsystem and part perform efficient development on their own, reworking occurs when the integrated subsystems and parts do not satisfy the target system performance.

Figure 2 shows an image of current vehicle system development. Here, the project leader (system designer) of the new model car summons the persons in charge (PIC) of each subsystem area to meetings to investigate the target performance of the overall vehicle. At these meetings, each PIC presents the target performance values of the subsystems investigated separately to that point. However,

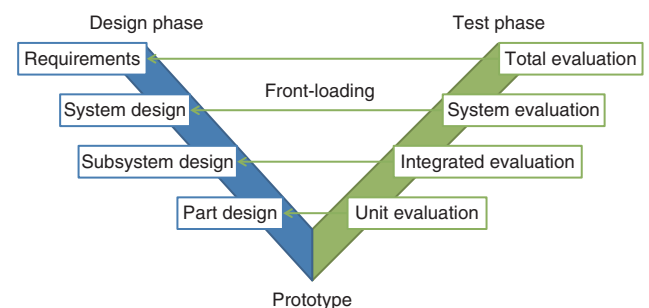


Fig. 1 MBD V-model

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when the target performance of the overall system is not achieved or a tradeoff relationship with another subsystem performance is discovered, the target performance values presented by each PIC remain pending, and the staffs of each subsystem area reinvestigate the target performances before the next meeting. This type of reworking is often repeated multiple times.

These issues in conventional system development stem from insufficient technical knowledge needed to achieve the target overall system performance and a failure to organize the relationships between the various performances. Target values with unclear prospects for achievement are merely nonbinding goals, so trial and error is repeated as long as the development period lasts. Conversely, if prospects for achievement can be reached from the start, the system designer can strategically choose the target performances from among the achievable Pareto optimal solutions, and request the needed performances from the subsystem staffs.

Therefore, to address the issues of conventional system development that involves much reworking, the following development process is proposed that enables determination

of the target system performance and the means for achieving that performance from the initial development stage.

- (1) Determine a reasonable development order.
- (2) Predict whether the target system performance can be achieved.
- (3) Search for other solutions in the event the target performance is not achieved with the initial investigation structure.
- (4) Quickly extract the most reasonable solution from among multiple solution candidates.

2. Overview of Development Process

Figure 3 shows an overview of the process proposed by this study. First, the method described in chapter 3 is used to sort the relationships between the target performances of the target system and the specifications and characteristics of each subsystem that comprise the target system in order to determine a reasonable development order. By modeling and visualizing each element and its dependencies, the order in which each element should be determined can be analyzed, and process enhancements can be proposed.

Next, the contents of the sorted relationships are clarified to predict whether the target system performance can be achieved. Specific methods and index values for determining characteristics, such as physical relationships, laws and regulations, trends, design procedures and timing, and designer knowledge and experience, are sorted based on the classification described in section 4.1. through hearings with designers, and the simulation environment is prepared.

However, the more complex the relationships between each system and subsystem performance, the harder it is to determine a development order completely free of reworking. Rework elements in Fig. 3 indicate the range of relationships between characteristics for which reworking occurs even when a reasonable development order is achieved through sorting. Reworking indicates the presence of a self-referencing relationship when obtaining a certain performance or characteristics value, where

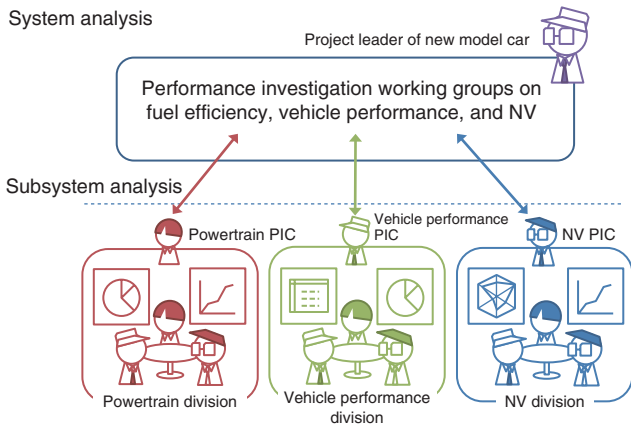


Fig. 2 Complexity of current product development phase

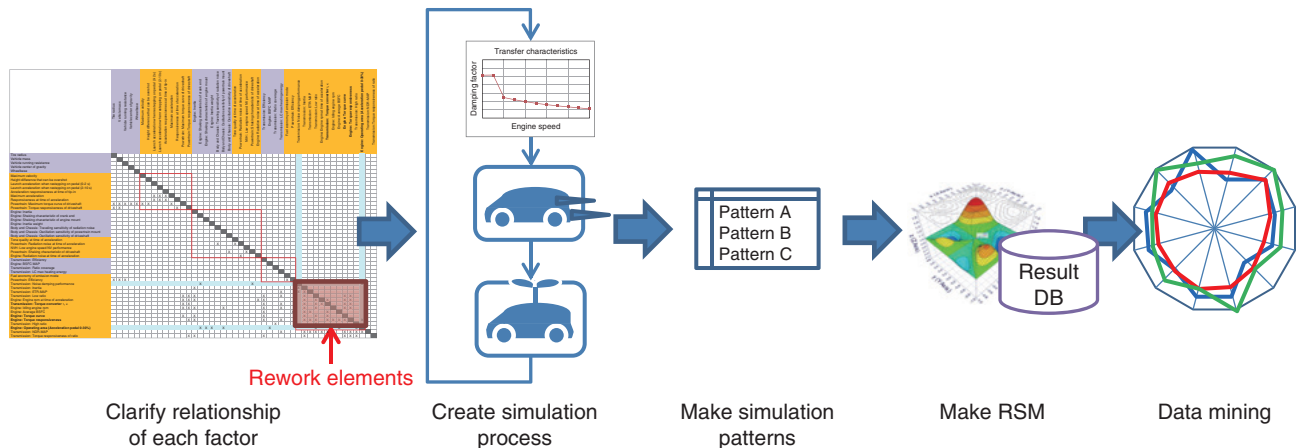


Fig. 3 Process overview

the preconditions for performing calculation include the characteristic value in question. Even when this kind of relationship exists, a solution can be obtained analytically as long as the calculation logic can be expressed by a simple equation such as shown in the upper part of the analytical approach in Fig. 4. However, when the logic includes a complex derived relationship such as the simulation shown in the lower part of the analytical approach in Fig. 4, it is not easy to analytically obtain a solution. In addition, since there are cases where no solution exists, or where multiple solutions exist, this paper performed solution search by statistical methods so that other solutions can be found when the target performance is not achieved. In the statistical approach in Fig. 4, when reworking occurs between characteristics, the process sets the initial conditions (initial values) and convergence conditions for the characteristics that are the rework factors, and searches for the initial values that satisfy those convergence conditions. However, when there are multiple rework factors, the search range becomes extremely large and there is no guarantee that the sought-after solution can be reached within the specified calculation time. In addition, there are also many cases in which the solution cannot be narrowed down automatically because of a tradeoff.

Therefore, the data mining technology developed by Ariyoshi et al.⁽⁵⁾ described in section 4.4. is used to quickly extract the most reasonable solution from among multiple solution candidates. The method developed by Ariyoshi et al. makes it possible to create a model by response surface methodology (RSM) that achieves accurate prediction from the calculation results, and that RSM can be used to efficiently search data in a short period of time.

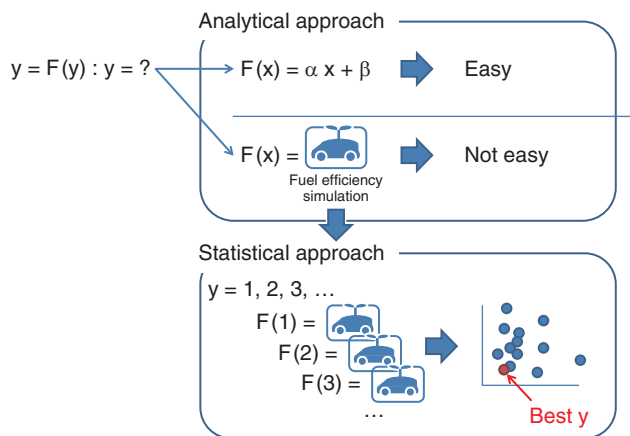


Fig. 4 Approaches toward of solution search

3. Sorting of Relationships Using DSM

A design structure matrix (DSM) is used to sort the relationships between the specifications and characteristics of each subsystem comprising the target system⁽⁶⁾. A

DSM is a simple method for performing both analysis and management of complex systems. Use of a DSM to model and visualize each element and the dependencies between elements of an arbitrary system enables analysis of the system configuration, which also facilitates management of each element. In addition, system enhancement proposals can be made by rearranging the order in which each element is determined based on these analysis results⁽⁷⁾.

Typical DSM modeling procedures are shown below. First, all elements comprising the target system are extracted and arranged equally in order of determination in the vertical and horizontal directions of a square matrix. Next, the elements in each column are input to the row elements that they influence with values indicating the respective influence dependency. This is called numerical DSM [Fig. 5(a)]. Rearranging the order of each DSM element enables sorting into a modularized element that reduces reworking, so this can be expected to effectively enhance the existing determination process for the target system.

A DSM is a matrix with the system and subsystem specifications, characteristics and target performances needed for development as elements, and is used to arrange the order in which each element is designed in order to optimize the development order. In the following chapters, this process is used to predict the system performance from the relationship between each element. However, these relationships cannot be expressed by simple values in the manner of the influence dependency needed by numerical DSM. Therefore, this paper used binary DSM [Fig. 5(b)], which models only simple dependencies, to determine the optimal development procedure. The detailed relationships were then determined separately as described in the following chapter. The DSM modeling procedure was performed as follows.

- (1) Arrange the elements needed for vehicle development in the same order in the vertical and horizontal directions of a square matrix.
- (2) When determining the vertical axis elements (outputs), mark the required horizontal axis elements (inputs).
- (3) Rearrange the element order to reduce the occurrence of marks in the upper right half of the matrix, while paying attention to performance modularization for the entire vehicle. Strive to minimize the range of the element group for which reworking cannot be avoided.
- (4) Use the order of the partitioned vertical axis elements as

Element	A	B	C	D
A				1
B	3		3	
C				
D	2		1	

(a) Numerical DSM

Element	A	B	C	D
A				x
B	x		x	
C				
D	x		x	

(b) Binary DSM

Fig. 5 DSM

the element determination order.

A typical system consists of multiple subsystems, and has a tree structure down to each individual part. If the level of each element extracted from the system differs greatly, subsequent description and partitioning of relationships becomes more complex, and arranging the elements becomes difficult. Therefore, it is necessary to extract elements on the same or higher levels as much as possible, and to show the relationships between these elements.

In addition, when setting the degree of influence with a DSM, it is not easy to describe relationships in a manner that takes all influences into account. While there may be relationships between each element, when the degree of influence is small, adjustments should be made to remove these small influences in order to reduce the DSM complexity. Furthermore, regarding elements for which the degree of influence is unclear in advance, the process should be adjusted (tailoring) in accordance with practical operations, by taking these relationships into account at the start of investigation, and then removing them from subsequent investigation.

There is no absolutely correct solution with regards to the partitioning of each element. Elements can be mechanically rearranged using a partitioning algorithm such as the reachability matrix method⁽⁸⁾ or identify loops by path searching⁽⁹⁾, but the approach toward modularization and the allowable rework range differ according to the target system, so DSM partitioning should reflect the intentions of the system designer.

As shown above, even if relationships can be sorted using a DSM, this does not mean mechanically extracting elements, adding relationships, and deciding the optimal determination order. The system designer and the staffs of each element should work together closely to construct and tailor the DSM.

However, even if the design order of each element to be developed can be sorted using a DSM, there will be element groups for which reworking cannot be avoided.

4. Derivation of Solutions by Data Mining

Technology was developed to perform predictive calculations in advance using various alternate solutions for element groups for which reworking cannot be avoided, and to quickly derive the most reasonable solution from among multiple solution candidates. Specifically, rework element group values are sampled over a wide range within the range of the design restrictions. Element group prediction simulations are then performed using these sampled values as the initial values, and interactive data mining is used to search for solutions from among the obtained multiple results.

4.1. Definition of Relationships between Elements

The relationships between each element were clarified and simulations were performed based on those

relationships to predict and visualize system performance.

The presence of influence between each element was demonstrated using a binary DSM that modeled only simple dependencies, but this DSM does not express the details of these relationships. Therefore, in order to clarify the relationship between each element, grouping was performed as shown by the specific examples below, and the inputs and outputs, calculation logic, and evaluation criteria needed to calculate numerical solutions were determined. Some of the groups (classifications) used to sort the relationships between elements are described below.

Constraint relationships refer to items determined as target values from market demands such as laws and regulations, comparison with competitors (benchmarking), and trends, and items with little or no freedom of design due to cost, production, and other limitations. Changes to these elements involve review of preconditions, so they have a large influence on other elements.

Subordinate relationships refer to items that have clear rules and principles and are determined uniquely according to the preconditions. These include items with clear decision algorithms, such as those determined according to simulation results.

Design policy and conditional relationships refer to items determined inversely from simulation results. These are items for which multiple solutions may appear, and for which the solution selection guideline is unclear.

Free design relationships refer to items determined more or less based on staff knowledge and experience, and from trends. These are items for which generalization is a challenge, and for which new relationships other than extracted items may occur.

As shown by the examples above, the basis for calculating each element is clarified by grouping the relationships between elements, making it easier to obtain the understanding of related parties when changes to calculation conditions occur.

4.2. Calculation of Each Element Value

Each element value is calculated in the order determined by the DSM. Regarding element groups for which reworking cannot be avoided, calculations are repeated multiple times based on design of experiments (DOE) among the elements comprising the largest loop. The values of the elements or element groups that are the reasons for reworking are sampled comprehensively within the design space as initial values, and these sample values are used to calculate subsequent element values using the previously determined calculation logic in the order determined by the DSM (Fig. 6).

The specific calculations handled here are performed at arbitrary levels ranging from simple spreadsheets to combinations of multiple analysis tools in conformance with the contents investigated beforehand in each performance area.

Each element is calculated in accordance with the DSM

determination order using the element values calculated in the previous stage and the initial values.

Each element value calculated using different initial values is linked and compiled into a database (DB), and visualization is performed using analysis tools in accordance with the purpose.

4.3. Method for Finding the Optimal Solution

The optimal solutions for the system designer and PIC are searched from each element compiled in the DB. Sorting is performed to minimize the range of the rework element group subject to optimization. However, when investigating overall system performance such as in this case, there are many inputs so multi-aspect issues cannot be avoided.

Various algorithms, ranging from analytical to statistical approaches, have been proposed to search for multivariate, multi-aspect optimal solutions⁽¹⁰⁾. However, when performing multi-aspect optimization with four or more dimensions, it is a challenge to obtain the optimal solution analytically within a practical number of calculation times⁽¹¹⁾.

Moreover, in actual design, reconciliation is performed based on implicit knowledge that is not easily quantified. For this reason, it is not easy to obtain the understanding of PIC with conventional methods that automatically search and narrow down the optimal solution.

Therefore, instead of a method that obtains the optimal solution analytically or statistically, this paper used data

mining technology that enables the system designer and PIC to derive the solutions to be obtained from among each element compiled in the DB in an interactive manner via a graphical user interface (GUI).

4.4. Solution Search by Interactive Data Mining

Figure 7 shows the interactive data mining flow proposed by Ariyoshi et al. The data search procedure is as follows.

- (1) An RSM is applied to the previously created calculation results DB to approximate the data within the design space in order to increase the solution density.
- (2) Grouping is performed to bundle the data in step (1) above into each related area.
- (3) The typical values of the grouping results for each area are shown in a radar chart, so the user selects the preferred group.
- (4) The data of the selected group is extracted, and grouping is performed again within this extracted data. The typical values of these results are shown in a radar chart.
- (5) The user repeats steps (3) and (4) to narrow down the solutions until the sought-after solution is obtained.

The radar charts for each area are linked in the DB, so when one radar chart is selected, the results are also reflected to the radar charts for other areas. In this manner the optimal solution is derived by narrowing down the data in consideration of the influence of each area on other areas.

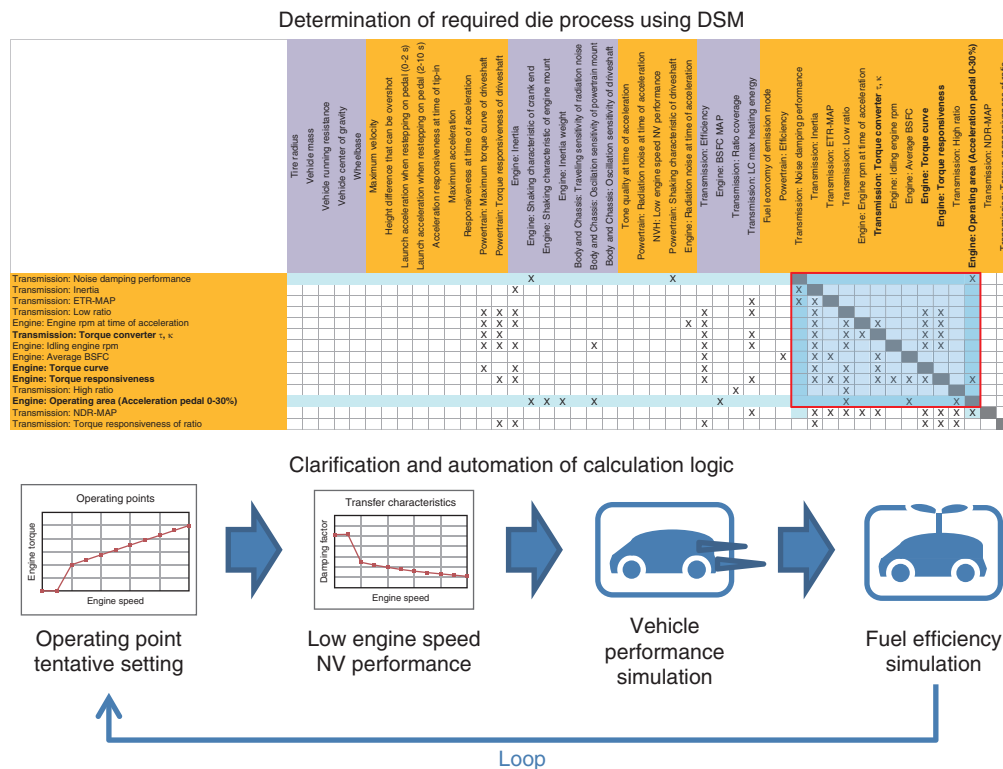


Fig. 6 Coordination process of DSM and simulation

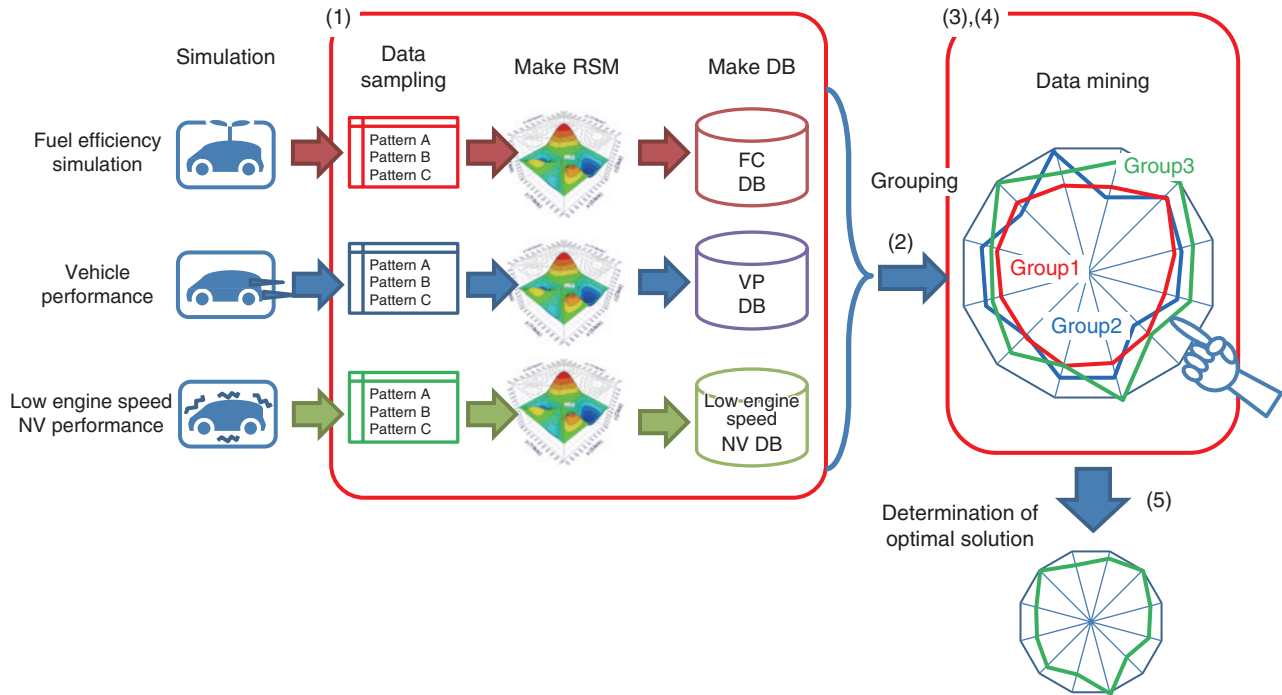


Fig. 7 Interactive solution search

5. Application to Actual Development

The process described in chapters 3 and 4 above was applied to a model currently under development and the effects were confirmed. The performance of a conventional engine vehicle was investigated using the proposed process for the areas of fuel economy, acceleration from standstill, and low-engine-speed booming noise. Figure 8 shows the DSM created by extracting the elements needed for the investigation.

The elements extracted in this investigation were narrowed down to typical items such as vehicle body specifications, running resistance, engine and transmission specifications and characteristics values, and control specifications in order to enable design and management in the initial stages of actual development. Elements with predetermined values set due to constraints other than extracted elements, that is to say elements with performance designs that cannot be changed, are indicated by gray. Conversely, elements that can be derived from the relationships with other elements in the DSM are indicated by orange. Gray elements are constraints, so these are basically shifted to the front part of the DSM, but there may also be cases where gray items are intentionally set in the middle part from the viewpoint of modularization. The created DSM was then partitioned from the viewpoints of reducing reworking and promoting modularization as described in Chapter 3. Figure 9 shows these results together with the rework and modularization areas.

As a result of partitioning, the three upper areas enclosed by red frames in the figure could be grouped as areas that enable modularization of acceleration from standstill, low-engine-speed booming noise, and fuel economy performance. In addition, element groups for which reworking cannot be avoided were gathered in the bottom right area enclosed by the red frame for the next optimal solution search.

Concrete performance design was performed using the sorted DSM as a reference. This investigation treated the engine characteristics as fixed values in subsequent processes in order to limit the target area. The rework area elements included many transmission area specifications and control specifications that play a central role in power transmission. In addition, the rework elements were determined to be the three elements of the recommended engine operating points (engine speed, engine torque), the torque converter characteristics (τ , κ), and the final gear ratio.

Next, relationships between elements within the rework area were clarified, and the calculation logic and evaluation criteria were determined.

Figure 10 shows an example of the logic used to calculate the transmission damping performance. An input/output equation approximation model was created beforehand using an analysis model that calculates the crank-end torque fluctuation based on the cylinder pressure measured on an engine bench. This approximation model was set as the crank-end excitation characteristics.

The recommended engine operating points (engine speed, engine torque) were input to the crank-end excitation characteristics, and the crank-end torque fluctuation at each engine speed was calculated.

Here, when the target wheel-axle excitation characteristics are set as the constraint, the damping performance demanded of the transmission can be obtained from the crank-end torque fluctuation and the wheel-axle excitation characteristics.

The evaluation criterion was that the calculated damping performance could not exceed the damping ratio of torque converters that can be used by the model to which the process was applied. The created calculation logic was mounted using a program and stored as a calculation module.

A calculation module was created in the manner described above for each element in the rework area, and calculation was performed by linking the respective inputs and outputs. The initial values of the three rework elements

were comprehensively sampled within the design space. The results of performing calculations within the rework area one time in the DSM order were taken as one sample, and approximately 400 samples were calculated. Figure 11 shows some of the calculation results.

Figure 11 shows an example of a tradeoff relationship where elements related to fuel economy performance, acceleration from standstill, and low-engine-speed booming noise mutually influence each other. Figure 11(a) shows the fuel economy performance in JC08 mode on the vertical axis and the 0-100 km/h acceleration time at wide-open throttle on the horizontal axis. The fuel economy is better and the acceleration time is shorter towards the upper left of this graph, so this shows good performance.

However, there are multiple samples that show good results, so two of these points were extracted and linked, and the damper performance was plotted as shown in Fig. 11(b). These figures show the damping factor on the

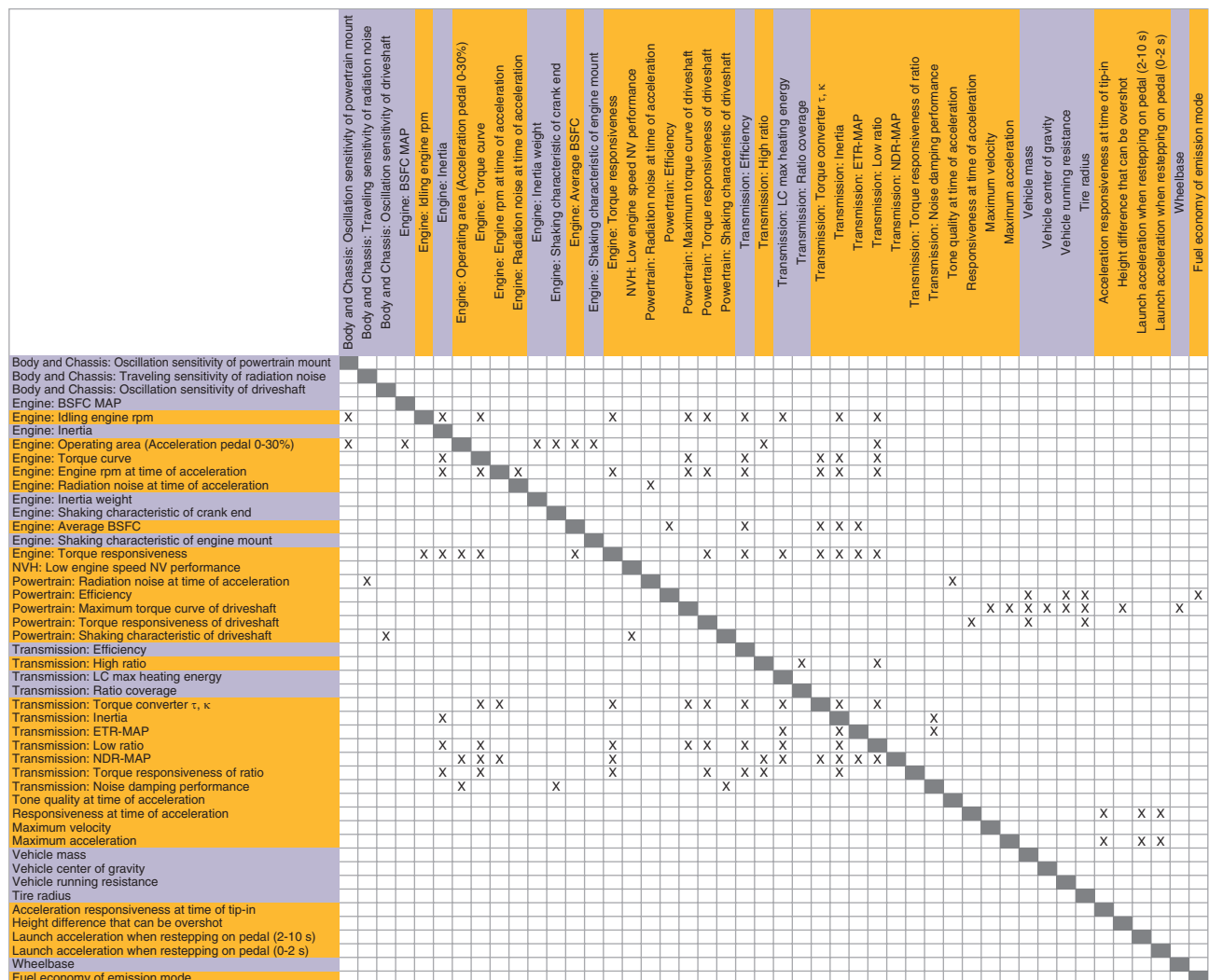


Fig. 8 DSM data before sorting

vertical axis and the engine speed on the horizontal axis. This shows that the damper performance differs even at the same fuel economy and acceleration from standstill, indicating that the usable torque converters also differ.

In addition, Fig. 11(c) shows the acceleration characteristics at that point. This figure shows the torque converter characteristics on the vertical axis and the acceleration response (acceleration after 0.5 s when accelerating at wide-open throttle from standstill) on the horizontal axis, and shows that the performance becomes better toward the right side. These results show that the two extracted samples exhibit the best fuel economy and acceleration time, but that other samples are better when viewed in terms of acceleration response.

Confirmation of simulation results of different levels from various viewpoints in this manner enables the system designer and PIC to discuss performance. However, the method of comparing the results separately for each sample point takes time to find the optimal solution.

Therefore, data preference-type data mining technology was applied to complete the process. An accurate RSM was created from approximately 400 calculated samples,

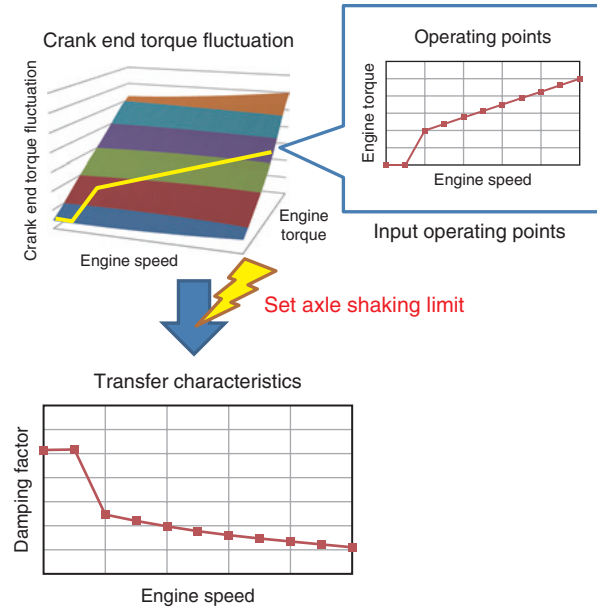


Fig. 10 Example of related definition

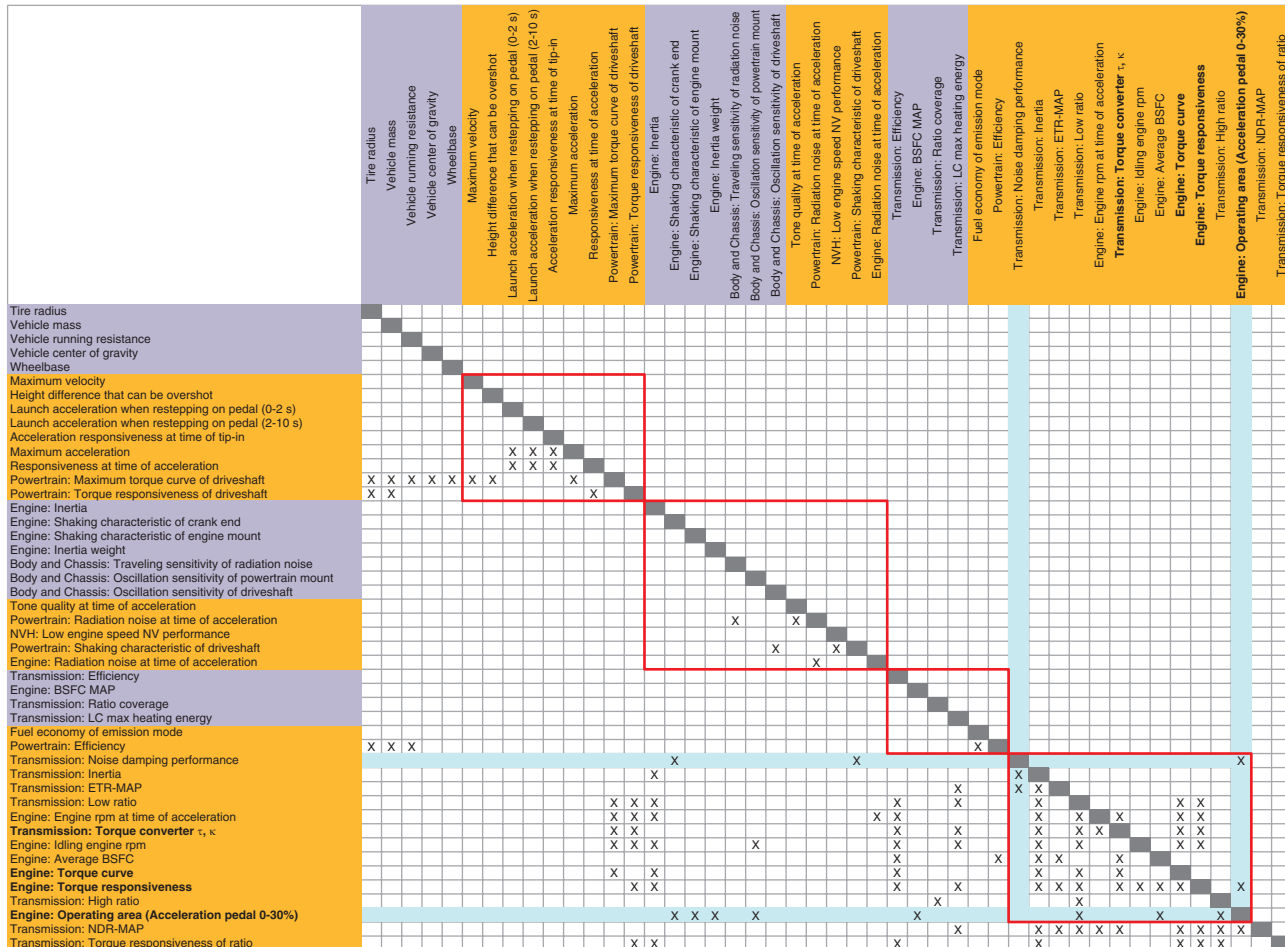


Fig. 9 DSM data after sorting

and this RSM was used to efficiently search the data. Figure 12 shows the process of narrowing down by data search, and demonstrates that as the process proceeds from

left to right, the optimal solution is narrowed down from among the solution space. The radar charts in the top part of the figure show typical performances and characteristics

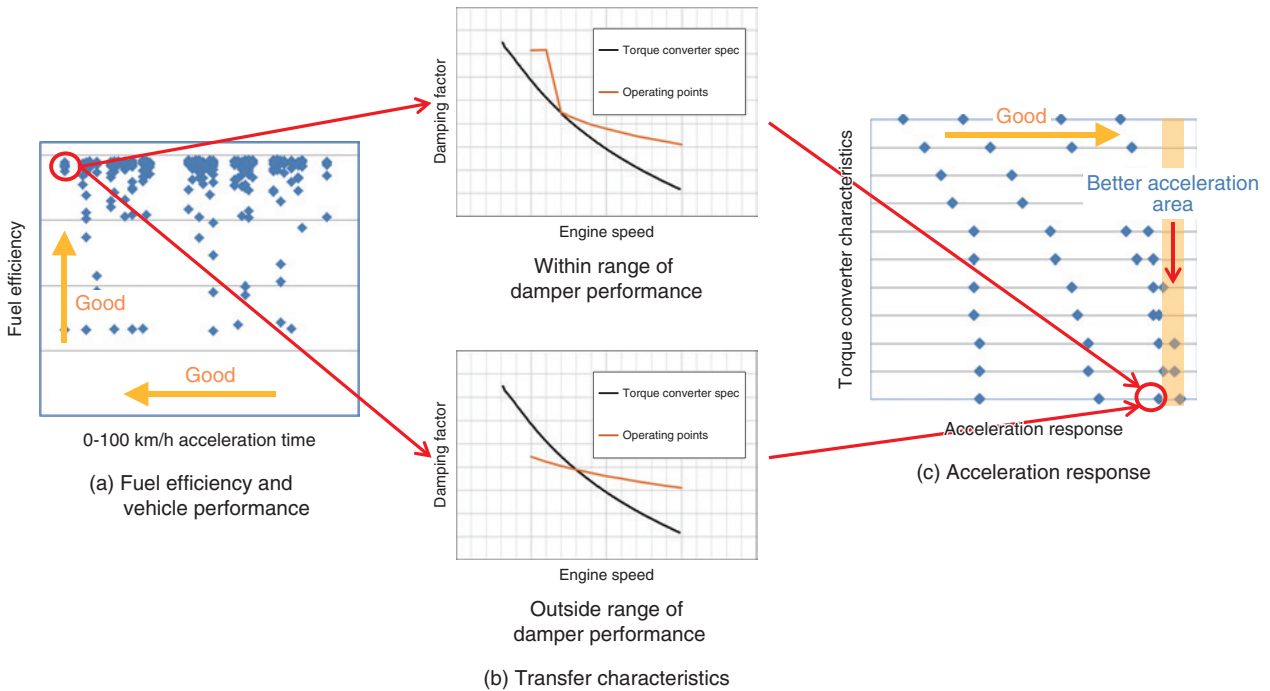


Fig. 11 Example of tradeoff relationship

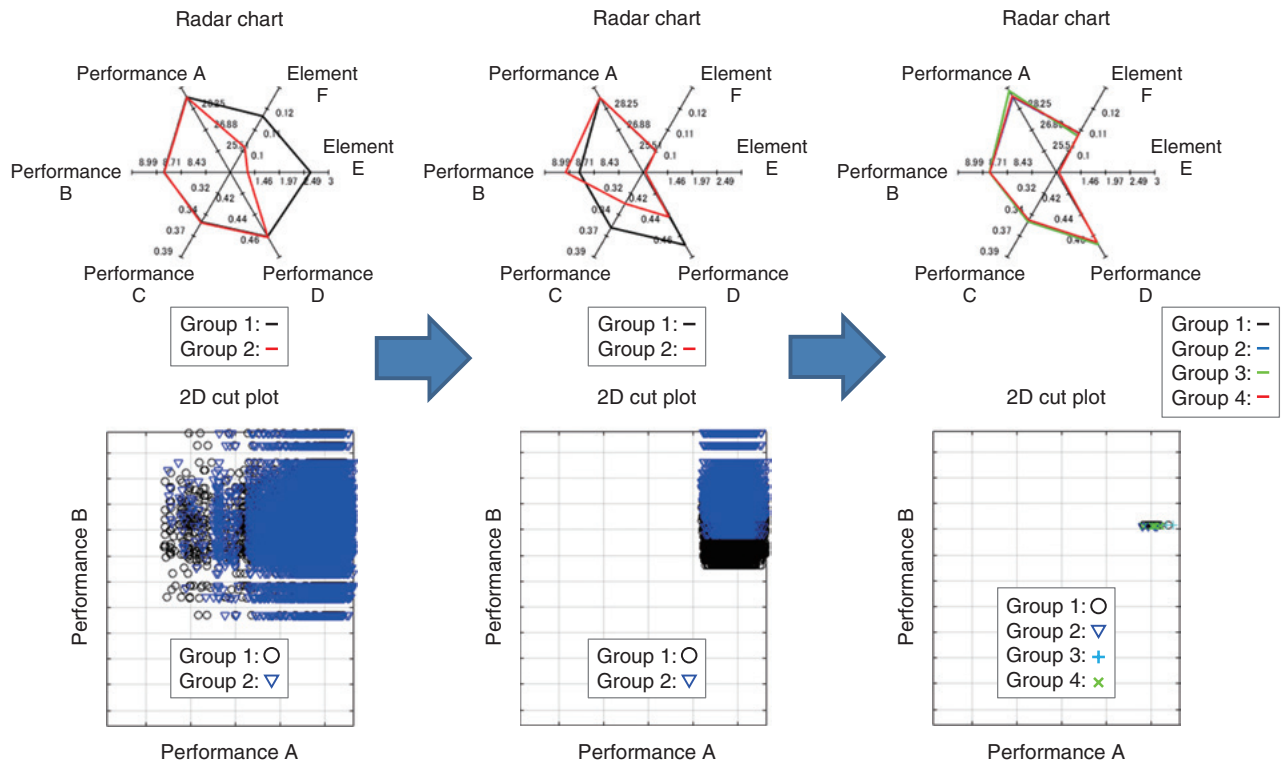


Fig. 12 Data mining process

to be confirmed together, such as fuel economy, acceleration from standstill, the torque converter type to be used, etc.

The 2D cut charts at the bottom cut out and plot only Performances A and B. The large number of sample points are automatically clustered, and the typical values are shown in the radar charts. The left figure shows the initial clustering results for the entire solution space, and shows two groups: Group 1 and Group 2. Here, Group 2 that has small Element E and Element F values was selected, and then clustering was performed again using only the selected Group 2 solutions to show the two groups in the center figure.

As a result of repeating this process, only solutions with roughly equal performance and characteristics remained as shown in the right figure. In this manner, the system designer and PIC were able to narrow down the solution with the optimal balance from among the solution space by continuing to select the radar chart that exhibits the desired tendencies while observing the balance between each performance value.

However, there are still many uncertain factors in the initial development stage and characteristics also change easily, so target performances coordinated beforehand using simulations in the initial stage may not necessarily be achieved by the actual component. Therefore, it is necessary to set robust solutions even in the stage when initially adjusting the target values.

As described above, an development process without reworking was proposed. This process consisted of sorting and calculating the influence of various performances in the fuel economy, acceleration from standstill, and low-engine-speed booming noise areas, and using data mining technology to determine the target system performance and the means for achieving that performance in the initial stage of powertrain system development.

6. Conclusion

The following development process was proposed that combines DSM, simulation calculation, and data mining technology in order to perform request-type development without reworking by determining the target system performance and the means for achieving that performance from the initial stage of development.

- (1) A DSM was used to clarify the relationships between system and subsystem performances and characteristics, and a reasonable development order was determined.
- (2) Logic for calculating the relationships between performances and characteristics was established, and enabled prediction of whether the target system performance can be achieved.
- (3) Simulations were performed in a wide design space for rework elements based on (1) and (2) above, and made it possible to search for other solutions in the event

the target performance is not achieved with the initial investigation structure.

- (4) Data mining technology was introduced to the vast calculation results, and enabled rapid extraction of the most reasonable solution from among multiple solution candidates.

Using powertrain system development as an example, performance was investigated for the areas of fuel economy performance, acceleration from standstill, and low-engine-speed booming noise, and it was verified that the system designer and PIC were able to narrow down the optimal balance solution.

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Design Philosophy of Honda WANDER STAND CONCEPT

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ABSTRACT

Anticipating changes in mobility needs stemming from changes in living environments and the further evolution of technology, Honda has examined the value of new forms of personal mobility, and has proposed the Honda WANDER STAND CONCEPT as a mobility aid embodying the concept of making the process of getting from place to place using the device as enjoyable as taking a stroll. The Honda WANDER STAND CONCEPT is a small, automated-driving electric vehicle that can be quickly and easily ridden by any user, and that makes it possible to comfortably move from place to place without any of the concerns associated with driving.

A working model of the Honda WANDER STAND CONCEPT was exhibited at the Tokyo Motor Show held in November 2015, and visitors were asked to complete a questionnaire regarding the unit. The results of this questionnaire demonstrated that the public was receptive to the styling and the concept proposed for the unit.

1. Introduction

Today, rapid changes are occurring in living environments in the advanced nations, and the nature of our daily activities and the form of the towns and cities in which we live are also beginning to change. These changes are diverse, and include the rapid decline of birthrates and the aging of populations, growing environmental awareness and the entrenchment of measures to reduce CO₂ emissions, the concentration of populations in cities⁽¹⁾, the consequent design of urban spaces to reduce dependence on cars⁽²⁾, and the evolution of IT services that make it possible to perform tasks without leaving the home. In Japan, although attempts are being made to promote mobility among citizens, such as an effort to increase pedestrian space in Kyoto⁽³⁾, reports indicate that both old and young throughout the country are less mobile⁽⁴⁾.

Analysis of “mobility” has indicated two main types of human mobility: Mobility as a derivative demand, involving travel to necessary destinations such as work or school; and mobility as a self-motivated demand, engaged in for the purpose of movement itself, for example going on drives and taking trips⁽⁵⁾. Modalities of public transport in cities where the population is concentrated, such as trains, buses, and taxis, were developed in order to satisfy the former demand. Compared to the state of provision of public transportation, the provision of infrastructure (including roads and parking facilities) necessary for transportation by

car, a personal mobility modality, tends to be inadequate. In addition, the abovementioned evolution of IT services is reducing the amount of mobility as a derivative demand, and is considered to be producing a movement away from the use of automobiles⁽⁶⁾.

The focus was therefore placed on mobility as a self-motivated demand.

Honda has been conducting actual proof experiments⁽⁷⁾ of next-generation personal mobility modalities as an initiative to increase value in the area of mobility as a self-motivated demand. The company has positioned increased quality of life (health, vitality, a sense of satisfaction, etc.) as a value for mobility. The research discussed in this paper focused on strolling as one of the values offered by mobility and embodied and verified the concept in a vehicle. It proposes a concept of mobility in which going where one wishes while enjoying urban spaces and making new discoveries within everyday scenes is positioned as the value of mobility.

2. WANDER Concept

The representative users of this form of mobility were projected to be people who want to make new discoveries as they wander the city streets, whether they are residents of the city or come from outside.

The projected areas for use of the vehicle were city areas mainly used by pedestrians, and it was therefore designed

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as a two-person mobility modality more compact than a normal car. The necessary elements for enjoyment of urban spaces were considered to be:

- (1) Operability using automated driving, to allow users to enjoy the scenery without the nuisance of operating the vehicle;
- (2) Sufficient compactness to allow the vehicle to enter narrow alleys; and
- (3) Maneuverability making it possible to stand still and turn around in the same way as pedestrians.

The speed range for use of the vehicle was projected as ranging from ultra-low speeds to around 30 km/h.

Because the word “wander” encompasses the three elements above and incorporates the sense of freely moving from place to place as one does when strolling, the concept was termed “WANDER.”

3. WANDER STAND Concept

3.1. Packaging and Applied Technologies

Automated driving research at Honda is currently focusing on efforts to realize a mobile, accident-free society⁽⁸⁾. In Japan, there is also discussion of certain “National Strategic Special Zones” being dedicated to fully automated driving⁽⁹⁾. The development of the new vehicles assumed a scenario in which fully automated driving would be allowed in such National Strategic Special Zones within Tokyo by the time of the 2020 Tokyo Olympics. This provided a vision of easier mobility based on automated driving.

Turning to Honda’s thinking regarding the size and maneuverability of the vehicle, while ultra-compact vehicles for urban use frequently feature a tandem-seat layout to reduce overall width, the new concept prioritized conversation and the sharing of experience, and projected a two-person vehicle in which the occupants would sit side-by-side, with a compact layout of a total length of 2000 mm and a total width of 1250 mm. In terms of the drive system employed, the vehicle would be an electric vehicle fitted on all four wheels with Honda Omni Traction Drive⁽¹⁰⁾ (termed “HOT Drive” below) to allow the freedom of motion, like walking, appropriate to the WANDER concept. In order to make fully automated driving compatible with the parallel motion and turning operations available from HOT Drive, rather than a steering wheel, accelerator, and the other elements of the existing man-machine interface, an interface of a type available only in a vehicle capable of fully automated driving was devised. This did away with the need for the operation of pedals, contributing to an increased degree of freedom in terms of potential sitting postures.

It was considered that increasing the ease of entering and exiting the vehicle would be an important factor in making it easy to use. The vehicle was provided with an ultra-low floor and a front-opening door (taking the flow of entering and exiting the vehicle into consideration) to help facilitate quick entry. A high seat position was employed in order to minimize changes in posture when entering and exiting the

vehicle. A tall body was employed to make it comfortable for users of any height to ride in the vehicle, and the cabin was provided with a full head-up display to enable for users to monitor navigation data while also enjoying the scenery in all directions (Fig. 1).

3.2. Styling Design

Because the purpose of the vehicle was to offer free mobility in the streets via automated driving, attempting to ensure that both occupants and surrounding pedestrians felt secure was a priority. The vehicle was designed based on the styling concept “simple, clean, and modern” in order to help ensure that it would blend into both historical streetscapes and modern urban areas and occupants would not look unnatural when riding in it. The design sought to ensure that pedestrians and others around the vehicle would feel safe despite the fact that it was driven automatically. The use of a design in which the rear of the vehicle is more broadly covered than the front and the sides boosts visibility from other vehicles and offers a sense of security (Fig. 2). The synchronization of the lights at the front and rear with the operation of the vehicle makes

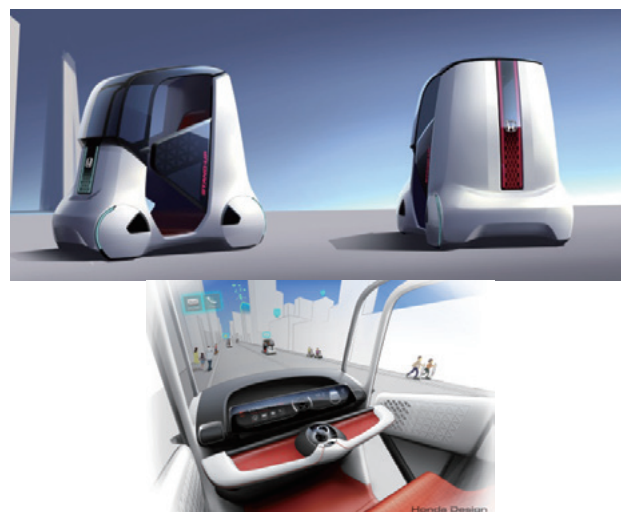


Fig. 1 Final rendering



Fig. 2 Final exterior

it possible for people to see when the WANDER STAND CONCEPT has switched to automatic operation. To help ensure that pedestrians are not intimidated by the wheels, the design has concealed the tires, a strongly iconic feature of conventional vehicles, and established a connection between sensors and lights. This gives an impression of advancement, and projects a sense of security to pedestrians (Fig. 3). This design was achievable because the use of HOT Drive made it unnecessary to impart a steering angle to the wheels.

The extended vertical headlight emphasizes the tall packaging which is a distinctive feature of the vehicle, and provides an iconic representation of the fact that it represents a modern concept of mobility, different from conventional vehicles (Fig. 4).

3.3. Interior Design

The design of the vehicle was not predicated on driving, but rather assumed scenarios in which passengers enjoyed the scenery while stopping here and there as they would when strolling, occasionally enjoying conversations with other people on the street. The design therefore set the occupants' eye-point at a height that would offer occupants of all statures the same line of sight as pedestrians, and an angled riding posture, making it possible for users to easily enter and exit the vehicle. To help enable this riding posture, the WANDER STAND CONCEPT uses seats fitted with slides

that enable them to be moved up and down and a layout in which the table is like a bar counter (Fig. 5). Anticipating the use of the vehicle on shopping excursions, the layout features space at the occupants' feet for storage of bags and packages.

The interface design helps to ensure that users are able to enjoy the scenery without being distracted by driving operations or focusing on maps. Two interface systems have been employed. The main source of information for occupants is a head-up display that provides a variety of information linked to changes in the external scenery. The vehicle has also been provided with a large center display that offers operating data and enables communication between the vehicle and the occupants. A universal joystick control has been positioned in the center of the table as an interface to assist the automated driving function, allowing anyone, whether right- or left-handed, to freely and intuitively guide the vehicle's motion. This welcoming space, easy to use by all, has been designed to provide an interior that will continue to inspire users to take the vehicle out again and again (Figs. 6 and 7).



Fig. 3 Light on corner



Fig. 5 Modern sliding seat



Fig. 4 Iconic headlight



Fig. 6 Universal instrument panel

3.4. Color and Material Design and Interior/Exterior Coordination

The color concept for the vehicle was set as “Japanese Modern,” with the aim that overseas visitors using the vehicle to get around would experience a sense of Japanese tradition and Japanese hospitality. Given that one of the characteristics of the vehicle was that the interior can be viewed at a glance from the exterior, three colors – white, vermillion, and black – have been used to achieve a seamless coordination of interior and exterior. On the exterior of the vehicle, the use of a white color in which golden highlights shine not only emphasizes the distinctive styling of the vertical keynotes, but offers a texture in which the color also appears to change depending on the viewing angle, making the unique movement that HOT Drive gives the vehicle visible even from a distance (Fig. 8). Vermillion has been used as the floor and seat color to offer users a welcoming impression, as if they were on a red carpet, until they take their seats. The distinctive seat fabric is jointly manufactured with Hosoo Co., Ltd., a company that possesses extensive textile manufacturing technologies, located in Kyoto’s Nishijin district. Skillful weaving of multiple threads with differing textures produces a design in which the pattern changes minutely with the viewing angle (Fig. 9). This is also an element that promotes enjoyment of the unique vehicle motion offered by HOT Drive.

As another feature to give the vehicle a uniquely



Fig. 7 Center display and joystick controller



Fig. 8 3D white color

Japanese feel, the exterior front center and rear center sections, the interior door panels, and the roof feature a triangular pattern based on the design of a traditional Japanese transom (Fig. 10).

3.5. Public Receptiveness to Vehicle Concept and Styling

The Honda WANDER STAND CONCEPT and WANDER WALKER CONCEPT were exhibited at Tokyo Motor Show 2015 (Fig. 11). Visitors were invited to take a questionnaire survey in order to help enable study of public



Fig. 9 Textiles for seat fabric

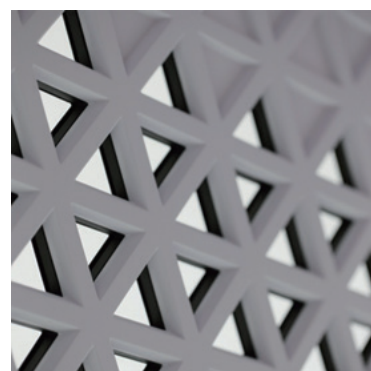


Fig. 10 Triangle pattern



Fig. 11 Tokyo Motor Show 2015

receptiveness to the concepts and styling that had been applied. Figure 12 shows the survey results. A total of 215 responses were received, with 94 comments concerning the Honda WANDER STAND CONCEPT.

When asked which vehicle they would prefer to ride a bicycle, a motorized bicycle, the WANDER STAND, or the WANDER WALKER (a one-person personal mobility aid), more than 60% of respondents selected the WANDER STAND. Asked for what kind of situation they would use the WANDER STAND, approximately half responded “For travel” or “For leisure.” Five comments related to receptiveness to the concept, for example asking when the vehicle would be mass produced and indicating a desire to ride in one. Eleven comments concerned automated driving, for example indicating a desire to try it, and commenting that using the vehicle looks effortless. Nine comments concerned the compactness of the two-person vehicle, indicating, for example, that a couple, or a couple with a child, could enjoyably ride in it. Four comments mentioned technological issues, for example anxiety regarding automated driving technology. Expressions of anxiety regarding automated driving had been anticipated, but overall, most comments were hopeful regarding future prospects, and the survey verified that the public would be receptive to the concept and styling of the Honda WANDER STAND CONCEPT.

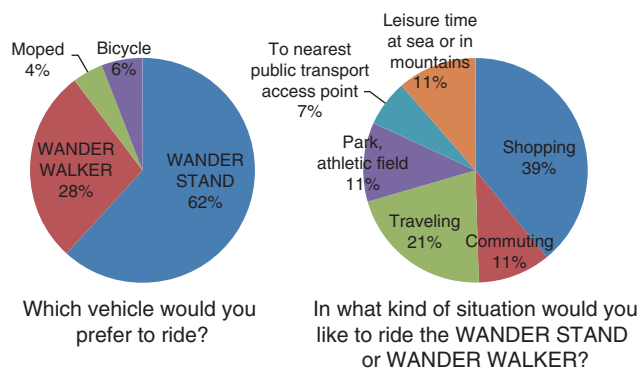


Fig. 12 Questionnaire result (215 respondents)

4. Conclusion

Conventional personal mobility modalities have been designed to satisfy the conditions for mobility as a derivative demand, and their designs have sought to realize efficiency. The WANDER CONCEPT has been proposed in order to stimulate and increase the desire for mobility as a self-motivated demand, and the concept has been embodied in projected vehicles. The public's receptiveness to this new concept and its styling was demonstrated when these vehicles were exhibited at Tokyo Motor Show 2015.

Honda will continue to advance its research and development towards the realization of a society in which

all people are able to move freely, wherever they wish, with a feeling of complete security.

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Design Philosophy of Honda WANDER WALKER CONCEPT

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ABSTRACT

Advanced nations are undergoing a major shift in direction from “car-centered” to “human-centered” urban design. Essential functions are being concentrated and city centers are becoming more compact, while initiatives to provide and expand pedestrian areas are intensifying. As we welcome an era in which the number of elderly citizens and others experiencing restricted mobility will increase and populations will undergo further urban concentration, expectations are increasing for new personal mobility aids that will supplement cars and public transport.

The Honda WANDER WALKER CONCEPT was developed to offer an “urban micro-mover” that would encourage all users to leave their homes and become mobile. Seeking a package design that would contribute to reducing infrastructural barriers, the development team achieved a size that would make it easy for users to access urban infrastructure. The WANDER WALKER CONCEPT is 540 mm wide with a minimum turning radius of 1,000 mm, making it capable of turning in its place. Twin-lever steering has been employed to ensure intuitive operation that is easily understood by all users.

In order to reduce psychological barriers to leaving the home, the WANDER WALKER CONCEPT is provided with an information and communications platform including a “Safety Watch” application, and an electrically operated seat with a “Welcome Mode.” Under the concept of “Urban Construction Style,” through the use of a simple twin-frame structure the styling design of the vehicle achieved a stylish yet unselfconscious design that would fit into the urban environment. The use of a frame structure also made it possible to provide a freely usable cargo space in the interior of the unit.

1. Introduction

Committed to a vision of realizing the joy of freedom of mobility and a sustainable society where people can enjoy life, Honda is working towards the realization of a society in which every individual is able to freely move anywhere they desire with a feeling of complete security.

Today, advanced nations are experiencing a rapid decline in birth rates and the aging of their populations (in particular Japan), a rise in environmental awareness and the entrenchment of measures to reduce CO₂ emissions, the concentration of populations in cities⁽¹⁾, and the consequent design of urban spaces to reduce dependence on cars⁽²⁾.

In concrete terms, urban design measures include the creation of high-density residential areas in compact urban spaces in which functions essential to daily life are concentrated, and the examination and implementation of measures such as “road diet” schemes that reduce the number or width of vehicle lanes in order to help ensure space for pedestrians.

This trend, together with the expansion in IT services that make it possible to conduct necessary activities without

leaving the house, has created a new issue, the concern that the spheres of life of city residents, and in particular those with limited mobility, will narrow, and that possibilities for social contact will be lost.

Through actual proof experiments of next-generation personal mobility modalities^{(3), (4)}, Honda has identified the maintenance and improvement of quality of life (QOL), for example through an increased sense of health and vitality, as a new value for mobility.

Based on the findings and experience gained from these actual proof experiments, the impetus to create a new form of urban personal mobility grew stronger, and development efforts were launched.

The completed concept model was called the Honda WANDER WALKER CONCEPT, and was made public at the Tokyo Motor Show 2015.

2. Design Concept

Projecting independent elderly people around the age at which they will return their driving licenses, whose numbers are expected to increase in cities in the future, as

* Automobile R&D Center

target users, the development of the concept commenced based on JIS standards for mobility scooters, which have a long history of use as mobility aids in pedestrian spaces.

During Honda's actual proof experiments of next-generation personal mobility modalities, analysis of the opinions of users of mobility scooters and their experience of operating market mobility scooters in city streets and in stores showed that users faced psychological barriers, infrastructural barriers, and physical barriers, and these represented factors that impeded the carefree and comfortable use of the vehicles.

As shown in Fig. 1, psychological barriers included being looked at by other people, concern about causing trouble or being a burden to other people when going out, and opposition from the family to elderly people leaving the house by themselves.

Infrastructural barriers included the status of roads, for example differences in level, the width of automatic ticket gates, and stores that could not be entered while riding the vehicle, in addition to a lack of space to store the vehicle at home and few places to park the vehicle when out.

The physical barriers were declining eyesight, hearing, and physical strength, and a concern among users that their legs would become weaker if they were to use a mobility aid regularly.

Working to formulate designs and develop technologies that would help to reduce these barriers, the development team for the WANDER WALKER CONCEPT sets out to create a one-person urban micro-mover that would enable users to feel comfortable in getting out into lively city streets.

that would make it possible for the vehicle to be used indoors and outdoors and to access elevators, trains, and other urban infrastructure. A number of turning and drive methods were available for the realization of these goals. Front-wheel steering was set as a precondition, to help ensure that drivers and pedestrians around the vehicle could easily judge vehicle behavior when turning. It was necessary to increase the steering angle of the front wheel in order to restrict the minimum turning radius to no more than 1,000 mm, and a front-wheel-drive system was therefore employed to make effective use of drive force when the vehicle is turning in its place. A tire diameter of 260 mm and a minimum ground clearance of 80 mm were set to help ensure that the vehicle was able to smoothly ride over the differences in level and uneven surfaces that are still encountered in urban areas (Fig. 2). Elements of vehicle design, including ease of entry and exit and the appropriateness of the user's eye point, were considered based on these predetermined conditions (Fig. 3).

Like the user's sitting position, the steering method is also an important element in considering package design. The joystick-type control used in some mobility scooters was initially a candidate for the steering method. This method presented the advantages of making it possible to control the vehicle with one hand and presenting few restrictions in terms of space, but it was judged that the use of this method would make it more challenging for users attempting to support themselves at such times as when riding over rough surfaces at a speed of 6 km/h or braking suddenly.

By contrast, when the handlebar steering method used in

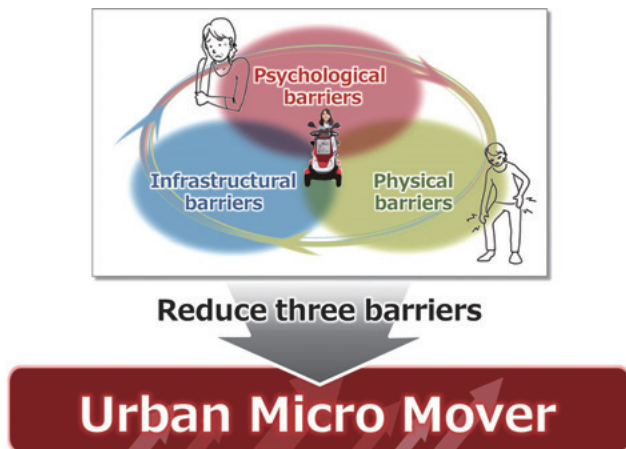


Fig. 1 Grand concept

3. Package Design

In order to realize a package design that would contribute to reducing infrastructural barriers, the target for the vehicle's width was set at no more than 550 mm, and the target for its minimum turning radius at no more than 1,000 mm, a size

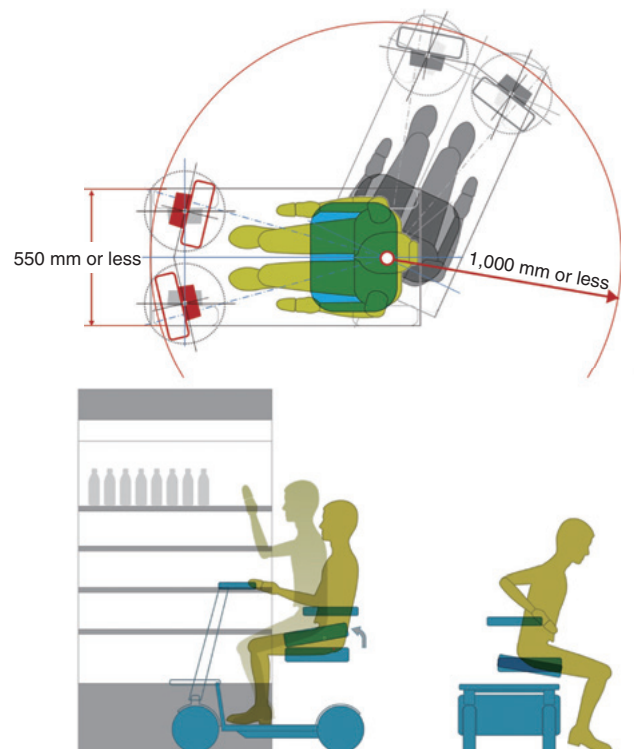


Fig. 2 Package concept

some mobility scooters is employed, loads on the upper body are naturally absorbed. This is not the only advantage of the handlebar steering method. The users of mobility scooters are treated as pedestrians, and it is necessary for other pedestrians around them to be aware of their intentions. The handlebar steering method, as also used on bicycles, has the advantage of allowing people close to the vehicle to readily perceive the rider's intentions. Based on these considerations, a twin-lever steering (TLS) system, to be discussed below, was adopted for the WANDER WALKER CONCEPT.

A questionnaire survey of mobility scooter users conducted during actual proof experiments of next-generation mobility modalities indicated that the main source of dissatisfaction with the vehicles was the limited number of spaces to carry packages, and the small volume of those spaces. The WANDER WALKER CONCEPT was designed to provide a cargo space that makes it possible to carry a 12-roll pack of toilet paper or a five-box pack of tissues: everyday items that the elderly have trouble transporting.



Fig. 3 Package prototype

4. Styling Design

Psychological barriers can be considered to be the main barriers among the three types of barrier impeding the use of vehicles of this type. Because the use of personal aid equipment can cause others to regard the user as somehow disabled, people may avoid its use if they can, or they may tend to avoid going out because of a sense that devices such as wheelchairs are a nuisance to people around them.

Development team therefore sets "Urban Construction Style" as the styling concept for a stylish but unselfconscious design that would help prevent users from feeling any embarrassment, and sought a form that would blend into the urban environment.

Using contemporary urban landscape designs that create comfortable and pleasant city spaces as a reference, development team attempted a fusion of designs of conveyances that travel at a maximum speed of 6 km/h.

The lightness and see-through structure of the simple twin-frame configuration, in which left and right frames enclose the vehicle space, blend into the streetscape like a piece of modern street furniture (Fig. 4). The empty spaces in the vehicle's structure were also designed to function as free cargo spaces with unlimited usage. A distinctive feature of the configuration is that even if packages or bags are placed inside the twin-frame structure, it does not affect the vehicle's styling (Figs. 5 and 6).

Solid tires do away with the worry of punctures. The rims feature a geometric pattern composed of hexagons and squares. The use of holes in the geometric pattern sought to convey a sense of lightness and agility (Fig. 7).



Fig. 4 Final exterior rendering

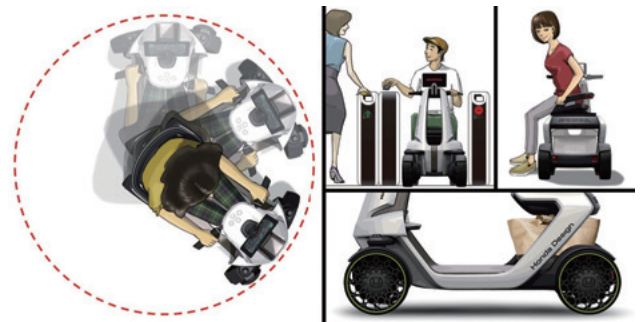


Fig. 5 Features of Honda WANDER WALKER CONCEPT



Fig. 6 Final exterior



Fig. 7 Airless tire

5. Interface Design

It was necessary to attempt to ensure that the operation of various devices would not in itself become a barrier to the use of the vehicle for its target elderly users, who would be beginning to be aware of a decline in their physical powers with advancing age. However, taking elderly users into consideration and making the functions and interface overly obvious would result in people hesitating to use the functions of the unit and thereby become a psychological barrier. The development team therefore aimed to realize an interface based on universal design (UD) principles, combining intuitive operation that is easy for anyone to understand with a welcoming feeling achieved through the use of new technologies. The distinctive features of this design are TLS, a seat with a user assist mechanism linked to the user recognition system, and the “Safety Watch” application. Based on the belief that the use of information and communication technology (ICT), for example links between mobile devices, and an information and communications platform for Intelligent Transport Systems (ITS), such as V2X and automatic charging and automatic parking, will be important for personal mobility aids in the future, the design of the WANDER WALKER CONCEPT incorporates an 8-inch tablet as the foundation for a communications system (Fig. 8).

TLS is an original Honda steering method that alternates between front and rear. Input is intuitive, and the system’s ability to be operated by means of the elongation and contraction of biarticular muscles without twisting of the joints has been proven to be effective in reducing fatigue⁽⁵⁾. At the same time, the use of steer-by-wire control further reduces steering force. The steering angle of the front wheels is controlled in response to the degree of movement of the handlebars to the front or rear; the vehicle can be turned in place only by turning the handlebars as far as possible when the vehicle is stationary. TLS is also provided with a handlebar function that helps to enable the user to maintain their riding posture, offering a new steering mechanism that is easy even for the elderly to use (Fig. 9).

To make the flow of procedures for getting into the vehicle smooth for users with weak legs, the vehicle is

provided with a seat with a “Welcome Mode.” When the user approaches the vehicle holding the tablet that functions as the key, the lights flash and the seat rotates outward to the user’s position and tilts 10 degrees downwards in order to assist the user in taking their seat. In addition, when the user sets the tablet in a predetermined position, the vehicle automatically goes into standby mode. The vehicle therefore does not automatically switch to a state in which it can immediately be driven; the purpose of this is to help prevent unintended takeoffs, for example if the user accidentally operates the exposed controls for advancing the vehicle.

Objections from elderly people’s families represented a psychological barrier to use of the vehicle. A “Safety Watch” application was therefore developed during actual

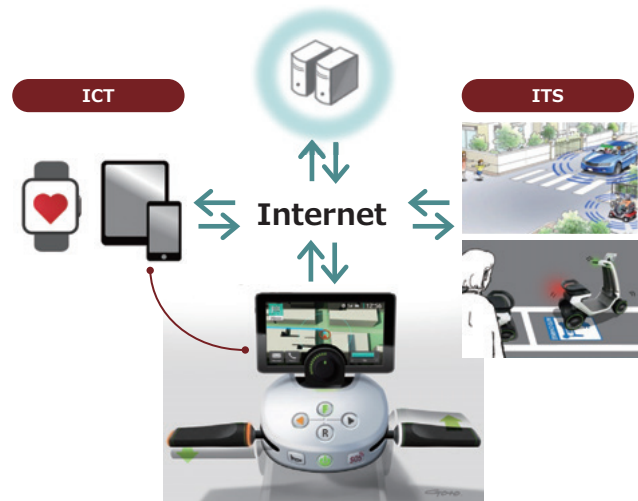


Fig. 8 Connection via ICT and ITS



Fig. 9 Pivot turn mode realized by TLS

proof experiments of next-generation mobility modalities. The results of a survey of users and their families indicated that the sense of security offered by knowing that the elderly user is being watched over and the sense of security offered by watching over the elderly user are reciprocal, suggesting that the application could be an effective tool in supporting the elderly in going out by themselves.

The use of a variety of other applications, for example a traffic alert application, applications to provide information, and health management software, can also be considered in addition to the application described above. These would reduce the psychological barriers to going out in the vehicle, and make it possible for users to go where they want to go, rather than merely to where they are able to go. This is the starting point for the joy of movement, allowing everyone to freely move anywhere they wish with a feeling of security.

6. Color Design

Based on the concept of urban style with a sense of warmth, the color concept for the vehicle was set as “Urban Natural Coordinate.” This enabled an expression that offers a sense of functional beauty in addition to aesthetic beauty (Fig. 10). A combination of white, presenting an image of advancement, with accent colors has been employed for the vehicle. The white used as the exterior color is contrasted with a warm, dark-toned brown wood-grain pattern, emphasizing the vehicle’s framework. A fluorescent yellow-green has been used as an accent on the sides of the tires to make going out fun, and the use of a reflective material for the color contributes to increased visibility. The vehicle’s main white pearl color has gold highlights that make the styling stand out when the vehicle is turning in its place or executing other unique movements. The coating, which highlights the design, is made up of two coats. The first base

coat is solid white, while the main content of the second base coat is glittering glass flakes and gold interference pearl pigment. This creates a hue that glitteringly reflects the sunlight to emphasize the solidity of the styling, while also blending into contemporary urban streetscapes.

7. Concept and Social Acceptance

The Honda WANDER WALKER CONCEPT was exhibited at the Tokyo Motor Show 2015 (Fig. 11). A questionnaire survey conducted at Honda’s Personal Mobility Concept booth received 215 responses. These responses contained 40 comments concerning the Honda WANDER WALKER CONCEPT. Approximately 80% of these were from males in their 20s, indicating a greater than anticipated interest in the concept among young people, and suggesting the potential for the concept to be developed as a



Fig. 11 Tokyo Motor Show 2015

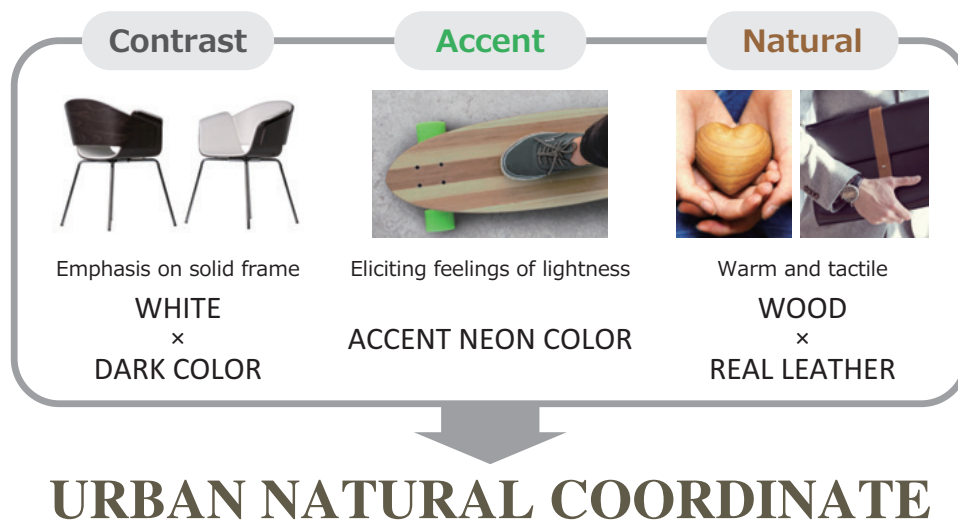


Fig. 10 Color concept

new form of personal mobility aid.

Four comments concerned the reception of the concept, for example that it took Japan's aging society into consideration; one respondent indicated that they would want their parents to use the vehicle. Thirty comments concerned the reception of the vehicle's functions, positively evaluating its small turning radius, compact size and cargo space. Six comments concerned the reception of the vehicle's styling. Two comments added the qualification that the vehicle was a mobility aid for the elderly, but opinions were generally positive, with praise for the excellence of the simple and non-tiring design.

Eleven comments were received from journalists, and articles on the concept were published in 15 media outlets.

As a proposal that seeks to respond to the special social issues facing Japan as a "super-aging" society, the concept resonated strongly with Motor Show visitors. A selection of representative comments is presented below.

"This is a great initiative for an aging society. Honda's stance of appealing to the social responsibility of car companies is praiseworthy." "It's a concept that has gone back to the starting point of the car while looking towards the future. Our aging society has been waiting for a mode of transportation like this one." "The level of maneuverability of the compact body is impressive." "TLS is a technology that is well-suited to the elderly." "An insightful design based on a monotone color scheme that can be used at any time, place, or opportunity." These responses demonstrate the receptiveness of the public to the proposed concept and design.

8. Conclusion

The development team for the Honda WANDER WALKER CONCEPT made the most of the design techniques and sensibility employed when designing Honda cars, and created an appealing personal mobility aid that responds to the mobility needs of people living and going about their business in compact urban spaces, which will contribute to creating livelier cities.

Looking towards the holding of the Tokyo Olympics and Paralympics in 2020, the transport services environment is targeting a diverse range of users and beginning to undergo considerable change. In November 2016, the Ministry of Land, Infrastructure, Transport and Tourism established the Committee for the Review of the Use of Mobility Scooters on Public Transport⁽⁶⁾, to reconsider necessary conditions for users and for mobility scooters themselves when the scooters are used on trains and elsewhere in the public transport system. The committee has commenced its reconsideration of the conditions for the use of mobility scooters when using public transportation.

In the future, based on analysis of the public's reaction at the Tokyo Motor Show 2015 and responses to the associated questionnaire, in addition to changes in social trends, the development team intends to create a functioning

model of the Honda WANDER WALKER CONCEPT that takes potential mass production into consideration, making it possible to refine the unit's design and functions in the real world, and to study its potential as a communication tool that helps to connect people with society.

Honda will continue to advance research and development efforts towards the realization of a society in which all people are able to move freely wherever they wish with a feeling of complete security.

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Honda Smart Home System Experiment Results

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ABSTRACT

An experiment has been carried out over a period of approximately three years using the Honda Smart Home System, which comprises micro combined heat and power generation system in combination with photovoltaic power generation, electric vehicle, and Smart e Mix Manager and is conceived to reduce CO₂ emissions from ordinary homes to one-half and then eventually to zero. The purpose of the experiment was to study the effectiveness of the system in reducing CO₂ emissions from the home, to examine the issues involved in reducing CO₂ emissions without lessening the quality of life, and to explore possible approaches to achieving zero CO₂ emissions. At a building identical to the experimental house, electric power was purchased from an electric utility company and a highly efficient gas hot water heater was used to make hot water on the assumption that a conventional household was consuming the same amount of electric power and heat as the household where the experiment was conducted. The result was that the experimental house achieved a 51% reduction in CO₂ emissions. When solar water heaters, biofuel, and other such renewable energy sources were used, a potential direction was found for achieving zero CO₂ emissions.

1. Introduction

There are calls at present to reduce emissions of greenhouse gases (GHG) that are one of the major causes of global warming. The use of photovoltaic (PV) power and other such forms of renewable energy is being pursued as one measure to that end⁽¹⁾. Measures are also being taken to build systems of local production for local consumption of energy⁽²⁾. This involves the use of distributed energy sources in an area-wide fashion within communities of a certain size so that in normal times the systems function to conserve energy and in times of emergency they can function to assure the supply of energy. There are movements among local governments to use such systems to revitalize local industry as well as to seek out ways to strengthen local communities against disaster. Within this context, Honda Motor Co.,Ltd and Saitama City entered into an agreement to implement the E-KIZUNA Project to use a diversity of energy sources and a variety of electric-powered vehicles in an initiative leading to proposals for a disaster-resistant environmental city of the future⁽³⁾. Specifically, this calls for the construction of houses for experimental use on land owned by Saitama City where experiments are being conducted to examine the local production of energy for local consumption and the possibilities for a community of homes capable of functioning independently, as well as to verify the effectiveness of the Honda Smart Home System⁽⁴⁾

(HSHS) as a supply of energy for household use that is also capable of dealing with emergencies.

2. Purpose

- (1) The first purpose is to demonstrate the effectiveness of the HSHS, with the objective of halving CO₂ emissions.
- (2) The second purpose is to propose a direction to pursue in order to achieve reduction of CO₂ emissions to zero in 2020.

3. Experiment Method

An overview of the experiment's measurement and observation period, the objects of measurement and observation, the equipment used, and other such details, is given in the following.

3.1. Overview of the Period of the Experiment

The buildings and measurement systems were completed at the end of April 2012, and occupancy of the buildings began in May. After a period for confirmation and adjustment of measurement system operation, the data from July onward was used for analysis in the experiment. The electric vehicle (EV) was used from December 2012 to the end of November 2014. The experiment was performed after obtaining approval from the Honda Bioethics Committee.

* Power Products R&D Center

3.2. Object of the Experiment

The experimental house is a two-story, light-gauge steel prefabricated house manufactured by Sekisui House, Ltd. The exterior appearance of the house is shown in Figs. 1 and 2.

Table 1 shows a summary of the house specifications. The building has a heat loss coefficient [$2.7 \text{ W}/(\text{m}^2 \cdot \text{K})$ or lower]⁽⁵⁾ that satisfies the revised “Builder Judgment and Standards Regarding the Rationalization of Energy Use Involving Residences” announced by Saitama City in 1999.

One air conditioner for heating and cooling is installed in each room of the house, and Table 2 shows a summary of the air conditioner specifications. The summary below presents published figures.

Table 3 shows a summary of the gas fan heater and hot water heater specifications. Table 4 shows a summary of the hot water floor heating specifications. These items of heating equipment are installed on the first floor.

A summary of the residential facilities in the experimental house is given in the following:

- LED lighting and a 24-hour ventilation system were used.
- A micro combined heat and power generation system was used as the heat source for the hot water supply and hot water heating equipment.
- In December 2013, the Japanese-style room on the first floor was equipped with a gas fan heater.
- A gas range was used for kitchen equipment.
- City gas (13A) was used for the gas.



Fig. 1 Southwest side appearance of experimental house



Fig. 2 Northwest side appearance of experimental house

Table 1 Experimental house specifications

Total floor space	171.13 [m^2]
Heat loss coefficient	2.03 [$\text{W}/(\text{m}^2 \cdot \text{K})$]
Building orientation	South

Table 2 Air conditioner specifications

Installation location	Cooling capacity	Heating capacity
1st floor living room	7.1 [kW]	8.5 [kW]
1st floor Japanese-style room	2.2 [kW]	2.5 [kW]
2nd floor master bedroom	2.2 [kW]	2.5 [kW]
2nd floor south bedroom	2.2 [kW]	2.5 [kW]
2nd floor north bedroom	2.5 [kW]	2.8 [kW]

Table 3 Heater specifications

Heater type	Installation location	Heating capacity
Hot water heater	1st floor Japanese-style room	3.5 [kW]
Gas fan heater	1st floor Japanese-style room	5.81 [kW]
Hot water heater	1st floor bathroom	4.1 [kW]

Table 4 Hot water floor heating specifications

Installation location	Heating area
1st floor kitchen	2.4 [m^2]
1st floor living room	8.0 [m^2]
1st floor dining room	7.2 [m^2]

3.3. Equipment Used and Equipment Comparison

The equipment used in this experiment is described below together with the equipment envisioned for comparison.

3.3.1. Micro combined heat and power generation system

The micro combined heat and power generation system (MCHP) comprises a generator unit that uses a gas engine generator and a hot water tank unit with a built-in auxiliary boiler. All the electricity generated by the generator unit is consumed in the house to which the MCHP is connected. When surplus electric power is produced, it is converted into hot water using a heater built into the generator unit. The experimental house was equipped with a system that has the same performance as the ECOWILL (a registered trademark of Osaka Gas Co., Ltd.)⁽⁶⁾, which is a commercially available MCHP, but it was given the additional functionality of operating independently to generate power on command from the Smart e Mix Manager (SeMM)⁽⁴⁾ during a power outage. Table 5 shows the generator unit specifications and Table 6 shows the hot water tank unit specifications.

3.3.2. Photovoltaic power generation

The photovoltaic power generation system uses solar cells and a power conditioner (PCS) manufactured by Honda Soltec Co., Ltd. The electricity generated by the PV system is consumed in the house to which the PV system is connected. When surplus electric power is produced, it

Table 5 MCHP power generation unit specifications

Power generation output	1.0 [kW]
Exhaust heat output	2.5 [kW]
Gas consumption	4.21 [kW]

Table 6 MCHP hot water tank unit specifications

Hot water temperature	Approx. 75 [°C]
Hot water storage tank capacity	90 [L]
Reheating capacity	12 [kW]
Heating capacity	17.4 [kW]
Maximum gas consumption	50.0 [kW]

is supplied to an electric utility company by reverse power flow. The general specifications for the solar cell array are shown in Table 7 and those for the PCS in Table 8.

3.3.3. Smart e Mix Manager

The SeMM is a device for the integrated control of heat and electric power when the MCHP, PV, and EV are used in combination. It also has the functionality to start up the MCHP during power outages and supply electric power, combined with that from the PV system, to designated wall sockets. Figure 3 shows a schematic diagram of the HSHS.

3.3.4. Vehicle

A FIT EV⁽⁷⁾ was used as the main means of transportation for the experimental house residents. An on-

Table 7 Photovoltaic array specifications

Manufacturer	Honda Soltec
Cell type	CIGS
Nominal maximum output	4.44 [kW]

Table 8 PCS specifications

Manufacturer	Honda Soltec
Rated capacity	4.0 [kW]
Power conversion efficiency	94.5 [%]

board logger was installed on the FIT EV to record such measurements as distances traveled and amount of charge.

3.4. Measurement

For the present experiment, certain measurements were made every second under approximately 200 headings, including electric power, pressure, temperature and flow rate, in order to analyze the house, the MCHP, and the PV in terms of the amounts of electric power, the amounts of heat demand and supply, and environmental conditions. Since the amount of heat could not be measured directly, it was calculated from the measured temperature, pressure, and flow rate.

3.5. Method for Evaluating CO₂ Reduction Effect

In the present experiment, data was actually collected for only one household. It was reasoned that the CO₂ reduction

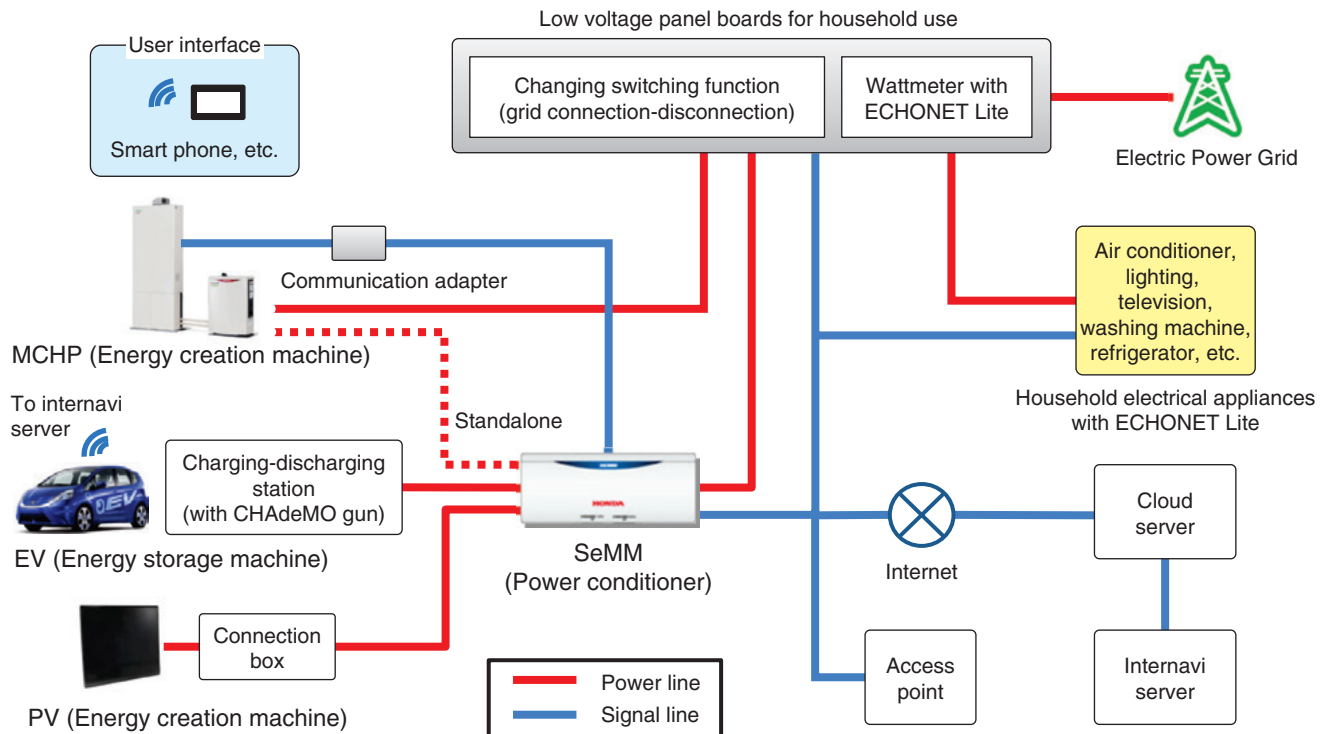


Fig. 3 System configuration of HSHS

amount due to the HSHS could be appropriately compared against measurements made in a house with entirely the same living environment and lifestyle but without HSHS. The conditions and method were therefore configured as shown below.

- Assume that there is a conventional house where HSHS has not been introduced and estimate the amounts of electric power and gas purchased in order to satisfy the same energy demand as in the experimental house. This house will serve as the object for comparison of the CO₂ reduction amount, and it will be designated the conventional house. The differences between this conventional house and the experimental house in the actual electric power and gas purchase amounts are the energy reduction effects. They are converted to CO₂ reduction amounts using the CO₂ emissions per unit of their respective energy sources. Table 9 shows the CO₂ emissions per unit⁽⁸⁾ used for this calculation.
- The electric power demand in the conventional house is considered to be the sum of the “electric power and lighting” used by means of the wall sockets or other such sources in the experimental house and the power consumed by air conditioners. This electric power is purchased from the electric utility company. The amount of electric power purchased W_{vhc} is obtained from the “electric power and lighting” amount W_{dpl} and the “air conditioning” power consumption W_{acu} by means of Eq. (1).

$$W_{vhc} = W_{dpl} + W_{acu} \quad (1)$$

- In the conventional house, gas was purchased from the gas company, and a high-efficiency water heater and gas range were used for hot water and cooking. Rooms were heated and cooled using air conditioners and hot water heating from the high-efficiency water heater. For the high-efficiency water heater⁽⁹⁾, equipment capable of hot water supply and hot water heating that was commonly available on the market in 2015 was selected. The general specifications are shown in Table 10.
- The conventional house heat demand Q_{dhc} was obtained from the experimental house “hot water supply” demand Q_{hws} , the experimental house “reheating” demand Q_{rh} , the experimental house “floor heating” demand Q_{wh} , and the high-efficiency hot water heater efficiency η_{ej} using Eq. (2).

$$Q_{dhc} = (Q_{hws} + Q_{rh} + Q_{wh}) / (\eta_{ej} / 100) \quad (2)$$

- The conventional house amount of gas purchased V_{vhgc} was obtained from the conventional house heat demand Q_{dhc} , the amount of heat from city gas Hu_g , the amount of gas consumed by the experimental house gas fan heater V_{gfr} , and the amount of gas consumed by the experimental house gas range V_b using Eq. (3).

$$V_{vhgc} = Q_{dhc} / Hu_g + V_{gfr} + V_b \quad (3)$$

Table 9 CO₂ emissions per unit

Type	CO ₂ emissions per unit
Electric power	0.192 [kg-CO ₂ /MJ]
City gas 13A	0.057 [kg-CO ₂ /MJ]
Gasoline	2.322 [kg-CO ₂ /L]

Table 10 High-efficiency water heater specifications

Energy consumption efficiency	Energy consumption efficiency	93 [%]
	Hot water supply	95 [%]
	Hot water heating	87 [%]

Table 11 Vehicle specifications

		Experimental house	Conventional house
Name		FIT EV	FREED
Model		ZAA-ZA2	DAB-GB3
Dimensions	Full length	4,115 [mm]	4,215 [mm]
	Full width	1,720 [mm]	1,695 [mm]
	Total height	1,580 [mm]	1,715 [mm]
Mass		1,470 [kg]	1,290 [kg]
Fuel consumption (JC-08)			16.6 [km/L]
		106 [W·h/km]	
Battery	Type	Lithium ion	
	Capacity	20 [kW·h]	

- The distance traveled by the vehicle used at the conventional house was made the same as the distance traveled by the FIT EV used at the experimental house. A Honda Freed⁽¹⁰⁾, which is about the same overall size as the FIT EV, was chosen as the vehicle. Table 11 shows the general specifications of the Fit EV and the Honda Freed.

3.6. Residents

The household consisted of a family of four, with a husband and wife, both of whom work, an adult child who works, and a second child who is a university student.

4. Results

The amount of electric power purchased, the amount of gas purchased, and the heat demand as measured in the experimental house are shown in the following.

4.1. Amount of Electric Power Purchased and Amount of Gas Purchased

Figure 4 shows the amounts of electric power and gas purchased per year during the measurement period and as averages for the entire period, categorized by type. Annual data is for the period from July to June of the following year. Electric power was used within the household and for EV charging. In order to clarify the differences in quantity involved, these are shown separately as electric power (without vehicle) used within the household for “electric power and lighting” and “air conditioning” and as electric

power (EV charging) used for “EV charging.” The amount of PV power generated shows “PV power usage” used in the household and the remainder sent to the electric utility company by reverse power flow as “PV reverse power flow” separately in order to clarify the differences in quantity involved. The reduction in the amount of electric power purchased corresponded to the amount of PV power generated and used in the household. The electric power sent to the electric utility company by reverse power flow was consumed through that company, and the power generated at the power plant can be reduced by that amount, so these amounts were shown as negative figures. The amount of gas is the total amount consumed by the MCHP, gas range, and gas fan heater. In this figure, the portion from July 2014 to March 2015 is nine months’ worth of data. The figures at the far right combine the average monthly figures for the year to produce average annual figures for the total measurement period. The total amount of electric power and gas purchased amounted to approximately 68,000 MJ.

4.2. Energy Demand

Figure 5 shows the energy demand figures from Fig. 4 recategorized by use. The national averages⁽³⁾ for 2013 are also shown on the right end of the figure for reference.

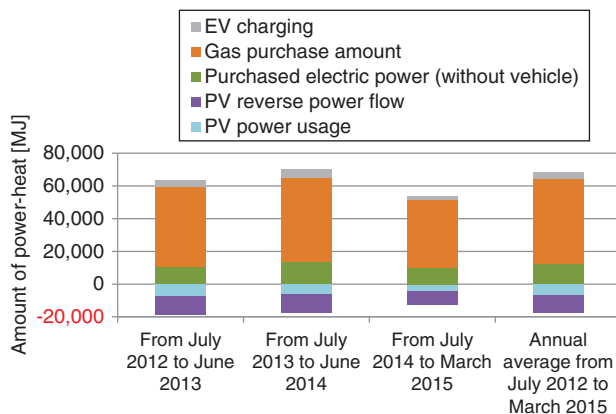


Fig. 4 Electricity and gas annual purchase amount

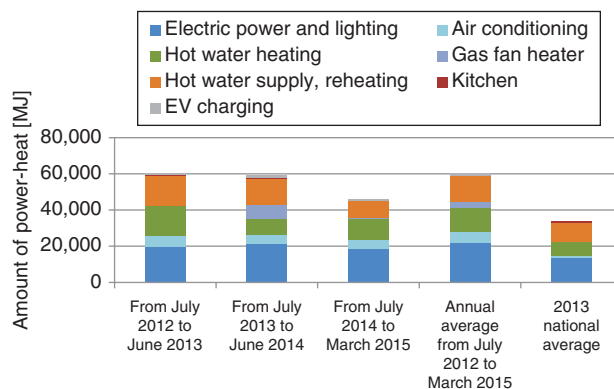


Fig. 5 Annual power and heat consumption

The amount of gas consumed by the MCHP cannot be categorized by use, so it was shown as heat demand divided between “hot water supply and reheating” and “hot water heating.” Figures other than those are shown as the “amount of electric power supplied” and the “amount of gas purchased.”

The annual demand for electric power and heat, excluding EV charging, amounted to approximately 61,000 MJ. This figure is lower than the total purchased amount shown in Fig. 4 because the demand for “hot water supply and reheating” and “hot water heating” is shown in the amount supplied by the MCHP, so that the figure is influenced by the MCHP gas-to-heat conversion efficiency.

The annual demand for electric power and heat came out larger than the national average of approximately 36,000 MJ by a factor of 1.7. The consumption for “electric power and lighting” and for “air conditioning” was greater than the national averages.

In “electric power and lighting,” the component percentages of consumption as a whole were generally the same as the national averages. However, the amounts consumed were 1.7 times greater than the national averages. Moreover, “air conditioning” was 6.4 times greater than the national average, and “heating” (in the experimental house, this was hot water heating and the gas fan heater) was twice the national average.

Figure 6 shows electric power consumption, heat demand, and gas consumption by application and by month. “EV charging,” “hot water supply and reheating,” “hot water and gas heating,” “air conditioning,” and “electric power and lighting” show the amounts of electric power consumed by the respective applications and their heat

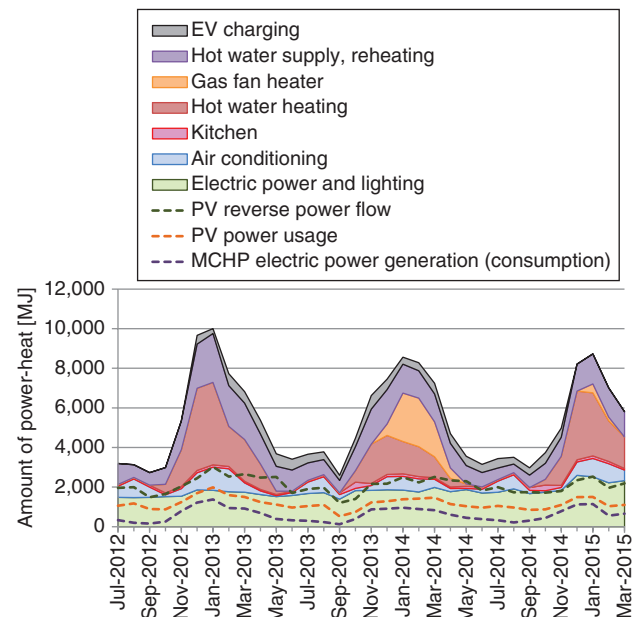


Fig. 6 Monthly electricity and heat consumption by application

demand. They are presented in a stacked area graph. The PV power generated is divided and shown as “PV power usage” and “PV reverse power flow.” The PV power generated and MCHP power generated are shown in a stacked line graph. Showing consumption of electric power, the total of “air conditioning” and “electric power and lighting” and the total of “PV power usage,” “PV reverse power flow,” and “MCHP power generated” are very similar figures in every month.

In December 2013, a gas fan heater was introduced into the Japanese-style room on the first floor to replace the gas fan heater and hot water heater with the aim of raising the efficiency per quantity of heat used for heating. The amount of electric power for air conditioning and the heat demand for the period from January to March 2013 (before introduction of the gas fan heater) and the period from January to March 2014 (after introduction of the gas fan heater) were compared. As a result, it was found that the electric power for air conditioning was reduced to 63% of what it was before introduction of the gas fan heater. However, use of the gas fan heater caused gas usage to increase, and the heat demand with the use of floor heating and the gas fan heater increased to 131% of what it was before introduction of the gas fan heater. Consequently, the electric power and heat consumption for heating during this period increased to 114% of what it was before introduction of the gas fan heater, and no reduction effect was obtained with regard to electric power and heat demand.

4.3. Electric Power Consumption

As can be see in Fig. 6, electric power consumption is greater in the summer and winter, due to air conditioning (heating and cooling) consumption.

Figure 7 shows hourly electric power consumption excluding EV charging in January (winter), May (intermediate period), and August (summer). In every case, consumption has peaks in the morning and evening, and is greater at night in particular.

Figure 8 shows the relationship between the average outdoor air temperature per month and the electric power

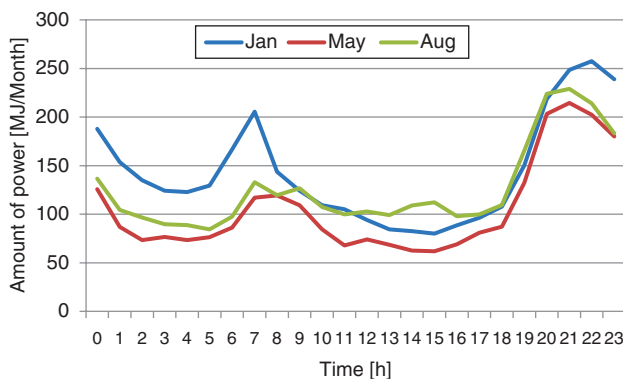


Fig. 7 Hourly electric power consumption with EV charging excluded

consumption in those months. The difference in air conditioning power consumption is not due to the difference between heating and cooling but is rather influenced by the outdoor air temperature.

4.4. Gas Consumption

Figure 6 shows that the consumption of gas in winter is approximately five times greater than in summer, since, in addition to kitchen use, it is used as a heat source for the hot water supply and for heating.

Figure 9 shows hourly MCHP gas consumption for January, May, and August. Regardless of the season, consumption was greater in the morning and evening, when more was consumed for hot water supply and reheating.

4.5. Heat Demand

Figure 5 shows that heat demand accounts for one-half of all energy consumption and that heat demand for heating makes up one-half of total heat demand. This can be broken down as follows.

4.5.1. Hot water heat demand for heating

Figure 10 shows monthly heat demand for hot water used for heating. Heating with hot water used from October

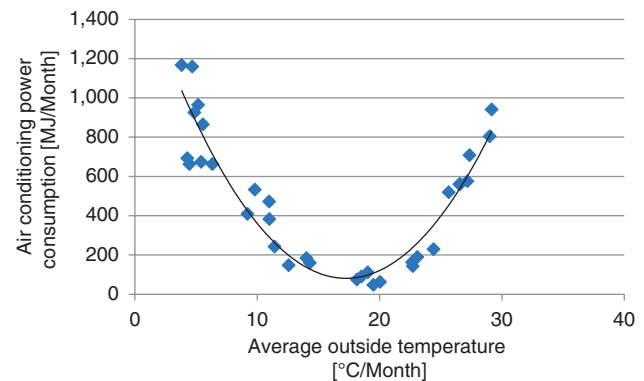


Fig. 8 Average air temperature and air conditioning power consumption

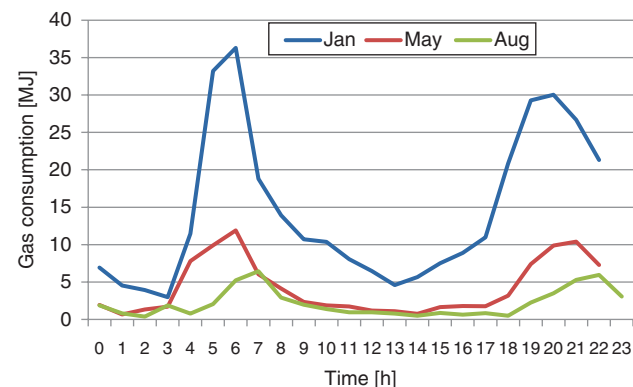


Fig. 9 Hourly MCHP system gas consumption by month

to May of the following year and consumption reached a maximum of 4,200 MJ/month. December 2013 showed a lower level than that month in other years because the hot water heater was changed to a gas fan heater.

Figure 11 shows the relationship between the average outdoor air temperature for a month and the hot water heat demand for use in heating in that month. Hot water heating began to be used from around when the average outdoor temperature fell below 20°C, and the demand increased as the average outdoor air temperature dropped.

4.5.2. Hot water supply and reheating heat demand

Figure 12 shows the hourly hot water supply and reheating heat demand for January, May, and August. Residents commented that they bathed in the morning, and this was supported by the higher morning demand. There was also daytime demand in the summer, seen in August, which differed from consumption for heating.

4.6. Reduction Effect on CO₂ Emissions

Figure 13 shows CO₂ emissions by type for the conventional house and for the experimental house, which were evaluated by the method described in Section 3.5. from the energy consumption shown in Figs. 4, 5, and 6. In the experimental house, the use of the MCHP meant

that gas consumption was higher and CO₂ emissions, therefore, increased by a corresponding amount. On the other hand, power generation by the MCHP and PV meant that the amount of electric power purchased was lower, and CO₂ emissions decreased by a corresponding amount. On balance, CO₂ emissions from the experimental house showed a decrease of 51% compared to the conventional house.

Figure 14 shows application-specific CO₂ emissions by month. In this figure, the CO₂ emissions for “air conditioning,” “electric power and lighting,” and “EV charging” are from electric power consumption, while the CO₂ emissions for “hot water and gas heating” are from the consumption of gas used for hot water and heating. For “hot water supply and reheating” and “hot water heating,” MCHP gas consumption was differentiated by the percentage of heat demand under each heading and the CO₂ emissions were calculated accordingly. These were all represented in a stacked area graph. Since the CO₂ reduction effect cannot readily be categorized by application, it was categorized instead by equipment type and displayed in the graph accordingly. The “EV use effect” is the difference in CO₂ emissions between the gasoline vehicle that is the

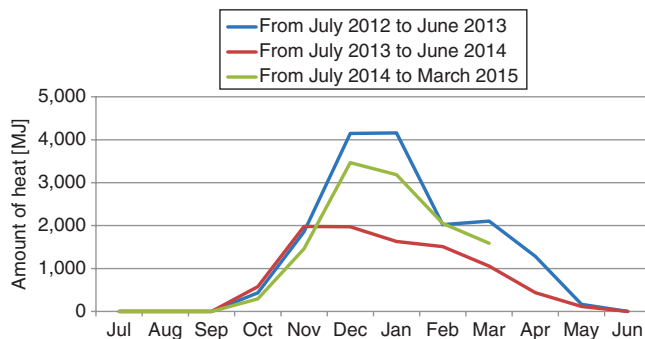


Fig. 10 Monthly hot water heat consumption for heating

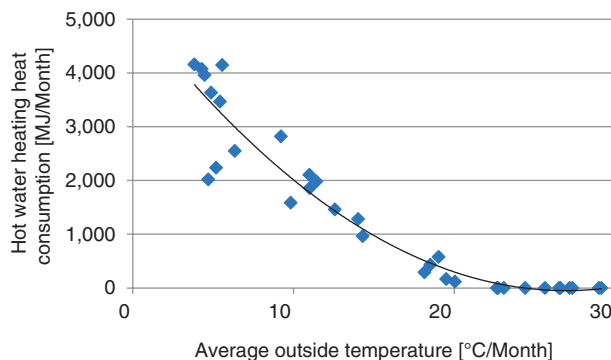


Fig. 11 Average outside temperature and hot water heat consumption for heating

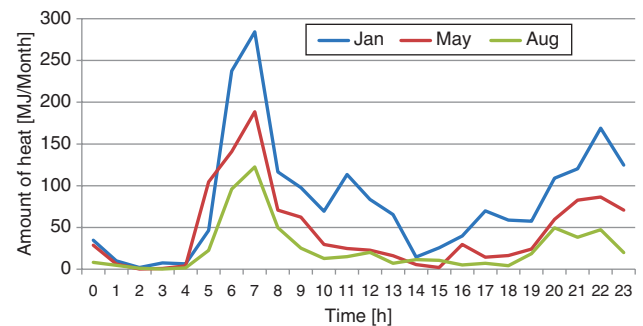


Fig. 12 Monthly hot water supply - Reheating heat consumption by hour

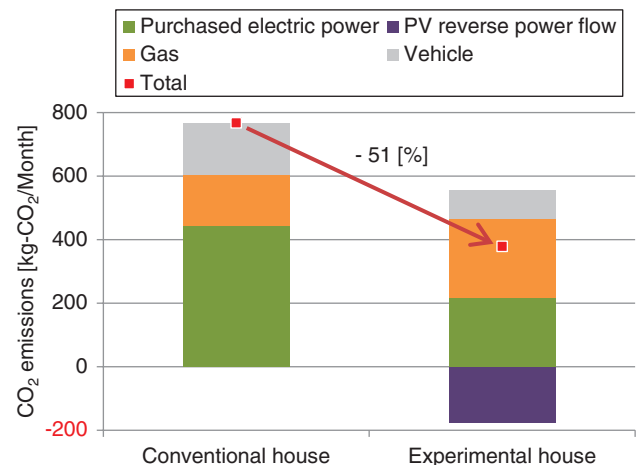


Fig. 13 CO₂ emissions by type

object of comparison and the FIT EV for travel over the same distance. The “MCHP use effect” is the difference in CO₂ emissions from the amount of power purchased from an electric utility company if an amount corresponding to MCHP power generation were to be purchased instead, when CO₂ emissions corresponding to the amount from gas consumption for MCHP power generation are subtracted. The amount of gas consumed for MCHP power generation in this case is separated out from the amount of gas consumed by the MCHP generator unit according to the ratio of the amounts from power generation and heat generation. The resulting figure displayed corresponds to the amount from power generation. The “PV power usage” represents the amount of CO₂ emissions from the portion of PV-generated electric power consumed within the household that would have been emitted if that amount of electric power had been purchased from the electric utility company. The “PV reverse power flow electric power” likewise represents the amount of CO₂ that would have been emitted if the amount of PV-generated power that went to reverse power flow had been generated by the electric utility company. These are shown by dotted lines in the stacked graph that represent the CO₂ reduction amounts.

While the seasonal variation in CO₂ emissions from “electric power and lighting” and “hot water supply” is small, the variation in “air conditioning” and “hot water and gas heating” is large. The total reduction in CO₂ emissions from MCHP, PV, and EV came out as a figure close to the

sum of emissions from “electric power and lighting” and “hot water supply.” In the spring and autumn, in particular, when energy consumption within the household is lower, those figures were practically identical. Viewed from the perspective of the contribution to CO₂ reduction, the PV has a significant effect throughout the year while the MCHP had a lesser effect in the summer, when heat consumption is lower and operating times are shorter.

Figure 15 shows CO₂ reductions as percentages of the annual average. The PV has a major effect, and when the effects of the PV generated electric power used within the household and the reverse power flow of electric power to the electric utility company are combined, they account for 72% of the total CO₂ reduction.

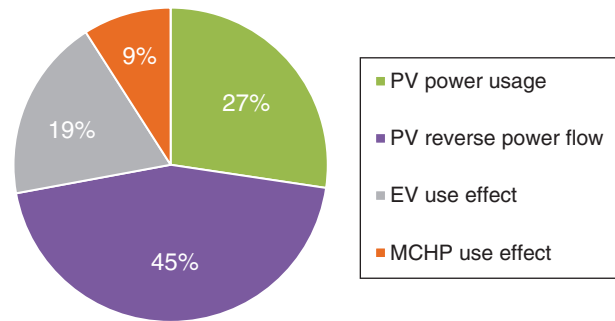


Fig. 15 CO₂ reductions

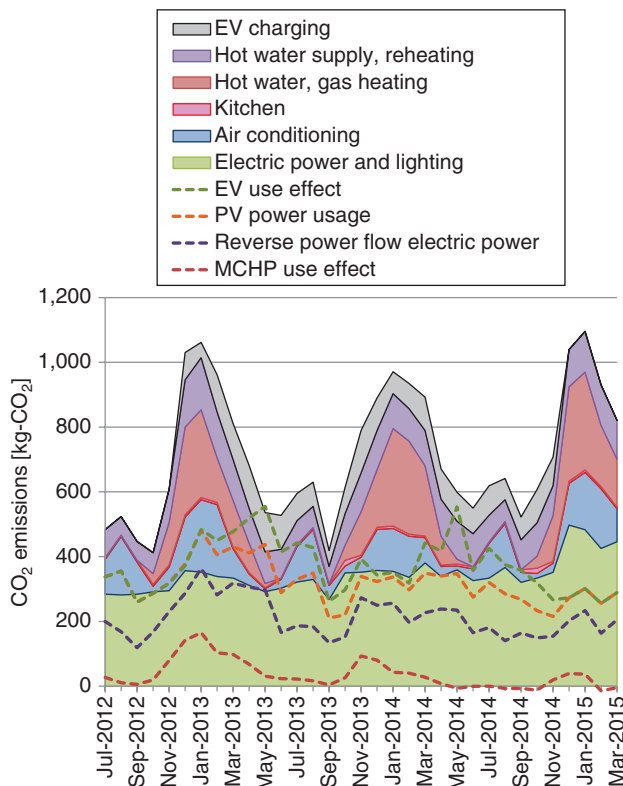


Fig. 14 Application-specific CO₂ emissions by month

5. Discussion

It became apparent in Section 4 that the PV is highly effective in reducing CO₂ emissions. However, if PV output were to be increased with the aim of achieving a still higher CO₂ emissions reduction effect, and the amount of reverse power flow in the region as a whole were to increase, there is a possibility that restrictions would be placed on PV output⁽¹⁾. This means that zero CO₂ emissions cannot readily be achieved simply by increasing PV output. The approach to be examined, therefore, will be a method that takes local production of energy for local consumption into account and works to achieve zero CO₂ emissions in households where the amount of electric power purchased and the amount of reverse power flow of power generated by PV are normally the same.

It became clear in Section 4 that it will be necessary to reduce the CO₂ emissions originating in heat demand that is supplied mainly by purchased gas. The following are conceivable methods for practical implementation of this aim.

- (1) Use solar water heaters, heat pumps, and other such heat source equipment that is more highly efficient to reduce the amount of gas purchased.
- (2) Replace purchased gas with biofuel or other such renewable energy sources.

For method (1), the possibility of using a solar water heater to reduce CO₂ emissions from gas purchased for hot water supply, reheating, and heating will be examined.

Figure 16 shows average monthly heat absorption efficiency for a vacuum tube-type solar water heater from measurements made separately from the present experiment.

Based on these results, the effect of installing an additional solar water heater like that in Table 12 is estimated, with the conditions shown below.

- The heat energy supplied by the MCHP auxiliary boiler is replaced entirely by the heat generated in solar water heaters. Operation of the auxiliary boiler is stopped by a corresponding amount.
- When heat demand is exceeded by the heat generated by solar water heaters, the operating hours of the MCHP generator unit are shortened by an amount that will result in that surplus heat not being generated.
- When the amount of electric power generated is reduced due to the shortened operating hours of the MCHP generator unit, that reduction will be made up with power purchased from the electric utility company.

Figure 17 shows the results from simulation of CO₂ emissions. If the experimental house has 4 m² of solar water heater installed, a CO₂ reduction effect of 10% can be anticipated by comparison without the installation. The operating hours of the MCHP generator unit become shorter

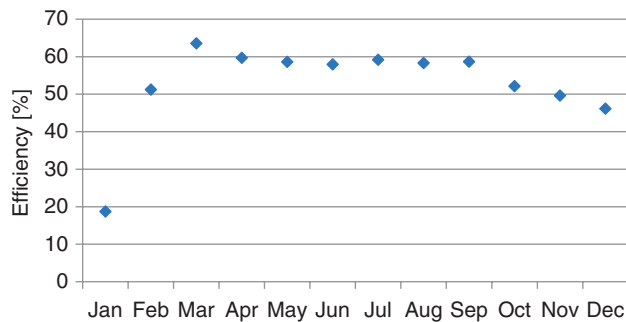


Fig. 16 Measured heat absorption efficiency of solar water heaters

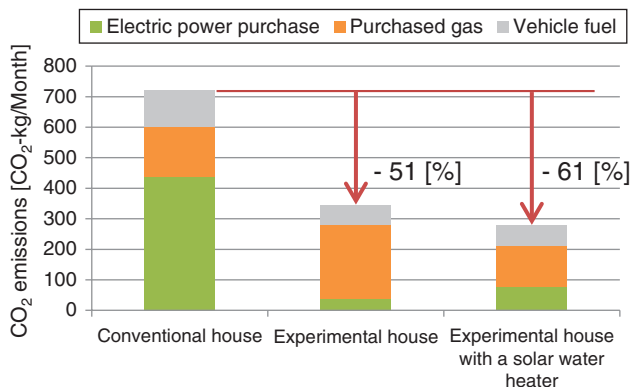


Fig. 17 Simulated CO₂ reduction effect of HSHS

Table 12 Solar water heater

Heat collection area	4 [m ²]
Endothermic efficiency	50 [%]
Amount of insolation	By experimental house data

than when a solar water heater is not installed. Although this causes the amount of electric power purchased to increase, the amount of gas purchased decreases, reducing CO₂ emissions as a whole.

Next, Fig. 18 shows an estimation of the CO₂ reduction effect obtained when the solar water heater area is further increased.

Even when the area of the solar water heater is increased to 32 m², the CO₂ emissions reduction rate remains 71%. This is due to the trade-off between the reduction in CO₂ emissions resulting from the effect of the solar water heater and the increase in CO₂ emissions accompanying the increase in the amount of electric power purchased. It was found, therefore, that CO₂ emissions could not be reduced to zero solely by installing additional solar water heaters.

In order to achieve zero CO₂ emissions from the purchase of gas, it is necessary to satisfy the heat demand within the household without using any fossil fuels. The following methods may be considered as possible means for this:

- Replace gas with hydrogen produced by renewable energy, biofuel, or some other such fuel that can be considered to have zero CO₂ emissions; or
- Replace heat source equipment with heat pumps or other such electrically driven equipment, and operate it using PV-generated electric power.

For method (b), an estimate is made of the PV output required in order to reduce CO₂ emissions to zero in the case of all heat being supplied by heat pumps, and in the case of it being supplied by a solar water heater of 4 m² together with a heat pump as an auxiliary heat source. The efficiency of the heat pump, which was measured separately from the present experiment, is shown by the data in Fig. 19.

The estimation results are shown in Table 13. It was found that a PV output nearly double that of the experimental house is required even when the combination of solar water heater and heat pump is used as the heat

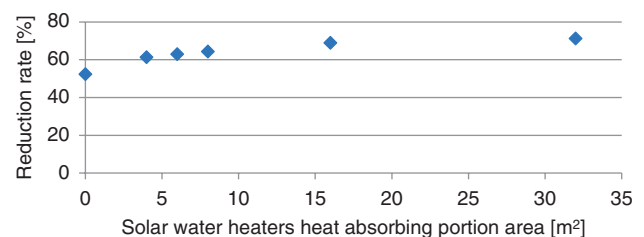


Fig. 18 Simulated CO₂ reduction effect of solar water heaters

Table 13 Estimated PV output required to achieve zero CO₂ emissions with heat pump

Conditions	PV output [kW]
Heat pump	10.1
Heat pump with solar water heater 4 [m ²]	7.1

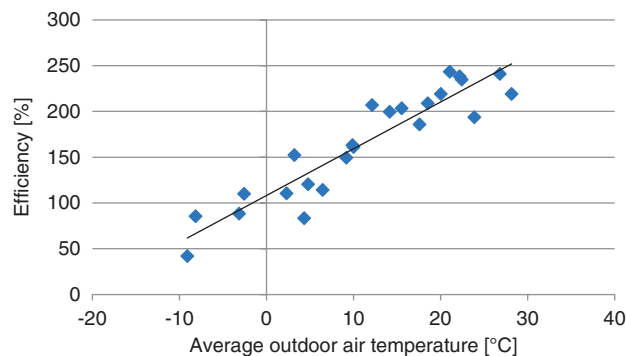


Fig. 19 Measured efficiency of heat pump

source equipment instead of the MCHP. Implementing this would require an installed PV area that is greater than the roof area, indicating that this cannot readily be adopted as a practical solution.

It was found from the above information that when the amount of electric power purchased and the reverse power flow of PV electric power are generally the same, zero CO₂ emissions cannot readily be achieved with the HSHS even when the following changes are made.

- The MCHP is replaced with a heat pump.
- The MCHP is replaced with the combination of solar water heater and heat pump.

In order to achieve zero CO₂ emissions, therefore, it will be necessary to add other approaches for future consideration of method (2) given at the start of Section 5, which was to replace purchased gas with a fuel that is considered to have zero emissions of CO₂.

6. Conclusion

The first purpose and objective were achieved by the determination of the CO₂ emissions reduction effect obtained when using HSHS and the one-half reduction in CO₂ emissions obtained as a result. It was also found that CO₂ emissions from energy used for heating and other such heat source applications will be crucial to the movement toward zero CO₂ emissions.

In the next step of the experiment, a solar water heater is scheduled to be added to the present form of the HSHS in order to verify the CO₂ emissions reduction effect described in Section 5. However, Section 5 also indicated that this addition alone would not be able to achieve zero CO₂ emissions. Therefore, an approach using hydrogen produced by renewable energy, biofuel, or other such fuel that can be

considered to have zero CO₂ emissions has been proposed as a further future step.

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■ Author ■



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Establishment of Prediction Method for Fatigue Durability Applied for Exhaust System in Motorcycle by Coupled Simulation of Vibrating Stress and Thermal Stress

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ABSTRACT

A method applicable to a design of a motorcycle exhaust system in its early design process that predicts fatigue strength was developed. In this prediction method, vibrating stress, thermal stresses, stress resulting from component assembling of the exhaust system and deterioration of strength of materials originated from fatigue by high temperature are simultaneously taken into account. For the prediction of vibrating stresses, flexible multibody dynamics was applied to get accuracy of vibration characteristics modeling for the entire motorcycle structure and for exciting forces delivered from the engine vibrations. Thermal conduction analysis and thermal deformation analysis based on finite element method (FEM) were applied for the prediction of thermal stresses caused in the exhaust system components. For prediction of the thermal stresses, temperature distributions on the surfaces of the exhaust system components are required. To estimate the temperature distributions, heat transfer analysis, which is composed of a one-dimensional (1D) engine cycle simulation and an air flow simulation for the air around the vehicle by 3D-CFD, was implemented. For the validation of the developed method, correlations between the results of simulations and actual measurements were analyzed for the vibrating stress, the temperature distribution, the thermal stress and the fatigue safety factor. High correlations were yielded between them proving the validity of the new method.

In this paper, some of the application examples of newly developed method to an early stage of the development process are also presented. Through the examples, it is shown that the application of the new method effectively help designers implement feasible measures for potentially frail areas against fatigue. As a consequence, the new method enhances the efficiency of the development of the exhaust system.

1. Introduction

Recently, due to advancement of CAE analysis technology and increased computer processing speed, the use of CAE in the development of motorcycles and automobiles is increasing. However, it is practically impossible to obtain necessary analysis accuracy when a structural body is composed of a number of parts or when a part is affected by a number of physical phenomena. One of such a physical phenomenon challenging for analysis by simulation is the fatigue strength of an exhaust system of a motorcycle.

The vibrating stress and the thermal stress occur simultaneously in the exhaust system components of a motorcycle. At the same time, the fatigue limit of material lowers along with an increase of temperature. It is necessary to simultaneously take into account such physical

phenomena for simulation of fatigue strength of exhaust system components.

In the conventional product development process, the fatigue strength of an exhaust system component was evaluated in three stages.

The first stage is the design. In this stage, the fatigue strength was evaluated according to the vibrating stress and the thermal stresses estimated by FEM analysis. However, there were the following three issues. The first issue was that the frequency response analysis was used for estimation of vibrating stress. The frequency response analysis is a linear analysis. Accordingly, an appropriate modeling was not performed for parts having non-linear stiffness characteristics such as a rubber bush or gasket, which were used to support exhaust system components, or the exciting force resulting from engine vibrations. The second issue

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was that an appropriate boundary condition for estimation of thermal stresses was unobtainable. The data representing temperature distribution in the exhaust system are required to estimate thermal stresses. However, there was no technical means to estimate temperature distribution. Accordingly, the temperature distribution data taken from a similar motorcycle were used as the boundary condition. The third issue was that the vibrating stress and the thermal stresses were independently evaluated. For proper evaluation, the thermal stress and the vibrating stress should be superimposed, and evaluated on the fatigue limit diagram.

The second stage is the measurement of performance of a prototype motorcycle on a dynamometer. In this stage, the fatigue strength was evaluated by the stresses those were measured by strain gauges. The strain gauge allows simultaneous measurement of the thermal stress and the vibrating stress. The allowable temperature of a general strain gauge, however, is lower than the maximum temperature of the exhaust system components. So, an accurate measurement at a high-temperature part cannot be conducted without special technique.

The third stage is the durability test using prototype motorcycle. As mentioned above, the evaluation of durability of an exhaust system in the design stage and the stage of performance measurement on the dynamometer was inadequate. As a result, the exhaust system components were often damaged during the durability test. Such damage in the durability test causes not only a delay of product development, but also an increase of man hours for the countermeasure. Accordingly, there had been a strong demand for a method that allows prediction of fatigue strength of an exhaust system in the early stage of product development.

There are a large number of past studies on the fatigue strength of exhaust system components. There are several studies on the simulation of fatigue strength in the design stage⁽¹⁾⁻⁽³⁾. However, such studies concern only on the vibrating stress without taking into account the thermal stress. Meanwhile, there are some studies in which both of the thermal stress and the vibrating stress are considered^{(4),(5)}. However, such studies are focused on evaluation and improvement of fatigue strength in the stage of tests using prototype vehicle, not yet allowing estimation of stress in the design stage. Studies for an accurate prediction of fatigue strength of exhaust system components in the design stage had not been conducted.

In view of the above, the authors conducted studies with a target at development of a method for prediction of the fatigue strength of a motorcycle exhaust system in a design stage at a practical accuracy. The feature of this project is the attempt for prediction of the fatigue strength of an exhaust system by linking the flexible multibody dynamics, 3D-CFD, 1D engine cycle simulation and the structure analysis by FEM. The recently developed simulation method and the results of validation are reported in this paper.

In addition to that, the examples of applications of

the newly developed method in the product development process are introduced to clarify effectiveness of this simulation method.

2. Overview of Fatigue Strength Prediction

2.1 Structure of Motorcycle Exhaust System and How Stresses Occur

The structure of a motorcycle exhaust system and how the stresses occur should be explained before explaining the simulation. The overview of a motorcycle and enlarged pictures of areas where the exhaust components are connected to the vehicle body are shown in Fig. 1.

The exhaust system of a motorcycle is usually composed of exhaust pipes and mufflers. A gasket is provided where the pipe and the muffler are connected for sealing of exhaust gas and insulation of vibration as shown in Fig. 1 (a). The exhaust pipe is attached to the vehicle at the cylinder head and the mount bracket in the main frame. The exhaust pipe is bolted to the cylinder head via flanges. Metal gaskets are

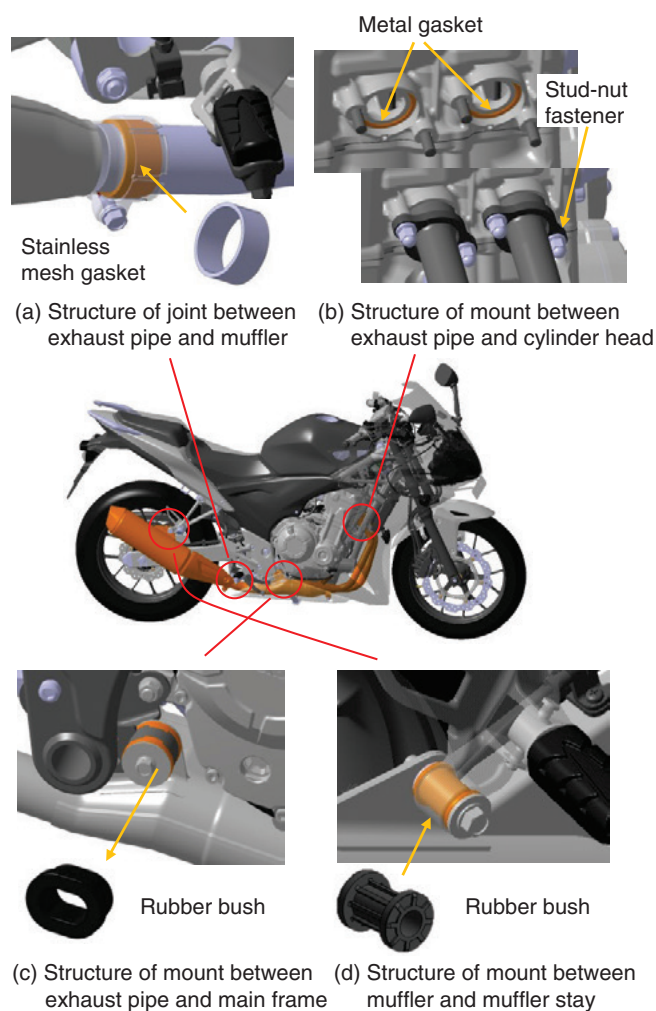


Fig. 1 Connecting structure between vehicle and exhaust system in motorcycle

provided between the exhaust pipe and the cylinder head as shown in Fig. 1 (b). Rubber bushes for mounting are provided between the exhaust pipe and the main frame as shown in Fig. 1 (c). The muffler is attached to the main frame via a mount bracket on a seat rail. Rubber bushes are provided between the muffler and the seat rail as shown in Fig. 1 (d).

The exhaust system expands due to the passing of hot exhaust gas. Under such conditions, when the thermal expansion is firmly restricted due to the stiffness of frame or engine, a high stress occurs in the exhaust system. To control such a thermal stress, the rubber bush is shaped like an elongated ring with the major axis in the direction the exhaust pipe elongates. The rubber bush in the muffler also has a construction that prevents excessive constraint. Consequently, the stiffness of rubber bush is designed non-linear and differs depending on the direction. Due to the non-linearity, the stiffness in the connecting part changes depending on the thermal deformation and the displacement from the vibrations. Consequently the natural frequency of exhaust system also changes due to the influence of thermal deformation and vibrations.

In automobiles, an insulating device such as a spherical joint or a metal bellows is provided in the part of an exhaust pipe close to the engine to prevent transmission of engine vibrations to the exhaust system. In motorcycles, however, such an insulating device is not provided because it affects the appearance, space layout, and kinetic performance of the vehicle. As a result, stresses resulting from engine vibrations tend to occur in the exhaust system of a motorcycle. Moreover, the natural frequency of an exhaust system when installed in the vehicle often falls somewhere between 50 Hz and 300 Hz, which overlaps with the frequency band of the vibrations occurring from reciprocating system in an engine. Consequently resonating vibrations often occur in the exhaust system, causing a significant lowering of the fatigue strength.

In addition to that, three kinds of thermal stress occur in the exhaust system and lower fatigue strength. One is the stress that occurs from constraint for thermal expansion as mentioned earlier. The second is the stress occurring from temperature distribution in the exhaust system. The third is the stress occurring from the difference of a linear expansion of materials. In the exhaust system components of a motorcycle, stainless steels having various linear expansion coefficient are often welded together to achieve both goals i.e., the production cost and the appearance. Such a construction causes a high thermal stress when hot, and reduces fatigue strength.

The last issue is that the fatigue limit of the material lowers along with an increase of temperature. Especially in the part of the exhaust pipe close to the engine or near the catalyst, the surface temperature becomes higher than 500 degree C. At such a high temperature, the fatigue limit of the material lowers to a half compared to that at a room temperature.

2.2. Requirements in Prediction Method

When predicting the fatigue strength of exhaust system components, the structural features and how the stresses occur mentioned in the preceding paragraph should be considered. The authors considered the following requirements for the simulation.

- (1) Simulation should be able to represent the vibrating stress including under condition in resonance.
 - (1)-a Dynamic characteristics of major components in a motorcycle including those in the exhaust system should be represented.
 - (1)-b An appropriate damping for each component part is set, and the vibration amplitude when resonating should be correctly represented.
 - (1)-c An exciting force in accordance with operating conditions of an engine should be considered.
 - (1)-d Changes of connection stiffness that depends on the thermal deformation and the displacement from vibrations should be considered.
- (2) Simulation should be able to represent following three kinds of thermal stresses.
 - (2)-a Stress occurring from constraint for thermal expansion
 - (2)-b Stress occurring from temperature distribution
 - (2)-c Stress occurring from difference of linear expansion of materials
- (3) The stress occurring from fastening with bolts for assembling the exhaust system should be considered.
- (4) Fatigue safety factors should be calculated simultaneously taking into account the above stresses 1-3.
- (5) At that time, the lowering of the fatigue limit when the material becomes hot should be considered.

2.3. Outlines of Fatigue Strength Prediction Method

As a solution for the issues, the authors have developed the simulation method in which the flexible multibody dynamics, the 3D-CFD, the 1D engine cycle simulation, and the structural analysis are linked. The relationship among the factors affecting the fatigue strength of exhaust system components and the domain each analysis method covers are shown in Fig. 2.

2.4. Test Motorcycle

The sport-type motorcycle equipped with a 399 cm³ parallel twin cylinder engine was used for the development of simulation. The test vehicle and the engine that is equipped in the vehicle are shown in Fig. 3 and its major specifications are shown in Table 1. The engine has a balance shaft to reduce coupling force occurring from the crankshaft having crankpins 180 degree apart. Accordingly, first-order coupling vibrations generated by the rotation of the crankshaft are reduced. The second-order vibrations, however, occur from rotating of the crankshaft. In motorcycles having an engine of this configuration, resonating vibrations tend to occur in the exhaust system due to the second-order vibrations.

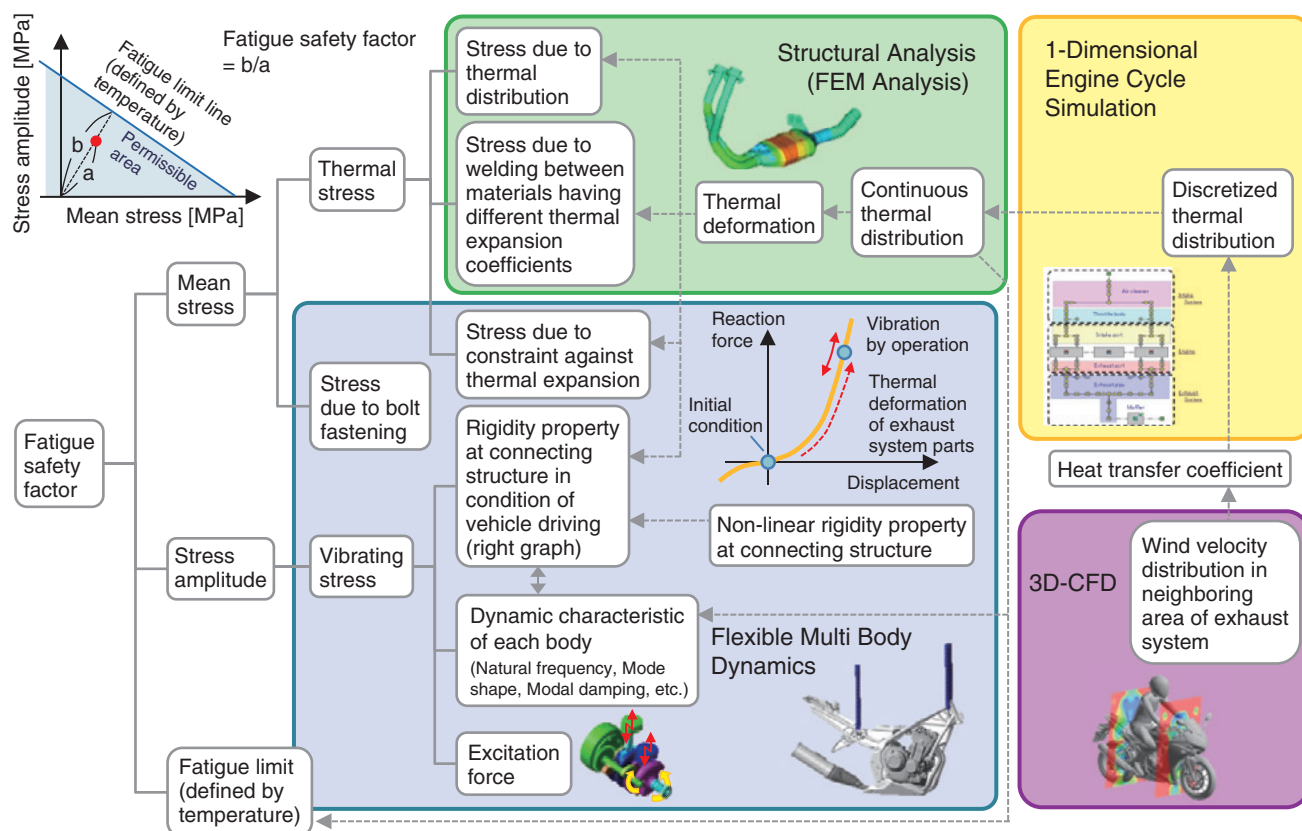


Fig. 2 Relationships among involving factors in fatigue safety factor and regions assigned by four simulation technology

Table 1 Specifications of test motorcycle used in verification tests

Specifications	
Engine type	4 stroke, Liquid-cooled, Parallel twin-cylinder
Displacement	399 cm ³
Bore and Stroke	67.0 mm × 56.6 mm
Crank shaft type	180 deg crankshaft
Balance shaft type	1st harmonic, 1 shaft
Max. brake power	34 kW at 9500 rpm
Max. brake torque	37 Nm at 7500 rpm
Frame type	Diamond
Engine mount type	Rigid mount



Fig. 3 Test vehicle and equipped engine

3. Vibrating Stress Prediction Method

3.1. Method Selection

In the field of industry, the frequency response analysis is generally applied for prediction of vibrating stress. In the meantime, as a dynamic analysis method for complex and large-scale vibration systems, the flexible multibody dynamics⁽⁶⁾ (FMBD) is recently used^{(7), (8)}.

The comparison of analysis capabilities of the frequency response analysis and the FMBD with respect to the function necessary for prediction of vibrating stress in the exhaust system is shown in Fig. 4. The frequency response analysis is a linear analysis. It is therefore almost impossible to accurately simulate engine vibrations or non-linear stiffness characteristics in the connecting part between exhaust system components. Meanwhile in the FMBD, the equation of motion and the geometric constraint equation between the components are coupled, and motions and vibration of the components are calculated by the time-history numerical integration. Thereby, dynamics in a reciprocating system and non-linear stiffness characteristics in the connecting part can be directly described in those equations, and correctly solved.

Due to above-mentioned reasons, the authors decided to employ the time history response analysis base on the FMBD.

3.2. Clarification of Parts that Affect Stresses Occurring in Exhaust System Components

It is considered that the stress of exhaust system components should be affected by a large number of components of a motorcycle. From the standpoint of analysis efficiency, however, it is preferable to have the analysis model composed of less number of parts. The authors therefore conducted experiments to clarify parts that are necessary for estimation of stress in the exhaust system components. Parts are removed one after another from the complete motorcycle. At the same time, changes of the acceleration response and the vibrating stress at various points of the exhaust system are measured. Fig. 5 is the picture showing a scene in the experiment. The motorcycle without the front and rear wheels and suspension systems is suspended by soft springs from the ceiling. Thus operating the engine under various conditions, the parts having a significant influence to the stress in the exhaust system components were clarified.

In the experiment, the stress in the exhaust system components was virtually unaffected by the front and rear suspension systems. The resonance frequency of a suspension system is generally around several Hz, which is adequately lower than the resonance frequency of an

exhaust system. In such a case, the two vibration systems can be deemed as independent from each other. The authors therefore concluded that it was not necessary to consider the suspension system in the analysis model. Similarly concluded that it was not necessary to take into account the fuel tank, throttle body, etc.

The authors finally concluded that it was efficient to have the vibrating stress simulation model composed of five parts, which are; the engine, the main frame, the seat rail including the muffler mount bracket, the muffler, and the exhaust pipe.

3.3. Modeling of Dynamic Characteristics of Each Component of Analysis Model

For accurate simulation of the dynamic behavior of an exhaust system, it is necessary to have the natural frequency, the vibration mode shape and the modal damping of each component in the analysis model accurately simulated. In this project, the mode synthesis that allows reduction of calculation time while maintaining accuracy is employed. So, each component is modeled as a dynamic super element that is made from FEM model.

In the FMBD method using super elements, the displacement of the center of gravity point of the elastic body and the modal displacement are calculated. The stress is not directly calculated. Therefore, the stress in the exhaust system components is calculated from the modal displacement and the mode shape.

Meanwhile in the temperature zone where the exhaust system components are in operation, a change of Young's modulus, which is dependent on the temperature, cannot be ignored. Young's modulus in the FEM model was set in accordance with the temperature of each component.

3.4. Modeling of Stiffness Characteristics of Connecting Element

The stiffness characteristics of the connecting element were measured prior to the modeling. As the connecting

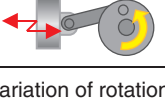
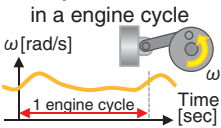
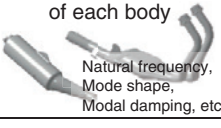
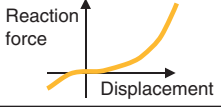
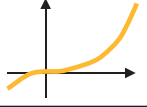
Needed function for simulation		Analysis method	Frequency response analysis	FMBD
Excitation force	Excitation force by unbalanced reciprocating system 	Applicable	Superposition of sinusoidal wave $F = \sum a \sin \omega t$	Applicable
	Variation of rotational velocity of a crankshaft in an engine cycle 	Not applicable		Direct modeling 
Dynamic characteristic	Dynamic property of each body  Natural frequency, Mode shape, Modal damping, etc.	Applicable	Applicable	Applicable
	Non-linear rigidity property at connecting structure 	Not applicable Linearization is needed	Applicable	Direct modeling 

Fig. 4 Comparison between frequency response analysis and flexible multibody dynamics regarding demonstration capabilities required for dynamic response representation of exhaust system

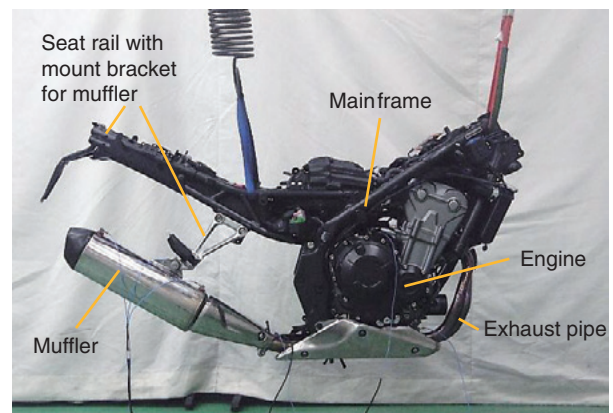


Fig. 5 Photographic view of experimental setting to identify the parts influential to the vibrational stresses on exhaust system in motorcycle

part is always vibrating, the dynamic stiffness was measured rather than the static stiffness.

As an example, the test equipment for measurement of stiffness characteristics of the muffler gasket is shown in Fig. 6. The test equipment is composed of a body and two weights. The body is fixed on the shaking table, and muffler gaskets are installed between the body and the weights. The construction of the areas where the gaskets are installed is same as that of the area where the actual exhaust pipe and the muffler are connected. In the experiment, the shaking table was sweep-accelerated at constant displacement amplitude and the acceleration response of the weights was measured. The input amplitude was altered in six levels. The measured acceleration of a weight is shown in Fig. 7. The input frequency is on the horizontal axis, and the acceleration response of a weight is on the vertical axis. Six lines of various inputs are drawn in the graph. The lowering of resonance frequency of the system along with an increase of amplitude is seen in the graph. The data indicates that the stiffness characteristics of the muffler gasket are non-linear.

In this project, the modeling of stiffness characteristics of the connecting element was conducted using springs and dampers having non-linear characteristics. The spring constant and the damping coefficient were determined using the FMBD model in which the test equipment was simulated. The analysis model is shown in Fig. 8. The body

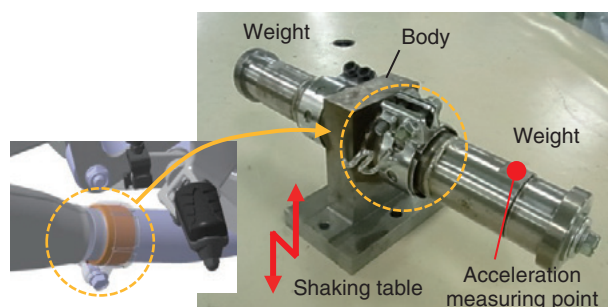


Fig. 6 Measurement of dynamic rigidity of muffler gasket

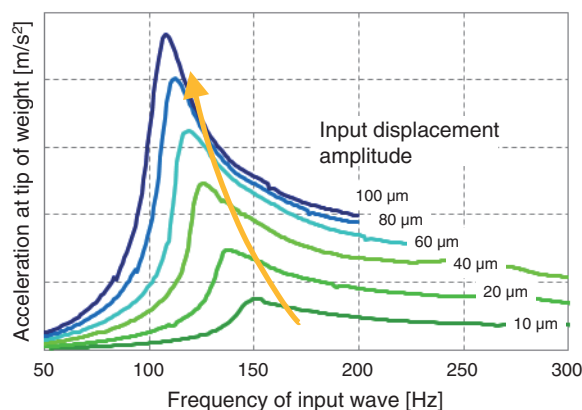


Fig. 7 Response acceleration of tip of weight

and the weights are modeled as super elements. The springs and the dampers are installed between the body and the weights. Using the model, the characteristics of the springs and the dampers were determined in such a manner that the response of weights reproduced the test results.

The acceleration response waveforms of a weight in the analysis model after setting the characteristics are shown in Fig. 9. The use of this model allows simulation of changes of stiffness characteristics in the connecting part that depend on the displacement from vibrations and the thermal deformation. The changes of resonance frequency, which depends on the amplitude of vibration, can also be simulated.

3.5. Excitation Force of Engine

In this project, the reciprocating system is represented by multibody model consists of pistons, piston pins, connecting rods and a crankshaft. Moreover in this reciprocating system model, the changes of pressure in the cylinder within a cycle of reciprocating mechanism are taken into account. In doing so, the motions of the reciprocating mechanism are faithfully simulated, allowing accurate modeling of excitation force of the engine.

3.6. Vibrating Stress Analysis Model

The overview of the vibrating stress analysis model

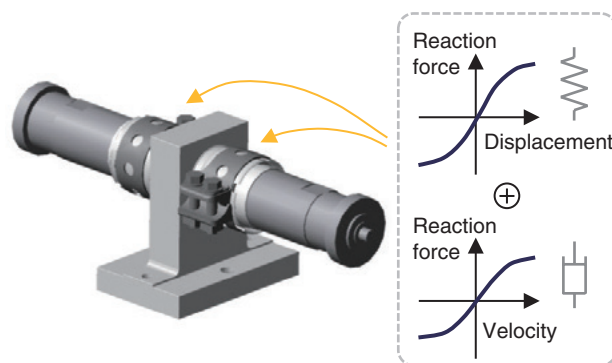


Fig. 8 Flexible multibody model used for setting dynamic rigidity property of muffler gasket

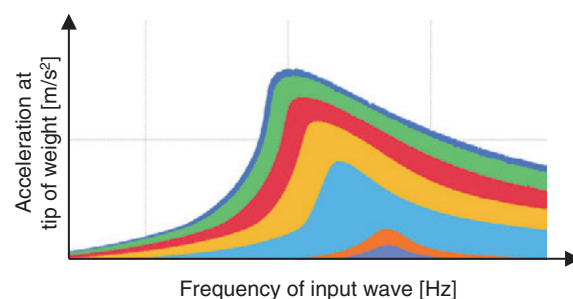


Fig. 9 Response in acceleration measured at tip of weight calculated by flexible multibody model analysis

is shown in Fig. 10. The components of the model are the engine, the main frame, the seat rail including the muffler mount bracket, the muffler, and the exhaust pipe. Those components are connected via non-linear springs and dampers to represent stiffness characteristics of the connecting element. At that time, the bolt fastening force in the position where the exhaust pipe is attached to the cylinder head is also taken into account. A multibody model that represents reciprocating system is incorporated into the crankcase for modeling of the excitation force. The main frame is mounted on the ground via the flexible spring.

The analysis takes approximately 8 hours with the engine sweep-operated from 1000 rpm up to 10000 rpm. This calculation time is allowable when applying it in the design stage.

3.7. Correlation of Entire Behavior of Exhaust System

In the analysis model, it is necessary to allow simulation of not only regional stress conditions but also vibration mode shape in the entire exhaust system. For validation of the analysis model, the correlation was analyzed in the behavior of entire exhaust system when the engine was operating.

The behavior of entire exhaust system was measured with four triaxial accelerometers. The accelerometer locations are shown in Fig. 11. The acceleration was measured while sweep-operating the engine from 1000 rpm up to 10000 rpm.

This motorcycle is equipped with a balance shaft to reduce vibrations of the primary component of crankshaft rotating frequency. Hence the acceleration response of the exhaust system at first order component is small. For the correlation validation, the acceleration response data of the secondary component of crankshaft rotating frequency was used. The acceleration response waveforms acquired from the experiment and the simulation are shown in Fig. 12. In the matrix, the horizontal direction indicates measurement points, and the vertical direction shows the direction of

acceleration. A graph is composed of two parts. The lower graph shows acceleration and the upper graph indicates phase of the vibration. The engine speed is on the horizontal axis. The standard of the phase is the response waveform measured by an accelerometer, which is attached to the engine, in the Z direction. The red lines show measurement and the blue lines indicate the simulation by the analysis model.

It can be understood from viewing of the entire data that the vibrations of the exhaust system are occurring in three directions. The simulation is in a good agreement with the measurement both in the acceleration amplitude and the phase. That indicates that the simulation is in a good agreement with the vibration mode of actual exhaust system. The simulation is also in a good agreement regarding the resonance in the proximity of 4000 rpm and 8000 rpm. The error of prediction of resonance frequency in this method is within $\pm 10\%$. The prediction error of peak acceleration is within $\pm 20\%$.

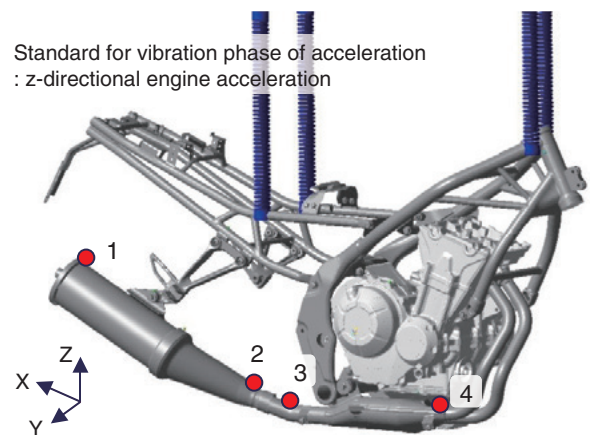


Fig. 11 Installation positions of tri-directional accelerometers

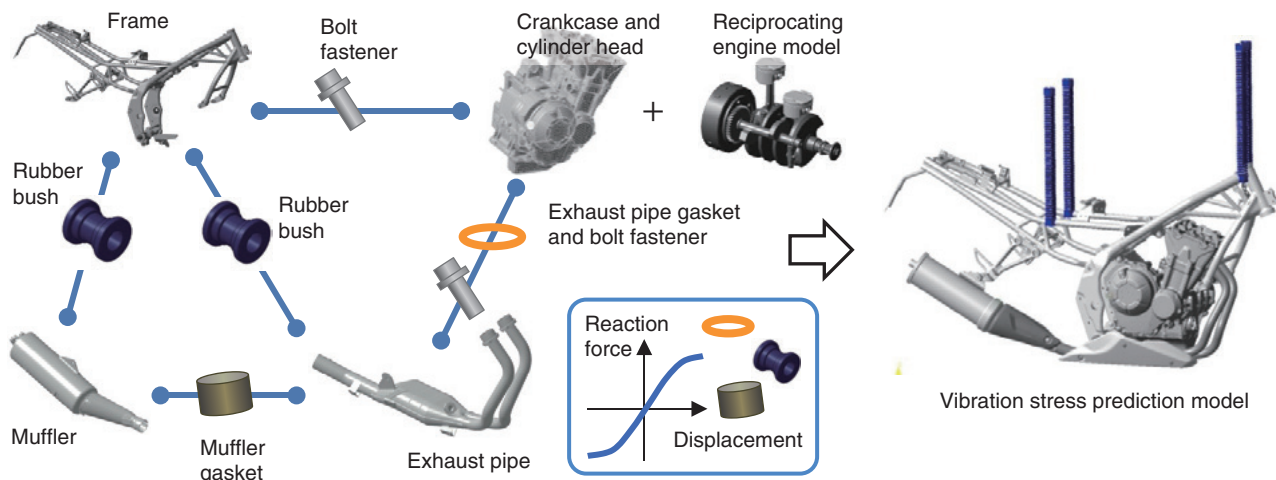


Fig. 10 Flexible multibody model to simulate dynamic responses and vibration stresses of exhaust system of motorcycle

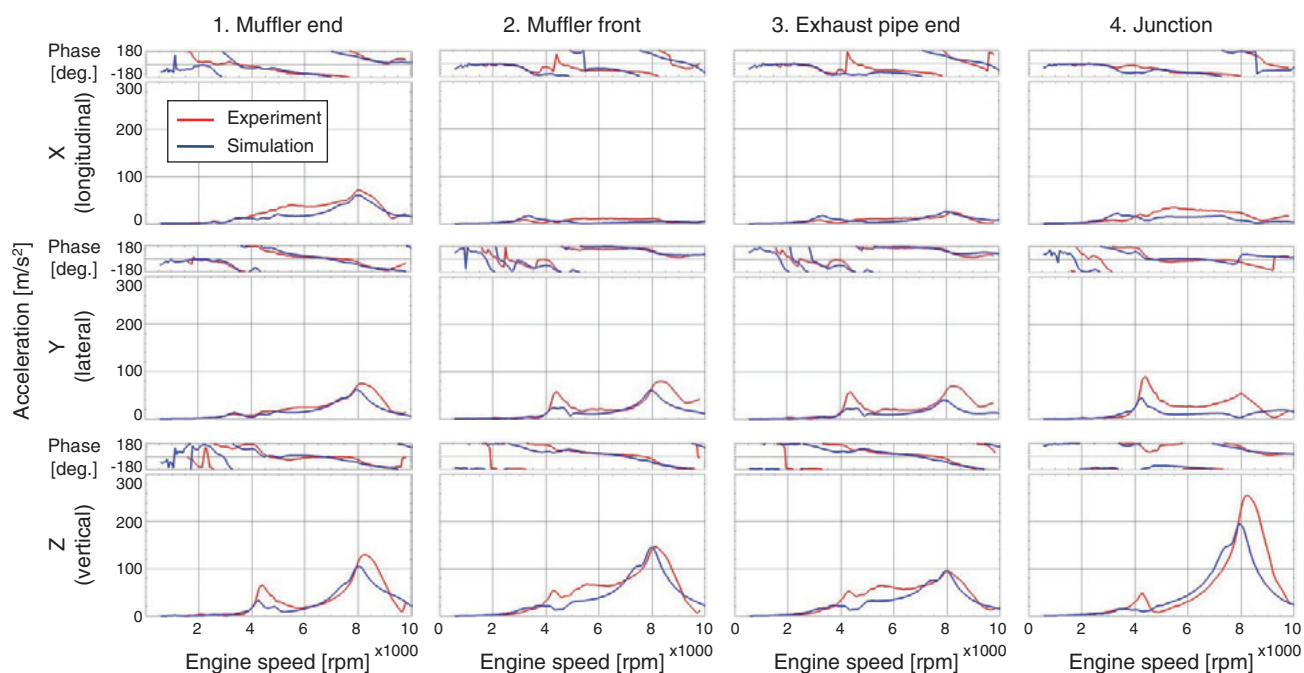


Fig. 12 Correlation of acceleration responses between experimental and calculation results measured at various locations in exhaust system

3.8. Correlation of Vibrating Stress

The correlation of vibrating stress was analyzed. The stress was measured at the bended radius section of the muffler mount bracket. A strain gauge was used for the stress measurement. The strain gauge location is indicated in Fig. 13. For evaluation of the fatigue strength of the exhaust system, an overall stress should be used. Accordingly, the correlation was validated using overall strain amplitude.

The strain data acquired from the experiment and the simulation are shown in Fig. 14. In the graph, the horizontal axis represents engine speed and the vertical axis shows the strain amplitude. The red line is measurement and the blue line is the simulation. In the experiment, the strain amplitude increases along with an increase of engine speed, and peaks at 9500 rpm. After that, along with an increase of engine speed, the strain amplitude decreases. The simulated strain amplitude is in a good agreement with the measurement in the resonance frequency and the quantity at each engine speed.

From the above findings, a good correlation between the simulation and measurement is seen in the behavior of entire exhaust system and the regional stress. The authors therefore conclude that the vibrating stress prediction method proposed in this project is appropriate.

4. Prediction of Thermal Stress

4.1. Prediction of Temperature Distribution in the Exhaust System

To predict thermal stress of an exhaust component, it is

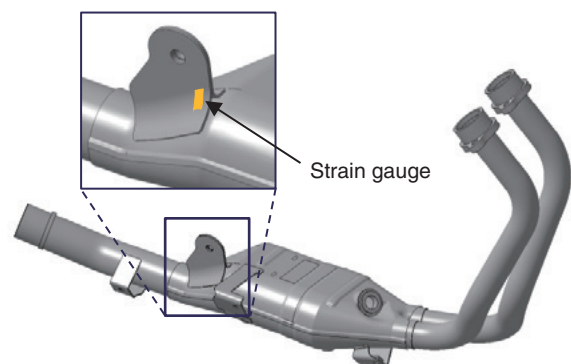


Fig. 13 Measurement point of stress amplitude using strain gauge

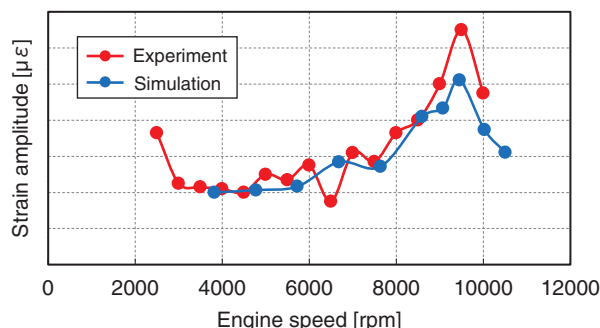


Fig. 14 Correlation between experimental and simulation results of strain amplitudes measured at exhaust pipe stay

necessary to estimate temperature distribution in the exhaust system. There are the following two methods for that. One is by the 3D CFD and the other is by the 1D engine cycle simulation⁽⁹⁾. Among those, the direct estimation of temperatures in the exhaust system by the 3D CFD requires extensive calculations, takes a very long time, and is not practical to apply in the early stage of design. Accordingly, the 1D engine cycle simulation was chosen in this project.

The temperature of an external surface of exhaust system depends on the balance of the heat input from the exhaust gas that passes inside, the heat conduction inside of the exhaust system component, and the heat release from the external surface to the ambience. In the 1D engine cycle simulation, the Colburn equation⁽¹⁰⁾ is used for the calculation of heat transfer from the exhaust gas to the internal surface of exhaust pipe. As the heat transfer from the exhaust gas to the internal surface of exhaust pipe is less affected by external factors, a highly accurate simulation is possible by that equation. In the meantime, the heat transfer from the external surface to the ambience is affected by the flow of air around the motorcycle. Accordingly, when developing a motorcycle having a very different shape from the conventional model, the existing data of heat transfer coefficient on the external surface was inapplicable and it was making an accurate simulation of temperature distribution unattainable by the 1D method.

In this project, the flow rate of air stream on the external surface of exhaust system is calculated by the 3D CFD, and converted to the heat transfer coefficient by the empirical formula⁽¹¹⁾. The calculation conditions of the 3D CFD used for the simulation of air flow rate distribution on the external surface of the exhaust system of the test motorcycle is shown in Table 2, and the calculated data in Fig. 15.

The overview of the 1D engine cycle simulation model⁽⁹⁾ is shown in Fig. 16. The distribution of heat transfer coefficient calculated by the 3D CFD is “mapped” on the external surface of the exhaust components in the model. The temperature distribution on the surface of exhaust component is simulated taking into account the heat transmission from the exhaust gas to the internal surface, the heat conduction within the component, and the heat transfer from the external surface to the atmosphere in the gas cycle simulation.

4.2. Correlation between Exhaust Gas Temperature and Surface Temperature of Exhaust System Component

The temperature distribution estimation method was validated. The exhaust gas temperature and the outer surface temperature of exhaust system components were measured on the chassis dynamometer and compared to the calculated result. The shift position was set at the 6th speed and the engine was fully loaded. The wind velocity in the chassis dynamometer room was controlled to be equal to the vehicle speed. The temperatures of exhaust system components and the exhaust gas were measured with thermocouples. The

Table 2 Conditions applied for 3D-CFD analysis

Solution type	Steady state
Working fluid	Air
Compressibility	Incompressible
Turbulent model	Realizable k- ϵ
Cell type	Hybrid (hex, tetra, pyramid)
Cell size	2-8 mm
Total number of cells	25,000,000
Vehicle speed	160 km/h

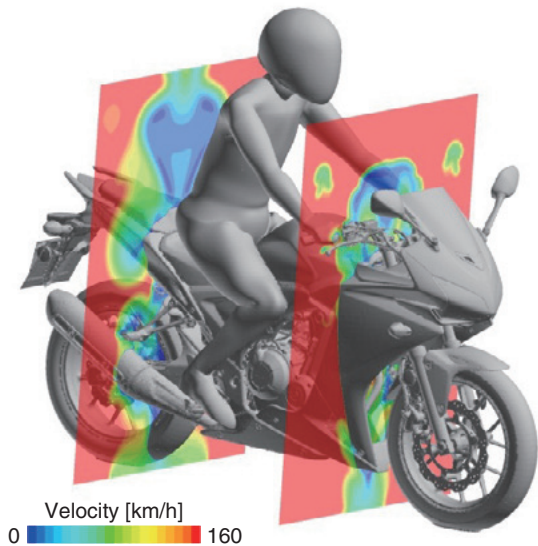


Fig. 15 Calculation results of wind velocity around the vehicle

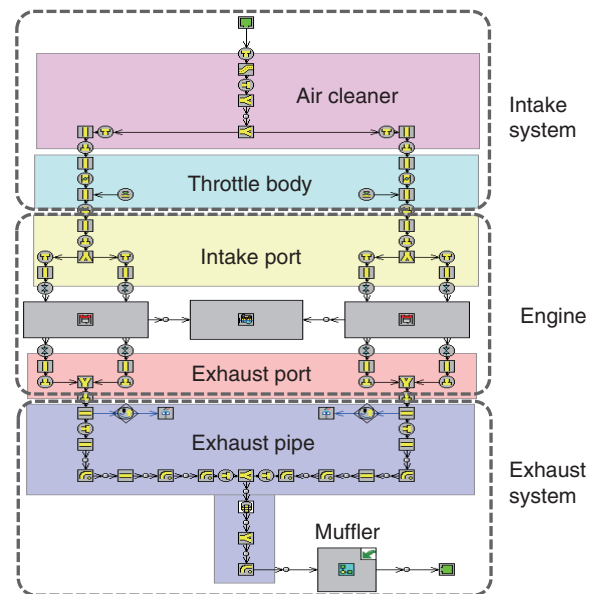


Fig. 16 Overview of one-dimensional simulation model for gas flow analysis

pictures of the motorcycle when measuring temperature and the locations of thermocouples are shown in Fig. 17.

The exhaust gas temperature and the external surface temperature when the engine speed is 9000 rpm are shown in Fig. 18. In the graphs, the positions of the measurement are shown on the horizontal axis and the vertical axis refers to the temperature. The red lines show experimental results and the blue lines show simulation results.

Though a small variance is seen between the measurement and the simulation results, the authors consider that the proposed temperature estimation method is acceptable for prediction of fatigue strength of exhaust system components.

4.3. Thermal Stress Estimation

In this project, the thermal stress occurring from the temperature distribution and the thermal stress occurring from the difference of linear expansion coefficient of materials were estimated by the heat conduction analysis and the stress analysis using the FEM model. The temperature distribution estimated by the 1D simulation was used as the boundary condition in the FEM analysis.

The estimated thermal stress is shown in Fig. 19. In the picture, high stresses are shown in the pipe-to-manifold welding and the catalyst-to-casing spot welding. As such areas tend to be damaged during durability tests, sufficient considerations should be given in the design stage.

The stress occurring from the constraint for thermal expansion of the exhaust component is calculated in the vibrating stress simulation model. By this method, the

thermal stress is calculated as mean stress component in the vibrating stress simulation.

4.4. Correlation of Thermal Stress

The part in the exhaust system that was most vulnerable to breaking in the durability test was the welding bead toe part. An accurately simulation of stress distribution around

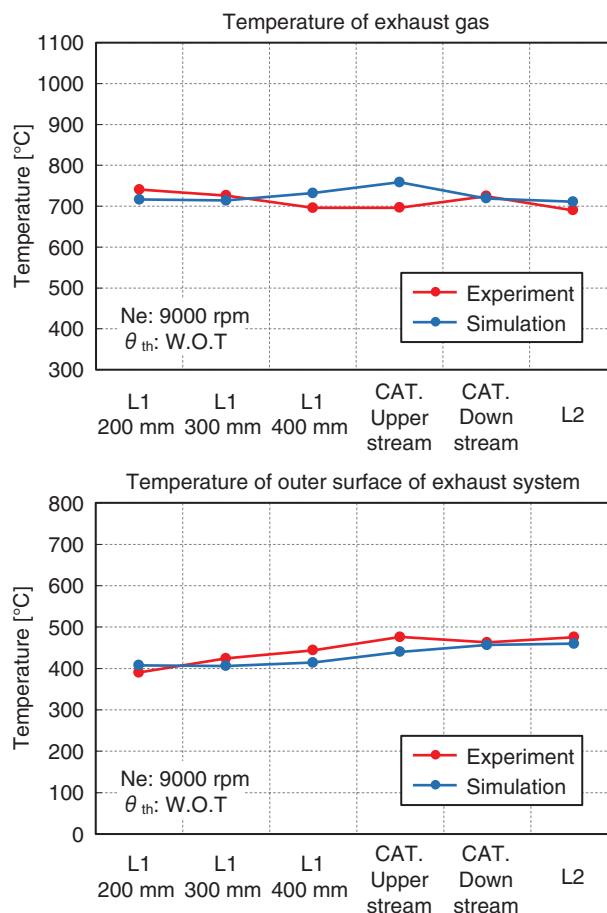


Fig. 18 Correlation of temperature distributions between experiment and simulation on exhaust gas and outer surface of the exhaust system

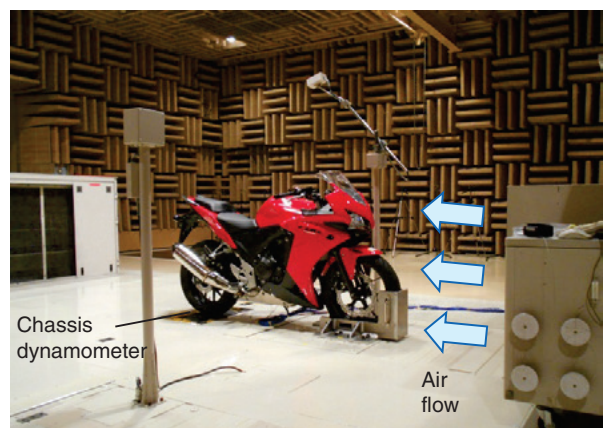


Fig. 17 Measurement of thermal distributions of exhaust gas and outer surface of the exhaust system

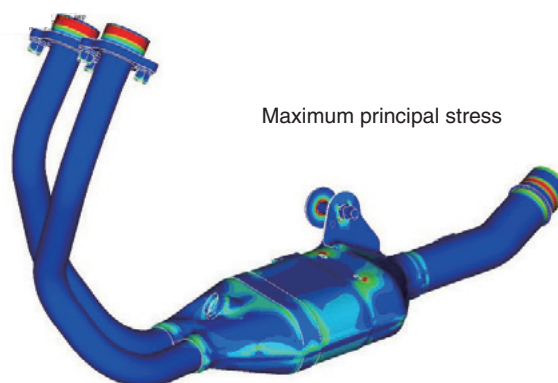


Fig. 19 Results of thermal stress analysis

that part is important when evaluating stress in the exhaust system. With respect to the thermal stress in the proximity of welding of stainless steel materials having different linear expansion, which is the characteristics unique in motorcycles, the correlation between the experiment and the simulation was analyzed.

The correlation was analyzed using a test piece. The test equipment and the test piece are shown in Fig. 20. The two stainless pipes having a different linear expansion coefficient are connected by MIG welding. The test piece was heated in the oven and the strain distribution in the proximity of welding beads was measured. The strain distribution was measured by the strain gauges laid lengthwise on the surface of the pipe from the welding bead toe part. The oven was set at a temperature 153 degree C above the room temperature.

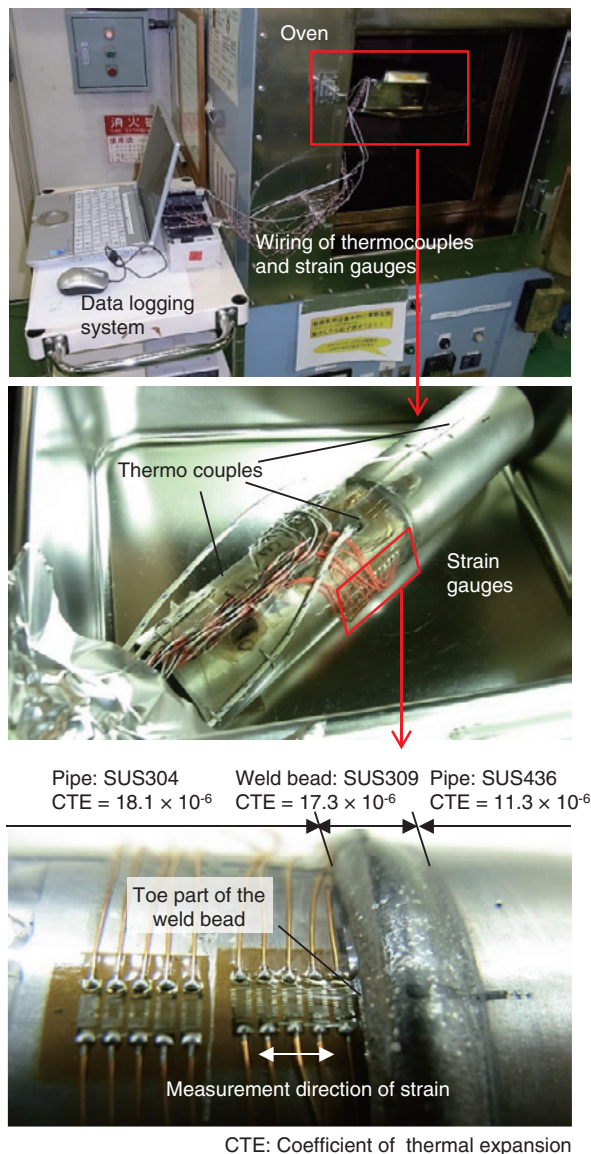


Fig. 20 Measurement of strains generated by heat deformation adjacent to the weld bead

The FEM model of the test piece and the analysis result for the thermal stress are shown in Fig. 21. The model is composed of shell meshes. The size and the pattern of the mesh were selected considering balance between the calculation speed and the analysis accuracy. The measured and simulated strain data are shown in Fig. 22. In the experiment, the strain is highest near the welding bead toe part, and gradually decreases as the length from the toe part increases. A good correlation is seen between the measurement and the simulation.

From the above findings, the authors have concluded that the method we propose in this project for estimation of the temperature distribution and the thermal stresses is appropriate.

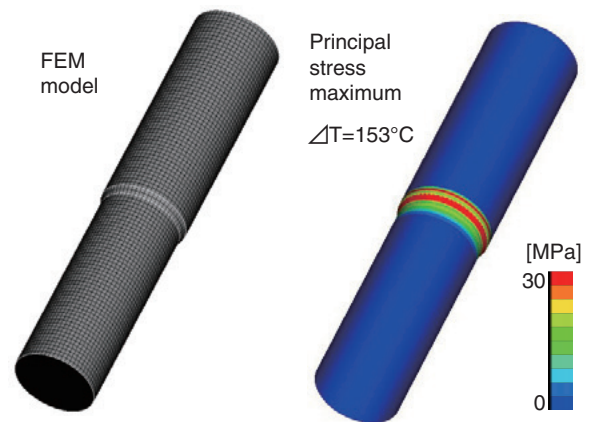


Fig. 21 Simulation result of FEM model of a test piece to investigate thermal stress distribution adjacent to weld bead

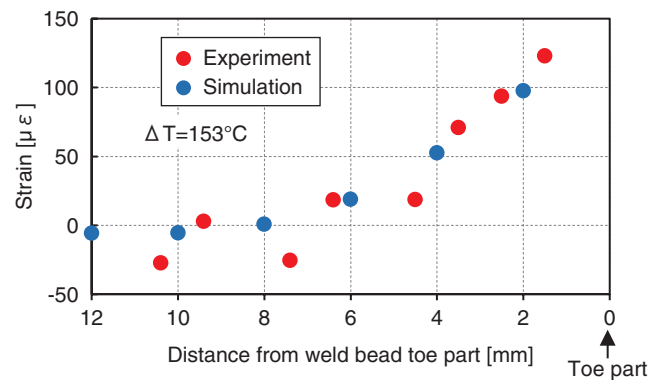


Fig. 22 Correlation of strain generated adjacent to weld bead by heat deformation

5. Calculation of Fatigue Safety Factor

5.1. Calculation Method

The procedure for calculation of the fatigue safety factor is shown in Fig. 23. From the analyses conducted so far, the time-history vibrating stress and the static thermal stress

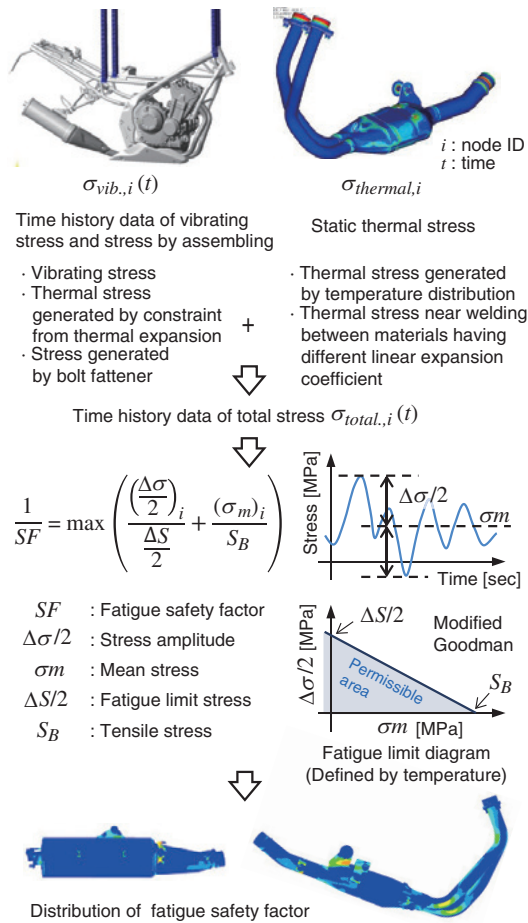


Fig. 23 Process of calculating fatigue safety factor

have been acquired. Firstly the time-history total stress is determined by superimposing the two stresses.

Then, using the total stress, mean stress and stress amplitude are calculated. Lastly the fatigue safety factor is calculated by plotting the mean stress and the stress amplitude on the fatigue limit diagram. In the calculation of the fatigue safety factor, the temperature dependency of the fatigue limit of materials and the creep limit are taken into account.

5.2. Correlation in Fatigue Safety Factor

The correlation in the fatigue safety factor was analyzed. The strain in the exhaust pipe mount bracket of the 399 cm³ test motorcycle was measured while sweeping the engine speed from idling up to 10000 rpm. The location of the strain measurement was the same as that in the vibrating stress analysis. Then, the mean stress and the stress amplitude were calculated and the acquired data were plotted on the fatigue limit diagram. The fatigue limit diagram on which the stresses are plotted is shown in Fig. 24. In the diagram, the horizontal direction indicates mean stress, and the vertical direction indicates stress amplitude. Also, to check for general applicability of the simulation method, the fatigue safety factor was measured and simulated in three other prototype motorcycles having a different engine displacement in addition to the 399 cm³ motorcycle. The data are shown in Fig. 24. In the diagrams, the red dots are the measurement and the blue dots are the simulation.

The estimated fatigue safety factor at a number of positions in a number of models shows a good agreement with the measured one. As shown in Fig. 2, the fatigue safety factor was calculated through many steps. Although

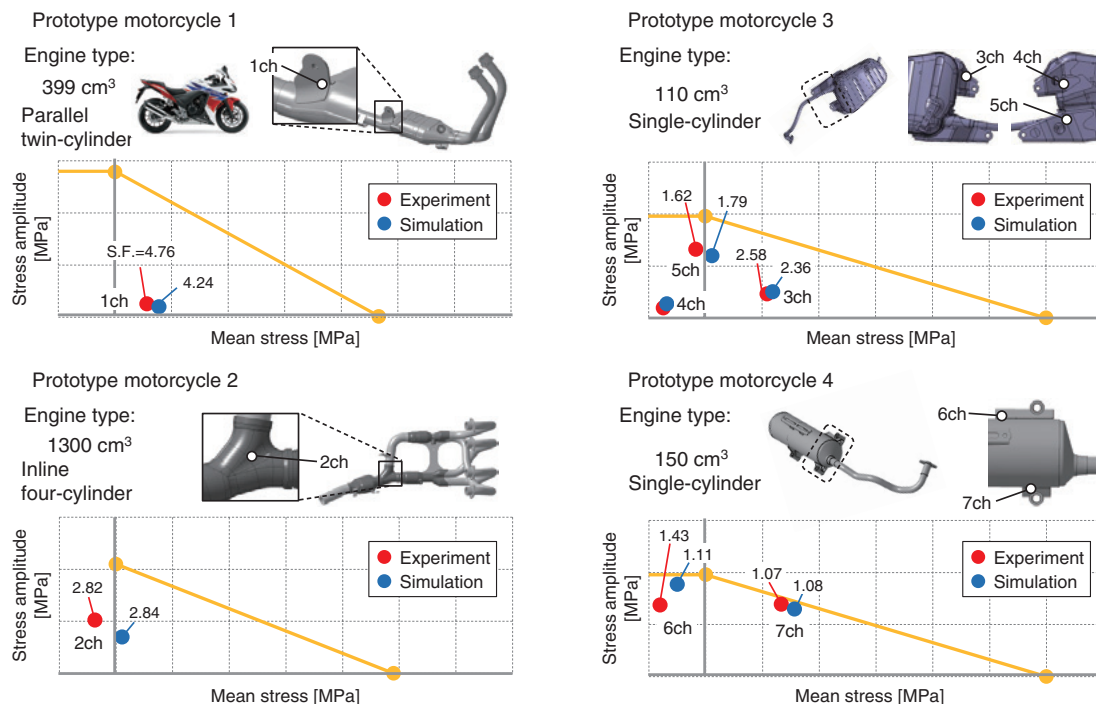


Fig. 24 Correlation between experimental and simulation results on fatigue safety factors

the calculated safety factor therefore contains cumulative errors, the error of estimated fatigue safety factor in this project is less than $\pm 30\%$.

From the abovementioned findings, the authors have concluded that the method for prediction of the fatigue safety factor including the estimation of vibrating stress, temperature distribution and thermal stress is appropriate and adequately applicable to motorcycles in general.

6. Example of Application to Product Development

This chapter introduces an example of application of the developed simulation method to a product development. The subject vehicle is frame-mount-engine type motorcycle equipped with a parallel-twin-cylinder engine.

In its early design stage, the fatigue safety factors were estimated for the components of the on-screen prototype of the exhaust system. Through the results of the investigation using the simulation, it was found out that some of the fatigue safety factors represented nonconforming values at the mount bracket of the exhaust pipe and at the mounting stay for the exhaust pipe covering. The contour representation of the distribution of fatigue safety factors for the exhaust pipe is shown in Fig. 25. Suppose that the prototype vehicle were produced based on this initial design and the durability tests were conducted, it is presumed that the breakages would occur at the sections mentioned above. Accordingly, the design changes were applied to fix these potential flaws. Iterative approach of repeating design changes followed by evaluations of fatigue safety factors for revised designs by developed simulation tool was applied. This iteration was performed until the fatigue safety factors for concerned portions exceed the required values.

Two examples of applied countermeasures are shown in the following. In Fig. 26, the shapes of mounting bracket for the original design and for the design after the changes are applied, together with the contour representations of the fatigue safety factors, are shown. In the initially designed configuration plan, the fatigue safety factor did not meet the

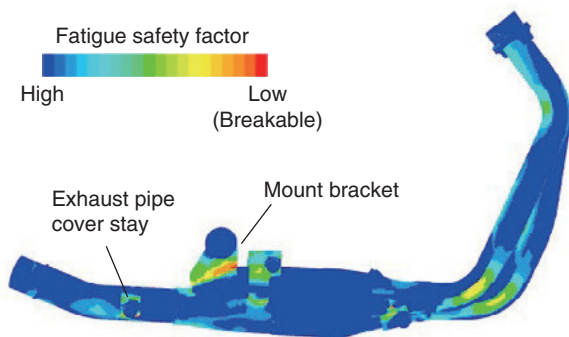


Fig. 25 Contour map of distribution of durability safety factor on the surface of the exhaust pipe

requirement at the curved area. It was because combined effects of bending stress brought by vibrations of the exhaust pipe and the stress arose from the thermal deformations. To cope with the potential design defect occurring in this area, the radius of the curvature was enlarged to mitigate the concentration of the stress. Thus, the fatigue safety factor of this curved area had become satisfactory to meet the requirement. Introducing the next example, the shapes of mounting stay of the exhaust pipe covering for the original design and for the design after the changes, together with the contour representations of the fatigue safety factors, are shown in Fig. 27. In the initially designed configuration plan, the fatigue safety factor did not meet the requirement at the curved area. It was because combined effects of vibrational stress generated in the system, composed of the exhaust pipe covering and its stay, and the stress arising from the difference of thermal expansion between the exhaust pipe and its covering. In addition, the lengths of two legs of the stay were different. Relatively speaking, a stay with shorter length has a structurally higher rigidity and bears higher load and is disadvantageous in terms of durability strength. Moreover, a smaller curvature tends to induce a concentration of stresses. To cope with the potential design defect in this area, the stays are redesigned to have the same leg lengths and the curvatures of the stay legs were enlarged. In addition, the welding points between the stay and the exhaust pipe were changed and the width of the stay was broadened. Thus,

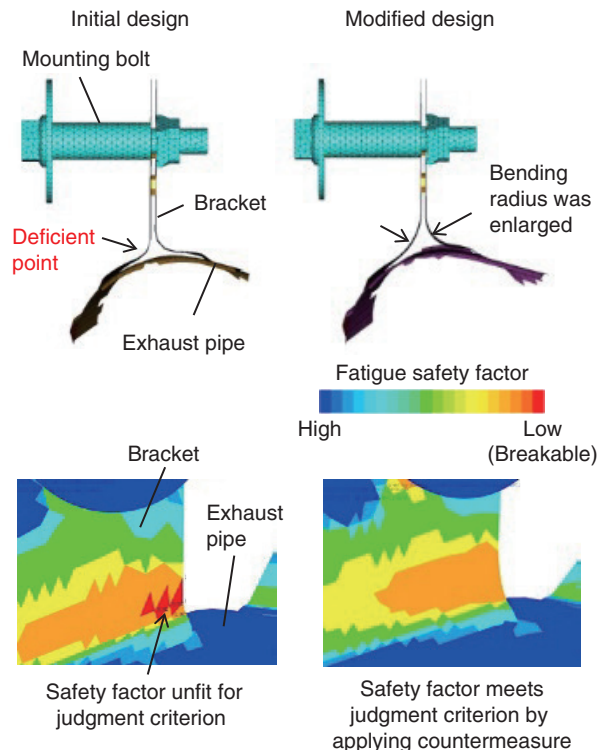


Fig. 26 Difference appeared in contours showing fatigue safety factor before and after application of countermeasures in designing for exhaust pipe mounting bracket

the fatigue safety factor of the concerned curved area had become to meet the requirement.

The comparison of the flow charts between the existing development flow and the redesigned flow applying the developed fatigue strength estimation method is shown in Fig. 28. By applying the fatigue strength estimation method to a design stage of a development, most of the evaluations were made possible to be finalized in the design stage of the development without carrying over those evaluations into the test and redesigning processes using actual prototype vehicles. As a consequence, the application of the developed method realized an increased capability of fatigue durability prediction in design stage and a reduction to minimize the fatigue-originated breakages experienced when testing prototype vehicles. That has enabled us to minimize the frequency of conducting durability tests necessary for developing an exhaust system.

Furthermore, in the conventional procedures, the information on stress available for reference used in the investigations for substituting shapes of the parts were limited to the data only for small localized areas. However, by applying the prediction method developed in this study, the designers are provided with the stress distribution data of whole area of the part as well as the ratio of vibrational stress to thermal stress and the information on

the mechanical behavior including natural vibration modes. The provided information can be utilized in the shape-change countermeasure investigations and the investigation efficiency is drastically enhanced. The other aspect to be noticed is that, by conventional approach, it had been required enormous efforts to make a precise prediction on the effects some shape change might bring about in surrounding areas of changing points. As a consequence, there used to be a case that, after applying a change in certain area of a part, different area on the same part broke in the durability test. On the contrary, using the developed prediction method, it is possible to comprehensively overlook the fatigue durability property for whole areas of a part under investigation and the case described above can be also evaded.

The examples introduced here were only a fraction of the examples actually experienced. There are many other successful application examples that have contributed to our developments. In recent years, other than a fatigue breakage prevention tool, the method is also utilized as an instrument for design investigation of weight reduction. The developed method is used to find out the area on a part that is unnecessarily bulky because of application of an excessively large magnitude of fatigue safety factor. As described above, the fatigue prediction technology developed in this study has been successfully implemented

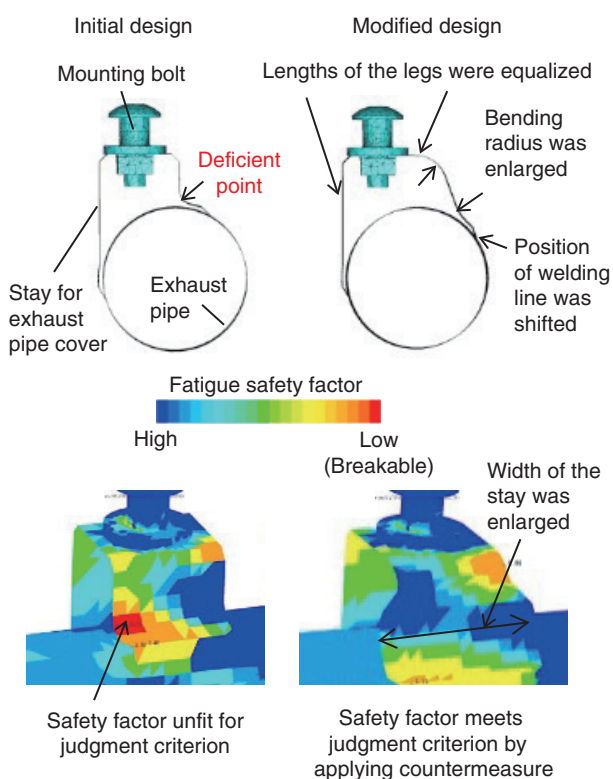


Fig. 27 Difference appeared in contours showing fatigue safety factor before and after application of countermeasures in designing for exhaust pipe cover stay

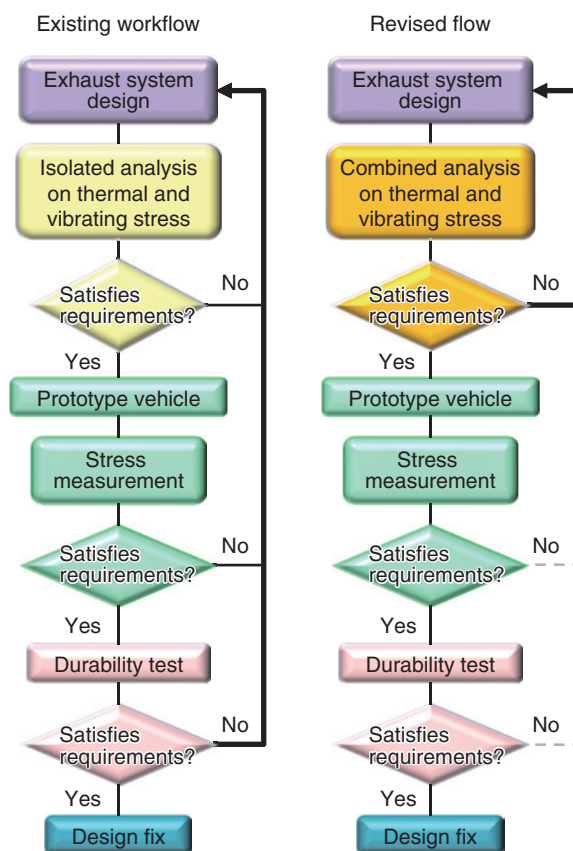


Fig. 28 Existing flow and revised flow of anti-fatigue durability evaluations

in the actual product development process and its significant contribution to the development efficiency has been proved.

7. Conclusion

The following has been achieved through this project.

1. Establishment of fatigue strength simulation method for exhaust system

In this study, the following three methodologies are coupled to make predictions on fatigue durability of an exhaust system in a motorcycle in its early stage of the development.

- (1) Simulation of vibration stress in the exhaust system based on the flexible multi-body dynamics
 - (2) Estimation of surface temperature of exhaust system based on the 3D CFD and the 1D engine cycle simulation
 - (3) A method to calculate safety factor on the fatigue limit diagram by totaling the thermal stress and the vibrating stress
2. Verifications of validity and applicability of the developed prediction method

Verification of validity and applicability of the developed method was conducted using four test motorcycles. The predicted values of fatigue safety factors yielded by applying the developed method exhibited high correspondences with actual measured values. Based on the results, validity and applicability of the developed method was proved.

3. Empirical proof tests on effectiveness in an exhaust system development

The developed method was applied to an actual vehicle development in its early stage of the design process. The potential areas of fatigue breakages were identified and the countermeasures were implemented prior to the prototype vehicle production. By following this procedure, it was made possible to minimize the number of trials for durability tests of the exhaust system components. Moreover, the states of stresses and mechanical behaviors of the entire exhaust system components were made visually represented. Through that information, the designers were able to get the knowledge efficiently for the investigation on necessary changes of the part shapes. This eventually leads to provide high efficiency of decision making on an appropriate shape to achieve high fatigue durability.

Through the achievements mentioned above, it was proved that our established method is truly effective for overall efficiency enhancement of an exhaust system development.

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Development of Method of Prediction of Lifespan of Rubber Tubes in Oscillation-absorbing Sections of Automotive Engines in Extremely Low-temperature Environments

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ABSTRACT

A method of prediction of the lifespan of rubber air flow tubes that connect the air cleaner and the engine and absorb engine oscillation that will contribute to efficient development has been developed.

First, software was selected that would make it possible to numerically simulate the displacement behavior of rubber and the strain generated in the material, and an elastic-plastic CAE model was employed as an analytic model to enable simulation of the nonlinearity of rubber materials.

Tensile tests were then conducted in order to determine the stress-strain characteristics of the materials, and the results were reflected in CAE property data. S-N diagrams were formulated based on the results and results for number of cycles to fracture in unit bellows test pieces in response to repeated oscillation.

Study of the correlation between actual test results and CAE predictions (for bellows behavior and the position of cracks in the bellows) indicated a strong correlation, with a coefficient of determination of 0.92.

The developed method has provided an analytic technique for the prediction of the number of cycles to fracture in air flow tubes based on CAE stress values, making it possible to predict the lifespan of the products.

1. Introduction

Air flow tubes (AFT) (Fig. 2) fitted in automotive engine chambers (Fig. 1) fulfill the function of absorbing engine oscillation. The use environment for these AFT differs depending on the region of use and the vehicle model, and it is therefore necessary to develop AFT tailored to specific conditions. In addition, when acrylonitrile-butadiene rubber mixed with polyvinyl chloride (NBR/PVC), acrylic rubber (ACM), or epichlorohydrin rubber (ECO) is used as the AFT material, the glass transition temperature ranges from around -35°C to -45°C. This means that extremely low-temperature environments (-20°C to -35°C) do not correspond to the range for complete elasticity of the AFT.

Conventionally, CAE techniques have been employed in the super-elastic range in order to predict stiffness in relation to negative pressure in the engine. However, the material characteristics of rubber materials change in different temperature environments, and because there

is no clear method of prediction of the lifespan of the materials in relation to engine oscillation, examining the status of absorption of engine oscillation under different temperature conditions at the desk is not an easy task. Studies involving extremely low temperatures using CAE at the desk are particularly challenging because the super-elastic models used in conventional CAE are not able to accurately reproduce the behavior of rubber in extremely low-temperature environments.

A method of prediction of the lifespan of the bellows sections of the tubes under a variety of environmental conditions was therefore necessary for the efficient development of AFT.

Past research⁽¹⁾⁻⁽³⁾ had not discussed methods of lifespan prediction in extremely low-temperature environments; the present research therefore examined a new method, involving unit oscillation tests of the bellows sections of AFT in extremely low-temperature environments, and simulations using elastic-plastic models.

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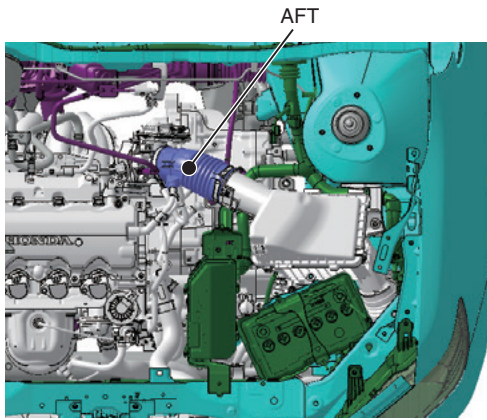


Fig. 1 Engine chamber

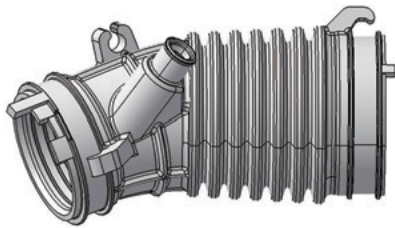


Fig. 2 AFT

2. Concept of Elastic-plastic Model

2.1. Concept of Selection

2.1.1. Selection of analysis methods

In order to accurately predict the lifespan of AFT, which are components that absorb engine oscillations, in extremely low-temperature environments, it is necessary to possess an accurate knowledge of the physical properties of rubber materials. When AFT absorb engine oscillations, their bellows sections deform, and strain is generated. Conventionally, when making judgments regarding the lifespan of materials, the relationship between strain and the number of cycles to fracture under a fixed environment (an ozone concentration of 50 pphm and a temperature of 40°C)⁽⁴⁾ has been used. However, because the stress generated by strain in rubber materials differs considerably⁽⁵⁾ depending on their shape and the temperature environment, it has not been possible to make judgments regarding lifespan based exclusively on strain.

The research under discussion therefore focused on the relationship between stress and temperature, and elucidated the relationship between stress and number of cycles to fracture.

2.1.2. Reason for selection of elastic-plastic model

Figure 3 shows the relationship between stress and strain for a standard rubber test piece in a tensile test. The amount of engine oscillation was controlled through the engine mount system. The results show that the amount of strain

produced in the AFT was 50% or less. The super-elastic models used in conventional CAE studying rubber materials use curves with two inflection points, like the curve shown in Fig. 3, as modeling parameters. However, the range in which AFT manufactured from the rubber material providing the analytic parameters are normally used is the area bounded by the broken line in Fig. 3, close to the first inflection point. The characteristics of the stress-strain curve for this inflection point are approximated by the stress-strain curve for an elastic-plastic body shown in Fig. 4. The present research therefore focused on the range in which strain was 50% or less, and considered that an elastic-plastic model using an elastic model for the stress-strain characteristic of rubber in the elastic range and a plastic model for the stress-strain characteristic of resin in the plastic range would be an appropriate choice as a rubber material model making it possible to accurately reproduce the material's nonlinearity.

2.2. Identification of Stress-strain Curves for Elastic-plastic Model

Rubber test pieces shaped as thin rectangles rather than dumbbells (Fig. 5) were employed to determine stress-strain curves to be used for elastic-plastic models in extremely low-temperature environments. When tests are conducted using dumbbell-shaped test pieces, the cross-sectional area of the neck section changes as shown in Fig. 6 close to the glass transition temperature, and the stress distribution becomes distorted, leading to deviations in tensile stress. The use of thin rectangular test pieces makes it easy to calculate the cross-sectional area for each displacement, and to calculate

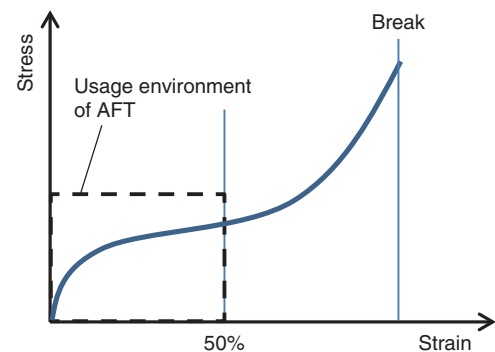


Fig. 3 Stress-strain curve for rubber

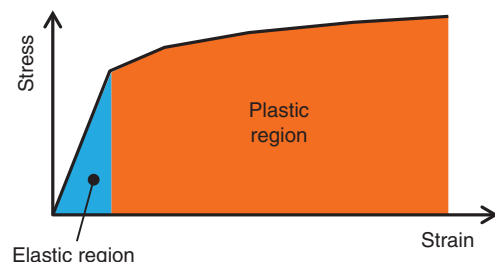


Fig. 4 Elastic-plastic model

the necessary true stress for the analysis. A low tensile speed of 12.5 mm/min was employed in order to control the generation of heat. The circles in Fig. 9 show the stress-strain curves for different temperatures obtained from tests.

2.3. Correlation between Test and CAE Figures

CAE was conducted using Abaqus for CATIA V5.

The elastic-plastic model employed to reproduce the stress-strain characteristics of the rubber material using CAE used physical property parameters for the material in the

elastic and plastic ranges. For the purposes of correlation, JIS tensile tests (Fig. 7) and tests modeling these using CAE (Fig. 8) were conducted, and the results for temperatures between -25°C and -35°C were obtained, as shown in Fig. 9.

Based on Fig. 3, the range up to a strain of 2% was considered the elastic range, with that from this point onwards considered the plastic range. The elastic range was defined using the initial modulus of elasticity and Young's modulus, and the plastic range was defined using data from a table of results for true stress against plastic strain at multiple points. For the elastic-plastic model input values, values from CAE analyses were compared with test values for stress-strain characteristics for different materials and different temperatures, and correlations were verified via parameter fitting. Figure 9(a) shows stress-strain curves for NBR/PVC at different temperatures based on tensile test results and CAE analyses. Figure 9(b) shows the correlation between the test stress figures and CAE stress analysis results for different materials and different temperatures at various levels of strain in tensile tests. This study of the correlation between test and CAE results shows a strong correlation, with a coefficient of determination of 0.99.

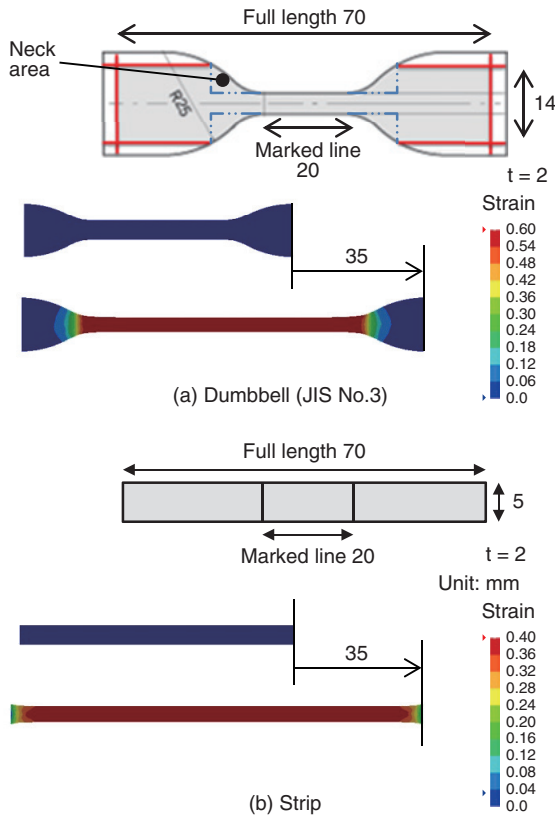


Fig. 5 Test piece

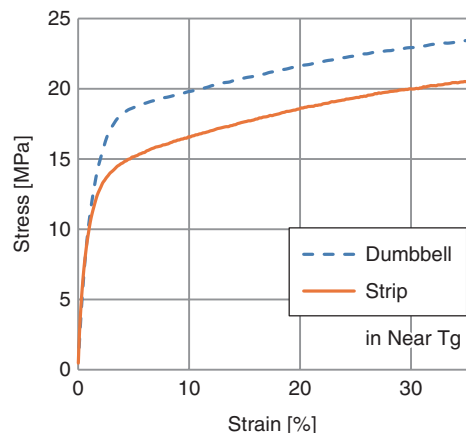


Fig. 6 Difference in stress

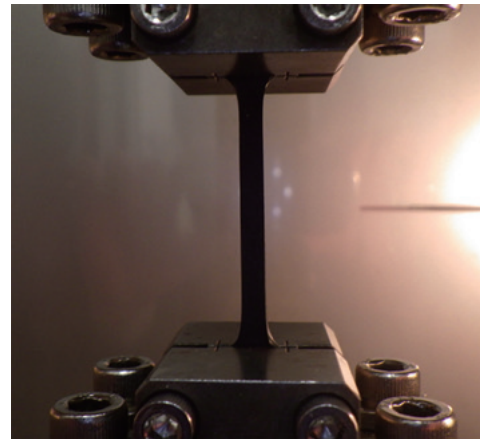


Fig. 7 JIS tensile test

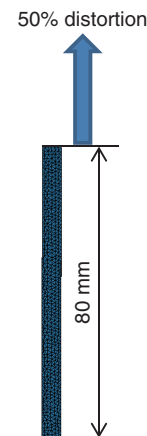
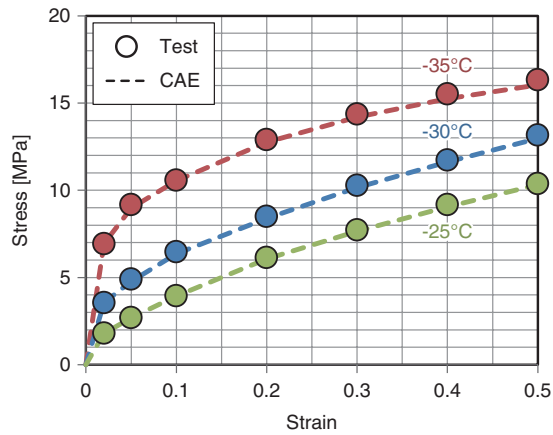
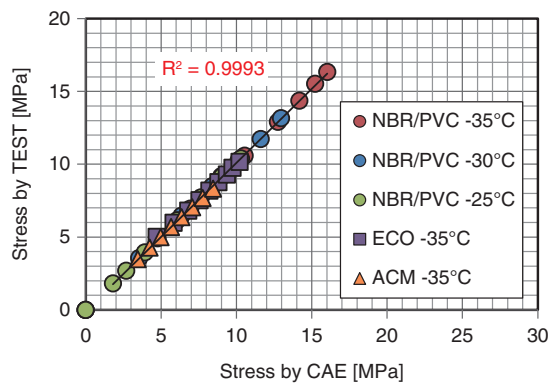


Fig. 8 JIS tensile test (CAE)



(a) Stress-Strain Curve



(b) Correlation between CAE and test results for each material

Fig. 9 Results of tensile test

3. Test Method

3.1. Extremely Low-temperature Unit Oscillation Test on Bellows

These tests were conducted in order to obtain accurate information regarding changes in the shape of the bellows section of the AFT and the number of cycles to fracture. As shown in Fig. 10, straight rubber tubes with three folds in the bellows section were produced, and these test pieces were subjected to unit oscillation tests at extremely low temperatures using a servo pulser with a low-temperature thermostat bath (the EHF-LV010KI-A04SP manufactured by Shimadzu Corporation, shown in Fig. 11).

In order to study the behavior of the test pieces and the number of cycles to fracture in different directions, the test pieces were vibrated and tested while fixed in a number of different positions, as shown in Fig. 12: Vertically (0°), with tensile and compression loads applied in an up-down direction [Fig. 12(a)]; horizontally (90°), with a shearing load applied in a left-right direction [Fig. 12(b)]; and diagonally (45°), with a composite tensile, compressive, and shearing load applied [Fig. 12(c)]. Tests in these directions were also conducted at different temperatures

in order to study the temperature-dependence of behavior and number of cycles to fracture in each direction. These tests were conducted separately for NBR/PVC, ACM, and ECO as materials. Engine oscillation behavior during actual acceleration and braking were measured, and a trapezoid wave with a plateau period was employed in the tests as shown in Fig. 13 in order to simulate this behavior.



Fig. 10 Unit body bellows test piece

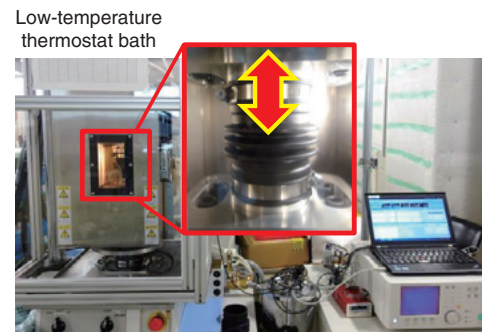


Fig. 11 Strength-testing apparatus

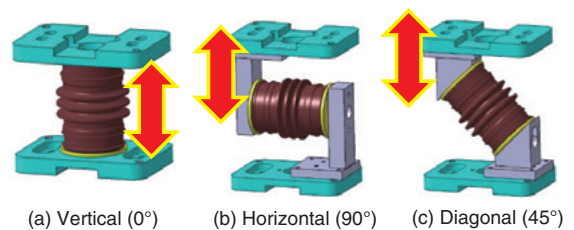


Fig. 12 Direction of oscillation

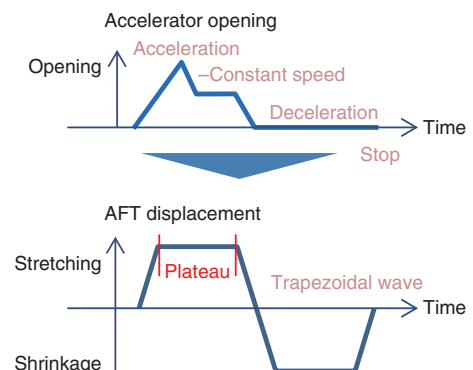


Fig. 13 Waveform for oscillation test

3.2. Extremely Low-temperature Oscillation Tests on AFT Products

Oscillation tests (Fig. 14) were also conducted in extremely low-temperature environments in order to study the correlation of test and CAE results for complex product shapes (Fig. 2).

The tests were conducted at -35°C , using sine waves of a frequency of 2 Hz. The amplitude and direction of the waves were set to generate the maximum displacement in engine oscillation.

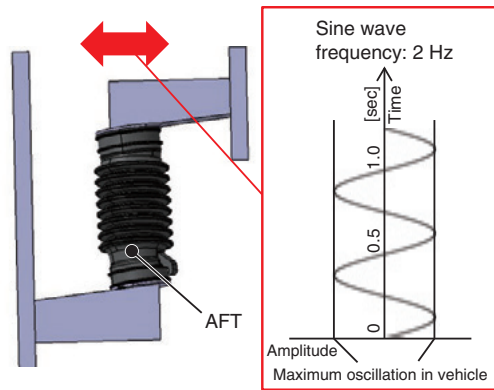


Fig. 14 Strength-testing apparatus for actual AFT

4. Test Results and Considerations

4.1. Test Results

Using unit bellows test pieces, repeated oscillation tests of the bellows were conducted at extremely low temperatures for the three displacement directions described above. Figure 15 shows results for the number of cycles to fracture of the AFT.

Figure 16 shows photographs of the positions of the fractures following each oscillation test.

4.2. CAE Analysis Results

Figure 16 shows the results of CAE analyses of von Mises stress (the maximum generated stress) generated in the bellows in response to a forced displacement input to the unit bellows test pieces for the same three displacement directions as discussed in Section 4.1. Figure 16 shows contour diagrams of the stress generated under each oscillation condition.

4.3. Considerations

In the test results, the order of the directions of oscillation in terms of the number of cycles to fracture, from lowest to highest, was diagonal direction [Fig. 12(c)], followed by up-down direction [Fig. 12(a)], and left-right direction [Fig. 12(b)]. In the CAE results, the order of the directions of oscillation in terms of stress values, from highest to lowest, was diagonal direction, up-down direction, and left-right direction; the higher the stress values,

the lower the number of cycles to fracture. In addition, the positions of the fractures for each oscillation direction in Fig. 16 correspond to the positions of maximum stress and fracture from CAE results for maximum stress.

Figure 18(a) shows an S-N diagram that plots the relationship between the generated stress in CAE analyses and the number of cycles to fracture in actual tests based on the results shown in Figs. 15 and 17. As this diagram shows,

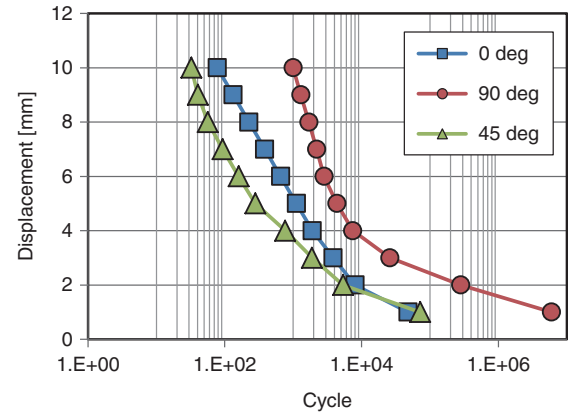


Fig. 15 Results of oscillation test

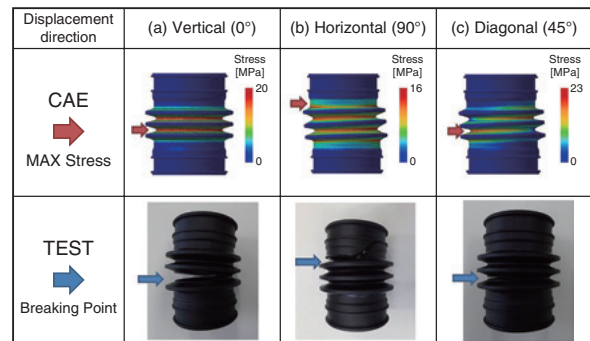


Fig. 16 Comparison between oscillation test results for test and CAE

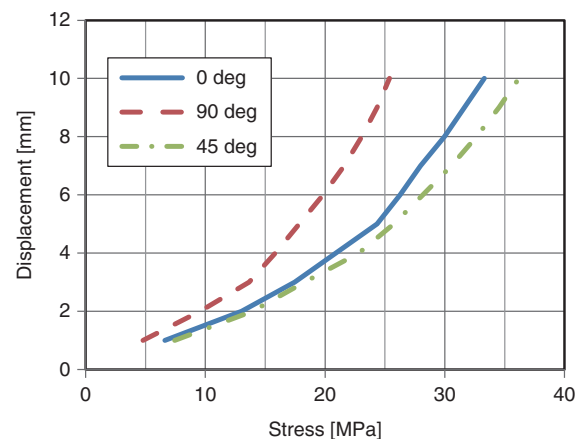
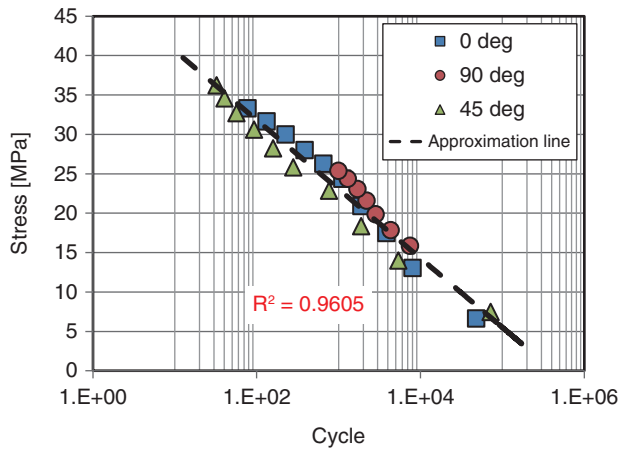


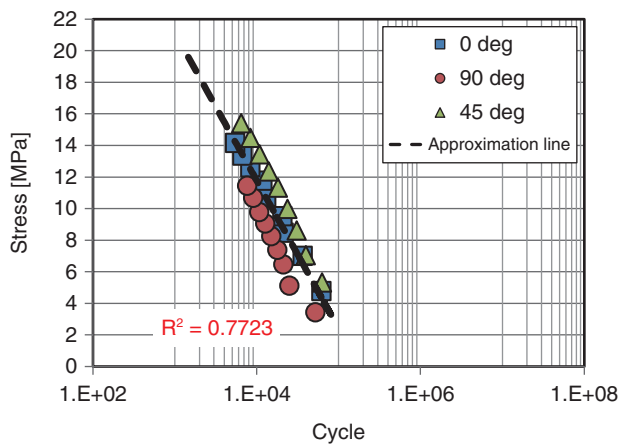
Fig. 17 CAE results

there is a strong correlation between stress and the number of cycles to fracture, with a coefficient of determination of 0.96.

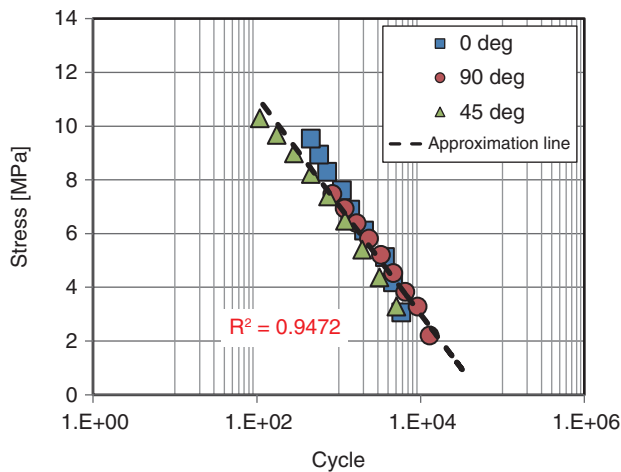
The results of plotting lifespan curves for each of the other materials [Figs. 18(b) and (c)] are considered to



(a) Results for NBR/PVC



(b) Results for ACM



(c) Results for ECO

Fig. 18 Stress and number of cycles to fracture for each material

demonstrate the potential to predict the lifespan of AFT in relation to oscillation based on the number of cycles to fracture of the material, as determined by the calculation of stress values using CAE.

5. Verification

In order to verify the validity of these lifespan curves, stress analyses and extremely low-temperature AFT oscillation tests (Fig. 14) were conducted using actual AFT product shapes (Fig. 2). Figure 19 shows a graph plotting results for the number of cycles to fracture of the AFT in actual oscillation tests against the predicted lifespan curve.

Figure 20 shows the correlation between the number of cycles to fracture in the extremely low-temperature unit bellows oscillation tests (Fig. 11) and the number of cycles to fracture estimated from CAE stress values [Fig. 18(a)]. There is a strong correlation between the results, with a coefficient of determination of 0.92.

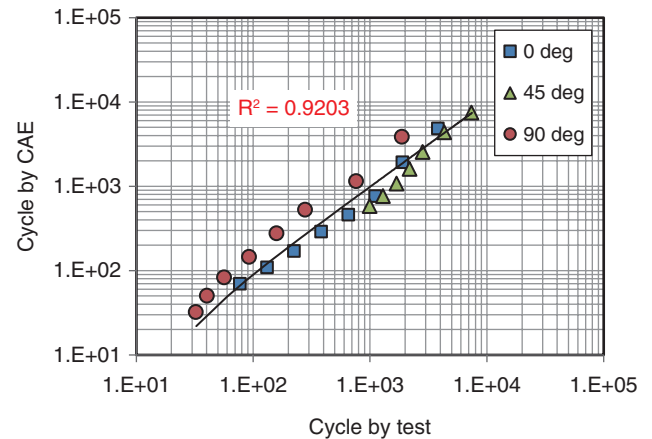


Fig. 19 Correlation between CAE and test for each material

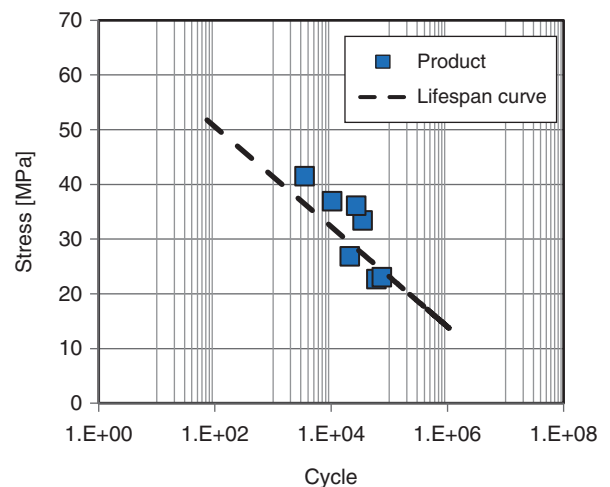


Fig. 20 Lifespan curve and test results

The results discussed above indicate that when considering the durability of AFT in relation to repeated oscillation in extremely low-temperature environments during the development process, it is possible to predict the number of cycles to fracture using lifespan curves for different materials plotting number of cycles to fracture against generated stress, based on the results of numerical analyses of stress conducted in advance for actual product shapes using CAE.

6. Conclusion

The research discussed in this paper realized a good correlation between predictions of the lifespan of bellows sections in AFT in extremely low-temperature environments and test results.

- (1) High-accuracy material characteristic curves were obtained by modifying the measurement method employed at extremely low temperatures;
- (2) Extremely low-temperature oscillation tests were conducted using unit bellows test pieces; and
- (3) S-N diagrams were generated using unit test and CAE results.

The results of this research have made it possible to predict the number of cycles to fracture for AFT from the results of numerical analyses of stress generation in AFT products (Fig. 2) using CAE.

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Non-linear CAE Technology to Balance Handling and Ride Comfort

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ABSTRACT

Honda's in-house vehicle dynamics simulation tool has been enhanced to balance ride comfort and handling performance more effectively. To deal with the characteristics that affect ride comfort, ride comfort tire model was newly introduced and its parameter identification method was developed for better reproduction of vibration transmission characteristics from road to tire. In order to consider suspension vibration transmissibility from tire to vehicle body, simulation models for rubber bushings, ball joints, and dampers were newly developed that can consider amplitude and frequency dependence. Their parameter calculation methods were also developed. In addition, to enhance the reproducibility of a vibration originated from engine, a hydraulic engine mount model that can reproduce amplitude and frequency dependence as well as its parameter identification method were developed. By incorporating the above technologies into our vehicle dynamics simulation tool, it became possible to study design more detailed specifications to balance handling and ride comfort performance at early stages of vehicle development.

1. Introduction

Contemporary vehicle development is expected to create attractive products that balance driving pleasure and comfort as well as good fuel efficiency and safety. In addition, enhancement of the product development process to support rapidly changing and diversifying customer demand is a common issue for vehicle manufacturers.

Under these circumstances, CAE is increasingly important to enhance the efficiency and quality of vehicle development. CAE is used in the early phases of vehicle development to study design concepts that balance opposing performances such as fuel efficiency and dynamic performance. CAE is also widely applied in later development stages, for example to predict vehicle performance and enhance specifications in the stage where vehicle and subsystem characteristics are reflected in the drawings. Multibody simulation (MBS) is commonly used to evaluate vehicle dynamics performance areas during this process.

MBS is a nonlinear numerical simulation method that calculates the dynamic behavior of mechanical systems comprising multiple parts. MBS solvers formulate equations of motion from inertia properties of bodies and kinematic constraints or force elements between parts, and solve these equations numerically to simulate the complex behavior of

mechanical system⁽¹⁾.

Honda began employing MBS in the vehicle dynamics performance area from the latter half of the 1980s. Honda initially applied MBS to evaluate suspension kinematics, for example prediction of alignment characteristics during the suspension stroke. Thereafter, use was expanded to various analyses, such as vehicle handling simulations and flexible multibody dynamics simulations⁽²⁾. In the area of handling performance, MBS is used to study the characteristics of parts that make up subsystems such as the suspension to meet subsystem requirements as determined by the design concept. Design employing MBS to predict performance feasibility in parallel with parts layout is now standard practice.

Handling and stability as aspects of handling performance are mainly studied by analyzing the motion of rigid bodies up to 5 Hz. In contrast, the study of ride comfort, which has a trade-off relationship with handling performance, deals with vibration phenomena up to around 50 Hz, making the expression of nonlinear parts a key factor. The nonlinearity of parts such as the rubber bushings, dampers, and tires and changes in the vibration transmission characteristics due to the amplitude and frequency of vibration have a large impact on the transient response when running on an uneven road. Because this calls for detailed modeling of these parts when studying

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ride comfort, it is not a simple matter to consider the balance between handling and ride comfort performance in the product planning phase, and ride comfort has generally been studied using prototype vehicles in the later stages of development.

However, as mentioned above, ride comfort performance should essentially be designed in parallel with handling performance. This is because enhancement of both types of performance calls for a good balance between responsiveness and isolation characteristics, and to achieve this balance it is necessary to appropriately set subsystem requirements and design the specifications of subsystem parts before determining the overall vehicle layout.

Against this background, the authors expanded the functions of an MBS design tool conventionally used to predict handling performance by adding model elements related to ride comfort. This made it possible to study handling and ride comfort performance in parallel for the same parts specifications in the initial phase of vehicle development.

2. Development Goals

Ride comfort refers to the phenomenon where input from the road is transferred to the vehicle body and steering system via the tires and suspension and perceived as vibration by vehicle occupants. Figure 1 shows typical phenomena that influence ride comfort.

For example, on an undulating road, sprung-mass vibration of 1 to 2 Hz in frequency occurs with relatively large amplitude of several mm to 10 mm or more. Drivers perceive this as a float feel, so the balance between front and rear natural frequency is designed to achieve rapid convergence of sprung-mass vibration.

Even when running on a smooth-surfaced road, unevenness of the road surface may generate suspension strokes with amplitude of 1 mm or less. Under these conditions the nonlinear characteristics of parts such as the dampers, ball joints, and rubber bushings result in a

boulevard jerk feeling with vibration of approximately 3 to 8 Hz, which is an issue for ride comfort.

In addition, rotational shimmy of the steering system is a form of vibration where imbalance between the tires and wheels generates periodic longitudinal input to the suspension. This input frequency is proportional to the running speed, and occurs in the range of approximately 10 to 20 Hz, which is near the longitudinal natural frequency of the unsprung mass.

As this indicates, ride comfort phenomena occur over a wide range of amplitudes and frequencies. To enhance performance in this area, it is therefore necessary to design characteristics that contribute to ride comfort in the target frequency and amplitude ranges, using models able to accurately reproduce parts with nonlinear characteristics.

Furthermore, to perform these studies in the initial phase of development, it is desirable to quickly build models from parts specifications, predict handling and ride comfort performance using the same model, and study parts specifications that balance the respective performance demands.

Therefore, the following three goals were set to address the above issues.

- (1) Realization of the ability to express the nonlinearity of parts within the range of amplitudes and frequencies relevant to ride comfort phenomena.
- (2) Realization of functionality that enables designers to study performance for typical ride comfort phenomena with simple operations.
- (3) Realization of accuracy sufficient for studying parts specifications.

3. Model Expression related to Ride Comfort Characteristics

Figure 2 shows the main elements comprising the dynamic stiffness of the suspension in the vibration transmission paths from the tires to the body.

Vehicle motion is excited by input from the tires, so

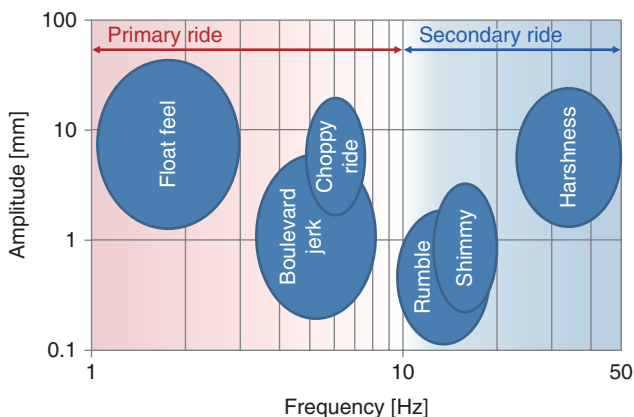


Fig. 1 Typical ride comfort feeling

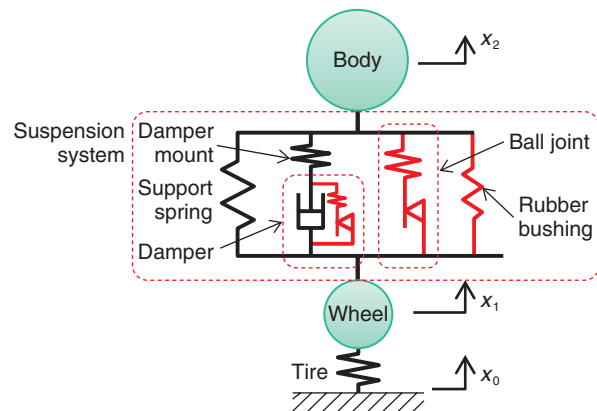


Fig. 2 Quarter-car model including hysteresis elements in suspension components

the reproduction accuracy of the forces generated on the tires has a large influence on the predictive accuracy of simulations for both handling and ride comfort performance.

In addition, the forces generated on the tires move the vehicle body via the suspension arms, springs, and dampers, but it is known that the friction of the joints and the stiffness of the dampers and rubber bushings in the transmission paths fluctuate greatly depending on the amplitude and frequency of the input. Furthermore, the characteristics of the engine-mount bushings that support the engine also greatly affect ride comfort and NVH. Incorporation of these characteristics models into the MBS design tool would make it possible to study ride comfort and NVH characteristics, which have a trade-off relationship with handling, together with handling in the initial phase of development, thus enabling creation of design proposals that achieve a good balance of conflicting performances. This chapter gives an overview of this modeling method.

3.1. Ride Comfort Tire Model

Due to increasing demand for better fuel efficiency, rolling resistance is being reduced by reducing the thickness of tire structures and using rubber compounds with low hysteresis loss. As a result, characteristics related to handling and ride comfort performance are changing greatly. Examples of computer simulation tire models (hereafter, tire models) that can reflect these changing tire characteristics include RMOD-K or FTire models that use small tread segments to reproduce both detailed tire contact patch behavior and whole tire structural responses, and FEM tire models that reproduce details down to the internal structure of tires⁽³⁾⁻⁽⁶⁾.

Honda adopted the theory of Magic Formula (MF)⁽⁷⁾ published in 1987 to study handling performance when beginning to apply MBS to chassis development. Since then, in-house tools have been used to create an MF tire model from measured characteristics, and the tire model has been enhanced while examining performance in the nonlinear region where lateral and longitudinal forces are combined, such as BIT (Braking In Turn).

In addition, Honda quickly focused on the 1st-order lag characteristics equation (1) of the lateral force transmission due to lateral deflection of the tire, theoretically clarified its influence on both vehicle dynamics performance and the driver-vehicle system⁽⁸⁾⁻⁽¹⁰⁾, and incorporated it into the tire model.

In vehicle development it is increasingly important to simultaneously consider the nonlinear characteristics and frequency dependency of the bushings, the stiffness of the body, and other characteristics in the initial phase of development when deciding the suspension member layout and the body framework, in order to find the best trade-off solution between ride comfort and handling performance^{(2), (11), (12)}. As a result, it was necessary to expand the functions of Honda's tire model to handle even higher-frequency vibration issues.

$$\tilde{F}_{y0}(t) = \int_0^t F_{y0}(\tau) \frac{1}{T_i} e^{-\frac{t-\tau}{T_i}} d\tau \quad \text{: Magic Formula with 1st order lag} \quad (1)$$

where

$$\left\{ \begin{array}{l} T_i(\alpha|F_z) = \frac{\frac{\partial F_{y0}}{\partial \alpha_y} |_{\alpha_y=0}}{K_y V} \quad \text{: Time constant} \\ F_{y0}(t) = D_y \sin \{ C_y \arctan \{ B_y \alpha_y - E_y [B_y \alpha_y - \arctan(B_y \alpha_y)] \} \} \end{array} \right.$$

When focusing on reproducibility of tire force input when running on a typical road with a frequency range up to approximately 50 Hz and undulation amplitude up to approximately 10 mm, the MF-Swift model⁽¹³⁾ that simulates behaviors of elastically supported tire tread rings excited by road undulation offers good calculation speed and is highly compatible with handling performance. Honda is therefore working to standardize this MF-Swift tire model and its application scheme in development.

The MF-Swift model is accurate enough to reliably reproduce measured cornering characteristics such as longitudinal force, lateral force, and SAT, as shown in Fig. 3. With regard to ride comfort characteristics, the model can accurately reproduce the enveloping characteristics of tires, that is to say the behavior of the tread surface to envelope road surface unevenness at low speeds, as shown in Fig. 4.

In addition, the reproduction of transient tire forces when the tires roll over a cleat in the road also shows good approximation of the peak values, frequency, and convergence of vibration (Fig. 5).

As tire measurements for the identification of tire models need large-scale, specialized engineering facilities, Honda had mainly relied on tire manufacturers and/or external organizations for these measurements. However, a tire tester was newly developed and introduced to measure tire characteristics that are not affected by test

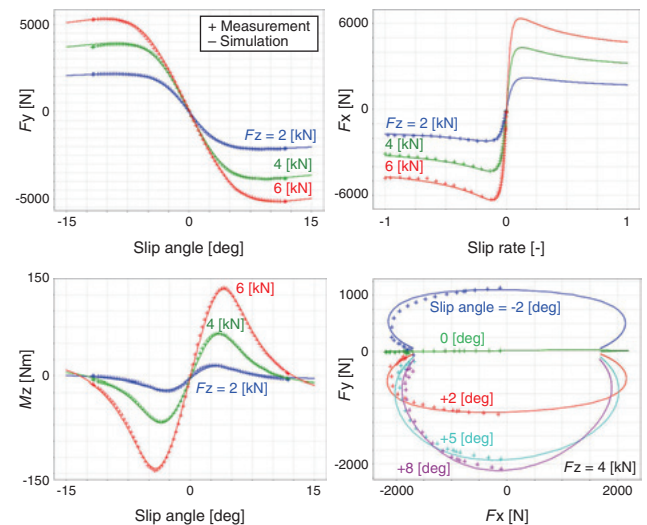


Fig. 3 Example of force and moment characteristics of tire 175/65R15, IP = 220 [kPa], measurement and its simulation with MF-Swift

facilities or conditions. This enabled the formulation of models using the test results and accurate comparison of tire characteristics⁽¹⁴⁾. The use of MF-Swift models helps to accurately predict input forces from tires, which is important for the study of ride comfort phenomena, and enables theoretical examination of the trade-off between handling and ride comfort.

3.2. Suspension Friction Model

When addressing vibration phenomena, measures are put into effect from low frequencies that have a broad influence. An effective approach for addressing the low-frequency ride comfort phenomena of boulevard jerk and rumble is to reduce the friction included in the ball joints and dampers

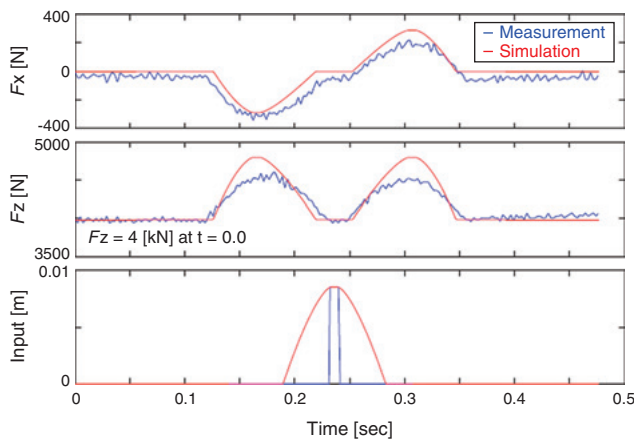


Fig. 4 Example of enveloping behavior of tire 175/65R15 with $F_z = 4$ [kN], $IP = 220$ [kPa], $V = 0.27$ [km/h], obstacle of 8.5×8.5 [mm], measurement and its simulation with MF-Swift

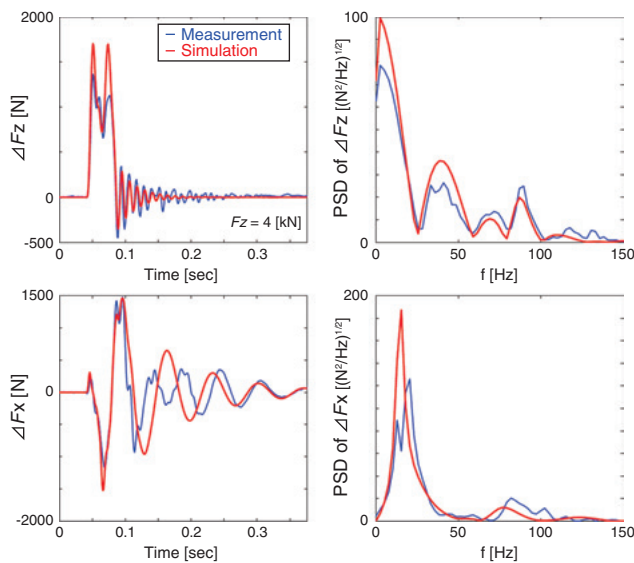


Fig. 5 Example of 2D (15 × 15 [mm]) 90 [deg] cleat response of tire 175/65R15 $V = 20$ [km/h]

that contributes to these phenomena during the suspension stroke, in order to suppress the increase in the natural frequency of the suspension in the small amplitude range⁽¹⁵⁾.

3.3. Modeling of Damper Force

Reducing the damping force is an effective means of addressing the boulevard jerk and rumble phenomena produced by the dampers, but this also reduces the handling performance, so it is not easy to achieve a good balance.

However, if the damping force at small velocity area that contributes to ride comfort and the damping force needed to suppress vibration for handling behavior could be adjusted separately, this would enable studies to achieve a better balance between the two demands.

Therefore, the force produced by the dampers was divided into a component originating in friction and one originating in the oil as shown in Fig. 2, and the nonlinear series spring elements in the dampers were taken into consideration. Use of this model made it possible to reproduce the damper force, which changes according to the frequency and amplitude, as shown in Fig. 6, and to separately design the damper characteristics in the range of frequencies and amplitudes associated with handling and ride comfort phenomena.

3.4. Rubber Bushing Model

The rubber bushings are the structural elements that exhibit the next-greatest degree of nonlinearity after the tires and dampers. The spring characteristics of the damper mount rubber and the compliance bushings in particular contribute greatly to the handling characteristics, and the amplitude dependency and frequency dependency of these characteristics also contribute to ride comfort performance.

Figure 7 shows a schematic of the applied model⁽¹⁶⁾. This model consists of three nonlinear elements: a displacement-dependent spring element in parallel mainly used to study handling performance, and amplitude- and frequency-dependent elements mainly used to study ride comfort performance.

This model is compatible with MBS, and can be used to analytically study dynamic stiffness together with time-

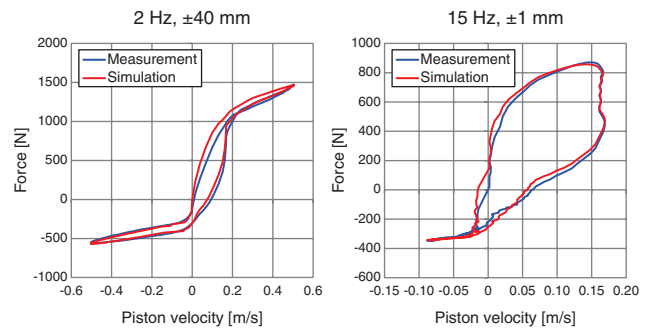


Fig. 6 Example of damper force characteristics with different input amplitude and frequency

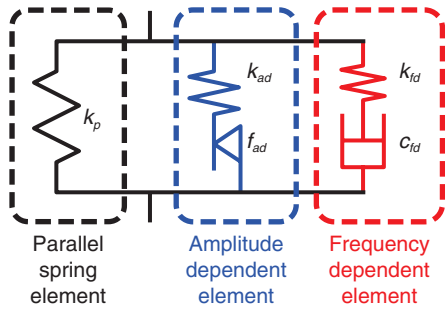


Fig. 7 Rubber bushing model for vehicle dynamics performance design

series simulations that take a certain amount of calculation time, enabling more efficient study of performance.

When the friction force f_{ad} and the series spring constant k_{ad} are given, the storage stiffness $K_{ad}'(a)$ and loss stiffness $K_{ad}''(a)$ of the amplitude-dependent element can be determined from the following equations where a denotes vibration amplitude.

$$K_{ad}'(a) = \frac{f_{ad}}{a} \tanh \frac{a}{a_{ad}} \quad (2)$$

Table 1 Parameters for rubber bushing model

Static stiffness K	7500 N/mm
Dynamic multiplication d	1.24 (± 0.05 mm, 100 Hz)
Loss factor L_f	0.05 (± 0.5 mm, 15 Hz)
Freq. dependence ratio r_{fd}	690 N/mm/100 Hz

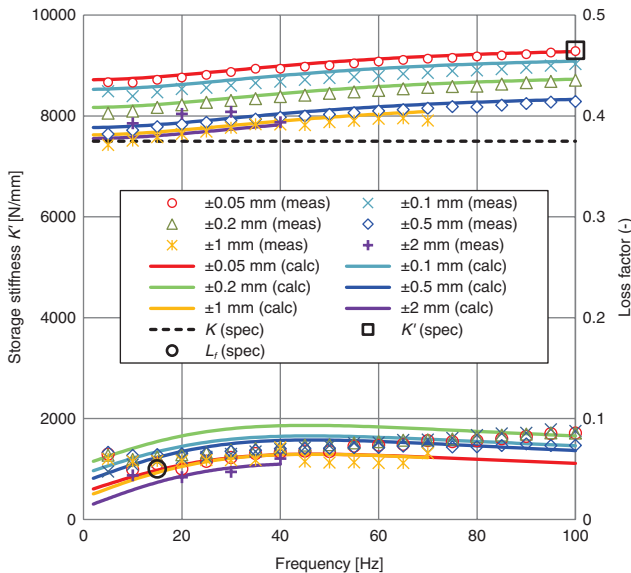


Fig. 8 Comparison between measurement and calculation results for dynamic stiffness of rubber bushing

$$K_{ad}''(a) = \frac{4f_{ad}}{\pi a^2} \left(a - a_{ad} \tanh \frac{a}{a_{ad}} \right) \quad (3)$$

Likewise, when the viscous damping factor c_{fd} and the spring constant k_{fd} are given, the storage stiffness $K_{fd}'(\omega)$ and loss stiffness $K_{fd}''(\omega)$ of the frequency-dependent element can be determined from the following equations where ω denotes vibration frequency.

$$K_{fd}'(\omega) = \text{Re} [G(j\omega)] = \frac{\frac{c_{fd}^2}{k_{fd}} \omega^2}{\left(\frac{c_{fd}}{k_{fd}} \right)^2 \omega^2 + 1} \quad (4)$$

$$K_{fd}''(\omega) = \text{Im} [G(j\omega)] = \frac{c_{fd} \omega}{\left(\frac{c_{fd}}{k_{fd}} \right)^2 \omega^2 + 1} \quad (5)$$

By calculating the given parameters for Eq. (2) to Eq. (5) from the minimum parameters that express the dynamic characteristics of the part (Table 1)⁽¹⁶⁾, the dynamic stiffness of the rubber bushing can be reproduced and applied to study performance as shown in Fig. 8.

3.5. Hydraulic Mount Model

The use of hydraulic mounts to enhance ride comfort and quietness has become the main solution for the mounting of automotive power plants. The basic structure and principle of operation of these mounts are described below (Fig. 9). The interior of the main rubber of a hydraulic mount is filled with fluid, and closed off with a diaphragm. A separator is used to form two fluid chambers, with an orifice running between them. This orifice is called a shake orifice, and the damping characteristics can be determined by the shape of the main rubber and the inner diameter and length of the orifice. When the main rubber is deformed by a load, fluid passes through the orifice, producing damping effects at a specific frequency due to the frequency response of the fluid mass within the orifice. This frequency is normally set near the resonant frequency of the power plant.

The resonance of the shake orifice produces damping effects, and at the same time the internal pressure of the mount rises, which increases the dynamic stiffness with small amplitude and would increase the amplitude

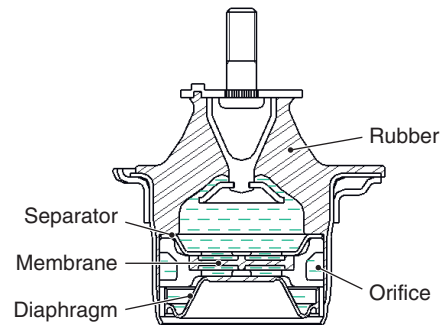


Fig. 9 Basic structure of hydraulic engine mount

dependency of the dynamic characteristics. To address this issue, low dynamic multiplication materials are generally applied for the main rubber, and a liquid pressure-absorbing mechanism such as a membrane is also often used.

To accurately model these hydraulic mount characteristics, the actual phenomena are reproduced in the simulation by using a combination of multiple-degree-of-freedom models as shown in Fig. 10.

Here, F_1 , F_2 , and F_3 are respectively defined to reproduce the static characteristics of the main rubber, the frequency dependency of the main rubber, and the expansion stiffness and amplitude dependency produced by the hydraulic effect. The total of these forces is the transmitted force relative to the mount displacement ($X - X_g$).

The above process makes it possible to reproduce the complex characteristics of hydraulic mounts in simulations (Fig. 11). This increases the predictive

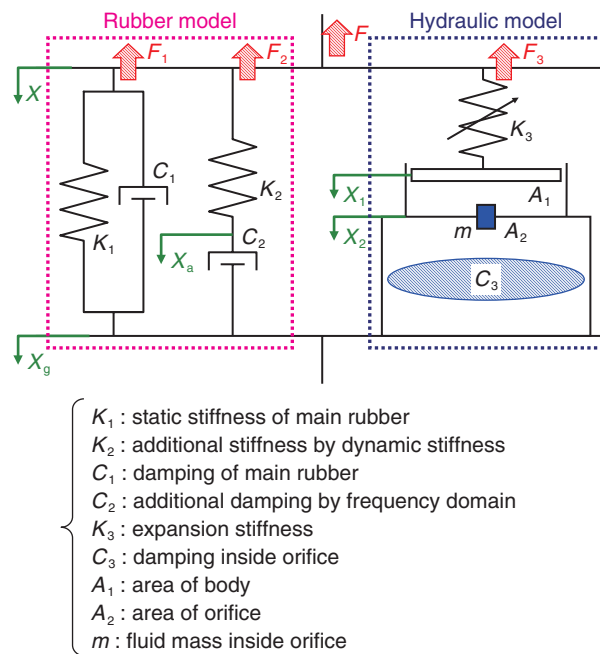


Fig. 10 Hydraulic engine mounting model for MBS

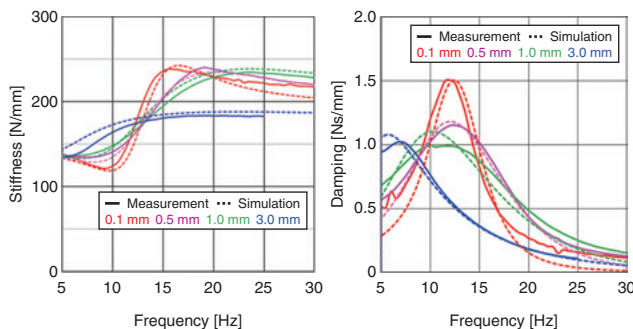


Fig. 11 Characteristics of hydraulic engine mounting model

accuracy of simulations under conditions of non-steady-state displacement that approximate actual operating conditions in the initial design phase, and enables setting of the optimum engine mount positions and sizes, which are challenging to change from the intermediate design phase onward (Fig. 11).

4. Dynamic Performance Design Tool

Studies using MBS that take into account the model elements discussed in the previous chapter are needed to design specifications of parts that balance handling and ride comfort.

However, it is not easy for designers to create models with the degree of detail needed to perform studies, run simulations, and analyze the results. Therefore, dedicated software was created that enables designers to easily perform this series of operations.

Figure 12 shows an example of screens used to define bushing characteristics in the MBS tool for designers. When the designer defines parts specifications such as static stiffness and dynamic stiffness, those characteristics are automatically applied to the simulation model.

In addition, Fig. 13 shows an example of an excitation test screen. The designer simply inputs the test conditions

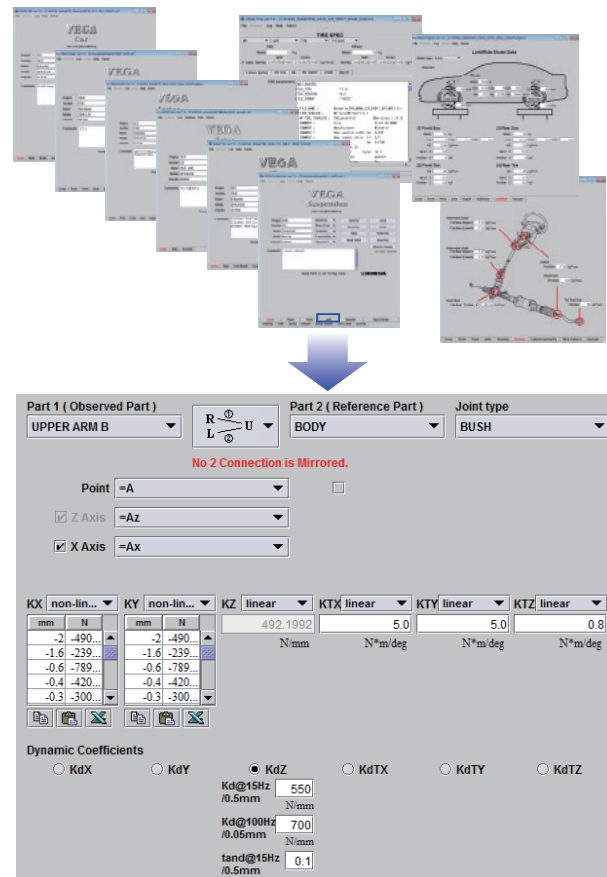


Fig. 12 Joint definition panel of MBS tool for designer

such as the subsystem assembly conditions, the excitation points, and the excitation waveform, and the software automatically generates a simulation model based on the model definitions and test conditions (Fig. 14), and runs the simulation.

Analysis screens are provided with menus offering simulations of typical vehicle tests and bench tests. The designer can run the necessary simulations and study parts specifications that achieve a balance between handling and ride comfort.

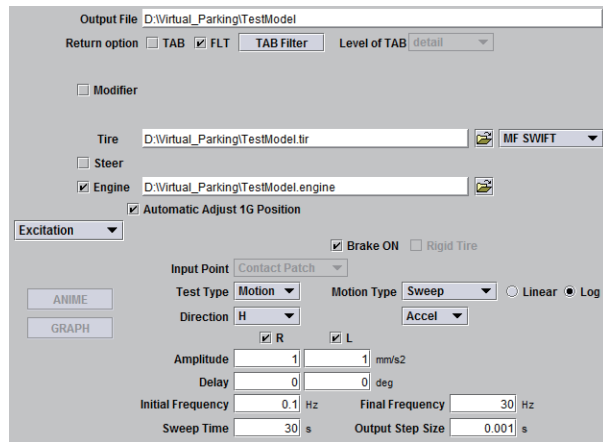


Fig. 13 Chassis excitation test panel in MBS tool

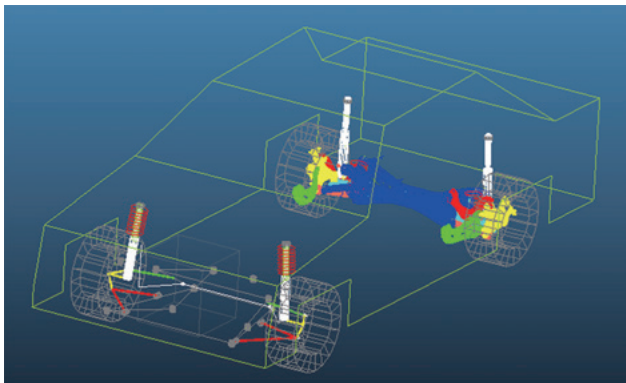


Fig. 14 MBS full vehicle model

5. Case Studies

5.1. Balancing Yaw and Roll Phase Delay and Flat Ride Performance

The phase delay of the vehicle yaw rate, lateral acceleration, and roll angle in response to steering are important characteristics that affect the feeling of responsiveness of steering operations. When steering operations produce a cornering force on the tires, first yaw motion is generated, followed by lateral motion and roll

motion. When the phase difference between this series of motions is large, the driver loses a sense of responsiveness and nimbleness in steering operations.

For example, correction of the roll phase delay relative to yaw motion is achieved by optimally designing specifications such as the roll center height in the initial phase of development, and then optimizing specifications such as the roll stiffness in later phases of development to enhance the ride feel. However, the roll center height is subject to layout constraints such as the minimum height above ground, and if the roll stiffness is adjusted by means of the suspension springs, ride comfort performance in the float feel region is affected. In the actual development process it is necessary to determine parts specifications in consideration of these opposing performance demands.

Figure 15 shows an example of evaluation results for yaw and roll phase delay and vertical vibration transmission characteristics in a midsize vehicle, with the front suspension specifications as the design variables.

Looking first at the spring rate, increasing this parameter reduces the phase delay, but also increases the natural frequency of the sprung mass, which reduces the ride comfort performance. Next, modification of the stabilizer stiffness and the roll center height makes it possible to reduce the phase delay with little change in the ride comfort performance, but in practice it is necessary to consider durability and layout constraints. By understanding the contribution of the characteristics of each part and studying their combinations in this manner, the proposed specifications can be determined as shown in Fig. 15.

Other parameters that affect ride comfort performance, such as the dynamic multiplication of the suspension bushings, can also be made the subject of design focus, and specifications from suspension kinematics to dynamic stiffness can be examined theoretically while evaluating handling and ride comfort performance.

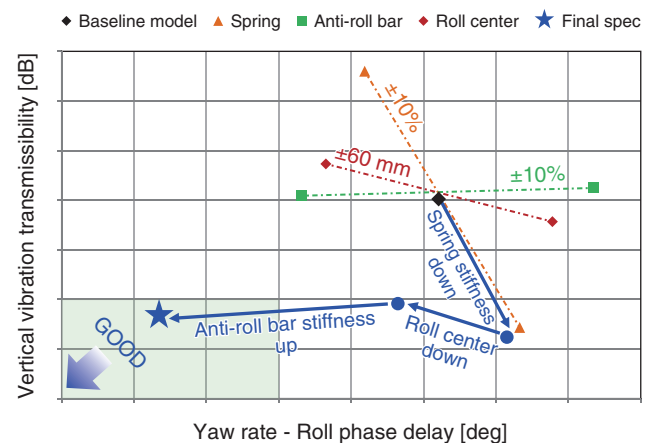


Fig. 15 Example of parameter study in which spring, anti-roll bar and roll center height are design parameters

5.2. Balancing Ride Comfort and Idling Vibration

Idling vibration is the phenomenon where periodic excitation force corresponding to the engine idling speed is generated around the engine roll axis and transmitted to the vehicle body via the engine mounts and other parts.

Therefore, the basic principle for addressing this issue is to separate the frequency of the excitation force and the resonant natural frequency of engine mount system. However, lower idling speeds are demanded from the perspective of fuel efficiency, so if the mount stiffness is reduced to help separate the natural frequency from the excitation force frequency, ride comfort performance declines in the primary region.

For this reason, when addressing idling vibration, the level of vibration is reduced by the rotational elastic axis of the engine mount system closer to the engine center of gravity in order to align the rotational elastic axis and the torque roll axis.

In addition, with regard to ride comfort relative to vertical input from the road, it is known that the level of sprung-mass vibration can be reduced by adjusting the distribution and positioning of mount stiffness to couple the vertical mode and pitch mode⁽¹⁷⁾.

Figure 16 shows an example of evaluation of the vertical vibration transmission properties of the vehicle body when the tire contact patches of a compact car are excited in the vertical direction. Test and simulation results were compared for two specifications: the baseline specification and the specification where the engine side mounts use hydraulic mounts with the characteristics shown in Fig. 17. In Fig. 17, K_d represents storage stiffness and K_i represents loss stiffness. These spring constants change according to the frequency, and are plotted for each amplitude. By positioning the damping peak of the hydraulic mounts close to the resonant frequency of the power plant, the vertical vibration level of the vehicle body can be reduced while at the same time realizing the low spring characteristics needed to secure idling vibration performance.

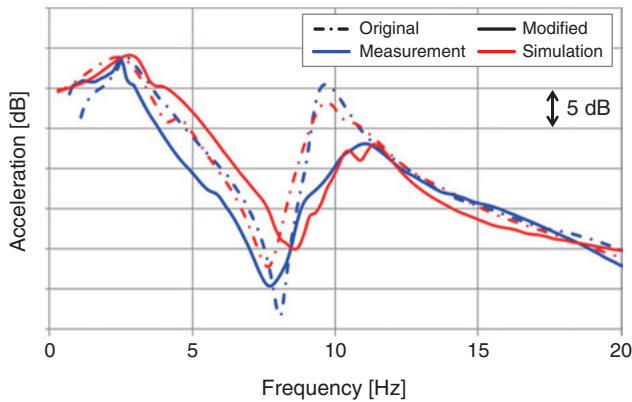


Fig. 16 Example of body vertical acceleration response in four-poster test

This demonstrates that the use of parts models enabling study of multiple functions makes it possible to accurately predict the performance of the complete vehicle in the initial phase of vehicle development, and to quantitatively evaluate the effects of using parts.

5.3. Harshness

Harshness is the phenomenon where transient vibration and noise are produced in the vehicle body when the vehicle passes over differences in level such as road seams or bridge joints. The backward inclination angle of the wheel center trajectory ϕ , the damper inclination angle θ , and the longitudinal stiffness of the suspension during the vertical suspension stroke are known to contribute to harshness performance as shown in Fig. 18^{(18), (19)}. When designing the suspension, changes in the virtual arm angle and the damper inclination angle need to take into account factors such as the positioning of the driver's heel point and the hip points of the rear-seat passengers, the balance with lowering of the floor, and changes in the vehicle posture

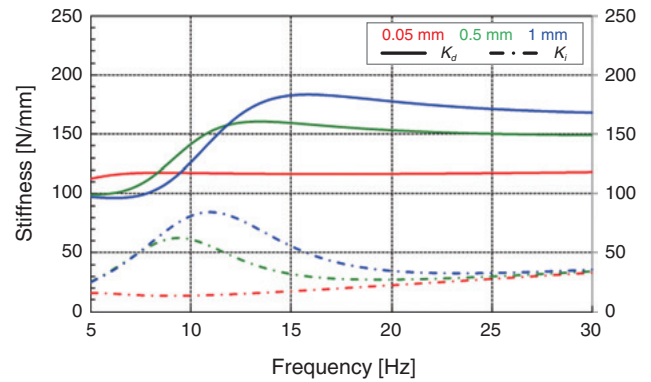


Fig. 17 Example of dynamic characteristics of hydromount bush model

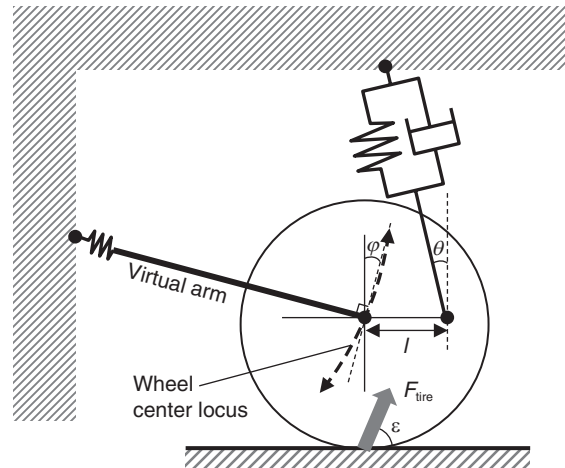


Fig. 18 Side view of conceptual suspension model with virtual arm and spring-damper

when accelerating and braking. Regarding longitudinal stiffness of the suspension, in order to achieve a balance with requirements related to handling performance, in many cases harshness performance is secured by reducing the dynamic multiplication or increasing the damping of the compliance bushings.

As an example, Fig. 19 compares the driving test and simulation results the rear suspension of a midsize vehicle. There is good correspondence between the calculated values and measured values, indicating that the developed calculation model makes it possible to judge the viability of, and performance prospects offered by, design proposals in consideration of the dynamic characteristics of the parts.

Figure 20 shows the results of a simulation study of the relationship between body input force and the design variables (damper inclination angle θ and virtual arm angle φ), based on the verified simulation model. The total body input force is reduced when the virtual arm is positioned so that the wheel center trajectory has a backward inclination and the damper inclination angle θ is negative (forward inclination).

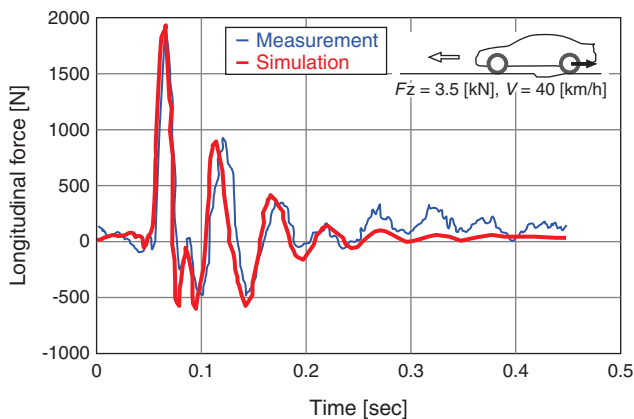


Fig. 19 Example of body longitudinal force ($V = 40$ km/h)

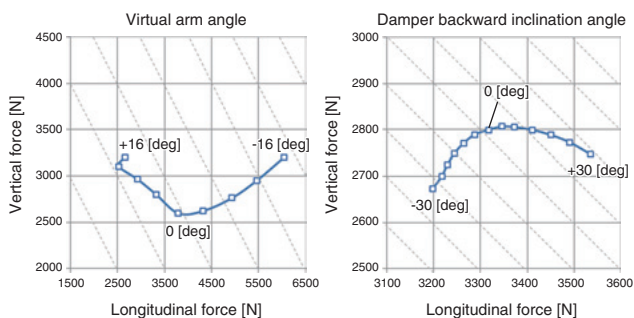


Fig. 20 Relationship between body input force, virtual arm angle, and damper inclination angle

6. Conclusion

This paper discussed the addition of model elements related to ride comfort to a design tool conventionally used to study handling performance in order to conduct studies aimed at achieving a balance between handling and ride comfort performance.

- (1) Specifications used in the models of bushings, dampers, and tires, which are parts that affect ride comfort performance, can be applied as is to study ride comfort performance in parallel with handling performance.
- (2) The developed tool enables simplified study of the rubber bushing, suspension damper, and engine mount characteristics needed to study ride comfort.
- (3) Comparison of the results of performance predictions using the developed tool with the results obtained in actual vehicles demonstrated that the developed tool possesses sufficient accuracy for initial studies, and can be used to study specifications that balance handling and ride comfort performance down to the level of detail of individual part specifications.

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Downsizing Technology for Chassis Parts Fastening Structure

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ABSTRACT

Downsizing technology for fastening structures was developed to realize a low-cost, next-generation, lightweight, high-performance suspension. The key technologies developed to achieve this were strength class 12.9 and 14.9 high-strength bolts that can also be applied to the vehicle body, and technology that controls the limit friction characteristic of the fastening joint surface. When the strength is increased with conventional thermal refining steel, the delayed fracture characteristic decreases, so a high-strength hexagon flange bolt was developed that uses patenting-treated pearlite steel to realize both increased strength and an enhanced delayed fracture characteristic. Technology for controlling the limit friction characteristic of the fastening joint surface made it possible to secure stable slipping resistance by optimization of micro unevenness, evaluated using developed accurate friction coefficient measurement technology, which helped to downsize the fastening structure. These two technologies were applied separately and in combination as appropriate to each fastening position in the 2016 model year CIVIC, and helped to achieve a lightweight and low-cost suspension.

1. Introduction

Needs are increasing to reduce automobile weights from the viewpoints of enhancing fuel economy performance and vehicle dynamics. In addition, as multi-link suspensions are being increasingly adopted even in C category vehicles in order to enhance product appeal, lightweight, high-performance suspensions are demanded.

The suspension is the link mechanism that joins the vehicle body with the tires and wheels, and the link mounts are typically fastened using bolts and nuts via rubber bushings. The number of bolt fastening positions increases when using a multi-link format, so realization of a lightweight and compact link mount-fastening structure would contribute greatly to reducing vehicle weights.

The link mount rubber bushing characteristics greatly affect the isolation of vibration from the road surface while driving, driving stability performance, and ride comfort performance. Specifically, a large spring ratio is demanded, with a low dynamic spring constant to isolate vibration, a high transverse direction spring constant to enhance driving stability, and a low torsional direction spring constant to enhance ride comfort. Reducing the rubber thickness in the transverse direction typically increases the transverse direction spring constant, which makes it possible to lower

the rubber hardness, and lowering the rubber hardness reduces the torsional direction spring constant. Use of a smaller bushing diameter enables increasing the spring ratio, but also tends to reduce the rubber volume, so durability is an issue. Therefore, this study focused on increasing the spring ratio by reducing the bushing inner diameter while maintaining the rubber volume, and investigated methods of downsizing the fastening structure bolts.

In addition, the suspension bolt fastening points are the base points when designing the surrounding part layout, so reducing the bolt size also increases the freedom of the surrounding part layout design.

Conventional approaches toward bolt downsizing typically focus only on increasing the bolt strength. For example, consider the case when downsizing the strength class 10.9 high-strength M14 bolt (point A in Fig. 1) typically used for chassis fastening to M12 size. When increasing the strength of a strength class 10.9 M12 bolt (point a in Fig. 1) to obtain an equivalent allowable axial force, a strength class 12.9 M12 bolt (point b in Fig. 1) has insufficient allowable axial force, so the strength needs to be increased to strength class 14.9 (point c in Fig. 1). In addition to enhancement of the reduced delayed fracture characteristic^{(1),(2)} described hereafter, there are also such issues as securing internal thread strength and fastened

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component flange surface strength and reducing the fastening torque, limiting the positions where downsizing can be achieved simply by increasing the bolt strength.

This study therefore focused on a bolt-fastening structure with bushings, and investigated downsizing methods that do not rely solely on increasing the bolt strength. Figure 2 shows a conceptual drawing of the load state generated on a fastening structure with bushings. The bolt size is designed so that the product of the bolt axial force F and the slipping limit friction coefficient μ of the fastening joint surface exceeds the maximum input force F_v applied to the fastening structure as shown in Eq. (1). Here, the slipping limit friction coefficient is defined as the ratio between the maximum input force at which slipping does not occur and the bolt axial force.

$$F_v < \mu F \quad (1)$$

Increasing the bolt strength is being considered to secure the needed axial force without increasing the bolt size, and enhancing the slipping limit friction coefficient μ of the joint surface is being considered to increase the allowable

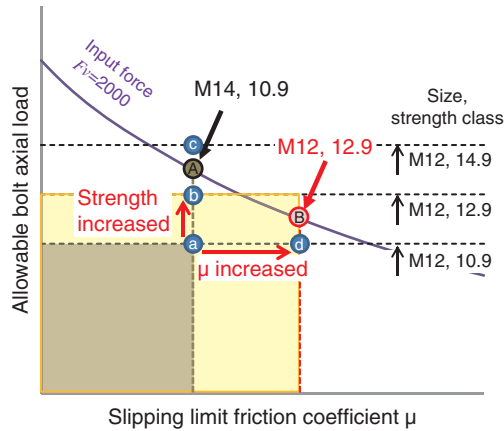


Fig. 1 Relationship between bolt axial load, slipping coefficient and bolt size

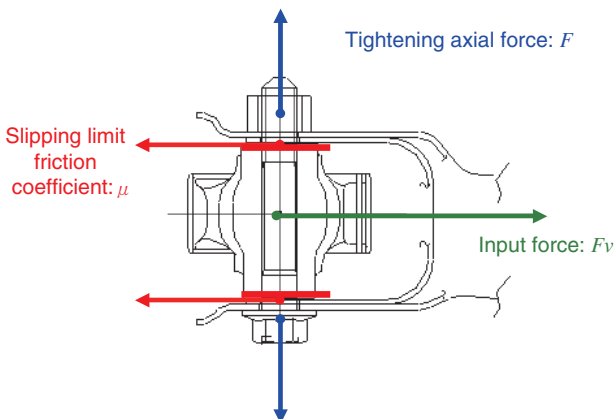


Fig. 2 Schematic illustration of bolt fastening structure

load⁽³⁾⁻⁽⁵⁾. Expressed in terms of Fig. 1, the bolt strength can be increased from point d in Fig. 1 by increasing μ , which makes it possible to withstand the maximum input force with the strength class 12.9 bolt axial force (point B in Fig. 1). This shows that these two approaches are useful for reducing the bolt size, and that the application range can be expanded and even greater weight reduction effects can be expected by applying these approaches separately or in combination as appropriate to each fastening position.

2. Development Issues and Goals

2.1. High-strength Bolts

When applying high-strength bolts, it is known that delayed fracture occurs more frequently as the material strength increases. In particular, the delayed fracture characteristic tends to decrease in the tensile strength range in excess of 1200 MPa. Delayed fracture occurs due to corrosion reaction during actual operation, and is thought to be due to occluded diffusible hydrogen. Bolts used in corrosive environments such as chassis parts have a particularly high need for an enhanced delayed fracture characteristic.

Materials that can be applied to high-strength bolts and enhance the delayed fracture characteristic are broadly classified into two types. The first is thermal refining steel wire that uses alloy addition and heat treatment. However, the material unit cost is high and securing stable heat treatment quality is an issue. The other type is steel that uses an isothermal transformation process called patenting treatment (hereafter, patenting steel), such as piano wire. Compared to thermal refining steel wire, patenting steel provides the same strength with a better delayed fracture characteristic, and Honda fastens the engine component using strength class 16.9 high-strength bolt materials⁽⁶⁾. However, patenting steel bolts have a high material strength, so as shown in Fig. 3, when the drawing processing rate before bolt forming is high, head forming is an issue for manufacturing hexagon flange bolts. Conversely, when the drawing processing rate is low, it is necessary to increase the shaft-forming processing rate during bolt forming, which increases the load applied to the heading die. Higher material strength during forming than that of thermal refining steel is needed to satisfy the strength for the drawing processing and subsequent plastic deformation, so achieving both strength and formability is an issue. Therefore, high-strength bolts that use patenting steel have so far been limited to only hexalobular internal types that have an easily formed bolt head shape. Hexalobular internal shape chassis fastening bolts have gained popularity in Europe and America, but in consideration of global deployment, there is a high need for hexagon flange bolts that can be tightened without changes to existing processes (Fig. 4). Bolt wire materials that do not use thermal refining have been developed, but the deformation resistance during heading is high, so application has been limited to

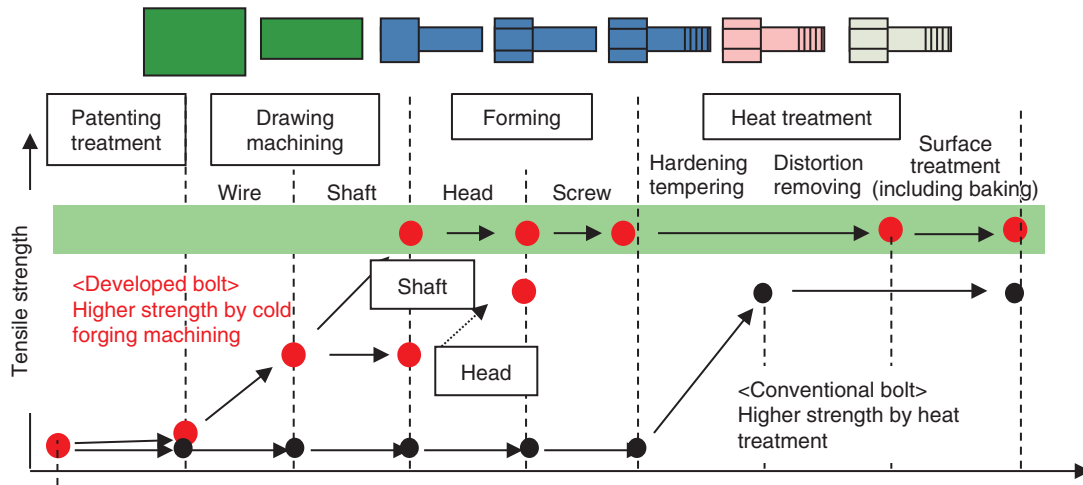


Fig. 3 Relationship between bolt manufacturing process and strength transition



(a) Hexalobular internal bolt

(b) Hexagon flange bolt

Fig. 4 Shape difference between hexalobular internal bolt and hexagon flange bolt

strength class 8.8 or lower from the viewpoints of securing formability and tool life⁽⁷⁾.

Development of high-strength bolts therefore focused on strength class 12.9 and 14.9 hexagon flange bolts that exhibit sufficient delayed fracture resistance in a corrosive environment that simulates the actual chassis part operating environment, and have thermal stability equivalent to that of conventional materials.

2.2. Control of the Limit Friction Characteristic of the Fastening Joint Surface

The limit friction characteristic of the fastening joint surface has been verified for various fastening combinations. However, there has been a lack of systematic organization, as there is no generally established method for measuring the slipping limit friction coefficient, and experimental values measured using different methods are often referenced. The reasons for this are thought to be that many environmental factors (test piece, jig, load application method, cleanliness, etc.) influence friction coefficient measurement, which leads to an increase in variance, and it is a challenge to define and determine the slipping start point. In particular, the bolt fastened structure with bushings used in the load input environment during actual vehicle driving has a complex structure and input

load configuration, so analysis of the fastening joint surface slipping behavior is a challenge.

As a result of these circumstances, consensus has not been achieved regarding the factors influencing enhancement of the slipping characteristic of the fastening joint surface or their effects. Therefore, when developing technology for controlling the friction characteristic of the fastening joint surface, the goal was to systematically evaluate the influence of the surface properties on the limit friction characteristic of the fastening joint surface, and to realize a low-cost method for achieving a high slipping limit friction coefficient.

3. Development of High-strength Bolts for Vehicle Body

3.1. Steel Type Selection

When selecting the material, highly versatile carbon steel wire for cold heading was investigated as the basic composition in consideration of cost, availability, and recyclability. As the main factor influencing the strength of carbon steel wire for cold heading is the carbon (C) content, the composition was selected in consideration of the influence on patenting treatment and subsequent drawing processing. Table 1 shows the material compositions set by this development for the strength class 12.9 and 14.9 bolts. The patenting treatment performed in this development aimed to reduce the C content in order to achieve an even and refined pearlite structure and provide a ferrite matrix with the goal of enhancing the balance between strength and ductility. Drawing processing was performed after patenting treatment to help secure strength by further refining the structure. As mentioned above, the drawing processing rate setting is the key to hexagon flange bolt forming. Figure 5 shows the results of setting a drawing processing rate that enables bolt forming. The optimal drawing processing rates were set for the strength class 12.9 and 14.9 bolts.

Table 1 Material composition

Bolt Strength class	Composition mass%				
	C	Si	Mn	P	S
12.9	0.54 -0.61	0.15 -0.35	0.60 -0.90	≤0.030	≤0.030
14.9	0.69 -0.76	0.15 -0.35	0.60 -0.90	≤0.030	≤0.030

3.2. Strain-removing Heat Treatment

Much dislocation accumulates in the ferrite layer of the bolts formed by this development due to strain introduced by drawing processing and heading processing. For this reason, when heading is simply performed the movable dislocation density is high, so the relaxation characteristic is low and a large amount of hydrogen is trapped by the accumulated dislocation, with the result that the delayed fracture characteristic decreases. Bluing treatment (strain-removing heat treatment) was then performed to enhance settling due to strain introduction during forming and to enhance the delayed fracture characteristic. Figure 6 shows the change in the material internal structure due to bluing treatment. When strain accumulates, dislocation also accumulates and electron beams are not easily transmitted, so these areas are shown by black contrast. Comparison of the metal structure before and after heat treatment at 350°C confirmed that ferrite layer dislocation is reduced and strain is released. However, when bluing treatment was performed at 450°C, spheroidization of the cementite and the start of decomposition of the pearlite structure were confirmed. This decomposition of the pearlite structure due to low-temperature heat treatment is thought to result from structural changes driven by the large amount of dislocation introduced by the strengthening process⁽⁸⁾. Figure 7 shows the results of verifying the relationship between the tensile

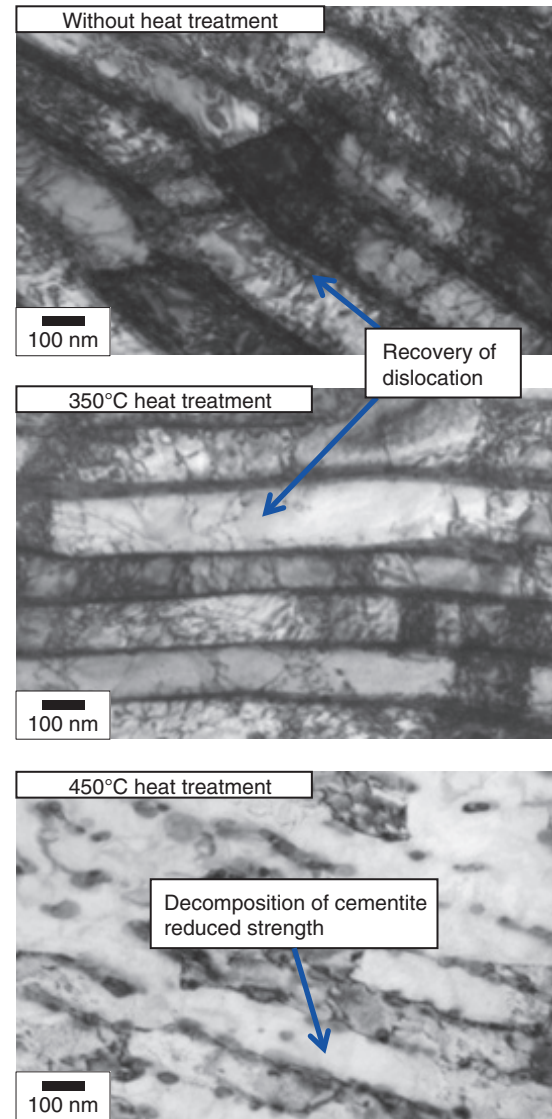


Fig. 6 Photographs of metal structure before and after heat treatment

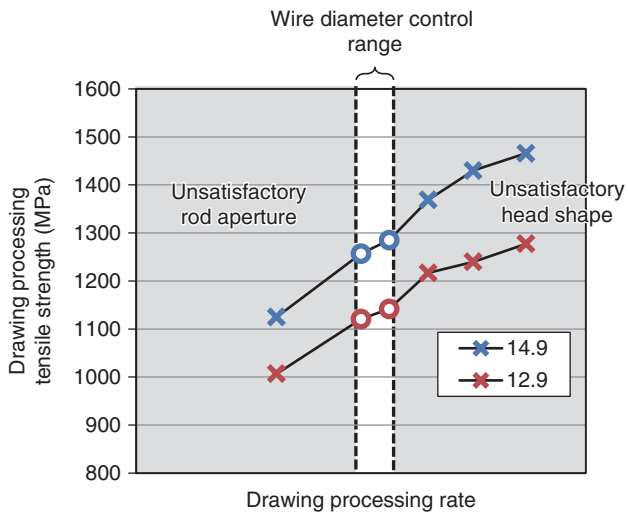


Fig. 5 Relationship between tensile strength and wire drawing rate

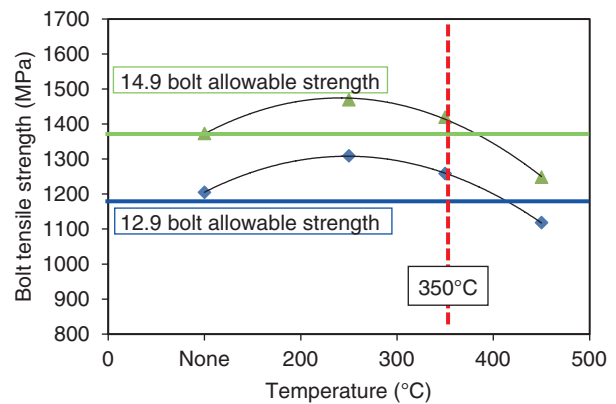


Fig. 7 Relationship between tensile strength and heat treatment temperature

strength and the bluing treatment temperature. These results show that the strength tends to drop rapidly at a temperature of 400°C or more. High thermal stability can be secured by performing bluing treatment at a temperature high enough that the strength does not fall below the target bolt strength class. Thus, this development set the bluing temperature to 350°C in consideration of quality variance during heat treatment.

The typical operating environment is assumed to be normal temperature or higher, so as it is necessary to secure thermal stability, the heat treatment stability enhancement effect of bluing treatment was verified. The developed bolts were tightened at room temperature, left for 100 h at 150°C, and the remaining fastening axial force was measured. The results confirmed that strain-removing heat-treated bolts had 20% higher remaining axial force and thermal stability equal to that of strength class 10.9 thermal refining bolts (Fig. 8).

3.3. Delayed Fracture Characteristic

Delayed fractures are produced by hydrogen that has entered the steel due to external factors such as corrosion or plating. The delayed fracture characteristic was verified by tests that apply tensile stress at a fixed load of 0.9 times the tensile strength in the condition with hydrogen forcibly input to the steel materials by the electrolytic hydrogen charging method, and then using the electrolytic hydrogen charging method to measure the critical hydrogen concentration at which fractures do not occur. Figure 9 shows the diffusible hydrogen curve when dehydrogenation treatment is performed on the developed bolt by heat treatment under various temperature conditions. Fractures do not occur for specifications with dehydrogenation treatment performed at 200°C, which indicates that the diffusible hydrogen content, which corresponds to the peak area up to 250°C, influences delayed fracture. It was therefore decided to perform subsequent evaluation of hydrogen content up to 250°C.

The delayed fracture resistance was quantitatively evaluated by measuring the critical hydrogen content (Hc) and the entered hydrogen content (He). Hc was measured

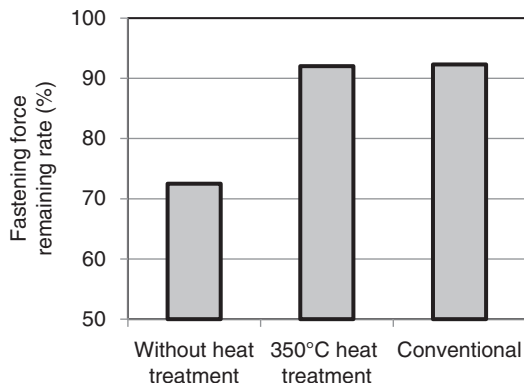


Fig. 8 Effect of strain-removing heat treatment

using the electrolytic hydrogen charging method. Here, an anode potential was generated on the steel material surface under various charging conditions to force hydrogen into the steel materials. He was calculated by having hydrogen enter the bolts in a rust endurance test simulating the actual vehicle environment, and then measuring the hydrogen content immediately after the test.

The delayed fracture resistance increases together with the Hc/He value. Figure 10 compares the Hc/He measurement results for the developed bolts with the values for conventional thermal refining bolts of the same strength. This shows that the developed strength class 12.9 and 14.9 bolts have better delayed fracture characteristics than the thermal refining bolts. In addition, the developed bolts have a sufficiently large critical hydrogen content relative to the hydrogen content that entered in the test simulating the actual vehicle rust environment, which confirmed that the developed bolts are high-strength bolts that can be applied to chassis parts.

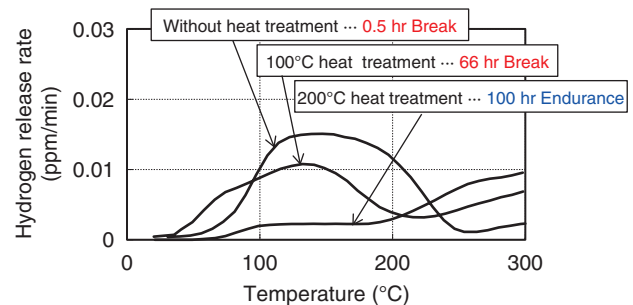


Fig. 9 Relationship between hydrogen release rate and temperature

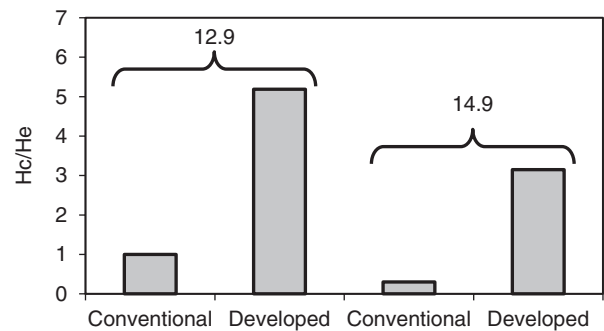


Fig. 10 Comparison of Hc/He between conventional and developed bolts

4. Realization of a Bolt Fastening Joint Surface with a High Slipping Limit Friction Coefficient

4.1. Establishment of Technology for Evaluating the Limit Friction Characteristic of the Joint Surface

This study attempted to establish a method that enables

stable evaluation of the slipping characteristic of the joint surface by developing and modifying a test apparatus with a simple structure that models the target fastening structure.

Figure 11 shows a schematic illustration of the measuring apparatus used by this development to evaluate the limit friction characteristic. This jig simulates the bush clamping structure, and has a structure that clamps a test piece B pipe part with two test pieces A. The test specimens were the typical fastening combination used in mass production, with cathodic electrodeposition-coated strength class 440 MPa steel plate selected for A, and SWCH10R⁽⁹⁾ selected for B. The prescribed axial force was applied beforehand to B by bolt fastening, and then external force simulating the actual vehicle input was statically applied at a rate of 1 mm/min in the direction perpendicular to the initial axial force. The limit friction characteristic of the joint surface was evaluated from the relationship between the external force and the relative displacement between A and B obtained at this time. Figure 12 shows the results obtained by this test. In addition to the relationship between

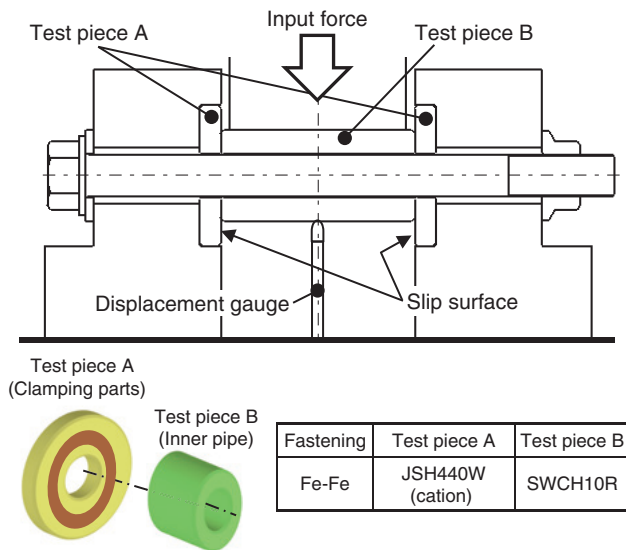


Fig. 11 Developed slipping test method and specimens

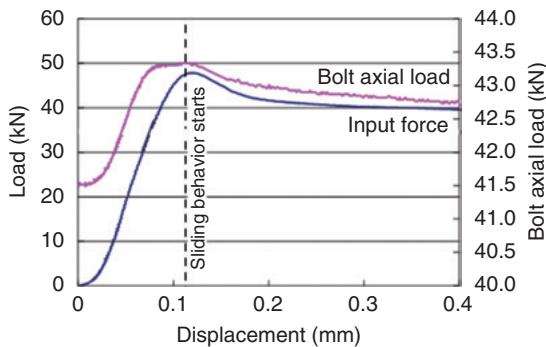


Fig. 12 Relationship between bolt load and displacement

the external force and the displacement, the bolt axial force also fluctuated during the test. The axial force fluctuation also exhibits an inflection point at the external force inflection point, and then gradually decreases thereafter. Therefore, the maximum axial force position was defined as the slipping start point, and the slipping limit friction coefficient was calculated. Note that the test method used in this development was modified in various ways to suppress test result variance. These modifications include stabilization of the shaft when applying external force, stabilization of the fastened component fixing position and fixing method, and stabilization of the displacement measurement position. The results of multiple tests confirmed a stable relationship between the external force and displacement as shown in Fig. 13.

4.2. Influence of Joint Surface Properties on Limit Friction Characteristic

Static verification of the limit friction characteristic with respect to different surface properties obtained by various control methods was performed to verify the influence of the surface properties on the limit friction characteristic of the joint surface. Table 2 shows the surface property control methods used by this development. Samples with various surface properties were created by varying the setting parameters for these methods. Figure 14 shows the relationship between the surface roughness (R_k) and the slipping limit friction coefficient of the joint surface. R_k is a parameter that indicates the level difference of the core part of the roughness curve excluding the peaks and valleys⁽¹⁰⁾. A slipping test was performed with $n = 3$, and the average values were plotted. The slipping limit friction coefficient was confirmed to increase together with R_k , and the friction coefficient tended to decrease in the range where R_k is 30 μm or more. While this suggests that increasing the surface roughness increases the friction coefficient, excessive surface roughening treatment reduces the friction coefficient.

When slipping occurs between contacting solids, shearing typically occurs at the junction of the real contact region. Labeling the junction shearing strength as S ,

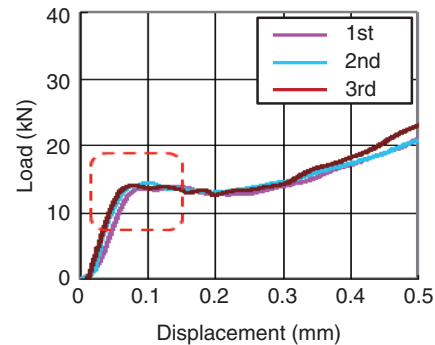


Fig. 13 Relationship between load and displacement for developed method

Table 2 Methods for controlling surface roughness

Sample	a	b1, b2	c1, c2	d1, d2, d3
Process	As forged	Forming	Wet blast	Air blast
Method	-	Form-rolling	Sprayed pressure (up to 5 MPa)	Sprayed air pressure

the friction force F_s due to shearing is expressed by the following equation.

$$F_s = ArS \quad (2)$$

Here, Ar is the real contact area. When Ar increases, the friction force due to shearing also increases, indicating a high friction coefficient. This is thought to suggest that appropriate surface roughening treatment increases the real contact area, while excessive surface roughening treatment reduces it.

In addition, two specifications (b1 and c1) with an R_k around 10 μm can be confirmed in Fig. 14 that have greatly differing slipping limit friction coefficients despite having nearly the same R_k . Figure 15 shows the results of

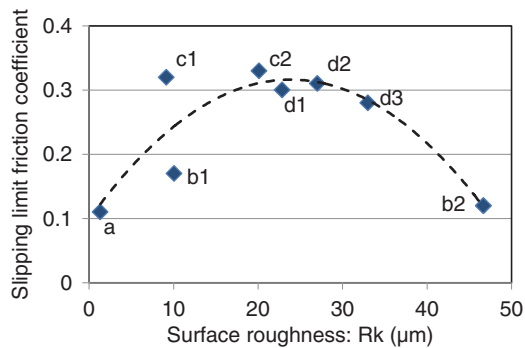
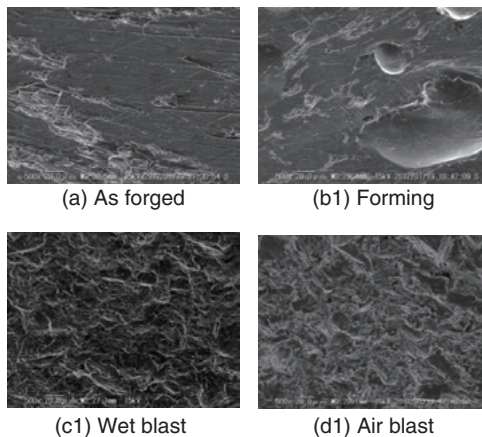
Fig. 14 Relationship between R_k and slipping limit friction coefficient

Fig. 15 SEM images of inner pipe surface after phosphate treatment

Table 3 Laser processing conditions

Frequency	Rated output	Peak output
30 Hz	30 W	20 kW

comparing the surface properties using SEM images of the micro-structure in order to verify the difference between these specifications. Figure 15 (b1) is the image after surface roughening treatment by heading, and shows coarse unevenness. In contrast, Fig. 15 (c1) confirms that micro unevenness is densely distributed by wet blast treatment. This comparison suggested that micro unevenness has a large influence on enhancement of the limit friction characteristic of the joint surface, so this influence was verified as described in the following section.

4.3. Clarification of Mechanism Contributing to Enhancement of Slipping Limit Friction Coefficient

The limit friction characteristic of different specifications created by applying surface unevenness by fiber laser treatment was evaluated to analyze in detail the influence of micro unevenness on the limit friction characteristic of the joint surface. The reason why fiber laser treatment was used here was that micro unevenness can be applied in a regular and stable manner by making arbitrary input parameter settings. Surface roughening treatment such as blast treatment applies coarse unevenness as well as micro unevenness, so it was thought that this would hinder verification focused only on the influence of micro unevenness. Table 3 shows the fiber laser processing conditions. The convex height was set to 1.25 μm . Figure 16 shows the slipping coefficient measurement results under the same conditions. The results in this figure confirmed that the limit friction characteristic at the chassis fastening axial force level tends to be higher than the setting at a pitch of 0.12 mm or less. This indicates that densely distributing the micro unevenness enhances the limit friction characteristic, even at a micro unevenness height of 1.25 μm .

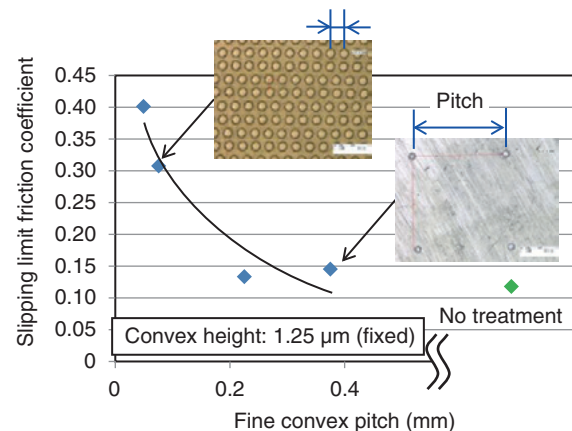


Fig. 16 Relationship between convex pitch and slipping limit friction coefficient

These results clarified that applying μm -order micro unevenness has a large influence on enhancement of the limit friction characteristic joint surface, and that dense distribution of this type of micro unevenness is an effective means of enhancing the limit friction characteristic, even for specifications controlled using simple methods such as shot blast or wet blast treatment.

4.4. Evaluation of Repeated Input Resistance

The repeat limit friction characteristic when 30 Hz repeated alternating input is applied was verified using a test apparatus similar to that used in the abovementioned static test in order to verify the limit friction characteristic under conditions approximating the actual vehicle input state. Slipping judgment consisted of confirming that apparent macro slipping did not occur and that the drop in fastening axial force was 10% or less after applying the repeated input 100,000 times. The critical load at which slipping does not occur at the set alternating input load was verified, and that critical load was converted into the repeat slipping limit friction coefficient.

Figure 17 shows the repeat slipping limit friction coefficients when repeated input conditions are applied. The developed specification with micro unevenness exhibits a high repeat slipping limit friction coefficient, albeit slightly lower than the static slipping limit friction coefficient evaluated in the previous sections. These results showed that the micro unevenness specification set by this development is useful for achieving a high limit friction characteristic even in an actual vehicle input environment.

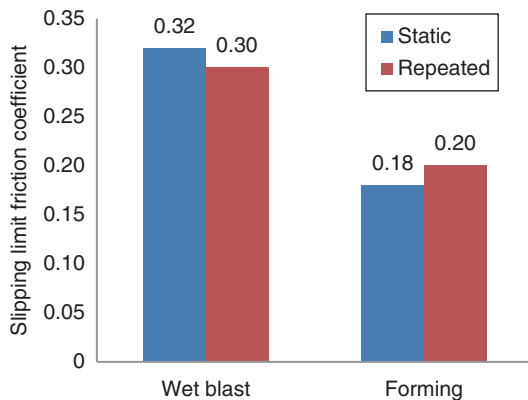


Fig. 17 Results of repeated force input test on Fe-Fe part with micro unevenness

5. Effects of Application

The high-strength bolts and technology for controlling the limit friction characteristic of the joint surface developed by this study were applied to multiple fastening positions of the 2016 model year CIVIC suspension parts. Figure 18 shows an example of positions where this technology

was applied. The developed technology was confirmed to achieve a weight reduction effect of approximately 0.05 kg per location, including the bolt, bushing, and surrounding parts. Figure 19 shows the weight reduction effects achieved by combining the two key technologies developed by this study. The weight reduction effect of technology to enhance the limit friction characteristic of the fastening joint surface is approximately 0.6 kg per vehicle. The combination of high-strength bolts and this key technology was confirmed to provide an additional weight reduction effect of 0.2 kg per vehicle, for a total weight reduction effect of 0.8 kg per vehicle. This indicates that even greater weight reduction effects can be obtained by using the technologies established by this development in combination. The weight reduction effects presented here are for the model used in this study, and similar compound effects can also be expected by applying the technologies separately or in combination as appropriate to other models. In addition, the downsizing effects achieved by this development not only enhanced vehicle fuel economy, but downsizing the bushing outer diameter by approximately 9% also helped to increase the parts layout freedom.

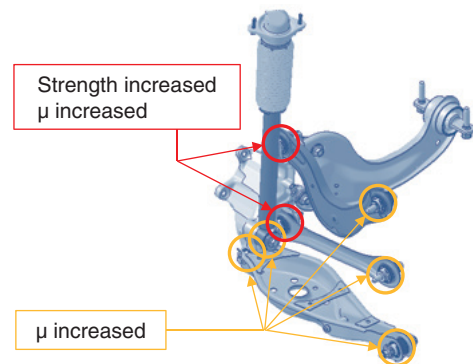


Fig. 18 Illustration of 2016 model year CIVIC rear suspension with weight reduction technology

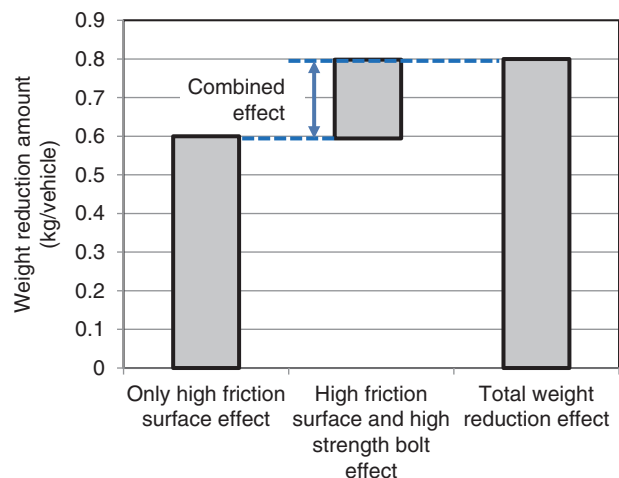


Fig. 19 Illustration of weight reduction effects for 2016 model year CIVIC rear suspension

6. Conclusion

High-strength bolts with a good delayed fracture characteristic and technology for enhancing the limit friction characteristic of the fastening joint surface were developed to downsize the fastening structure used to fasten chassis parts via bushings. In addition, it was shown that even greater downsizing effects can be obtained by using these two development technologies separately or in combination as appropriate. The obtained results are summarized below.

- (1) To achieve both strength and formability, the optimal drawing processing rate for the bolt wire material was set, and the C content corresponding to strength classes 12.9 and 14.9 was respectively set to 0.54 wt.%C-0.61 wt.%C and 0.69 wt.%C-0.76 wt.%C.
- (2) Bluing treatment was performed at 350°C to relieve the strain introduced in the drawing and bolt heading processes. This enhanced the relaxation characteristic and helped ensure the thermal stability needed for chassis parts. In addition, the delayed fracture characteristic was verified in consideration of the corrosive environment during actual operation, and the developed bolt was confirmed to have sufficient delayed fracture resistance.
- (3) Regarding the limit friction characteristic of the joint surface, μm -order micro unevenness was shown to be an effective means of enhancing the slipping limit friction coefficient. In addition, wet blast and shot blast treatments were confirmed to be effective methods for achieving this at low cost.

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Image Sensor Based V2V Optical Wireless Communication System

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ABSTRACT

In order for multiple autonomous vehicles to interact, they will need to identify one another according to their relative position and ID. With this objective, a camera-based system for vehicle-to-vehicle communication was developed to detect vehicle IDs and relative position by imaging optical signals loaded with an ID and blinking at high speed. Digital information is sent with modulated light that changes in phase, and phase changes are detected by lock-in pixel camera to receive the signals. The system also detects distance by shifting the relative sampling timing. The implementation can receive 640 bps signals simultaneously on seven channels and detect distances up to 600 m with an uncertainty of ± 5.6 m.

1. Introduction

With the advances in IT, various parties are currently working on implementation of autonomous vehicles. In order for them to move autonomously without colliding, such vehicles are equipped with multiple distance sensors, including stereo cameras, millimeter-wave radar, and laser rangefinders, to detect their surrounding environment.

In addition to the progress being made on standalone vehicles, vehicle-to-vehicle (V2V) communication is also evolving toward making traffic flow more smoothly and safely, allowing vehicles to alternate when merging into traffic, and to accelerate and decelerate without delay when following another vehicle. In Japan, the Ministry of Internal Affairs and Communications (MIC) is heading up research on systems utilizing V2V communication and roadside-to-vehicle (RTV) communication⁽¹⁾. Accordingly, systems using wireless communications and position information from the Global Navigation Satellite System (GNSS) are being researched (see Fig. 1).

These cooperative systems require communications that can sense relative positions accompanying individual identification by vehicle IDs.

Thus, stabilizing detection for relative positions during communications is an issue. The cooperative systems are built to constantly receive vehicle information for multiple vehicles within an approximately 300-m radius at a frequency of 10 Hz^{(2), (3)}. This vehicle information includes details such as vehicle position, speed, level of acceleration or deceleration, and traveling direction. In order to select vehicle information for a specific position among multiple vehicles, the system determines the information for a vehicle

in a specific position from the positions for all received vehicle information by comparing it with the system's own position. As the system uses GNSS to detect position, positions will be detected at reduced accuracy or not at all if the vehicle cannot be observed by multiple satellites due to the way GNSS detects positions. When downtown or in tunnels, this makes coordinated driving less feasible.

Distance sensors can be used to detect relative position, but have issues identifying individual vehicles. If the sensors could identify individual vehicles, they could obtain pre-registered vehicle addresses and communicate based on these addresses. However, distance sensors are for measuring relative positions up to approximately 100 m, and even if they could separate the detected objects based on their shape and movement by making detection points finer, individual vehicle identification is not possible without the ability to estimate license plates from the vehicle shape.

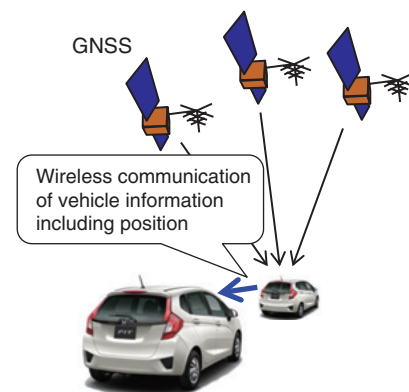


Fig. 1 V2V communication

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Thus, communication with a specific vehicle is not possible through any combination of distance sensors and mobile communication or near-field communication systems.

If a system were capable of both detecting relative position and identifying individuals, it could communicate with specific vehicles through next-generation technology like 5G and other various means of wireless communication. For example, if there were a system to read vehicle license plates and search addresses through the license plate numbers, it could communicate with any vehicle with readable license plates. By reading the license plate numbers from images, the system could also detect relative position. Trying to read numbers from long distances, however, requires a number reader with a telephoto camera as part of a complex system with anti-vibration and swivel mechanisms. Also, sensing vehicles in the neighboring lane would require license plate numbers mounted on the vehicle sides.

This paper therefore reports the evaluated performance of a system constructed using a camera to receive signals sent from transmitters with LED light sources (see Fig. 2), allowing it to simultaneously identify individual vehicles and detect their relative position.

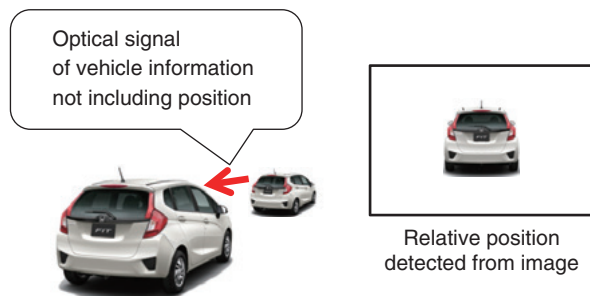


Fig. 2 Optical wireless communication camera

2. Prior Art and Issues

The camera-based methods of individual identification and position detection are each explained below.

In terms of individual identification, research has advanced on a method of camera-based optical wireless communication that changes the light source to decode lower resolution signals⁽⁴⁾⁻⁽⁸⁾. The system will be required to communicate simultaneously with the preceding vehicle and vehicles in the next lanes, and needs to be able to communicate with at least three vehicles simultaneously. It also needs to communicate with multiple light sources when detecting its own position from the light source position.

In terms of relative position detection, as stereoscopic views and other distance sensors can be used for short distances within approximately 100 m, the system uses a method that can detect relative position at longer distances. In terms of detection distances, the system requires 400 m for communicating with oncoming traffic when overtaking low-speed vehicles. As direction can be detected from

image positions if a camera is used, relative position can be detected if the distance is also detected simultaneously.

2.1. Optical Communication Method

Optical communication methods for receiving optical signals by camera (a two-dimensional light-receiving element) are roughly divided by camera types: general-purpose cameras, compound-eye cameras, direct readout cameras, and lock-in pixel cameras.

With general-purpose camera communication, the camera continuously captures optical signals, tracking the changes in light intensity sequentially by time and demodulating the signals⁽⁴⁾. As the cameras can normally capture images at 30 to 60 FPS, the time resolution for signals is roughly 16 to 33 ms. Thus, assuming three images are needed to decode a 1-bit signal, the signal rate will be 5 to 11 bits per second.

If the general-purpose color camera continuously captures optical signals, changes in light intensity can be tracked sequentially by time, and the signals then demodulated⁽⁵⁾. As the human eye is also sensitive to color changes, physical and psychological impacts will be the issue if multiple light sources are put in place.

There is a method of restoring signals from the striped scan line patterns generated when imaging a surface-emitting light source of varying intensity using a general-purpose rolling-shutter camera, as often seen in smartphones⁽⁶⁾. The light-emitting source needs to be of a certain length and perpendicular to the scan lines. This places size limitations on the light source, which is an issue for communications at a distance.

With compound-eye camera communication, elements capable of receiving optical signals are arranged two-dimensionally⁽⁷⁾. In this camera, a lens is placed in front of a two-dimensionally arranged light-receiving section of an infrared remote controller. This form of optical communication would be high speed as it can be used without alteration, but it is difficult to increase resolution.

With direct readout camera communication, the current generated by light incident to pixels is continuously retrieved at an external source, allowing conventional optical signal decoding technology to be applied without alteration⁽⁸⁾. This method requires a special imaging element. As the electrons generated by photoelectric conversion is continuously being read out as a current, sensitivity is low, and it is not easy to communicate at distance with a wide-field camera. Furthermore, communicating with multiple light sources is difficult.

With lock-in pixel camera communication, the camera detects the amplitude and phase of modulated light by synchronizing with the modulated light frequency, then storing the electrons generated by a photoelectric converter in different charge accumulators⁽⁹⁾. In this research, this was the method adopted to decode a phase-altered signal. This method can detect modulated light at frequencies exceeding camera frame rates. With it, frequencies can be set high and only the phase has to be changed, so the human

eye will not perceive any blinking. In addition, as it uses modulated light, candidate signal sources can easily be narrowed down to within a single frame. Also, as electric charge is integrated before detection, sensitivity is higher than with the direct readout method. While the transmission speed is slower than with direct readout, speed can be increased system-wide through use in conjunction with high-speed wireless technology, such as 5G, thus giving it the sensitivity to be usable at long distances. This explains our selection of this method.

2.2. Distance Detection Method

Camera-based methods for detecting distance are broadly classified into monoscopic vision, stereo vision, and Time of Flight (TOF).

In the monoscopic vision method, distance is detected from the location of features on the image of a subject whose position has already been determined⁽¹⁰⁾. Accuracy is good if multiple features are detected with sufficient distance between them.

Stereo vision detects distance based on the location of a single subject in images taken from multiple points of view⁽¹¹⁾. A wider camera separation or higher resolution is required to maintain measurement accuracy at longer distances.

TOF detects distance based on the time light requires to move through space. Here, the difference between when the light departs and when it enters the camera is multiplied by the speed of light to convert to distance. TOF was selected for use in this research as subject size is not a factor and distance can be detected up to a few hundred meters.

TOF is further subdivided into lock-in pixel camera TOF and draining-only modulation camera TOF.

The lock-in pixel camera method uses a pixelated camera with capacitor sections to store multiple electric charges and a photoelectric conversion section to convert light into electric charge. Here, the generated charges are distributed to the capacitor sections at differed timings. Distance is then detected by calculating the time required for the light

to move through space from the ratios of the number of charges stored in the multiple capacitor sections⁽¹²⁾. In order to detect the correct distances, the characteristics of each capacitor section have to be calibrated according to the reflected light intensity, ambient light intensity, and temperature. Calibrating for three parameters is not easy.

A draining-only modulation camera⁽¹³⁾ detects distance by repeatedly imaging for exposure periods of 100 ns or less. The photoelectrons can be discarded at arbitrary timing using the drain function. These cameras can detect the distance similar to the lock-in pixel camera by changing timing with multiple pixels. Compared to the lock-in pixel camera, sensitivity at the same area is reduced due to the discarded loads. By changing the timing in chronological order and imaging multiple times, this method obtains the same results as the lock-in pixel camera without changing the timing between pixels. As light source intensity is higher imaging direct light than it is in reflected light, allowing detection at lower sensitivity, the draining-only system was used.

In summation, the system adopted for this research uses three capacitor sections for phase detection, leaving the remaining capacitor section available to detect distance.

3. Fundamentals of the System

This system consists of a light projector to project modulated light carrying a signal, camera-based light receivers to decode the signals from the modulated light and detect the TOF pulses for distance detection, a modulator to send the signals to the light receivers, and a demodulator and oscillator to decode the signals (see Fig. 3). With frequency stabilized through GNSS, the oscillator enhances distance detection performance and, if times are synchronized, decreases the delay times in detection. When GNSS data is not available, the system is still capable of detection, however with longer detection times.

3.1. Pixel Functions

The pixel circuit for a single pixel is constructed as

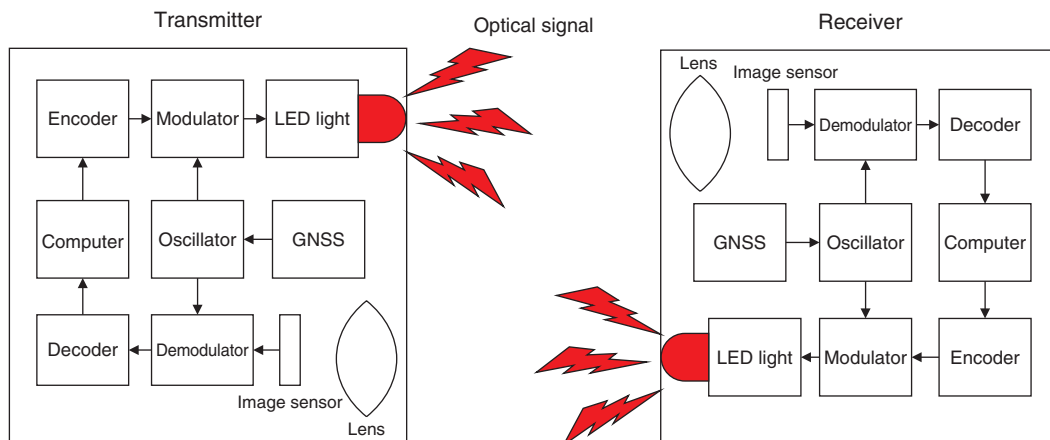


Fig. 3 System configuration

shown in Fig. 4. The photoelectric conversion section converts light into electric charge. The ends of the transfer gates are connected to the capacitor sections, and the generated charges are stored (integrated) each time a transfer gate opens. The end of the drain gate is connected to the power source and purges the generated charges. Charges stored in the capacitor sections are converted to current by the readout circuit and read out of the pixel.

Three out of the four capacitor sections are used for communication, with the remaining one used for detecting distance. Transfer gates TX1, TX2, and TX3 are sequentially opened at staggered timings, storing charges a total of eight times in their respective capacitor sections. The transfer gate cycle for gates TX1, TX2, and TX3 includes a brief period when gate TX4 is opened, during which the other transfer gates are closed (see Fig. 5).

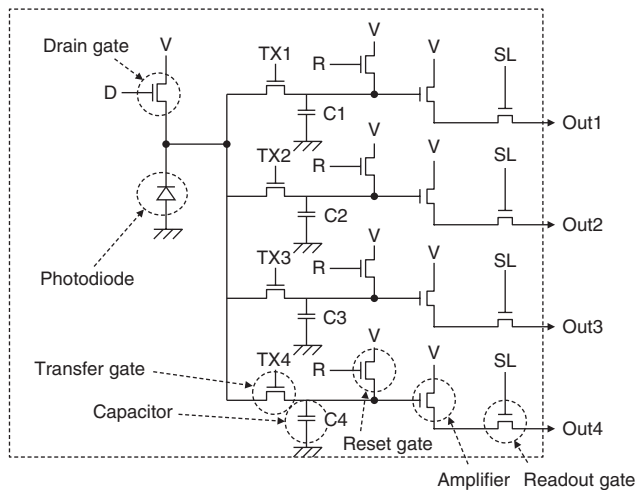


Fig. 4 Pixel circuit

3.2. Transmission Waveform

The signal pulse of 36- μ s cycles (27.8 kHz) as used for communication is taken as the basic waveform SP for transmission (see Fig. 6). This pulse reiterates a consecutive 24 times, then rests for eight empty cycles. This 32-cycle period is taken as one bit period, called a signal block (SB). When the signal pulse phase is changed from the previous block, the change is expressed in binary with a 0 or 1. When 0, the phase is advanced by $2/3\pi$ (120°); when 1, it is retarded by $2/3\pi$ (see Fig. 8). Thus, as long as the phase stays within the range of $\pm 60^\circ$, noise will not result in an error. The start of the signal sequence is made leader by continuing at the same phase. Subsequently, sections with changing phases are expressed in binary, ending with a period with no signals. These pulse sequences of 84 pulses with two leaders and two periods of no signals comprise a single signal sequence. In order to fill the sequence cycle out into the 100 ms⁽²⁾ required for V2V communications, an additional adjustment period is inserted after the periods of no signals. Thus, the system can receive 10 IDs per second.

The 36- μ s basic waveform includes the 200-ns pulse signal sequence used for detecting distances. The timing for the start of the pulse sequences shifts by 25 ns for each signal block, bringing it back to the original position every 80 iterations.

3.3. Optical Signal Detection Method

The phase and amplitude for the pulsed light are derived from the values sampled for one cycle of modulated light in the three capacitor sections (see Fig. 7). As levels are insufficient after just one sampling, however, eight samples are taken for each cycle to calculate the phase and amplitude as the integrated value of the signal of the same timing.

Phase changes are detected and decoded to binary for the binary signals (see Fig. 8). Phase is assigned in $2/3\pi$ increments to limit error across three samples.

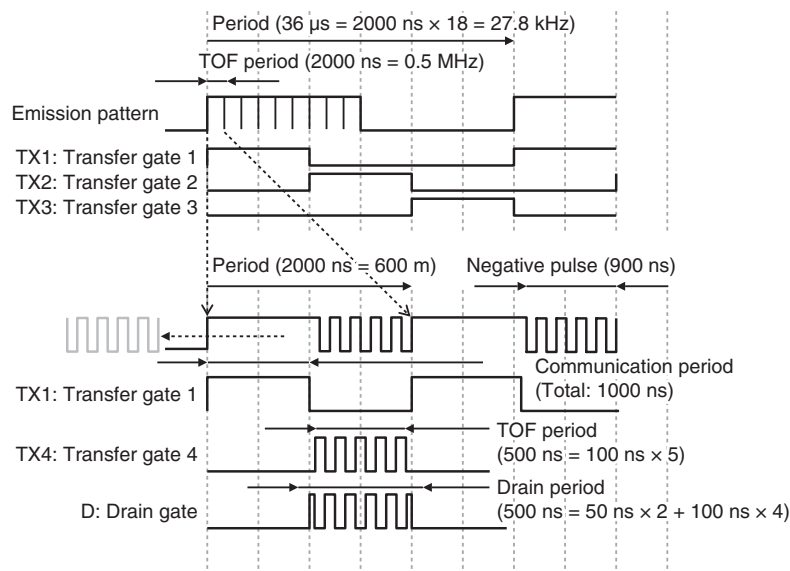


Fig. 5 Transfer gate timing

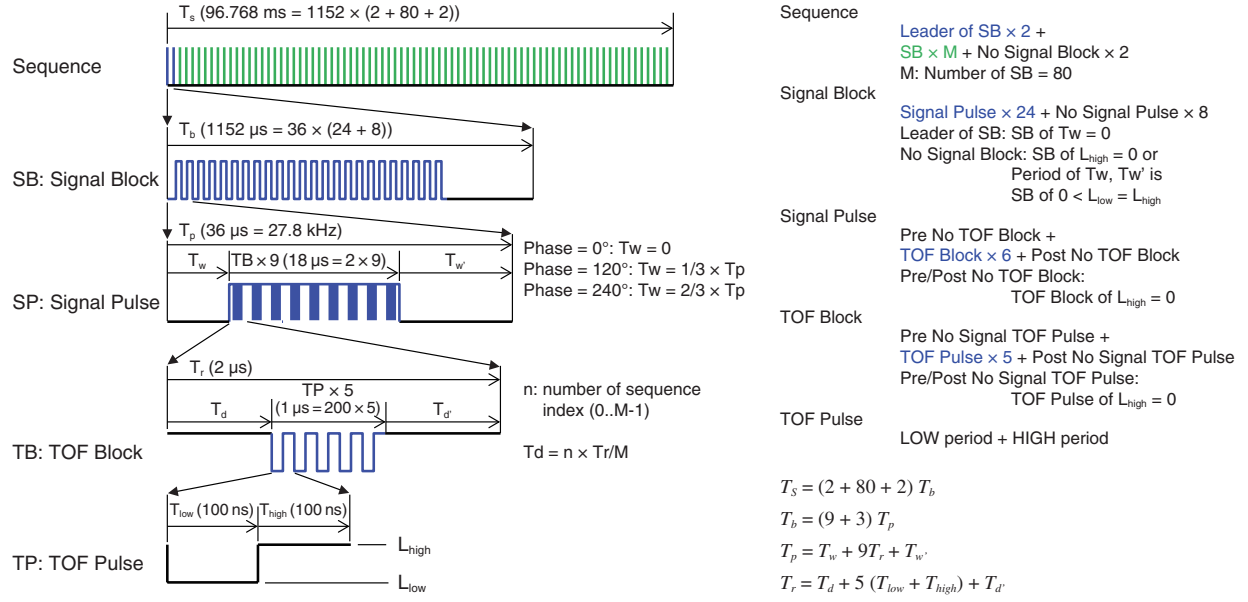


Fig. 6 Wave configuration

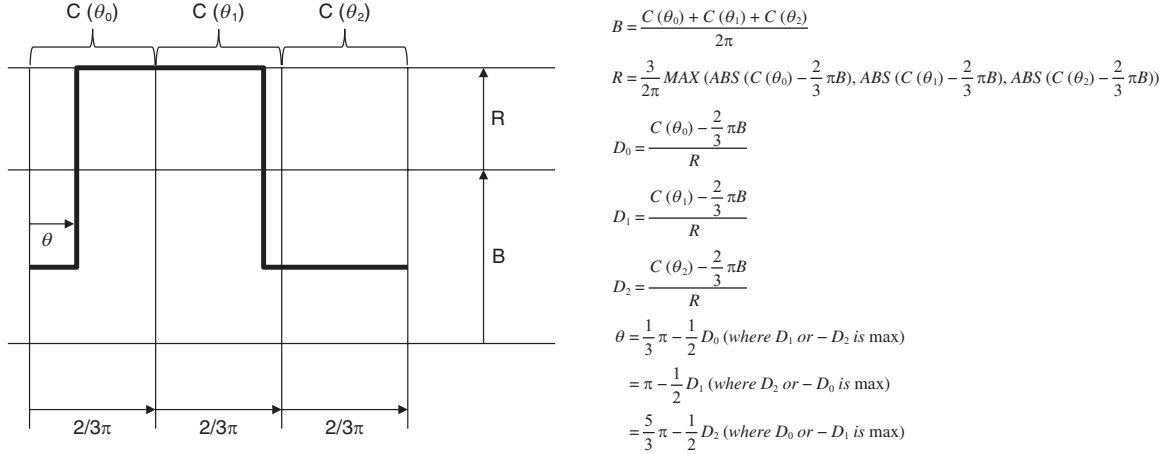
 Detecting phase θ of three integrated values obtained by shifting integration period


Fig. 7 Phase calculation

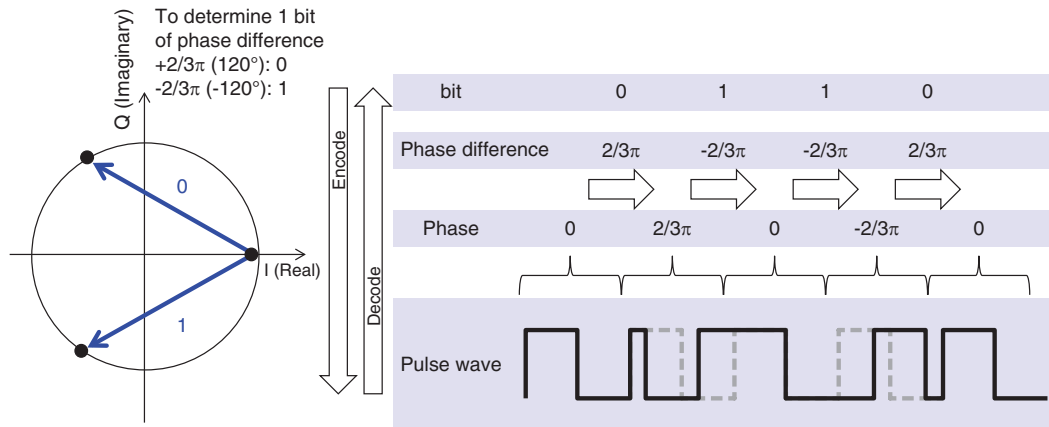


Fig. 8 Encoding and decoding process

3.4. Distance Detection Method

In order to permit measurement of longer distances, the distance detection method involves systems communicating with each other via light-emitting sources, similar to secondary surveillance radar⁽¹⁴⁾ as used in aircraft (see Fig. 9). When detecting distance with reflected light, light intensity decreases in inverse proportion to distance to the fourth power when the size of the light source as captured on the image is one pixel or smaller. With direct light, however, intensity decreases in inverse proportion to the distance squared, giving it an advantage

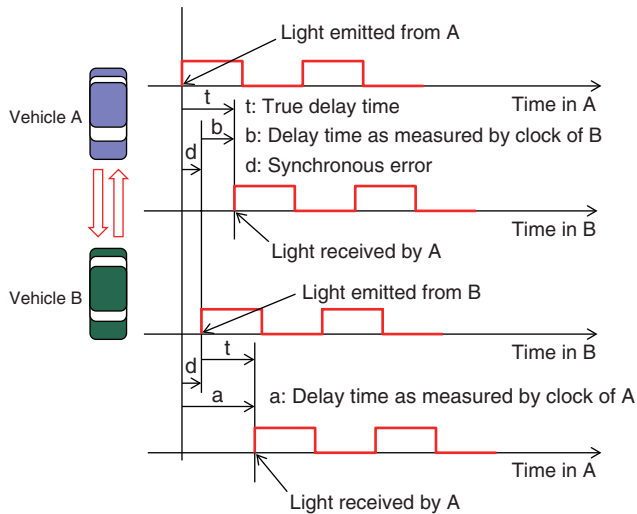


Fig. 9 Distance calculation

at longer distances. Unlike secondary surveillance radar, signals cannot be sent back after a set delay period from when the signal was received as the signals are sampled discretely. Therefore, each system sends the other signals continuously and independently, communicating to one another the delay times as they are detected in order to cancel out synchronization error and calculate the true delay time. Taking b as the delay time measured by the clock of Vehicle B when Vehicle B receives the pulsed light emitted by Vehicle A, a as the delay time when Vehicle A receives the pulsed light emitted by Vehicle B, and d as the synchronization error between the clocks of the two vehicles, the true delay time t can be solved as:

$$t = b + d$$

$$t = a - d$$

Canceling out the unknown d , t can thus be calculated as:

$$t = (a + b)/2$$

In measuring delay time for the pulsed light, the relative sample timings will gradually change, and delay time is calculated from the degree of coincidence. Gate output is at its lowest when the five gate timings coincide with five negative pulses, then increases with each shift in timing. Thus, the system finds the timing with the lowest output and sets it as the delay time (see Fig. 10). With single pulses, a short pulse increases accuracy in estimating peak position but reduces detection sensitivity, so pulses are sent in repeated bursts in order to increase sensitivity and detection accuracy. In addition, negative pulses are united with the

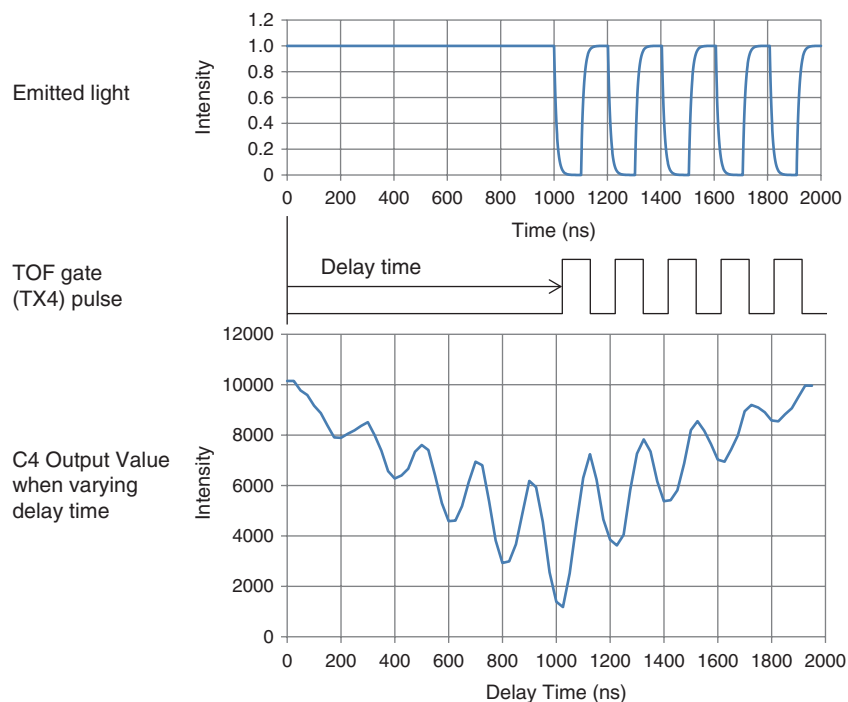


Fig. 10 Delay time detection

communication pulses to allow distance to be detected during communication without separating communication and distance measurements.

4. System Implementation

4.1. System Configuration

Each side of the system will consist of a communication distance camera, lens, projector, arbitrary waveform generator, GNSS oscillator, and calculator (see Fig. 11).

The GNSS oscillator outputs a 10-Hz pulse signal and 1-PPS pulse signal at synchronized times.

The arbitrary waveform generator outputs a clock signal used by a camera synchronized to 10 MHz and a communication distance waveform signal projected by the projector.

The communication distance camera consists of an imaging element, control board, and interface board. The imaging element sends image signals to a computer via the control board and interface board.

The projector emits LEDs as a communication distance optical signal, driven by light pulse signals input from the arbitrary waveform generator.

4.2. Pixel Structure

The same imaging element is used as in the differential camera⁽¹⁵⁾. It uses square pixels with 12- μm sides for an effective image size of 640 pixels horizontally and 480 vertically.

4.3. Region of Interest (ROI)

Image readout is broken into small regions to speed up the process. Up to eight small regions can be specified.

Regions can be subdivided into any number of detection regions and tracking regions. For this study, detection regions were set to a size of 64×24 pixels, scanning for regions of modulated light from top left to bottom right. When a region of modulated light is detected, the light is decoded into seven tracking regions of 8×8 pixels each. As the detection regions scan for modulated light at 27.8 kHz, no continuously bright sunlight or low-frequency blinking lights will qualify as candidate light sources.

4.4. Projector

For the projector, 10 near-infrared LEDs outputting approximately 1 W each were arranged in series across spaces of 940 nm and 860 nm. The half-value angle is approximately $\pm 35^\circ$, with no diffusion plate used. V2V communication is expected to use a 940-nm array, and RTV communication is expected to use an 860-nm array. As light would be visible in red at 860 nm and visible red light is prohibited by law, communication with visible light at 860 nm would be unsuitable. Meanwhile, 940-nm light results in a less sensitive imaging element, but is not visible. Thus, a 940-nm wavelength was selected for the on-board LED.

5. System Performance

5.1. Communication Performance

The system was operated with a 64-bit data component and 12-bit error correction with 4 bits of additional information. One sequence was 80-bit with no error correction. As 10 sequences per second can be received simultaneously across seven channels, the channels were 640 bps each for a total band of 4,480 bps. Only the ID was

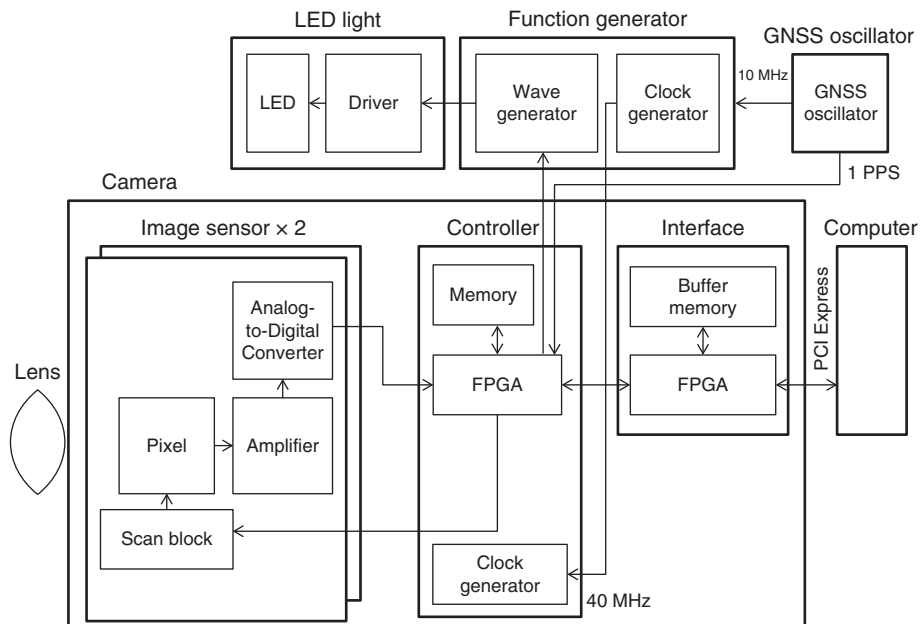


Fig. 11 Block diagram

listed in the data component. Depending on application, the data component will be loaded with a number indicating the data transmission order and vehicle speed information.

To measure communication performance, timing for a pulse generated by the arbitrary waveform generator was shifted over one cycle with the camera and projector set facing each other. Performance was then evaluated by shifting the detected pixel position.

Standard uncertainty for phase detection values was confirmed to be well within the $\pm 60^\circ$ range of binary detection at 5.4° or less within 1/8 intensity from the reference intensity.

A lens with a 56.5° horizontal angle of view and infrared transmission filter results in a maximum communication distance of 118 m outside in sunlight. Light at wavelengths of 800 nm and above can pass through the infrared transmission filter. Maximum communication distance is estimated to be expandable to 300 m or more just by narrowing the upper and lower irradiation angles with a lens or reflective mirror.

5.2. Distance Detection Performance

As with communication performance, distance detection performance was evaluated by changing the delay time and converting to distance.

Standard uncertainty for distance detection was 5.6 m or less within 1/8 intensity from the reference intensity. Distance was detectable within a range of 600 m. Variation due to temperature change is within ± 0.3 m at a temperature range of 5°C to 35°C . Calibration is unnecessary within this temperature range and likely even outside this range.

5.3. Tracking Performance

To evaluate performance in tracking a moving light source, the camera was loaded onto a rotating stage to evaluate performance in tracking a changing angular velocity.

When decoding signals in a small region with ROI, the light source was trackable at moving speeds of 480 pixels/s

and lower. For example, using a lens with a 60° horizontal angle of view, the camera could track a light source moving at an angular velocity of 68° per second. Given that the angular velocity of a vehicle traveling at a relative speed of 50 kph 1 second before passing is 23° per second, 68° is sufficient.

6. Sample Applications of the Communication Camera

As it will likely take some time for intercommunicating V2V and RTV communication systems to attain prevalence, an assumed application is automatic parking at homes, simply installing optical beacons locally as a positioning method.

Using optical beacons at known locations installed at seven points on interior walls, the system was tested for ability to calculate its own position (see Fig. 12). Using the optical beacons, the system was usable in both day and night settings.

Mounting the camera with a fisheye lens⁽¹⁶⁾ calibrated for a 180° horizontal angle of view, the camera detected position and posture of two or more light sources by detecting distance to the light sources from an elevated position in view of the light sources. As the camera was faced horizontally in anticipation of being mounted on vehicles, it detects position with just two or three light sources visible ahead. If the cameras are restricted to in-cabin mounting, the number of light sources can be reduced by pointing the camera toward the roof.

Standard uncertainty for position was 0.10 m, and that for angle was 0.64° . The camera requires 200 ms to determine position, at which point it can detect position at 250 Hz as long as it can track the light source. In terms of accuracy, the camera detects position as accurate as with existing methods⁽¹⁷⁾, while angle accuracy is an order of magnitude higher, and detection time was 1/5 that of existing methods. This demonstrates the method to be superior to existing methods.

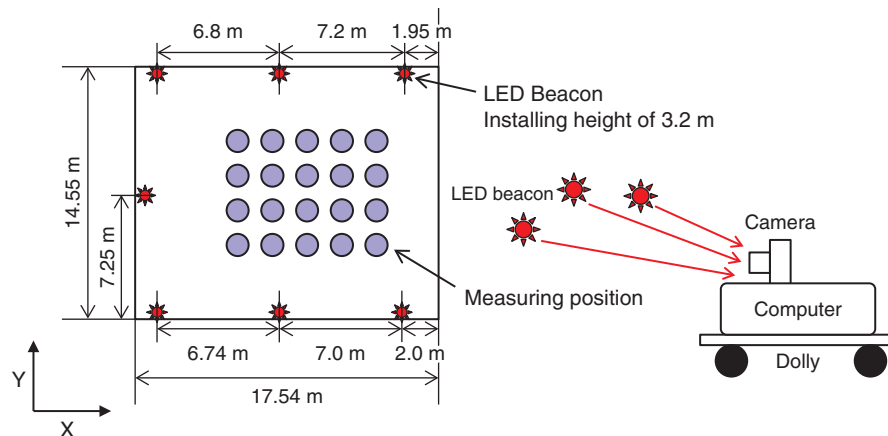


Fig. 12 Position detection

7. Conclusion

In this paper, a communication system receiving an optical signal sent by LED-based light source with a camera, assuming a vehicle-based application, was proposed. By imaging a pulsed light with superimposed digital signals using a lock-in pixel camera, the following performance was achieved:

- (1) Could receive seven optical signals at 640 bps simultaneously
- (2) Could track at a maximum angular velocity of 480 pixels/s
- (3) Could detect distances of 0 to 600 m with an uncertainty of 5.6 m

Additionally, with an assumed application of parking assist, a positioning system using the communication camera was constructed. The following performance was confirmed within a range of approximately 15 m:

- (4) Standard positioning uncertainty of 0.10 m
- (5) Standard angular uncertainty of 0.64°

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Vehicle Performance Design System Using Data Mining Techniques towards Model-based Development

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ABSTRACT

A vehicle performance design system has been developed that makes it possible to assign target values for different performance demands at the initial stage of the vehicle development process. Based on the concept of model-based development, this system determines appropriate target values through the application of data mining techniques.

The system uses the data available at the time of vehicle development, and comprises the following three techniques:

- (1) A response surface generation technique that helps enable high-accuracy predictions based on small amounts of data;
- (2) A decision-making technique that helps enable high-efficiency data searches that reflect the user's intentions; and
- (3) An impact analysis technique that makes it possible to identify specifications that have the great effect on performance from the relationships between performance and specifications.

It was demonstrated that the application of this system to vehicle dynamic performance simulations enables determination of target performance values and specification values simply and at high speed.

1. Introduction

Today, the automotive design process is expected to satisfy a range of demands proceeding from consideration of factors such as environmental and safety performance, and the process of development of the various elements necessary to the realization of target performance has become large-scale and complex. At the same time, the shortening of the development period and the reduction of cost continue to be important considerations.

A global survey of the automotive market shows that the market expands at a rate of 4% per year, and that factors such as the expansion of markets in emerging nations are changing the sales structure in regional markets. In order to respond to this situation, it is necessary to expand vehicle lineups, leading to modification of the flow of development throughout the automotive industry. The area of digital development is particularly active, and the importance of model-based development (MBD), involving virtual development based on input including simulation and test data and entailing as little creation of actual prototypes as possible, is increasing.

The research discussed in this paper sought to develop a vehicle performance design system that would help to realize MBD in order to increase development efficiency by making it possible to identify multiple target performance factors such as fuel efficiency and acceleration performance, and derive the related specifications, at the initial stage of development.

2. System Configuration

The research discussed here created a vehicle performance design system that will contribute to the development of next-generation vehicles using data mining techniques, realizing a style of development in which optimal target values are set with consideration of the tradeoff between multiple areas of performance.

The developed system (referred to simply as "the system" below) uses data-driven simulations employing a database (DB)^{(1), (2)}; accuracy and responsiveness are therefore important factors. Figure 1 shows an overview of the developed vehicle performance design system.

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As Fig. 1 shows, the system comprises a computer-aided engineering (CAE) simulation section, which simulates vehicle performance, a target performance display section incorporating the DB, and three data mining technologies. The flow of system procedures is as follows.

- (1) The specifications that form the input for the system are sampled using a design of experiments method.
- (2) Training data is formulated by conducting CAE calculations.
- (3) Using the training data, response surfaces and DB are created using machine learning technology.
- (4) Using the DB, optimal target values are determined with consideration of the tradeoff between multiple areas of performance.

The DB is made up of specification values used as input for CAE and performance values obtained from simulation results. Table 1 shows an example of a DB array consisting of five columns of specification values on the left and three columns of performance values on the right. The vehicle performance design system can be operated by using this DB. (1) to (3) below provide an overview of the data mining techniques employed to actualize the vehicle performance design system.

(1) Generation of response surfaces

In order to design target performance, it is necessary for the system to create DB containing large-scale data from the results of CAE calculations.

Figure 2 compares the conventional method and the response surface approximation method in generating data for 100,000 cases and creating a DB, showing the differences in calculation between the conventional method and the proposed method. The use of conventional CAE calculations necessitates an impractical calculation time for the creation of the DB. It was therefore necessary to develop a method of creating DB in the shortest time possible. Using a limited number of CAE calculation results (around 200 cases as training data), the proposed method employs machine learning techniques in order to generate a response surface model (RSM). The use of this RSM to calculate data for 100,000 cases enables a major reduction

Table 1 Database matrix

	Factor					Performance		
	1	2	3	4	5	1	2	3
1	460.91	2026.05	0.42	1418.89	2533.81	6.15	136	4.34
2	448.78	1775.30	0.42	1368.02	2511.51	6.19	137	4.41
3	459.69	2204.39	0.44	1417.41	2500.43	4.81	135	4.26
4	439.90	1924.32	0.44	1346.49	2487.21	5.13	137	4.37
5	458.57	1850.84	0.42	1348.45	2527.38	5.07	136	4.35
:	:	:	:	:	:	:	:	:
200	450.60	1745.68	0.44	1409.86	2590.14	5.38	138	4.39

in calculation time compared to the conventional method. Further details will be provided in Section 2.1.

(2) Decision making (Interactive data selection)

Determining the optimum solution that reflects the user's intentions from the 100,000-case DB is not a simple matter. The system therefore employs the following two data selection methods:

- (a) A direct search technique that searches for specification values directly from performance values; and
- (b) A group search technique that makes it possible to interactively narrow down data while surveying trends in the data using a clustering technique.

Users are able to select and implement the decision-making technique that suits their application from these two search techniques. Further details will be provided in Section 2.2.

(3) Impact analysis (Analysis of impact factor)

Impact analysis is an analytic method that makes it possible to determine the characteristic relations between performance and specification values whether the data is linear or nonlinear or the data environment features multiple specification values.

When nonlinear relationships between variables straddle multiple specifications, visualization of the relationships is challenging. The method visualizes relationships using partial dependence plots⁽³⁾, which are able to display characteristics even for multiple specifications. The degree of effect of each specification value on performance is calculated as impact factor, which is easily understood intuitively, and displayed on a graph. Further details will be provided in Section 2.3.

2.1. Generation of Response Surfaces

A response surface is a mathematical model that correlates a performance value, y , representing system output data, with a specification value, x , representing input

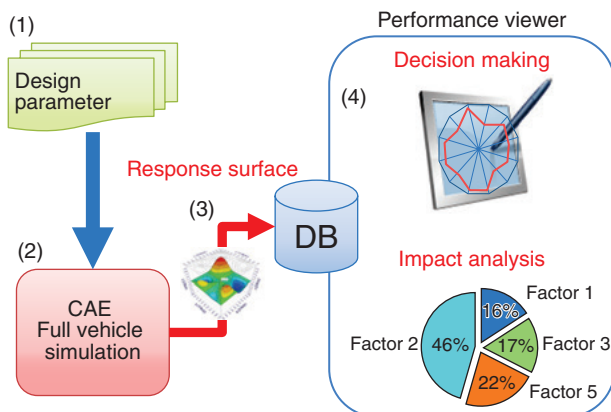


Fig. 1 Vehicle performance design system

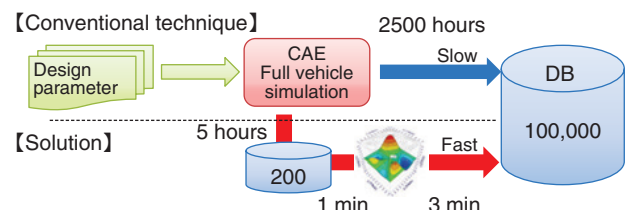


Fig. 2 Effect on response surface

data. This data might be experimental data or data generated by engineering simulations. Because conducting tests and simulations entails costs in terms of time and labor, it is often not possible to obtain data for a large number of cases. If a response surface can be generated for data for a limited number of cases, it is then possible to simply predict an output, $\mathbf{y}$, for a new input, $\mathbf{x}$. The case data necessary for the generation of a response surface is known as training data.

Terming modeling error ε , the relationship between input and output can be expressed by Eq. (1).

$$\mathbf{y} = M(\mathbf{x}) + \varepsilon \quad (1)$$

Here, $M(\mathbf{x})$ is the response surface. Output $\mathbf{y}$ is normally a vector quantity, but will be considered as a scalar quantity below, because modeling can be performed in relation to each component.

Various methods are available for the generation of response surfaces, $M(\mathbf{x})$. Because it was necessary to generate models of high approximation accuracy from limited training data in the research under discussion, the PCE-LAR method⁽⁴⁾ and the Kriging method were combined to produce the PCK method. Terming the function generated by the PCE-LAR method $f(\mathbf{x})$ and the function generated by the Kriging method $Z(\mathbf{x})$, $M(\mathbf{x})$ can be expressed as Eq. (2).

$$M(\mathbf{x}) = f(\mathbf{x}) + Z(\mathbf{x}) \quad (2)$$

As Fig. 3 shows, the PCE-LAR method expresses global changes in output, and the Kriging method expresses local changes in output. The modeling of both global and local changes makes it possible to realize a high level of approximation accuracy⁽⁵⁾. The PCE-LAR method and the Kriging method will be discussed in greater detail below.

2.1.1. PCE-LAR method

The PCE-LAR method combines modeling of global system response using polynomial chaos expansion (PCE) and high-accuracy approximation using least angle regression (LAR). In the PCE method, model $f(\mathbf{x})$ is expressed as a linear combination of orthogonal polynomials. For a scalar input x , $f(x)$ is given by Eq. (3).

$$f(x) = \sum_{i=0}^N a_i \Phi_i(x) \quad (3)$$

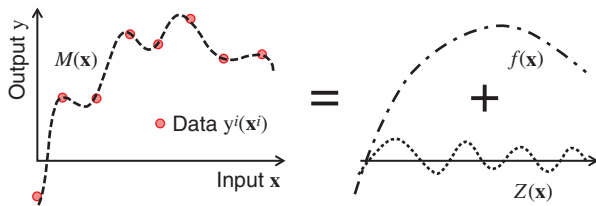


Fig. 3 Response surface model $M(\mathbf{x})$ as sum of global trend $f(\mathbf{x})$ and local fluctuation $Z(\mathbf{x})$

Here, a_i is a coefficient, N is the highest-order polynomial, and $\Phi_i(x)$ is the i -th orthonormal polynomial. In the research under discussion, Legendre polynomials were employed as the orthogonal polynomials. Because the range of the Legendre polynomial input x is $[-1, 1]$, it was necessary to convert the input to the relevant range in advance. In Eq. (3), all Legendre polynomials, from the 1st to the N th, are employed.

The coefficient a_i is determined using the generalized least squares method. If the number of training data points, $(\mathbf{x}, \mathbf{y})$ is termed n , n should be equivalent to or lower than the number of coefficients. An increase in the number of components of input $\mathbf{x}$, M , or the highest order of input $\mathbf{x}$, N , results in an explosive increase in the number of coefficients. For example, when $M=10$ and $N=4$, if all of the Legendre polynomials up to the 4th order are considered, the number of coefficients necessary for modeling exceeds 1,000. This means that the number of necessary training data points increases explosively when M or N increases.

LAR algorithms are employed in order to avoid this issue. LAR algorithms determine polynomials of the necessary order for the generation of the model in order of importance while referring to the training data. This enables considerably reducing the number of $\Phi_i(\mathbf{x})$ functions, i.e. the number of coefficients, employed in the model. This reduction of the number of coefficients employed also increases approximation accuracy. The reason for this is as follows: When an LAR algorithm is not employed, the explosive increase in the number of terms makes it hard to use terms up to the higher orders; when an LAR algorithm is used, the model is generated using polynomials of the necessary order only, making it possible to use high-order terms without the risk of explosive increase in the number of terms.

2.1.2. Kriging method

The Kriging method describes variation from an average. This variation is regarded as conforming to a stochastic process. The variations at two points in the training data, $\mathbf{x}^i$ and $\mathbf{x}^j$, are taken as having correlation with a correlation function, R_{ij} , which is dependent on the distance $|\mathbf{x}^i - \mathbf{x}^j|$. Gaussian-type, exponential-type, and Matérn-type R_{ij} exist. The appropriate type in relation to the training data was distinguished in the research under discussion. Because every type incorporated a parameter, the value of the parameter was determined using optimization computations.

The variation modeled by the Kriging method, $Z(\mathbf{x})$, is given by Eq. (4).

$$Z(\mathbf{x}) = \mathbf{r}(\mathbf{x}) \mathbf{R}^{-1} \Delta \mathbf{y} \quad (4)$$

Here, $\mathbf{r}(\mathbf{x})$ is the row vector of components of n generated by correlating $\mathbf{x}$ and training data $\mathbf{x}^i$, $\mathbf{R}$ is the $n \times n$ matrix making up R_{ij} , and $\Delta \mathbf{y}$ is the column vector of components of n given by subtracting $f(\mathbf{x}^i)$ from training data y^i .

In the actual calculations, the creation of the response surface is divided into first and second stages. In the first stage, the order of precedence of candidates for the polynomial $\Phi_i(\mathbf{x})$ used to construct $f(\mathbf{x})$ is determined using LAR algorithms. In this stage, $f(\mathbf{x})$ remains undetermined. The reason for this is that the function $f(\mathbf{x})$ optimized using the PCE method in isolation will not necessarily correspond to the function $f(\mathbf{x})$ optimized using the combination of the PCE and Kriging methods. In the second stage, the appropriate form of the correlation function is first distinguished. Next, following the order of precedence determined in the first stage, the error of $M(\mathbf{x})$ as determined using the combination of the PCE and Kriging methods is evaluated while adding the $\Phi_i(\mathbf{x})$ functions used to constitute $f(\mathbf{x})$ one by one. The response surface $M(\mathbf{x})$ is determined using the combination of $f(\mathbf{x})$ and $Z(\mathbf{x})$ that produces the least error.

2.1.3. Verification of method

The developed method was applied to data obtained from fuel efficiency and acceleration simulations, and approximation error was studied. Figure 4 shows modeling results for simulations of three fuel economy modes (US-Hot, JC08, and EUDC). Results obtained using a high-accuracy method [anisotropic Kriging (AKR)] provided by a commercially available response surface generation tool are also shown for comparison. The horizontal axes of the graphs show the number of training data points, and the vertical axes show relative error (%). In order to calculate relative error, data differing from the training data was employed, and relative error was determined for each data point and averaged for all data points. For all the fuel economy modes, approximation error becomes smaller in the order AKR→PCE-LAR→PCK. Figure 5

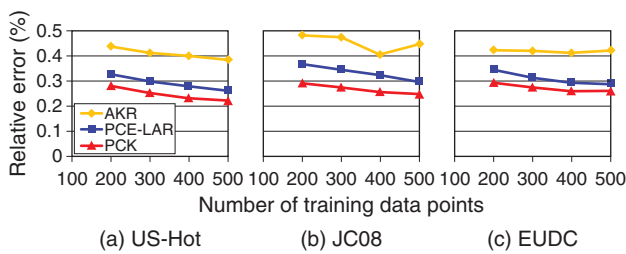


Fig. 4 Relative error plot for three fuel economy modes

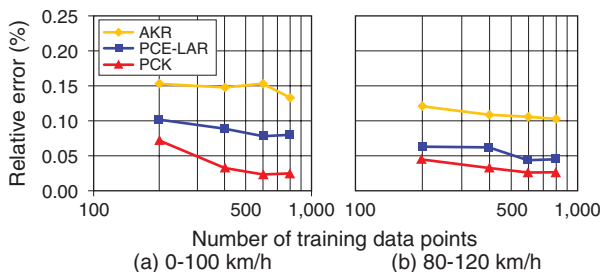


Fig. 5 Relative error plot for two acceleration modes

shows results generated by modeling acceleration time for two acceleration modes (from 0-100 km/h and from 80-120 km/h) when the method was applied to acceleration simulation data. The magnitude of error is half or less than that for the fuel efficiency data. In terms of the difference between the methods, as in the case of fuel economy data, approximation error becomes smaller in the order AKR→PCE-LAR→PCK.

The results discussed above indicated that for both fuel economy and acceleration simulation data, the use of the PCK method reduced error by comparison with the PCE-LAR method. Figure 6 shows models obtained from mode fuel economy simulations, produced in order to assist in understanding the difference between the methods. The horizontal axis shows normalized design values, and the vertical axis shows mode fuel economy values expressed in terms of variation in relation to a reference value. The PCE-LAR method accurately captures the overall trend across the input value range. However, there are considerable local discrepancies between values from data and values predicted by the model. The PCK method, in which the PCE-LAR method is used together with the Kriging method, considerably reduces these discrepancies. Depending on the specific characteristics of the data, the reduction might not be as marked as in this example, but response surfaces generated by the PCK method, combining the PCE-LAR and Kriging methods, tend to display more robust performance than response surfaces generated using the PCE-LAR or Kriging methods in isolation.

2.2. Decision Making

While a user may wish to determine vehicle performance target values using a large-scale DB, the fact that suitable vehicle performance data is not easy to obtain presents an issue. The user therefore employs decision making to conduct a high-speed search for vehicle performance data using a minimal number of operations.

The research under discussion employed two search techniques, direct search and group search.

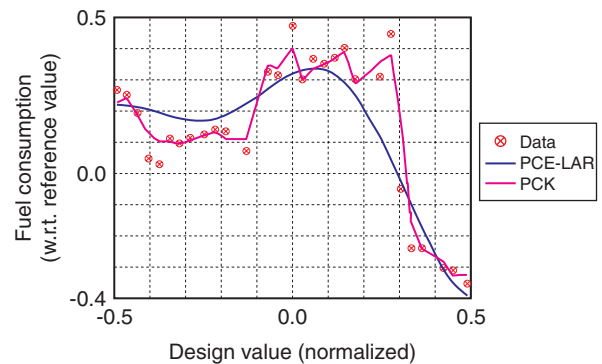


Fig. 6 Comparison of simulation data and response surface model predictions (PCE-LAR and PCK)

2.2.1. Direct search: High-speed search method

The high-speed search method calculates the Euclidean distance between vehicle performance data in the DB and vehicle performance target values, and selects data for which the distance is lowest as solutions. Figure 7 shows the search technique using three-dimensional data as an example. The target for the user is the data marked with a star; because the data space is sparse, the method takes the data that most closely approximates this data as the search solution. Data can be sparse or dense within the formulated DB, and if the appropriate search solution does not exist within the DB, the accuracy of the solution will be low. However, because the search is approximate, calculations can be conducted at high speed.

2.2.2. Direct search: High-accuracy search method

The high-accuracy search method searches for optimum solutions using response surfaces generated in order to create large-scale DB. Figure 8 shows an example of the generation of a response surface from three-dimensional data and its use by the search technique in order to search

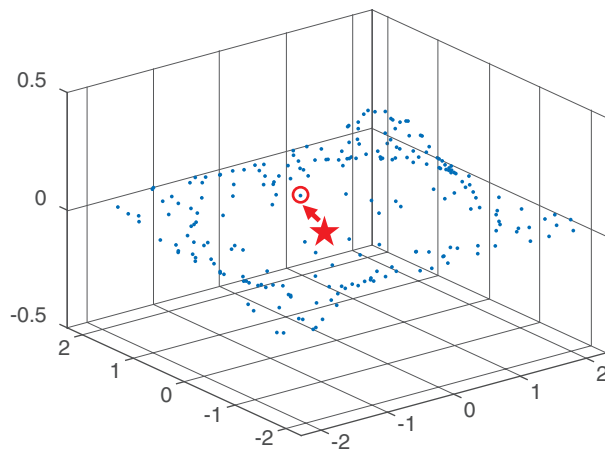


Fig. 7 High-speed search method by data exploration

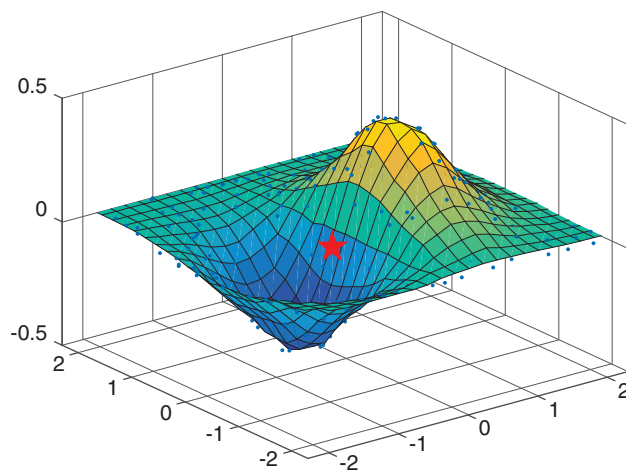


Fig. 8 High-accuracy search method by response surface

for a solution. The target for the user is the data marked with a star. In this method, the data within the DB space is mathematically complemented, and the solution is sought by calculating optimum values based on this complemented data. The method searches for data that approximates an error of 0 between performance values determined by the calculation of response surfaces using the gradient method and vehicle performance target values.

The issue for this method is that local optimization might occur due to the effect of the initial value, and it therefore might not be possible to find the true optimum solution. However, local optimization can be avoided by randomly changing the initial value 100 times in advance, making it possible to realize true optimization. A lengthy calculation time is necessary for the solution of the optimization problem in this method, but the accuracy of the solutions determined is high.

2.2.3. Group search

The group search technique clusters the data within the DB and displays representative data (the centroid for each class) on radar charts. Using these radar charts, the user is able to search for the solution by viewing the data and narrowing it down. Figure 9 shows an example of the process of narrowing down the data. The data within the DB is clustered and expressed as the five classes on the left-hand side of Fig. 9. The user selects one of the classes from among these five. The right-hand side of Fig. 9 shows the result of this selection. Because performance values in the DB (P1-P12) are displayed for each term on each radar chart, the user is able to narrow the data down to the desired performance values while taking the overall performance balance into consideration.

Fuzzy C-Means⁽⁶⁾ was introduced as the clustering method in the research under discussion in order to expand the range of data selection by the user. The Fuzzy C-Means method does not determine class dependency, which is the standard mode of operation for conventional clustering algorithms. Rather than classifying specific data points into a single exclusive class, the method considers each data point to belong to multiple classes, and expresses separation boundaries fuzzily using degrees of assignment, which indicate the degree to which each data point belongs to particular classes. Figure 10 shows the differences in the separation boundaries between three classes for the conventional and the Fuzzy C-Means methods. The left-

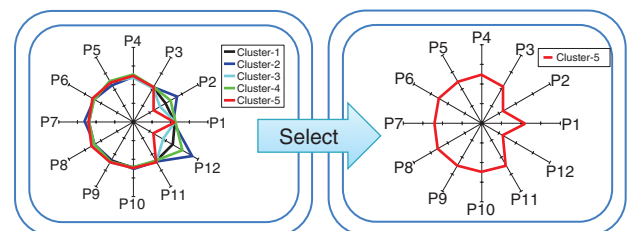


Fig. 9 Decision making

hand image in Fig. 10 shows the result of clustering based on a general determination of dependence, while the right-hand image shows the result of fuzzy clustering using the degree of assignment for data around the separation boundaries between the classes.

Fuzzy C-Means algorithms function as follows:

- (1) The degree of assignment of element i to class j , u_{ij} , is initially randomly set.
- (2) Using the current degree of assignment, u_{ij} , the centroid vector, $\bar{x}_i$, is calculated for each class. m is a fuzzy parameter that determines fuzziness, satisfying the condition $m > 1$. n is the number of feature vectors for machine learning, and element x_j is the aggregation of data. The centroid vector, $\bar{x}_i$, is given by Eq. (5).

$$\bar{x}_i = \frac{\sum_{j=1}^n (u_{ij})^m x_j}{\sum_{j=1}^n (u_{ij})^m} \quad (5)$$

- (3) Using the calculated centroid vector $\bar{x}_i$, the degree of assignment of element x_j to class i , u_{ij} , is updated. $\|\cdot\|$ is the Euclidean norm, and c is the parameter for the number of classes. The degree of assignment, u_{ij} , is given by Eq. (6).

$$u_{ij} = \left(\sum_i^c \left(\frac{\|x_j - \bar{x}_i\|}{\|x_j - \bar{x}_i\|} \right)^{\frac{2}{m-1}} \right)^{-1} \quad (6)$$

- (4) The centroid for each class is recalculated, and clustering is completed when there is no longer any change in the centroids in any class. If there is change in a centroid, the process is recommenced from step (2).

2.3. Impact Analysis

In many cases, performance values are functions of multiple specification values. Because of this, it is not easy to determine the effect of each variable. Partial dependence plots are therefore employed in order to visualize the relationship between performance and each variable. Partial dependence plots are graphs that express the effect of variables other than the variable being focused on as averages. Figure 11 shows examples of partial dependence plots. The figure shows five partial dependence plots in which performance y is expressed in terms of five

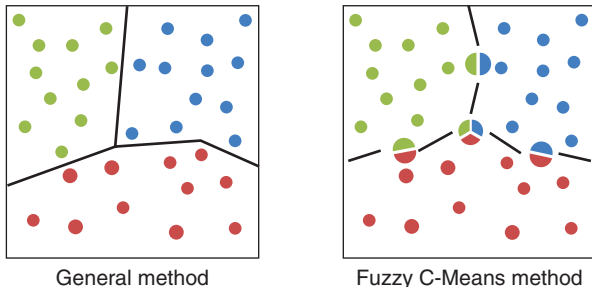


Fig. 10 Difference between clustering methods

specifications. The horizontal axes show the specifications, and the vertical axes show performance y .

Because y is a function of variables termed Factors 1 to 5, even if, for example, Factor 1 was to take a specific fixed value, the value of y would continue to fluctuate if Factors 2 to 5 changed within the scope of their upper and lower limit values. Repeating the operation of determining the average of the fluctuating value of y and plotting this value against the value of Factor 1 makes it possible to obtain the characteristic shown as the solid black line. This expresses the average characteristic of change in y in relation to Factor 1. The two broken red lines show the positions of the 25th percentile and the 75th percentile of the amplitude of fluctuation in y , providing a yardstick for the magnitude of the fluctuation in y due to other variables. The same processing can be used to create partial dependence plots for Factors 2 to 5.

Figure 11 shows that the characteristic of y in relation to Factor 2 is basically linear, indicating a linear relationship between the two. By contrast, the characteristic of y in relation to Factors 1, 3, and 5 describes a curve, indicating a nonlinear relationship with these variables. The characteristic of y in relation to Factor 4 is basically horizontal, indicating that this variable has almost no effect on performance. The intervals between the broken red lines show that all of the variables are affected by the other variables. In the case of variable 2, there are two areas in which the interval between the red broken lines becomes narrower, indicating that in these points the effect of the other variables is diminished.

Impact factor is used to help enable quantitative comparison in order to make the magnitude of the effect of each specification easier to understand. Impact factor is an index that expresses the changes in performance values when specifications are changed, and is determined by partially differentiating performance values with respect to specifications. Terming the response surface function f , the impact factor, g_i ($i=1, 2, \dots$), of Factor x_i ($i=1, 2, \dots$) is expressed as follows:

$$g_i = \frac{\partial y}{\partial x_i} \cong \frac{\partial f(x_1, x_2, \dots, x_n)}{\partial x_i} \quad (7)$$

Like performance values, impact factor is a function of multiple variables, making it challenging to compare impact factor between variables. The average value of g_i is therefore employed as the Impact factor of variable x_i .

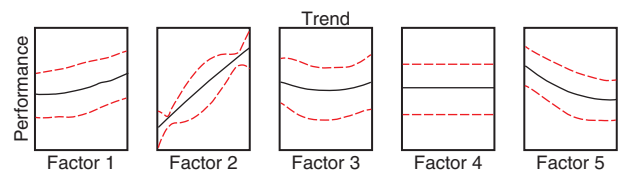


Fig. 11 Trend curves for parameters

Figure 12 shows the calculated Impact factor in descending order as a bar graph. In this example, Factor 2 displays the highest impact factor, two to three times greater than those of Factors 1, 3, and 5, while the impact factor of Factor 4 is basically 0. Unlike partial dependence plots, this method is suited to comparison of the magnitude and ratio of the effect of each variable.

Figure 13 shows the impact factor ratio as a band graph. The impact factor of Factor 2 is similar to the impact factor of Factors 1, 3, and 5 combined.

The selection of variables and the magnitude of change in values when the design is changed are determined on the basis of this impact factor data.

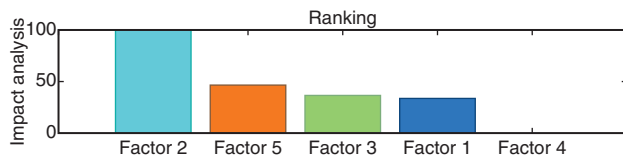


Fig. 12 Ranking of contribution ratio

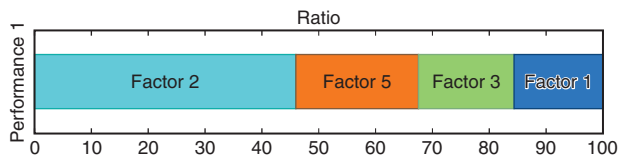


Fig. 13 Contribution ratio expressed as band graph

3. Application

To provide an example of the application of the vehicle performance design system developed in the present research, this section will discuss an analysis of a vehicle dynamic performance simulation using the system.

Data for the five specification values forming the input was sampled using a design of experiments method, and performance values were calculated using CAE in order to generate training data. Data for 400 cases was prepared for this example of system application. Response surfaces were generated using the completed performance value and specification datasets in order to create a large-scale DB.

PCE-LAR, as discussed in Section 2.1., was employed for the generation of response surfaces. In this example, three types of response surfaces were generated for three performance values. The use of these response surfaces not only makes it possible to conduct high-speed calculations for large-scale DB, but also to calculate specifications from performance and analyze impact factor. The results of three analyses will be discussed below.

3.1. Direct Search

Here, the user specifies target values in relation to three aspects of performance. As an example, specification values

will be calculated for the three performance values. Figure 14 shows a radar chart for the three target performance values.

Figures 15 and 16 show the results of calculation of specification values in relation to the three target values shown in Fig. 14. Figure 15 shows the results of the high-speed direct search method, and Fig. 16 shows the results of the high-accuracy search method. Both Figs. 15 and 16 are expressed as radar charts made up of the five specification values, and show five alternative solutions using the target

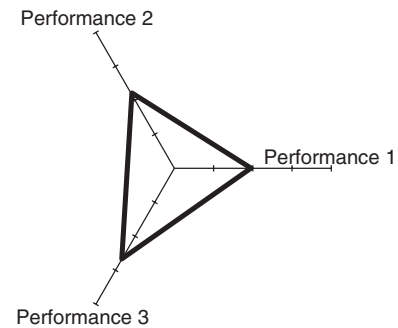


Fig. 14 Target performances

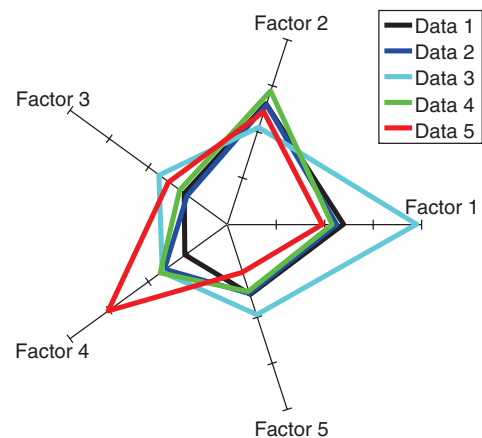


Fig. 15 Direct search results (High-speed search)

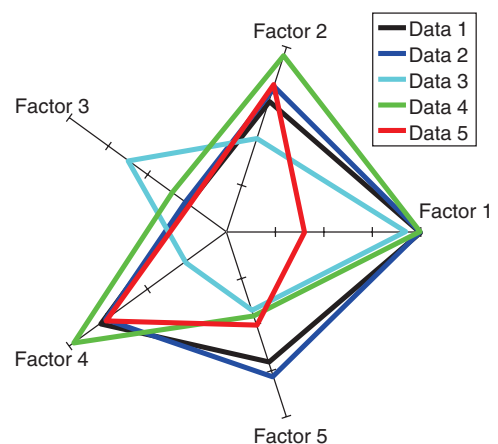


Fig. 16 Direct search results (High-accuracy search)

values specified in Fig. 14.

The user simply has to specify the target performance values, and the system presents a combination of specification values that satisfies the performance targets. The time necessary to produce alternative solutions on the same workstation was 0.043 sec for the high-speed search method and 13.89 sec for the high-accuracy search method.

The user is able to select the desired specification values from the alternative solutions shown in Figs. 15 and 16. In addition, the use of two direct search methods makes it possible to obtain different candidates for solutions even for the same performance values.

3.2. Group Search

Group search is a method of data selection using data in the DB.

The user is able to search for specification values while verifying performance values using radar charts. Because this search method uses a clustering technique, it does not directly identify the desired performance values from among large volumes of data, but uses class data to help enable simpler decision making, in stages. Figure 17 shows the flow of decision making using class data.

- (1) The 100,000-case data in the DB is clustered into five classes and expressed as a radar chart of performance values.
- (2) The user selects the class made up of the desired performance values. As a result, data for 100,000 classes is reduced to data for 21,485 classes.
- (3) The 21,485-class data is reclustered into five classes and expressed as a radar chart of performance values.
- (4) The user selects the desired class from among the five performance value classes. As a result, the data is reduced to data for 7,306 cases. This narrowing-down operation is continued in order to further reduce the amount of case data.

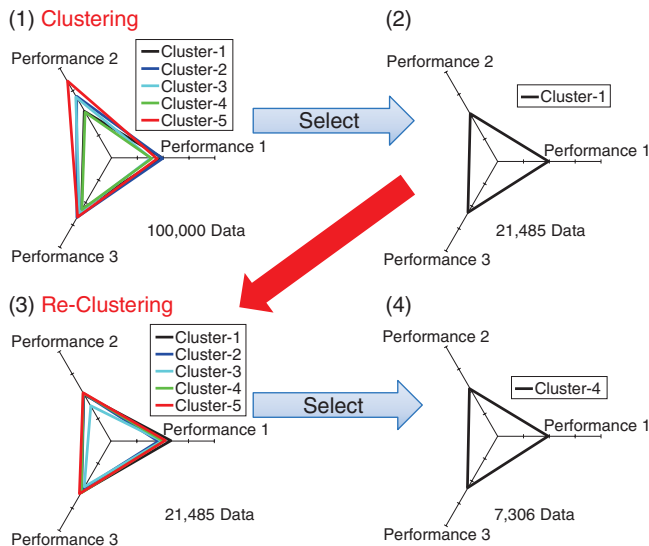


Fig. 17 Decision making flow

The use of data classified into five representative classes makes decision making simpler. Proceeding with decision making reduces the amount of data, advancing the process of narrowing down.

Figure 18 shows the final results of narrowing down the data.

The image on the left-hand side of Fig. 18 shows the result obtained by narrowing down performance, while the image on the right-hand side shows the result obtained by narrowing down alternative solutions for specification values to five.

In this way, the group search method makes it possible to narrow down the data while verifying the balance between different aspects of performance using radar charts.

3.3. Analysis Using Impact Factor

By making it possible for users to identify specification values in relation to the desired performance through decision making, and to select viable options from data for alternative solutions for specification values, the system is able to increase the efficiency of the development process against conventional methods. The system also makes it possible for the user to set detailed specification values using the results of impact analysis. This process will be explained below.

Figure 19 shows the results of an impact analysis in relation to performance (Performance 1). The upper section of Fig. 19 shows the order of precedence of the Impact factor of each specification, while the lower section shows partial dependence plots for the five specifications

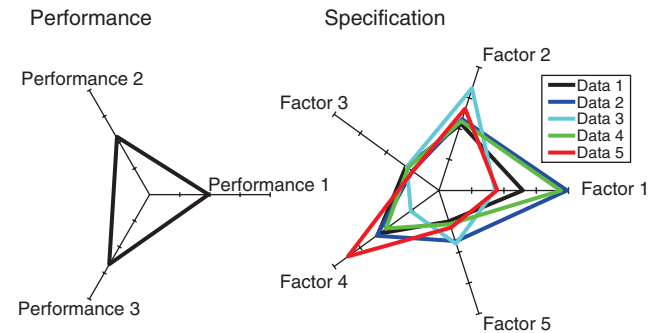


Fig. 18 Decision making results

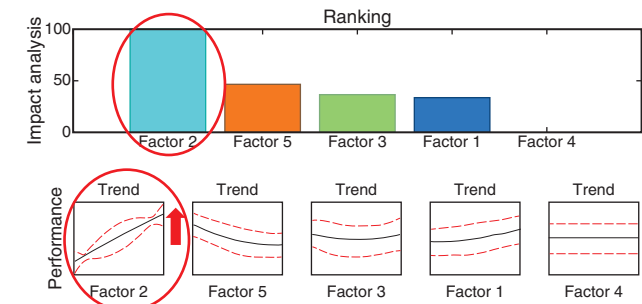


Fig. 19 Results of Impact analysis

in relation to performance. The upper section of the figure indicates that performance could be boosted by changing the parameters of Factor 2. Using the results of the partial dependence plots shown in the lower section of the figure, it is possible to identify the direction for the setting of parameters. The results shown here indicate that increasing the value of Factor 2 will increase performance.

Figures 20 and 21 show changes in performance when the value of Factor 2 is increased. Figure 20 shows a radar chart for the five specifications; Data 2 is produced by changing Factor 2 from its setting in Data 1. Figure 21 shows the change in performance when the value of Factor 2 is changed. The results show that performance is boosted against Data 1 in Data 2.

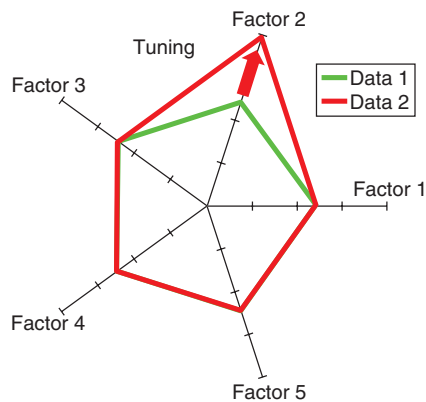


Fig. 20 Factor setting

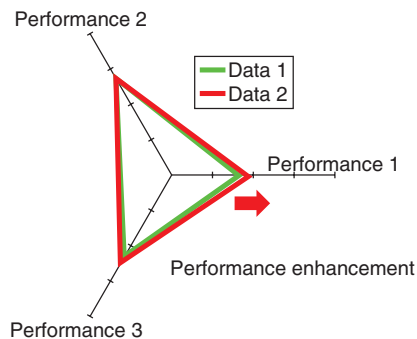


Fig. 21 Performance results

4. Conclusion

The research discussed in this paper developed a vehicle performance design system able to assign target values for different performance demands at the initial stage of the development process using three data mining techniques:

- (1) A response surface generation technique that helps enable high-accuracy predictions based on small amounts of data;
- (2) A decision-making technique that helps enable high-efficiency data searches that reflect the user's

intentions; and

- (3) An impact analysis technique that makes it possible to identify specifications that have the greatest effect on performance from the relationships between performance and specifications.

It was demonstrated that the application of this system to vehicle dynamic performance simulations enables users to conduct analyses simply and to determine performance values and specification values at high speed.

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Strength Design Method of Metallic Separator for Fuel Cell

Katsunori SAIKI*

ABSTRACT

A strength design method was developed to form channels using ultrathin stainless steel foil in the structural design of metallic separators for a fuel cell. This method focused on the work hardening characteristic of stainless steel, and defined the deformation behavior of the metallic foil as a function of the strain value, based on the strength-related physical property values of the metallic materials. The shape range that can be formed with metallic foil was obtained using a forming limit diagram and localized strain values calculated from measurements of the material physical property values. In addition, the mechanical stress produced on the metallic separators of the fuel cell during fuel cell stack power generation was obtained from localized strain values calculated by micro Vickers hardness measurements. The metallic foil life with respect to mechanical degradation could be evaluated by plotting this stress value in the material fatigue limit diagram.

1. Introduction

One of Honda's goals is to realize use of fuel cell vehicles (FCV) on the same level as vehicles equipped with internal combustion engines. Honda's "Man Maximum Machine Minimum" philosophy calls for reduced size and weight and increased power density of the fuel cell stack mounted in FCV in order to provide a comfortable cabin befitting a sedan. A fuel cell stack contains hundreds of layered separators and membrane electrode assemblies (MEA) (Fig. 1), and reduction of the metallic separator thickness and a fine channel shape design are needed to reduce the size and weight of the fuel cell stack and realize a higher power density⁽¹⁾. This calls for ultrathin stainless steel materials.

However, thickness reduction and cracks easily occur when ultrathin stainless steel foil is stamped into a fine channel shape (Fig. 2).

It is necessary to avoid localized plastic deformation

and perform uniform work hardening in order to reduce this occurrence of thickness reduction and cracks.

Furthermore, during fuel cell stack power generation, load fluctuation occurs due to swelling and shrinking of the MEA located between the separators and pressure fluctuation occurs due to the gas and coolant. This load and pressure fluctuation produces constant stress fluctuation on the metallic separators of the fuel cell. Thus, strength design is needed to help prevent separators formed by ultrathin stainless steel foil from experiencing plastic deformation and fatigue failure due to the stress produced during power generation (Fig. 3).

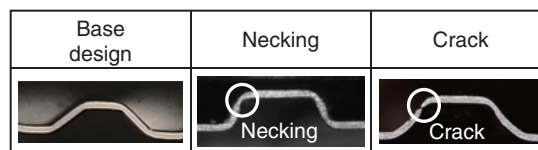


Fig. 2 Issue due to forming

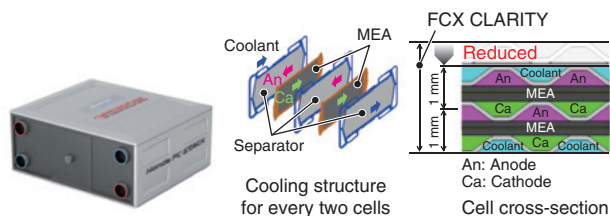


Fig. 1 FC stack and structure

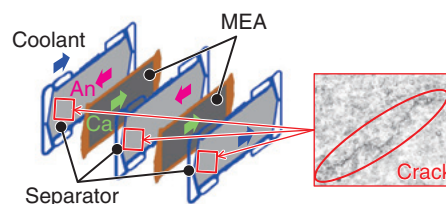


Fig. 3 Issue due to plate fatigue

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Therefore, this study focused on the work hardening characteristic, and defined the strain and stress from the physical property value of the material hardness. This enabled life evaluation by drawing the material fatigue limit curve from the stress values obtained in this manner.

2. Issues with Conventional Confirmation Method

2.1. Assessment of Forming Limit

The conventional method for achieving stainless steel foil stamping without localized thickness reduction and cracks is to create a large number of samples by stamping while varying the design parameters (flat length, corner R, angle, channel depth, and channel pitch) shown in Fig. 4. The plastic deformation limit is assessed by checking for localized thickness reduction and cracks by means such as cross-sectional observation of the created samples.

A forming limit diagram (FLD) (Fig. 5) is typically used to assess the plastic deformation limit of stamped products. The FLD shown in Fig. 5 demonstrates that localized thickness reduction does not occur when the strain is less than the forming limit curve (FLC), and that noticeable thickness reduction and cracks occur when the

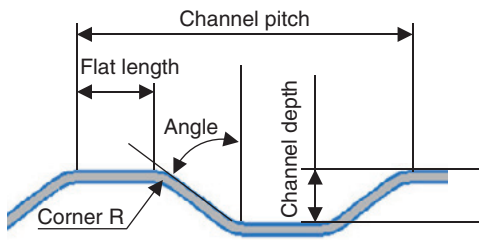


Fig. 4 Design parameters

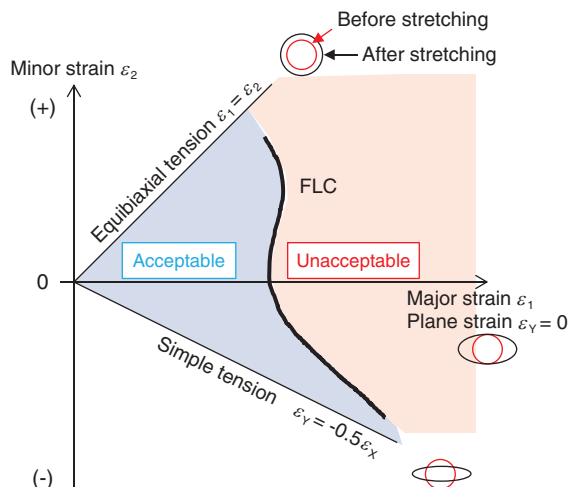


Fig. 5 FLD

strain is greater than the FLC. For this reason, FLD are widely used as a simple means of assessing the formable range of stamped products.

The scribed circle test method (Fig. 6) used when drawing the FLD stamps a steel sheet with a pre-drawn reference graphic ranging in size from $\phi 5$ mm to $\phi 20$ mm into the prescribed shape. The amount and direction of change in the pre-drawn reference graphic due to stamping are measured and plotted within a coordinate system that uses the Major strain ε_1 as the horizontal axis and the Minor strain ε_2 as the vertical axis. This enables understanding of the deformation state at the evaluation target location⁽²⁾.

The separator has a fine and complex shape, however, which makes it a challenge to microscopically draw this reference graphic in the separator location to be evaluated from the viewpoints of identifying deformation locations and measurement accuracy. Therefore, obtaining accurate separator data is an issue for performing fine and accurate design.

2.2. Assessment of Fatigue Limit

When performing strength design for a separator formed by stainless steel foil, it is necessary to accurately understand localized stress in order to help prevent the occurrence of plastic deformation and fatigue failure due to the stress produced during fuel cell stack power generation.

One method for calculating localized stress is the stress measurement method using a strain gauge. This method attaches a strain gauge in the specified location and calculates the stress from the measured strain value. However, it is a challenge to attach a strain gauge to the fine surface of the separator and confirm the actual behavior during power generation. The reasons for this are first that power generation is performed in the condition with a strain gauge wired into a fine channel, which hinders the supply of gas needed for fuel cell power generation to the channel. In addition, when a strain gauge is attached to ultrathin metallic foil, the measured localized stress produced on the separator during power generation includes the reinforcement effect of the strain gauge, which is not appropriate from the viewpoint of measurement accuracy.

Therefore, understanding the localized stress produced on the separator during fuel cell stack power generation was an issue.

Before stretching	After stretching	Definition of 'strain'
		Major strain $\varepsilon_1 = \ln(d_1/d_0)$ Minor strain $\varepsilon_2 = \ln(d_2/d_0)$

Fig. 6 Scribed circle method

3. Strength Design Method for Separator Channel

As localized strain measurement is a challenge with the conventional method, the authors focused on the work hardening characteristic of stainless steel foil, and attempted to derive the localized strain measurement from the hardness measurement as a method for more accurately and quickly assessing the localized deformation state of the separator and the fatigue failure life due to stress.

The localized strain values of the stamped separator were calculated from the hardness and the thickness reduction rate in order to assess the forming limit. The plastic deformation limit was assessed using these strain values and the FLD. Next, the localized strain values were calculated from the localized stress produced on the separator and the stainless steel foil hardness in order to assess the fatigue limit. The fatigue failure life was assessed using this value and the fatigue limit diagram. In this manner, a strength design method for the separator channel was developed by defining the stress produced on the separator and the deformation behavior as functions of the strain value.

4. Technology for Prediction of Plastic Deformation Limit Using FLD

First, stamping simulation was used to predict the strain locations and directions in order to clarify the localized strain locations and strain direction of a separator channel stamped into the prescribed shape. The formable shape range was then obtained by applying the strain values calculated from micro-hardness measurement of the obtained localized deformation locations to the FLD.

4.1. Analysis of Strain Produced on Separator during Stamp Forming

4.1.1. Confirmation of strain behavior

Since fuel cell separators have a fine channel shape, stamping simulation was used to predict the localized deformation locations and directions from the viewpoint of measurement accuracy.

Stamping simulation was performed for the shape samples shown in Table 1. The simulation conditions and analysis results for that typical shape are shown below. Analysis was performed over a range simulating part of the separator channel shape to identify localized deformation locations in the plane direction of the fine channel shape (Fig. 7).

In addition, analysis was performed with the minute mesh size setting of 1 μm to confirm locally occurring strain.

Figure 8 shows the results of predicting the locations where localized deformation occurs in the plane direction of the fine channel shape.

The greatest strain was identified to occur at the locations marked by red circles in Fig. 8. Next, Fig. 9 shows the simulation results illustrating the direction of localized strain occurrence in the A-A' plane in Fig. 8.

Table 1 Conventional method samples

	Sample 1	Sample 2	Sample 3	Sample 4	Sample 5	Sample 6	Sample 7	Sample 8
Flat length	1.00	1.00	1.00	1.00	0.00	0.00	0.53	0.34
Angle	1.00	1.07	1.10	1.11	1.07	1.10	0.80	0.89
Corner R	1.00	0.83	0.67	0.50	0.83	0.67	0.67	0.83
Channel depth	1.00	1.00	1.00	1.00	1.00	1.00	0.99	0.99
Channel pitch	1.00	0.96	0.92	0.88	0.48	0.44	0.76	0.76
Judge	Good	Good	Good	Good	Good	Good	Good	Good

The direction of the planar deformation produced by forming was analyzed by simulation. The results showed that at the locations where the largest localized strain occurs as marked by red circles in the A-A' plane in Fig. 9, the deformation extends in the direction of the Major strain ε_1 axis shown in Fig. 5. In a wave-shaped channel, the largest localized deformation was confirmed to occur at the corners R marked by red circles in Fig. 8, which corresponds to the Major strain ε_1 axis direction of the FLD.

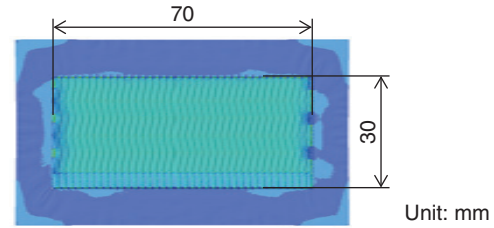


Fig. 7 Simulation method

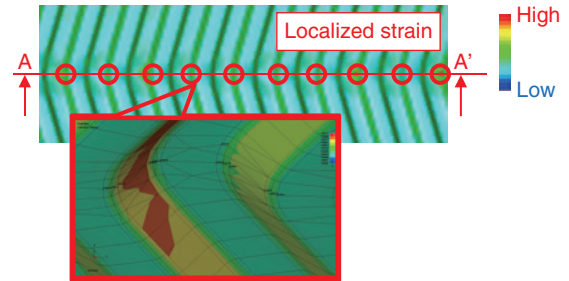


Fig. 8 Localized strain point in plane direction

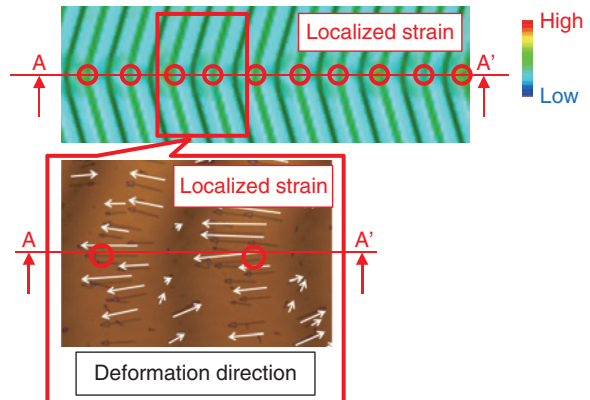


Fig. 9 Deformation direction of plane

4.1.2. Mathematical expression of stainless steel foil micro-hardness and strain

This study focused on the work-hardening characteristic of ultrathin stainless steel foil to perform strength design for the separator channel shape. The work-hardening characteristic is the characteristic whereby the hardness changes due to strain. An attempt was made to use this work-hardening characteristic to derive the localized strain measurement, which is a challenge with the conventional method, from the hardness measurement.

A micro Vickers hardness gauge that can measure microscopic locations was used to obtain the localized strain of a separator with a fine channel shape. The hardness test method conformed to the JIS Z 2244 (Vickers hardness test - Test method)⁽³⁾ specification. The shape of the diamond indenter used in the micro Vickers hardness test was a square pyramid with an angle between opposite faces of 136°. The relationship between the stainless steel foil micro-hardness and strain was obtained by creating test pieces with the prescribed strain applied by tensile tests and measuring the stainless steel foil hardness by the micro Vickers hardness test method. The test pieces for the tensile test used 13B specimens that conform to the JIS Z 2201 (Tensile test piece for metallic materials)⁽⁴⁾ specification.

Micro-hardness was expressed mathematically as a function of the strain value to create a formula for the work-hardening characteristic of stainless steel foil. Work hardening is a phenomenon that occurs when the plastic strain of the material produces the slipping motion of dislocation, and the dislocation density within the material increases together with that plastic strain. Therefore, work hardening was treated as dislocation density to formulate the relationship between work hardening and strain. This dislocation density and the true stress have a positive correlation. In addition, the true stress can be formulated as a function of strain from the tensile strength characteristic of the material, so the relationship between the dislocation density and strain was obtained. This means that work hardening can be expressed as a function of strain. The detailed results are shown below.

First, the dislocation density ρ and the true stress σ have a positive correlation as expressed by Eq. (1)⁽⁵⁾. Here, M is the Taylor factor, ϕ_D is a constant, G is the modulus of rigidity, and b is the size of the Burgers vector.

$$\sigma = M\phi_D Gb \sqrt{\rho} \quad (1)$$

As work hardening is produced by the slipping motion of dislocation, there was also thought to be a relationship between micro-hardness Hv due to work hardening and the true stress σ . In addition, the Vickers hardness and Rockwell hardness obtained with a high test load and the stress have a linear relationship for a typical steel sheet. The relationship between micro-hardness and strain was obtained from tests.

$$f(Hv) = A * \sigma + B \quad (2)$$

The true stress σ and true strain ε_P can be expressed as

a function of the nominal stress σ_0 and nominal strain ε_0 ⁽⁶⁾. Equations (3) and (4) are shown below.

$$\sigma = \sigma_0 * (1 + \varepsilon_0) \quad (3)$$

$$\varepsilon_P = \ln(1 + \varepsilon_0) \quad (4)$$

The micro-hardness can be expressed as a function of the nominal stress and nominal strain by substituting Eqs. (3) and (4) into Eq. (2).

The Vickers hardness can be formulated as a function of the strain value by using a Swift-type constitutive equation (5) that provides good approximation with respect to the true stress-true strain curve for typical stainless steel sheet described hereafter in section 4.1.4.

Here, σ is the true stress, ε_P is the true strain, and F , b , and n are constants. Work-hardening coefficient is F , work-hardening exponent is n , and strength-related physical property values of the materials were derived from the Swift-type equation.

$$\sigma = F(b + \varepsilon_P)^n \quad (5)$$

Here, the symbol f represents the function, A and B are constants, F is the work-hardening coefficient, and n is the work-hardening exponent.

$$f(Hv) = A * F[b + \ln(1 + \varepsilon_0)]^n + B \quad (6)$$

Figure 10 plots the relationship between strain and hardness for the ultrathin stainless steel materials applied to the fuel cell separator, and shows the formulated Eq. (6) results expressed as a solid line. The constants A and B were obtained from tests.

It was confirmed that hardness could be calculated as a function of the strain value as shown in Fig. 10.

These results show that conversion to the microscopic strain values produced on the separator based on the stainless steel foil hardness, which is a strength-related physical property value, makes it possible to understand the localized deformation state even in a separator with a complex and fine shape.

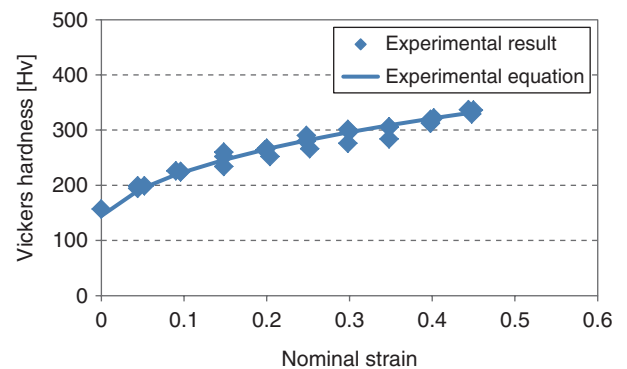


Fig. 10 Vickers hardness results under various strain conditions

4.1.3. Mathematical expression of stainless steel foil thickness reduction rate and strain

Stainless steel has a work-hardening characteristic, and the localized strain produced by plastic deformation can be defined as a function of the thickness reduction rate under constant volume conditions, so the localized strain was derived from the results of thickness measurement by cross-sectional observation. The relationship between localized strain and the thickness reduction rate for the ultrathin stainless steel foil applied to the fuel cell separator was obtained as follows.

The relationship between the strain and the thickness reduction rate was derived experimentally by creating samples with the prescribed strain applied by tensile tests and then measuring the thickness.

The amount of strain ε applied to the sample by the tensile test is expressed by Eq. (7)⁽⁶⁾. Here, L_0 is the initially evaluated length and L_1 is the evaluated length at the prescribed strain amount.

$$\varepsilon [\%] = (L_1 - L_0) / L_0 * 100 \quad (7)$$

In addition, labeling the initial thickness as T_0 and the thickness of a sample subjected to the prescribed strain as T_1 , the thickness reduction rate ΔT is expressed by Eq. (8)⁽⁶⁾.

$$\Delta T [\%] = (T_1 - T_0) / T_0 * 100 \quad (8)$$

The test pieces for the tensile test used 13B specimens that conform to the JIS Z 2201 (Tensile test piece for metallic materials)⁽⁴⁾ specification.

In addition, labeling the strain in the plane directions as ε_1 and ε_2 and the strain and true strain in the thickness direction as ε_3 , ε_3 can be expressed by Eq. (9) and Eq. (10)⁽⁶⁾.

$$\varepsilon_3 = \ln (T_1 / T_0) \quad (9)$$

Equation (10) was then derived using constant volume conditions.

$$\varepsilon_3 = -(\varepsilon_1 + \varepsilon_2) \quad (10)$$

From Eq. (8), Eq. (9), and Eq. (10), the thickness reduction rate ΔT can be expressed by Eq. (11).

$$\Delta T [\%] = \{1 - \exp [-(\varepsilon_1 + \varepsilon_2)]\} * 100 \quad (11)$$

Figure 11 plots the relationship between the strain and thickness reduction rate for ultrathin stainless steel foil, and shows the formulated Eq. (11) results expressed as a solid line.

It was confirmed that the thickness reduction rate could be calculated as a function of the strain value as shown in Fig. 11.

The above results were used to develop a method for quantifying the microscopic strain produced during stamping by confirming the thickness reduction rate, which can be measured in microscopic locations.

Based on the results in sections 4.1.2 and 4.1.3, application to FLD was investigated to obtain the formable range of ultrathin stainless steel foil.

4.1.4. Application to FLD

It was thought that measurement of strength-related physical property values such as the localized hardness and thickness reduction rate of ultrathin stainless steel foil at the localized deformation locations of the separator shown in section 4.1.1 and use of these values to calculate the localized strain values would also enable the use of FLD to design the channel shape of a separator formed by ultrathin foil. This was then verified.

The strain calculated from the localized hardness and thickness reduction rate for the conventionally manufactured samples shown in Table 1 and the Stören-Rice theory for predicting localized necking⁽⁷⁾ were used to draw FLC on the FLD shown in Fig. 5.

Equation (12) expresses the Stören-Rice theoretical formula for predicting localized necking.

$$\varepsilon_1 = [3\beta^2 + n(2 + \beta)^2] / [2(2 + \beta)(1 + \beta + \beta^2)] \quad (12)$$

Strain ratio β : $\varepsilon_2 / \varepsilon_1$

n : work-hardening exponent

Plastic deformation occurs when a material is pulled, and localized necking refers to thickness reduction in the thickness direction immediately before breaking. For thin sheet materials, the point at which localized necking occurs can be treated as the forming limit. The forming limit of ultrathin stainless steel foil was theoretically predicted by using a theoretical formula of localized necking occurrence to draw an FLC.

Here, material strength tests were performed on the stainless steel foil to obtain the work-hardening exponent and other characteristics, and a mathematical formula was created to express the relationship between the stress and strain of the stainless steel foil. The relationship between stress and strain is typically obtained by performing static tensile tests using tensile test pieces.

Thus, the stress-strain curve (S-S curve) of the stainless steel foil applied to the fuel cell separator was obtained by static tensile tests (Fig. 12).

The tensile test method conformed to the JIS Z 2241 (Method of tensile test for metallic materials)⁽⁸⁾ specification.

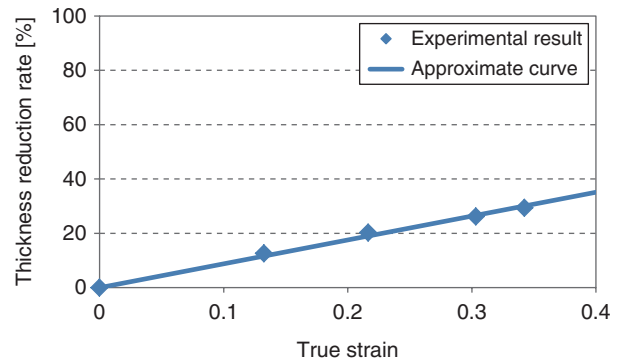


Fig. 11 Thickness reduction rate results

The test pieces for the tensile test used 13B specimens that conform to the JIS Z 2201 (Tensile test piece for metallic materials)⁽⁴⁾ specification.

Figure 12 shows the test results obtained by least square approximation using a Swift-type equation. The Swift-type constitutive equation (5) is known to provide a good approximation of the true stress-true strain curve of typical stainless steel sheet⁽⁹⁾.

In Fig. 12, T is the vertical direction, L is the flow direction, and D is the 45° direction relative to the rolling direction of the stainless steel foil.

The true stress-true strain curves for the T, L, and D directions trace almost the same line, so the material does not exhibit anisotropy. Furthermore, the test results could be approximated using a Swift-type equation, and the strength-related physical property values of the stainless steel foil needed to calculate the stress could be obtained.

The strain calculated from the localized hardness and the thickness reduction rate was plotted on the FLD (Fig. 13).

Of the obtained results, the blue plots indicate points where localized necking did not occur, and red plots indicate points where localized necking occurred.

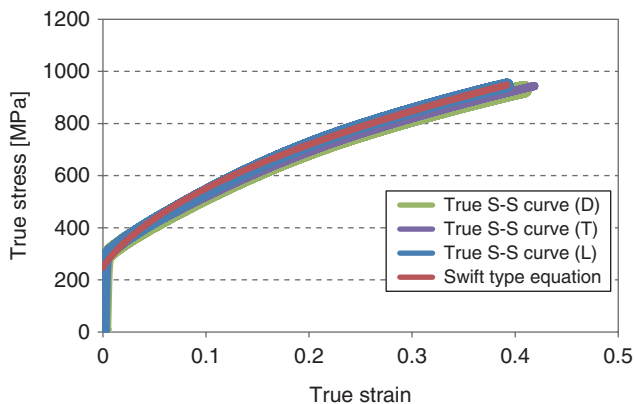


Fig. 12 True stress-true strain curve

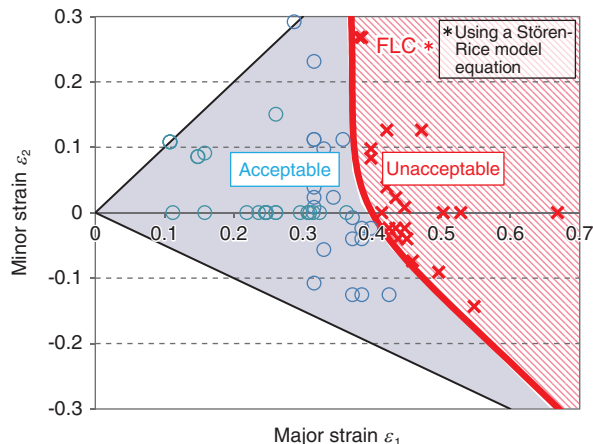


Fig. 13 FLD results

The respective plots made it possible to confirm whether localized necking occurred with the FLC on the Major strain ε_1 axis as the boundary. This clarified that FLD can also be applied to the channel shape of a separator formed by stainless steel foil by using the strain values calculated from the microscopic hardness of the material.

5. Technology for Predicting Life Using a Fatigue Limit Diagram

Next, the stress locations and degree were predicted by finite element method (FEM) analysis to obtain the stress produced on the separator during fuel cell stack power generation. Strain values calculated from micro-hardness measurements were converted into stress and compared with the prediction results to investigate the prediction accuracy. It was thought that the separator channel life relative to mechanical degradation could be evaluated by plotting the stress produced on the separator during power generation in the fatigue limit diagram.

5.1. Analysis of Stress Produced on the Generator during Power Generation

5.1.1. Mathematical expression of stainless steel foil micro-hardness and stress

Since it is a challenge to obtain localized stress values with the conventional method, localized stress values were calculated from the micro-hardness as described in chapter 2 to understand the stress produced on the separator during power generation.

First, the relationship between micro-hardness and stress was quantified. Equation (2) was formulated by first formulating the relationship between the stress and strain stainless steel foil by Eq. (5), and then converting the strain value calculated from the micro-hardness into the stress.

The strain and stress produced on a separator with fine channels formed by ultrathin stainless steel foil can be measured by measuring the micro-hardness and the thickness reduction rate.

5.1.2. Results of stress analysis using FEM

Load fluctuation occurs due to swelling and shrinking of the MEA located between the separators and pressure fluctuation occurs due to the gas and coolant during fuel cell stack power generation. This means that stress fluctuation is constantly produced on the metallic separators of the fuel cell (Fig. 14).

Various conditions exist inside a fuel cell according to the power generation state. The MEA swells when the fuel cell interior is a wet environment, and shrinks when it is cold or dry. Load fluctuation is produced on the adjacent separators when the MEA swells and shrinks due to power generation. In addition, gas and coolant flow in the separator channels, and pressure fluctuation is also produced on the separators due to fluctuations in the gas and coolant internal pressures during power generation. FEM analysis

was performed on these load- and pressure-induced stress fluctuations on the separators (Fig. 15).

The mechanical stress produced on the metallic separators of a fuel cell during fuel cell stack power generation was quantified using the fatigue limit diagram of stainless steel foil and stress values calculated from the micro-hardness, which is a strength-related physical property value of the material, in stress locations identified using FEM.

Figure 16 shows the correlation between the FEM calculation results of the stress produced at various locations within the separator plane and the stress calculated from the hardness measured at the same locations.

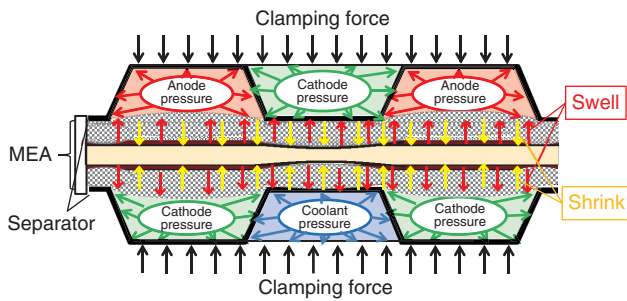


Fig. 14 Image of stress occurrence

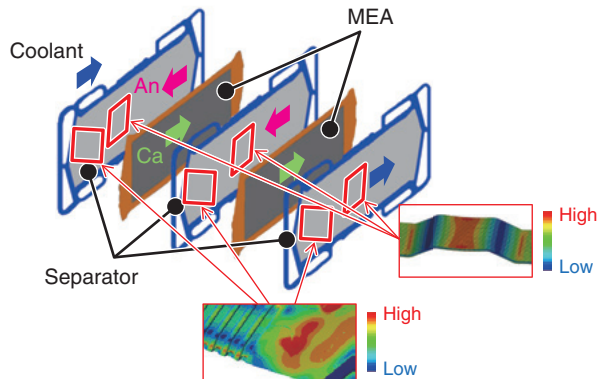


Fig. 15 Stress occurrence location by FEA

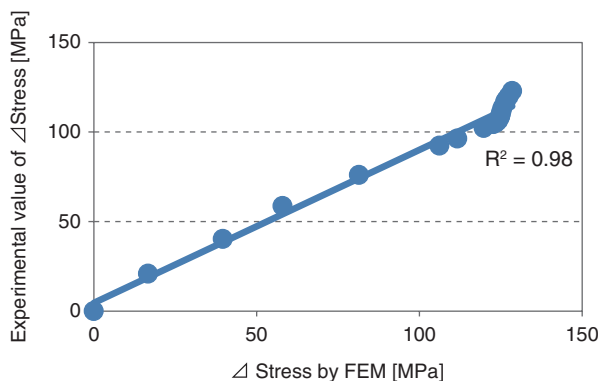


Fig. 16 Test results and correlation between FEM stress simulation and measurement results

The FEM stress calculation results and the stress results calculated from the micro-hardness were confirmed to have a high degree of correlation with a correlation coefficient of 0.98.

Based on the stress obtained by FEM and tests, application to the fatigue limit diagram was investigated to realize simple prediction of the mechanical life of a separator formed by stainless steel foil.

5.1.3. Application to fatigue limit diagram

A typical method for calculating locally produced stress is the stress measurement method using a strain gauge. However, as already described in chapter 2, it is a challenge to attach a strain gauge to the fine surface of the separator and confirm the actual behavior during power generation. It was conjectured that simple life prediction could be achieved by using the stress value calculated from the micro-hardness as the stress produced during fuel cell stack power generation, and plotting this stress value in the fatigue limit diagram.

When performing strength design, the produced stress needs to be the material strength limit or less. Therefore, the strain amplitude-number of repetitions to breaking curve (ϵ -N curve) of the ultrathin stainless steel materials applied to the fuel cell separator was obtained, and the fatigue limit diagram of the ultrathin stainless steel materials was drawn. The ϵ -N curve was used to evaluate the stainless steel foil fatigue by controlling the strain value with a pulley-bending fatigue test. The fatigue test method conformed to the JIS Z 2274 (Method of a rotary bending fatigue testing of metals)⁽¹⁰⁾ specification.

A fatigue limit diagram is a method for evaluating material fatigue strength that takes into account the influence of the average stress when fluctuating stress is received. This diagram shows the average stress on the horizontal axis and the stress amplitude on the vertical axis, and enables evaluation of the range within the yield limit and equal to or less than the fatigue limit curve as the area in which fatigue failure does not occur.

As described above, the stress values calculated from micro-hardness measurement were plotted in the fatigue limit diagram to confirm that the stress produced during fuel cell stack power generation is within the fatigue limit curve (Fig. 17).

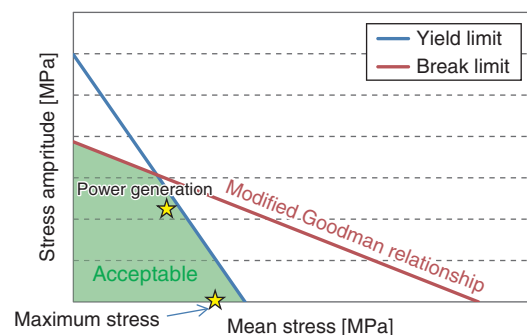


Fig. 17 Fatigue limit diagram

It was confirmed that separator channel life with respect to mechanical degradation could be evaluated by using strain values calculated from the microscopic hardness of the stainless steel foil.

6. Conclusion

This paper focused on the work-hardening characteristic of the ultrathin stainless steel foil applied to fuel cell separators. A method was developed that enables simple prediction of the available strength design range by understanding the strength-related physical property values of the materials. This method enables prediction in terms of both shape forming and life evaluation from the viewpoint of separator structure feasibility.

- (1) It was clarified both experimentally and theoretically that the stress produced on a thin metallic sheet and the deformation behavior of the sheet can be defined as a function of the strain value based on the metallic material hardness, which is a strength-related physical property value.
- (2) Measurements of localized hardness within the separator plane could be converted into stress and strain values, which confirmed that the FLD and fatigue limit diagram can be applied to the strength design of a separator channel shape formed by thin sheets.

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Detailed Structural Analysis of Pd Particles through Use of XAFS and STEM-EELS in Tandem

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ABSTRACT

In order to realize more effective use of palladium, one of the precious metals employed in catalysts for the conversion of automobile exhaust gases, it is important to possess an understanding of the detailed structure of the metal particles. However, it has not proved easy to visualize the structure of palladium particles in three dimensions using individual analytic techniques. A detailed analytic technique was therefore created through the use of X-ray absorption fine structure analysis, scanning transmission electron microscopy and electron energy loss spectroscopy, and first principle calculations in tandem. The application of this analytic method made it possible to formulate a model of palladium particles incorporating nano-scale data. The formulated model showed that palladium oxide in three different valence states displays a unique structure, forming aggregates with microscopic clusters of approximately 3 nm as the minimum units.

1. Introduction

With the current worldwide growth of environmental awareness, demand is increasing for vehicles that achieve both low levels of exhaust emissions and high levels of fuel economy. The greater thermal efficiency realized in increasing the fuel economy of internal combustion engines reduces exhaust gas temperatures, necessitating increased low-temperature activation of three-way catalytic converters (TWC). At the same time, reduction of the volume of use and the realization of more efficient use of platinum (Pt), rhodium (Rh), and palladium (Pd), the precious metals employed in TWC, is a pressing issue from the perspective of avoiding resource risk. Honda has responded to the issue of avoiding resource risk by making effective use of Pd, the supply of which is comparatively stable. From the 2012 Accord onwards, Honda's catalysts have used no Pt and half the previous volume of Rh⁽¹⁾. The importance of Pd, which supplements conversion performance in order to make up for the reduction in Rh, is therefore increasing. In comparisons of single-element Pd and Rh catalysts (containing no other additive elements), while the catalysts display identical initial performance, the NO_x conversion performance of the Pd catalysts displays a particular decline following aging. This is believed to be due to

nano-scale changes in the structure of the Pd particles, which function as reaction sites for oxidation-reduction reactions of substances in the exhaust gas. The development of catalysts in which the Pd particles retain their structure even after aging is therefore considered to be an important focus in realizing the effective use of Pd. In order to do so, it will be necessary to obtain a detailed understanding of the nano-scale structure of Pd particles. If it were possible to model and visualize the initial structure of the Pd particles in three dimensions, it would become feasible not only to conduct reaction simulations and other simulations using accurate surface structures, but it could also be expected to be possible to propose highly active Pd particle shapes that could be retained following aging.

A number of research teams have previously reported on observation of the shape of supported Pd nano-particles using electron microscopy⁽²⁾⁻⁽⁴⁾, and studies of the microscopic behavior of Pd on carriers using synchrotron radiation analysis techniques⁽⁵⁾⁻⁽⁶⁾. X-ray absorption fine structure (XAFS) obtained using synchrotron radiation offers data concerning the average coordination environment around the element being focused on in an entire test material, but because the three-dimensional coordination environment is reduced to one dimension, the method offers insufficient spatial data, and it is not easy to realize a three-dimensional

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model structure through analysis of the obtained spectra alone. Analysis using scanning transmission electron microscopy and electron energy loss spectroscopy (STEM-EELS) makes it possible to obtain data concerning local structure on the nano-scale, but it is not possible to observe lattices when the particles under observation do not have a clear crystal structure, and said particles are also often damaged by the electron beam, making it necessary to exercise adequate caution during observations. In addition, the identification of metastable phases for which reference materials cannot be obtained using EELS spectrum analysis represents a challenge. As a result of these characteristics of the methods, no reports are available concerning the accurate modeling of the particle structures formed by supported Pd.

2. Aims

Because it was difficult to model Pd particle structure using either of the analytic techniques XAFS or STEM-EELS in isolation, it was necessary to create an analytic method by coupling the two techniques. First, one-dimensional data concerning average elemental defects and periodicity were obtained using XAFS, and compared with a cluster model in order to reflect atomic-level data in three-dimensional data concerning particle size and other parameters; a structural model was hypothesized in accordance with the results of STEM observation. This was compared with STEM-EELS results using spectrum simulations based on first principle calculations, in order to verify the validity of the abovementioned structural model. The development of this coupled analytic method made it possible to model Pd particles without inconsistencies.

3. Test Method

3.1. Manufacture of Test Materials

Because EELS uses transmission electron beams, analysis using the method is hindered if other particles exist in the background of those under observation. $\alpha\text{Al}_2\text{O}_3$ (99.9%; manufactured by Kojundo Chemical Laboratory Co., Ltd.), which has a simple geometry, was therefore selected as the carrier. Palladium nitrate (manufactured by Kojima Chemicals Co., Ltd.) was employed as the Pd precursor.

Palladium nitrate corresponding to 1 wt% of Pd was supported on $\alpha\text{Al}_2\text{O}_3$, following which it was calcined in air at 450°C for three hours in order to remove the nitric acid component. Below, the particles formed by Pd when supported on $\alpha\text{Al}_2\text{O}_3$ will be referred to as “unknown Pd particles.”

3.2. Test Apparatus

Employing a PF-AR NW10A beamline (Experimental Proposal Number: 2012G586) located at the High Energy Accelerator Research Organization’s Institute of Materials

Structure Science, the manufactured test material was studied via Pd K-edge XAFS using the transmission method. A Pd foil and palladium oxide (PdO) were employed as reference materials. REX2000 Ver. 2.5.9 (Rigaku Corporation)⁽⁷⁾ was used for extended X-ray absorption fine structure (EXAFS) analysis. The structural analysis functions of MedeA (Materials Design, Inc.) were employed in studying the coordination number of the modeled unknown Pd particles⁽⁸⁾.

The HD-2700 Cs-corrected STEM-EELS 200 kV manufactured by Hitachi High-Technologies Corporation was employed for STEM-EELS, and the CASTEP module of Material Studio Ver. 5.5 (Dassault Systemes Biovia) was employed for EELS spectrum simulations based on first principle calculations⁽⁹⁾⁻⁽¹¹⁾.

4. Results and Discussion

4.1. XAFS Analysis Results

An EXAFS analysis was conducted using the following steps in order to study the change ratio of the coordination number and the distance between the coordination elements for the unknown Pd particles.

- (1) Using McKale parameters⁽¹²⁾ based on spherical wave approximation, curve fitting was applied simultaneously to the nearest neighbor Pd and the next-nearest neighbor Pd atoms in the spectrum following Fourier transform of EXAFS of the Pd foil reference material. Analytic ranges for the forward baseline for the absorption edge (the pre-edge), EXAFS extraction, Fourier transform, and Pd foil curve fitting were set to help ensure that the results of the analysis converged on valid figures in comparison with the crystal structure of Pd foil⁽¹³⁾.
- (2) Curve fitting was applied simultaneously to the nearest neighbor O, nearest neighbor Pd [Pd (1)] and next-nearest neighbor Pd [Pd (2)] atoms in the spectrum following Fourier transform of EXAFS of the PdO reference material under the conditions described in Step (1). As in Step (1), the analysis ranges for pre-edge, EXAFS extraction, Fourier transform, and PdO curve fitting were again adjusted in order to help ensure that results for strength converged on valid figures in comparison with the crystal structure of PdO⁽¹⁴⁾.
- (3) With the analysis ranges from Step (2) as the initial values, the validity of the analysis ranges was studied by applying them to a further EXAFS analysis of Pd foil.
- (4) Steps (1) to (3) were repeated until analytic results for Pd foil and PdO matched the respective crystal structures with an error of no more than 1%.
- (5) Coordination numbers were set for each coordination sphere based on the bulk coordination numbers derived from the crystal structures of Pd and PdO, and Debye-Waller factors (indicators of thermal atomic oscillation and disturbance of the crystal structure) were derived.
- (6) The Debye-Waller factors for each coordination sphere of the unknown Pd particles were assumed to be

Table 1 Fitting results for Pd foil and PdO reference

	Coordination atom	Ideal values		Results		Change ratios	
		Coordination number	Coordination distance (nm)	Coordination number	Coordination distance (nm)	Coordination number (%)	Coordination distance (%)
Pd foil	Pd (1)	12	0.274	12.0	0.274	0	0
	Pd (2)	6	0.387	6.00	0.387	0	0
PdO reference	O	4	0.203	4.01	0.203	0.25	0
	Pd (1)	4	0.305	4.00	0.305	0	0
	Pd (2)	8	0.343	7.98	0.343	-0.25	0

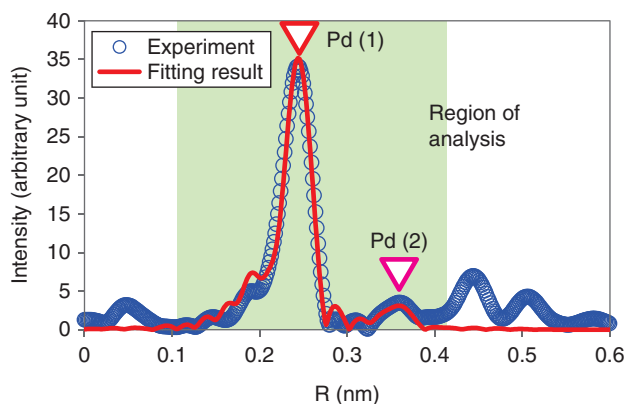
identical to values for the reference materials, and the Debye-Waller factor values for the reference materials were set for subsequent analyses.

- (7) An analysis of the unknown Pd particles was conducted using the analysis range determined in Step (4).

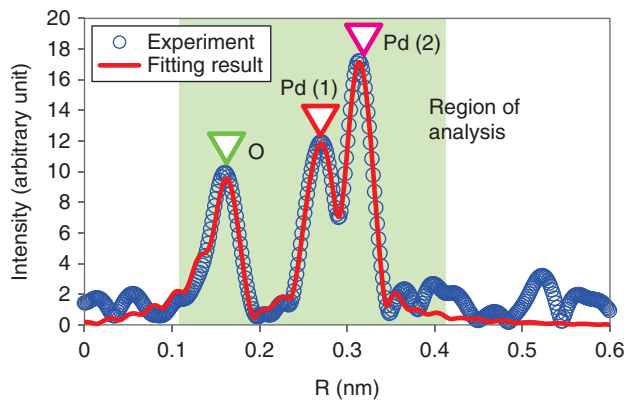
Figure 1 shows the results of curve fitting for the Pd foil and PdO reference materials. Figure 2 shows the respective crystal structures, and Table 1 shows the parameters obtained. Test values and calculated values match well in the region of analysis, and there is a good correspondence between each parameter and values obtained from the crystal structures, with errors of less than 1%. Coordination numbers were set based on bulk coordination numbers, and Debye-Waller factors were derived under the applied

analytic conditions.

Figure 3 shows results for the PdO reference material and the unknown Pd particles following Fourier transform of EXAFS data. There is no significant difference in relation to nearest neighbor O atoms, but the peak strengths for nearest neighbor Pd and next-nearest neighbor Pd are lower for the unknown Pd particles, suggesting that a Pd elemental defect or a decline in periodicity due to a disturbance of the coordination distance or a reduction in particle diameter compared to bulk. The research then attempted to obtain results that would allow modeling of the unknown Pd particles by setting values for the reference



(a) Pd foil



(b) PdO reference

Fig. 1 Radial distribution functions and fitting results

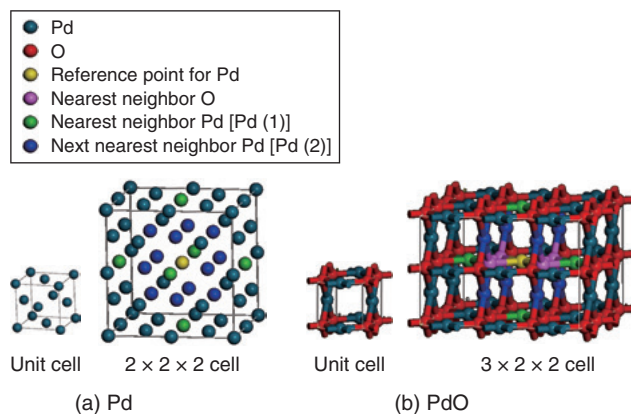


Fig. 2 Crystal structures and coordination environments of Pd and PdO

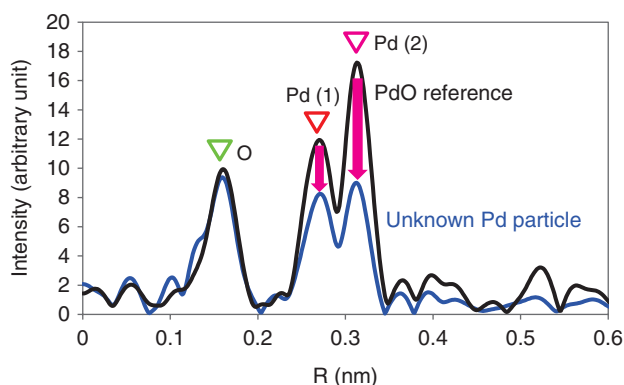


Fig. 3 Radial distribution functions for PdO reference and unknown Pd particle

Table 2 Fitting results for PdO reference and unknown Pd particle

Atom	PdO reference		Unknown Pd particle		Coordination number defect (%)	Change ratio of coordination distance (%)
	Coordination number	Coordination distance (nm)	Coordination number	Coordination distance (nm)		
O	4	0.203	3.68	0.204	7.9	-0.2
Pd (1)	4	0.305	3.27	0.304	18.3	0.1
Pd (2)	8	0.343	6.16	0.343	23.0	-0.1

material PdO as the values for the Debye-Waller factors and aggregating parameters for EXAFS strength changes as coordination numbers. Figure 4 shows curve fitting results for the unknown Pd particles, and Table 2 compares the parameters obtained from the analysis with the PdO reference material. The analysis of the unknown Pd particles was conducted in the same region of analysis as the analysis of the PdO reference material. Figure 4 shows that there is a good correlation between test values and calculation values.

The analysis results indicated that the bond length for the unknown Pd particles was the same as that for the PdO reference material, but that the coordination number for O was 8% lower, the coordination number for Pd (1) was 18% lower, and the coordination number for Pd (2) was 23% lower.

4.2. Modeling of Unknown Pd particles

The modeling of the unknown Pd particles proceeded according to the following steps, with comparison of cluster models based on the results obtained from XAFS analysis.

- (1) Multiple models with different cluster sizes were formulated based on the crystal structure model for the PdO reference material (Fig. 5).
- (2) The coordination numbers of the nearest neighbor O atoms, the nearest neighbor Pd atoms, and next-nearest neighbor Pd atoms were analyzed and plotted in relation to cluster size (Fig. 6).
- (3) Curve fitting was applied using cluster size and coordination number as functions.
- (4) Cluster sizes for each of the coordination elements

Table 3 Cluster size analysis results for unknown Pd particle

Coordination atom	Coordination number (experimental result)	Cluster size (nm)
O	3.68	3.0
Pd (1)	3.27	2.1
Pd (2)	6.16	2.1

were derived from the coefficients obtained in Step (3) and the coordination numbers obtained from the XAFS analysis results (Table 3).

- (5) The size of unit cells [Fig. 7(a)] was determined based on the amount of elemental defects obtained from the PdO reference material (Table 2), and Pd atoms were removed based on the cluster size obtained in Step (4) [Fig. 7(b)].
- (6) The structure produced in Step (5) was clustered to produce the same cluster size for O as obtained in Step (4) [Fig. 7(c)].

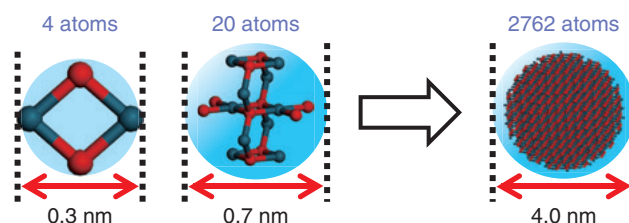


Fig. 5 Cluster models of PdO

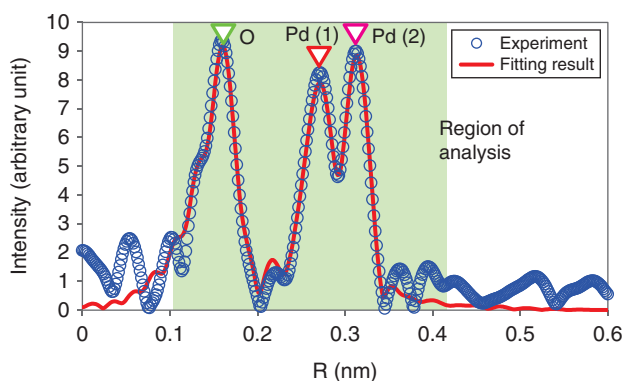


Fig. 4 Radial distribution function and fitting result for unknown Pd particle

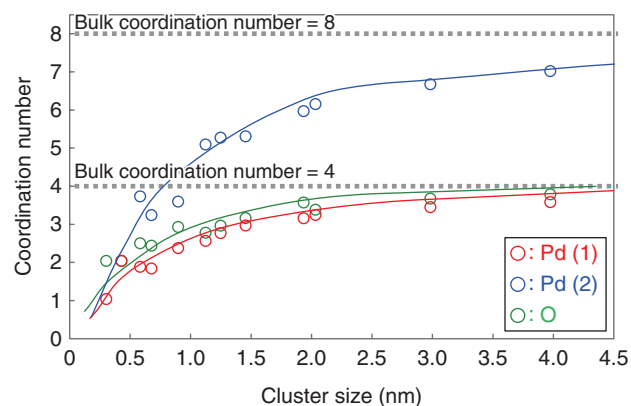


Fig. 6 Cluster size-dependence of coordination numbers for PdO

(7) Cluster models were formulated using the various combinations of positions of elemental defects in Steps (1) to (6); the respective coordination numbers were verified, and a model was selected in which there was no inconsistency with the results of XAFS analysis.

In the curve fitting of the coordination number and cluster size, Eq. (1) was employed to find the asymptotes to the bulk coordination numbers as the cluster sizes increased.

$$Y = N - A \times \text{EXP}(-X/\tau) \quad (1)$$

Here, Y is the coordination number for each element, N is the bulk coordination number, X is the cluster size, and A and τ are coefficients expressing the relationship between the coordination number and cluster size.

Figure 6 shows the results of curve fitting of coordination number and cluster size for each cluster model. The results indicate that Eq. (1) made it possible to basically reproduce the tendency of the results for coordination number and cluster size. Next, Eq. (2) was employed to derive cluster size from arbitrary coordination numbers.

$$X = -\tau \times \ln[(Y - N)/A] \quad (2)$$

When employing Eq. (2), coordination numbers derived from the EXAFS analysis were used for Y , and parameters derived from Eq. (1) were used for τ , N , and A . The cluster size estimated from the coordination number of O, X_O , was approximately 3.0 nm, and the cluster size estimated from the coordination numbers for Pd (1) and Pd (2), X_{Pd} , was approximately 2.1 nm (Table 3). The maximum cluster size was therefore approximately 3.0 nm.

Because there were numerous Pd defects and the Pd cluster size was lower than the O cluster size, modeling commenced by removing Pd. The unit cells in the PdO reference material were taken to be $2 \times 2 \times 2$ super cells, and, after Pd was removed to create areas equivalent to

Table 4 Modeling results for unknown Pd particle

Coordination atom	Coordination number (experimental result)	Coordination number (cluster model)	Change ratio of coordination number (%)
O	3.68	3.61	1.9
Pd (1)	3.27	3.19	2.4
Pd (2)	6.16	6.22	1.0

Table 5 Settings for calculation of EELS spectrum using CASTEP

Item	Setting
Functional	GGA PBE
Spin polarized	Checked (Use formal spin as initial)
Metal	Checked
Energy cutoff	610 eV
SCF tolerance	5.0×10^{-7}
k-point separation	0.04 1/Å
Pseudopotentials	On the fly
Energy range of core level spectroscopy	25 eV
Instrumental smearing	1 eV

cluster size X_{Pd} , O was clustered to produce the cluster size X_O . While referring to the structure obtained from STEM-HAADF observation (to be discussed below), the resulting clusters were randomly positioned, and a bulk model and a surface model of the unknown Pd particles were formulated [Figs. 7(d) and (e)].

The formulated cluster models corresponded well to XAFS analysis results, with change ratios of less than 2.5% (Table 4).

4.3. STEM-EELS analysis Results

Up to this point in the analyses, it had been predicted that the unknown Pd particles would display a variety of local structures as a result of elemental defects and differing cluster sizes. Because the formulation of reference materials displaying these local structures would represent a challenge, reference spectra for comparison with test results were formulated using first principle calculations, and employed to assist in making judgments in analyses of local structures. Measurements were conducted together with EELS spectrum simulations using the following process.

- (1) Multiple structures for Pd-O compounds were selected from the Inorganic Crystal Structure Database (ICSD).
- (2) Using CASTEP, EELS spectra were calculated for the K shells of the O atoms.

The unit cells employed in the calculation model were created by introducing core holes in single O atoms in $2 \times 2 \times 2$ super cells and resetting symmetry for each of the Pd-O compound unit cells selected in Step (1).

Table 5 shows the calculation conditions.

- (3) EELS spectra for O K-edge absorption were obtained from EELS spectrum measurements. Because irradiation with an electron beam over an extended period would

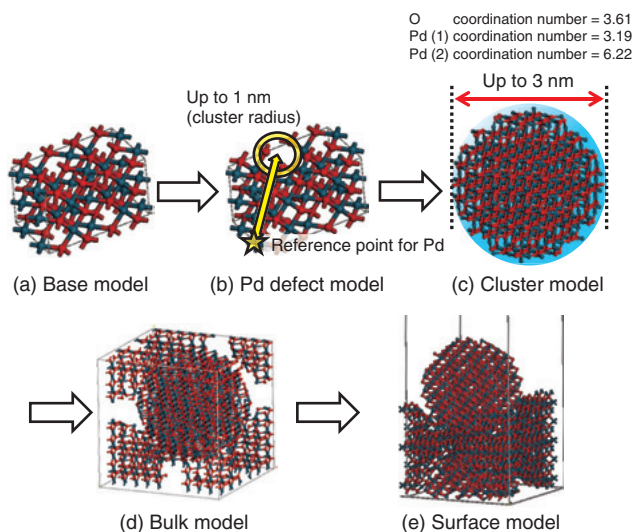


Fig. 7 Example of procedure for formulation of structural model of unknown Pd particle

change the shape of the particles, particular care was exercised in the measurements conducted to obtain EELS spectra; the authors attempted to obtain spectra within the measurement domain after adjusting the electron beam to near-field observation. When changes in the particle shape were observed following the measurements, the data was considered to be extrinsic local data, and fell outside the scope of the considerations.

- (4) Spectra obtained from tests and simulation results were compared, and candidates for local structures were identified.

Figure 8 shows a high-angle annular dark field (HAADF) image of the unknown Pd particles. Brightness in HAADF images increases in proportion with electron density, and areas in which heavy elements such as Pd are present and areas in which atomic density is high appear bright. The unknown Pd particles are distributed across a wide area, appearing to cover the $\alpha\text{-Al}_2\text{O}_3$, and a large number of particles of around 3 nm can be observed.

Figure 9 shows an example of the results obtained by identification of local structures through comparison of EELS spectra obtained in tests and the results of EELS spectrum simulations using first principle calculations. There is a good match between the test results and the results of spectrum simulations. Of 20 measurement areas, excluding six areas in which there was a change in the shape of the particles following measurements, local structures of the unknown Pd particles could be identified and broadly divided into three types as Pd_2O in five areas, PdO in six areas, and $\text{Pd}_{3.5}\text{O}_4$ in three areas.

The analysis demonstrated that the unknown Pd particles form aggregates of particles of around 3 nm, and display three types of local structure with different states of oxidation.

4.4. Validity of Model and Discussion

XAFS analysis suggested that the unit cluster size of the unknown Pd particles was approximately 3.0 nm; this matched the cluster size observed in STEM-HAADF images (Fig. 8).

In addition, a cluster model formulated to display no inconsistencies with the results of XAFS analysis showed local structures formed by different Pd oxides (Fig. 10), and this corresponded with local structure data obtained from EELS.

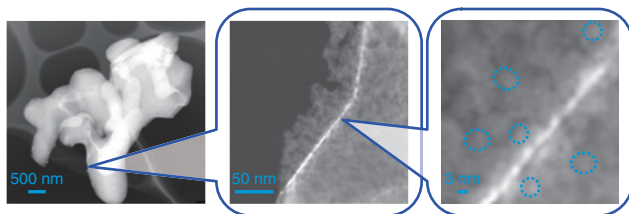


Fig. 8 STEM-HAADF image of unknown Pd particle

This model is backed up by XAFS, providing overall average structure data, and STEM-EELS results, providing local structure data, and represents a three-dimensional model of Pd immediately following support on a carrier.

This model shows that immediately following support

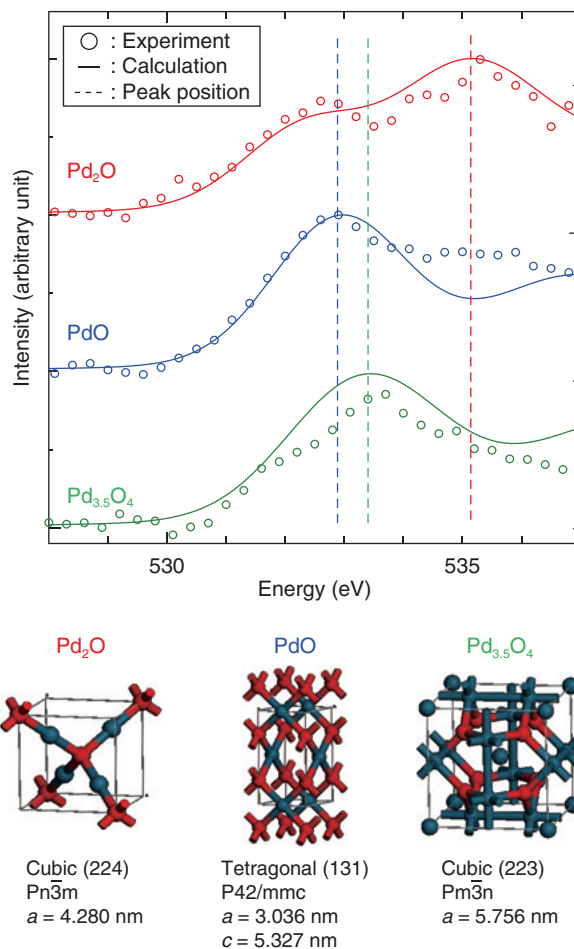


Fig. 9 EELS results for unknown Pd particle

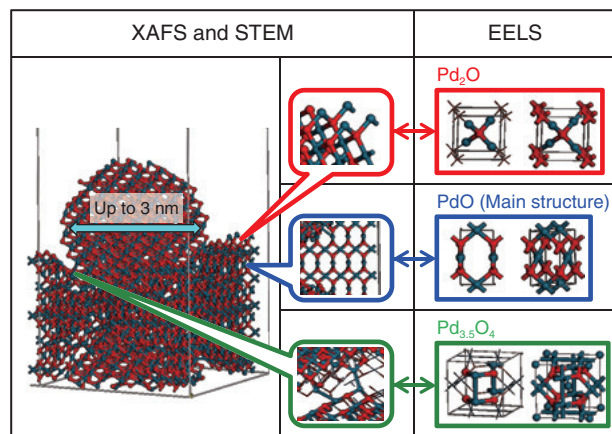


Fig. 10 Structural model based on XAFS, STEM and EELS

on a carrier, Pd forms aggregates of microscopic PdO unit clusters. Because of this, unlike bulk PdO there are gaps between clusters, and it can be conjectured that this makes it possible to actively move oxygen atoms in and out of the structure. Together with the fact that multiple valence states exist as local structures, this can be considered to indicate that immediately following support on a carrier, the Pd catalyst will display a high level of activity with regard to catalytic reactions for which oxidation-reduction reactions are essential.

This suggests that from perspectives other than maintaining the level of activity, maintaining microscopic PdO clusters that readily absorb and desorb oxygen and attempting to ensure the existence of multiple valence states as local structures represents one direction towards the effective use of Pd.

5. Conclusion

A direction for modeling unknown structures based on XAFS results obtained using synchrotron radiation has been established. The validity of the models was studied using STEM-EELS and EELS spectrum simulations based on first principle calculations in tandem. As a result, it was possible to determine a detailed structural model of unknown Pd particles supported on $\alpha\text{Al}_2\text{O}_3$, which had previously not been possible using the individual technologies.

The unknown Pd particles were shown to be structured from microscopic PdO clusters of around 3 nm, and displayed local structures produced by Pd in three different states of oxidation as a result of elemental defects within the clusters and on their surfaces.

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Research on Skutterudite-type Thermoelectric Materials for On-board Waste Heat Recovery System

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ABSTRACT

Skutterudite-type thermoelectric materials were researched with the goals of enhancing the performance of a lightweight and compact thermoelectric waste heat recovery system and scaling-up to mass production. Measures such as reducing the starting material cost and purity and increasing the manufacturing process yield were promoted, and prospects were achieved for mass production scale manufacturing of high performance thermoelectric materials (P-type thermoelectric properties $ZT > 0.8$, N-type thermoelectric properties $ZT > 1$) at approximately 1/100 the current proto-type manufacturing cost. Material nano-structuring was performed to enhance the properties, and the results showed that rare-earth metals added to skutterudite-type thermoelectric materials oxidize easily, and that the carrier concentration in the materials changes due to this oxidation, which has an acute effect on the thermoelectric characteristics. The effects of material nano-structuring in an environment that suppresses oxidation and adjustment of the carrier concentration enhanced the thermoelectric properties by 23.8%.

1. Introduction

Fuel economy regulations aimed at reducing CO₂ emissions are increasing in number and severity around the world, and further reduction of vehicle energy loss is demanded. The energy balance of a typical gasoline fuel automobile is such that only approximately 25% of the total energy is used as running energy, approximately 40% is discarded as exhaust gas, and approximately 30% is discarded as waste heat for purposes of engine cooling⁽¹⁾. Waste heat utilization technology that effectively uses this discarded heat will play an important role in future enhancement of vehicle fuel economy. In particular, technology that regenerates waste heat into electricity is attracting attention as an important technical development item in consideration of future motorization.

There are a number of methods for regenerating waste heat into electricity. Examples include an on-board Rankine cycle⁽²⁾ and an on-board Stirling engine⁽³⁾, but these systems have complex structures that are large and heavy, so practical application has not been achieved. In contrast, regarding waste heat recovery technology that

uses thermoelectric conversion, Honda has previously reported the verification results of a lightweight and compact thermoelectric system concept with a characteristic vacuum structure⁽⁴⁾. In addition, many automobile makers are also working to develop such a system⁽⁵⁾⁻⁽⁹⁾. However, the thermoelectric materials used in recovery systems that use thermoelectric conversion typically have lower conversion efficiency compared to other waste heat recovery methods⁽¹⁰⁾, and this is thought to be a factor delaying commercialization of these systems. In addition, a lightweight and compact system design and assembly technology that includes modularization are needed to create a thermoelectric conversion system that is suitable for mass production. Moreover, thermoelectric material development to enhance properties and scale-up production, and system integration that takes into account the energy balance of a vehicle are also important. Furthermore, there are various development issues such as the need to secure durability and reliability. This paper focuses in particular on thermoelectric material development among these various issues.

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2. Overview of Thermoelectric Material Research

When a temperature difference is applied to a material, an electromotive force is generated due to the Seebeck effect. Materials for which this effect is particularly pronounced are called thermoelectric materials. Thermoelectric materials include P-type materials in which electricity flows from the hot side to the cold side, and N-type materials in which electricity flows from the cold side to the hot side. Electric power regeneration can be achieved by connecting thermoelectric materials in series via electrodes. The basic concept of an on-board thermoelectric conversion system is a structure that generates electric power by using the high temperature waste heat from the exhaust gas and the coolant used for engine cooling to apply a temperature difference to thermoelectric materials.

Equation (1) below expresses the energy conversion efficiency η for regenerating input heat energy into electricity.

$$\eta = \left(\frac{T_h - T_c}{T_h} \right) \left(\frac{\sqrt{1 + ZT} - 1}{\sqrt{1 + ZT} + T_c/T_h} \right) \quad (1)$$

Here, T_h (K) is the temperature on the hot side and T_c (K) is the temperature on the cold side of the thermoelectric element. ZT is expressed by Eq. (2) below as a figure of merit that indicates the thermoelectric material properties.

$$ZT = \left(S^2 \sigma / \kappa \right) T \quad (2)$$

Here, S is the Seebeck coefficient, σ is the electric conductivity, and κ is the thermal conductivity. In addition, κ can be expressed as follows by the carrier thermal conductivity κ_c , which uses carriers such as electrons and holes as thermal energy propagators, and the lattice thermal conductivity κ_{ph} , which uses phonons, or lattice vibration, as energy propagators.

$$\kappa = \kappa_c + \kappa_{ph} \quad (3)$$

From Eq. (2) and Eq. (3), the thermoelectric material ZT is expressed as follows.

$$ZT = \left[S^2 \sigma / (\kappa_c + \kappa_{ph}) \right] T \quad (4)$$

Enhancement of ZT involves increasing the numerator $S^2 \sigma$, which is called the power factor (PF) or reducing the denominator $\kappa_c + \kappa_{ph}$ in Eq. (4). However, S , σ and κ_c are all physical property values derived from the carrier conduction, so for example, increasing the numerator induces an increase in the denominator. This complex interrelation makes it a challenge to simply enhance

ZT . In contrast, κ_{ph} is an independent parameter derived from phonon propagation, so methods that reduce κ_{ph} are typically used as the thermoelectric material design method. As a concrete example of such a method, B. Poudel et al. performed dry mechanical milling on BiTe-type thermoelectric material powder and then used the spark plasma sintering method to manufacture a nano-structured sintered compact. The results showed that thermoelectric properties can be enhanced by reducing $\kappa_{ph}^{(11)}$, and since then there have been many reports of research on nano-structured thermoelectric materials.

This development targeted gasoline vehicles, and focused on skutterudite-type thermoelectric materials which, among materials suited to the waste heat recovery temperature of a gasoline vehicle, are inexpensive, high performance, and do not contain harmful elements. MX_3 compounds ($M = \text{Co, Rh, Ir}$; $X = \text{P, As, Sb}$) have a skutterudite-type crystal structure, and CoSb_3 in particular is known to exhibit thermoelectric characteristics (Fig. 1). In addition, the $\text{RM}'_4\text{X}'_{12}$ ($R = \text{La, Ce, Eu, Yb}$; $M' = \text{Co, Fe}$; $X' = \text{Sb}$) with the atoms filled in the cage part in the figure reduces κ_{ph} through a special lattice vibration called a rattling effect, so it is known to exhibit high thermoelectric properties in the medium to high temperature range⁽¹²⁾. The yellow, green and blue atoms in the figure indicate Sb, M' and R , respectively.

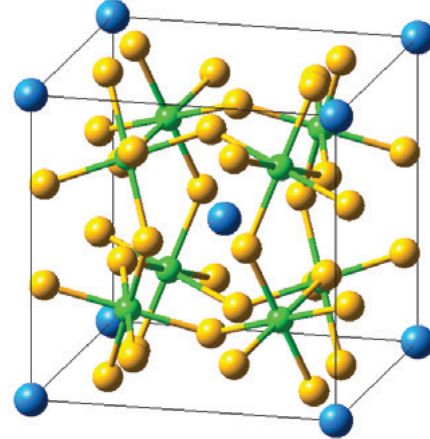


Fig. 1 Schematic illustration of skutterudite-type crystal structure

3. Research Goals

Most thermoelectric materials thus far, including skutterudite-type thermoelectric materials, have been small-scale synthesized materials on the level of several tens of grams, such as those described in academic papers, and these have all been synthesized using high-purity reagents. As mentioned above, thermoelectric-based waste heat recovery technology has not yet been used for mass

production by any automobile maker, so of course mass production technology for these materials also has not been established. Therefore, this development first considered the purity of the starting materials that can actually be obtained from suppliers, set a tentative target value of 50 million $2 \times 2 \times 2$ -mm thermoelectric elements per year (equivalent to an annual production scale of 3000 kg), and verified a manufacturing process that enables mass production. Depending on the amount of thermoelectric materials used in the thermoelectric generation system, this target quantity is enough to support a production scale of approximately 100,000 units/vehicles per year. In addition, approaches have also been reported for enhancing the properties of skutterudite-type thermoelectric materials using the abovementioned nano-structuring, but these studies have all been limited to small scale (laboratory scale) verification^{(13), (14)}. As mentioned above, low conversion efficiency is an issue for on-board thermoelectric waste heat recovery systems, so the effects of nano-structuring on the developed material were confirmed and analyzed in detail in consideration of mass production scale manufacturing.

4. Mass Production Scale Manufacturing of Thermoelectric Materials

The thermoelectric material manufacturing process begins from selection of the starting materials, and proceeds through the dissolution synthesis, homogenization, powderization, sintering, and dicing processes. Laboratory scale synthesis consists of encapsulating high-purity reagents as the starting materials in an ampule-type quartz tube and performing melting heat treatment. However, mass productivity was taken into account, and the manufacturing process for the developed material was streamlined by applying gas-atomization in place of synthesis, homogenization, and powderization.

The starting materials assumed for mass production contain various impurities, so the influence of these impurities was confirmed. A slight drop in performance was observed when the high-purity reagents were replaced with starting materials that contain impurities, but the performance was restored for the most part by finely adjusting the material composition again (Fig. 2).

Granulation used gas-atomization as mentioned above, and this enabled the suppression of the effects of inter-lot variance (composition changes) during material synthesis, which is unavoidable with small-lot manufacture. When synthesis is performed by gas-atomization, a powder grain size distribution exists. Therefore, the influence of powder grain size on performance was confirmed. The results showed that when powder with a grain size of 20 μm or less is used, the oxygen concentration increases and the thermoelectric material properties drop compared to when using powder with a larger grain size (Fig. 3). The reason for this drop in conversion efficiency when

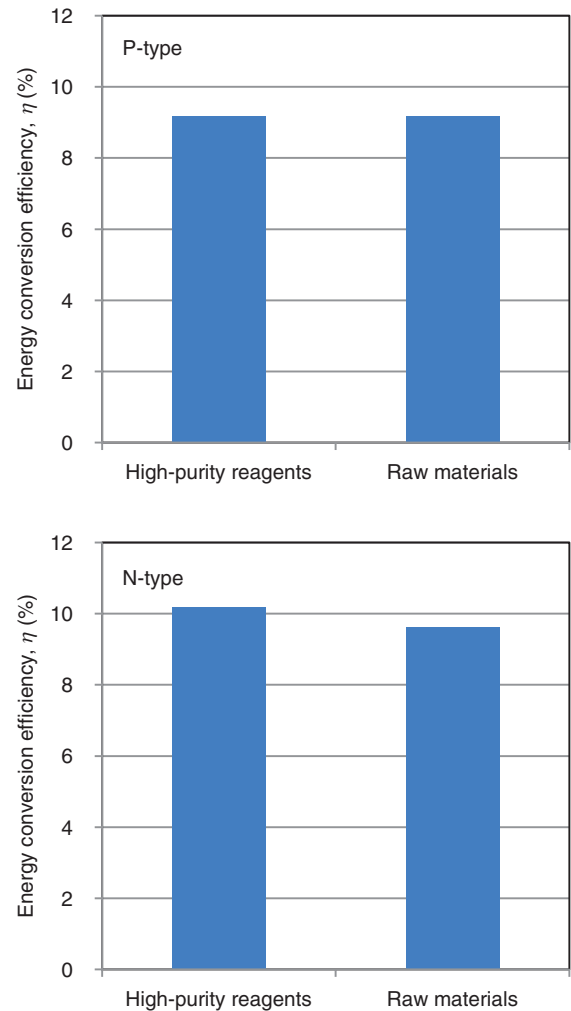


Fig. 2 Thermoelectric performance versus purity of starting materials for using gas-atomization

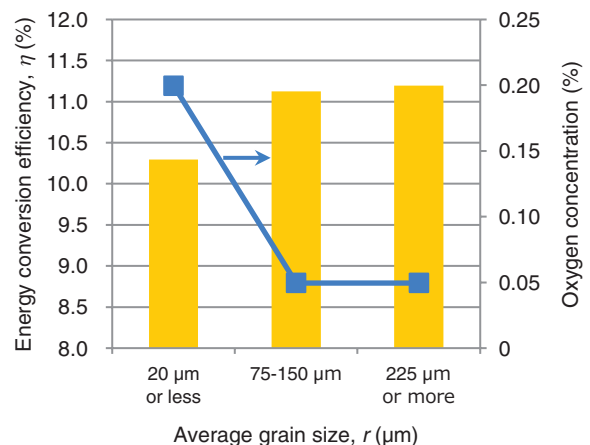


Fig. 3 Thermoelectric performance and oxygen concentration versus gas-atomized powder radius

the oxygen concentration increases is described in the following section. Much powder with a grain size of 20 μm or less was present in the initial manufacturing process condition, but subsequent optimization of the manufacturing process parameters, such as the injection pressure and nozzle hole diameter, enabled us to reduce the amount of powder with a grain size of 20 μm or less.

To enhance the tact time in the next process, sintered compact manufacture, it is necessary to increase the diameter of the sintered compact. Here, the key is how to realize the manufacture of homogeneous sintered compact with a large diameter. Uneven firing of the sintered compact becomes an issue as the diameter increases, so it is important to ascertain the maximum sintered compact size that can be homogeneously fired. This development established technology for manufacturing a $\phi 200$ sintered compact using the spark plasma sintering method. It was assumed that nano-structured powder would be sintered as described in the following section, so the spark plasma sintering method was selected to enable quick sintering in a short time so that as little grain growth as possible occurred in the nano-structured powder during the sintering heat treatment. Figure 4 shows the results of evaluating the

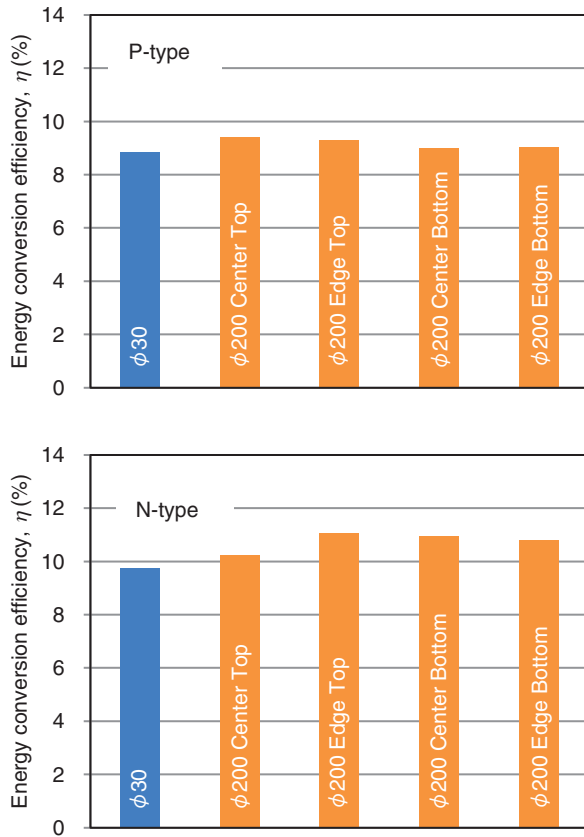


Fig. 4 Energy conversion efficiency of P-type and N-type sintered skutterudite thermoelectric materials 200 mm in diameter

thermoelectric properties at various locations of the sintered compacts. There was little or no difference in properties at any of the locations, indicating that good, homogeneous sintered compacts can be manufactured.

This series of thermoelectric material developments enabled scale-up manufacture of a thermoelectric material with thermoelectric properties that exceed those of a $\phi 30$ sintered compact, as shown in Fig. 5. The main factor enhancing performance was a vast reduction in the amount of powder used that had a grain size of 20 μm or smaller (Fig. 3). The obtained sintered compact was sliced, polished and then diced to create thermoelectric elements with a square pillar shape, and it was verified that the material can be processed to a size with a cross-sectional area of 1 mm $\times$ 1 mm. When this manufacturing process is used to produce the tentative target quantity of material, the effects of reducing the starting material cost, increasing the yield, and economies of scale due to mass production enabled us to lower the cost to approximately 1/100 that of the materials supplied as a prototype.

When this new thermoelectric element is installed in the thermoelectric generation system currently being developed, and the US 5 cycle (U.S. automobile fuel economy labeling method) driving mode average value is estimated through the method used in past studies⁽⁴⁾, a fuel economy enhancement effect of approximately 1% can be expected for the system with the greatest cost-effectiveness, including the early warming effect during engine start. In addition, enhancing the input heat amount, the heat transfer efficiency, and other thermoelectric system properties can further increase the waste heat recovery performance. Effects can also be anticipated such as the use of waste heat to supply the power consumed by always-on sensors, for which demand is expected to increase due to self-driving and other applications in the future.

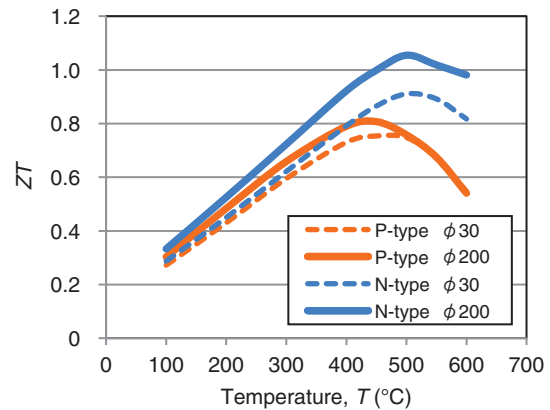


Fig. 5 Resultant ZT properties for developed skutterudite thermoelectric materials that can be mass produced

5. Enhancement of Properties by Material Nano-structuring

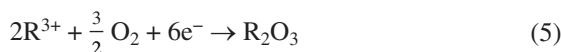
Gas-atomization was used to prepare several kg of N-type skutterudite-type thermoelectric materials with rare-earth metals added, and then a planetary ball-mill was used to perform nano-structuring.

Figure 6 shows electron back scatter diffraction (EBSD) images of the material structure before and after ball-milling. The average crystal grain size of the gas-atomized powder is approximately 15 μm , and the average crystal size after nano-structuring is approximately 300 nm. This shows that pulverization by a planetary ball-mill produces a nano-structure with a grain size approximately 1/50 and a grain volume approximately 1/125000 those of gas-atomized powder.

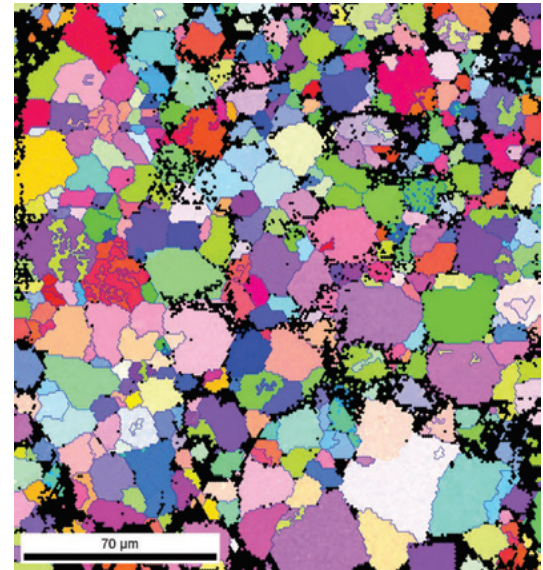
Figure 7 shows the temperature dependence of each physical property value that characterizes the thermoelectric properties indicated by Eq. (4). The primary goal of nano-structuring is to reduce only κ_{ph} and enhance performance. However, physical property values other than κ_{ph} also change before and after nano-structuring, such as for example the change in σ shown in Fig. 7(a). The two likely factors behind this drop in σ are thought to be the effect of the inhibition of electrical conduction by nano-structuring and the effect of the change in the carrier concentration. Regarding the effects of nano-structuring, Fig. 8 shows the test results of σ dependency on crystal grain size for different milling conditions. Ball-milling greatly reduced σ , but no clear dependency of the milled material σ on the crystal grain size was observed thereafter.

Regarding the influence of the change in carrier concentration, the oxygen concentration was confirmed to increase by 0.18 wt%O as shown in Table 1. The tests were performed using a controlled atmosphere glove box or similar facility, but this increase is thought to be due to oxidation by residual oxygen that could not be removed. The material structure was observed using transmission electron microscopy (TEM) and energy dispersive X-ray spectrometry (EDX) to confirm the oxide format. The results showed that rare-earth oxides had formed in the materials and were unevenly distributed along the crystal grain boundaries as shown by a, b and c in the figure (Fig. 9 and Table 2).

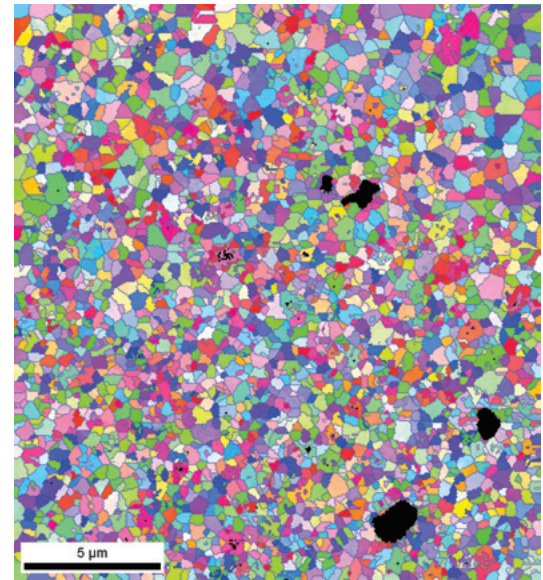
The rare earth oxidation reaction is expressed by the following equation.



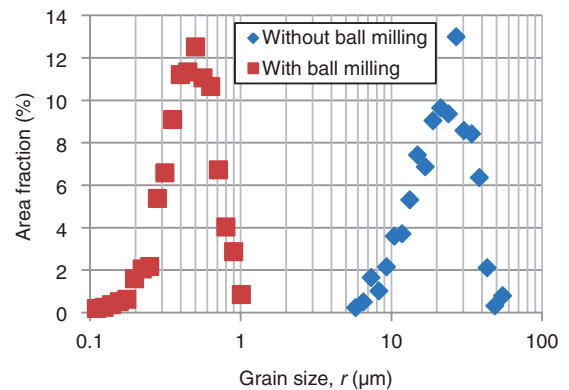
This reaction equation (5) shows that electrons are consumed when rare-earth oxides are generated. This material is an N-type material, so when electrons are consumed, the carrier concentration in the material decreases, or in other words, σ drops. This is presumed to be the reason for the drop in σ shown in Fig. 7(a).



(a) EBSD images without ball-milled material



(b) EBSD images with ball-milled material



(c) Grain size distribution

Fig. 6 EBSD images of N-type skutterudite thermoelectric materials, and resultant grain size

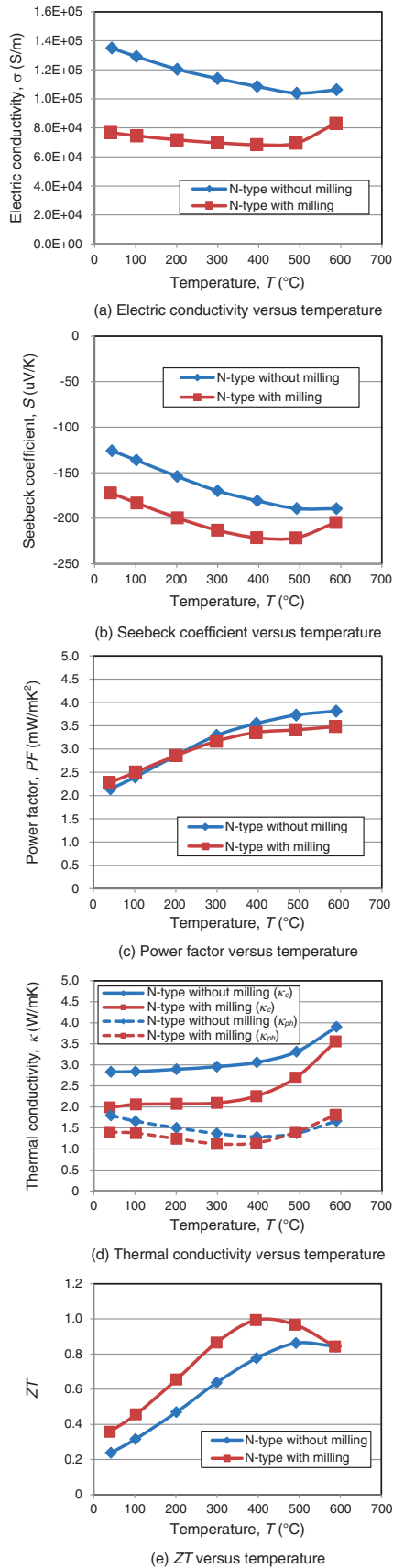


Fig. 7 Temperature dependence of thermoelectric properties derived from ball-milling process

Table 1 Oxygen concentration of thermoelectric materials before and after ball-milling treatment

	Before milling	After milling
Oxide content	0.14	0.32

(wt%O)

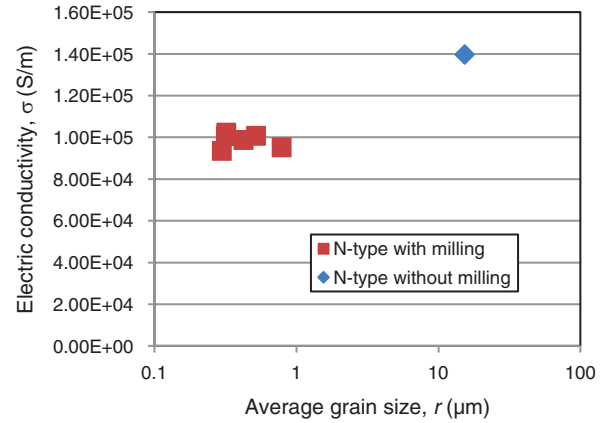


Fig. 8 Electric conductivity versus average grain size at RT. Except for the data with an average grain size of 15.28 μm, the other data were derived after several ball-milling processes

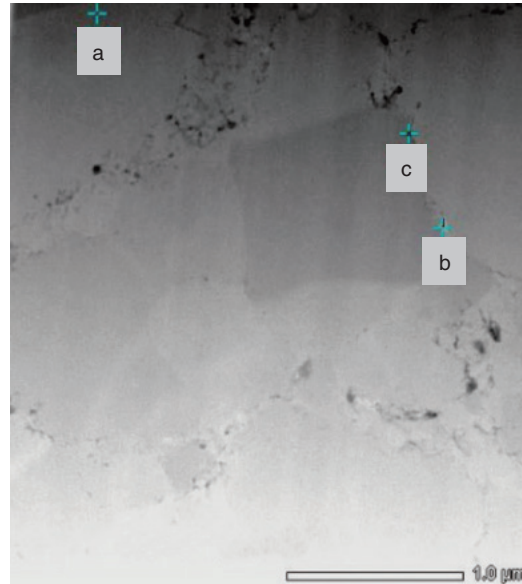


Fig. 9 TEM image of ball-milled materials microstructure

Table 2 Chemical composition analysis results for each point derived by EDX equipped with TEM

Point	O	Yb	Fe+Co	Sb	Others
a	0.8	0.8	34.3	63.1	Bal.
b	31.1	23.9	17.8	27.2	Bal.
c	19.3	2.8	21.9	44.9	Bal.

(atom%)

Equation (4) shows that ZT decreases when only σ drops, but the absolute value of S increases after ball-milling as shown in Fig. 7(b), so as a result $S^2\sigma$ is maintained [Fig. 7(c)]. Material general theory holds that as the carrier concentration in the materials increases, σ increases and S decreases, so $S^2\sigma$ indicates a certain maximum carrier concentration value⁽¹⁵⁾. The results of this development are thought to be that because while the carrier concentration changed, it remained within the ideal $S^2\sigma$ range. However, according to the Wiedermann-Franz law ($\kappa_c/\sigma = LT$: L = Lorentz number, T = temperature), σ and κ_c have a proportional relationship. This means that a drop in σ results in a reduction in κ_c , so ZT increases according to Eq. (4). Therefore, the change in the carrier concentration due to this process has the overall effect of increasing ZT .

Next, κ_{ph} was examined. When performing material design of thermoelectric materials, it is important to reduce κ_{ph} as mentioned above, and the two methods of adding rare-earth metals (La, Yb, Ce, etc.) and material nano-structuring are used for skutterudite-type thermoelectric materials. Rare-earth metals are added to induce special lattice vibration to produce a rattling effect that reduces κ_{ph} . However, as the materials oxidize, the added rare-earth metals form oxides, so the amount of added rare-earth metals in the material lattice decreases and κ_{ph} increases. In contrast, nano-structuring reduces κ_{ph} , so the balance between rare-earth metal addition and material nano-structuring determines the amount of increase or decrease in κ_{ph} . The temperature dependence of κ_{ph} shown in Fig. 7(d) indicates that κ_{ph} decreases after milling as a result of both of these methods. This indicates that the effect of material nano-structuring exceeds the increase in κ_{ph} due to rare-earth oxide formation.

These results show that when nano-structuring is performed on skutterudite-type thermoelectric materials, κ_{ph} is reduced as initially intended, and the thermoelectric properties ZT are also enhanced as shown in Fig. 7(e) as a compound effect of κ_c also decreasing. Here, the ZT area value increased by 23.8% (11.3% in terms of energy conversion efficiency) over the range from 100°C to 600°C. Scale-up of this nano-structuring is currently being investigated.

6. Conclusion

This paper reported an approach to creation of a manufacturing process that enables mass production of skutterudite-type thermoelectric materials for an on-board thermoelectric waste heat recovery system. This research achieved prospects for creating a material manufacturing process with an annual production scale of 100,000 units/vehicles that enables a reduction in the manufacturing cost to approximately 1/100 the prototype cost. In addition, nano-structuring was performed on skutterudite-type thermoelectric materials in consideration of mass production scale manufacturing, and the results showed that the effects

of nano-structuring and changing the carrier concentration enhanced the thermoelectric properties by 23.8%.

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Editorial Afterword

The Honda R&D Technical Review commenced publication of its Web edition five years ago. The number of registered users of the Web edition has now exceeded 22,000. Prior to the commencement of publication on the Web, the total publication volume for the Technical Review was around 8,000 copies, including both the Japanese and English editions. The significant increase in the number of readers with the publication of the Web edition is proof that the journal has attracted many readers throughout the world beyond the confines of Honda, which is a source of great satisfaction to us.

During the five years since the commencement of Web publication, one thing that has impressed me through my activities as a member of the Editorial Committee has been a rapid increase in the variety of technological areas and technological methods treated in the Technical Review. In the process of peer-reviewing, I come upon many more papers that I cannot fully understand based on a single reading, and that force me to consult a variety of references. This can be considered evidence of today's rapid evolution of modes of transportation and the diversification of technologies in line with new social trends.

On the other hand, consideration of history in a variety of areas shows us that flourishing diversification is often narrowed down by a process of culling. This also applies to technology. As a consequence, it is critical for engineers to maintain a focus on the essential aspects of developing technologies. It is my hope that younger engineers will develop the ability to perceive these essential aspects by broadening their scope to take in areas of technology beyond their own specialization, and I will be delighted if the Honda R&D Technical Review is able to provide them with opportunities to do so.

Tomoo SHIOZAKI

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ISSN 1881-6223

Published by:
Honda Motor Co., Ltd.
2-1-1, Minami-Aoyama, Minato-ku,
Tokyo 107-8556, Japan

Edited by:
Honda R&D Co., Ltd.
Honda R&D Technical Review Editorial Committee

Printed:
Mar. 1, 2017

Honda R&D Co., Ltd.