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Technical Review



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Comment on the Cover

Through Honda's products, I want to respond to the hearts and minds of a whole new generation of people.

I want Honda to continue to be the cool, cutting-edge company it has become – one that's always in the global limelight, and never afraid to meet the expectations of the people of the world.

Honda's 'new stage' is being formed right now, by us, as we speak.



Tsubasa Tsukasaki
Asaka R&D Center

Development of New Model CIVIC Sedan

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Abstract

The new CIVIC was developed under the theme “a vehicle to provide unique sensations and excitement.” The developers were aiming for a completely new driving sensation and appeal. The power plant was newly developed, the platform and suspension were redesigned and the vehicle’s fundamental performance was enhanced in an innovative way. Moreover, its unique design, never before seen in the sedan concept, was combined with a people-friendly package.

Factors that are often difficult to balance – safety and novel exterior design, comfortable running performance and stability, advanced interior design and ease of use, high output and fuel economy – were mastered one by one, setting new standards and using innovative technologies to create not just a vehicle with originality, but one that is highly practical. In other words, the developers have come up with a high quality, next generation compact sedan with the never-before-seen CIVIC sensation.

1. Development Concept

The original CIVIC launched Honda’s automobile business, and people the world over see the CIVIC as a symbol of progress. Our mission is to develop that image further using the new 8th generation CIVIC. The CIVIC is currently being sold in 160 countries worldwide, and the roles it plays in each country were due for a major change.

In this Internet age, global information can be obtained quickly. People demand products (and not just vehicles) with unique design that provide innovative performance and package.

To respond to this type of customer demand, in the new CIVIC we chose the development concept, “a vehicle that provides unique sensations and excitement,” to express how its unique presence and innovative performance far surpass conventional concepts.

2. Global Formation

For the American market, the CIVIC was given a sporty appearance, to meet the needs of the market’s largest segment, Generation Y (now around 25 years old) which has replaced Generation X (now around 40 years old) and the baby boomers (now around 55 years old). For the markets in Japan and the rest of Asia, the median age of sedan buyers is slightly higher, and thus the CIVIC has been given a more luxurious feel (Fig. 1). In addition, the power plants are configured to meet regional needs, with 1.6L, 1.8L or 2.0L gasoline engines, as well as IMA and CNG (natural gas) vehicles that appeal to those who are environmentally conscious (Table 1).

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Fig. 1 Worldwide style variations

Table 1 Worldwide power plant variations

Power plant variation	North America		C., South America	Asia	EC	Japan
	CIVIC	ACURA CSX	CIVIC	CIVIC	CIVIC	CIVIC
1.8 L 140 Ps 5AT	●		●	●	●	●
1.6 L 125 Ps 6MT				1.6/1.8L	5AT 5/6MT	
2.0 L 155 Ps 5AT		●		●		
IMA 110 Ps CVT (Motor power included)	●			●	●	●
CNG 115 PS 5AT	●					

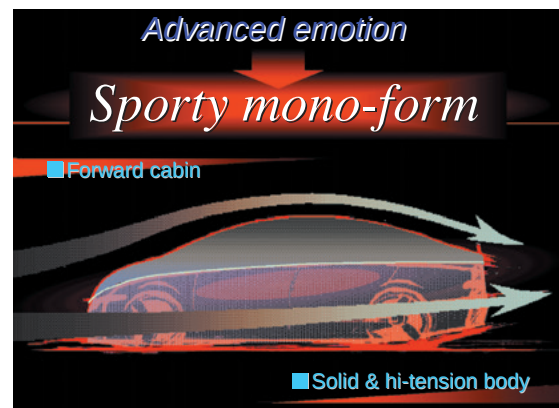


Fig. 2 Exterior design concept

3. Exterior

3.1. Design

The “sporty mono-form” exterior design concept frees the vehicle from the conventional sedan image and adds additional style. While carefully maintaining its current ergonomic features, such as head clearance, trunk room and visibility, the new CIVIC also provides best-in-class collision performance with the unique new presence of its mono-form, flowing upper body; and a solid, strong underbody (Fig. 2).

3.2. Dimensions

In the upper body, the front glass was moved forward 260 mm while compressing the overall height, creating a mono-form that also provides high-level collision performance. Changes to the body frame structure and a wider application of high tension steel both enhanced collision performance and achieved a new look. In the underbody, the wheel base was extended 80 mm, and the tread was widened. 16-inch wheels were made the standard, and the wheel arch gap was made the smallest found in a compact-class car. This produced a lower body that is strong and solid (Fig. 3).

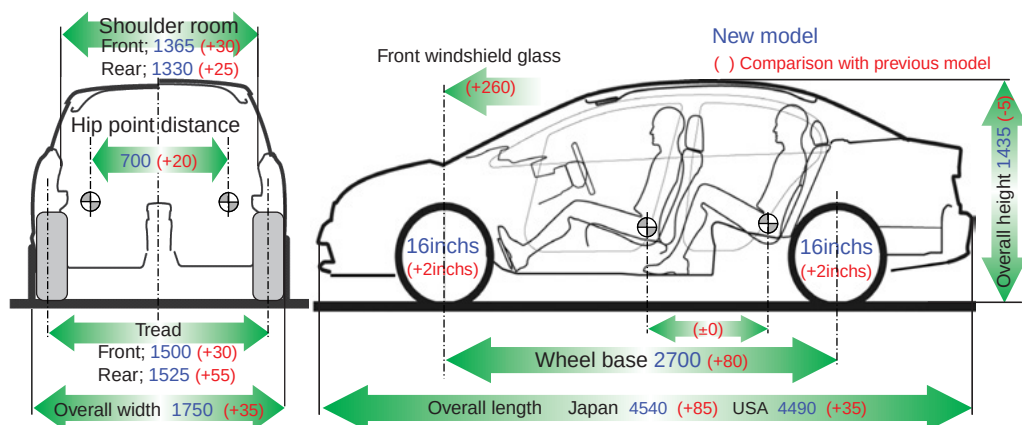


Fig. 3 Basic dimensions (mm)

To highlight the underbody's solidity and sporty high-quality feel, high caster front suspension geometry was used to increase steering camber, and the tire was offset from the wheel arch (Fig. 4).

Vehicle stability was enhanced by implementing such measures as bending the outer panel's rear wheel arch flange (Fig. 5), and the gap between the wheel arch and upper edge of the front and rear tires was narrowed to 50 mm, the smallest in the global compact-car class.

3.3. New Link Mechanism Flat Wiper

The front window was moved forward to achieve the mono-form design, expanding the window glass surface. To ensure adequate wiper coverage as well as to enhance the wipers appearance and quality, a new style of wiper, linked opposite-type flat wipers, was developed (Fig. 6).

Conventionally, opposite-type wipers either have a large gap between the stationary positions of the left and right wipers, or have pantograph links to avoid interference during operation. The wipers developed during this project combine a proximity pivot, output link and connection link to create a system that controls the speed of the wiper on the front passenger seat side. This achieves a low stationary position and an elegant appearance.

The design arm and flat blade were used to enhance the wipers elegant appearance with an integrated feel. In consideration of maintenance costs, the wiper blade was made a single replaceable structure (Fig. 6).

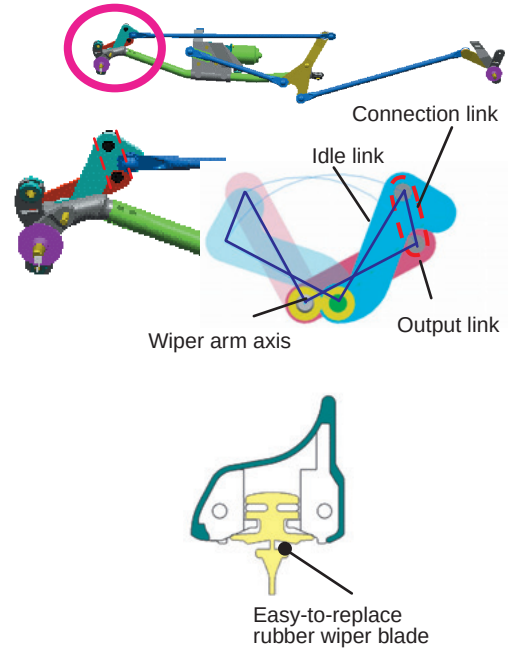
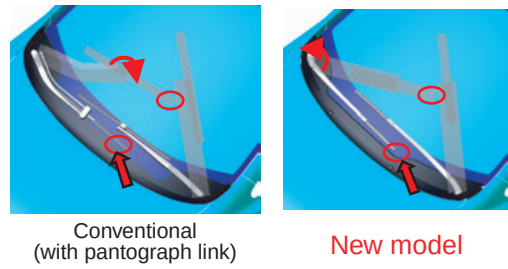


Fig. 6 Flat wiper with new linked system

4. Interior

4.1. Design

To realize the concept of a “multiplex cockpit”, a user-friendly interface with separate functions was required. To achieve this, driving, ease of operation and ease of use were studied, focusing on the driver's needs, and a comfortable, functional cabin space with a new sense was created (Fig. 7).

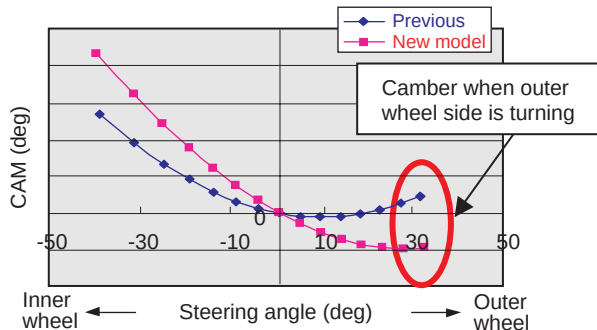


Fig. 4 High caster front suspension geometry

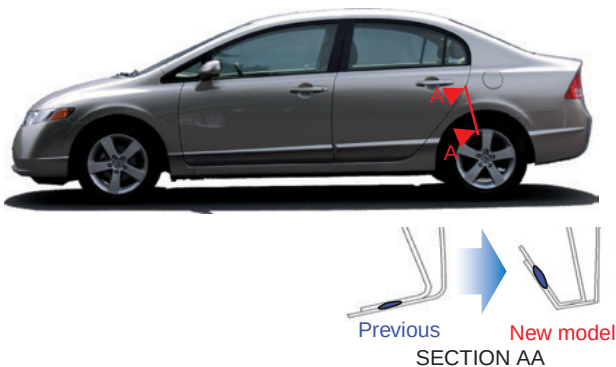


Fig. 5 Rear wheel arch bend



Fig. 7 Interior design concept

4.2. Meters

When designing the meter area, the features of the mono-form exterior were put to practical use. Meter functions are divided between information that is a priority during driving (the speedometer) and other information. After analyzing the location of monitors that are easy to see lettering size and colors, it was decided to create a dual-layer configuration to provide enhanced visibility and a futuristic look (Fig. 8).

4.3. Control System

The elliptical and compact steering wheel, modularized new compact shift lever and Z-arm hand brake are all located in close proximity. This layout provides for smooth and user-friendly operation (Fig. 9).

4.4. Console

Because the control system was centralized, it was possible to provide large, easily accessible console storage and door pockets, which are especially useful since these are usually accessed blindly while driving. The center console includes a cup holder that can hold two large travel mugs, and large storage areas both in front of and behind the gear shift lever that do not interfere with shifting. In addition, below the sliding armrest there is a storage space that can hold 25 CDs (Fig. 10).

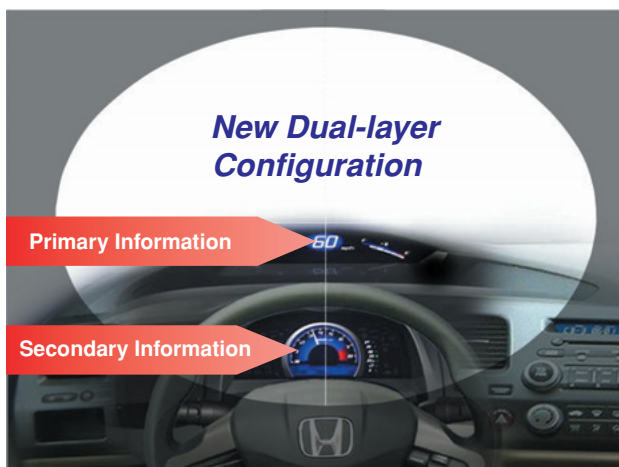


Fig. 8 Meter arrangement



Fig. 9 Control package

4.5. Seat

The seats provide the feeling of being enveloped “in the seat,” rather than just sitting “on the seat.” A dual layer structure increases the “soft solid” portion of the seat that comes in contact with the passenger, while the “hard solid” under layer provides support. This dual solid structure provides high quality and a sporty feel. The internal structure comprises a new highly rigid frame and a plastic self-adjusting plate that fits to the body. This structure makes it possible to drive for long periods without feeling fatigued (Fig. 11).

4.6. Driving Position

One aim in developing the new CIVIC was to create the “best (driving) position for everyone.” A comfortable driving position was achieved by standardizing the tilt-telescoping steering wheel, a sliding armrest, easy-to-use lever seat height adjuster and organ-type acceleration pedal, features available for all grades (Fig. 12).

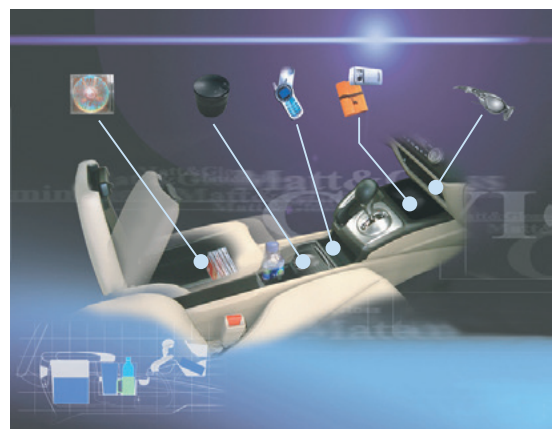


Fig. 10 Center console storage

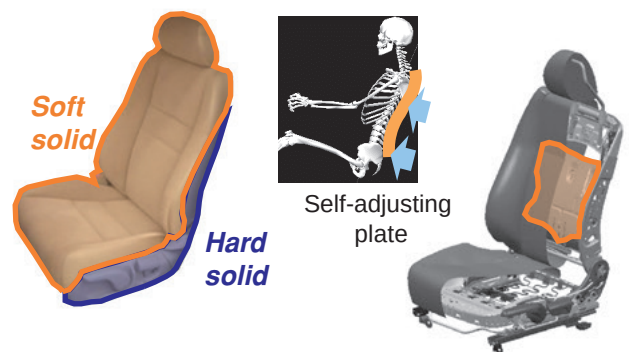


Fig. 11 Dual solid sheet

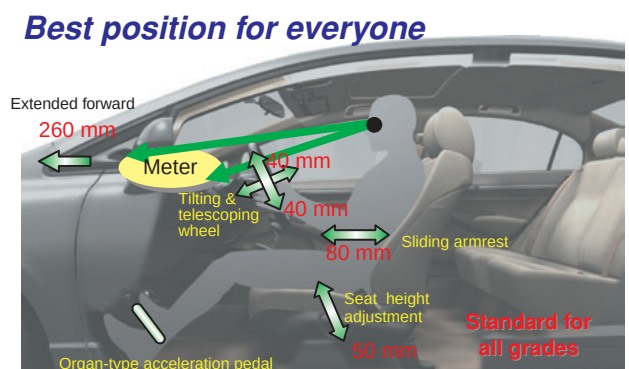


Fig. 12 Mechanics of driving position

5. Power Plant

A new 1.8L (for some regions, 1.6L) i-VTEC engine equal to the high output of engines one class above while having best-in-class fuel economy was developed⁽¹⁾ along with an ultra-compact 5AT, and 5MT (6MT) transmissions, all with enhanced drive feel.

By combining these features with the newly adopted drive-by-wire control, a sporty driving feel was developed, with good acceleration feel and a quick accelerator response (Fig. 13).

The 2.0L i-VTEC engine installed in the Accord has been used for some regional needs. An IMA vehicle with enhanced fuel efficiency and battery performance, and a CNG vehicle that uses natural gas for fuel were prepared to demonstrate environmental performance and innovative technology. These various uses were further proof of their strength and versatility (Fig. 14).

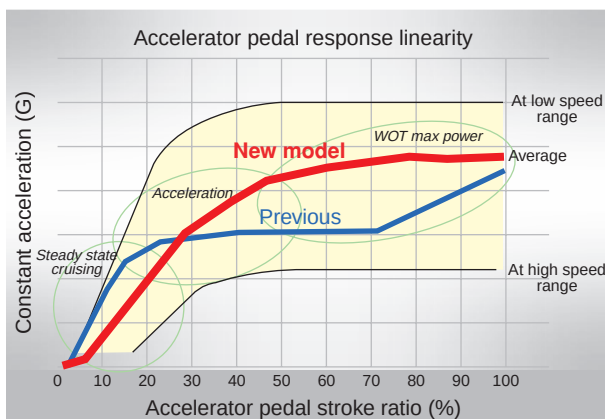


Fig. 13 Accelerator pedal response

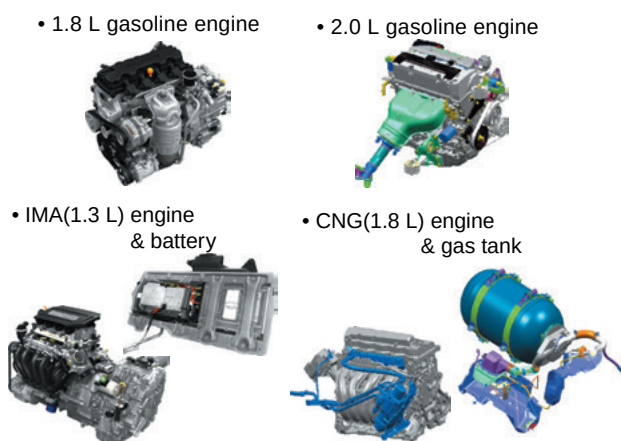


Fig. 14 Worldwide power plant variations

6. Ride and Handling

To produce the CIVIC's distinctive ride and handling, the developers wanted to come up with both linear and direct handling that were compatible with a smooth ride. Many thorough simulations and tests were carried out concerning the body that supports the suspension, and the results showed a 35% increase over the previous model in torsion rigidity, enhancing the suspension performance (Fig. 15).

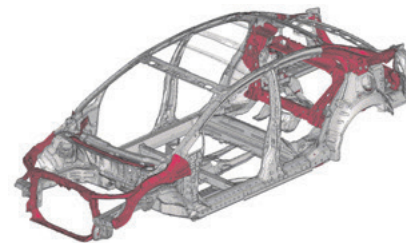
6.1. Front Suspension

Increasing the caster angle and caster trail amounts enhanced turning stability, linear toe compliance performance and straight-line stability (Fig. 16). A linear and direct steering feel was obtained by reducing the center offset amount and optimizing characteristic damper, spring and bush settings, thereby suppressing kickback and torque steer.

6.2. Rear Suspension

An enhanced damper lever ratio with separately located suspension springs, a reduced damping force and optimized bush characteristics increased ride comfort. A longer upper arm length, rear tilt and deviation against rear knuckle bush rotation axis, etc. (Fig. 17) raised the linear characteristics of camber changes against bump toe change, and raised lateral ground contact rigidity, thereby enhancing turning performance.

Furthermore, a comprehensive review of detailed parts, including the use of a damper valve for steering kickback, changes to hydraulic controls, and rigidity increases were made, producing a stable feeling as well as exhilarating handling with a linear and direct feel (Fig. 18).



Reinforced body parts

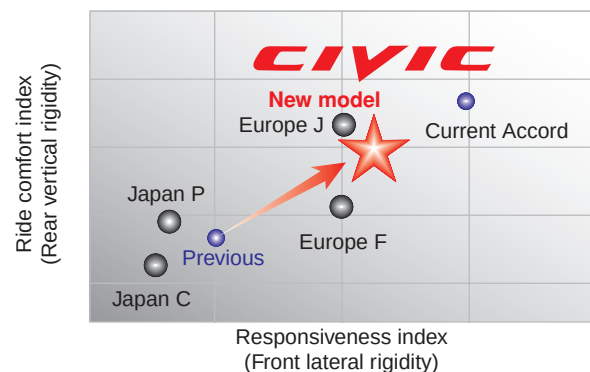


Fig. 15 Body rigidity

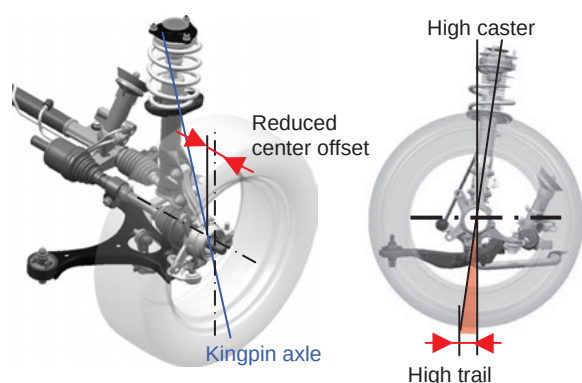


Fig. 16 Front suspension

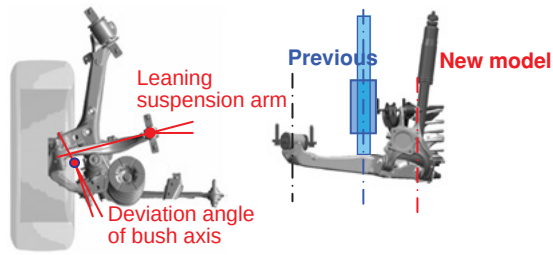


Fig. 17 Rear suspension

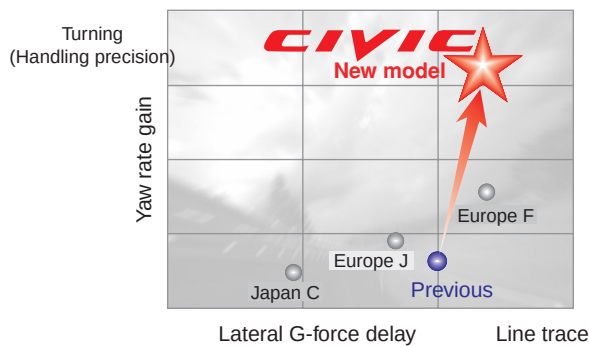


Fig. 18 Handling

7. Engine Sound Creation

To express the sporty and comfortable feel of the CIVIC, developers aimed “to create an engine sound that was integrated with acceleration.” This was achieved by reducing 2nd order booming noise and creating sound using 4th order intake noise, which rises linearly with acceleration (Fig. 19).

7.1. Noise Cleaning

7.1.1. Use of a torque rod mount method

For the engine mount system, a torque rod mount method⁽²⁾ was used in which the torque rod receives driving reaction force in the longitudinal direction.

The torque fluctuation vibration which operates in the rotational direction around the crankshaft is received longitudinally. This reduces idle vibration and booming noise at low engine speeds, e.g. when taking off. With the torque rod’s double vibration-prevention effect, booming noise inside the vehicle and engine noise are both reduced (Fig. 20).

7.1.2. Application of light-weight sound absorbing material

Reviewing a layered structure of the conventional noise insulation material and applying materials with advanced sound absorbing performance both enhanced body quietness and reduced body weight⁽³⁾ (Fig. 21).

7.1.3. Structure preventing sound penetration

A structure preventing noise penetration from the outside was created by using a sound-absorbing door hole seal (Fig. 22) and a double door seal, by reducing body plate joint clearances and the number of holes, and by applying urethane foam inside each pillar.

7.2. Sound Creation

In addition to implementing the measures outlined above, the intake system’s silencer device was used to create an engine sound that matches the acceleration.

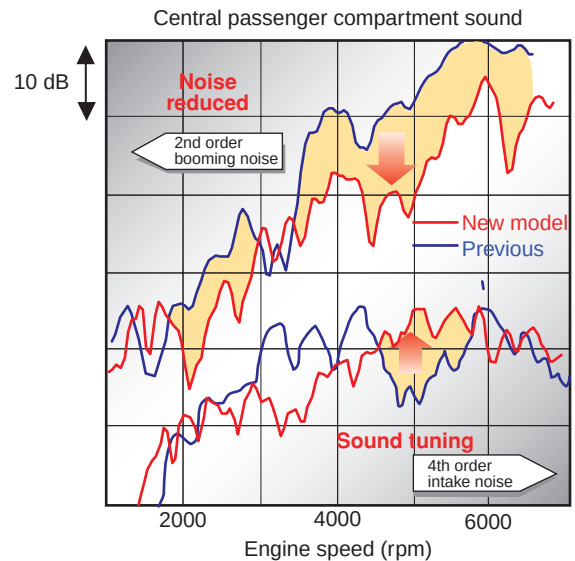


Fig. 19 Engine sound creation

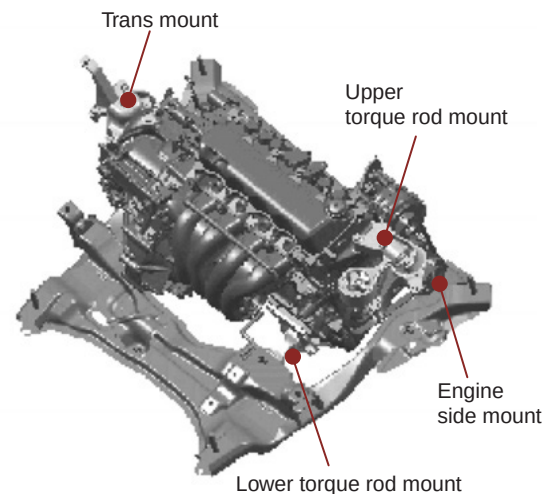


Fig. 20 Torque rod mount system

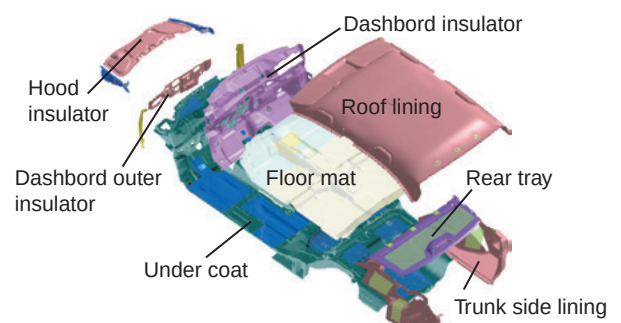


Fig. 21 Use of sound-absorbing materials

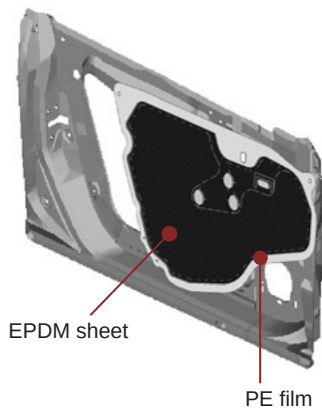


Fig. 22 Sound-absorbing door hole seal

8. Collision Safety Performance

In collision safety evaluations carried out according to J-NCAP, US-NCAP, EURO-NCAP, and IIHS standards, top performance was achieved. Furthermore, the target was not only reducing damage to the CIVIC and its occupants when struck by other vehicles, but also to reduce damage inflicted on the other vehicle. An Advanced Compatibility Engineering (ACE) body⁽⁴⁾ (Fig. 23) was used, along with tailored blank welding on the side frame, which helps control the deformation mode during a collision. This produced a structure that is highly efficient in impact energy distribution and absorption.

As a result, the structure increases the efficiency of the occupant restraint system, reduces vehicle body deformation and enhances occupant restraint performance, thus achieving best-in-class collision safety performance.

In addition, the active headrest, a Honda first, is a standard specification on front seats (Fig. 24). The structure harnesses the force of the occupant hitting the seat back during a rear-end collision. If the force is great, a link mechanism inside the seat back goes into operation, the headrest is raised and moved into an

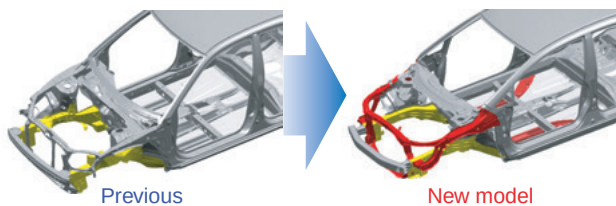


Fig. 23 Advanced compatibility engineering body

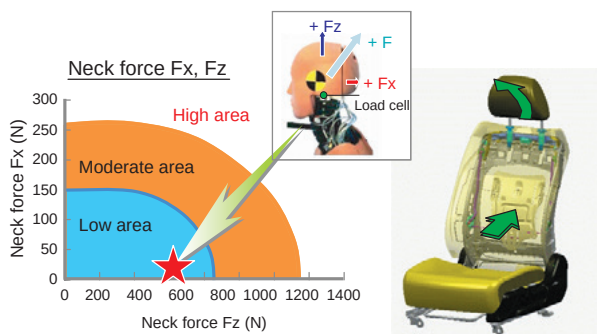


Fig. 24 Active headrest

appropriate forward tilt. By adopting an inertial lock to maintain the headrest position, the head's rear movement is kept to a minimum, thereby mitigating shock to the neck.

9. Conclusion

We have paid close attention to the development of even the smallest details, working to create a vehicle with the unique CIVIC sensation and excitement.

We are confident that we have created a next generation compact sedan that combines a "One of a kind" feeling of unique distinction, with being "Number one" in innovative performance.

References

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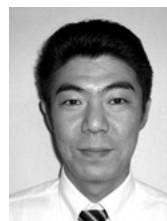
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Development of 1.8L i-VTEC Gasoline Engine for 2006 Model Year Honda CIVIC

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ABSTRACT

Honda has developed a lightweight and compact next-generation 1.8L i-VTEC 4-cylinder gasoline engine that offers a superb balance of power and environmental performance (high fuel economy and low emissions). The i-VTEC system in the new engine provides high power when the vehicle is accelerating by operating the valves with a cam that produces maximum power. When the engine is operating under a low load, e.g. when cruising, the system switches to a cam with delayed valve closure timing and simultaneously opens the electronically-controlled throttle in order to reduce pumping losses and increase fuel economy. The 2006 CIVIC in which the new engine is fitted has achieved a high level of fuel economy, obtaining 17.0 km/l in Japanese 10-15 mode, a 5% increase over the level required by Japanese 2010 fuel economy standards. The use of a variable-length intake manifold has balanced low- to medium-speed torque with high power. The integration of the cylinder head with the exhaust manifold and the employment of a high-density close-coupled two-bed catalyst and high-accuracy adaptive air-fuel control has enabled the achievement of an emissions level 75% below the level required by Japanese 2005 regulations. In addition, noise and vibration have been reduced by employing a lower block structure that increases the rigidity of the crankshaft support structure.

1. Introduction

The reduction of exhaust emissions is an ongoing requirement of the engine development process. More recently, additional issues have become increasingly important focal points of engine development: conserving energy resources by reducing fuel consumption and enhancing environmental performance by reducing CO₂ greenhouse gas emissions. Honda has continued to enhance environmental performance by developing a series of next-generation engines, including 2.0L and 2.4L 4-cylinder i-VTEC

engines⁽¹⁾ that combine VTEC with a VTC mechanism to provide continuous camshaft phase variations, and a 3.0L V6 i-VTEC engine⁽²⁾ that comes equipped with a variable cylinder system that allows cylinder idling.

Honda has now developed an R18A 1.8L 4-cylinder i-VTEC gasoline engine that has a new mechanism enabling delayed closure of the intake valves to balance enhanced fuel economy with high power and low emissions. This paper will discuss the innovative technologies that have been used in the new engine.

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2. Development Aims

The development concept for the new engine is the achievement of a superb balance between the environmental performance (high fuel economy and low emissions) and the increased torque performance required of Honda next-generation engines. The following were established as specific development aims:

- (1) Fuel economy of 17 km/L in 10-15 mode
- (2) Emissions levels 75% lower than required by 2005 exhaust gas standards
- (3) Low- to medium-speed torque allowing smooth driving under standard usage
- (4) Balance between increased quietness and weight savings

3. Overview of Engine and Main Specifications

Figure 1 shows an external view of the R18A i-VTEC engine and Table 1 shows its main specifications as compared to those of a D17A engine. The reduction of pumping loss by delaying closure of the intake valves is known to be an effective means of increasing fuel economy in gasoline engines⁽³⁾. The new engine discussed in this paper has been provided with a new variable valve mechanism that switches cams depending on engine load, enabling increased fuel economy under low engine loads through delayed closure of the intake valves, and increased power under high loads.

The new engine has also been made more lightweight and compact, with a 5 kg weight reduction and 13 mm shortening in length compared to the former D17A engine.

Figure 2 shows a cross-section of the cylinder head. As in the D17A engine, the intake and exhaust rocker arms are supported by a single rocker shaft. The valve-included angle of the cylinder head has been reduced from 46° to 34° to make the combustion chamber more compact.

In addition, integration of the cylinder head with the exhaust manifold, as shown in Fig. 3, has reduced the heat-radiating surface area, thus allowing faster activation of the catalyst.

Hydraulic-hydraulic switching continues to be used in the valve system, however, an aluminum alloy rocker arm was used for the first time in a VTEC engine. The new rocker arm weighs approximately 35% less than the former cast iron rocker arm.

Table 1 Engine specifications

Engine code	R18A	D17A
Cylinder configuration	In-line 4-cylinder	In-line 4-cylinder
Bore × stroke (mm)	81 × 87.3	75 × 94.4
Displacement (cm ³)	1799	1668
Compression ratio	10.5 : 1	9.9 : 1
Valve train	SOHC i-VTEC Inlet delayed closure	SOHC VTEC-E 1 intake valve inactive
Number of valves	4 per cylinder	4 per cylinder
Valve diameter (mm)	In. 32 Ex. 26	30 26
Cylinder offset (mm)	12	0
Intake manifold	Variable intake system	Conventional
Gasoline	Regular (RON91)	Regular (RON91)
Max. power (kW/rpm)	103/6300	96/6300
Max. torque (Nm/rpm)	174/4300	155/4800

The lightweight alloys (aluminum, magnesium) formerly used to manufacture the head cover have been replaced by plastic, which has contributed to reducing the sound level radiated from the valve system and achieved a weight reduction of approximately 40%.

In order to reduce weight as well as to cope with the increased power, hot-forged, high-strength cracked connecting rods, as shown in Fig. 4, have been used in the drive system. The use of a material with approximately 50% higher fatigue strength than that of the former material has enabled the cross-sectional area of the connecting rods to be reduced by approximately 20%. Integrating the rod and the cap and manufacturing the connecting rods by cracking after forging allows the rods to be fitted together by means of the concavities and convexities of the cracked surface, eliminating the requirement for a locating dowel pin. This has enabled the bolt pitch to be reduced by 1 mm, resulting in the weight reduction of approximately 13% in the connecting rod unit.

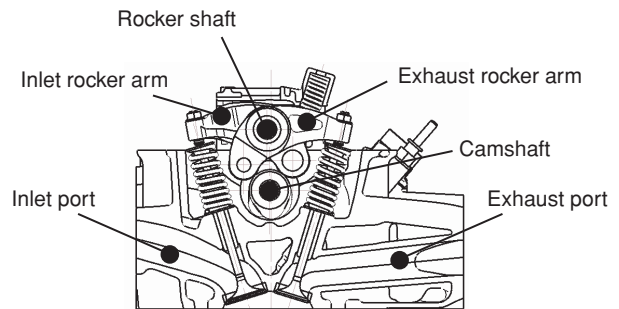


Fig. 2 Configuration of cylinder head



Fig. 1 R18A i-VTEC gasoline engine

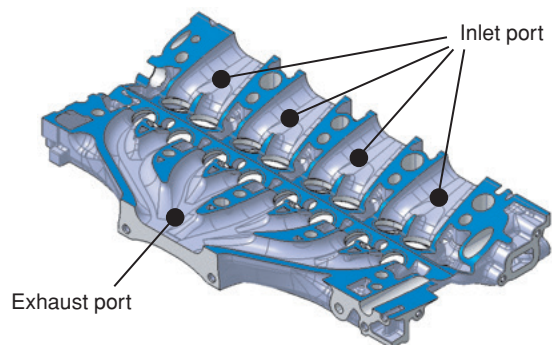


Fig. 3 Section view of exhaust manifold-integrated cylinder head

Figure 5 shows the camshaft drive system. Optimizing the positioning of the chain has enabled its width to be reduced by 1.9 mm, as compared to the former model. The chain guide and chain tensioner arm are entirely manufactured from plastic and aluminum alloy has been used for the tensioner body, resulting in a weight reduction of approximately 35% in the drive system.

Figure 6 shows the configuration of the variable-length intake manifold, and Fig. 7 shows the open and closed position of the bypass valves and illustrates the change in the effective length of the intake manifold. This plastic variable-length intake manifold has allowed an increase in torque at low to medium speeds and higher power at high speeds. In addition, the new circular cross-section of the surge tank has reduced radiated noise and allowed the thickness of the tank walls to be reduced by 30% against those of the former model.

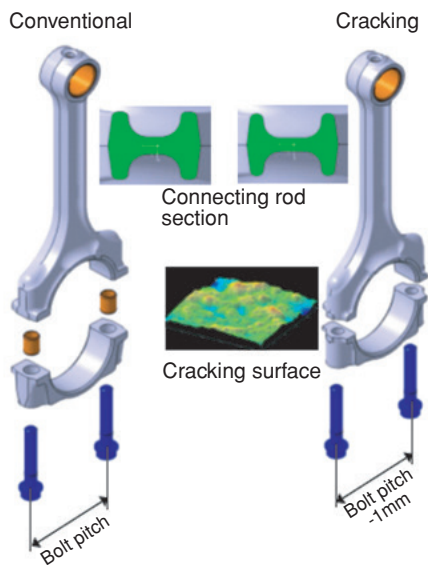


Fig. 4 Comparison of conventional and cracking connecting rod

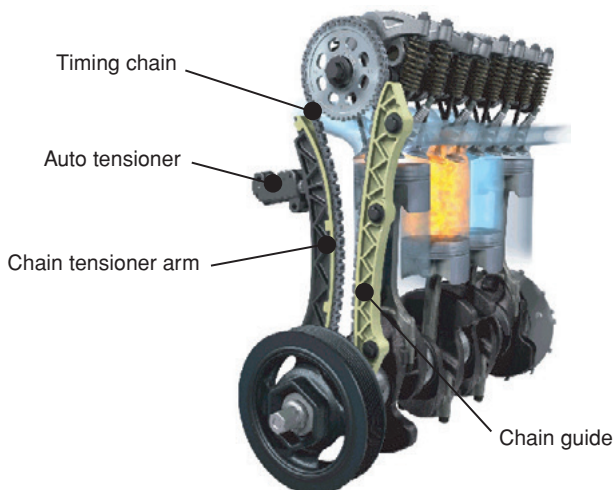


Fig. 5 Camshaft drive system

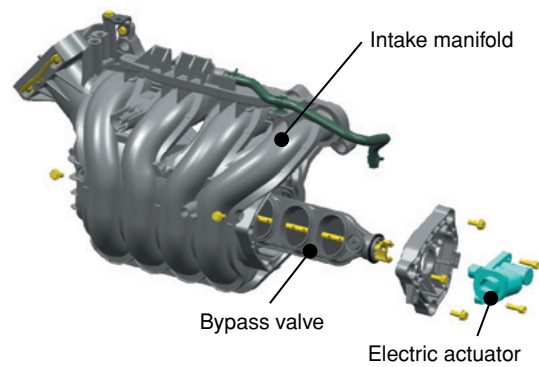


Fig. 6 Configuration of variable intake manifold

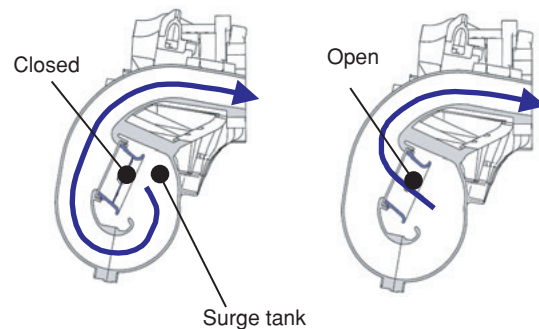


Fig. 7 Motion of bypass valves

4. Power Performance

Figure 8 shows the engine's power characteristic. A variety of technologies have been used to increase power: Use of the variable-length intake manifold has increased filling efficiency, the shapes of the intake and exhaust ports have been optimized, the compression ratio has been increased, knocking performance has been enhanced by making the combustion chamber more compact and by using a piston oil jet, and friction has been reduced. These measures have increased the ratio of torque to displacement by 4% against that in the former engine.

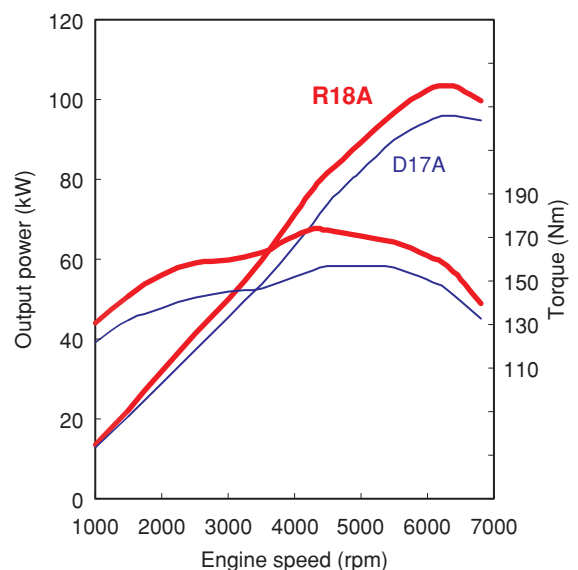


Fig. 8 Engine performance

5. i-VTEC Mechanism

5.1. i-VTEC Intake Valve Closure Delay Mechanism

At high load (acceleration) conditions, the i-VTEC system used in the new engine controls the intake valves by a cam designed to produce maximum power (high output cam, below). When the engine is under low loads (when the vehicle is cruising, etc.), the system switches to another cam (delayed closure cam, below) that delays the closure of one bank of intake valves, and at the same time opens the electronically controlled throttle. This reduces pumping loss and increases fuel economy⁽⁴⁾. The delayed closure cam has effectively reduced the compression cycle and essentially is able to increase the expansion ratio versus the effective compression ratio. A high expansion cycle has long been known to be effective in increasing fuel economy, but also causes filling efficiency and the effective compression ratio to decline, resulting in reduced power. The mechanism proposed to resolve this issue in the new engine was optimal switching between the high output cam and the delayed closure cam in response to driving conditions by means of the i-VTEC system.

Valve switching in the i-VTEC system is conducted by means of the connection and disconnection of the rocker arms via the hydraulic pressure exerted on the synchro pistons built into the arms. In the new engine, this switching mechanism has only been applied to one rocker arm of the two intake valves.

Figure 9 shows the basic configuration of the valve train and the hydraulic passages. Figure 10 shows the configuration of the rocker arms and the cam profile. This is a hydraulic-hydraulic switching mechanism that employs two hydraulic passages and is identical to the mechanism used in the CIVIC Hybrid cylinder idling system⁽⁵⁾.

When the high output cam is operating, synchro piston A is disconnected by the hydraulic pressure exerted on synchro piston B, and the intake valves are operated according to cam profile A (Fig. 10(a)).

As shown in Fig. 10(b), when the delayed closure cam is in operation, the hydraulic pressure that is exerted on synchro piston A links rocker arms A and B, and the intake valves are operated according to cam profile B.

A three-way solenoid spool valve has been used to ensure responsiveness in switching between the two hydraulic passages. This hydraulic-hydraulic switching mechanism enables i-VTEC switching in the low engine speed/low hydraulic pressure range,

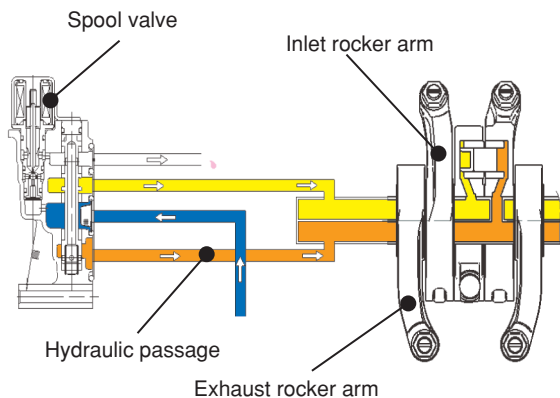


Fig. 9 Valve train and hydraulic passage

thus making it possible to receive the fuel economy benefits of the delayed closure cam at low vehicle speeds. To reduce weight, an aluminum alloy rocker arm has been used for the first time in the hydraulic-hydraulic switching system in the new engine.

5.2. i-VTEC Switching Control

Figure 11 shows the operational range of the delayed closure cam. Switching to the delayed closure cam is conducted at low engine speeds and low loads, which are the normal vehicle cruising conditions. Apart from this – at engine start, when the engine is idling, and when the engine is operating under high loads and at high speeds – the high output cam operates. The necessity for switching is judged on the basis of a variety of data, including engine speed, degree of throttle opening, vehicle speed and the engine's water temperature.

As Fig. 12 shows, control of the degree of throttle opening and the ignition timing ensures that this load-dependent switching is smooth, with no torque steps. This has enabled the operational range of the delayed closure cam to be extended to approximately 90% of WOT torque.

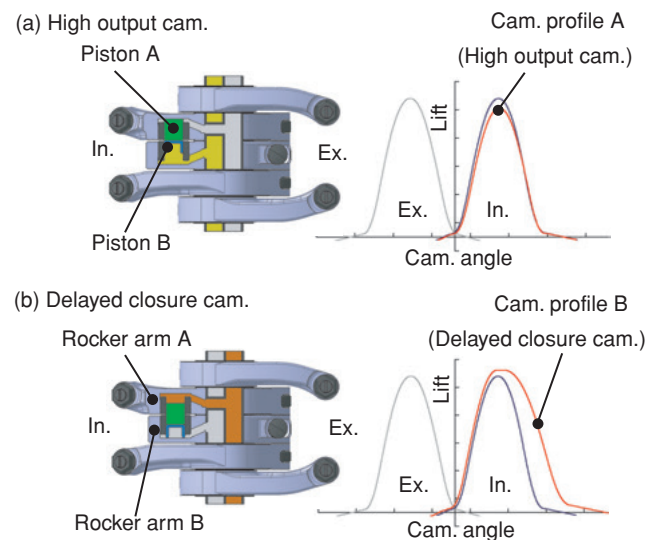


Fig. 10 Rocker arm and cam. profile

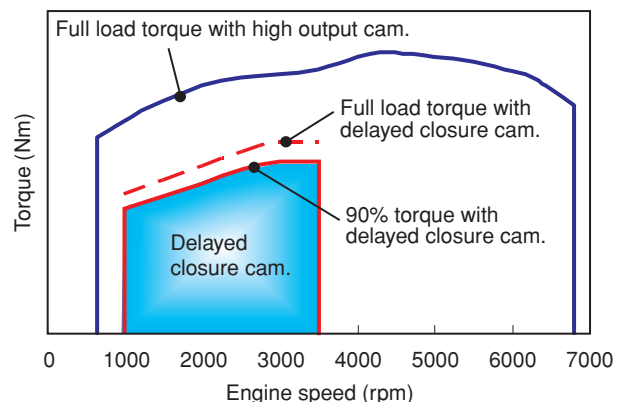


Fig. 11 Operational region of delayed closure cam.

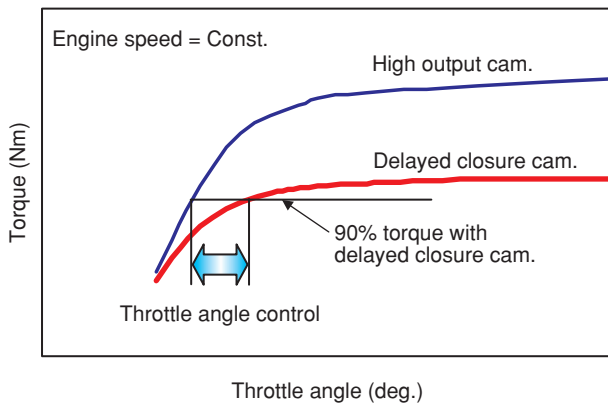


Fig. 12 Throttle angle control at i-VTEC switching

6. Friction Reduction Technologies

In addition to the use of delayed closure of the intake valves to increase fuel economy, the new engine incorporates the technologies that have been employed in the i-VTEC series to date to reduce friction: An offset cylinder configuration, roller-follower rocker arms, ion-plated piston rings and shot peening of the piston skirts⁽⁶⁾, followed by a molybdenum disulfate coating. Further technologies applied to reduce friction include a reduced cam chain width, reduced chain tensioner spring load, reduced cam journal surface roughness and reduced piston ring tension.

The application of these technologies has reduced friction in the new engine by 10% against that of the D17A engine.

7. Engine Fuel Economy and Vehicle Fuel Economy

Figure 13 shows brake-specific fuel consumption (BSFC) against total engine displacement. The reduction in pumping loss, brought about by delayed closure of the intake valves and the usage of technologies to reduce friction, have reduced BSFC by 6% against that in the former engine. This has resulted in the achievement of an engine fuel economy equivalent to that of a lean-burn engine even though it operates at a stoichiometric air-fuel ratio.

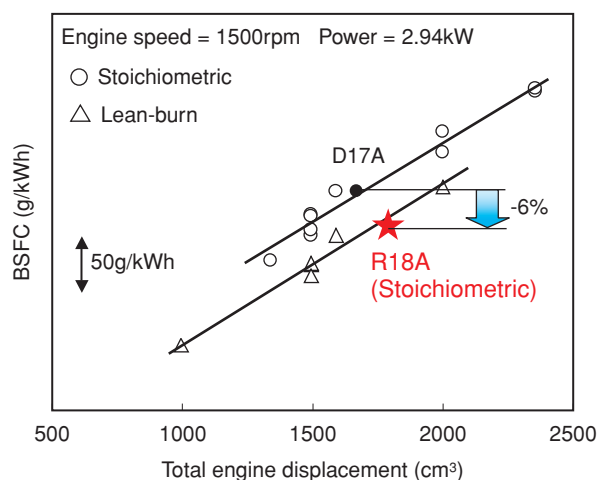


Fig. 13 Enhancement of BSFC

Other techniques have been used to reduce fuel consumption in the vehicle in which the engine is fitted, including expanding the AT lock-up clutch's operational range, which has resulted in the achievement of a 17 km/l fuel consumption rate in the Japanese 10-15 mode, representing a 5% improvement of 2010 fuel consumption standards (Fig. 14).

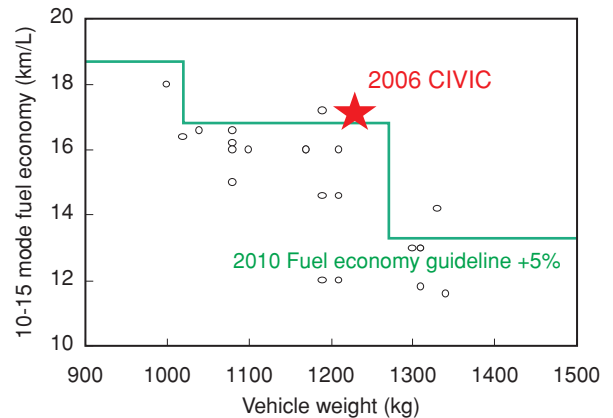


Fig. 14 Relationship between vehicle weight and 10-15 mode fuel economy in production stoichiometric gasoline vehicles

8. Technologies to Reduce Exhaust Emissions

8.1. Emissions Processing System

The exhaust system of the new engine is shown in Fig. 15. A closed coupled, two-bed, three-way catalyst has been fitted to the cylinder head, which has an integrated exhaust manifold. This has reduced heat mass and enabled rapid activation of the catalyst. In addition, a universal exhaust gas oxygen sensor (UEGO) has been positioned upstream and a heated exhaust gas oxygen sensor (HEGO) downstream from the front bed of the catalyst to allow highly accurate adaptive control of the air-fuel ratio, thereby achieving excellent purification performance. This has allowed significantly reduced amounts of precious metals to be used in the new engine, compared to the former engine.

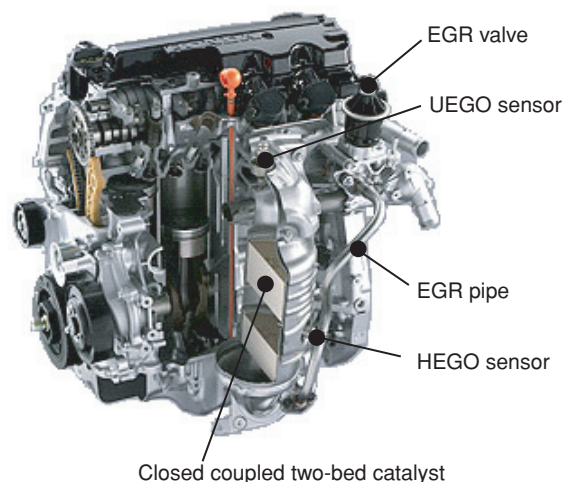


Fig. 15 Exhaust system

As shown in Fig. 16, the cone shape at the front of the catalyst has been optimized using computational fluid dynamics (CFD) to ensure that the exhaust gas strikes the catalyst uniformly, to maximize the catalyst's purification capability and to limit catalyst degradation. The channel has been restricted immediately after the cylinder head outlet. A bell mouth shape has been used to make the passage between the restriction and the catalyst smooth to ensure the exhaust gasses take an optimal flow direction, reducing vortex loss on the walls and resulting in excellent gas flow.

Figure 17 shows images of exhaust gas from each cylinder striking the UEGO during the exhaust stroke. To enhance air-fuel ratio control, the positioning of the UEGO sensor has been optimized to allow uniform detection of the exhaust gas from each cylinder.

8.2. EGR System

Figure 18 shows the configuration of the EGR system. To eliminate unburnt hydrocarbons from the EGR passage in the new R18A engine, EGR gas is removed after passing through the catalyst.

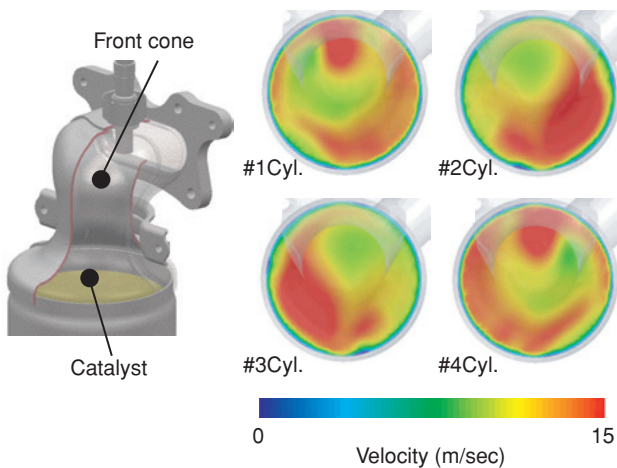


Fig. 16 Optimized exhaust gas feed to catalyst

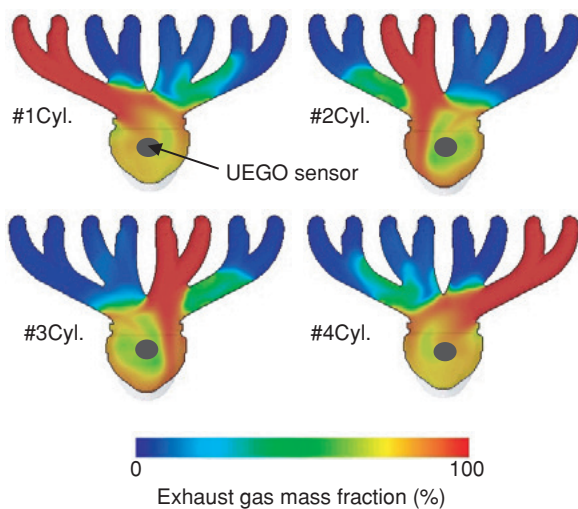


Fig. 17 Optimized UEGO sensor positioning

Figure 19 shows the results of an analysis of the EGR gas flow in the intake manifold around the area where the gas is introduced. The results indicate that a counterflow is generated downstream from the throttle valve when the opening angle of the valve is low. This counterflow carries EGR gas to the throttle valve and causes it to become dirty. The positioning of the EGR introduction joint in the intake manifold and the direction of the EGR gasses' introduction have therefore been optimized to prevent EGR gas from directly striking the throttle valve.

Figure 20 shows CFD results for sudden closure of the throttle. Optimizing the shape, length and orientation of the EGR

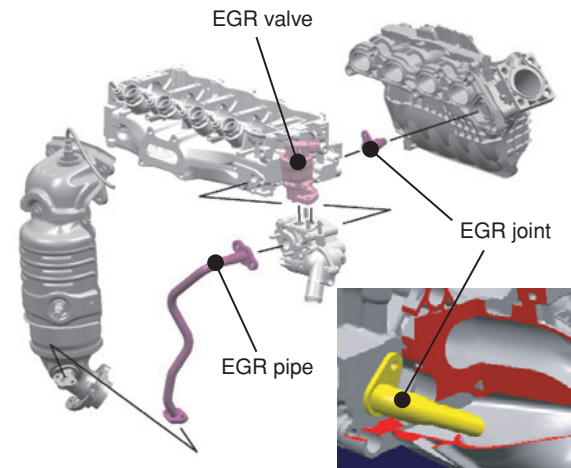


Fig. 18 Configuration of EGR system

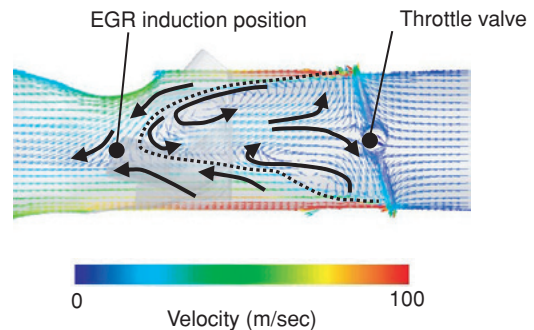


Fig. 19 Analysis of EGR gas flow around throttle valve

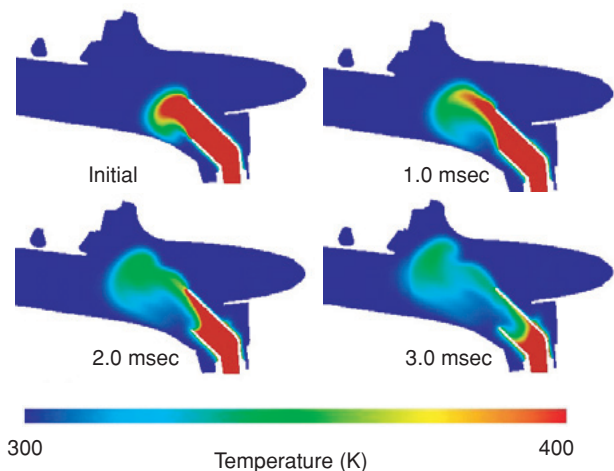


Fig. 20 Analysis of EGR gas flow at sudden throttle closing

introduction joint has prevented EGR gas from directly striking the intake manifold wall, reducing the effect of heat on the plastic manifold. In addition, the mixing of EGR gas and new gas has been promoted and EGR distribution to each cylinder has been optimized.

A 6-hole injector has been used to enhance the fuel spray droplet characteristics in order to reduce the amount of fuel that adheres to the intake port and combustion chamber walls, and to reduce unburnt hydrocarbon emissions at engine start and when the catalyst is being heated. The new engine has also been provided with highly accurate electronic EGR valve control and air-fuel control using air flow sensors, map sensors and an electronically controlled throttle. These various technologies have reduced emissions to a level 75% below 2005 standard levels.

9. Technologies to Reduce Noise and Vibration

A frequent issue in in-line 4-cylinder engines is a muddy sound (rumble) occurring between 2000 and 4000 rpm when the vehicle is accelerating. Measures were taken to resolve this issue in the new engine.

Rumble is mainly caused by structure-borne sound from cyclic engine vibration originating from the motion of the crankshaft, which is transmitted to the vehicle body via the engine mounts. The primary components of this sound are between 200 and 800 Hz⁽⁷⁾⁻⁽⁹⁾. To reduce this sound, the measures listed below were implemented to reduce the vibration originating in the crankshaft and the vibration transmitted from the crankshaft to the engine mounts.

- (1) Control of the crankshaft's primary bending deformation caused by the inertia of reciprocating parts
- (2) Increased rigidity of the area around the crankshaft bearings
- (3) Increased bending rigidity of the engine-transmission assembly
- (4) Reduction of the oil clearance of the crankshaft bearings

Figure 21 shows the relationship between the crankshaft's bending stiffness and the bending moment exerted on the crankshaft.

To reduce the crankshafts primary bending deformation, a

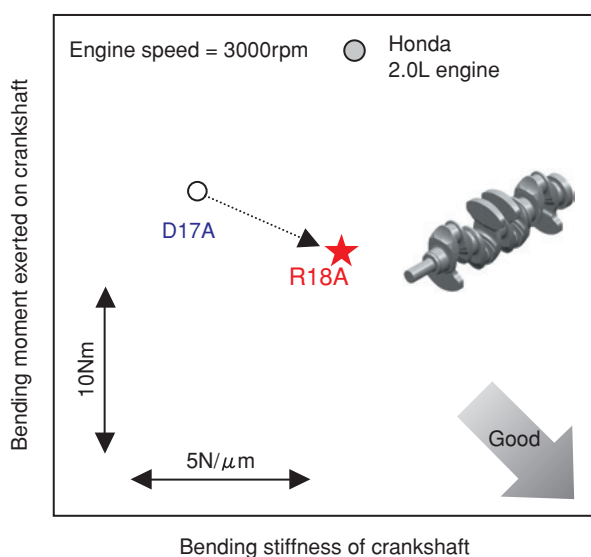


Fig. 21 Bending stiffness of crankshaft and bending moment exerted on crankshaft

counterweight was fitted to reduce the bending moment to a level less than or equivalent to that of the former engine. In addition, the pin width was reduced and the area around the web was reinforced, resulting in the achievement of a level of bending stiffness equivalent to that of a Honda 2L engine.

Increasing the stiffness of the crankshaft bearings was achieved by using a lower block that integrates the crankshaft's bearings and external wall into a ladder frame configuration. Figure 22 shows the cylinder block and lower block.

To enhance the bending rigidity of the engine-transmission assembly, the coupling between the engine and the transmission, the major factor in the bending mode of the power plant, was reinforced, while still keeping in mind the goal of weight reduction. As a result, the power plant's bending resonance frequency has been increased by approximately 20% against that of the former engine.

To reduce the oil clearance of the crankshaft bearings, although the bearing sections of the lower block usually use FC (Cast iron), FC casting has been eliminated in the R18A engine to reduce weight. This has increased the rate of change of oil clearance in relation to engine temperature, but adjusting the crank bearings' range of tolerance has allowed crank vibration to be reduced.

Figure 23 shows mid-frequency engine vibration against engine speed. By implementing measures (1) to (4), mid-frequency engine vibration at medium engine speeds has been reduced by approximately 10 dB against that of the former engine, and weight reduction has been balanced against reduced rumble.

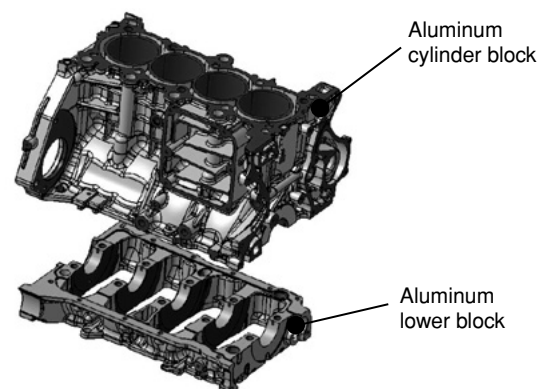


Fig. 22 Cylinder block and lower block

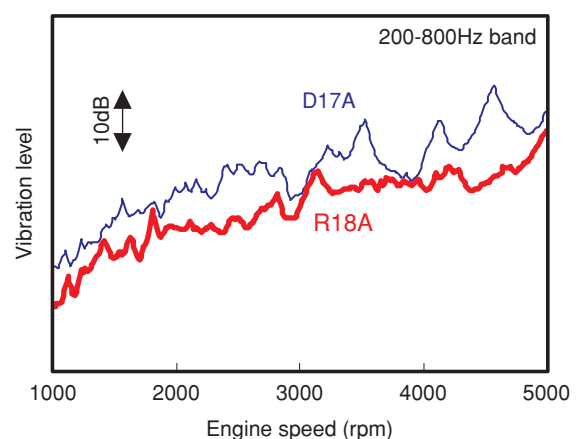


Fig. 23 Vibration level at the engine mount bracket end under full load

10. Conclusion

An i-VTEC mechanism allowing switching between a high output cam and a delayed closure cam in response to driving conditions has been developed. The following are significant findings in the development:

- (1) The torque versus displacement ratio has been increased by 4% and the unit fuel economy by 6% against that of the former engine.
- (2) The vehicle in which the new R18A engine is fitted has achieved a best-in-its-class fuel economy of 17 km/L in the Japanese 10-15 mode (an improvement of 5% against 2010 fuel economy standards).
- (3) The integration of the cylinder head with the exhaust manifold and the use of a closed coupled two-bed catalyst and highly accurate adaptive air-fuel ratio control have allowed the achievement of an emissions level 75% below that required in the 2005 standards.
- (4) The new engine is lighter, quieter and more compact than the former engine.

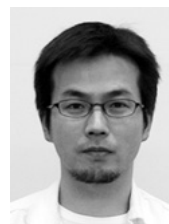
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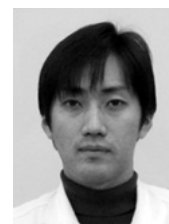
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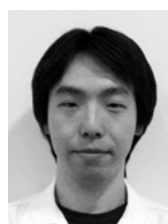
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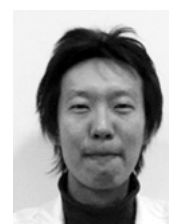
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Development of IMA System for New CIVIC Hybrid

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ABSTRACT

The 2006 year model CIVIC Hybrid was developed to surpass its former model in both fuel economy and acceleration performance.

The combination of an advanced variable cylinder management system employing 3-stage i-VTEC technology and an IMA motor has resulted in enhanced fuel economy through increased regenerative energy and motor-only operation. A variety of modifications have been made to the IMA components to increase fuel economy and power: Flat wire coils for superior motor torque, modification of current collecting plates in the battery for reduced resistance, and an increased CVT pulley diameter for enhanced capacity and reduced energy loss.

The application of these technologies has achieved both a fuel economy of 31.0 km/L in the 10-15 mode and an acceleration performance of 0-100 km/h in 12.1 seconds.

1. Introduction

Fuel-efficient, low-emission vehicles are attracting increasing attention while the importance of environmental protection increases. Honda has responded by developing hybrid systems in order to increase fuel economy: The ultra-low fuel consumption Insight⁽¹⁾ reached the market in 1999, followed by the CIVIC Hybrid small sedan⁽²⁾ in 2001, and the ACCORD Hybrid⁽³⁾, a medium-size sedan for the US market, in 2004.

With the model change of the CIVIC base vehicle, a new IMA system was also developed for the CIVIC Hybrid. The goals behind the new development were to achieve even higher levels of fuel economy, an important selling point of the CIVIC Hybrid, while also pursuing superb driving performance. This paper discusses the technologies employed to achieve these increases in fuel economy and power.

2. Development Goals

With the aim of balancing fuel economy and driving performance, the targets for the development of the new system were to achieve a fuel economy of more than 30 km/L in the 10-15 mode and an acceleration performance of 0-100 km/h in 12 seconds.

3. System Configuration

The configuration of the complete hybrid system is shown in Fig. 1. Figure 2 shows an external view of the power plant.

The layout of the main components is identical to that of the 2003 year model CIVIC Hybrid ("former model" below): The engine compartment features a thin IMA motor to supply drive power and conduct power regeneration. This IMA motor is sandwiched between the engine and the CVT, while an Integrated Power Unit (IPU), containing high voltage electrical components, including the inverter unit used to drive the motor, a high-voltage battery pack, and an air-cooling system, all which are located in the trunk behind the rear seatbacks. A cable to transmit drive power

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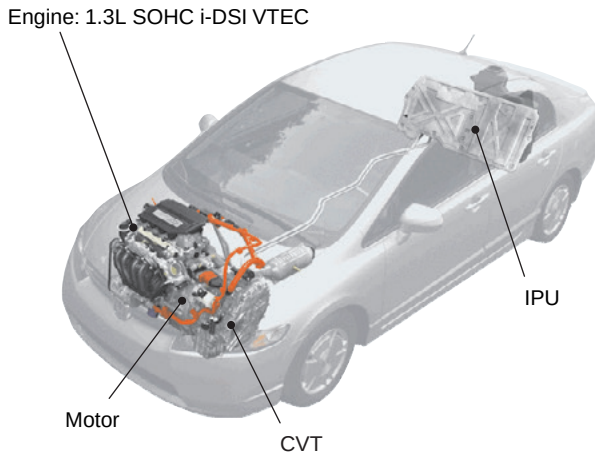


Fig. 1 IMA system configuration

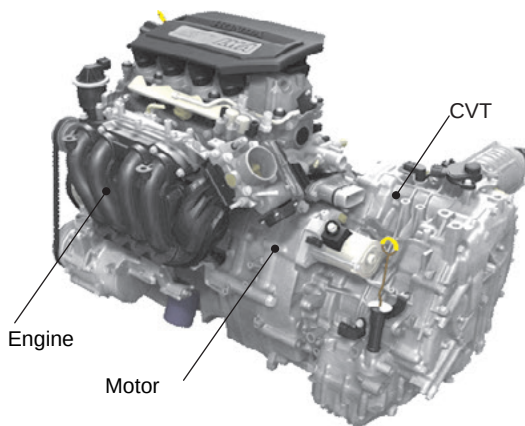


Fig. 2 View of IMA power train

and regenerated energy between the motor and the inverter runs underneath the floor of the vehicle, outside the cabin space.

4. Technologies Employed to Balance Fuel Economy with Driving Performance

4.1. Engine

The assist provided by the motor enables an engine of lower displacement to be used in the CIVIC Hybrid than in the base gasoline vehicle to reduce fuel consumption. The 1.3L SOHC i-DSI engine, with the modified Variable Cylinder Management (VCM) system has been employed in the 2006 year model CIVIC Hybrid.

This system previously deactivated valves in three cylinders to increase the amount of regenerative energy during deceleration, but the new CIVIC Hybrid has been redesigned to deactivate valves in all four cylinders to further reduce pumping loss.

3-stage i-VTEC has been employed to achieve higher power output. The rest of this section will discuss the functioning of the 3-stage i-VTEC mechanism, which an overview is shown in Fig. 3.

The new mechanism features three rocker arms on the intake side (low-speed, high-speed and idling) and two rocker arms on the exhaust side (operation and idling). Three hydraulic channels are used to switch these rocker arms. These enable valve timing and lift to be varied in order to achieve excellent power characteristics from low to high engine speeds, and enable valves

to deactivate as required to provide the optimum engine operating mode. In the new system a high-speed cam has also been employed and the valve diameter has been increased to achieve higher power.

The number of spool valves employed has been increased to two, and the hydraulic channels used for inactive valves have been separated to increase the reliability of reactivation, enabling valves in all the cylinders to be deactivated.

Deactivation of valves in all four cylinders reduces pumping loss as compared to figures when valves are active in all cylinders, and reduces engine friction by a maximum of 66%, as shown in Fig. 4.

4.2. Driving Mode and System Operation

The combination of the 3-stage i-VTEC mechanism and IMA motor in the 2006 year model CIVIC Hybrid enables a variety of operating modes. The optimum deployment of these operating modes in actual driving patterns has enabled engine operation (and thus fuel consumption) to be reduced, it increases the amount of power regenerated during deceleration, and increases the power of the entire power plant, enabling high fuel economy to be reconciled with excellent dynamic performance.

The rest of this section will discuss the operating modes of the

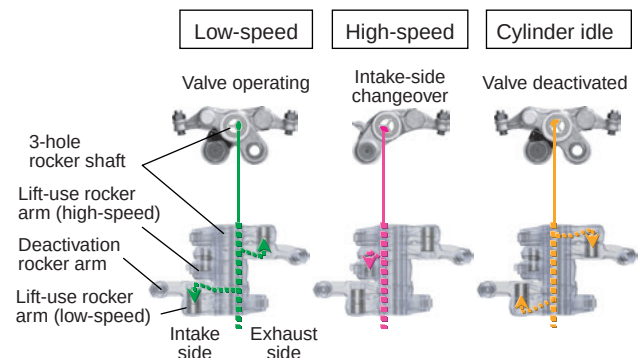


Fig. 3 3-stage i-VTEC mechanism

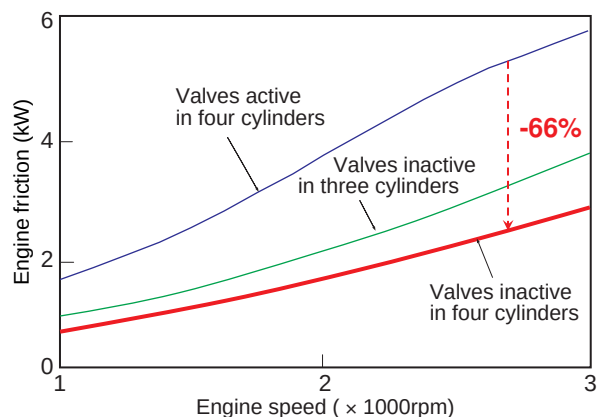


Fig. 4 Engine friction

system, as shown in Fig. 5.

When the vehicle goes from a stationary state (Mode 1) to start-up and acceleration (Mode 2), the IMA motor starts the engine from an idle stop state, and both the engine and the motor supply power to provide sufficient acceleration. When gentle acceleration is sufficient, the system uses Mode 3, and the vehicle accelerates using only the engine. In this mode the motor does not provide or regenerate power, but conducts flux weakening control to reduce running resistance.

When the vehicle has stopped accelerating and is cruising at a low speed, the system enters Mode 4. In this mode, only the motor operates the vehicle, using energy stored in the batteries. The valves in all cylinders are closed and all cylinders become idle, which stops the fuel supply and reduces pumping loss. This reduces running resistance, enabling vehicle operation using the motor only.

When the vehicle accelerates again, the system enters Mode 5, in which the valves become active and both the engine and the motor provide power for acceleration. When rapid acceleration is required at high engine speeds, Mode 6 switches valve operation from the low/medium speed-cam to the high-speed cam, enabling more power to be provided by the engine.

When the vehicle is cruising at high speed, Mode 7 is engaged, and the vehicle is operated using power from the engine only (low-speed cam). In this mode, the motor conducts flux weakening control to reduce running resistance.

When the vehicle decelerates, Mode 8 is engaged. In this mode, the motor regenerates energy, converting kinetic energy into electrical energy to be stored in the battery and employed as a power source when the vehicle is ready for the next accelerating or cruising cycle. Valves of all the cylinders become inactive at this time to reduce pumping loss and increase the amount of energy able to be regenerated.

When the vehicle is stationary (Mode 1), auto idle stop is engaged in order to stop fuel consumption when idling.

This dual combination of an engine which is able to provide

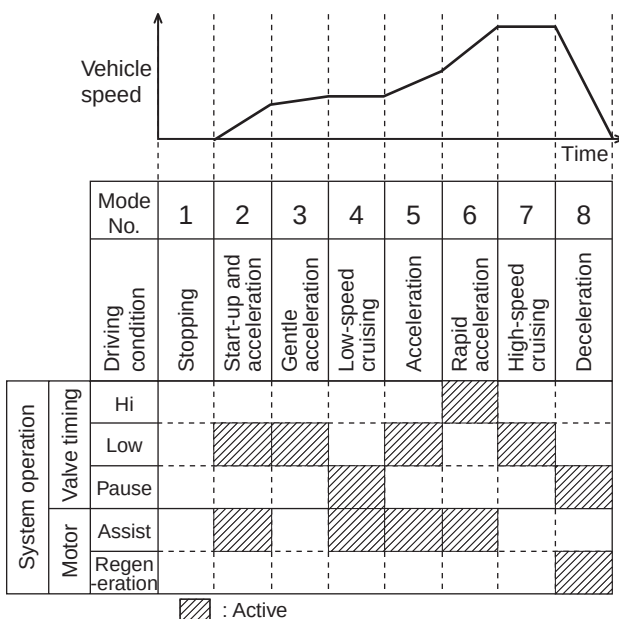


Fig. 5 Characteristics of 3-stage i-VTEC+IMA operation

high power and stable torque from the low-to-high engine speed ranges, and a motor which is able to recover kinetic energy as regenerated power and provide large torque from the ultra-low speed, enables highly efficient and high-power driving modes. These contribute to the balance of fuel efficiency and excellent acceleration achieved by the new CIVIC Hybrid.

4.3. Cooperative Regeneration Brake System

The cooperative regeneration brake system shown in Fig. 6 has been employed in the new CIVIC Hybrid to increase the amount of regenerative energy. This system is composed of a power unit that generates hydraulic pressure to ensure that stable hydraulic pressure is maintained even when the valves of cylinders are inactive, a hydraulic servo and a master cylinder that amplify and transmit the braking pressure applied by the driver, a solenoid valve and a hydraulic pressure sensor to control hydraulic braking power, and an ECU to calculate the required braking force. Apart from the power unit, these components are bundled together as a servo unit and positioned in the engine compartment.

Formerly, only a fixed amount of energy was regenerated with respect to the required braking force as determined by the pressure applied by the driver to the brake pedal, and hydraulic pressure control was employed to enable the shortfall in braking force to be supplemented by the hydraulic brakes. In the new system, control is applied to ensure that the maximum energy regenerated against the required deceleration force is always within the limit of the remaining storage capacity of the battery, and the hydraulic pressure of the brakes is controlled to ensure that the shortfall is made up by the brakes.

The application of this form of control has increased the vehicle energy recovery rate from 38% to 64%.

4.4. High Power Motor

The motor employed in the new CIVIC Hybrid is the same thin DC brushless type⁽⁴⁾ used in the previous model, but with modifications added to the rotor and stator to increase power and torque.

Torque has been increased by replacing the previous surface

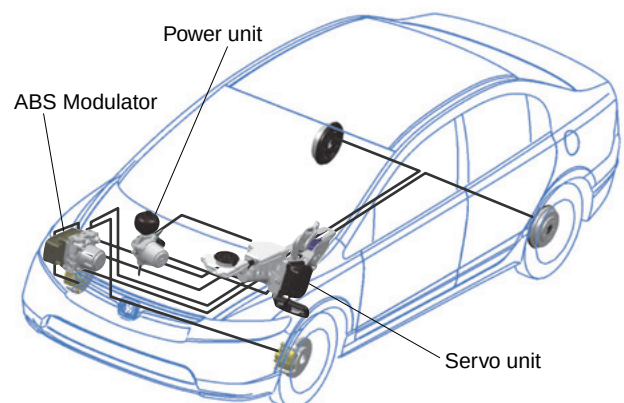


Fig. 6 Cooperative regeneration brake system

permanent magnet (SPM) with an interior permanent magnet (IPM) as shown in Fig. 7, reducing the size of the magnetic air gap and employing a high-performance magnet with high residual flux density.

In the stator, as shown in Fig. 8, the previous round wire coils have been replaced by flat wire coils to increase the space factor, the number of turns has been increased to increase the torque constant and the cross-sectional area of the wire coil has been increased to increase the allowable current. These modifications have resulted in higher torque.

The innovations discussed above have enabled the achievement of a 22% increase in regenerated power and a 14% increase in torque against the previous model with no change in the shape or dimensions of the motor. This has not only enhanced acceleration performance, but has increased the amount of energy that can be regenerated, contributing to increased fuel economy.

4.5. Technology for High Power Battery

The shapes of the current collecting plates in the battery module were modified (Fig. 9). The number of welding points to the electrodes was increased, the size and thickness of the electrodes were optimized and the amount of electrolyte in the electrolytic solution was increased. The reduction in resistance achieved by these methods has increased power density by 25%.

In addition, the number of modules in the battery pack has been increased from twenty to twenty-two. All of these modifications have been made to increase motor power and accommodate increased power regeneration. The use of a staggered module arrangement rather than the previous lattice arrangement has ensured cooling performance in addition to enabling the new model to be as easily installed as the previous model, despite the increase in the number of modules (Fig. 10).

4.6. CVT for High Power

A number of changes were made to the CVT in order to increase fuel economy and power.

The shift ratio range was expanded, and the ratio of the final gear was set high to reduce fuel consumption.

The distance between the axes of the drive and driven pulleys was increased from 143 mm to 156 mm and the diameter of the pulleys was increased to respond to increased power. The drive radius was also increased to enable the pulley clamp hydraulic pressure to be reduced, increasing CVT efficiency and contributing to increased fuel economy.

As a further measure in response to the increased power, the number of plates in the start clutch was increased from four to five to increase torque capacity (Fig. 11).

4.7. Air-conditioning System

Like the 2005 model year ACCORD Hybrid, the air-conditioning system in the new CIVIC Hybrid employs a dual scroll hybrid compressor driven by a DC brushless motor using power supplied by the high-voltage power source in addition to a conventional belt drive. This enables the idle stopping range to be increased for greater fuel economy while maintaining passenger comfort.

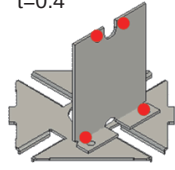
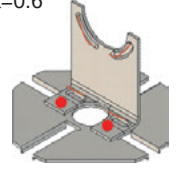
	Conventional	Developed
Collecting plate	$t=0.4$ 	$t=0.6$ 
Internal resistance	39.24mΩ	25.2mΩ

Fig. 9 Comparison of battery collecting plates

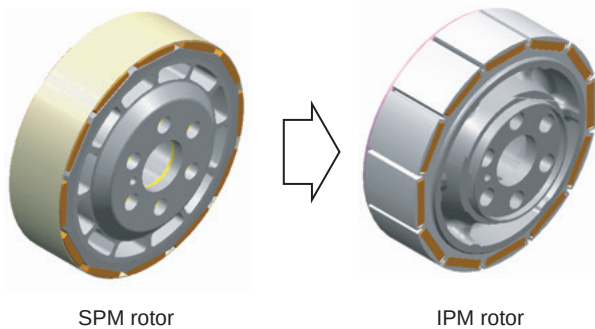


Fig. 7 Electric motor rotor

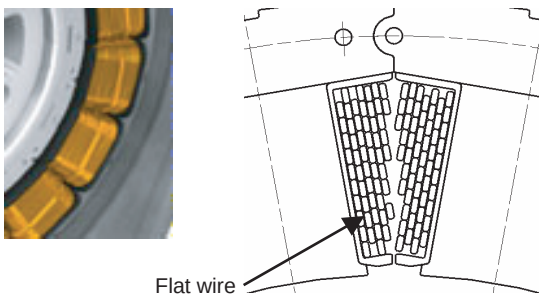


Fig. 8 Electric motor stator

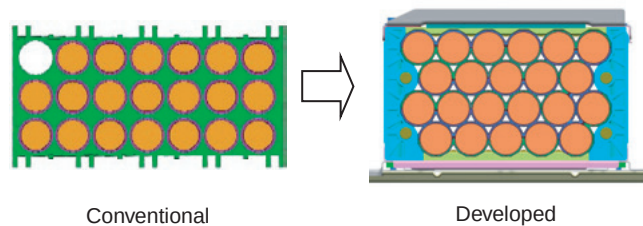


Fig. 10 Comparison of module arrangement

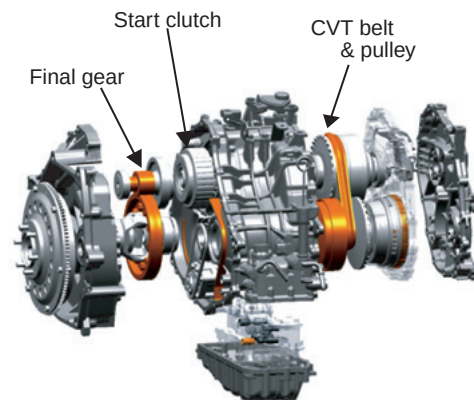


Fig. 11 CVT construction

In addition, an integrated thermo and humidity sensor (Fig. 12) has been used in-cabin, to enable the air-conditioning to be stopped when the temperature is high but humidity is low enough for no discomfort to be felt, or when there is a balance of temperature and humidity inside and outside the vehicle that will not cause the windows to fog, ensuring that passenger comfort is maintained while further increasing fuel economy (Fig. 13).

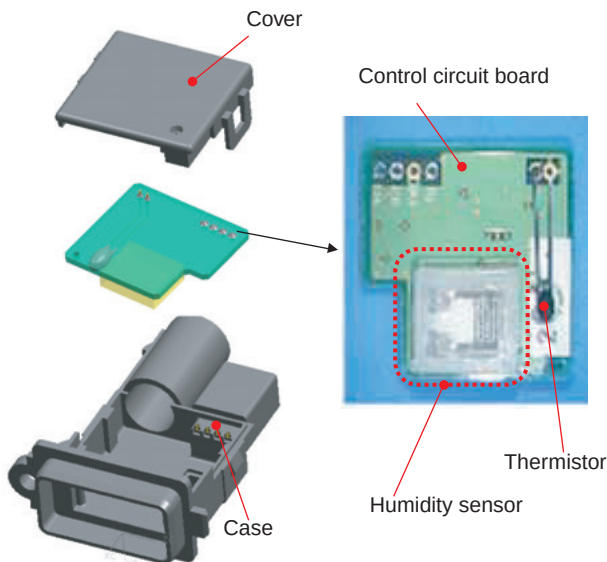


Fig. 12 Thermo & humidity sensor

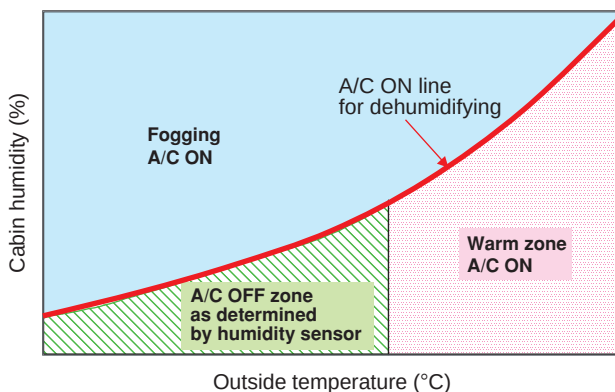


Fig. 13 Air conditioner operational chart

5. Fuel Economy and Driving Performance

The application of the technologies discussed above has resulted in the achievement of a 10-15 mode fuel economy of 31.0 km/L, representing a 5% increase against that of the previous model.

In addition, engine power and torque have been increased to 70 kW (+11%) and 123 Nm (+4%) respectively, and motor power and torque have been increased to 15.5 kW (+50%) and 123 Nm (+14%) respectively, when regenerating. These increases have resulted in the achievement of 0-100 km/h acceleration in 12.1 seconds (Figs. 14, 15).

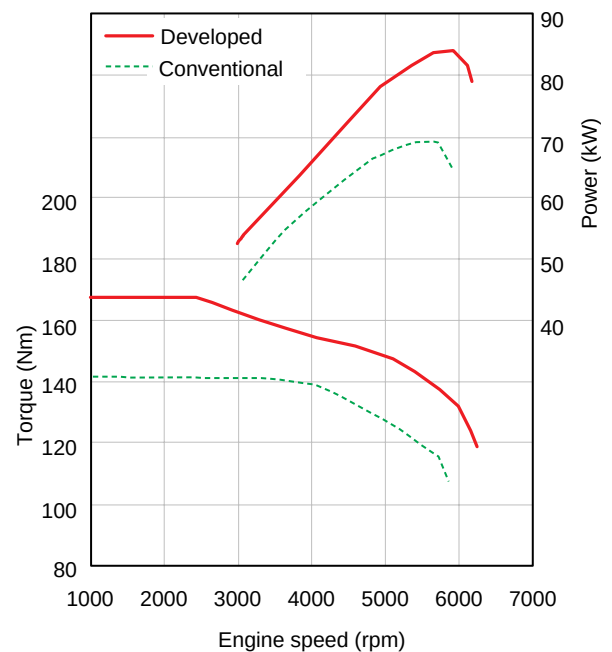


Fig. 14 Output performance of IMA system

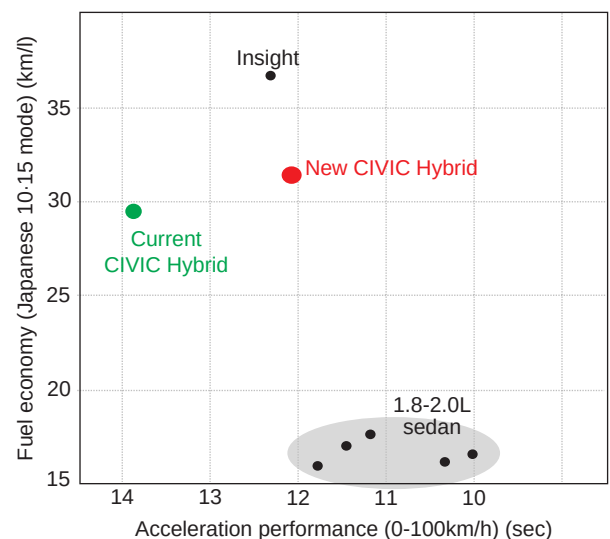


Fig. 15 0-100km/h performance/Fuel economy

6. Conclusion

The following technologies have been employed in the development of the 2006 year model CIVIC Hybrid in order to balance fuel economy and driving performance:

- (1) 3-stage i-VTEC has reduced pumping loss and increased engine power in the high-speed range.
- (2) Valves deactivation in all 4-cylinders has increased the amount of energy regenerated and enabled cruising on motor power.
- (3) A cooperative regeneration brake system has increased the amount of energy that is regenerated.
- (4) Modifications have been made to the components of the motor, battery and CVT to increase power and to respond to an increase in the amount of regenerated energy.
- (5) The use of a hybrid air-conditioner has enabled the range of the idle stop mechanism to be increased, and by adding air-conditioner control, employing a humidity sensor, the range in which the air-conditioner can be stopped has been increased.

As a result of the application of these technologies, the new CIVIC Hybrid has achieved an excellent balance between fuel economy and driving performance, with 31.0 km/L fuel economy in 10-15 mode (+5% against the previous model) and 0-100 km/h acceleration in 12.1 seconds (-1.8 seconds against the previous model).

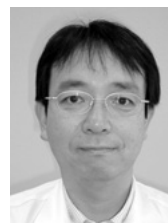
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Development of Inverter for 2006 Model Year CIVIC Hybrid

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ABSTRACT

Our goals in developing the inverter used in the CIVIC Hybrid Integrated Motor Assist (IMA) system were to reduce size and increase power. To increase the power, a C-Cu substrate was used to increase heat diffusion, resulting in a 6% rise at maximum power current compared with the former model. Several different technologies have been employed to reduce the size of the unit: A trench structure was used in the insulated gate bipolar transistor (IGBT) gates, the thickness of the chips was reduced, the diodes were spread out, a thin film capacitor was used and the control components of the motor and structural components were integrated. These changes resulted in a reduction of 47% in volume and 25% in weight compared with the former model.

1. Introduction

In light of the full remodeling of the CIVIC Hybrid, it was decided to develop an enhanced-performance inverter in-house. As shown in Fig. 1, the inverter is integrated into the Power Control Unit (PCU) in the Intelligent Power Unit (IPU), which is installed

on the back of the rear seat. Its role is power conversion between the motor and the battery. To enhance the power performance, fuel economy, and marketability of the vehicle, our goals were to increase the power and reduce the size of the inverter and other IPU parts. This paper describes the technologies used to accomplish those goals.

2. Technology for Achieving Increased Power

The maximum power current of the inverter is determined by the upper temperature limit of the semiconductor power chip (hereafter referred to as “chip”). In this development, the temperature rise of the chip was reduced and its maximum current value was increased by enhancing the heat radiation performance of the power module substrate.

2.1. High Heat Radiation Power Module Substrate

The power module substrate fulfills the functions of heat radiation, insulation, chip fixation and electric current conduction. Its heat radiation performance affects the temperature rise of the chip.

The temperature of the chip rises sharply when the vehicle is in motion such as engine starting that generates transient heat and continuous running that generates steady heat. The heat capacity

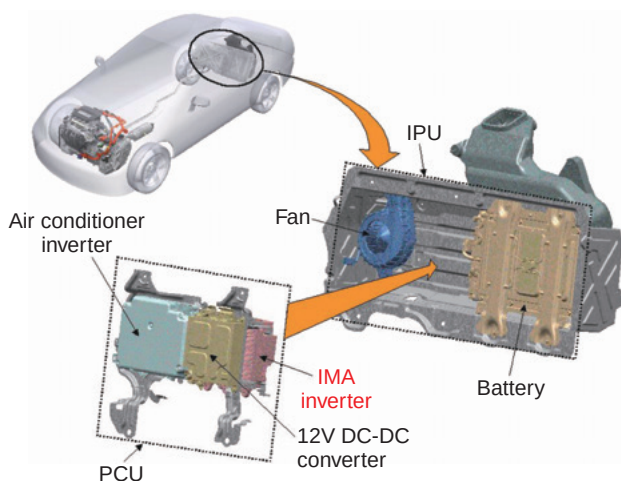


Fig. 1 Layout of IMA inverter

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and thermal conductivity of the material directly under the chip have a major influence on the temperature rise of the chip when transient heat is generated. As shown in the upper part of Fig. 2, the conventional power module substrate consists of a Direct Brazed Aluminum (DBA) substrate and a radiator plate (heat spreader), which are soldered together. The DBA substrate directly below the chip, in order from top down, consists of Al, AlN and Al layers. The relative thinness of the Al layers provides little thermal heat capacity. If the upper Al layer is made thicker, leading to increased thermal heat capacity, cracks are likely to occur in the soldering that connects the chip and the power module substrate. These cracks are possible because the coefficient of thermal expansion for Al is higher than that of AlN, which is made of ceramic, or of the chip, which is made of Si. Thus, the stress on the soldering increases if the thickness of Al is increased. As shown in the lower part of Fig. 2, the power module substrate developed is made of the following layers, beginning with the top: C-Cu, insulation and a heat radiator plate. C-Cu is a composite material made of a porous carbon substrate impregnated with Cu. It has a thermal conductivity equivalent to that of Cu, and a thermal expansion coefficient equivalent to that of Si. By arranging the thicker C-Cu layer directly below the chip, both a reduction in stress in the soldered section and high heat capacity were achieved. The heat generated in the chip is dispersed to the C-Cu, as shown in Fig. 3, and the transient thermal resistance is reduced in the area of the same chip, as shown in Fig. 4. At point A, when the engine is started, a reduction of

13% in transient thermal resistance was achieved in comparison to the conventional model.

In the case of steady heat generation, the heat radiation performance from a chip to the heat sink, which performs heat exchange, has a major influence on the temperature rise of the chip. The heat radiation performance in the power module substrate is determined by the thermal conductivity, distance and heat radiation area from the chip to the heat radiator plate, and can be expressed as thermal resistance. Equation (1) shows the thermal resistance equation.

$$R = \frac{L}{kA} \quad (1)$$

R is the thermal resistance, L is the thickness, k is the thermal conductivity and A is the area. The method of reducing the thermal resistance considering a single component is to use a material with high thermal conductivity, to increase the area through which heat passes and to decrease that area's thickness. However, the power module substrate is a multi-layered structure consisting of several materials, and the thermal resistance of the entire power module substrate can be reduced by dispersing heat to components with low thermal conductivity and by increasing the area through which heat passes. In this development, the C-Cu section, with high thermal conductivity, was made thicker to disperse heat, as shown in Fig. 5, and the area of insulating material with low thermal conductivity, through which heat passes, was made larger. High-

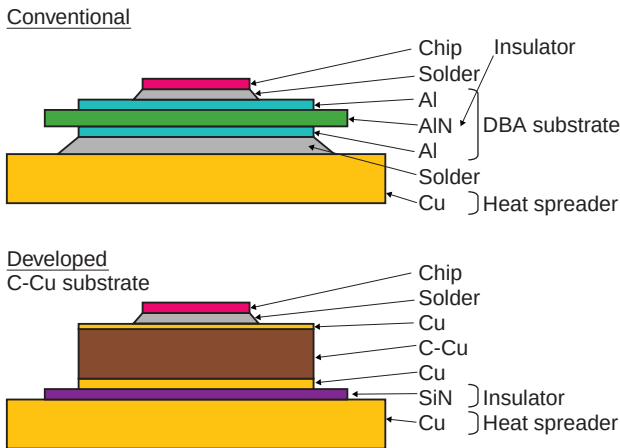


Fig. 2 Structure of power module substrate

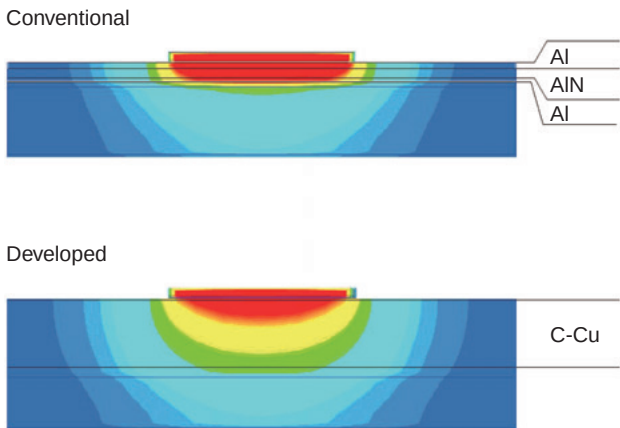


Fig. 3 Thermal simulation of transient conduction of heat

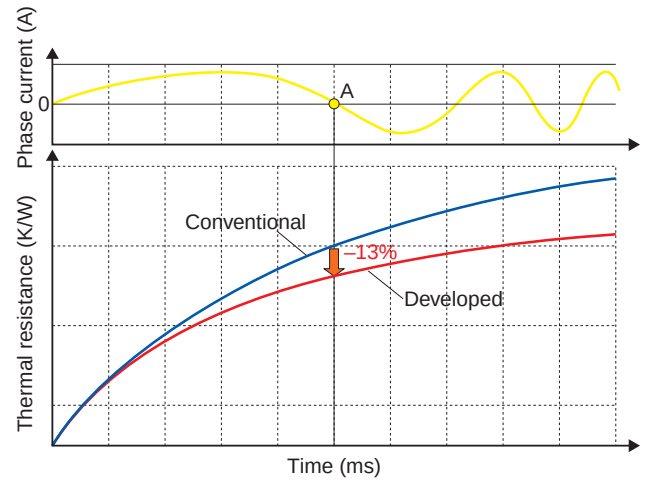


Fig. 4 Transient thermal resistance

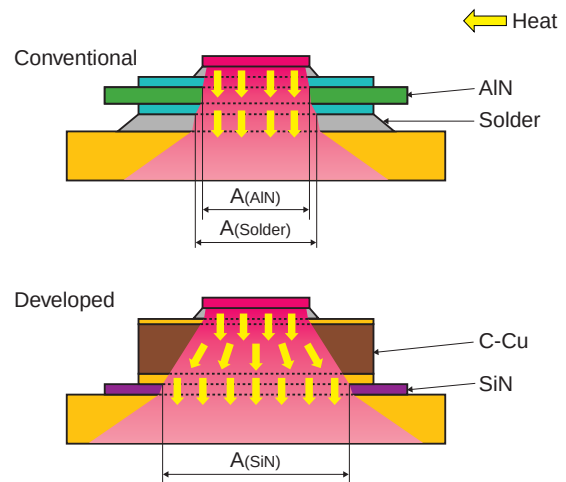


Fig. 5 Thermal diffusion

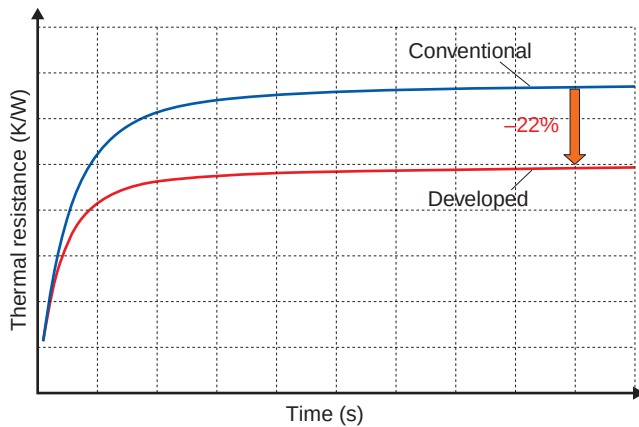


Fig. 6 Stationary thermal resistance

strength SiN was selected as the insulating material, and the section made of SiN was made thinner. The use of solder, which has low thermal conductivity, was avoided, and the C-Cu, insulating material, and heat spreader were brazed to form an integral body. Using these measures, as shown in Fig. 6, the steady state thermal resistance for the same chip area was reduced by 22% compared to the conventional structure.

3. Techniques for Achieving Compactness

Figure 7 shows the inverter and the functional components for motor control. Integration of components was the focus of attempts to achieve compactness. Individual items such as an inverter and other components have been made smaller in the past, but to achieve further compactness, functional components for motor control and inverter peripheral components were integrated, while at the same time eliminating connecting parts.

3.1. IGBT

The inverter uses IGBT as a switching device, allowing high power and high speed switching. The main current conducting section in the IGBT consists of pnp-type bipolar transistors, while the gates in the IGBT consist of Metal Oxide Semiconductor Field Effect Transistors (MOS FET). The IGBT possesses high speed switching characteristics in voltage drives of power MOS FET, and the high voltage and high current switching characteristics of a bipolar transistor.

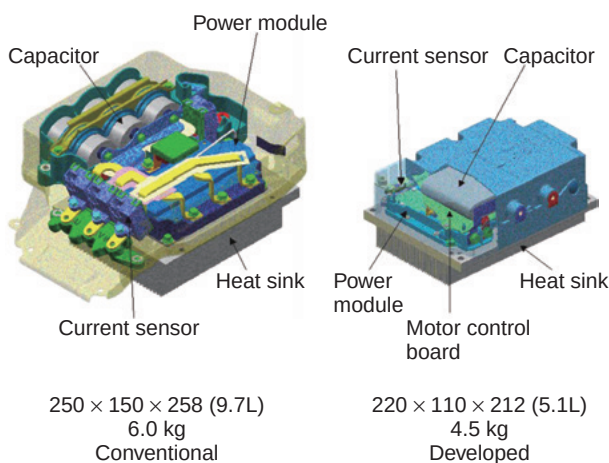


Fig. 7 Inverter configuration

Using techniques to reduce the size of the cells and the thickness of the chips and carrier accumulation technology, the loss per unit area in the IGBT was reduced and the chips were made more compact. The structure of the IGBT is shown in Fig. 8. In a conventional planar gate structure in which gates are formed on the surface of the chip, if the cells are made smaller, the current path narrows due to the expanded depletion layer, resulting in a steep increase in on-resistance. By adopting the trench gate IGBT structure, in which deep grooves of narrow width are made internally on the chip surface to form the gates on the side surface, the increase in on-resistance has been prevented and the cell size has been reduced. By arranging the gates with different spacing and by providing an insulating film in the wider spacing between gates, the carrier accumulation area was set in the N- layer and the modulation effect was enhanced⁽⁴⁾. On-resistance per unit area was reduced by reducing the resistivity due to the thinned chip and the adoption of the trench gate structure. The result was a reduction in chip surface of 46% compared to conventional structures.

The trench gate structure has several disadvantages, including a reduction in the short-circuit tolerance capability because of the increase in saturation current during short circuits, a delay in switching time and an increase in drive loss because of the increase in gate input capacity. By locally increasing the gate spacing in the adopted IGBT, the area of the MOS section is reduced, saturated current can be controlled and the input capacity can be reduced to the level similar to that found in a planar gate structure.

3.2. Diode

As shown in Fig. 9, a two-chip arrangement was used instead of the conventional one-chip arrangement, and heat was dissipated

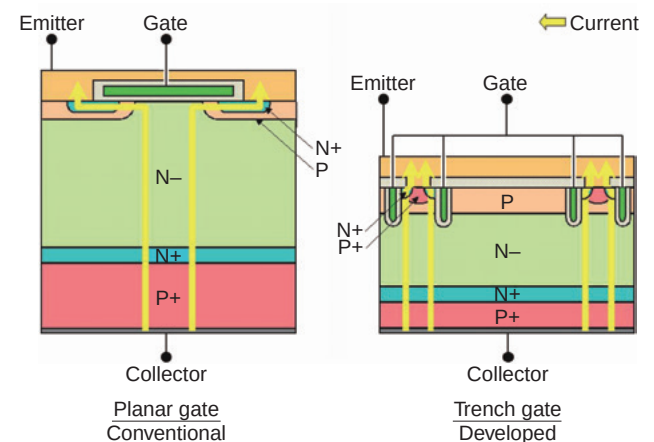


Fig. 8 Structure of IGBT

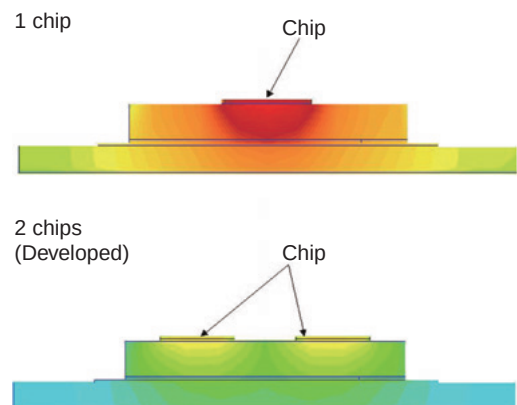


Fig. 9 Thermal simulation of conduction of heat

to enhance heat radiation performance. Thanks to the dispersed arrangement of chips, a reduction in thermal resistance of 50% was achieved in the same chip area, enabling a reduction in chip area of 44% compared to a conventional model.

3.3. Capacitor

Ways to reduce the electric capacitance were studied so that the capacitor could be made smaller. Usually, the IMA contactor is disconnected only when the current has been switched off, but sometimes it can be disconnected during current regeneration, as shown in Fig. 10, when the IMA system is in an unusual condition, and the voltage at inverter terminals rises abruptly⁽²⁾. Electric capacitance not only affects the voltage smoothing, but also the rise in voltage. If the electric capacitance is reduced, the voltage rise time decreases. In this inverter, the electric capacitance is reduced by compressing the response time for over-voltage protection by 90% compared to conventional models.

The electric capacitance of the conventional electrolytic capacitor was high, but since the equivalent series resistance (ESR) is high because of the electrolyte, the heat generated due to ripple current is large. The allowable ripple current and the electric capacitance are proportional to each other, as shown in Fig. 11. When the electric capacitance is reduced, the allowable ripple current of the electrolytic capacitor cannot satisfy vehicle requirements, and the capacitor cannot be made smaller than that of the conventional model. In contrast, the electric capacitance of the film capacitor is small, but since the ESR is also low, the allowable ripple current is high. A film capacitor with the minimum film thickness was used as the smoothing capacitor for vehicles in this inverter. As shown in Fig. 12, a reduction in capacity of 40% compared to conventional models was achieved by flattening the capacitor element shape.

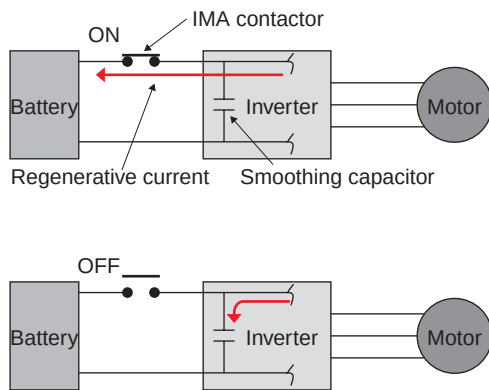


Fig. 10 IMA system diagram

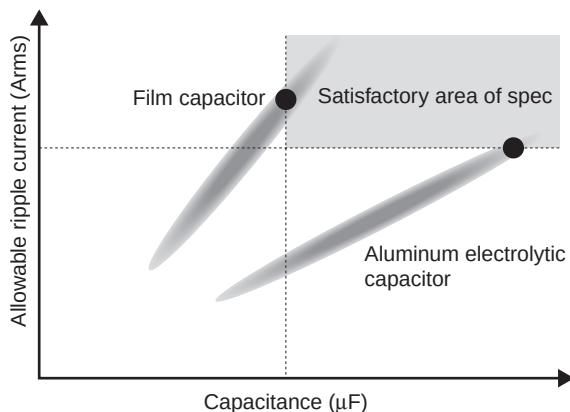


Fig. 11 Comparison of capacitor characteristics

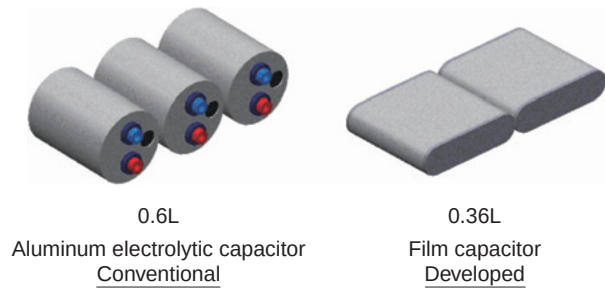


Fig. 12 Comparison of capacitor

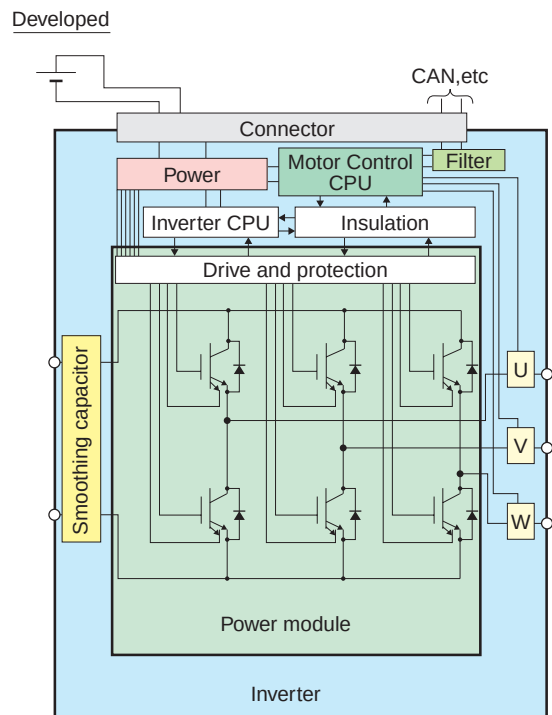
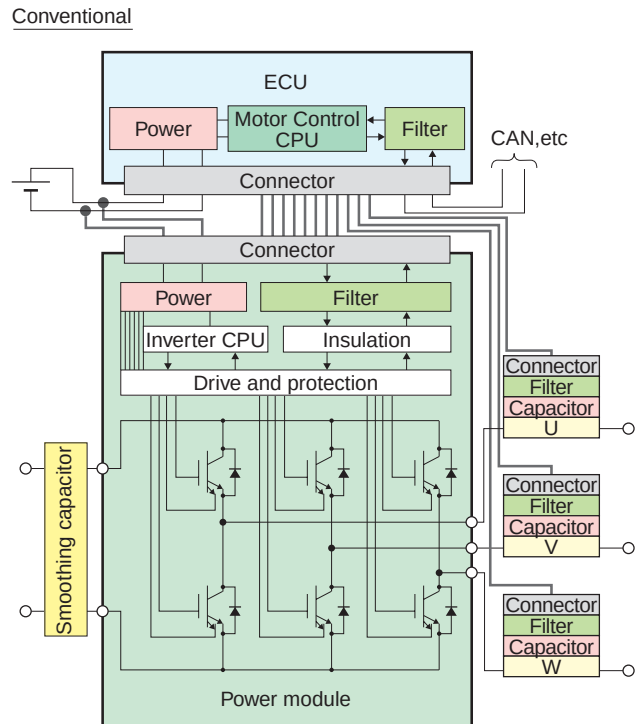


Fig. 13 Block diagram of inverter and peripheral parts

3.4. Integrating Motor Control Functions and Current Sensor

Figure 13 shows the block diagram of inverter and peripheral components. The motor control, battery control and cooling control functions were integrated in the conventional ECU, and many harnesses were used to connect the ECU and the components associated with motor control functions. To simplify the system components, the motor control functions and current sensors were integrated in the inverter, and the ECU and inverter were connected by CAN. The above measures enabled the elimination of harnesses and interface circuits, and allowed the sharing of the power source and filters.

Figure 14 shows the internal configuration of the inverter. The connectors and harnesses have been eliminated by changing from conventional connectors to pin terminals to connect the gate drive board, the motor control board and the current sensor board.

Figure 15 shows the structure of the connection between the gate drive board and the motor control board. The pin terminal assembly consists of a clamp housing, an adjustable housing and pin terminals. The adjustable housing is used to correct the offset in position between the gate drive board and the motor control board during assembly, and the pin terminal is normally inserted in the motor control board.

Figure 16 shows the construction of the connection between the motor control board and the current sensor board. The pin terminals are directly installed in the current sensor board, and connected to the motor control board by soldering. The offset in position of the motor control board and the current sensor board during assembly is corrected by adjusting the pin terminal position using the pin guide installed on the motor control board side. The current sensor volume has been reduced to 73% of that of the conventional model by sharing the power source and filters with the motor control board.

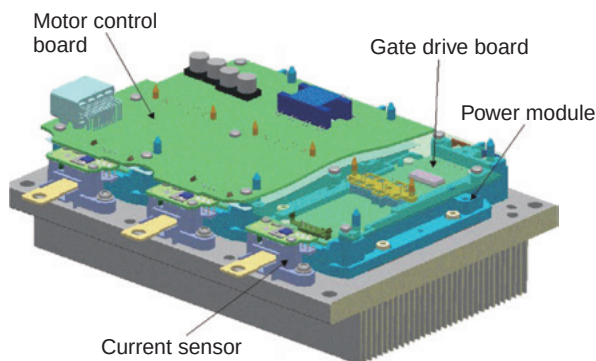


Fig. 14 Interior of inverter

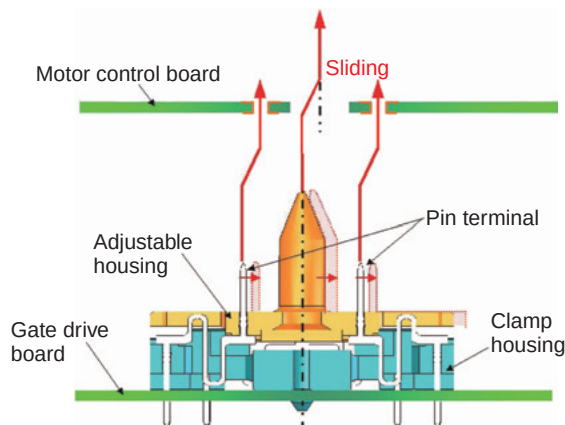


Fig. 15 Structure of pin terminal

3.5. Integration of Functional Components

Functions were concentrated in the capacitor unit to integrate electrical connections between components and fixation of components. Figure 17 shows the configuration of the capacitor unit. The capacitor and noise filters were fixed to the conventional inverter cover and connected to the same bus bar. The ground bus bar of the noise filter was extended up to the component to fix the capacitor unit in place, and is used for ground connection as well. The inverter cover was fitted with input and output terminal blocks, and the inverter output bus bar was extended, eliminating the conventional output bus bar. After adopting the measures above, the number of components was reduced, as shown in Fig. 18.

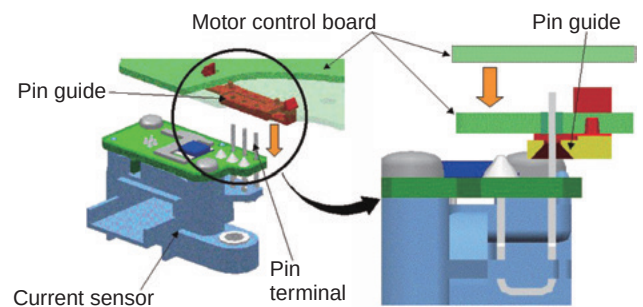


Fig. 16 Configuration of pin terminal

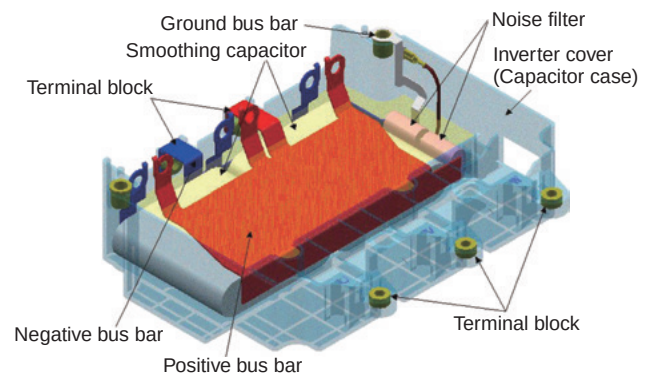


Fig. 17 Configuration of complete capacitor

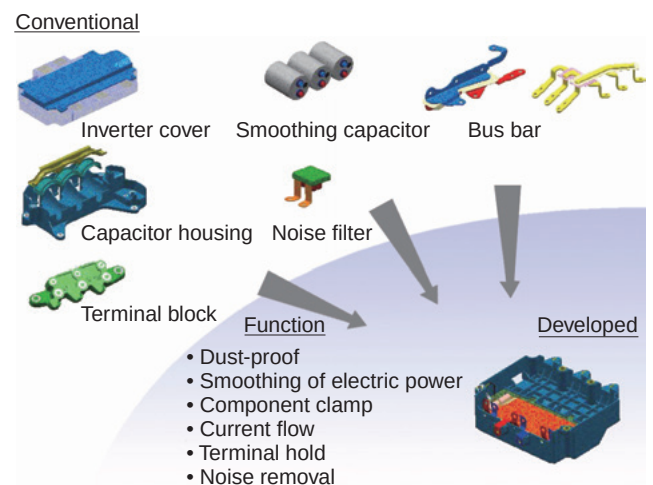


Fig. 18 Integration of capacitor components

4. Conclusion

An inverter for the new CIVIC Hybrid was developed, incorporating the modifications to conventional inverters listed below.

- (1) Inverter maximum power current was increased by 6% by adopting a C-Cu substrate with high heat radiation performance.
- (2) IGBT chip area was reduced by 46% by adopting IGBT trench gate structure and by using a thinner chip
- (3) Diode chip area was reduced by 44% by adopting a dispersed diode arrangement
- (4) Capacity of the capacitor was reduced by 40% by adopting a thin film flat capacitor
- (5) By integrating motor control function and the current sensor in the inverter, connectors and harnesses were eliminated, the number of components mounted on the board was reduced and the current sensor volume was reduced by 73%.
- (6) The number of components was reduced by integrating functional components.

As a result of adopting the measures above, the maximum power current was increased by 6%, the inverter volume reduced by 47% and the inverter weight reduced by 25%, meaning that target values were achieved. In this way, the technologies for achieving higher power and a more compact inverter have contributed to enhancing the power performance, fuel economy and marketability of the Honda IMA system.

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Development of Lightweight and Compact 6-speed Manual Transmission

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ABSTRACT

A high capacity 6-speed manual transmission has been developed based on the existing compact, well-established 5-speed manual transmission for 1.5-liter engines without increasing the overall length of the transmission. In order to ensure both high capacity and compactness, the synchronizer sleeve stroke of each gear has been reduced, and the specification and width for each gear was optimized. Gear noise was reduced by using a rigid, yet lightweight case shape designed using CAE analysis. The resulting 6-speed transmission has an overall length of 352 mm and total weight of 47 kg, making it light weight and compact.

In addition, a new structure was employed for the change link mechanism in order to enhance the gear-shift feel.

1. Introduction

Though 6-speed manual transmissions (hereafter 6MTs) were formerly installed mainly in large-displacement and sporty vehicles, market trends in recent years suggest that they are increasingly being installed into ordinary, small-displacement vehicles. Currently, this trend is centered in Europe.

Honda has already installed 6MTs into engines of 2.0 L or larger displacement as well. Now, however, a new 6MT has been developed for installation in compact vehicles with small-displacement engines. For that purpose, the new 6MT has been made more compact with lower friction. This article will give an introductory overview of this 6MT and the main elements of the techniques involved.

2. Development Goals

The development concepts for the new 6MT included “achieving a balance of high capacity and lightweight compactness,” “enhanced gear shift feeling” and “enhanced transmission efficiency.” The four items described below were identified as objectives for the purpose of realizing the concepts for development.

- (1) Achieve a 6MT having, at most, the same total length as the conventional 5MT used with 1.5-L engines to ensure that there is an adequate degree of freedom for it to be installed into small-displacement vehicles.
- (2) Achieve a balance between enhanced noise and vibration (NV) performance and lighter weight.
- (3) Provide a crisper, smoother gear shift feeling.
- (4) Achieve a friction level equal to or lower than the conventional 5MT.

3. Basic Construction and Specifications

The external appearance and major sectional views are shown in Fig. 1 and Fig. 2, respectively. The main specifications are shown in Table 1.

The 6MT gear train is a parallel dual-shaft configuration. Gears 1 through 6 are synchromesh gears that are always in contact. The reverse gear has a selectable sliding configuration. A high-capacity single cone synchronizer was adopted, and the arrangement of the bearings was optimized as a way of reducing friction in the gear train.

As shown in Fig. 1, a new structure was set up for the change link mechanism and situated in the upper part of the gear train.

* Tochigi R&D Center

Table 1 Specifications of transmission

Transmission type		6-speed Manual
Torque capacity (Nm)		Over 175
Gear ratio	1st	3.142
	2nd	1.869
	3rd	1.303
	4th	1.054
	5th	0.853
	6th	0.727
	Reverse	3.307
Weight (wet)	(kg)	47

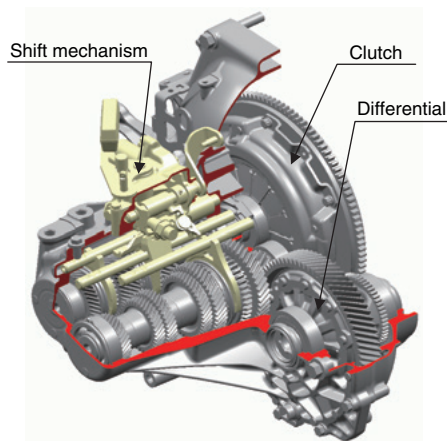


Fig. 1 View of transmission

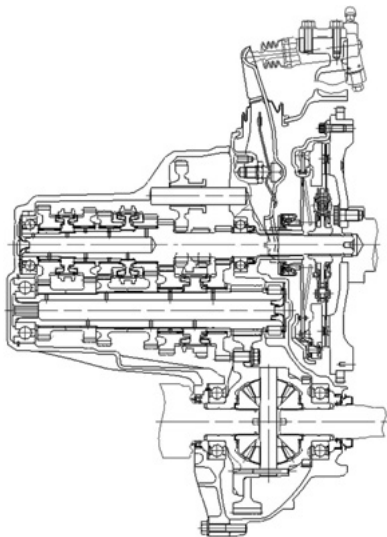


Fig. 2 Cross-sectional view of transmission

4. Techniques for Achieving Compactness

4.1. Gear Train Construction

Development proceeded with the objective of building a 6MT based on the existing 5MT for 1.5-L use and having the same length as the 5MT. This objective was achieved through the development of techniques for shortening the overall length so that in creating the 6MT, the expected overall lengthening would not occur (Fig. 3).

4.2. Shortening the Synchronizer Sleeve Stroke

The factors that determine the total length of a manual transmission include the dimensions of the gear and bearing components and the stroke space of the synchronizer sleeve (hereafter sleeve), which moves when the gears are changed.

The sleeve functions to actuate and synchronize the synchronizer ring. When the gear shifting is complete, the sleeve also meshes with the clutch gear and functions to transmit the power. The sleeve stroke is determined by the sum total of the dimensions necessary to fulfill those functions. Figure 4 shows the component elements of the sleeve stroke. In this development, the sleeve stroke was reduced by shortening the distance from completion of synchronizing to gear contact, as shown in Fig. 4.

That portion of the stroke was made shorter by changing the shape of the sleeve in first and second gears, and by revising the synchronizer ring shape for the remaining gears. The following describes the change of the sleeve shape for those remaining gears.

Conventional sleeves would use all their teeth for both their synchronizing and clutch gear meshing functions. In this development, however, the teeth used for synchronizing and the teeth used for meshing were separated and a relative difference in level was placed between them. This resulted in a shorter distance between synchronizing and contact with the clutch gear, achieving an approximate 5 mm shortening of the total length of the six gears. This was also effective in reducing the double-bump feeling, which will be discussed later. A structural comparison of the conventional sleeve and the newly developed one is shown in Fig. 5.

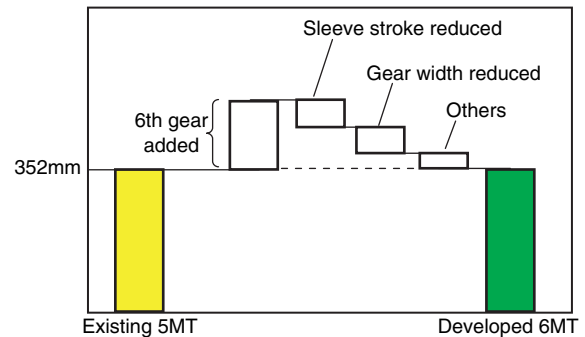


Fig. 3 Effect of reducing length of each part

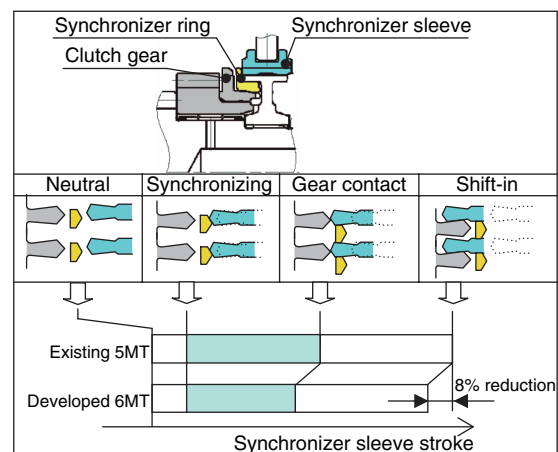


Fig. 4 Comparison of synchronizer sleeve stroke

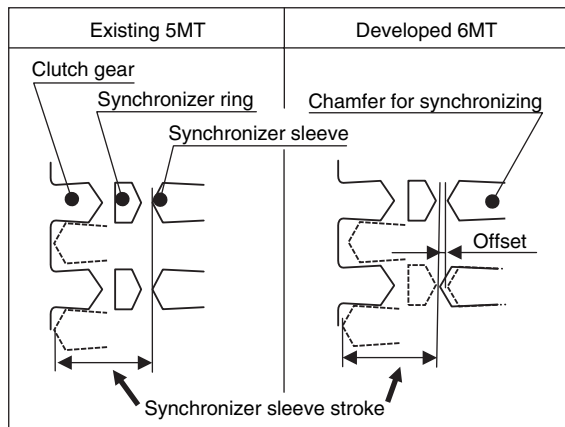


Fig. 5 Comparison of synchronizer sleeve shape

4.3. Reducing the Gear Width

There were concerns that reduction of the gear width would affect the durability of the gear itself and increase gear noise. In this development, some of the gears were made into small modules and the specifications were optimized in order to assure durability. This also increased the contact ratio, which affects gear noise. As a result, the gear width reduction yielded a 4 mm shortening of the total length.

5. Techniques for Achieving Lighter Weight

The use of techniques to achieve more compact dimensions and efforts to achieve lighter weight made it possible to develop a 6MT with a weight of 47 kg (wet), which is the same as that of a 5MT. A comparison of the total transmission weight is shown in Fig. 6. The main aspect of this weight-optimization effort to be described here is the design of the transmission case, which has the greatest impact on weight reduction.

Advances in CAE analysis technology have made it possible to use FEM models for analysis at the design stage. This means that stress, rigidity and behavior can be predicted accurately before fabricating a prototype. Using this technology not only enhances development efficiency, but also enhances accuracy in predicting the effects of means used to achieve objectives. In this development, the above-mentioned technology made it possible to create an effective design to achieve the dual objectives of rigidity and lighter weight, demands that are mutually contradictory⁽¹⁾.

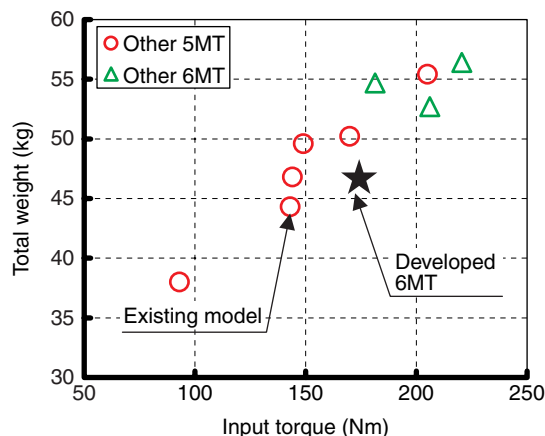


Fig. 6 Comparison of transmission weight

Making use of these analytical techniques, a new case structure was designed with the objective of raising rigidity to enhance NV performance with a minimum amount of weight increase. Analysis by FEM modeling of the case at the initial stages of design made it possible to predict vibration characteristics and stress. An effective rib layout was made and the case shape reevaluated accordingly to minimize surface vibration and thus mitigate the radiant noise⁽²⁾. As a result, the performance objective was achieved with a minimal weight increase relative to the base 5MT (Fig. 7).

The case example of analysis of vibration characteristics conducted for this development is illustrated in Fig. 8. The vibration characteristics obtained by the analysis are shown in Fig. 9.

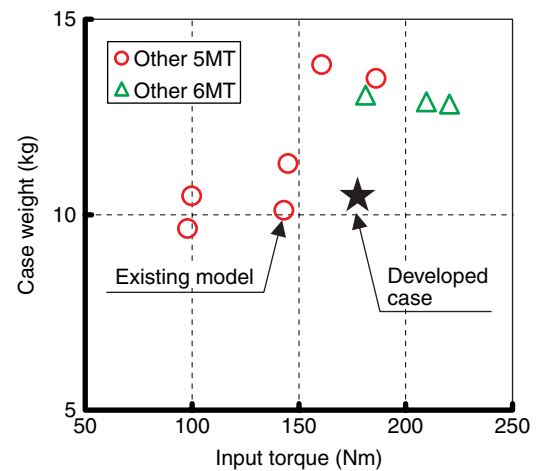


Fig. 7 Comparison of case weight

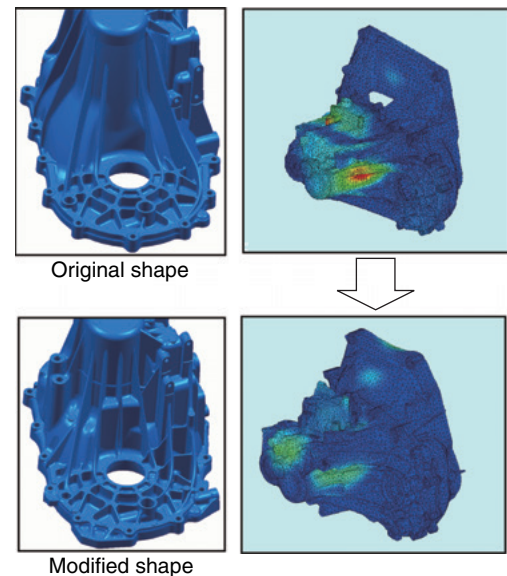


Fig. 8 Comparison of model shape

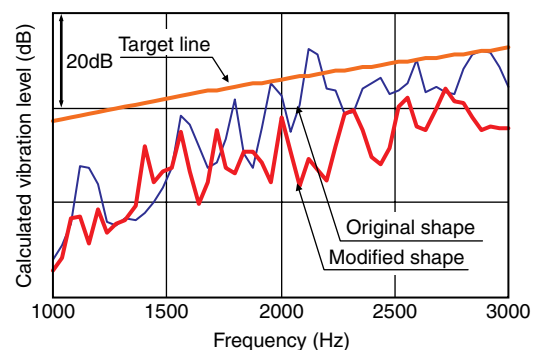


Fig. 9 Comparison of sound pressure level

6. Enhancing the Gear Shift Feeling

In order to realize the development objective of providing a crisper, smoother gear shift feeling, development focused on the following items:

- (1) Reducing the friction sensed when selecting a gear (lateral movement)
- (2) Reducing the effort involved in shifting gears (longitudinal movement)
- (3) Reducing the double-bump feeling

The change link mechanism was made compact by placing the shift selector and the reverse gear lockout as a unit in a single location (Fig. 10).

6.1. Reducing Friction during Gear Selection

Figure 11 shows the selector mechanism. The change link parts turn and slide during gear selection. This causes friction, a negative factor with regard to the shift feeling.

In order to reduce that friction, the development eliminated the link, which was the largest cause of friction during gear selection. Instead, the unit was designed so that the select cable would directly cause the select lever shaft to turn to select a gear position. The mount for that shaft was also equipped with a needle bearing to reduce the rotational friction. As shown in Fig. 12, these measures reduced the overall friction in the selector mechanism by 80% as compared to the conventional mechanism, thereby achieving smoother gear shifts with less sense of hysteresis.

Figure 13 shows the resulting selection effort characteristics.

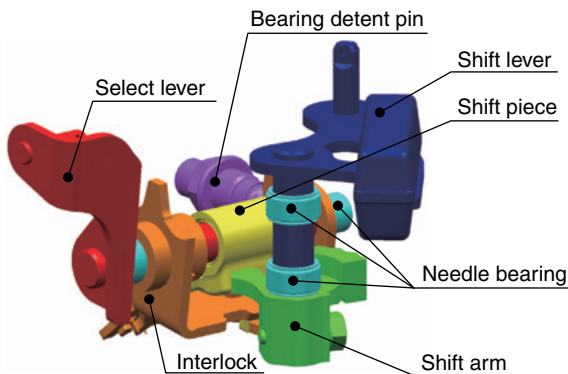


Fig. 10 View of gear change linkage

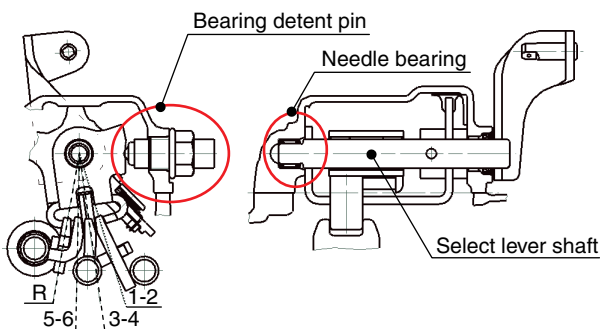


Fig. 11 Gear selector mechanism

6.2. Reducing the Effort of Shifting Gears

Conceivable means of reducing the effort needed to shift gears include not only raising the capacity of the synchronizer, but also increasing the lever ratio from the shift knob to the sleeve inside the transmission. Generally, however, increasing the lever ratio also requires lengthening the shift stroke, which tends to interfere with the light, sporty gear shift feeling.

To increase the lever ratio without lengthening the shift stroke, the shape of the operating part of the shift arm inside the change unit was altered. As shown in Fig. 14, two circles of different radii were combined to form a compound circle shape. This structure enables the lever ratio to be further extended, only when needed, for synchronizing that requires greater effort. Combined with the above shortening of the sleeve stroke, which served to increase the lever ratio, this structural modification enabled an approximate 14%

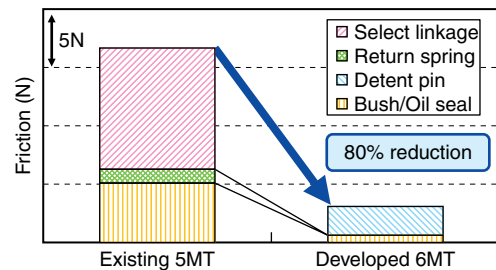


Fig. 12 Comparison of select lever friction

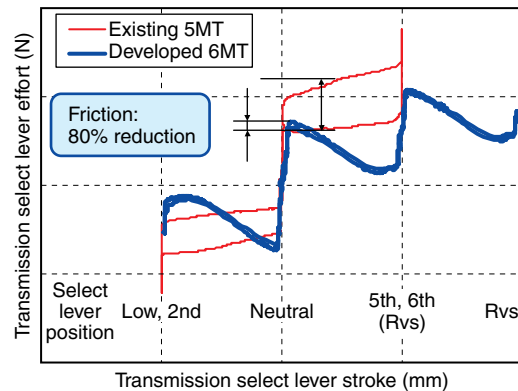


Fig. 13 Characteristics of select lever operation

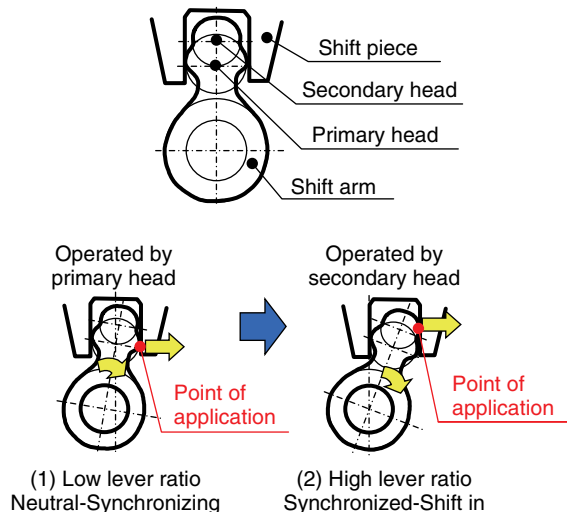


Fig. 14 Operation of variable lever ratio shift arm

reduction in the effort needed to shift gear. Figure 15 shows the effect on reducing the effort required to shift gears.

6.3. Reducing the Double-Bump Feeling

When using the gear shift, the load that occurs like a secondary kick-back after synchronizing ends is called the “double-bump” feeling. This double-bump degrades the shift feeling and occurs when the torsion deformation in the shaft being synchronized is released after synchronizing. This results in another differential rotation, which strongly resists the resultant sleeve-clutch gear contact. The resulting vibration is transmitted to the shift knob as the double-bump⁽³⁾. Figure 16 shows the principle of the double-bump effect.

The shortening of the synchronizer sleeve stroke, as described above, causes a shortening of the stroke and the time from when synchronizing ends to when the sleeve and clutch gear engage. This suppresses occurrence of the differential rotation, and has reduced the occurrence of the double-bump effect by approximately 30%.

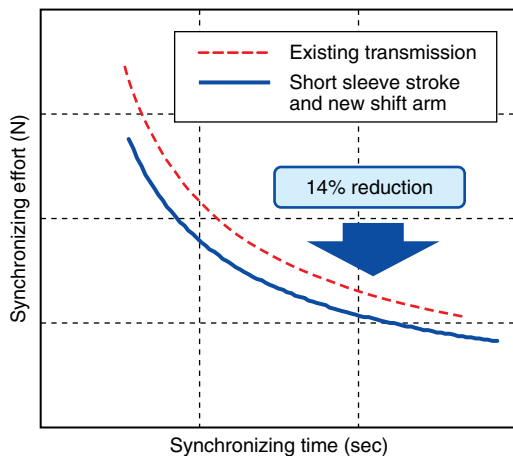


Fig. 15 Reduction of synchronizing effort

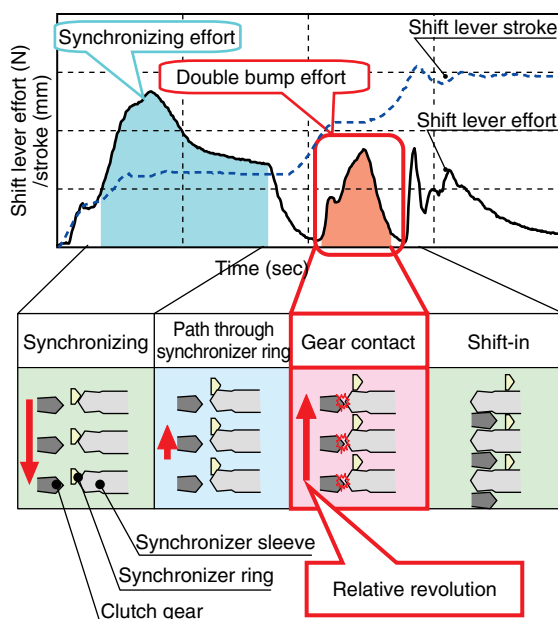


Fig. 16 Principle of double bump effort

7. Enhancing Transmission Efficiency

The factors that reduce transmission efficiency of a manual transmission include, for example, oil agitation resistance and friction in the synchronizer, bearings and other parts of the gear train.

The shift effort-reducing techniques used for this development were combined with the adoption of a high-capacity, single-cone synchronizer for 2nd gear. This made it possible to eliminate the double-cone synchronizer that was formerly used and reduced the rotational resistance occurring in the synchronizer. Conversion to a 6MT also involved reevaluation of the arrangement of bearings, and the roller bearings were consequently eliminated. The ribs located on the interior walls of the case were also made smoother, thus reducing the resistance caused by oil agitation.

The results are shown in Fig. 17. Although conversion to a 6MT caused some increase in friction, it was possible to reduce the friction felt when cruising at 60 km/h by approximately 13% as compared to the conventional 5MT.

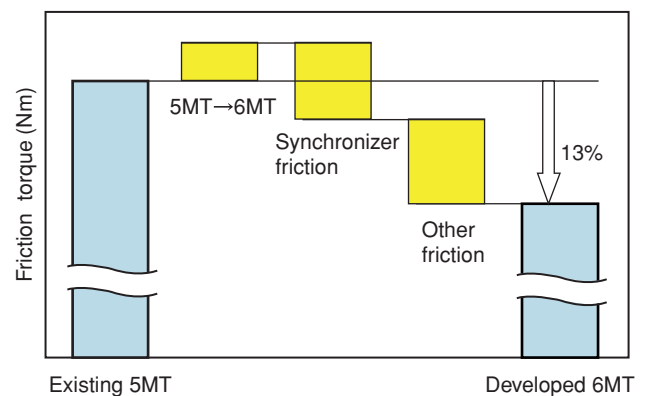


Fig. 17 Transmission friction torque at 60km/h

8. Conclusion

- (1) The total length of the transmission was made shorter by shortening the synchronizer sleeve stroke, and techniques for new gear specifications were established. The result of these methods was the development of a 6MT that has a total length equivalent to that of a conventional 5MT.
- (2) FEM modeling analysis was used to seek higher torque response and reduced vibration. At the same time, a case was developed with a shape that makes it the lightest in its class.
- (3) The friction in the selecting direction when shifting gears was reduced by approximately 80% by means of the structure of the change link mechanism and component technologies for friction reduction.
- (4) The synchronizer sleeve stroke was shortened and the change mechanism was enhanced to reduce the effort needed to shift gears. This also mitigated the double-bump feeling.
- (5) By means including optimization of the synchronizer and bearing specifications, the resulting transmission design, despite being a 6MT, achieved a reduction in friction of approximately 13% as compared to a conventional 5MT.

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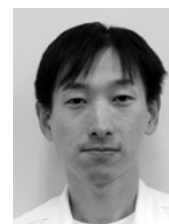
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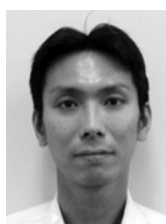
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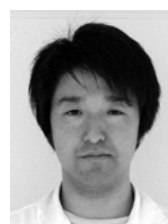
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Development of Plastic Extra Window

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ABSTRACT

A large, extra window made of plastic with a complex 3-dimensional shape has been developed for use on the 2006 European CIVIC. The extra window was produced by a double-injection process using transparent and black resins and fixed to the tailgate panel with urethane adhesive. To ensure scratch resistance, the full outer and transparent inner surfaces of the extra window have a hard coating. The hard coating with the best weathering resistance was chosen from among polysiloxane hard coatings with good scratch resistance. Using an automated flow coat process to apply the hard coat mean the hard coat film thickness could be controlled within the appropriate range, ensuring superior durability of the coating. The new window provides the driver with an enhanced rear view and gives the car a whole new look.

1. Introduction

Conventionally, inorganic glass has been used for automotive windows. Compared to other materials used for exterior parts, such as steel plate, aluminum sheeting and plastic, there is less freedom in shaping and machining inorganic glass, so window design has been more limited than that of other exterior parts.

Recently, however, there has been more research into and development of alternative materials⁽¹⁾ that can be used in more complex window designs, and the application of these alternative materials to quarter windows and tailgate windows has begun^{(2), (3)}.

One design feature of the new European CIVIC is a tailgate spoiler located high on the tailgate that includes an integrated “extra window.” Figure 1 shows the extra window in the new European CIVIC. This window is large, with a complex 3-dimensional shape, and the transparent panel built into the extra window provides the driver with enhanced rear visibility.

This paper introduces the technologies involved in the plastic extra window that has been adopted in the new 2006 European CIVIC.



Fig. 1 Rear view of the new European CIVIC

* Tochigi R&D Center

** Honda R&D Europe (UK)

2. Basic Structure of the Extra Window

Figure 2 shows the shape of the extra window. The transparent panel is deeply curved, and there is no dividing line between the transparent panel and the spoiler.

Figure 3 shows the inside structure of the extra window. In attaching the extra window to the tailgate panel, urethane adhesive is applied around the entire edge to ensure a water tight seal. The inside of the extra window has clips that provide temporary support until the adhesive cures, and a high mount brake light must also be attached to the inside of the extra window.

To achieve this design and inside structure with the fewest number of parts, the extra window is formed using transparent and black plastics and a double injection process to create an integrated extra window. Specifically, after injection molding the entire extra window using transparent plastic, black plastic is used for injection molding at designated locations on the inside. The black plastic on the inside gives the appearance of black ceramic around the periphery of the transparent panel, and at the same time contributes to an overall increase in rigidity of the extra window.

Ordinarily, the scratch resistance of alternative materials is much lower than that of inorganic glass. Therefore, a hard coat was applied to the entire external surface, which must be scratch-resistant because of its exposure, and to the inside surface of the transparent panel.

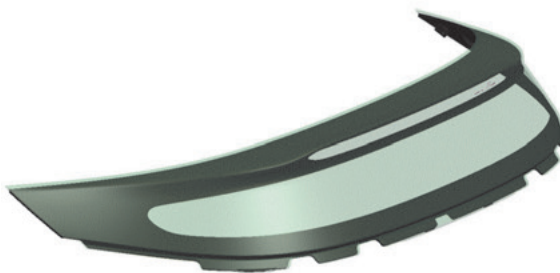


Fig. 2 Shape of extra window

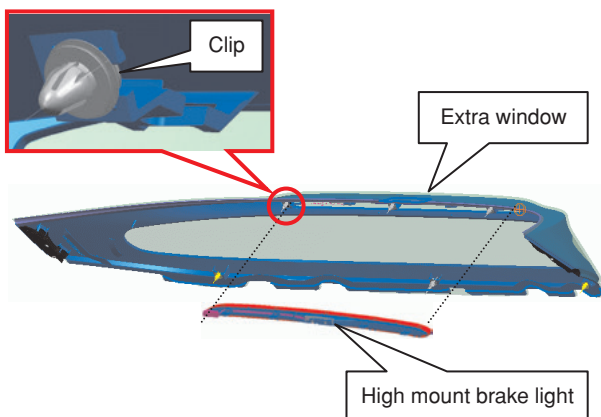


Fig. 3 Inside structure of extra window

3. Selection of Plastic Materials

A polycarbonate with advanced transparency and thermal stability was used for the transparent plastic. The transparent plastic was tinted green to match the colour of the rear window.

To prevent delamination of the black and transparent plastics after the double injection process, it was necessary to select a black plastic similar in type to the transparent plastic. In addition, to ensure the adhesiveness of the urethane adhesive, it was necessary to apply a solvent primer for plastics to the black plastic's adhesive surface. Therefore, a polycarbonate alloy was selected for the black plastic, providing chemical resistance superior to that of polycarbonate alone.

To confirm adhesion when the urethane adhesive was used with the black plastic, a shear adhesive test was performed. The results confirmed that the urethane adhesive had cohesive failure across the adhesive surface, and that there was sufficient adhesion. It was also confirmed that there were no plastic primer application irregularities caused by dissolution of the black plastic surface.

4. Selection of the Hard Coat Material

To qualify as providing enhanced rear visibility, the extra window's transparent panel must satisfy the abrasion resistance test stipulated in European regulations (ECE 43) on automotive safety glass material. To do this, the material had to have even higher scratch resistance than that of headlight lenses. Therefore, a polysiloxane hard coat was selected for the extra window rather than a conventional acrylic urethane hard coat often used for headlight lenses.

Moreover, because the extra window is designed with a comparatively large horizontal surface, it is exposed to more UV radiation than headlight lenses. Accordingly, the extra window's hard coat must have greater weathering resistance. Accelerated weathering tests were carried out on five types of polysiloxane hard coats, using a Sunshine Weather-O-Meter (SWOM) and a QUV Weather Tester (QUV). A durability comparison was made by timing the period until cracks appeared on the hard coat surface. The SWOM accelerated weather test conditions and test results are given in Table 1 and Fig. 4, and the QUV accelerated weathering test conditions and test results are given in Table 2 and Fig. 5. Compared to conventional polysiloxane hard coats (hard coats 1 – 4) used on headlight lenses, hard coat 5 had weathering adhesive resistance that was 4.5 times greater than that of the others in the SWOM tests, and 5 times greater than that of the others in the QUV tests. Therefore, hard coat 5 was selected for the extra window.

Hard coat 5 has a double layer structure, with a top layer and a primer layer. Each layer receives a thermal curing treatment, creating a 2-coat, 2-bake specification. Hard coat 5 has the two features, (1) and (2) below, and these features provide both the scratch resistance and weathering adhesive resistance required by the extra window.

- (1) By adding a UV absorber to the primer layer, it was possible to add a lot of UV absorber without any drop in the top layer's surface hardness.
- (2) By carrying out thermal curing treatment of the primer layer independently, it was easier to control the primer layer's film thickness and to obtain a stable hard coat adhesiveness.

Table 1 SWOM accelerating weathering test conditions

	Item	Test condition
Test subject condition	Substrate	Flat piece of black polycarbonate
	Coating thickness	Proposal thickness of hard coat supplier
	Coating condition	Proposal condition of hard coat supplier
SWOM condition	Basic condition	In compliance with JIS D 0205
	Back panel temp.	63±3°C
	Rain shower time	12 min. during 60 min.

Table 2 QUV accelerating weathering test conditions

	Item	Test condition
Test subject condition	Substrate	Flat piece of black polycarbonate
	Coating thickness	Proposal thickness of hard coat supplier
	Coating condition	Proposal condition of hard coat supplier
QUV condition	Basic condition	In compliance with JIS D 0205
	Light exposure temp.	63±3°C
	Condensation temp.	50±2°C
	Operating condition	Repetition of light exposure for 4 hours - condensation for 4 hours

Hard coat 5 is only produced in North America and because it has a short shelf life, supplying Europe from North America presented logistical difficulties. Therefore, in combination with the development of the extra window, European localization of hard coat 5 production was carried out, making a stable supply possible.

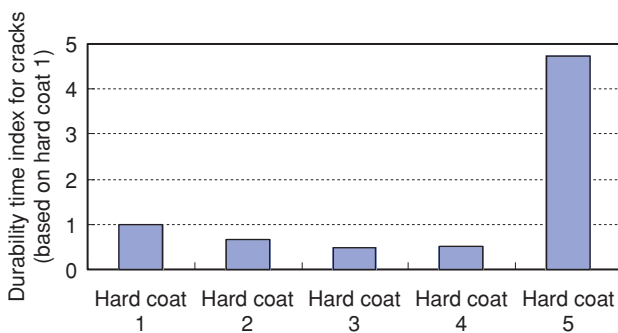


Fig. 4 SWOM accelerating weathering test results

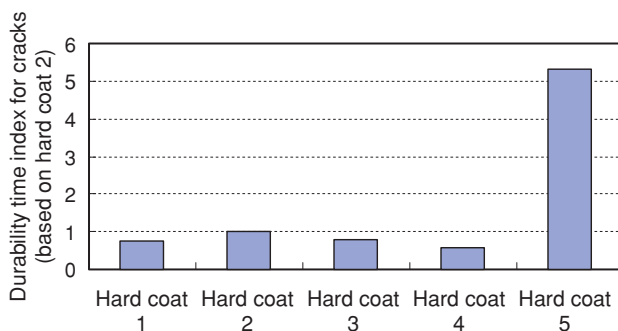


Fig. 5 QUV accelerating weathering test results

5. Hard Coat Film Thickness Control

To prevent scratching and environmentally induced deterioration under operating environment conditions, it was important to be able to control hard coat film thickness, as well as hard coat material selection. A flow coat process was used as the extra window's hard coat coating method, allowing comparatively uniform coating over complex shapes. In the flow coat process, the coating film is produced by having the coating liquid discharged from a nozzle and flow over the object being coated.

Furthermore, by optimizing the coating liquid parameters, coating liquid application conditions, and position in which the extra window was held during the flow coat process, the hard coat film thickness was controlled within the desired range. Automating the flow coat process also stabilized the hard coat film thickness.

Durability tests have confirmed that, under external factors and environmental conditions assumed for the operating environment, extra windows coated in this way are not subject to scratches, hard coat cracks, or yellowing of the transparent panel.

6. Dimension Accuracy and Appearance Quality Controls

Figure 6 shows the extra window's production flow.

In the hard coat's thermal curing treatment, the extra window is exposed to high temperatures for an extended period of time. To obtain the correct part dimensions after hard coat coating, it is necessary to control deformation due to residual stress during the thermal curing treatment or due to the window's own weight. By optimizing injection molding parameters in the double injection process, residual stress of parts after molding is controlled at a low level. Furthermore, by optimizing the shape of the jig that holds the extra window, deformation caused by the extra window's own weight is controlled.

In addition, the hard coat process is carried out entirely in a clean room to provide the high-quality appearance required of an exterior part.

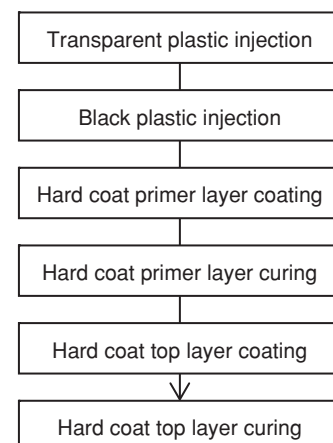


Fig. 6 Manufacturing flow for extra window

7. Conclusion

We have developed a plastic extra window that integrates the tailgate spoiler and window by using a polycarbonate coated with a polysiloxane hard coat. The extra window has been adopted in the new 2006 European CIVIC.

- (1) A double injection molding process using transparent and black plastics allowed for a distinctive design and inside structure. Use of a polycarbonate for the transparent plastic and a polycarbonate alloy for the black plastic ensured superior adhesion between the plastic layers, as well as good performance with the urethane adhesive.
- (2) A hard coat was applied to the entire external surface, which has to be scratch-resistant, and to the inside surface of the transparent panel. Among polysiloxane hard coats that are highly scratch-resistant, the hard coat with superior weathering resistance was selected.
- (3) Using an automated flow coat process to apply the hard coat meant the hard coat film thickness could be controlled within the appropriate range, ensuring superior durability of the coating under actual operating conditions.
- (4) To ensure correct dimensions, hard coat deformation during thermal curing treatment was controlled by optimizing the injection molding parameters and jig shape. In addition, an excellent appearance was achieved by carrying out the entire hard coat process in a clean room.

The adoption of this extra window enhances driver rear visibility, and at the same time, gives the car a whole new look.

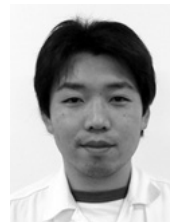
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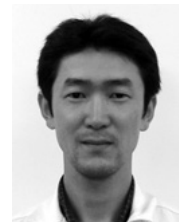
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Development of New ASIMO

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ABSTRACT

A New ASIMO humanoid robot with advanced physical capabilities that can work autonomously in a real-life environment, such as in an office, was developed in an effort to realize a robot that can coexist with and assist people in society. To attain these goals, by integrating more joints and additional sensors that help ASIMO recognize its surrounding environment, a new control system was developed. The following enhancements were achieved: Obstacle detection was enhanced and route guidance was added for greater mobility; ASIMO was equipped with the ability to run in both circular and straight directions; manipulative ability was enhanced by adding new functions that allow the robot to hand-carry light objects and to push other objects using a cart; and it was provided with the ability to better interact with people.

1. Introduction

Honda has for a long time been engaged in research on humanoid robots in its attempts to further innovative forms of mobility. Through their ability to coexist and cooperate with people, such robots are able to function usefully in society, which provides them with added value. In the course of this research, a variety of technologies were acquired, including findings from analyses of how humans walk, designs for actuator mechanisms for bipedal walking, and methods for controlling bipedal walking. These technologies were put to use in the world's first autonomous bipedal walking humanoid robot, the P2, the prototype of which was publicly announced in 1996⁽¹⁾. In November 2000, Honda introduced ASIMO, which was made the minimum size adaptable to the human living environment, with the aim of making it an actually useful robot that can function in close proximity to people⁽²⁾.

ASIMO became available for rent in 2001, and was exhibited in 30 or more countries around the world in 2002. Since then, Honda has continued to explore the interaction between people and robots in search of ways to make robots more useful. The Company has equipped them with human interactive functions that give them the capability of recognizing people's posture and actions, as well as distinguishing among individuals' faces so robots can interact more smoothly with humans⁽³⁾. Through activities like these, Honda has sought to further its dream of a humanoid robot that functions together with human beings; to illuminate new frontiers for exploration, and to develop a wellspring of technological know-how.

This article presents an overview of the new model ASIMO (Fig. 1) that was presented in December 2005, with a focus on the evolution of its functions.

* Wako Research Center



Fig. 1 Humanoid robot "New ASIMO"

2. Development Goals

Figure 2 shows a roadmap that leads to the ultimate goal for ASIMO, which is to make it "a robot that coexists and cooperates with people and performs useful functions in society." The development stage has been divided into three steps, assigned respectively to entertainment, public-use, and personal-use applications.

The new ASIMO is congruous to the second step. The goal in its development was to enhance its mobility and its ability to work autonomously in a real-world environment, such as an office. Practical jobs envisioned for ASIMO include reception and delivery. The objectives as shown below were defined in terms of movement, work, and human interaction; and technical development was conducted accordingly.

- (1) Movement Functions
 - It is able to move smoothly without running into people or walls.
 - It is able to move with agility.
- (2) Transport Functions
 - It is capable of transporting and delivering lightweight objects such as trays.
 - It is capable of transporting objects using a cart.
- (3) Human Response Functions
 - It is capable of quickly recognizing individual people and has the capacity to behave adaptively in interactive situations.
 - It is capable of walking hand-in-hand with a person.

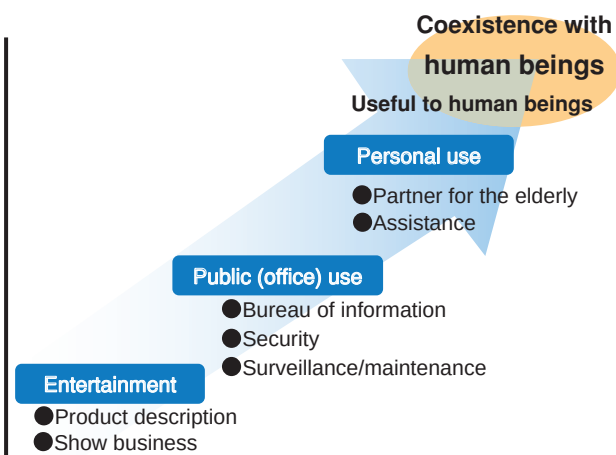


Fig. 2 Development of ASIMO

Table 1 Specifications for ASIMO

Height	1300mm	(in self-standing)
Weight	54kg	
Speed	0 - 2.7km/h (Walking) 6.0km/h (Running in straight line) 5.0km/h (Running in circular pattern)	
Sensors	· Stereo camera (for human) · Slit laser sensor (for floor shape) · Ultra-sonic sensor (for obstacles) · IC tag with optical communication unit	
Power section	51.8 V (typ) / 10Ah	(Li-ion 14 cells)

3. Main Specifications

3.1. Basic Specifications

The main specifications of the new ASIMO are shown in Table 1. Its height is 1300 mm and its weight is 54 kg. It has been given the walking speed of 2.7 km/h so that it can walk at the speed used when guiding a person to a destination. The exterior design carries on the innovative-yet-familiar image of the previous ASIMO, while developing it further and allowing for the installation of various devices. The power supply system has been changed to use a lithium ion secondary battery for greater compactness. On-board recharging has also been newly adopted for this system, to enhance usability.

3.2. Structure of Joints

The new ASIMO has been given a total of 34 degrees of freedom with the addition of one axis in the waist, two axes in each arm, one axis in each hand, and one axis in the head. The turning axis in the waist was added to cancel out the spinning in the torso that occurs when the robot's movement is speeded up. In the arms, the wrists were given two more axes, for a total of seven degrees of freedom. This increased the degrees of freedom for determining the appropriate position of the hands when working, thus greatly enhancing operability. The thumbs were given an additional degree of freedom to enable them to clasp lightweight objects. In the head area, one axis was added to the neck to enable it to tilt, thus increasing expressiveness when interacting with people. The legs have six degrees of freedom, as before. The structure of the joints and the degrees of freedom are shown in Fig. 3.

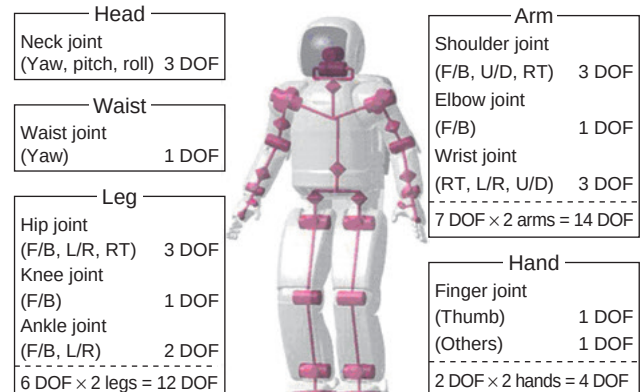


Fig. 3 Joint structure and DOF (Degree of Freedom)

3.3. Basic Control Systems

The new ASIMO systems have been revised in every respect to allow the robot to quickly recognize its surroundings and move with agility. Recognition of the surrounding environment is accomplished using a head-mounted stereo camera, as before, together with an added floor-surface sensor, ultrasonic sensor and an IC communication card combined with a photosensor. Integrated control systems were developed that are able to plan combined actions in response to greater amounts of sensor data. To enable agile movement, a higher-speed main CPU (10 times faster than the former CPU) was used; the speed of the internal communications system linking the main CPU with the sensors and motor drivers was increased (five times faster than before); and the motor driver response was stepped up (to four times faster than before). These measures significantly enhance the overall responsiveness of the system to situational changes.

4. Mobility Functions

It is necessary for a robot to move with agility in a real-world environment so that, for example, it does not bump into walls, desks or other structures, or run into people, and so that it will be able to move quickly to avoid hitting people. The new ASIMO has been given significantly enhanced navigational and obstacle-detection capabilities and new running control technology, which provides the basis for agile movement.

4.1. Route Guidance

ASIMO is guided along a route map that is set up in advance. It calculates its own position using a gyro sensor, checks it against its position according to landmarks placed on the floor surface, and makes corrections in real time to proceed on its course in the real-world environment (Fig. 4).

4.1.1. Landmark detection

As there are diverse lighting conditions in offices and other such real-world environments, it was necessary to make the robot more capable in that regard. Retro-reflective markers are employed as landmarks for this purpose, and an LED for infrared light irradiation installed at the robot's waist. Detection processing uses images captured through an infrared radiation pass filter (Fig. 5), and the shutter speed can also be adjusted automatically when images are being captured. This has enhanced the reliability of landmark detection even in locations that receive direct sunlight.

4.1.2. Self-position correction

The new ASIMO can correct its own position in real time while walking. When calculating its orientation using landmarks, it compensates for the yaw component that occurs during walking. This was done to increase the accuracy of its self-orientation. As it

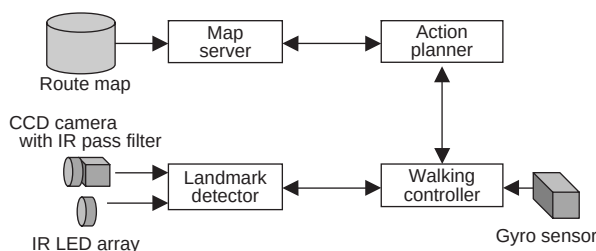


Fig. 4 Block diagram of navigation

became possible to correct its landing position while walking, the robot became able to move without stopping along the way.

4.2. Obstacle Detection

As shown in Fig. 6, the new ASIMO detects obstacles using three different sensors with different detection ranges. These are (1) the floor-surface sensor, (2) the ultrasonic sensor, and (3) the head-mounted stereo camera. Obstacle information output by the various sensors is used by the action planner to calculate obstacle positions relative to ASIMO. The robot has gained the ability to initiate deceleration or stopping behavior in response to distance, light and other such changes in the environment.

4.2.1. Obstacle detection by floor-surface sensor

The new ASIMO has been equipped with a floor-surface sensor so that it can detect objects that are on the floor in its path. It radiates several narrow beams of infrared light from its waist area onto the floor, and takes pictures of it using two CCD cameras also installed in the waist area. The images captured by the different cameras are measured by the light-section method; and for each measured point, the estimated height above the floor is calculated. Measurement points that are more than a certain height above the floor are considered to indicate the presence of an object on that part of the floor. These results are used to calculate the positions and shapes of objects within measurement range, which are then output as obstacle information. The robot is capable of automatically reducing its walking speed when the floor-surface sensor is unusable because it is receiving direct sunlight. At those times, it can use other sensors to decide whether or not to stop.

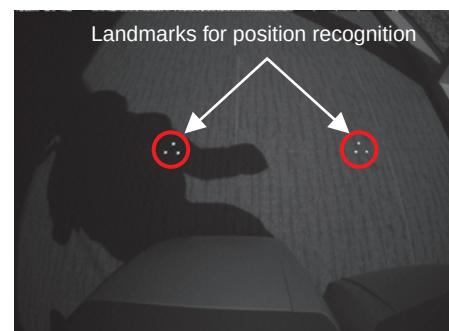


Fig. 5 Image of Landmark detection processing

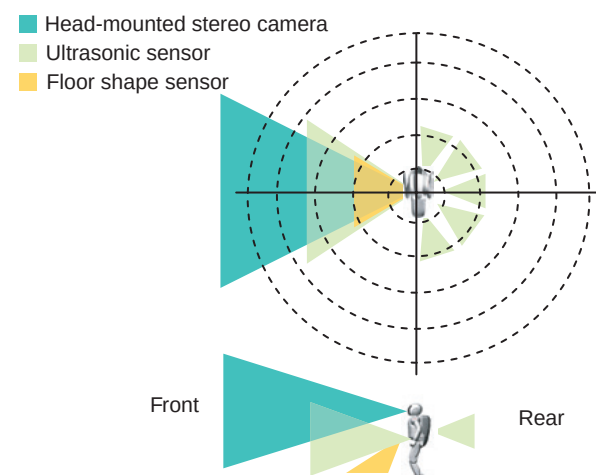


Fig. 6 Obstacle detection area

4.2.2. Obstacle detection by ultrasonic sensor

The new ASIMO is equipped with a total of six ultrasonic sensor units in the waist area, one aimed to the front and five to the rear. These detect transparent glass windows that cannot be detected by the floor-surface sensor or head-mounted stereo camera, as well as walls and other objects that are outside the range of the cameras.

4.3. Running Control Technology

The objective for the new ASIMO has been to achieve smooth, stable running as a basis for agile movement. Reduction of foot landing shock and alleviation of spinning in the torso or slipping of the feet were issues in this regard. Those issues were resolved, however, and the robot has achieved a speed of 6 km/h when running in a straight line and 5 km/h when running in a circular pattern (with a turning radius of 2.5 m). Figures 7(a) and (b) show the robot's running posture.

4.3.1. Reduction of foot landing shock

A stronger foot landing shock occurs when running than when walking. This can be effectively reduced by enhancing the responsiveness of compliance control. In this case, compliance control means that when force sensors (load sensors) attached to the foot soles detect the foot landing shock, the control system responds by adjusting the target angle for all of the motors in the legs to pull in the entire leg. This made it possible to limit the peak

foot landing shock to 2 G when running, as shown in Fig. 8, at a speed of 6 km/h (flight duration 0.08 sec, length of stride 525 mm) (Fig. 9). These values may be said to be at levels on a par with human beings. Future areas of enhancement include increasing absorptive capacity during movement over an uneven floor.

4.3.2. Generation of target motion to avoid spinning or slipping

The movement of a robot causes acceleration and deceleration in all parts of its body, which generates inertial forces. The robot is also subject to the force of gravity. The cumulative effect of all these forces is termed the total inertial force. The opposing force from the floor to the soles of the feet is called the ground reaction force. The magnitudes and lines of action of the total inertial force and the ground reaction force are identical.

The method previously used to balance the total inertial force and the ground reaction force, as shown in Fig. 10, was to have the robot hold its upper body perpendicular to the floor while thrusting the feet against the floor, to accelerate the upper body forward parallel to the floor. If this method were also applied to running, then the ground load would diminish when the robot was in flight and just before and after. This would cause the force of the feet thrusting against the floor to exceed the limit of the static frictional force, causing the robot to slip.

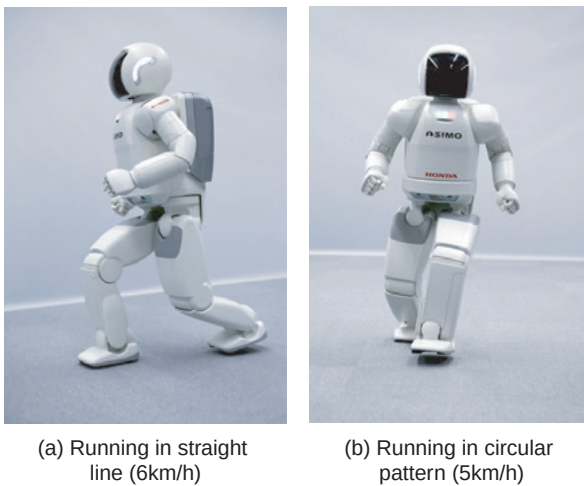


Fig. 7 Running posture



Fig. 9 Flight duration while running (High speed camera)

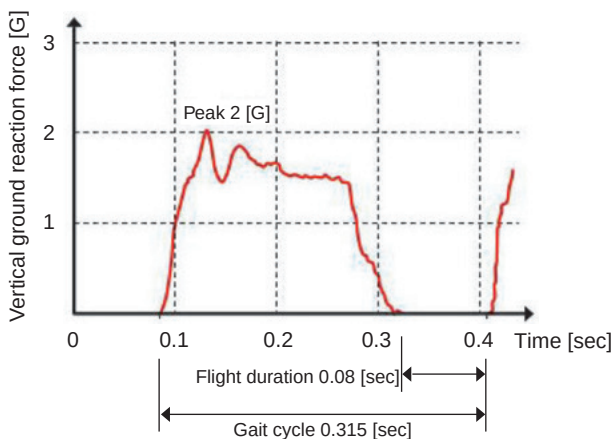


Fig. 8 Vertical ground reaction force at 6 km/h

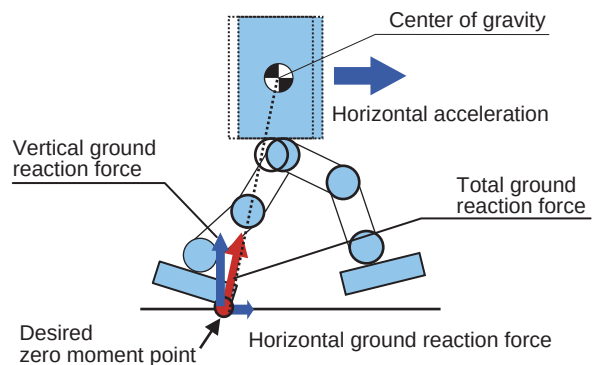


Fig. 10 Balancing motion via upper body horizontal acceleration

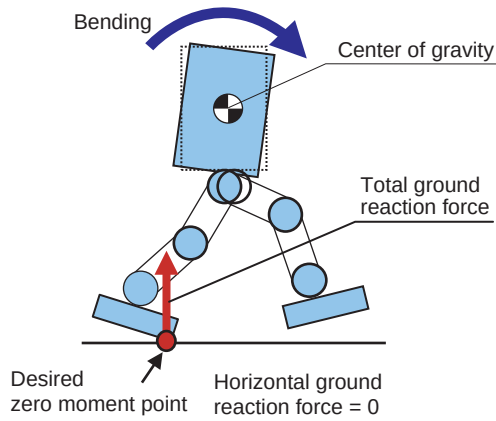


Fig. 11 Balancing motion via upper body rotation

When people sense that they might fall over, such as when standing or walking in a skating rink, they actively lean their upper bodies in the direction they are falling, as seen in Fig. 11. They use this reaction to try to balance themselves. This action does not depend on horizontal ground reaction force. When these two balancing actions are combined properly in accordance with the limit of the static frictional force, they can generate a target motion that prevents slipping.

Swinging the legs also generates spinning forces (the moment of force that tries to rotate around the vertical axis). The robot twists its upper body to cancel out these forces.

Both feed-forward and feedback controls use these actions, which prevents spinning and slipping while achieving highly stable walking (at a speed of 2.7 km/h) and running movement.

5. Work Functions

The new ASIMO has become capable of performing various services autonomously. These include transporting documents, beverages on a tray and other such lightweight objects (lightweight object transportation function), and guiding a cart along a route with some flexibility (cart guidance function).

5.1. Transporting Lightweight Objects

The new ASIMO uses force sensors to detect the load exerted on its hands when gripping a lightweight object. This has made it possible for the robot to independently judge the timing for opening and closing its hands. In addition to the detection of external forces using the force sensors, joint angle error information of the hands is also used to determine the grasping status of objects that it is transporting. Ranging information from its head-mounted stereo camera further detects whether it has documents or a tray; and it has been provided with the ability to judge the timing for lowering its arms after delivering something. The robot integrates this kind of visual and kinesthetic information with numerous other items of information to achieve a natural delivery style that adapts to the movements of the recipient.

One issue was how to limit the vibration in the robot's fingers caused by the foot landing shock when it is transporting a tray. This was solved by adopting vibration controls that feed back the acceleration component exerted on the fingers in a speed control loop, while also limiting vibration in the waist when the robot is walking while grasping something. As a result, the foot landing

shock at the tray position was reduced by 20% and the peak angular velocity on the pitch circumference was reduced by 40%, thus realizing stable transportation of lightweight objects.

5.2. Cart Guidance Function

Research on the movement of robots pushing carts or other wheeled objects had been conducted in the past⁽⁴⁾, but it was limited to linear movement. The new ASIMO was envisioned as being able to guide a cart that it pushes through offices or other such passages in a real-world environment. The aim, in other words, was to realize the capability for guidance along a path; for guidance on a route that involves right angles, such as turning sharp corners; and for dealing promptly with changes in paths and surface conditions of roadways, changes in transportation volume and other such factors, and adjusting speed accordingly. The following three technologies were developed for this purpose:

- (1) Technology for predicting the future movement of a cart.
- (2) Technology for determining movement of the robot's whole body in response to the predicted movement of a cart, taking into account the range where the robot cannot step, the full stretch of its arms and so on.
- (3) Technology for guidance speed control to avoid the range noted above in response to changes in the floor surface or changes in weight.

These technologies provide a foundation for making continuous revisions to action plans in real time. This has made it possible for the robot to guide a cart autonomously and without imposed limitations, without exceeding its action boundaries, while adapting to changes in the external environment.

6. Human Response Functions

The new ASIMO has various added capabilities. It can quickly recognize an individual person, and is capable of adapting its behavior to others in a real-time manner (human detection). It is also capable of actions that involve holding people's hands (shaking hands and walking hand-in-hand).

6.1. Human Detection

Images from the pair of CCD cameras installed in the robot's head were formerly processed to isolate objects resembling the human form. The human object information (position, ID, speed) would then be provided as output. The system developed for the new ASIMO functions with the addition of an IC communication card. This combination is capable of detecting the positions of people in all directions around the robot, while also acquiring information about those people.

The IC communication card system uses existing radio communications in conjunction with optical communications that makes use of the directional characteristics of light and separates all the directions into eight. As shown in Fig. 12, the system is now capable of detecting the direction to any person within a 4-m radius of the robot.

ASIMO has further been given the capacity to behave adaptively in interactive situations, such as when it is acting as a receptionist, by applying these various functions (Fig. 13). When functioning as a guide, for example, the robot adjusts its actions according to the distance from and direction of the other party. It can turn in that person's direction and speak facing the person directly, with its head angled to meet the person's line of vision and so on.

6.2. Shaking Hands and Walking Hand-in-Hand with a Person

Force sensors have been installed in the robot's wrists and used with compliance control, to extrapolate the direction and speed of travel desired by the person with whom it is interacting. Thus it can respond flexibly, interact adaptively such as shaking hands and walking while holding a person's hand (Fig. 14).

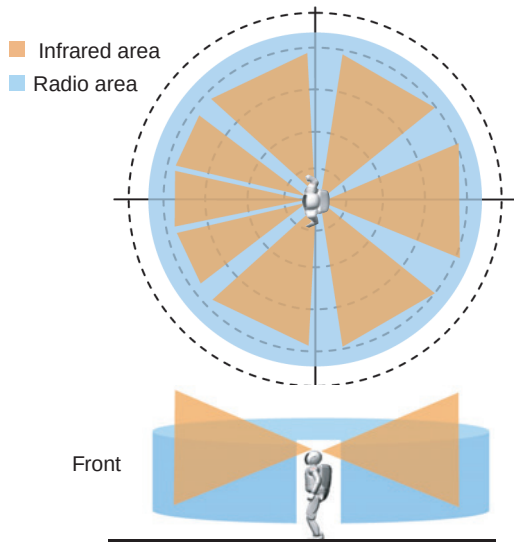
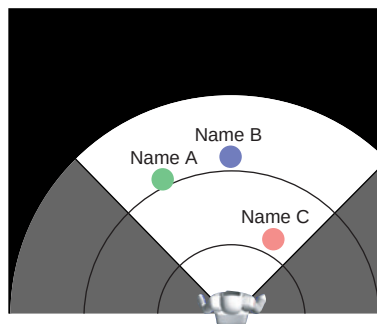


Fig. 12 Detection area of IC communication card



(a) Image processing



(b) Integration of IC communication card & Image processing

Fig. 13 Result of human detection



Fig. 14 Walking hand-in-hand

7. Conclusion

The development of the new ASIMO aimed at achieving a robot that could work autonomously in an office or other such real-world environment, and exhibit greater mobility. All this was intended as a step toward the goal of “a robot that coexists and cooperates with people and performs useful functions in society.” The number of joint axes was increased; new control systems equipped with sensors for perceiving the surrounding environment were installed; and new running control technology was applied, enabling it to move with greater agility and to carry out real-time interactions with humans. As a result, the robot has evolved into an autonomous receptionist, a guide in an office, and a practical worker that can use transportation tools.

Our intention at Honda is to further refine the technologies developed for the present ASIMO so that its range of practical applications can be expanded in the future.

The generous assistance provided in the course of the present research project by the people involved both inside and outside the Company is hereby acknowledged with gratitude.

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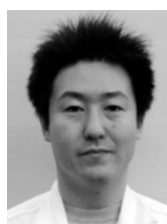
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Development of New Honda FCX with Next-generation Fuel Cell Stack

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ABSTRACT

In December 2002, Honda produced the first practical fuel cell vehicle, delivering models to the Japanese government and to the City of Los Angeles in the United States. To encourage greater public acceptance of fuel cell vehicles, Honda announced a new stack with enhanced performance in low temperatures in October 2003 and installed it in the New Honda FCX. This is the first fuel cell vehicle that can be used below freezing temperature. Vehicles were delivered to New York state in November 2004 and to Hokkaido in January 2005. This paper introduces Honda's fuel cell vehicle, the first to be used in both temperate and cold regions in major cities worldwide.

1. Introduction

Honda developed the Honda FCX fuel cell vehicle (FCV) as an ultra environment-adaptable vehicle and in December 2002 became the first to supply a fuel cell vehicle to the Japanese government and the City of Los Angeles, in the United States. Since then this vehicle has demonstrated that the fuel cell vehicle is an achievable technology, and a large part of society is now aware of this technology^{(1), (2)}.

The next step is to develop the technical elements that will help society to fully embrace hydrogen vehicles. One of the most important issues here has been driving performance at temperatures below freezing, a weak point for earlier fuel cells. In October 2003, Honda announced a stack capable of starting at below freezing temperatures. In February 2004, Honda conducted vehicle starting and driving tests at below freezing temperatures in Hokkaido, and

announced the results. This report introduces the new Honda FCX technology.

2. Development Details

The Development of the new Honda FCX began by using the frame of the 2002 model. The principle technical issue was ensuring the ability of the vehicle to start and drive at below freezing temperatures.

Other goals included enhancing driving performance, including power, fuel efficiency, and range. The vehicle's overall weight and dimension specifications were unchanged from the construction shown in Table 1. However, the fuel cell system was redesigned, with the goals of ensuring that the vehicle could start and drive at -20°C , achieving vehicle energy efficiency of at least 50%,

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Table 1 Major specifications

Model	2005 Model FCX	2003 Model FCX
Length/Width/Height	4165 × 1760 × 1645mm	4165 × 1760 × 1645mm
Wheel base	2530mm	2530mm
Vehicle weight	1670kg	1680kg
Number of passengers	4	4
Maximum speed	150 km/h	150km/h
Range (in LA4 mode)	430 km	355 km
Maximum motor output	80kW [109PS]	60kW [82PS]
Maximum motor torque	272 N·m [27.7kg·m]	272 N·m [27.7kg·m]
Fuel cell stack	PEMFC	PEMFC
Power assist	Ultra-capacitor 9.2F	Ultra-capacitor 8F
Hydrogen supply system	High-pressure hydrogen tank	High-pressure hydrogen tank
Hydrogen pressure	34.4 MPa	34.4 MPa
Hydrogen storage volume	156.6L	156.6L

increasing acceleration by 10%, and providing a vehicle range of at least 400 km. Development proceeded with the goal of winning the first type certification as an FCV in Japan to prove this technology for full-scale mass production.

3. Summary of Vehicle and Its Technology

3.1. Appearance and Basic Specifications

The exterior and interior design of the 2005 FCX are the same as in the previous model, but the body was covered with a bright and transparent-seeming blue base tone to project an image of newness and progressiveness, as seen in Fig. 1. The mascot logo, which is new to this vehicle, projects the image of electricity being produced from hydrogen and oxygen, thus contributing to the global environment. The vehicle is newly equipped with navigation and traction control systems, as well as a standard ABS brake system. The vehicle also has front McPherson struts, rear five-link multi-link suspension and an electronically controlled power steering system.



Fig. 1 Exterior design and mascot mark

3.2. Major Vehicle Equipment

3.2.1. Navigation system

The Honda FCX is equipped with the first navigation system to display hydrogen fueling stations. Such facilities are shown just as gasoline stations are, but with a unique “H2” emblem, and drivers can search for routes and get verbal driving instructions as well (Fig. 2). The NAVI system release coincided with the announcement of the California Hydrogen Highway network, and it earned praise for furthering the arrival of a hydrogen-based society.

The vehicle also has an HVAC system that performs well at below freezing temperature. This system warms the refrigerant directly using an electric heater, for quick and comfortable heating.

3.2.2. Traction control system

To ensure that the vehicle can drive safely on snow, the New Honda FCX is equipped with a traction control system, which is part of its package as an FCV capable of driving at below freezing temperatures. Maximum torque in an electric motor occurs directly after the motor begins to turn, and when taking off on slippery road surfaces, the tires are liable to spin. When driving on snow, the vehicle's new traction control system interprets the driver's intent precisely to ensure smooth takeoff.



Fig. 2 Navigation system

The drive system of the Honda FCX consists of a motor and deceleration gears. Without a clutch, gearshift, or reverse gear, the structure has fewer elements and is much simpler than that of a gasoline vehicle. The characteristics of the drive mechanism, which include minimal transmission loss and a motor that provides higher torque at low rotational speeds than gasoline engines, enable this vehicle to deliver torque to the tires quickly at takeoff and provide extremely agile power performance. By taking advantage of these excellent basic characteristics, the new traction control system can directly control motor torque without using the brakes in situations when the tires spin right after takeoff. As a result, the tires recover their grip quickly after starting to slip and show excellent responsiveness to subsequent demands for speed (Fig. 3).

The quiet, smooth acceleration of the Honda FCX on snow makes the driver keenly aware of its new power performance—quite unlike that of a vehicle with a conventional engine.

3.3. Power Plant System

The development objective of the power plant system was to further enhance the acceleration performance to achieve a sporty feeling, and achieve -20°C startup capability, while also improving fuel economy. Figure 4 shows the layout of the new FCX's power plant system.

3.3.1. Fuel cell stack

The new FCX fuel cell stack uses strong metal separators that are only 50% of the thickness of conventional carbon separators. Moreover, because of the use of integrated molded seals, the cell units are thinner. By using the springiness of stamped metal separators, construction has been simplified by merely encasing the stacked cells with panels, thereby reducing the number of

parts in the entire stack to nearly half. In this way, the fuel cell stack has been made lighter and more compact, and the output density has been increased to more than two times that of the conventional carbon separators stack (Fig. 5)⁽³⁾⁻⁽⁶⁾. The amount of ion exchange substrate that governs ion conductivity was increased significantly compared with that of the conventional fluorine electrolyte membrane. The result was that membrane resistance was halved, excellent proton conductivity was attained even below the freezing point, and a fuel cell stack that could start generating power at -20°C was achieved. Due to the newly developed aromatic electrolyte membrane, which does not soften or deform at high temperatures due to the high strength and high thermal resistance of the main chains, the fuel cell stack can be used at temperatures as high as 95°C .

3.3.2. Fuel cell system

To enhance the space efficiency of the fuel cell system box, the fuel cell stack was divided into two units. By utilizing features of the smaller units, the humidifier modules were integrated (Fig. 6). By efficiently laying out the entire system after integrating/discarding auxiliaries, a compact fuel cell system box was obtained, leading to better energy efficiency by reducing air and fuel pressure losses.

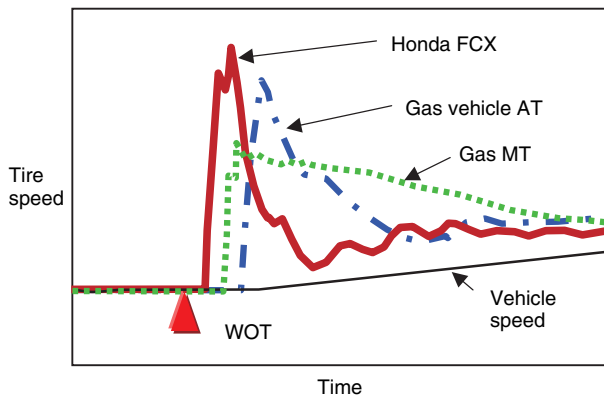


Fig. 3 Response of tire rotation on WOT

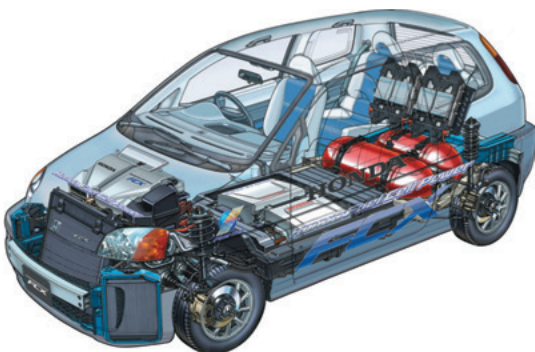


Fig. 4 Perspective view of Honda FCX

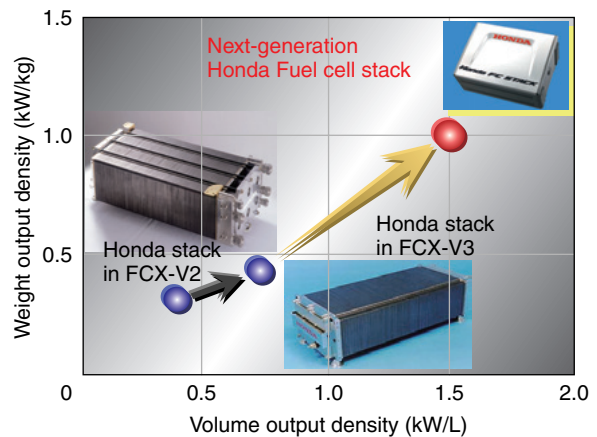


Fig. 5 Evolution of Honda stacks

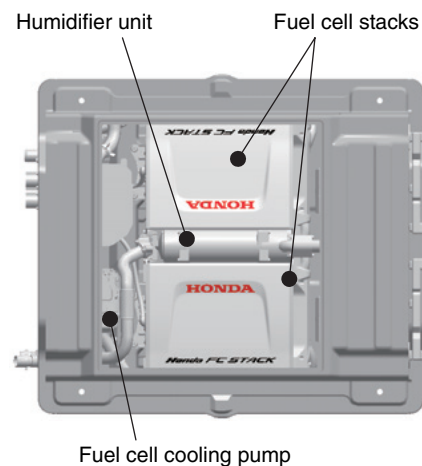


Fig. 6 Fuel cell system box configuration

3.3.3. Drive motor

In addition to raising the fuel cell's output as explained above, Honda also adjusted the output of the drive motor. Eddy current generation was controlled by magnetic partitioning to counter the increase in rotor magnetic flux fluctuation resulting from the combined use with a reluctance system. Furthermore, the use of heat-tolerant magnets and a magnetic circuit configuration designed for high output helped to achieve highly effective demagnetization suppression. As a result, maximum output was increased from the previous 60 kW to 80 kW while maintaining high efficiency over a wide range, meaning that driving potential was increased (Table 2, Fig. 7).

3.3.4. Ultra-capacitor

An ultra-capacitor is used as the auxiliary power source to provide a powerful drive. Conventional high-performance activated charcoal electrodes were used previously. The electrode wrapped-element construction has been reexamined, and the number of turns of the electrode was increased without increasing the size of the cell. This enabled a significant increase in the stored energy capacity. As a result, energy and power were enhanced by more than 10% compared to those of the previous capacitor (Fig. 8). The increased performance of the capacitor allows for increased fuel economy through more efficient energy management.

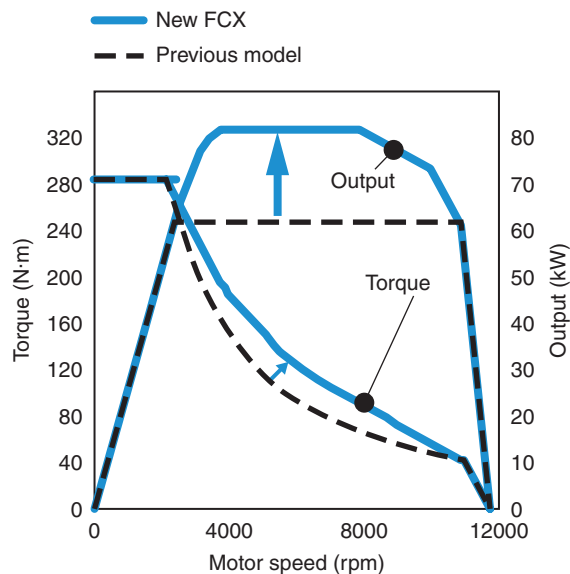


Fig. 7 Output torque characteristic

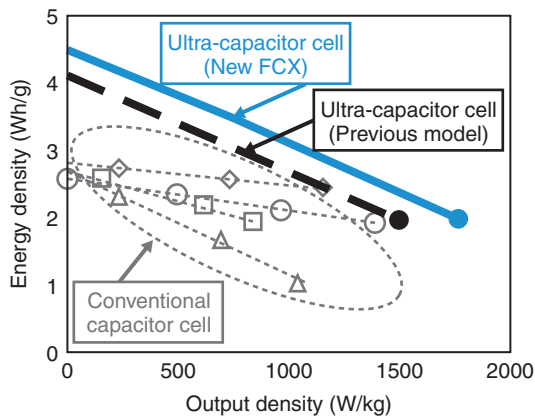


Fig. 8 Capacitor charging and discharge performance (cell unit)

Table 2 Comparison of drive motor performance

	New FCX	Previous model	EV-Plus
Max. output	80kW	60kW	49kW
Max. torque	272N·m	272N·m	275N·m
Max. rpms	11000rpm	11000rpm	8750rpm
Max. efficiency	97%	97%	96%
Avg. efficiency (LA-4 mode)	93% or greater	93% or greater	90%

4. Achieved Performance

4.1. Efficiency

As shown in Fig. 9, drive energy efficiency was enhanced by 22% compared to that of the previous model by increasing the power generating efficiency of the newly developed Honda FC Stack and by increasing the efficiency of the air supply system to the stack and the hydrogen circulation system. Better power efficiency and lower fuel consumption were realized by combining conventional high-efficiency energy management with other techniques, such as regeneration of the deceleration energy in the ultra-capacitor, and using a fuel cell auto idling stop system. Tank to Wheel energy efficiency of 55% was achieved, which is about three times higher than that of a gasoline-engine vehicle and about two times that of a hybrid vehicle.

4.2. Fuel Economy and Driving Range

Because the energy from 1 kg of hydrogen and 1 gallon of gasoline are approximately equivalent, the fuel consumption in CITY mode of equal-sized ICEV, HEV and CNG vehicles can be compared. Figure 10 shows the results. The fuel consumption performance of the FCX was twice that of the gasoline-engine and CNG vehicles, and 1.3 times that of the hybrid vehicle. The fuel

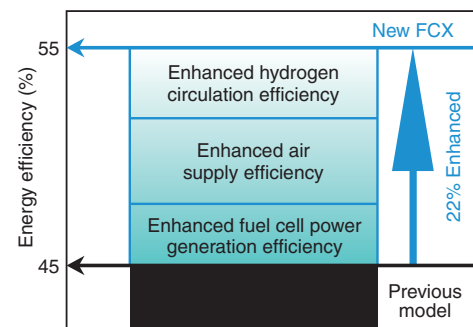


Fig. 9 Energy efficiency, LA-4 Mode

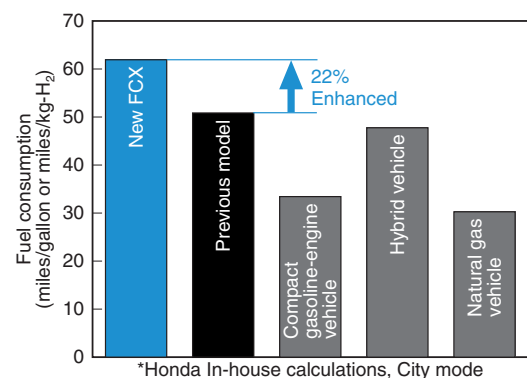


Fig. 10 Fuel consumption

consumption performance was enhanced by 22% compared to that of the previous model, while the driving range increased from 355 km to 430 km.

4.3. Acceleration

Figures 11 and 12 illustrate the output responsiveness at takeoff and the vehicle's takeoff acceleration characteristics. The expanded capacity provided by the ultra-capacitor and the increased output of the fuel cells allow the vehicle to take advantage of a drive motor that delivers maximum torque from low rotational speeds. This results in a shorter response time during acceleration as compared to a hybrid electric vehicle (HEV). Further, there is no transmission and no gear changing, so acceleration is exceptionally smooth. As a result, takeoff acceleration time has been reduced 13% and passing acceleration time 20% compared to the previous model (Fig. 13).

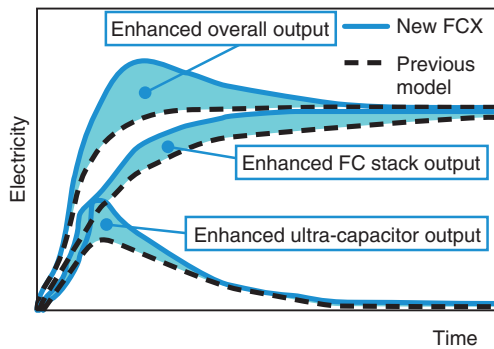


Fig. 11 Electricity at takeoff

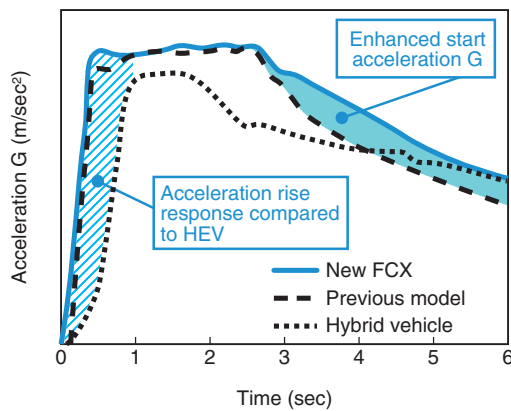


Fig. 12 Takeoff acceleration

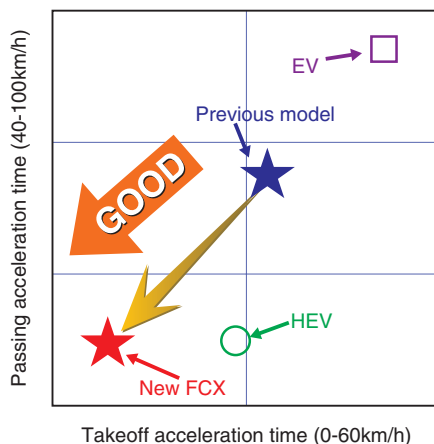


Fig. 13 Acceleration performance

4.4. Noise Countermeasures

The new model's air pump rotor has a solid construction, rather than the hollow construction of the previous FCX. This change suppresses air column resonance and reduces intake noise. In the interior, the noise of the air pump is reduced by 10 dB over the previous model. In addition, the part of the traction motor stator where the coil is wound is now made of molded resin, which increases the rigidity of the stator overall and cuts motor resonance vibration. In the interior, the noise of the motor is reduced by 6 dB over the previous model (Fig. 14).

4.5. Performance at below Freezing Temperatures

With conventional fuel cells, power cannot be generated below 0 °C. There are three basic reasons for this:

- The hydrogen ion conductivity of ion exchange membranes declines as the temperature decreases below 0 °C.
- The contact resistance of carbon separators increases significantly as the temperature decreases below 0 °C.
- Carbon separators have a low thermal conductivity and take a long time to warm up.

Below freezing, the next generation HONDA FC STACK has excellent ion conductivity and 25% lower intracellular contact resistance than previous stacks. The superior warming characteristics, due to the reduced heat capacity of the fuel cell and the fuel cell system, reduced the stack warm-up time by 80%. As well, with optimal water management of the generated water, start-up at -20 °C has become possible. The use of the FCX in low temperature environments, such as on snowy roads, is comparable to that of gasoline-engine vehicles, and the low temperature operation goal has been achieved (Fig. 15).

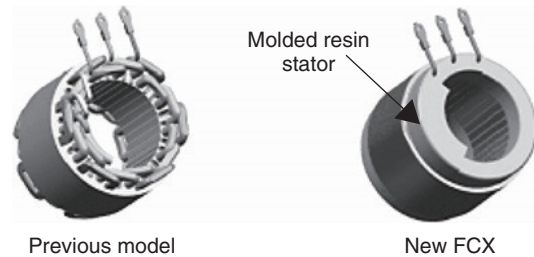


Fig. 14 Motor structure



Fig. 15 Test driving in snowy area

5. Certification

5.1. United States

In July 2004, the new model received sales authorization (as did the previous model before it) from the U.S. Environmental Protection Agency (EPA) and the California Air Resources Board (CARB). It has been recognized as the first fuel cell vehicle usable in cold areas. The Honda FCX has been registered as a zero-emissions vehicle (ZEV), meaning that it has no pollutant gas emissions at all (Fig. 16).

5.2. Japan

On March 31, 2005, the Ministry of Land, Infrastructure and Transport established technical guidelines establishing the world's first fuel cell vehicle safety standards, making it possible to apply for type certification. With officials from the Ministry of Land, Infrastructure and Transport observing, Honda conducted the nation's first fuel cell vehicle crash tests to ensure that the vehicle met crash safety standards. Figure 17 shows the tests in progress. These crash tests demonstrated that the Honda FCX fulfills the technical standards, and in June 2005, it received the first type certification for a fuel cell vehicle in Japan.



Fig. 16 ZEV label



Fig. 17 Crash tests

6. Conclusion

With the FCX, Honda has achieved extremely environmentally adaptable automotive technology. The result of these technical developments is that, for the first time, such a fuel cell vehicle is being sold in areas where temperatures fall below freezing in the winter. In November 2004, a new vehicle was delivered to the government of New York State, in the northeastern part of North America. Subsequently, a vehicle was delivered to the government of Hokkaido in January 2005. The FCX has proven that FCVs can be widely used in the major cities of the world.

However, there are still many issues with fuel cell vehicles and problems that need to be overcome, so R&D remains ongoing. Another important issue is the creation of an environmental society, and part of this issue is supplying the hydrogen. Yakushima Island in Kagoshima Prefecture has been the site of a proving test for the "recirculating hydrogen energy society" since April 2004. Honda is taking part in this project, which seeks to run fuel cell vehicles on hydrogen produced from hydroelectric power (Fig. 18).



Fig. 18 Test driving on Yakushima Island

Honda has become the first company to market a fuel cell vehicle, and will continue to innovate as a leader in environmental technology.

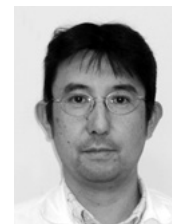
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Development of Snow Thrower Control System with Engine Provided of Electronic Governor System

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Nobuo YAMAZAKI*

ABSTRACT

An electronic control system has been developed to enhance snow removing performance, operability and fuel efficiency of medium-sized snow throwers mounting an engine provided with an electric governor. For efficient and perfect snow removal, proper control of working load and stable engine speed are required. The load control in this development uses engine throttle opening degree as a parameter of engine load detection, which enables real-time vehicle speed control which is required for load control. In a simulation of actual snow removal work, fuel economy of new system was enhanced 22% in the same area compared with that of existing hybrid snow thrower. In addition, mode control responding to usage is provided, which controls engine speed and vehicle speed automatically which was controlled manually in existing snow thrower. It has realized a user-friendly snow thrower for beginners.

1. Introduction

For those living in snowy regions, winter snow removal work is a serious issue. To ease the burden of this hard work, snow throwers have been developed for residential use.

Walk-behind snow throwers are composed of clearing parts that clear the snow and traveling parts that push the snow removal operational parts against the snow. The main components include the engine, which is the power source; an auger, which breaks up the snow; a blower, which throws the chopped snow; a chute, which guides the thrown snow; and a crawler, which moves the unit. Figure 1 shows the structure of the walk-behind snow thrower. During snow removal, the operator must carry out many operations – controlling the auger height according to the amount of snow, adjusting the chute in the direction and distance the snow is to be thrown, and controlling the traveling speed – which means that the operator must become familiar with the unit's operation.

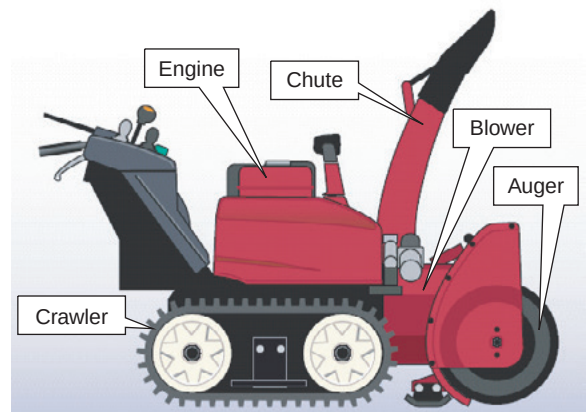


Fig. 1 Structure of walk behind snow thrower

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To make snow removal faster, simpler and more pleasant, Honda has developed the hybrid snow thrower HS1390i in 2000⁽¹⁾, adopting a series hybrid method, in which engine output is used to drive the snow removal parts and generate electricity, while the motor drives the traveling parts only, using the generated electrical power. The use of electronic controls achieves smooth start-offs and stops and smooth turning. At the same time, the engine speed that is carrying out the operation is detected, and load control automatically varies the traveling speed to the work load. When the engine speed drops in response to the work load, control is such that the traveling speed slows, reducing the amount of snow that is cleared, thereby reducing the load on the engine and minimizing the drop in engine speed.

This paper reports on the development of a control system introduced in the snow thrower HSM1590i, in which the iGX440 engine equipped with an electronic control function⁽²⁾ has been installed, and which incorporates new load and mode controls utilizing the features of an electronic governor⁽³⁾.

2. Snow Thrower Load Control

When a large amount of snow is stuck in the snow clearing parts, the engine load increases, and there are cases in which the engine speed does not just drop, but the engine actually stalls. In addition, if the snow is not completely cleared and the snow removal parts ride up onto the snow so that the body is floating, the snow removal parts can no longer be pushed against the snow, and the operator must repeat the operation after pulling the machine back. Furthermore, there are cases in which the snow removal surface is not completely cleared of snow and the snow removal procedure must be repeated. In these situations, work efficiency is poor, and it is necessary to control the amount of snow being cleared to enhance efficiency. By slowing the speed of the crawler, the amount of snow per unit time that goes into the snow removal parts can be reduced. That is to say, by controlling traveling speed in response to engine load, it is possible to operate efficiently.

2.1. Load Adjustments by the Operator

In the case of snow throwers that change engine power in the traveling parts using gears or Hydro Static Transmission (HST), traveling speed drops proportionally to engine speed. This relationship is shown in Fig. 2. However, in cases where there is an insufficient reduction in traveling speed against engine speed drop, together with an engine speed drop feel and snow removal parts riding up, the operator oneself will need to slow the traveling speed.

2.2. Load Control by Engine Speed

With the series hybrid snow thrower HS1390i, it is possible to freely control the traveling speed with the motor, regardless of the engine speed. That is to say, if it is possible to detect the engine load, the traveling speed can be automatically adjusted.

The engine installed in the working unit has a mechanism that automatically adjusts the carburetor throttle opening, maintaining a fixed engine speed. Many of these types of systems adopt a mechanical feedback control that applies a centrifugal governor mechanism. The GX390 engine fitted in the HS1390i uses the same mechanism. However, at the same time, an engine speed difference mechanically occurs in the GX390, according to the load. Figure 3 shows the engine's Wide Open Throttle (WOT) output characteristics and centrifugal governor's partial load output characteristics.

With the HS1390i, the centrifugal governor's characteristics are utilized, and engine speed changes are used as a means of judging engine load. The ratio of the reduction in traveling speed against engine speed is shown in Fig. 4. Using this control mechanism, it is possible to reduce engine stalls due to engine speed drops, and to reduce the occurrence of clearing parts riding up. However, when 9.4 kW at the full throttle output power of 3200 rpm is needed, centrifugal governor adjustments are carried out so that the operation is 3600 rpm at no load. When operating at partial load, the engine speed is faster than 3200 rpm, but if 3200 rpm is sufficient for the operation, there is wasted engine rotation, which leads to decreased fuel economy and more noise. Moreover, the blower motor speed changes according to engine speed fluctuations, which affects the distance snow is thrown.

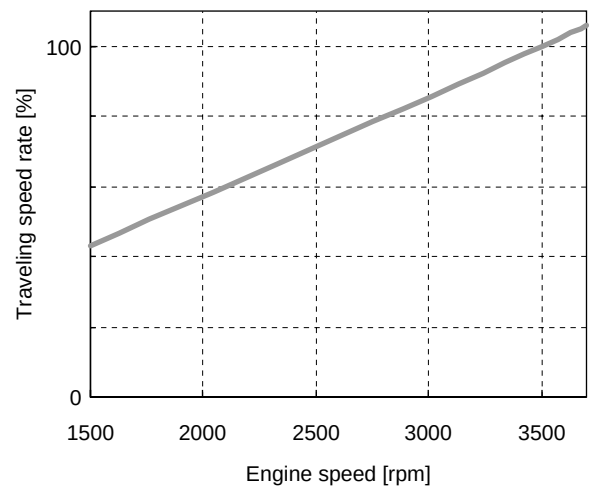


Fig. 2 Relationship between engine speed and traveling speed rate

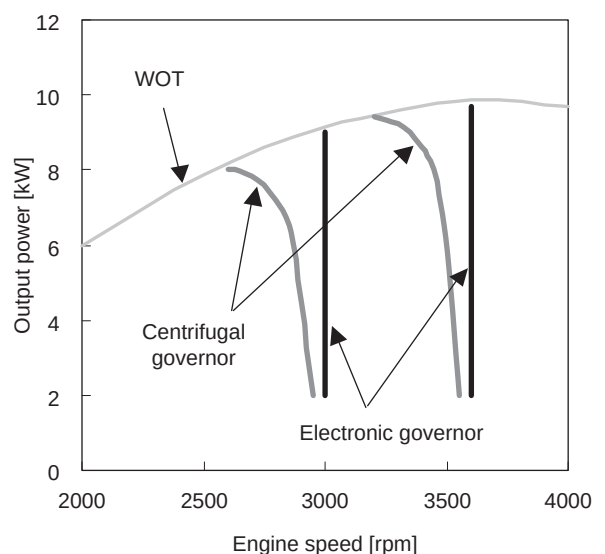


Fig. 3 GX390 engine performance

2.3. Load Control according to Engine Throttle Opening

Engines adopting the electronic governor system can operate at a fixed engine speed against target engine speed, regardless of load increases or decreases.

It is possible to obtain throttle opening information from the engine, when the engine installed has adopted an electronic governor controller system. According to this information, the engine load rate is controlled, and it is possible to operate at the set engine speed. This throttle opening is controlled by the electronic governor controller, and the throttle valve is driven by the stepping motor.

Figure 5 shows the relationship between the amount of snow cleared per unit time, as determined in practical use tests, and the stepping motor's step count controlling the throttle opening. When the stepping motor's step count is 102 pulses, the throttle is fully open. When the engine speed is 3600 rpm, the motor step count, in the no-load operational state where the clearing parts are only connected by the clutch, is 30 pulses. From these results, it is clear that 102 pulses is the maximum value at which engine speed changes do not occur, and 30 pulses is the state at which there is no operational load. At 3600 rpm, the motor step count with no operational load status is 30 pulses, but this value changes according to engine speed, so it is necessary to fully consider no operational load pulse counts for each engine speed.

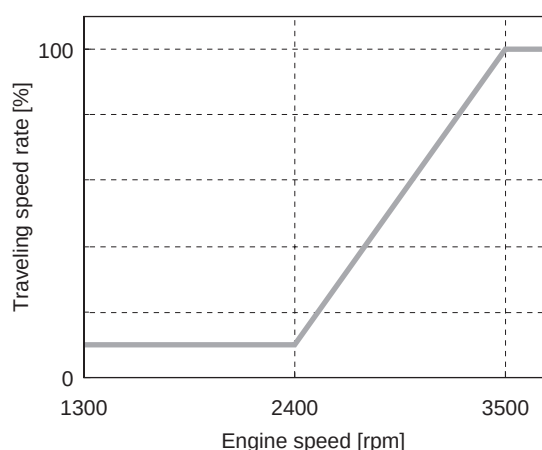


Fig. 4 Relationship between engine speed and traveling speed rate

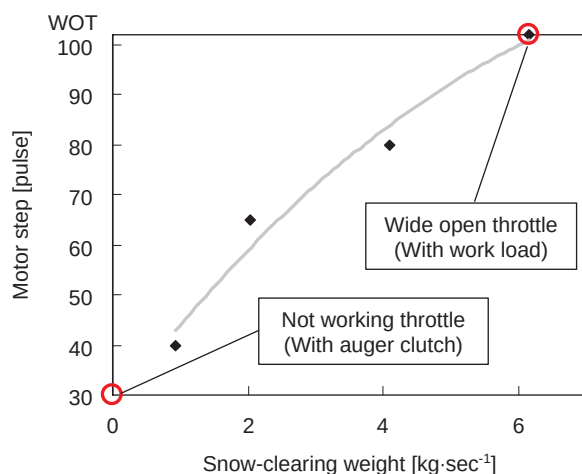


Fig. 5 Relationship between snow-clearing weight and motor step (Engine speed = 3600rpm)

Applying this to load control, step counts from 30 to 102 pulses become the range in which the throttle can follow the load. If traveling speed is controlled so that throttle opening is 30 pulses or more and 102 pulses or less, there will be no change in the engine speed. In addition, even when operational load changes occur, if the traveling speed can be varied so that the engine load, that is to say pulse count, is constant, the condition will be equivalent to rated operation. If engine load is stabilized, the engine throttle will not have to be fully opened and closed, and stable operation is possible.

In this way, by using throttle opening information, or stepping motor step count, in load control, the engine load state becomes known, traveling speed is controlled based on that load state, and load on the clearing parts is controlled before the engine speed drops enabling efficient operation.

3. Mode Control

When removing snow, the engine speed is set normally between 3000 rpm and 4000 rpm, speeds at which it is possible to obtain maximum engine power. This snow thrower is designed so that it is possible to operate at these engine speeds. When the snow thrower is not removing snow, for example when the clearing parts (auger and blower) are not operating while the unit is being moved, or when the unit is reversing during operation, little engine power is required.

For snow throwers that change speeds mechanically using gears or HST and use engine power to drive the traveling parts, even when no snow is being removed and there is little need for engine output, it is necessary to increase the engine speed to increase the traveling speed.

Snow throwers that adopt a series hybrid method have travel powered by the motor only, using generated electricity, so it is not necessary to have high engine speed to obtain fast traveling speed. Changing the set engine speed in response to the operational state, so that there are no unnecessarily high engine speeds, works to enhance fuel economy and to make the unit quieter.

With the HS1390i, fitted with the GX390 engine, by changing the governor spring load using wires, the engine target speed is set, and so to change the engine speed settings, it was necessary for the operator to carry out each operation.

With the newly developed control system, the use of the iGX440 engine fitted with an electronic control function that has adopted the electronic governor control system makes it possible to set the target engine speed, from the Electronic Control Unit (ECU) that controls the snow thrower system as a whole. Three operational modes are available as follows, adjustable to the way the snow thrower is used by the operator: Auto Mode, which automatically controls the traveling speed and engine speed to minimize operational procedures by the operator; Power Mode, which automatically selects the engine speed during operation, allowing more powerful snow removal; and Manual Mode, which allows the operator to control operations at will, with almost no automatic controls. In these modes, not only the target engine speed, but also the previously mentioned load control and other functions are switched.

3.1. Setting Engine Speed by Mode

The newly developed control system adopts a method wherein the operational state is recognized by the ECU, and the target engine speeds are adjusted automatically. The relationship between each mode and target engine speed is shown in Table 1.

Table 1 Operation models

Mode		Auto	Power	Manual
On working	Forward and auger on	Auto set 3000 [rpm]	Speed set by user 2000 - 4000 [rpm]	Speed set by user 2000 - 4000 [rpm]
	Reverse and auger on	Auto set 2000 [rpm]	Speed set by user 2000 - 4000 [rpm]	Speed set by user 2000 - 4000 [rpm]
Off working	Forward and auger off	Auto set 2000 [rpm]	Auto set 2000 [rpm]	Speed set by user 2000 - 4000 [rpm]
	Reverse and auger off	Auto set 2000 [rpm]	Auto set 2000 [rpm]	Speed set by user 2000 - 4000 [rpm]

3.2. Load Control by Mode

Load control when Auto Mode and Power Mode are selected reduces traveling speed based on throttle opening. However, when Manual Mode is selected, deceleration occurs only in proportion to the engine speeds shown in Fig. 2, so there is no feeling of the ECU arbitrarily reducing speed with respect to the traveling speed set by the operator. Figure 6 shows the traveling speed deceleration rate against the stepping motor step count of the engine throttle valve when Auto Mode is selected and when Power Mode is selected.

With the iGX440 engine, when the stepping motor's step count is 92 steps, the throttle is fully open.

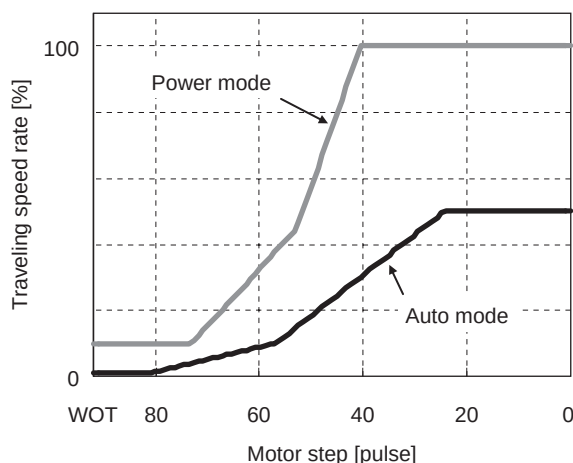


Fig. 6 Relationship between motor step and traveling speed rate

3.3. Auto Mode

In Auto Mode, by reducing traveling speed, clearing part ride-up is reduced and the engine speed, too, is kept at 3000 rpm, as shown in Table 1. The settings are such that quiet snow clearing is possible without traveling speed or engine speed adjustments.

Figure 7 shows data on engine speed, motor step count and traveling speed when clearing snow in Auto Mode. Engine speed is controlled at a fixed 3000 rpm by automatically reducing traveling speed as the motor step count, which expresses the engine load state, rises.

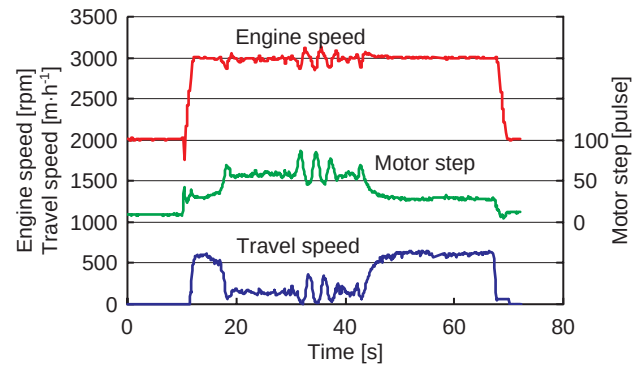


Fig. 7 Clearing snow in auto mode

3.4. Power Mode

In Power Mode, traveling speed is automatically adjusted to maintain engine output near its maximum. The operator can select engine speed at will, such as 4000 rpm, providing a maximum snow throwing distance; 3600 rpm, providing maximum engine output; or 2000 rpm, when quiet operation is desired.

Figure 8 shows data on engine speed, motor step count and traveling speed when clearing snow at an engine speed of 4000 rpm in Power Mode. Engine speed is controlled at a fixed 4000 rpm by automatically reducing traveling speed as the motor step count, which expresses the engine load state, rises.

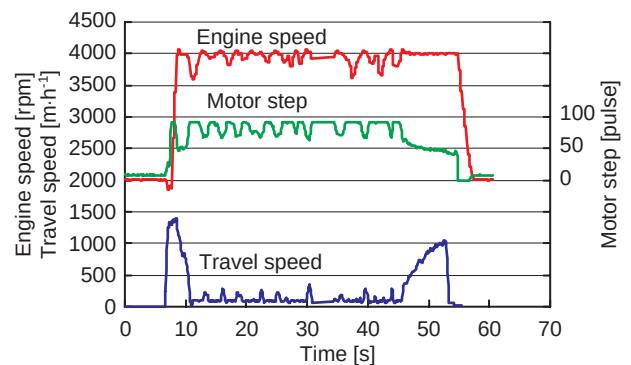


Fig. 8 Clearing snow in power mode

4. Fuel Economy Enhancement Effect

The system configuration is like that of the HS1390i snow thrower. Engine output drives the auger and blower, which clear the snow, while only the motor drives the crawler, which moves the snow blower, when a series hybrid method is used. In addition, there is data communication between the ECU, which carries out system control, and the iGX440 electronic governor control system. Figure 9 shows the system's schematic diagram.

In the newly developed system controls, there is an Auto Mode that automatically lowers engine speed when it is unnecessarily high by controlling the engine speed settings by mode. In addition, the HSM1590i snow thrower has better fuel economy than the HS1390i snow thrower, due to its better snow clearing efficiency. The benefits, with the same area and the same load rate, are outlined below.

Working conditions

- (1) Snow Clearing Surface Area: 128 m²
- (2) Load Rate: 80%
- (3) Deposited Snow: 0.4 m
- (4) Specific Gravity of Snow: 0.1
- (5) Actual Snow Clearing Width: 0.7 m
- (6) Engine Speed (HS1390i snow thrower): 3000 rpm
(HSM1590i snow thrower): Auto Mode

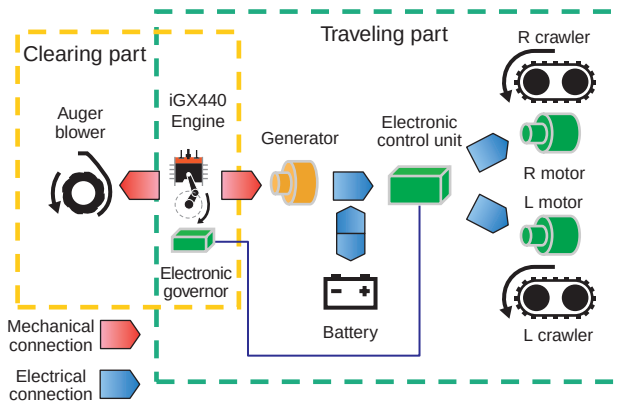


Fig. 9 System configuration

Maximum snow clearing capability at an engine speed of 3000 rpm is 79 ton/h and 92 ton/h, respectively. Snow removal operation speeds with an 80% load rate are 0.155 m/s and 0.18 m/s, respectively.

Tables 2 and 3 show fuel consumption when reverse speed for both is 0.5 m/s, a direction correction count is 56, and the time required to make the turns is 10s and 5s, respectively.

As a result, the fuel economy when clearing snow over the same surface area by the HSM1590i is enhanced 22% than that of the HS1390i. In addition, during actual snow removal, when operation loads are inconsistent, it is necessary to familiarize oneself with the unit's operation, in terms of the optimal traveling speed that allows maximum snow clearing capability. However, with the newly developed HSM1590i incorporating the control system, load control is carried out using the throttle opening, so the ECU automatically reduces traveling speed so that the load does not exceed the maximum snow clearing capability, meaning that even novice operators can remove the same amount of snow as an experienced operator.

Table 2 Fuel used by HS1390i

Mode	Time required [min]	Engine speed [rpm]	Fuel efficiency [L·h ⁻¹]	Fuel [L]
Clearing snow	19.7	3000	3.30	1.08
Reverse	4.0	3000	1.32	0.09
Turning	9.3	3000	1.32	0.21
Total	33.0			1.37

Table 3 Fuel used by HSM1590i

Mode	Time required [min]	Engine speed [rpm]	Fuel efficiency [L·h ⁻¹]	Fuel [L]
Clearing snow	16.9	3000	3.20	0.90
Reverse	4.0	2000	1.16	0.08
Turning	4.7	2000	1.16	0.09
Total	25.6			1.07

Figure 10 shows an external view of the HSM1590i snow thrower, fitted with the newly developed control system.



Fig. 10 External view of HSM1590i

5. Conclusion

The following advantages were achieved with the snow thrower incorporating the control system that utilizes the electronic governor controller features.

- (1) Stable snow throwing distances were achieved because constant engine speeds could be maintained, even with increases and decreases in load.
- (2) It was possible to detect load increases before the engine speed dropped, reducing instances of snow thrower ride-up and traveling speed adjustment operations by the operator.
- (3) Engine speed settings were automatically adjusted, and reduction of engine speed when high engine speeds were unnecessary resulted in a 22% enhancement in fuel economy and quieter operation.

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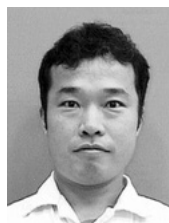
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Research on Technology Used in Predicting Traffic Congestion for Navigation System

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ABSTRACT

Today, most vehicle navigation systems use a Vehicle Information Communication System (VICS), which provides drivers with traffic information for specific areas. VICS is extremely useful in keeping drivers informed of current traffic conditions in the area they are driving in, but drivers also require information on the traffic conditions of roads leading to destinations in other VICS areas and recommendations on alternative routes based on the information. The fact that VICS information can now be used by private companies has allowed Honda to provide information on traffic conditions up to the driver's destination and to conduct research on the prediction of the vehicle's travel time using recorded VICS data. This paper discusses the developed prediction method.

1. Introduction

In the evolution of navigation systems, a function that accurately guides you to your destination has been demanded. There has been research into innovative map databases and route calculation methods to create optimal routes^{(1), (2)}, and functions were provided to guide the driver along the optimum route in an easy-to-understand way. In 1996, the Vehicle Information Communication System (VICS)⁽³⁾, which had been in development for some time, was put into service, allowing us to provide drivers with real-time traffic conditions as well as optimum routes using that information.

Traffic conditions are monitored in each prefecture by the local police, and by road administrators such as the Japan Highway Public Corporation, for road units. The information is collected by the Japan Road Traffic Information Center, batch processed and edited by the VICS Center so the data can be supplied to navigation units with VICS receivers. The VICS receivers obtain data using optical and radio wave beacons, FM waves and other means (Fig. 1).

Normally, real-time VICS information is transmitted only in a given geographical area (Fig. 2), and it is not possible to obtain information beyond the current area. Therefore, there are cases where, at the time the information is received the system shows smooth traffic ahead, but when the vehicle reaches that area, it has become congested (Fig. 3). This results in inaccurate arrival times being given by the navigation system.

In 2002, VICS data was provided to the public, and it became possible for telematics service companies to provide users with secondary processed data. This has led to the possibility of not only providing VICS information from other areas, but also of providing an optimum route while predicting an accurate arrival time, and overcoming the issue mentioned above.

In this project, the authors verified whether it was possible to predict traffic conditions, such as congestion, on specific roads by analyzing VICS historical information.

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** Honda Motor Co., Ltd.

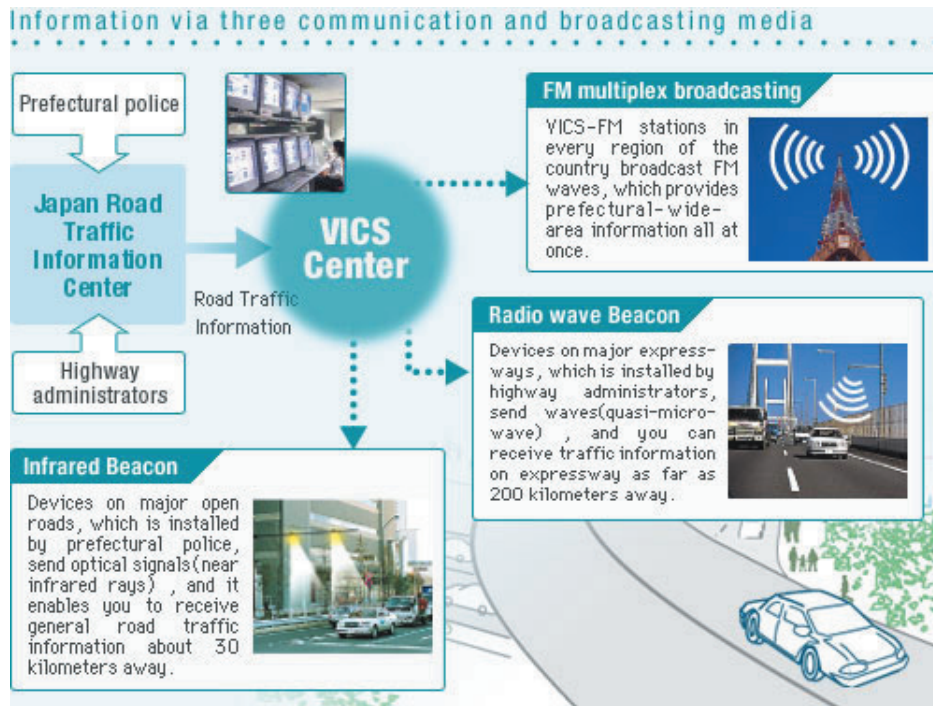


Fig. 1 VICS system schematic

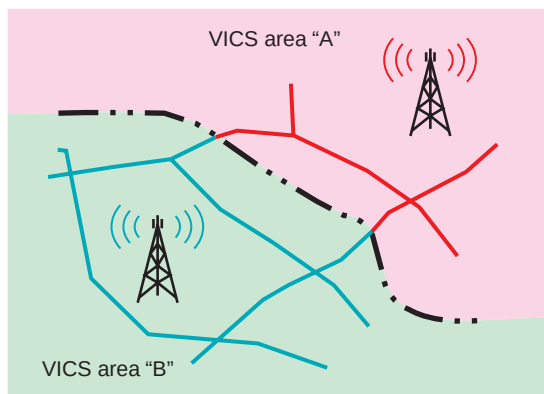


Fig. 2 Image of VICS areas

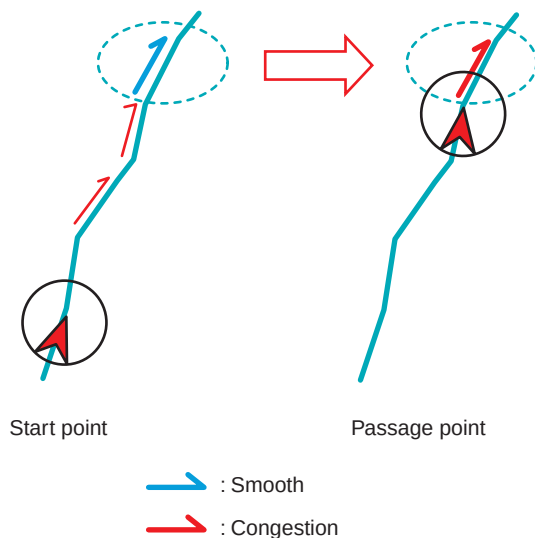


Fig. 3 Change in traffic status

2. Traffic Status Simulation

2.1. Verification Method

VICS establishes segments called "VICS links" for each section of road data, and for each link, the time (seconds) taken to pass through that link is provided (link passing time). Until now, methods to predict traffic conditions, such as congestion, have included those that simply determine conditions from historical data on days and time bands for an observed link, and those that carry out complex calculations from traffic flow over the observed link⁽⁴⁾⁻⁽⁷⁾.

In this project, the goal was to increase accuracy and make more efficient predictions, and the authors studied a method using searches of approximate values from historical data.

Creating a graph with time as the horizontal axis (24 hours) and link-passing time in an observed link as the vertical axis allowed researchers to calculate the passing time at an observed link every five minutes, when the VICS data was updated (Fig. 4). Here, the prediction base time t_b is set and the specific time period immediately prior to t_b is established as basic data to search (matching time width t_m) (Fig. 5). Calculations are made by

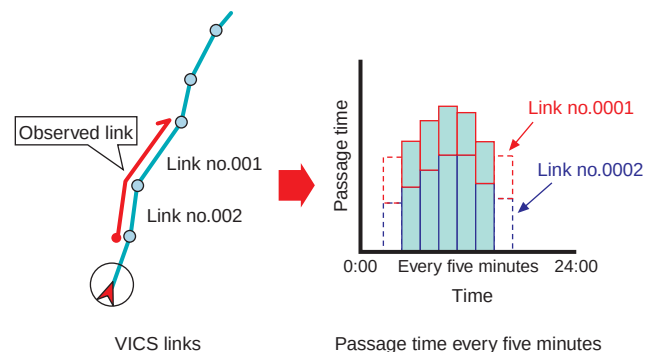


Fig. 4 Change in passage time in observed link

searching for approximate values that have the least difference among historical data. Because calculations with all historical data subject to the search would increase processing, the calculation range was limited to search time width, t_s , and calculations were carried out within that range. After searching approximate data series for those with the least differential (Fig. 6), the data after the extracted data series (predicted time) were set as the predicted value (Fig. 7).

In this method, the following major parameters are thought to cause fluctuations in accuracy.

- (1) The time width that was the basis for the search (matching time width t_m)
- (2) The time width that was subject to the search (search time width t_s)
- (3) The historical data group that was subject to the search (period, time, etc.)

In this simulation, all historical data were divided into periods. Search processing was carried out for every five minutes, and predicted values were obtained at 15-, 30-, 45-, and 60-minute intervals. The predicted values were then compared to the actual values, and the differences between these values were quantified into the mean square deviation, and evaluated.

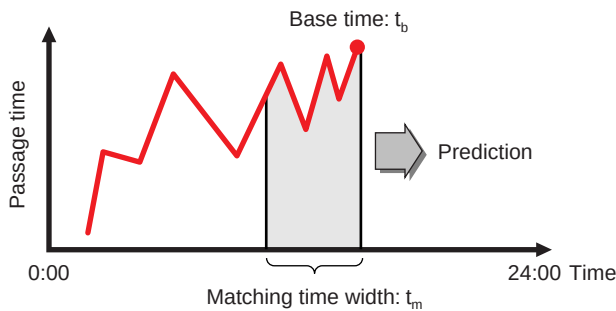


Fig. 5 Traffic status simulation procedure

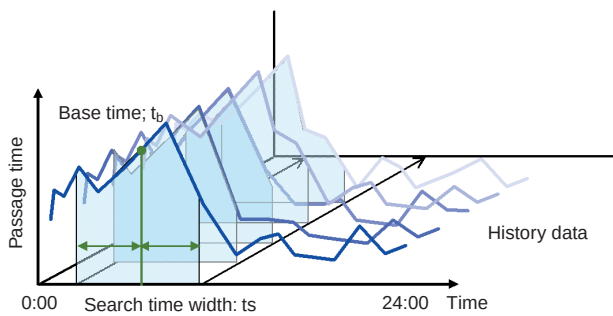


Fig. 6 Passage time each hour

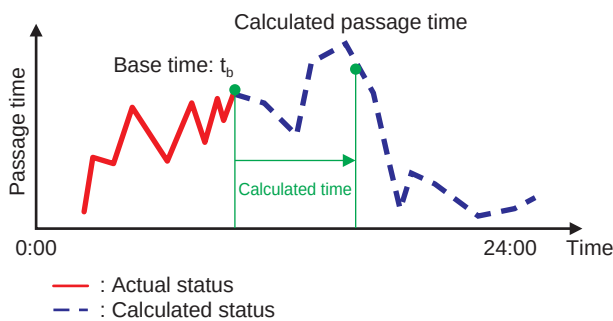


Fig. 7 Image of traffic status simulation

2.2. Subject Roads

When every VICS link is subject to predictions and searching, some links without apparent features, such as uncrowded roads, are included, and it was expected that data search quantity and processing load would be large. Therefore, it was decided to limit the subject links to those between a link where congestion easily occurs (immediately before and after an intersection, etc.) and the next link where congestion easily occurs. Looking at the level of congestion, several links are set in one series.

This paper introduces a verification example of National Highway 1 from the area around the intersection of the Akabanebashi Minami in Tokyo Minato Ward, to the area around Shirokane 1 intersection.

2.3. Verification of Prediction Simulation Parameters

First, as one parameter in prediction simulations, short (15 minutes, which is 3 points) to long (120 minutes, which is 24 points) matching time t_m ranges, which are the base data units for searching, were compared, with the reference point of the time subject to prediction t_b . The results showed that whichever has the shortest matching time width t_m has the smallest error, regardless of the prediction time.

The search time width t_s is another parameter, as the historical data within t_s is subject to search. Verification was made of the effect of the search time t_s range on accuracy. A simulation was implemented from a short t_s of 40 minutes to a longer t_s of 240 minutes. The results showed that there was no great error change due to search time width t_s , and that the effect was small. In the actual prediction simulations, it was learned that it was possible to reduce the calculation load by setting the previously mentioned matching time width t_m , and at the same time, making the t_s range small.

Finally, to investigate whether there was any effect due to the period of the historical data, which was the third parameter, the historical data were divided into 4 data groups by period.

- (1) The past year (March 2001 through February 2002)
- (2) The past 3 months (Dec 2001 through Feb 2002)
- (3) The past 1 month (Feb 2002)
- (4) The same month one year previously (Mar 2001)

Historical data would be an important factor if it was possible to achieve accuracy when the amount of data was small. This has a direct effect on processing load when making predictions on the national level.

In verification of this parameter, there was a tendency for slightly fewer errors when using (1) historical data from the past year, which has the longest period among the 4 data groups. However, it was learned that, in general, there was little effect due to the difference in periods.

From the results of verifications of these three major parameters, this prediction simulation indicated error sixty minutes later with approximately 12% to 13% better than when not making predictions, as shown in Fig. 8.

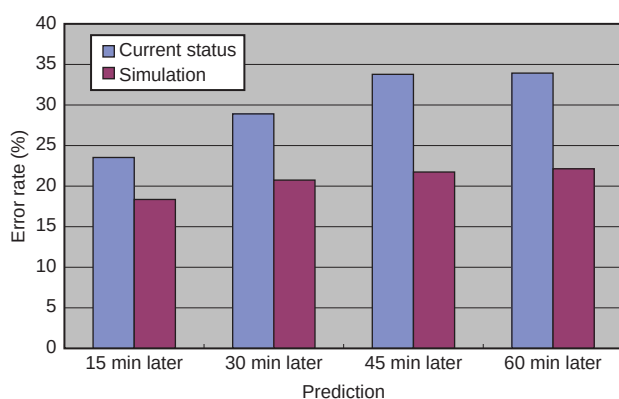


Fig. 8 Result of traffic status simulation

3. Conclusion

To actually provide prediction information using VICS historical data, initially there was concern that it would be necessary to have prediction simulation that searched and analyzed large amounts of data. However, in this evaluation, a variety of prediction parameters were verified, making it possible to greatly reduce data load. At the same time, it was learned that predictions closer to reality can be obtained by comparing this with “the application of current information to arrive time, as it is”, such as normal VICS information. The results of these verifications are currently being applied to congestion predictions provided by Honda’s information provision service, internavi Premium Club. The arrival time at the next link series is determined from the predicted passing time, and by repeating one after the other calculations of passing time after the arrival time, detailed predictions can be made for the whole course. A new initiative is to have the VICS Center collate travel data recorded by navigation systems, concerning roads where VICS information is not provided, process the data, and redistribute it to the navigation systems. This is known as the Probe Information Service, and it started in 2003. At present, due to the data quantity issue, this data is not subject to data prediction simulations. However, by collecting enough data for prediction verifications, and by making technological studies to further increase accuracy, in the future it should be possible to achieve a service that predicts traffic conditions on almost all roads heading to a destination.

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Development of Puncture-Resistant Tubeless Tire for Motorcycles

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ABSTRACT

A puncture-resistant tubeless tire has been developed using the same principles as the puncture-resistant double-chamber inner tube. The inside of the newly developed tubeless tire is a double-chamber structure. The new double-chamber tires were tested at high and low temperatures for puncture resistance and to determine a damage-size limit. They were also endurance-tested with a hole deliberately created in the tire, and positive results were attained. The high-temperature test has proven that the tire prevents air leakage after piercing with a pin as large as 9 mm in diameter. In addition, the endurance running tests have proven that the tire with pierce damage is able to run up to the tire wear limit, 20 000 km.

1. Introduction

Robert Thomson invented the pneumatic tire in 1845, but the first practical tire was not developed until 1888, by the Scottish veterinarian John Boyd Dunlop. Even after 120 years, pneumatic tires are still manufactured in large quantities. The merit of pneumatic tires is that they fulfill the four major functions of a tire, as follows:

- (1) Supporting the vehicle load.
- (2) Transmitting driving force and braking force to the road surface.
- (3) Reducing impact from the road surface.
- (4) Providing optimum elasticity for changing and maintaining direction.

The history of the pneumatic tire is also a history of anti-puncture measures, but these measures had not resulted in an effective solution.

Today's motorcycles present fewer mechanical malfunctions during operation, due to increases in their quality and durability.

However, generally, motorcycles do not carry spare tires, and therefore cannot be operated until repaired if a puncture occurs. The development of a high puncture-resistant tire was desired.

In 1996, Honda developed a puncture-resistant, sealant-filled double-chamber tube for motorcycle tires, and it has been marketed in Japan, India, Brazil and other countries.

2. Double-chamber Tire Tube

2.1. Structure of the Double-chamber Tire Tube

Figure 1 shows the structure of the double-chamber tube⁽¹⁾. The 180 degrees of the tube that faces the tread is double-chambered. That chamber is filled with an anti-puncture sealant. This sealant chamber and the air chamber are separate, and air pressure constantly presses against the sealant chamber. Because of this pressure, if the tube is punctured at any point by a nail or other object, sealant will be forced out of the puncture hole. Due to the contractile force of the rubber, fibrous materials in the sealant are

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trapped and plug the hole, preventing air leakage. Even if the motorcycle is left parked for a long time with the hole in the tire facing upwards, the sealant will maintain its grip around the hole and prevent air from leaking.

2.2. Sealing Mechanism

The sealing mechanism of the double-chamber tube is shown in Fig. 2. When a nail or other sharp object pierces the tread, it normally punctures through the sealant chamber to the air chamber. As air leaks from the hole in the tire, fibrous components in the anti-puncture sealant are carried to the hole by the liquid component. These fibrous materials are squeezed by the contractile force of the rubber. Microscopic particles and liquid fill the spaces between the fibrous materials, and the hole is plugged. Different types of fibrous materials are used to ensure excellent performance in plugging large and small holes. The liquid will not leak into the tire as long as it is pressed on by the air pressure in the tire. Figure 3 shows an image of the sealant sealing an actual puncture, clearly showing the fibrous material squeezed between the rubber.

2.3. Effectiveness of Double-chamber Tire Tube

To verify the effectiveness of the double-chamber tube in actual use, comparisons were conducted of 98 motorcycles used in newspaper delivery. Motorcycles for newspaper delivery are usually used under heavy conditions such as frequently starting, accelerating, and braking during relatively short travel distances. Figure 4 shows a comparison of results obtained when conventional tubes and double-chamber tubes were used for a period of four to six months each. The use of the double-chamber tubes resulted in a 90% reduction in punctures compared to the conventional tubes.

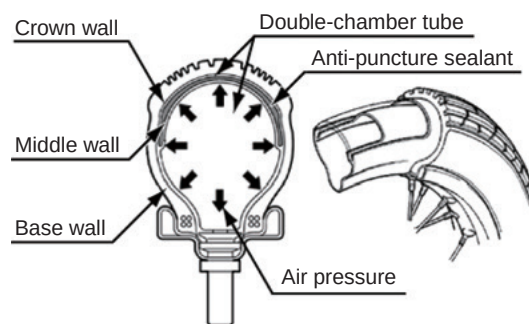


Fig. 1 Structure of double-chamber tube

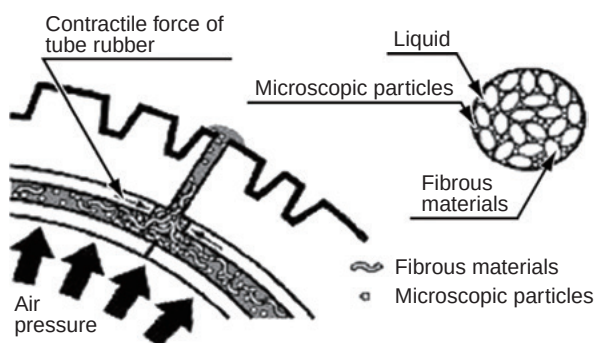


Fig. 2 Filling mechanism

3. Overview of Double-chamber Tubeless Tire

The double-chamber anti-puncture technology was applied to a tubeless tire to develop a puncture-resistant double-chamber tire.

Figure 5 shows the structure of the double-chamber tubeless tire (hereinafter referred to as "double-chamber tire").

A sealant chamber filled with anti-puncture sealant is formed on the tire inside against the tread. When the tire is fitted on the wheel, the sealant chamber and the air chamber are separate, and air pressure constantly acts on the sealant chamber. For this reason, no matter where a puncture penetrates the sealant chamber, air pressure will force the sealant out of the hole and prevent air from leaking. Even if the motorcycle is parked with the hole facing upwards, the anti-puncture sealant will maintain a seal. The tire can also be deflated for inspection or repair, and no sealant will come out from an air-valve.

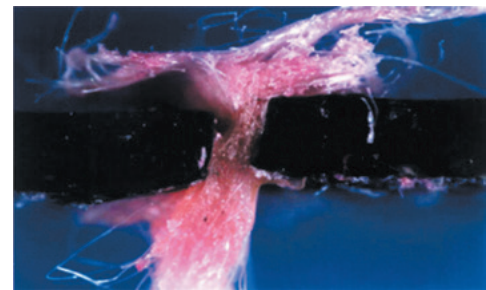


Fig. 3 Photo of filling situation

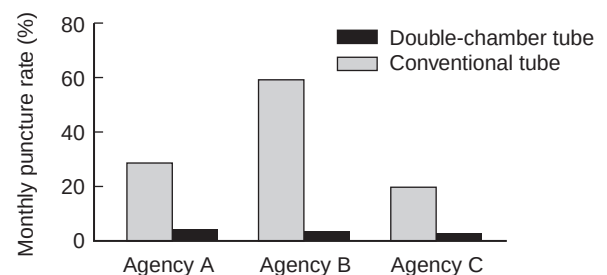
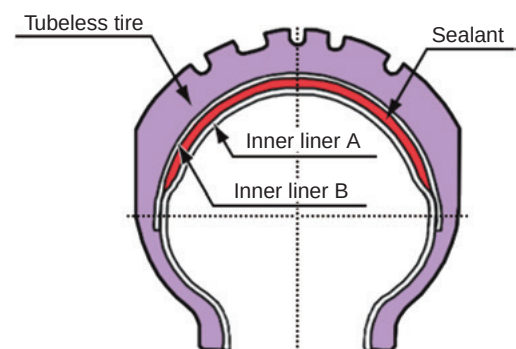


Fig. 4 Effectiveness of double-chamber tube



Inner liner added to conventional tubeless tire

Fig. 5 Double-chamber tire

Figure 6 shows a schematic diagram of technologies for dealing with punctures^{(2), (3)}, including the double-chamber tire. In preventing the leakage of air even when the tire is punctured, this technology differs significantly from conventional technologies that respond to punctures after the air has escaped.

4. Development of Double-chamber Tubeless Tire

4.1. Development Goals

To achieve an equivalent level of performance to that of double-chamber tubes, the following were set as development goals.

- (1) Ability to be manufactured without major changes in the conventional tire manufacturing process.
- (2) Ability to continue use for the standard life of the tire even when punctured by a nail.
- (3) Equivalent handling and stability to that of a normal tire.
- (4) Interchangeability with conventional wheels.

4.2. Development of Production Technology

4.2.1. Production process

To determine a method of forming the sealant chamber in the conventional tire manufacturing process^{(4), (5)} was an issue. Figure 7 shows the production process.

The forming stage for bias tires normally involves wrapping the various sheets from which the tire will be composed of around a drum and forming them into circular shapes. First, an air-impermeable butyl rubber sheet known as the inner liner is wound around the drum. This is followed by the tire carcass, formed from crossed plies of woven materials sandwiched between layers of rubber. Finally, the tread rubber, which will form the area that will contact the ground, is wound around the drum.

These untreated rubber tubes are subjected to heat and pressure in the vulcanizing process to form them into a tire. The tread pattern on the surface of the tire is formed by the application of pressure during the vulcanization process.

The sealant chamber of the double-chamber tire was manufactured using a rubber sheet; the rubber material forming the sealant chamber can be peeled away after forming the sealant chamber.

This concept of peeling away a layer after formation is the same for the double-chamber tube. The expansion rate of the tube in the vulcanization process is less than 10%. Talcum powder is used to eliminate an adhesion between the rubber sheets and allows the peeling of the rubber sheet.

However, as shown in Fig. 8, the expansion rate of the tire in the vulcanization process is around 150%. When using talcum powder, the rubber in the area without the powder adheres because the powder cannot cover the entire surface of the rubber sheet on a stretched tire.

Therefore, requirements of peeling are as follows:

- (1) To expand to the same extent as the rubber, in order to enable peeling after formation.
- (2) To have no effect on puncture-resistance performance.

Various materials were tested as the peeling sheet, as shown in Table 1. As a result, Film D satisfied both requirements, cost and durability. Forming the double-chamber tire was enabled by the selection of this material.

4.2.2. Homogeneous distribution of puncture sealant

Because the sealant in the sealant chamber is required to prevent air from escaping no matter where the tire is punctured, it should be homogeneously distributed. Diagonally opposed punctures on the tread of double-chamber tire were pierced using pins of 0.8, 3, and 5 mm in diameter. The pins were used to make two, four, and two, punctures respectively, eight in total. After running using this tire, the distribution of the sealant was inspected by a CT scan. Figure 9 shows the result of the distribution.

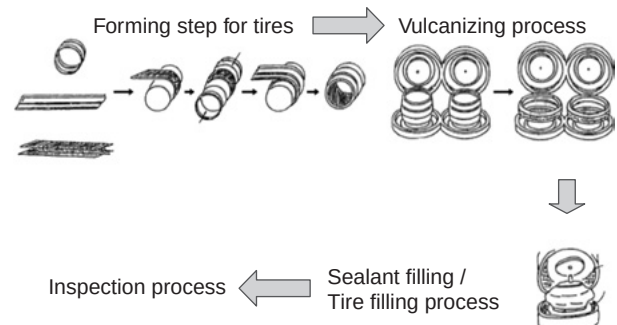


Fig. 7 Manufacturing process for tires

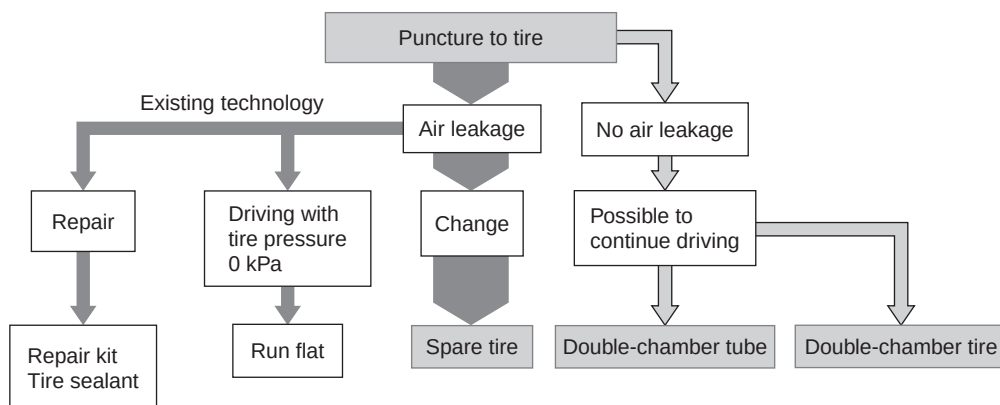


Fig. 6 Schematic diagram of technology dealing with puncture

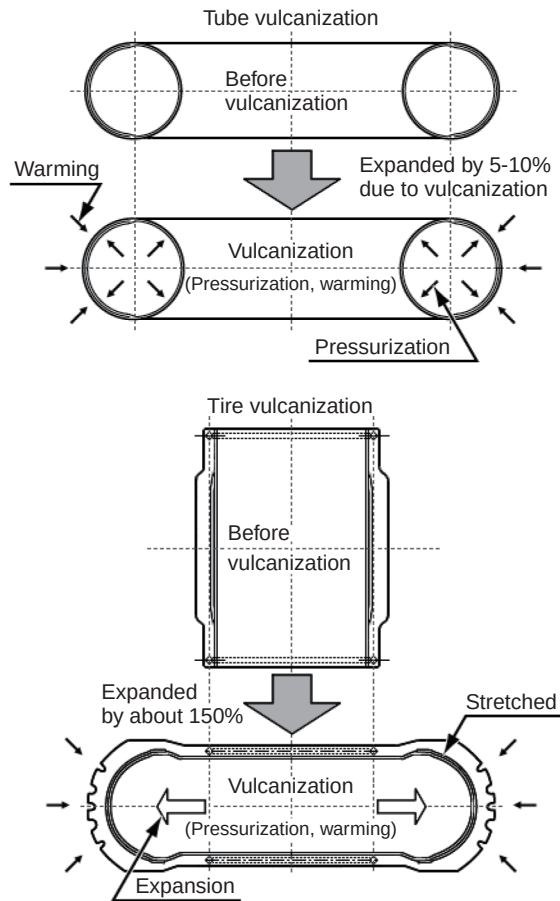


Fig. 8 Vulcanizing process

Table 1 Consideration of release agent

Test sample	Formability of fluid chamber	Initial anti-puncture performance	Anti-puncture performance after durability test
Talcum	Insufficient	Insufficient	—
Sheet A	Insufficient	Insufficient	—
Tape A	Sufficient	Insufficient	Insufficient
Film A	Sufficient	Insufficient	Insufficient
Film B	Sufficient	Insufficient	Insufficient
Film C	Sufficient	Sufficient	Insufficient
Film D	Sufficient	Sufficient	Sufficient

Measurement of the thickness of the tread showed that the sealant remained homogeneously distributed under these conditions.

4.3. Results of Puncture-resistance Test

An important function of the double-chamber tire is to be capable of driving after punctured. A drum endurance test was conducted using the pierced tire. As a result, the tire accomplished continuous drive for 10 000 km at 65 km/h drum operation speed as shown in Fig. 10.

Figure 11 shows the change in the tire pressure when the tube tire and the tubeless tire were rotated on the drum after a 2.8 mm diameter nail puncture.

The change in the tire pressure is minimal until the nail comes out. However, the nail comes out when the centrifugal force grows in a high-speed rotation, and the air leakage occurs. At that time, the air leakage of the tube tire was earlier than that of the tubeless tire. Moreover, when running at the same speed the angular velocity of the small radius tire was large and the centrifugal force acted strongly on the nail in the outer surface of the tire. Therefore, the stuck nail on the small diameter tire came out easily in a low-speed rotation.

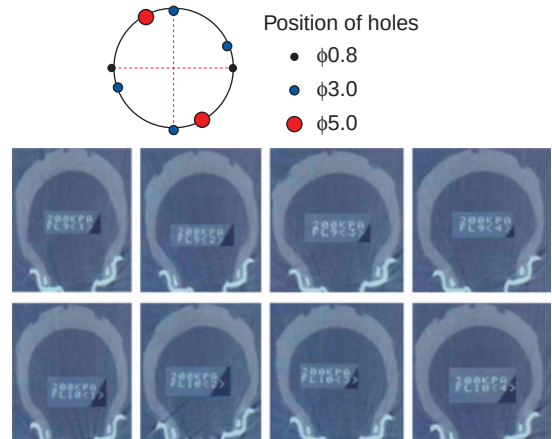


Fig. 9 Pictures of tire by CT scan

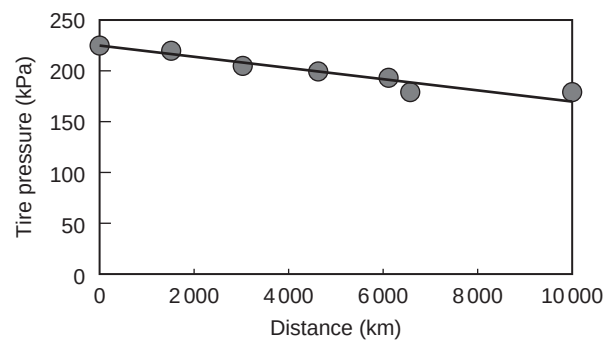


Fig. 10 Result of durability test on drum bench

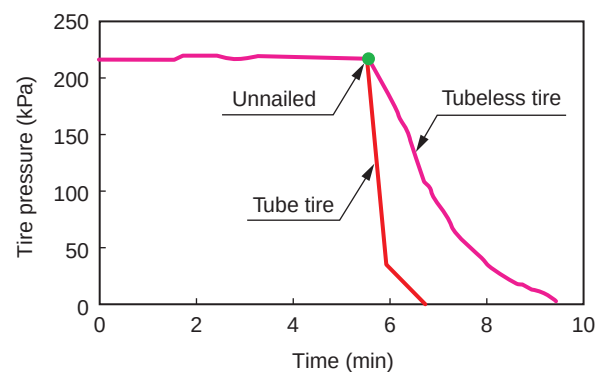


Fig. 11 Result of air leakage by drum bench test with sticking nail

4.3.1. Actual running durability test

Actual running durability tests were conducted on a pierced tire up to the tire wear limit of 20 000 km, and its air leakage was inspected. The sealant demonstrated stable plugging and actual running performances at limit tests under high and low temperatures.

Figure 12 shows the results of the actual running durability test conducted with a pierced tire. In actual running durability, the test motorcycle with the pierced tire ran 20 000 km on the road. To make the puncture conditions even more severe in this 20 000 km test, the hole was made with a blade of 2 mm in width instead of a round pin. After 12 000 km, the air pressure of the tire had declined by approximately 30%, but after being refilled to the correct pressure, the tire was able to be used until the tread wear limit of 20 000 km.

4.3.2. High-temperature test

Figure 13 shows the results of tests conducted at high temperatures with holes in the tire. In this test, nine holes were made in the same tire in ascending order of diameter with pins of 1 to 9 mm in diameter.

Every time a puncture was made, the motorcycle ran the range of 6 km, and the tire pressure was measured. When the 7 mm pin puncture was made there was a 15% drop in tire pressure, but since this was a large hole, a certain degree of time was required for the sealant to fill in the hole and stop air from escaping. The drop in tire pressure represented the amount of air that escaped during that period. The test was carried out in Death Valley in the United States under atmospheric temperature of 48 °C.

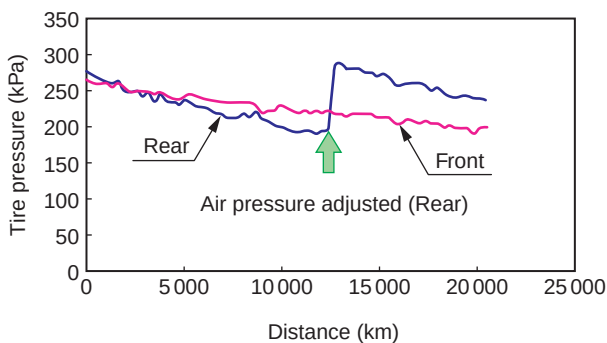


Fig. 12 Result of durability road test with tire with hole

Figure 14 shows the results of a 300 km durability test with a puncture made by a 3 mm diameter pin in the rear tire. The results were compared with the air pressure of the front tire without punctures. In Fig. 14, the drop in pressure was a result of the overnight drop in atmospheric temperature. Comparison with pressure changes in the front tire indicates that no air was leaking from the tire.

4.3.3. Low-temperature test

Figure 15 shows the results of test runs over 200 km at low temperatures. A puncture was made in the rear tire only. Because the rear tire is the drive wheel, at low temperatures, the temperature of the rear tire increases greater than the front tire, and hence its pressure increases greater, as the results indicated.

4.3.4. Handling and stability

The double-chamber tire developed for small sized scooters was tested for handling and stability. Because the double-chamber structure changed the stiffness of the tire case, the internal structure of the tire, including cord angle, was modified and the tire achieved handling and stability equal to those of a normal tire.

4.3.5. Verification of wheel interchangeability

The developed double-chamber tire has a double-chamber inside the tire facing the tread. The other part of the structure is almost the same compared with conventional tires. Tests of fitting the double-chamber tire on a conventional rim demonstrated that the new tire satisfied the development target of being interchangeable with conventional tires.

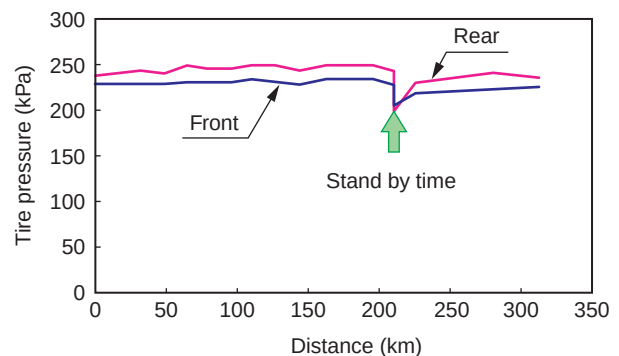


Fig. 14 Test run at high temperature with tire with hole

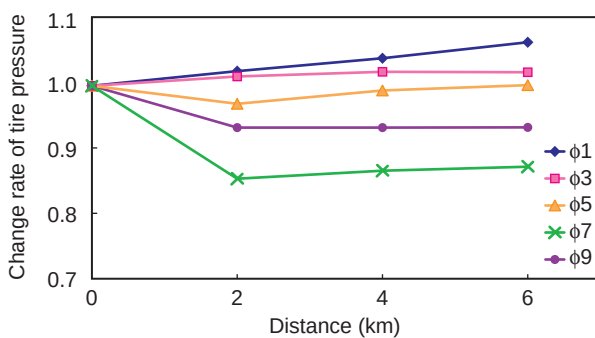


Fig. 13 Result of performance test at high temperature with tire with hole

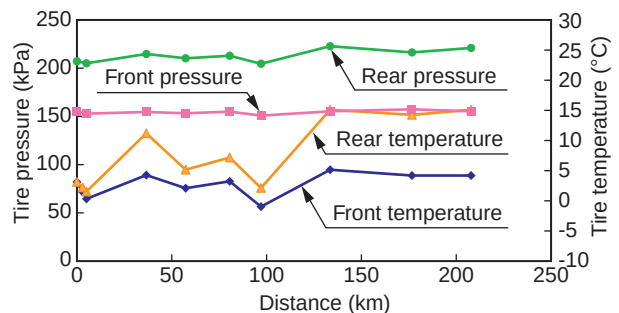


Fig. 15 Test run at low temperature with tire with hole

5. Comparison of Other Puncture-Resistant Technologies

Figure 16 shows some potential running distance of puncture-resistant tires punctured by a pin of 3 mm in diameter. A reinforced sidewall type and a Core type have been developed as “Run-flat tires” technologies for automobiles. The reinforced sidewall type is reinforced by the thick sidewall of tire. This tire can drive by supporting the vehicle mass by the thick sidewall even when air has completely leaked out. There are various types of Core type tires. This tire can drive by supporting the vehicle mass by the core equipped on the wheel rim. PAX tires are core-type tires.

Because double-chamber tires prevent air leakage, they are able to drive for longer distance than “Run-flat tires.”

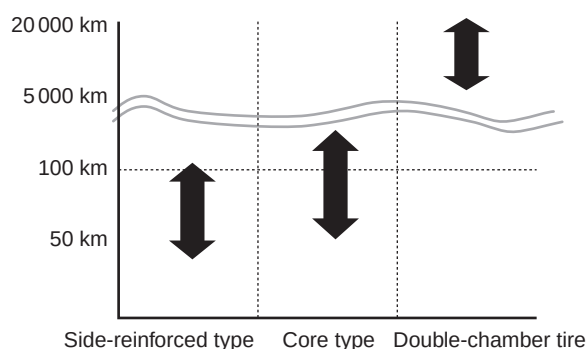


Fig. 16 Comparison of potential travel distance after puncture

6. Conclusion

A double-chamber tire containing a chamber filled with a puncture sealant has been developed to enhance the puncture resistance of small-diameter tubeless tires. Results in this development are as follows:

- (1) Specifications of double-chamber tire such as material and structure were established, and its puncture-resistance has been verified.
- (2) A manufacturing technique has been developed that does not necessitate major changes in the conventional manufacturing line.
- (3) A method of ensuring the handling and stability of the double-chamber tire has been determined.
- (4) Tests verified that the double-chamber tire could be fitted onto a conventional rim.

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Development of Technology to Create Surfaces with High-speed from Measurement Data

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ABSTRACT

Transformation of measurement data from parts into CAD models allows the data to be more widely used in the automotive development process. Surfaces are formulated from the data and fed back into the CAD model. These surfaces must correlate well with a low degree of error against the measurement data.

However, creating these surfaces requires considerable data processing time. A new algorithm for surface creation that combines high accuracy with high-speed processing has therefore been developed. This has resulted in a 90% reduction in the time required for surface creation with no significant increase in error.

1. Introduction

Two-dimensional (2-D) blueprints and three-dimensional (3-D) shape data have become common design tools in automobile development in recent years. As non-contact measurement tools have evolved, it has become possible to measure the dimensions listed on 2D blueprints (diameters, heights, etc.) as well as 3D surface forms, and the amount of information available has increased dramatically.

Transforming measurement data into surfaces and using it in CAD systems, where all automobile development starts, allows developers to use this increased amount of information (data), offering the following potential benefits.

- (1) Verification by 3D layout
Even products for which blueprints and CAD data do not exist can be verified by 3D layout.
- (2) CAM-based cutting programs
Cutting programs can be written for the form data derived from the products, making it possible to take the form of the product into account when machining it.
- (3) Edited forms reflected in design values
Product forms that workers have revised can easily be reflected in design values.

- (4) CAE analysis using measurement data

Form data for actual articles can be transformed into CAD data and undergo CAE analyses, enabling a functional comparison of design values and actual articles.

The ability to create surfaces that differ only slightly from measurement data is essential to reaping these benefits, and ordinarily it is accepted that this error should be within ± 0.1 mm.

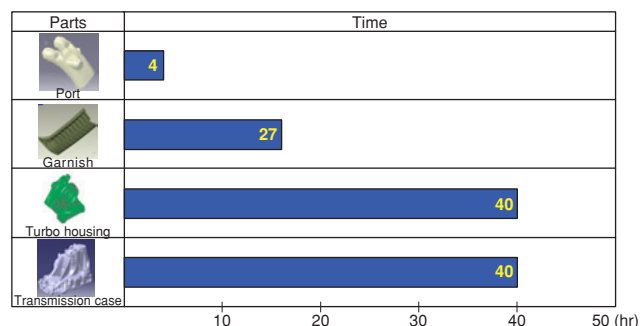


Fig. 1 Time required for creation of surfaces before software development

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However, keeping error within ± 0.1 mm when the measured form is complicated requires a great deal of time, particularly for components like transmission cases (Fig. 1), and this becomes an issue in the use of measurement data.

This article provides details on a newly developed algorithm for creating surfaces with the following two target values.

- (1) Error within ± 0.1 mm of measurement data for the surface created
- (2) Time required to create surfaces reduced by 90%

2. Current State of Surface Creation

2.1. Clarification of Processes Needing Modification

The process of creating surfaces from measurement data goes as follows.

- Measurement data editing
- Display of polygonal form
- Polygon editing
- Surface layout editing
- Creation of surface

First, to clarify the areas of the surface creation process that need modification, standard data (for a transmission case) was used to verify the time needed for each process (Fig. 2).

The results indicated that two of the five processes, (3) polygon editing and (4) surface layout editing, had a significant impact on surface creation time.

2.1.1. Polygon editing

Faults occur in triangular polygons created using the measurement data's 3D coordinates as apexes. The forms represented by these polygons do not adequately capture measurement data and the surface status of the product (especially glossiness) is inadequate for the measurement (Fig. 3). The objective of this process was to make up for these polygon faults and achieve smooth, continuous forms.

With the standard data, 308 polygon faults were confirmed. Selecting and editing each of these manually required 14 hours.

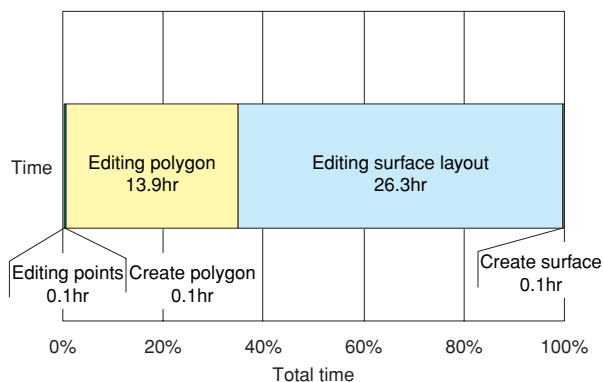


Fig. 2 Time required for each process in creation of surface

2.1.2. Surface layout editing

Details of this process (Fig. 4) are given below.

- (1) The curvature distribution of the form represented by polygons is calculated and contour lines are automatically created.
- (2) At the discretion of the operator, contour lines other than edges are manually edited to follow points of inflection.
- (3) To produce surfaces within the areas bounded by the contour lines, the regions (below, "patches") separated by the four sides are automatically created.
- (4) The arrangement of each patch is manually edited to minimize the changes in curvature within the patch.

These processes are performed to minimize the error between measurement data and the surface created, and also to minimize the size (i.e. the bytes) of the data.

To track the changes in curvature, and thereby minimize deviation from the measurement data, the number of points used to determine surface form must be increased. (These are known as control points.) However, the surface's control points increase in units of one column and one row, so that the data capacity increases proportionately to about the square of the number of control points (Fig. 5), which has an impact on the processing time of the various data. Results of verifying this process with the standard data indicated that the process took 26 hours.

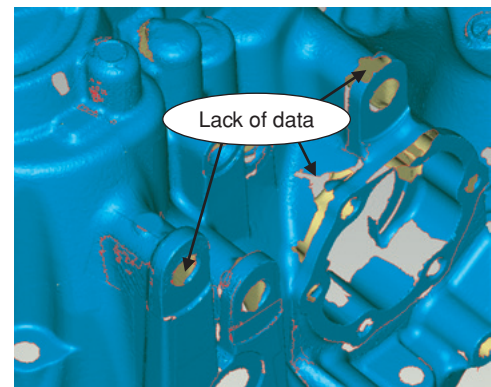


Fig. 3 Hole caused by lack of data

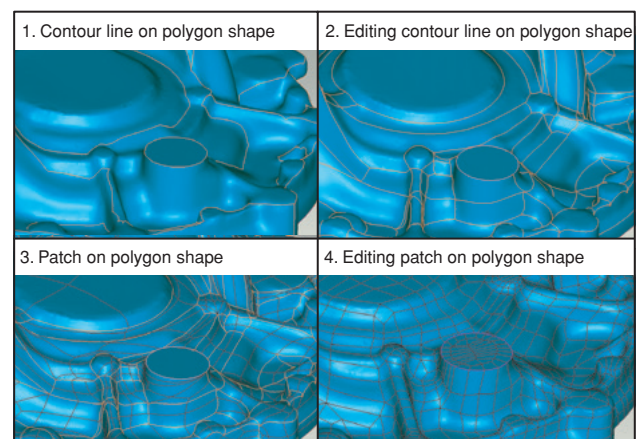


Fig. 4 Generation process of patch lay-out

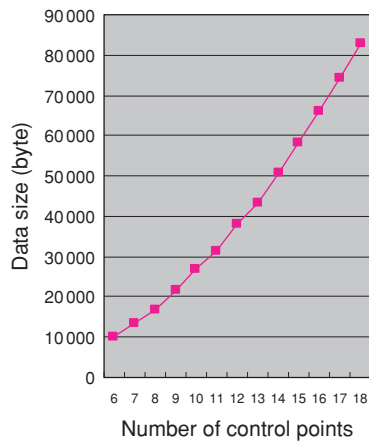


Fig. 5 Relation of control points and data size

3. New Technique

One factor common to these two processes, which increases the time it takes to create a surface, is a need for manual editing. Therefore the authors devised a technique that focuses on how to make manual tasks more efficient.

3.1. Polygon Editing

In this process operators have had to manually select and edit, one at a time, multiple areas with data faults where the shape or size differs. The authors felt that to solve this problem, ideally the operator would have to be able to edit multiple areas simultaneously and that it was important to be able to efficiently select areas where the same editing technique could be used. They devised the following technique based on those ideas.

3.1.1. Selecting parts to edit

In the new technique, the operator can define the conditions for selecting parts to edit and thereby select only polygon fault areas containing forms that meet those conditions. The conditions determine the form characteristics of the fault areas and include such factors as the number of polygon sides facing the fault area, the circumference of the fault area and the flatness of the outline of the fault area. Combining and defining such conditions enables the operator to select fault areas with the same characteristics and edit them all at once.

The defined conditions, moreover, can be saved in a file and used again to edit other measurement data polygons.

3.2. Surface Layout Editing

Until now, contour lines and patches that were automatically detected have been manually edited because the process sought to strike a balance between data capacity on the one hand, and deviations between the created surface and measurement data on the other. The authors have devised a new algorithm that enables efficient manual editing.

3.2.1. Creating contour lines

Creating contour lines has been very time-consuming because the operator has had to perform editing to follow points of inflection and parts found to be edges. To begin, curvature changes for the overall form, represented by the polygon, were calculated to find areas with significant curvature (Fig. 6), and contour lines were created to pass through the centers of these areas (Fig. 7). The operator is able to increase the number of contour lines as necessary.

3.2.2. Creating patches

Considering data size (bytes) and error between measured form and the surface created, it is preferable if the measurement form encompassed by one patch does not have significant curvature change when that patch is created. For that reason, when creating contour lines, patches are created with the contour lines as boundaries on polygons found to have significant curvature (Fig. 8), then patches are created in areas where the remaining curvature change is small. This means that within a single patch, there is no significant curvature change.

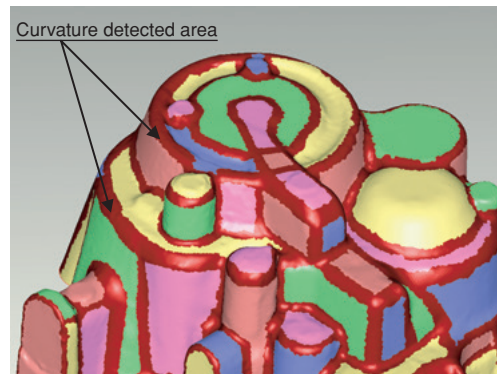


Fig. 6 Discrimination curvature

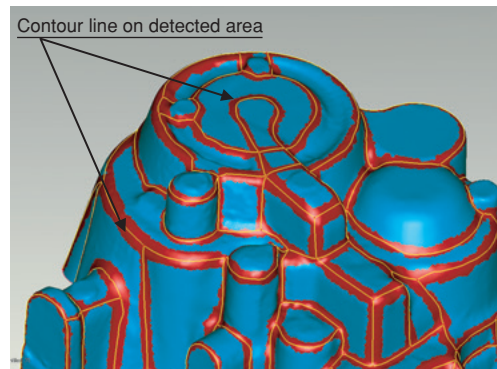


Fig. 7 Contour line created on areas of discrimination curvature

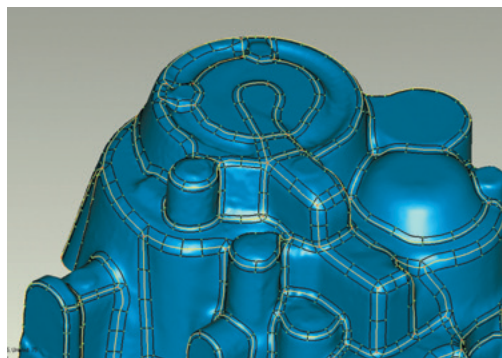


Fig. 8 Patch created on areas of greater curvature

4. Effectiveness

Applying the new algorithm to the existing processes had the following results: When using standard data, the time needed to edit polygons dropped from 14 hours to 1.2 hours and the time required for surface layout editing dropped from 26 hours to 2.6 hours. For all processes, the time it took to create a surface dropped from 40 hours to 3.8 hours. This meets the established target of reducing processing time using standard data by 90% (Fig. 9). Results of verification with other components showed that the time needed to create surfaces declined by 80% or more (Fig. 10) and that it has become possible to shorten the processing time while maintaining the precision of the surface created.

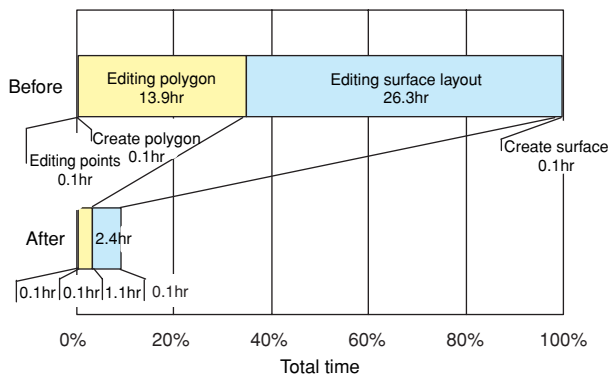


Fig. 9 Time required for each process in creation of surface before and after software development

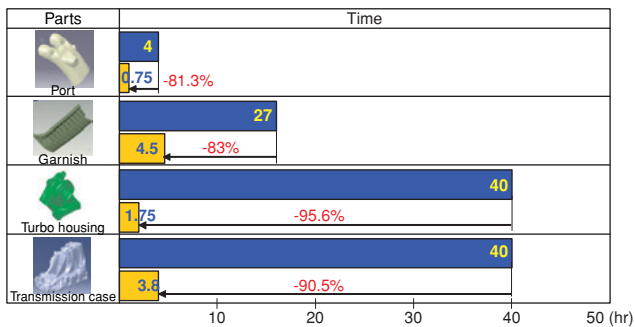


Fig. 10 Verification results of time for creation of surfaces

5. Conclusion

The authors have developed a new algorithm for surface creation that combines high accuracy with high-speed processing from product data.

- (1) Using condition definitions, to select in bulk polygon areas that need revision, enables the operator to reduce the time spent manually selecting areas to edit.
- (2) Curvature changes are calculated and contour lines are created from the areas found. These are the main areas with significant curvature from which patches are created. This enables the operator to reduce the time spent editing contour lines in arbitrary areas.

6. Afterword

The newly devised algorithm makes it possible to shorten the time needed to supply data and also makes it possible to supply data in a timely manner on demand, even when dealing with complicated forms like transmission cases.

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Development of System to Generate FE Models of Body-in-White

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ABSTRACT

A system to generate models for body-in-white analyses has been developed that reduces the number of man hours expended on the formulation of FE models for automotive body analyses. This system comprises a model management system and an automatic mesher. The model management system automatically registers the CAD models of body-in-white parts, data on the thickness and materials, and data on connections between parts, and can also manage the formulated FE models. The automatic mesher creates FE meshes from the registered CAD models. The new mesher provides more detailed reproduction of parts that affect calculation accuracy, including the shape of fillets, beads and flanges and spot welding connections, than the previous mesher. This has reduced the amount of correction required for the models.

1. Introduction

Large-scale automotive body analyses have recently begun to be widely used, and have contributed to reducing the time required for the development process by enabling performance to be evaluated with a high degree of accuracy before an actual vehicle is built. A preprocessor has already been developed that increases efficiency in formulating the FE models used in these large-scale body analyses, thus reducing the time spent on creating the models⁽¹⁾. An automatic mesher for use in generating FE models for body-in-white analyses has also been developed to make the process of generating FE meshes (hereinafter mesh) for various components more efficient^{(2) - (4)}.

However, a huge expenditure of man hours was still required for the collection of data on several hundred body-in-white components and the connections between them, meaning further increases in efficiency are needed to achieve still greater time savings in the development process. In addition, it was necessary to achieve more detailed reproduction of the shapes of components in meshes in order to increase the accuracy of the results of analyses,

and this would also necessitate an immense investment of man hours.

A system to generate FE models for vehicle body analyses, comprising a model management system able to automatically register models and an automatic mesher able to reproduce shapes in greater detail, was therefore developed to make operations more efficient.

2. Flow of Generation of FE Model

A body-in-white is made up of several hundred parts connected by welding and other means. These are reproduced by FE models to enable analyses to be conducted using the finite element method. Figure 1 shows the flow of procedures in generating the FE models.

First, data is collected on the components of the body-in-white. The information that is essential to generating an FE model is the CAD models of the components, data on thickness and materials, and data on connections between parts. The parts might be connected by spot welding, MIG welding or bolts, among other methods. Data on the combination and positioning of connected

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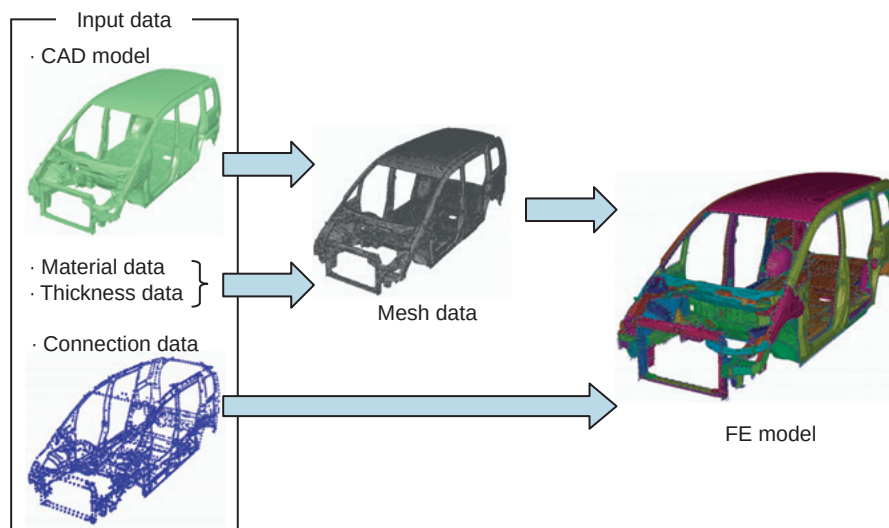


Fig. 1 Data flow of FE model generation

parts is also used.

Next, meshes are created from the CAD models of the components. Press-formed parts are reproduced using triangular and square two-dimensional elements. As the models are being generated, thickness and material characteristics are applied to the elements.

A preprocessor is then used to formulate FE models for use in robust, NV and crash analyses from the generated meshes and the connection data.

3. Issues Concerning the Previous Method

The aspects of generation of FE models for body-in-white analyses that required a large expenditure of man hours were examined.

3.1. Model Management

Although a DPM system⁽⁵⁾, which creates and manages digital mock-up data, was used to enable integrated management of the CAD models of the body-in-white parts and thickness and materials data required for generation of the meshes, a huge number of man hours were previously required to manually program the data for each of several hundred components.

In addition, connection data in technical drawings also previously had to be manually scanned and programmed. This also required a large number of man hours, as an entire body-in-white contains several thousand points of connection.

3.2. Mesh Generation

To achieve greater calculation accuracy, the meshes employed in vehicle body analyses are required to reproduce the original shapes in the CAD model in greater detail. Standard indexes, including warping, aspect ratio and angle, have been established for each solver to ensure element quality. Meshes are required to satisfy these element quality standards, because failure to do so may prevent calculations from being conducted or accurate results being obtained. Reducing the size of the elements generally increases the level of reproducibility of shapes and the quality of the elements, but this method also increases the number of elements, consequently increasing solver calculation time. To achieve detailed

reproduction with the minimum number of elements, it was previously necessary to manually program the edges of fillets, beads, and other elements along constraint edges. In addition, elements that did not satisfy element quality standards had to be corrected manually, and these manual operations also involved many man hours.

4. Development of Model Management System

A model management system was developed to increase the efficiency of the operations that had previously required a huge expenditure of man hours: CAD modeling of the body-in-white parts, collecting data on thickness and materials, programming connection data and generating the meshes. Figure 2 shows an overview of this system.

4.1. Parts Data Collection

The new system centrally manages the CAD models, the data on thickness, materials and connections, and the FE models formulated from the data. The system links to the DPM system, which registers the CAD models and thickness and materials data, enabling it to automatically collect the body-in-white parts data as a package. Even if the process of parts registration in the DPM system is not yet complete, the model management system is able to automatically collect data as parts are completed and to generate meshes.

4.2. Connection Data Collection

The development of a function for the transfer of data programmed in CAD models and other FE models enables connection data to be automatically programmed. When the connection data is incorrectly programmed, a summary for the specific area is output, enabling the programmer to determine and correct any flaws.

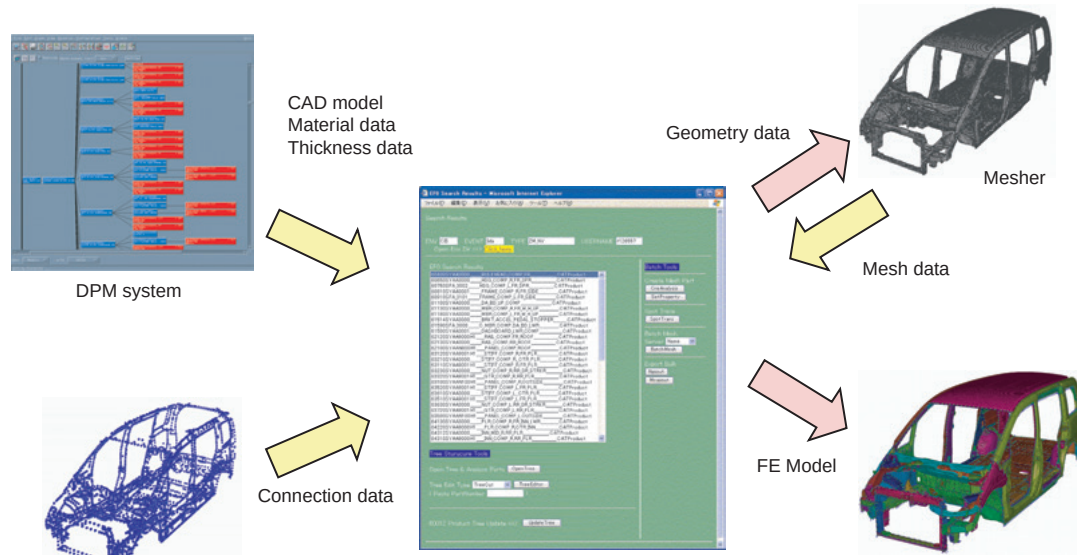


Fig. 2 System overview

4.3. Automatic Mesh Generation

The new system uses an automatic mesher (discussed below), enabling meshes to be generated from the registered CAD models in a single procedure. When the automatic mesher has completed a mesh, it outputs a summary report indicating the success or failure of mesh generation, element quality and other factors. On the basis of this data, the programmer is able to focus on correcting only meshes with flaws.

4.4. FE Model Output

The new system is able to output the generated meshes and the connection data as solver or preprocessor data, as required for robust, NV and crash analyses.

5. Modification of Automatic Mesher

It has become necessary to increase the level of reproducibility of components and the quality of mesh elements in order to increase analytic accuracy, but the previous automatic mesher was unable to automatically generate meshes satisfying these requirements. Modifications were therefore made to the automatic mesher.

5.1. Increased Detail in Reproduction of Fillets

Press-formed parts are generally composed of standard surfaces and the fillets at their boundaries. Reproducing the fillets is an important factor in reproducing the shape of parts using a mesh. However, fillets are not adequately reproduced by mesh when their width is less than the size of the elements. To enable automatic reproduction of this type of fillet, an algorithm was developed that can determine the fillet from the curvature of each face of the CAD model and other data, and automatically extract the edges of the boundaries. The fillet shape is reproduced in mesh by using these edges as constraint edges and aligning the edges of the elements with them. Figure 3 shows the results of applying this method. Depending on the shape of the face, the extracted edges may be in extremely close proximity, which has a negative effect on mesh quality. In this case, one of the edges is removed to reconcile detailed reproduction and element quality. Figure 4 shows the results of this process. This function ensures that constraint edges in close proximity do not produce meshes with a poor aspect ratio.

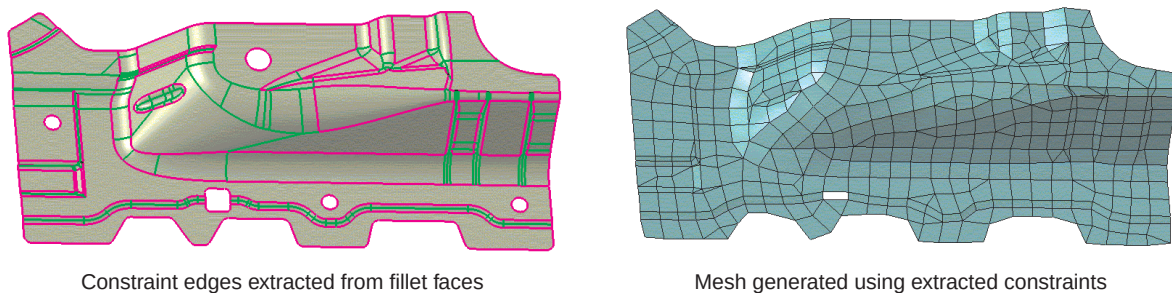


Fig. 3 Fillet representation

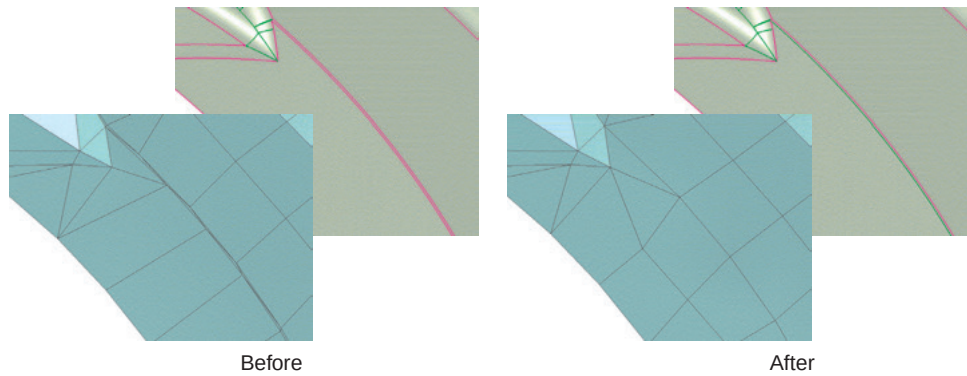


Fig. 4 Constraint simplification

5.2. Increased Detail in Reproduction of Beads

Bead shapes are generally made up of the type of faces shown in Fig. 5. Reproduction of the bead height is the important factor in reproducing beads in mesh. However, when the width of the bead is less than twice the size of the elements, only one layer of elements is used for the bead, and the height is not reproduced, as shown in Fig. 6. Even when the bead edge is set along a constraint edge to reproduce the height, there will be insufficient detail in the reproduction, as shown in Fig. 7. To enable automatic modeling of beads, an algorithm able to detect the shape of the bead from the configuration and curvature of the faces in the CAD model was developed. In addition, automatically forming a curve at the ridge of the bead (Fig. 8) and setting it along a constraint edge, enables automatic reproduction of the height of the bead in mesh, as shown in Fig. 9.

5.3. Generation of Spot Welding Pattern

The standard mesh pattern shown in Fig. 10 is generated to reproduce spot welds for stress evaluations of spot weld connections in robust analyses. The new mesher can automatically generate this standard mesh pattern at the points of connection shown in the registered connection data when generating meshes. Connection data is normally designed after completion of the connected parts, but in the new system, the meshes are formulated in advance and only the connections are remeshed as connection data is obtained.

5.4. Greater Detail in Reproduction of Convex Shapes

When a large element size is employed, the area of CAD models of convex shapes such as corners of flange not covered by the mesh increases. When there are spot welds in the model (Fig. 11), the mesh does not cover the spot weld connections, and connection elements are not generated (Fig. 12). This has a significant effect on the generation of specified mesh patterns at spot weld connections, particularly for robust analyses. A convex shape recognition algorithm was developed to ensure coverage of the mesh on these types of convex shapes. This algorithm enables the detection of convex sections consisting of edge connections with discontinuous tangents, which cannot be judged from the curvature of the edges, and detection of the necessary sections after exclusion of the curved sections of the faces. Positioning mesh nodes over the detected convex section prevents loss of the connection elements, enabling generation of a mesh that reproduces the original shape, as shown in Fig. 13.

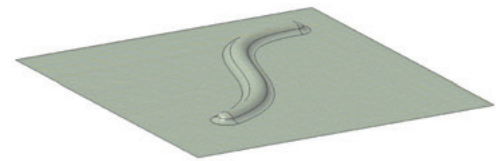


Fig. 5 Bead in CAD model

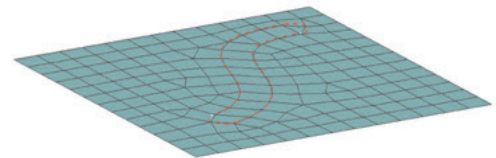


Fig. 6 Mesh generated by constraining edges only around the bead

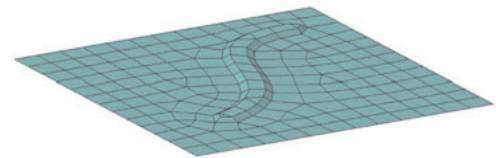


Fig. 7 Mesh generated by constraining all edges of the bead

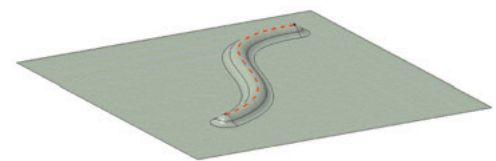


Fig. 8 Extracted ridge curve of the bead

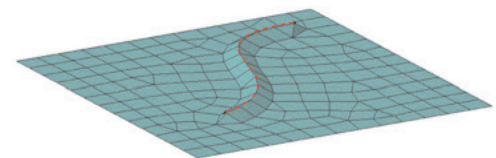


Fig. 9 Mesh generated by constraining ridge curve

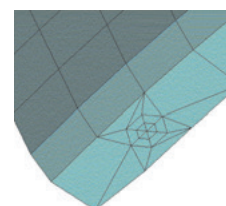


Fig. 10 Spot weld represented as hexagonal mesh pattern

5.5. Increased Element Quality

Constraint edges are programmed to increase the level of reproducibility of CAD shapes in the mesh, but when these edges are in close proximity it is difficult to generate uniform elements, and element quality declines. The new mesher detects elements that do not satisfy quality standards and either moves the nodes to the optimum position or removes them in order to increase the quality of the elements. As Fig. 14 shows, this process enables the achievement of a balance between highly detailed reproduction of shapes and element quality.

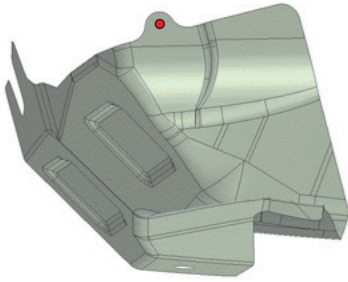


Fig. 11 Spot weld on convex flange

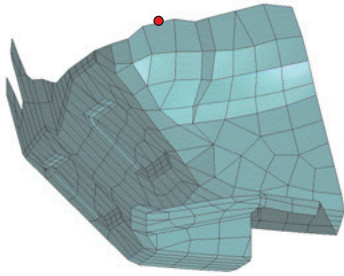


Fig. 12 Lack of elements at convex flange

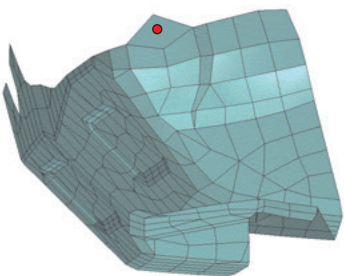


Fig. 13 Convex flange represented as mesh

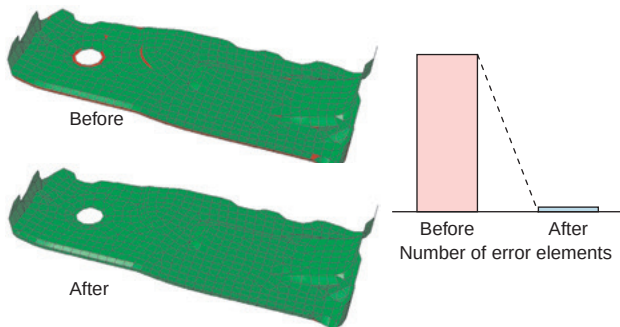


Fig. 14 Mesh refinement

6. Application of New System

The new system was used to generate the FE models used in an NV analysis. Figure 15 shows the man hours expended on model data registration and mesh generation, showing a reduction of approximately 80% against the conventional method. Figure 16 shows the results of the analysis using the models formulated by this system. The results are equivalent to those using the models formulated by the conventional method, and demonstrate that no calculation accuracy issues will be encountered in relation to models formulated by this system.

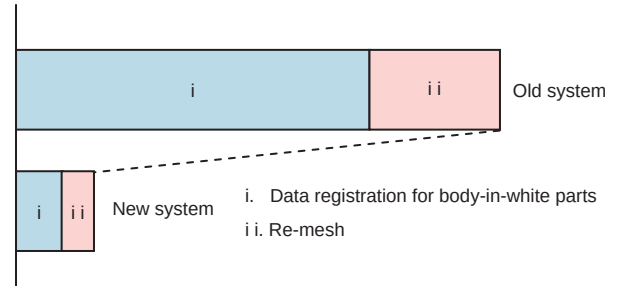


Fig. 15 Man-hour comparison

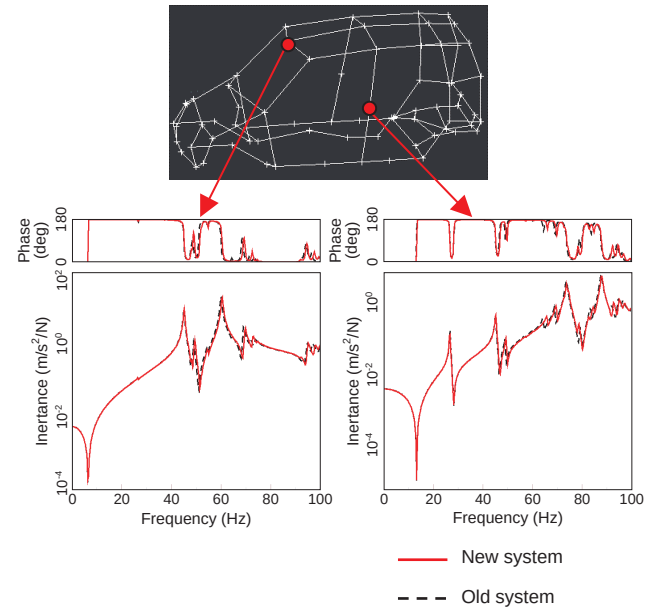


Fig. 16 Comparison of frequency response spectrum

7. Conclusion

- (1) Linking the model management system to the DPM has enabled the automated collection of parts data.
- (2) Procedures to increase the level of reproducibility in the mesh and enhance mesh quality, formerly conducted manually, have been automated.
- (3) The development of this model management system and an automatic mesher has reduced the man hours required for the generation of body-in-white FE models by approximately 80%.

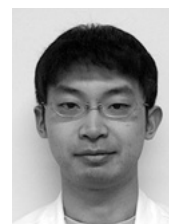
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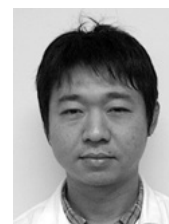
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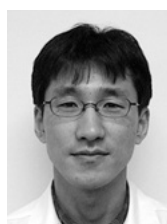
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Real-time Piston Temperature Measurement Using Electromagnetic Induction Telemetry

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ABSTRACT

As piston durability is a decisive factor in the limits of engine performance, piston temperatures are useful information for engine development. Because these temperatures are typically available only under constant-speed operating conditions, conventional material hardness measurements can no longer satisfy the diversifying requirements of piston temperature measurement. Therefore, we have developed a piston temperature measurement system that directly measures piston temperatures using a thermistor, which allows the transfer of measurement information at bottom dead center using a light, compact electromagnetic induction system. This measurement system can be applied to motorcycle engines, which are often subjected to high engine speeds, and allows measurement up to 15 300 r/min while ensuring endurance reliability of more than 10 hours.

In this study we introduce some actual examples of piston temperature changes measured by this electromagnetic induction system with the typical parameters altered when the engine is running. The phenomenon that the piston temperatures become less sensitive to changes of parameters under high engine speed conditions is thus explained by the difference in the heat transfer rate of the heat-input side and the heat-release side.

1. Introduction

The “Measurement method of piston temperature for high-speed gasoline engines”⁽¹⁾ by Takamatsu et al., which was announced by the Society of Automotive Engineers of Japan in 1998, involves transferring the signal output from a thermistor recessed in the piston at the location to be measured via a contact in the vicinity of bottom dead center, and converting it into temperature data. This measurement method has the ability to withstand 6 hours of use at a maximum engine speed of 12 000 r/min, and an error of 1% with respect to the temperature measured using a CA thermocouple.

However, because a contact was used to transfer the signal, it is difficult to measure the piston temperature in real-time when the engine is running at high speeds. Therefore, to address such issues, real-time temperature measurement technology using a contactless electromagnetic induction telemetry method was developed, enabling the data to be transferred in the vicinity of bottom dead

center without the use of a contact. This paper reports the results obtained using this technology to measure the effectiveness of the cooling oil jet, and the effect of the air-fuel ratio and ignition timing on the piston temperature, both of which are representative parameters of a running engine.

2. Principle of the Electromagnetic Induction Telemetry Method

Piston temperature measurement technology using the electromagnetic induction telemetry method was reported in 2000 by Matsui et al.⁽²⁾, and in 2002 by Inami et al.⁽³⁾ under the title “Temperature Measurement on an Operating CART Engine,” both at the Society of Automotive Engineers of Japan. Prior to this, Matsuura’s doctorate thesis⁽⁴⁾ included a detailed report^{(5), (6)} on this technology.

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The method described here is basically the same as those mentioned above. Figure 1 shows the configuration of each coil, which is based on the measurement principle used.

The transmitter and receiver coils are fixed to the cylinder block. The resonator coil, which forms a closed circuit with the thermistor, is installed on the piston so that the centers of both coils line up with each other when the piston is near bottom dead center.

When alternating current is passed through the transmitter coil, alternating current of the same frequency flows through the receiver coil as well. To reduce the effect of the relative positions of the coils on the received signal, two sets of transmitter coils and receiver coils are combined with each other.

When the piston approaches bottom dead center, current flows through the resonator coil via electromagnetic induction, causing the current flowing through the resonator coil to decrease. The magnitude of this reduction depends on the resistance of the thermistor, which indicates the piston temperature, thus enabling the piston temperature to be measured.

Figure 2 shows an actual example of the resonator coil and thermistor installed on a piston.

In this example, the recessed depth of the thermistor is 2 mm from the surface at the center of the piston crown.

3. Example of Real-time Measurement

Table 1 shows the main specifications of the engine used in these measurement examples.

Using the hardness method, it was found that the temperature of the third cylinder in a 4-cylinder engine was high, so a thermistor was recessed 2 mm into the surface at the center of the piston in

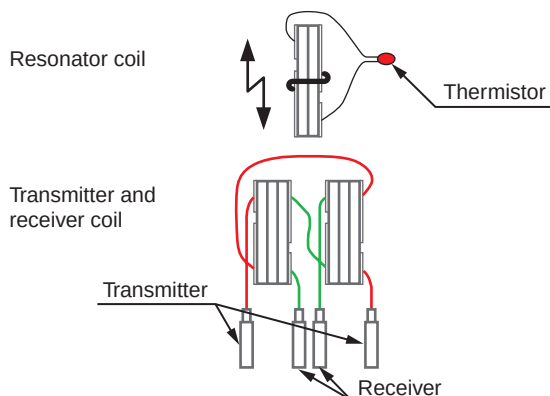


Fig. 1 Construction of electromagnetic telemetry method

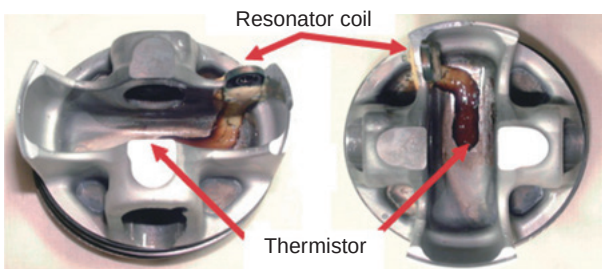


Fig. 2 Illustration of actual piston with measurement device

Table 1 Engine specifications

Engine type		Water cooled, DOHC 4-stroke 4-cylinder, gasoline
Bore × Stroke	(mm)	67 × 42.5
Displacement	(cm ³)	599
Compression ratio		12 : 1
Max. engine speed	(r/min)	15 300
Piston cooling jet	(mm)	φ0.9
Oil pressure	(kPa)	590

this cylinder, and measurement was performed at each set speed after a lapse of about 60 seconds to allow the data to stabilize. A description of the effectiveness of oil cooling of the back of the piston, fuel cooling, and changing of the ignition timing for an actual measurement example is provided below.

3.1. Piston Cooling

The ignition timing was set to minimum spark advance for best torque (MBT) for all engine speeds, the fuel-air ratio was set to the vicinity of 13 to obtain maximum output, and a comparison of the piston temperature with and without the cooling oil jet applied to the back of the piston was performed. The cooling jet had a 0.9 mm diameter nozzle and an oil pressure of 590 kPa. Figure 3 shows the temperature difference measured at this time.

Based on the measurement data, the piston cooling oil jet is very effective, demonstrating a temperature reduction at medium engine speed of 50 °C or more. Conversely, in the high speed region, the difference gradually diminished. One explanation for this is that when the engine is rotating at very high speeds, the lubricating oil at the bottom of the piston is violently agitated in the same way as when the oil jet is applied, even when a cooling jet does not exist.

3.2. Fuel Cooling

The method of lowering the piston temperature by setting the air-fuel ratio to the rich side of stoichiometric and utilizing the latent heat of evaporation of the fuel that is not related to combustion has been used empirically for many years to prevent seizure and melting of the piston due to knocking. When the vehicle is racing under severe conditions, the piston is cooled to help prolong the life of the piston for the duration of the race.

As an example, Fig. 4 shows the piston temperature at engine speeds of 11 700 and 15 000 r/min when the concentration of CO in the exhaust gas, instead of the air-fuel ratio, is changed.

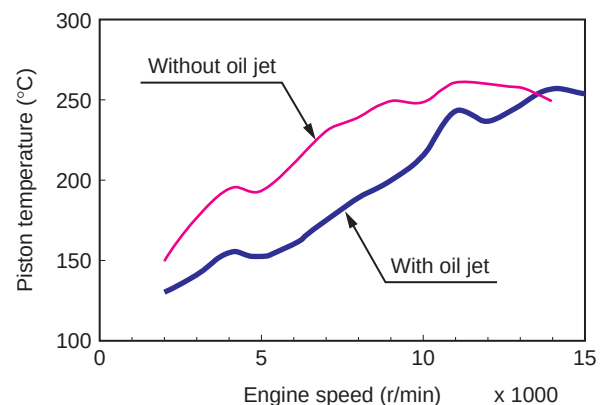


Fig. 3 Effect of piston cooling jet

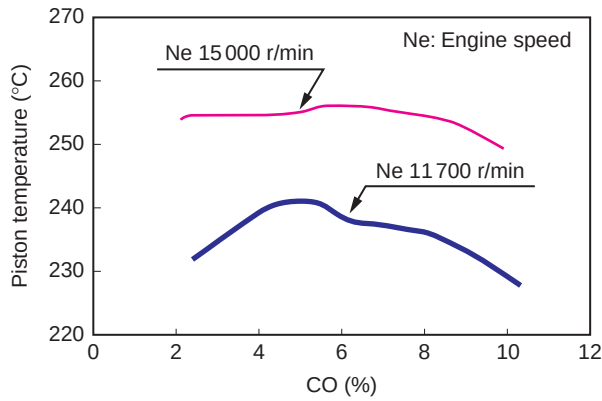


Fig. 4 Piston temperature against CO concentration

It can be seen from Fig. 4 that the temperature of the piston is greatly affected by the CO concentration when the engine speed is a relatively low 11 700 r/min, and is relatively unaffected when the engine speed reaches 15 000 r/min.

These results indicate that the conventional method based on experience, in which the CO concentration was purposely richened to ensure good piston durability, did not have a particularly large effect on the temperature at the center part of the piston at high engine speeds.

Figure 5 shows the data of Fig. 4 expressed in terms of the indicated mean effective pressure (IMEP). The piston temperature rises roughly in proportion to indicated work. When the CO concentration is set to the rich side, the piston temperature drops abruptly at an engine speed of 11 700 r/min, whereas it falls slowly at an engine speed of 15 000 r/min. On the other hand, the piston temperature remains virtually unchanged when the CO concentration is set to the lean side, as seen in the circled area.

3.3. Effect of Ignition Timing

Figure 6 shows the relationship between the maximum in-cylinder pressure and piston temperature when ignition timing is changed at engine speeds of 11 700 r/min and 15 000 r/min as described in Section 3.2.

When the ignition timing is advanced, the crank angle at which the maximum pressure is generated approaches top dead center and the maximum pressure increases. At this time, when the engine speed is an intermediate value of 11 700 r/min, the piston temperature, like the maximum pressure, rises along with the advance angle of the ignition timing. However, when the engine

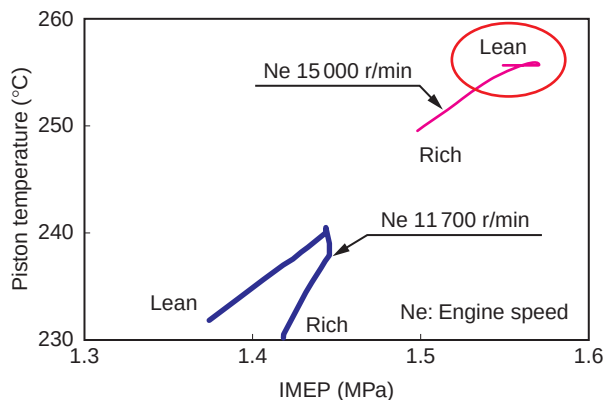


Fig. 5 Fuel cooling effect

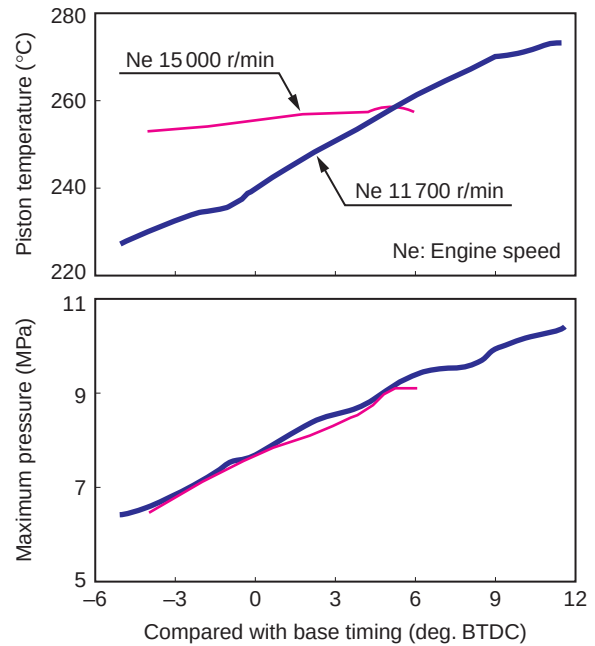


Fig. 6 Piston temperature and maximum pressure against ignition timing

speed is a high value of 15 000 r/min, the temperature is already sufficiently high when the ignition timing is retarded, and even if the ignition timing is advanced further to raise the pressure inside the cylinder, the piston temperature will not rise significantly.

Also, as shown in Fig. 7, the indicated mean effective pressure also changes along with the ignition timing. However, as demonstrated in Fig. 5, the piston temperature exerts relatively little effect on in-cylinder pressure and indicated work.

As mentioned above, the piston temperature is greatly affected by the cooling oil jet, air-fuel ratio and maximum in-cylinder pressure, or the indicated mean effective pressure determined by the ignition timing. However, this applies to an engine speed of around 11 700 r/min, and does not necessarily apply at high engine speeds of about 15 000 r/min. The question of whether this is due to an issue occurring during measurement of the piston temperature at high engine speeds, or whether it occurs unavoidably due to the balance between the heat input to the piston and the heat release from it, will be covered in the following section.

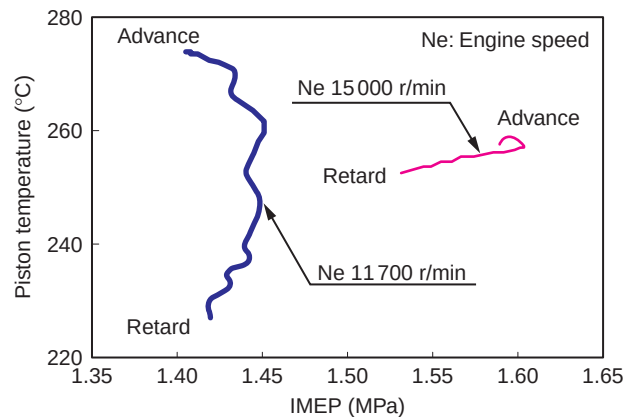


Fig. 7 Piston temperature against IMEP

4. Study of the Balance between Heat Input and Heat Release

Temperature measurement during this development project was carried out 60 seconds after the piston reached a steady-state temperature with each parameter changed. In this condition, heat transfer is also in a steady-state condition, so the total of heat input and heat release to and from the piston is zero. It is thought that when parameters such as air-fuel ratio, ignition timing and engine speed are changed, the heat transfer rate on the heat input side changes along with the in-cylinder temperature.

On the other hand, parameters such as the presence of an oil jet and engine speed can cause the heat transfer rate on the heat release side to change. Figure 8 shows a Microsoft Excel simulation of the change in the piston temperature according to the heat transfer rate difference on the heat input side. Here, the heat transfer rate on the heat release side was assumed to be constant, and the ambient temperature on the heat input side was changed. This demonstrated, of course, that the larger the heat transfer rate is on the heat input side, the more rapidly the piston temperature changes.

Because the piston temperature is determined by the balance between the input heat and the released heat, the piston temperature changes markedly with changes in air-fuel ratio and ignition timing in the medium speed region, but changes relatively slightly in the high speed region. This phenomenon can be explained as follows:

In the medium speed region, torque is increased and heat input time per heat transfer cycle is adequate compared to that of in the high speed region, so the piston temperature changes rapidly in response to changes in input heat. On the other hand, the amount of heat released is affected by the difference between ambient temperature and piston temperature, so it is ultimately determined by the heat transfer rate on the heat input side. Conversely, in the high speed region, torque is reduced and heat input time per cycle decreases, hence the piston temperature is relatively unaffected by changes on the heat input side. Therefore, it is largely determined by the heat transfer rate on the heat release side.

The piston temperature did not rise in the high speed region even when there was no oil jet. Likely, this occurs because the agitation due to the lubricating oil affects piston cooling to the same extent as oil jet cooling. This causes the heat transfer rate on the heat release side to increase and thus largely determine the piston temperature, even when the input heat changed as a result of a slight increase or decrease in engine speed.

5. Conclusion

The authors developed a technology for measuring piston

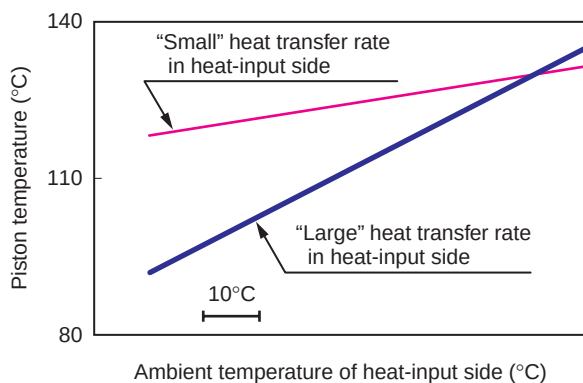


Fig. 8 Piston temperature against heat transfer rate

temperature using a contactless electromagnetic induction telemetry method and used it on motorcycle engines, which are run at high speed. As a result, it was possible to perform measurements up to 15 300 r/min reliably for a period of at least 10 hours. By using this measurement method to perform real-time measurement of piston temperature during actual running of an engine, the following results were obtained:

- (1) It was possible to quantitatively determine the effect of representative parameters, such as the piston cooling oil jet, air-fuel ratio and ignition timing, on the piston temperature.
- (2) In the medium speed range of 11 700 r/min, the temperature measured using this measurement method definitely changed according to changes in each parameter. In contrast, in the high-speed range of 15 000 r/min the temperature did not change very much when each parameter was changed. In other words, it appeared that a temperature saturation condition had been reached.
- (3) Numerical calculation was carried out to establish the cause of this phenomenon. It was found that the heat transfer rate during heat input to the piston and heat release from the piston had both changed. Consequently, the medium speed range can be considered as the range in which the heat transfer rate on the heat input side is dominant, whereas the high-speed range can be considered as the range in which the heat transfer rate on the heat release side is dominant.

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Research on Extended Expansion General-Purpose Engine

— Theoretical Analysis of Multiple Linkage System and Improvement of Thermal Efficiency —

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ABSTRACT

Research has been conducted on an extended expansion engine, using a multiple linkage system to increase the thermal efficiency of general-purpose engines. A four-jointed linkage was used between the connecting rod and the crank pin of a standard piston-crank system. The end of the linkage rotates at half the speed of the crankshaft, resulting in piston strokes unequal length in each revolution. The length of the expansion stroke is greater than that of the compression stroke, thereby providing an extended expansion cycle (Atkinson cycle).

Theoretical thermal efficiency and numerical simulations were used to determine the increase in thermal efficiency at a compression ratio of 8.5 and an expansion ratio of 12.3. The results show that this extended expansion increases indicated thermal efficiency 4.0% against a conventional engine, from 27.3 to 31.3%. In addition, the new design displayed increases of 2.0 and 2.7% in indicated thermal efficiency against naturally aspirated Miller cycle engines with early valve closure and late valve closure, respectively.

An air-cooled single-cylinder engine equipped with this mechanism was constructed and its thermal efficiency was compared in tests against a conventional engine in the same power class. The results of these tests matched the simulation results almost exactly, with the new engine displaying 4.1% higher thermal efficiency than in the conventional engine.

1. Introduction

Demands for enhanced fuel economy to prevent global warming and conserve fossil fuels are on the rise, not just for cars and motorcycles, but also in the work equipment sector. The compact general-purpose engines mounted in work equipment are typically air-cooled, often operated continuously at medium to high loads and may also use low-octane fuels in developing countries. For

these and other reasons, increasing the compression ratio can lead to knocking, so there is a limit to how much efficiency can be improved simply by increasing the compression ratio.

This research designed and manufactured a prototype extended expansion engine that uses a piston-crank system with a multiple linkage system. This system increases the expansion ratio without changing the compression ratio, a means of increasing thermal

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efficiency that is more suited to general-purpose engines. Running tests were then conducted using this engine. This paper provides a theoretical analysis and overview of this mechanism, and reports the predicted improvement of efficiency and the running test results.

2. Multiple Linkage Extended Expansion Engines

2.1. Background and History

Multiple linkage internal combustion engines have a long history, with the oldest practical engine considered to be the No. 2 engine manufactured by James Atkinson in 1886⁽¹⁾. This engine used a clever link design to set the expansion stroke longer than the compression stroke, thereby providing an extended expansion cycle (Atkinson cycle) in which the expansion ratio is greater than the compression ratio.

Various studies on multiple linkage internal combustion engines have been conducted since then⁽²⁾⁻⁽⁴⁾, but most of these studies involved variable compression ratio engines that aimed to improve fuel economy, mainly under partial load conditions in automobiles, and few studies concerned applications involving extended expansion engines⁽⁵⁾.

On the other hand, practical applications of extended expansion engines⁽⁶⁾⁻⁽⁸⁾ include the Miller-cycle engine, advocated by Ralph Miller in the 1940s. This engine used a conventional internal combustion engine with a high geometric compression ratio (expansion ratio), and eliminated the need for a complex linkage system by suppressing the amount of intake mixture by adjusting the inlet valve closure timing and lowering the actual compression ratio to less than the geometric expansion ratio. However, in practical engines, high-load applications required a supercharging system, and additional devices such as a variable valve mechanism were needed to help achieve performance over a wide range of engine speeds.

There are also cases where thermal cycles using this method have been called Atkinson cycles in the broadest sense of an extended expansion cycle⁽⁹⁾, but even though both types of thermal cycles include intake and compression strokes, they are different and the thermal efficiency also differs as described below. Therefore, this paper refers to the extended expansion cycle, which depends on the inlet valve open and close timing as the Miller cycle to distinguish it from the Atkinson cycle.

2.2. Mechanism and Principle

Table 1 lists the main specifications of the engine used in this research, and Fig. 1 shows an overview of the multiple linkage system. The specifications were determined through theoretical investigation of various linkage types in consideration of the total size including the linkage, a simple parts configuration, the target power output and the improvement of efficiency. Figure 2 shows the piston displacement relative to the crank angle of the test engine. Here, the expansion and exhaust strokes in the second half of the cycle are approximately 1.5 times the length of the intake and compression strokes in the first half of the cycle, thus indicating that an extended expansion cycle based on the Atkinson cycle can be achieved.

Figure 3 shows the main dimensions of the linkage system. The quantity of state of this multiple linkage system is expressed by the following equations, where θ_c is the crank angle.

$$x = l_4 \cos \phi_4 + l_2 \sin (\phi_1 + \alpha) + r_c \cos \theta_c \quad (1)$$

Table 1 Engine specifications

Engine type		Air-cooled 4 stroke OHC single-cylinder spark-ignition
Bore (mm)		64
Stroke (mm)	Intake and compression	42.1
	Expansion and exhaust	63.3
Displacement (cm ³)	Intake and compression	135
	Expansion and exhaust	204
Compression ratio		8.5
Expansion ratio		12.3
Fuel		Gasoline
Air-fuel ratio		12.5
Gross Maximum power (kW/rpm)		3.4 / 4000

$$l_4 \sin \phi_4 + \delta = l_2 \cos (\phi_1 + \alpha) + r_c \sin \theta_c \quad (2)$$

$$l_5 - r_c \cos \theta_c + r_p \sin (\theta_p + \gamma) = l_1 \cos \phi_1 + l_3 \cos \phi_3 \quad (3)$$

$$l_6 + r_p \cos (\theta_p + \gamma) + l_3 \sin \phi_3 = l_1 \sin \phi_1 + r_c \cos \theta_c \quad (4)$$

Assuming the following relationship,

$$\theta_p = \eta \theta_c + \gamma \quad (5)$$

The linkage system has a degree of freedom of "1," so the piston position relative to the crank angle can be determined freely, assuring its operation as a linkage system. Equation (5) represents the pivot shaft rotation driven by the crankshaft, and yields the value $\eta = -1/2$ for 1/2 gear reduction driving. This provides a phase difference with a period of $\theta_c = 720^\circ$ between the crankshaft and the pivot shaft, which causes the length of each piston stroke to differ.

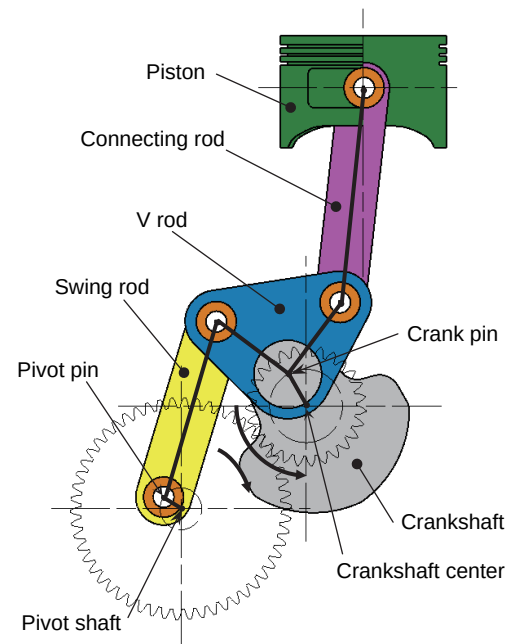


Fig. 1 Configuration of multi-link system

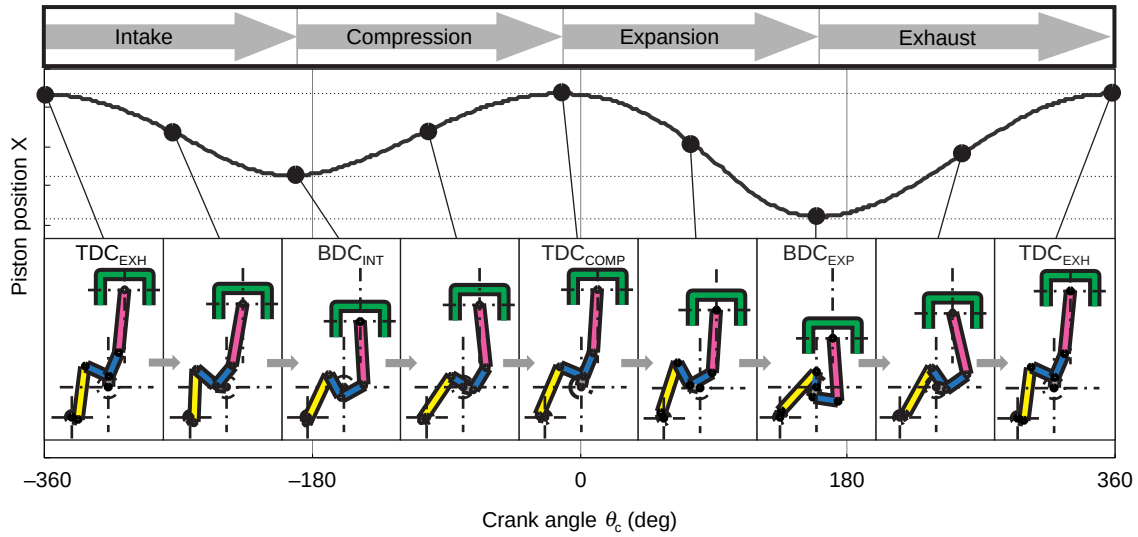


Fig. 2 Piston movement through one cycle

In addition, the initial phase difference γ is related to the top and bottom dead center positions of each stroke (Fig. 4). By applying the appropriate γ , the same piston position can be used for the compression top dead center position and the exhaust top dead center position, making it possible to obtain an extended expansion cycle where the expansion and exhaust strokes are

longer than the intake and compression strokes and the expansion ratio ε_{EXP} is larger than the compression ratio ε_{COMP} .

3. Predicted Performance

The increase in thermal efficiency provided by the extended expansion cycle was predicted by the following method.

3.1. Theoretical Thermal Efficiency

Equation (6) expresses the theoretical thermal efficiency for a general Otto cycle.

$$\eta_{th} = 1 - \frac{1}{\varepsilon_c^{\kappa-1}} \quad (6)$$

ε_c : Compression ratio (= Expansion ratio)

κ : Specific heat ratio (= Cp/Cv)

Expanding this to accommodate an extended expansion cycle with a constant pressure interval in the compression stroke, the theoretical thermal efficiency becomes as shown in Eq. (7) (Fig. 5).

$$\eta_{th} = 1 - \frac{1}{\varepsilon_e^{\kappa-1}} - \left\{ \frac{\varepsilon_e}{\varepsilon_c} (\kappa - 1) - \kappa + \left(\frac{\varepsilon_c}{\varepsilon_e} \right)^{\kappa-1} \right\} \frac{C_v T_0}{Q_{in}} \quad (7)$$

ε_e : Expansion ratio ($> \varepsilon_c$)

C_v : Specific heat at constant volume

T_0 : Compression start temperature

Q_{in} : Additional heat at constant volume

Figure 6 shows the theoretical thermal efficiency when the compression ratio is fixed and only the expansion ratio is increased, in reference to the compression ratio of 8.5 for a typical general-purpose engine. This shows that the increased expansion ratio of the test engine has the potential to improve the thermal efficiency by approximately 10% or more.

3.2. Numerical Simulation

The above-mentioned theoretical thermal cycle does not take into account factors such as the pumping loss involved with intake and exhaust or the heat loss from the cylinder wall during the expansion and compression strokes, so it differs from the actual

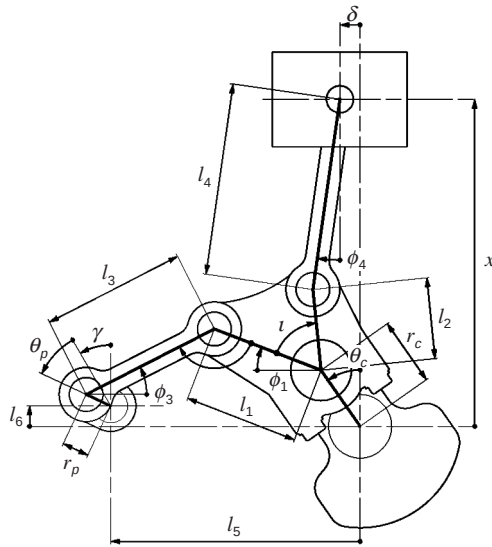


Fig. 3 Characteristic dimensions of linkage system

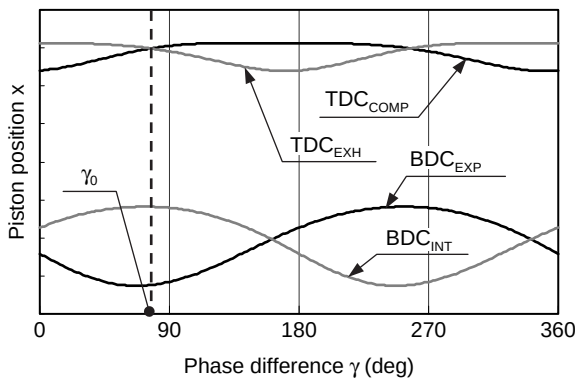


Fig. 4 Relationship between phase difference and each dead center

thermal cycle. To more accurately estimate the increase in efficiency provided by the extended expansion cycle, thermal cycle simulations⁽¹⁰⁾ were conducted using a numerical calculation model that took into account the intake and exhaust air flow as a quasi-one-dimensional flow, and the combustion inside the cylinder and heat transfer to the cylinder walls as a non-dimensional vessel. The functions of existing in-house software were expanded to perform the calculations, allowing the physical levels in each stroke to be calculated based on the operation of a multiple linkage type extended expansion engine.

Table 2 shows the specifications and operating conditions of the conventional engines used for comparison. The Type A conventional engine has the same intake stroke volume as the test engine, and the Type B conventional engine has the same calculated WOT output power as the test engine.

The heat released by combustion was calculated by approximating the time history of the mass fraction burned, based on actual measurement results using a Wiebe function⁽¹¹⁾, and all of the conventional and test engines were assumed to release the same amount of heat using the compression top dead center state as a reference. The total amount of heat supplied to the thermal cycle used the value calculated from the fuel combustion level of the test engine at 100% load and was the same for all specifications. The heat loss was calculated using the heat transfer coefficient

Table 2 Comparison with conventional engine specifications

	Extended expansion	Conventional	
		TYPE A	TYPE B
Bore x stroke (mm)	64x42.1 / 63.3	64x42.1	64x47
Compression ratio	8.5	8.5	
Expansion ratio	12.3	8.5	
Engine revolutions (rpm)		3060	
Air fuel ratio		12.5	
Supplied heat (J/cycle)		456	
Combustion efficiency		0.725	

Table 3 Calculated work and loss of each stroke

		Extended expansion	Conventional	
			TYPE A	TYPE B
Work or loss of each stroke (J/stroke)	Intake	-3.2	-3.9	-5.3
	Compression	-31.4	-32.9	-31.8
	Expansion	178.5	163.3	162.4
	Exhaust	-1.1	-0.7	-0.8
Heat loss from cylinder wall (J/cycle)		91.3	82.1	84.9
Indicated efficiency (net)		31.3%	27.6%	27.3%

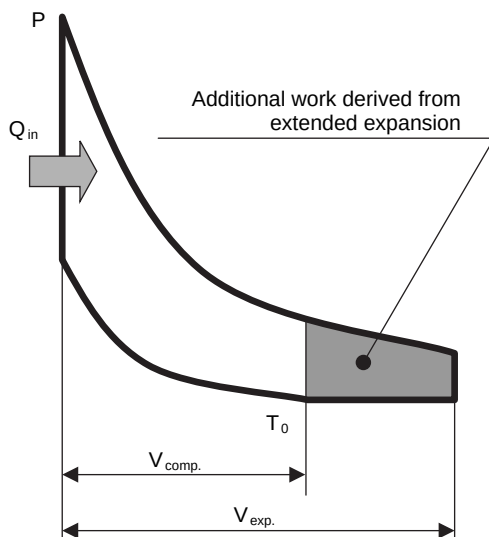


Fig. 5 Extended expansion thermal cycle

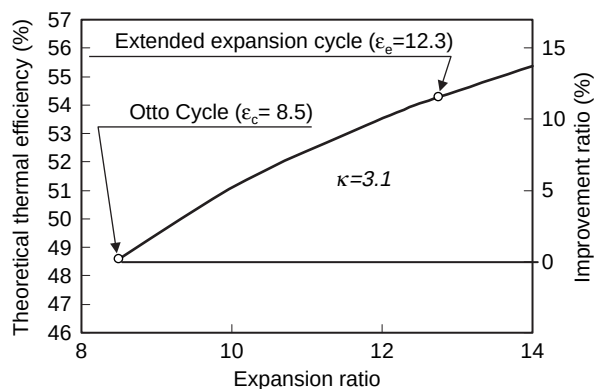


Fig. 6 Theoretical efficiency of extended expansion cycle

formulated by Woschini, et al.⁽¹²⁾, with the same cylinder wall temperature and other boundary conditions used for all engines. The air/fuel ratio, combustion efficiency, valve open and close timing and other operating conditions were also the same. In addition, the intake pressure loss used the value equivalent to when the butterfly valve is completely open to reduce differences in the choking pressure loss when opening the throttle.

The calculation results for each specification are described below. A comparison of the P-V diagrams for the test engine and the Type A engine (Fig. 7) shows that work derived from extended expansion is added at the end of the expansion stroke for the test engine due to the cylinder pressure during the extended expansion stroke. In addition, a slight increase is seen in the maximum cylinder

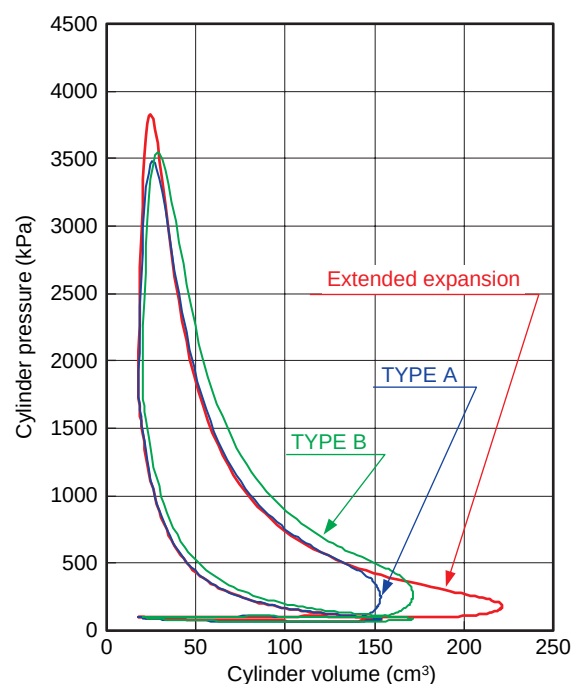


Fig. 7 Calculated P-V diagrams

pressure, but this is because the piston speed of the test engine near the compression top dead center is slower than that of the conventional engine.

To view a breakdown of the increase in efficiency, Table 3 shows the calculated values for the indicated output power of each stroke and the heat loss from the cylinder wall, and Fig. 8 shows the heat balance of each engine relative to the total heat supplied to engine. Note that all subsequent indicated output power values are the indicated output power (net) including the intake and exhaust strokes.

Based on the above results, the test engine was found to have the following characteristics as compared to the Type A and Type B engines.

- (1) The expansion stroke is longer, so the heat loss from the cylinder wall increases.
- (2) The exhaust stroke is longer, so the exhaust pump loss increases, although the absolute value is small.
- (3) The exhaust heat loss decreases due to the increase in the work derived from extended expansion.

These characteristics give the test engine an indicated thermal efficiency of 31.3%, which is 3.7 percentage points higher than the 27.6% for the Type A engine, and 4.0 points higher than the 27.3% for the Type B engine, confirming that the extended expansion cycle of the test engine can increase thermal efficiency, even in a more realistic thermal cycle that includes the intake and exhaust strokes and the heat loss from the cylinder wall.

3.3. Comparison with the Naturally Aspirated Miller Cycle

As mentioned above, most practical extended expansion engines are Miller-cycle engines that increase the expansion ratio through a combination of throttling the intake mixture air flow by varying the inlet valve close timing, and using a combustion chamber that has a heightened geometric compression ratio. This study used the above-mentioned numerical simulations to compare the indicated efficiency of the test engine's Atkinson cycle with those of single cylinder, naturally aspirated Miller cycle engines that use early and late inlet valve closure. Table 4 shows the specifications of the Miller-cycle engines used for the calculations.

The stroke volume of the Miller-cycle engines used for comparison was the same as the expansion stroke volume of the test engine. In addition, the expansion specifications of both engine

Table 4 Comparison with Miller cycle engine specifications

	Extended expansion	Miller cycle	
		TYPE C	TYPE D
Bore x stroke (mm)	64x42.1 /63.3	64x63.3	
Combustion chamber volume (cm³)	18		
Geometrical compression	8.5	12.3	
Expansion ratio	12.3	12.3	
Intake valve close	STD	51deg early closure	78.5deg late closure
Engine revolution (rpm)	3060		
Air fuel ratio	12.5		
Supplied heat (J/cycle)	456		
Combustion efficiency	0.725		

Table 5 Calculated work and loss of each stroke

		Extended expansion	Miller cycle	
			TYPE C	TYPE D
Work or loss of each stroke (J/stroke)	Intake	-3.2	-8.1	-7.8
	Compression	-31.4	-36.5	-37.9
	Expansion	178.5	179.2	177.1
	Exhaust	-1.1	-1.2	-1.2
Heat loss from cylinder wall (J/cycle)		91.3	94.6	95.8
Indicated efficiency (net)		31.3%	29.3%	28.6%

types were aligned by matching the geometric compression ratio of the Miller-cycle engines to the expansion ratio of the test engine. In the early inlet valve closure Miller cycle (Type C), the throttle was set to fully open and the valve open period and lift amount were proportionally reduced with the intake valve open timing fixed. In addition, in the late inlet valve closure Miller cycle (Type D), the throttle was set to fully open and the valve open period at maximum lift was extended with both the inlet valve opening timing and the maximum lift amount fixed. Operation was then compared using the same intake mixture air flow, that is to say the same feed heat amount as that of the test engine (Fig. 9).

Table 5 shows the calculation results for the indicated output power of each stroke, the heat loss from the cylinder wall, and the indicated thermal efficiency obtained by the simulation for each specification. The effects of extended expansion are clearly evident, with the expansion stroke work greatly exceeding that of the previous Type A and Type B conventional engines for all specifications. However, Type C and Type D engines both showed greater loss in the intake and compression strokes compared to that in the test engine. This is caused by the following factors:

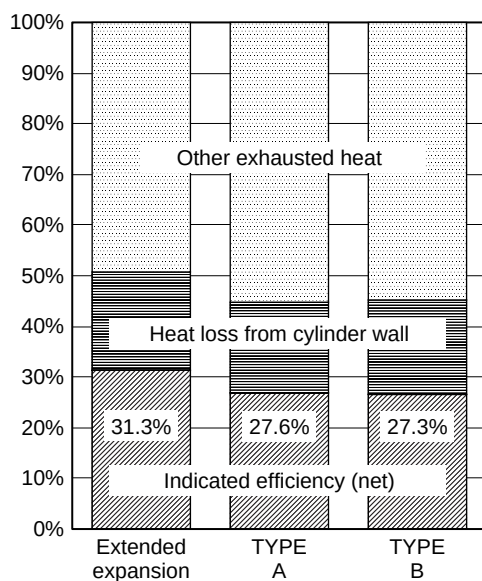


Fig. 8 Heat balance diagram

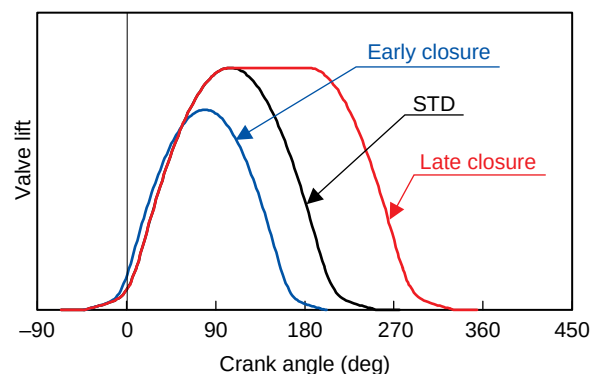


Fig. 9 Inlet valve lift curve

- (1) Type C and Type D engines have longer intake and compression stroke lengths.
- (2) When using a realistic valve lift curve, the inlet valve produces choking pressure loss in the low lift interval near the open and close timing.
- (3) For this reason, the pump loss increases during the intake period in the early closure cycle (Type C).
- (4) Similarly, pump loss occurs during the compression stroke due to intake blow-back in the late closure cycle (Type D).

The single cylinder, naturally aspirated Miller cycle engine increased the indicated thermal efficiency by 1.0 to 1.7 percentage points compared to the conventional engines, but the test engine increased the thermal efficiency even further. These numerical simulation results confirmed the advantage of the ideal extended expansion cycle (Atkinson cycle) achieved by the newly developed mechanism.

4. Test Results

Based on the results of the above investigations, a prototype extended expansion engine using a multiple linkage system was manufactured to verify the linkage system operation and assess performance during actual operation. These test results are described below.

4.1. Verification of Predicted Effects

The stroke work was calculated by measuring the cylinder's internal pressure during operation using a water-cooled piezoelectric pressure sensor mounted on the cylinder head of the prototype engine. Figure 10 compares the P-V diagram at 100% load with the above numerical simulation results. The test results show a slightly lower value for the maximum cylinder pressure, and some pressure change differences in the compression and expansion strokes can also be seen. These differences were determined to be due to the need for increased accuracy in the

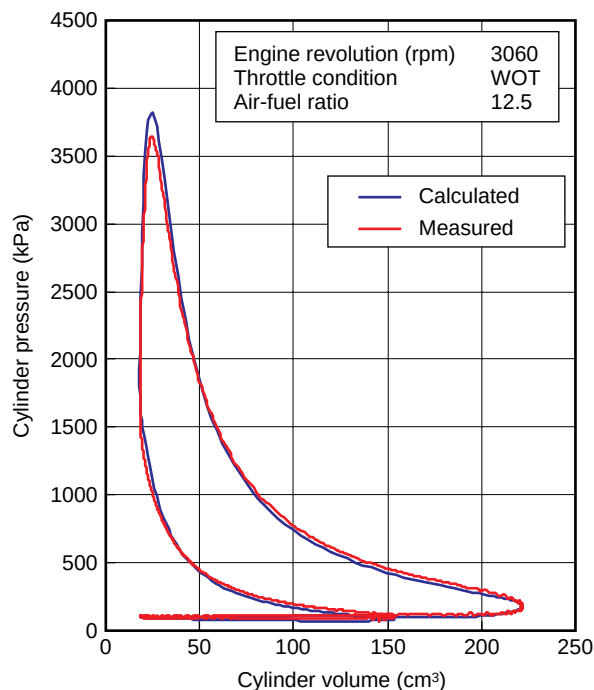


Fig. 10 Measured and calculated P-V diagrams

Table 6 Comparison of calculated and measured work

Work or loss of each stroke (J/stroke)	Calculated		Measured
	Intake	-3.2	
	Compression	-31.4	
	Expansion	178.5	
	Exhaust	-1.1	-1.5
Indicated efficiency (net)		31.3%	31.2%

Table 7 Test engine specifications

	Extended expansion	Baseline Otto cycle
Bore x stroke (mm)	64x42.1 / 63.3	68x45
Compression ratio	8.5	8.5
Expansion ratio	12.3	
Ignition timing (deg BTDC)	24	25
	(MBT on WOT)	
Air fuel ratio	12.5	
Gross maximum power (kW/rpm)	3.2/3060	

heat-generation model used for calculations in the former case, and in the heat-loss model in the latter case. However, the indicated output power and the indicated thermal efficiency calculated from the P-V diagram matched almost exactly (Table 6), thereby confirming the validity of the method for predicting the performance criteria described above.

4.2. Verification of Increased Efficiency

These test results were compared with those of a conventional engine in the same power class to verify the increase in thermal efficiency. A Honda air-cooled, single cylinder general-purpose engine with approximately the same gross maximum power was selected as the engine for comparison. Table 7 shows the engine specifications.

Figure 11 compares the indicated efficiency for each load at the same engine speed. This shows that the efficiency is increased over the entire load range, and the effects are even greater at higher loads. The indicated efficiency at 100% load is 4.1 percentage points

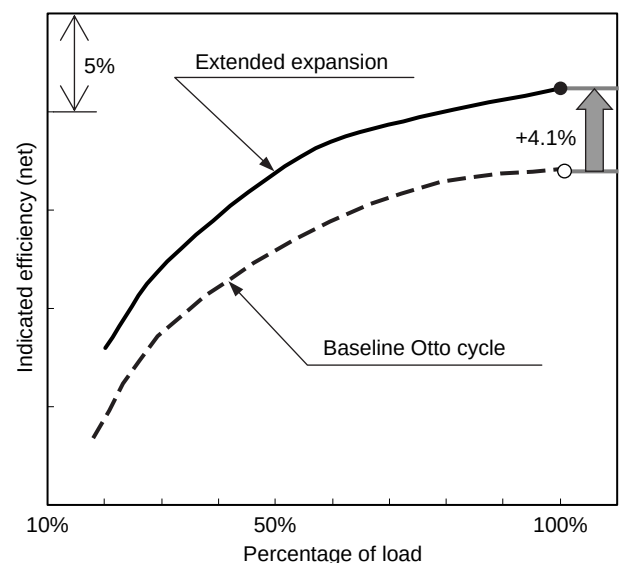


Fig. 11 Improvement of indicated efficiency (net)

higher than the comparison engine, which corresponds to an approximate 15% improvement of fuel economy as compared to the conventional engine. These results confirmed through actual engine operation the effects of applying this technology to general-purpose engines, which require improved efficiency in the medium to high load range.

5. Conclusion

Research was conducted on an extended expansion engine using a multiple linkage system with the aim of increasing general-purpose engine efficiency, and the following results were obtained.

- (1) Numerical simulations showed that an extended expansion engine with a compression ratio of 8.5 and an expansion ratio of 12.3 increases the indicated thermal efficiency by 3.7 percentage points compared to a conventional engine with the same intake volume, and by 4.0 percentage points compared to a conventional engine with the same power.
- (2) This extended expansion engine also increased the indicated thermal efficiency by 2.0 and 2.7 percentage points over naturally aspirated Miller-cycle engines having equivalent expansion specifications with early and late inlet valve closures, respectively.
- (3) The indicated thermal efficiency obtained from operation using a test air-cooled single cylinder engine was almost exactly the same as that obtained by the above simulation. The new engine increased the measured indicated thermal efficiency by 4.1 percentage points over a conventional general-purpose engine in the same power class, which corresponds to improvements in fuel economy of approximately 15%.

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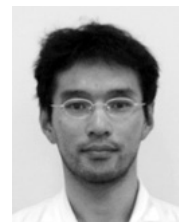
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Achievement of Enhanced Fuel Economy via i-VTEC Intake Valve Closure Delay Mechanism

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ABSTRACT

The conventional VTEC system has been modified with a mechanism enabling delayed closure of the intake valves as a new method of increasing fuel economy. The major characteristic of the new engine is the achievement of a balance between low fuel consumption and high power by means of switching between a delayed closure cam with delayed intake valve closure timing and a high output cam, depending on engine load. The intake valve closure delay mechanism employs the delayed closure cam when the engine is under low load in order to reduce pumping loss. The high output cam is employed when the engine is under high load, in order to produce high power. A variety of driving conditions were considered and in-cylinder flow simulations were used in setting the valve timing of the delayed closure cam. The delayed closure cam achieves a 6% reduction in engine fuel consumption compared to the high output cam. The application of the new VTEC mechanism has enabled the achievement of 17.0 km/L fuel economy in the Japanese 10-15 mode, representing a 5% increase on the level as stipulated by domestic 2010 fuel economy standards.

1. Introduction

Against a background of increasing concern about the environment, enhancing the environmental performance of automotive engines by increasing fuel economy and reducing emissions has become an important issue in the engine development process. CO₂ emissions can be reduced and fuel economy increased by increasing theoretical thermal efficiency, reducing pumping loss and reducing friction. Honda has worked to enhance environmental performance by developing lean-burn and direct cylinder injection engines⁽¹⁾ and by combining its VTEC system with a mechanism for continuous camshaft phase variation to produce a 4-cylinder i-VTEC engine⁽²⁾. Honda has also developed a V6 i-VTEC⁽³⁾ engine fitted with a variable cylinder system allowing cylinder idling.

Delaying closure of the intake valves to increase the expansion ratio to greater than the compression ratio is known to be one approach to increasing thermal efficiency. This approach is

discussed in the technical literature as the Atkinson and Miller cycles⁽⁴⁾⁻⁽⁶⁾, and has been applied in a number of engines^{(7),(8)}.

As indicated, delayed closure of the intake valves is an effective method of increasing fuel economy. However, the longer valve closure is delayed to increase efficiency, the greater the decline in charging efficiency and consequent decline in power at high-load, high-engine speeds necessitating the use of a turbocharger or a hybrid mechanism.

Honda has developed an innovation on this design in the R18A 1.8L 4-cylinder gasoline engine. The R18A engine has been provided with an i-VTEC mechanism that switches between normal and delayed closure of the intake valves to balance low fuel consumption, high power and low exhaust emissions⁽⁹⁾. In the development process, the effect of the intake valves' delayed closure on pumping loss and combustion stability and the mechanism's effectiveness in enhancing fuel consumption were examined.

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2. Intake Valve Delayed Closure

2.1. Effect on Thermal Efficiency

Figure 1 shows a comparison of the P-V diagrams of a conventional (Otto cycle) engine and an engine using an intake valve delayed closure mechanism.

In conventional engines, the compression and expansion ratios are almost identical. When closure of the intake valves is delayed, however, an over-expanded cycle is produced, in which the actual air-fuel mixture compression ratio in the intake stroke (the effective compression ratio) and the expansion ratio in the expansion stroke differ. An engine using this cycle has a higher thermal efficiency than Otto cycle engines do at the same compression ratio.

The thermal efficiency of the Otto cycle and over-expanded cycle engines as shown in Fig. 1 can be expressed as follows:

Otto cycle engine:

$$\eta_o = 1 - \frac{1}{\epsilon_o^{\kappa-1}} \quad (1)$$

Where: Compression ratio $\epsilon_o = V_{o1}/V_{o2} = V_{o4}/V_{o3}$
Specific heat capacity ratio κ

Over-expanded cycle engine:

$$\eta_e = 1 - \frac{\beta \epsilon_e^{\gamma-1} \epsilon_e^{\kappa} + \kappa (\epsilon_e' - \epsilon_e) - \epsilon_e'}{\epsilon_e^{\kappa(\beta-1)}} \quad (2)$$

Where: Effective compression ratio $\epsilon_e = V_{e1}/V_{e2}$
Expansion ratio $\epsilon_e' = V_{e4}/V_{e3}$
Degree of explosion $\beta = P_{e3}/P_{e2}$

Figure 2 shows the results of thermal efficiency calculations using Eqs. (1) and (2) when the expansion ratio is set at 10.5. The effective compression ratio varies in the case of the over-expanded cycle engine and the compression ratio varies in the case of the Otto cycle engine. The degree of explosion was set at 4.5 on the basis of engine tests. These results indicate that the expansion ratio determines changes in thermal efficiency. At the same compression ratio, an engine with a high expansion ratio would display superior fuel economy. In addition, the lower effective compression ratio in the over-expanded cycle reduces the temperature in the cylinders during compression, which could be expected to reduce heat loss and prevent knocking.

2.2. Effect on Pumping Loss

When a gasoline engine is under low loads, the throttle valve is normally restricted to reduce air flow. When the engine is operating in this state, air is taken in under negative pressure, causing pumping loss. Figure 3 shows a diagrammatic representation of the

difference in pumping loss between a conventional engine and an engine using delayed closure of the intake valves. As shown in (b), using a delayed closure cam for the valve closure causes some of the air that is taken in during the intake stroke to reenter the intake manifold in the first half of the compression stroke, changing the actual compression commencement point from A to A'. This enables air flow to be adjusted with less restriction of the throttle valve, which reduces pumping loss.

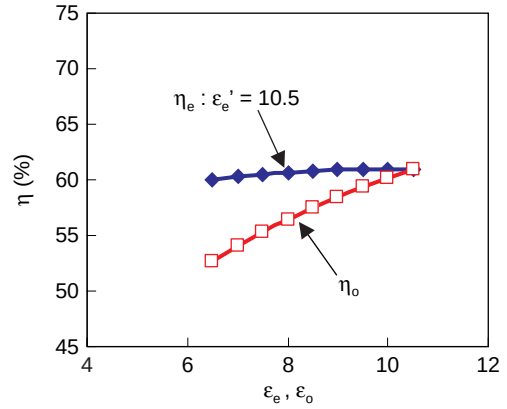


Fig. 2 Influence of ϵ_e and ϵ_o on cycle efficiency

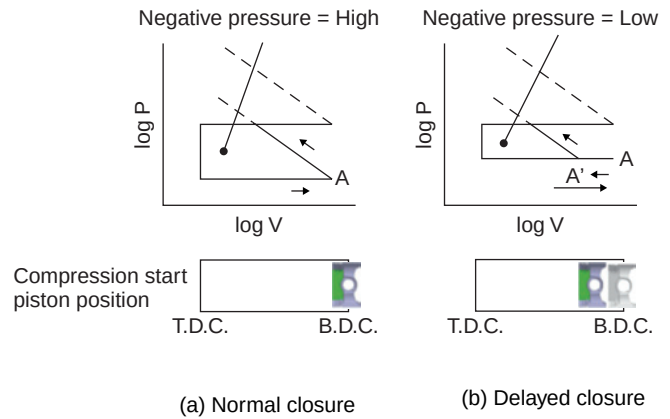


Fig. 3 Influence of intake valve closure timing on pumping loss

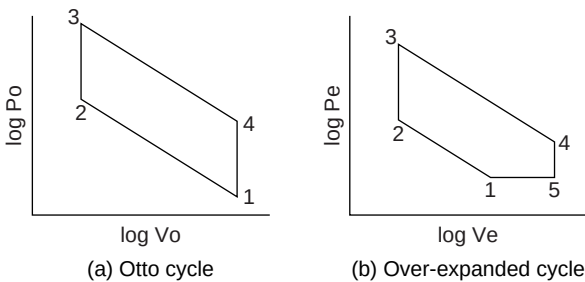


Fig. 1 Comparison of P-V diagram between Otto cycle and Over-expanded cycle

3. Test Equipment

Table 1 shows the main specifications of the R18A i-VTEC test engine.

The test engine used two intake valves, one of which was operated by switching between the high output and delayed closure cams. Different delayed closure cams with a variety of valve closure timings were used in the tests. Valve lift for intake valve opening and closing timing was set at 1 mm. Figure 4 shows the various intake valve closure (I.V.C.) timings used for the high output and delayed closure cams and the cam profiles.

Table 1 Engine specifications

Type	R18A
Cylinder configuration	In-line 4-cylinder
Bore × Stroke (mm)	81 × 87.3
Displacement (cm ³)	1799
Compression ratio	10.5:1
Valve train	SOHC i-VTEC Inlet delayed closure
Number of valves	4 per cylinder
Valve diameter (mm) In./Ex.	32/26
Cylinder offset (mm)	12
Intake manifold	Variable intake system
Gasoline	Regular (RON91)
Max. power (kW/rpm)	103/6300
Max. torque (Nm/rpm)	174/4300

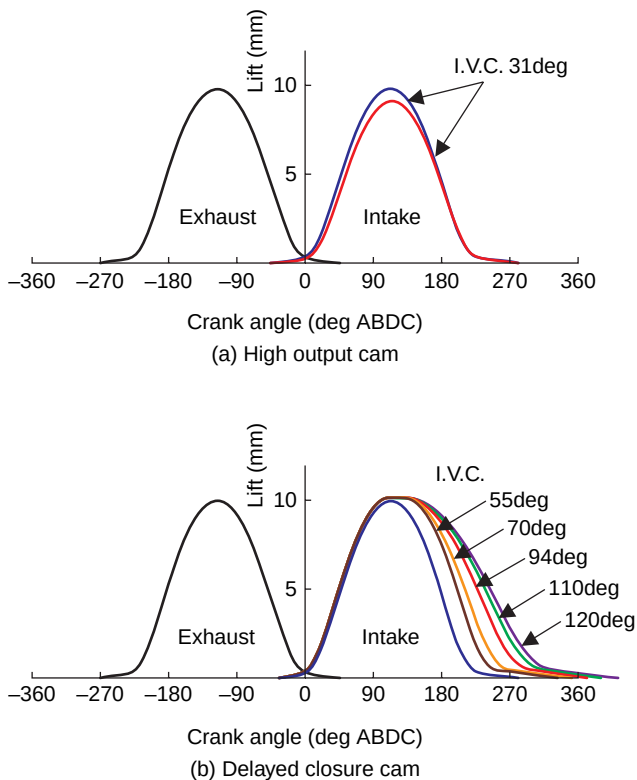


Fig. 4 Applied I.V.C and cam profiles

The pressure waveform is based on samples taken at every 1 deg of crank angle using a pressure sensor (Kistler 6052B) and a charge amp (Kistler 5011B).

4. Results and Observations

4.1. Effect of Delayed Closure Cam on Pumping Loss

Figure 5 shows a pressure diagram for the high output and delayed closure cams at the same output. The I.V.C. was 31 deg (ABDC) for the high output cam and 94 deg for the delayed closure

cam. In the case of the delayed closure cam, air was returned to the intake manifold in the first half of the compression stroke, delaying the commencement of compression. In addition, the average negative pressure during the intake stroke was closer to atmospheric pressure, and pumping loss was reduced.

Figure 6 shows the ratio of the volumetric efficiency (η_v) of the high output and delayed closure cams at wide-open throttle, defined as the intake air reduction rate. When the delayed closure cam was operating, 25-35% of the intake air was returned to the intake manifold, reducing the apparent displacement. Switching between cams as required via the i-VTEC mechanism balanced high power during high loads with reduced fuel consumption during low loads.

4.2. Effect of Delayed Closure Cam on Thermal Efficiency

Figure 7 shows the effect on the indicated specific fuel consumption (ISFC) rates of I.V.C. of 31, 94 and 120 deg. A gross figure that includes a fraction for pumping loss has been used for the indicated mean effective pressure (IMEP)⁽¹⁰⁾.

Because the compression ratio of the test engine was set at 10.5, the ISFC values do not change significantly with changes in the I.V.C., but there is approximately a 0.5% reduction in the case of the delayed closure cam. This is believed to be because the change

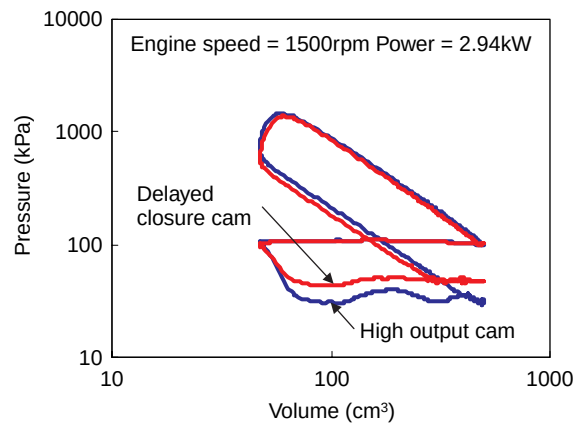


Fig. 5 Effect of delayed closure inlet valve timing for reduction of pumping loss

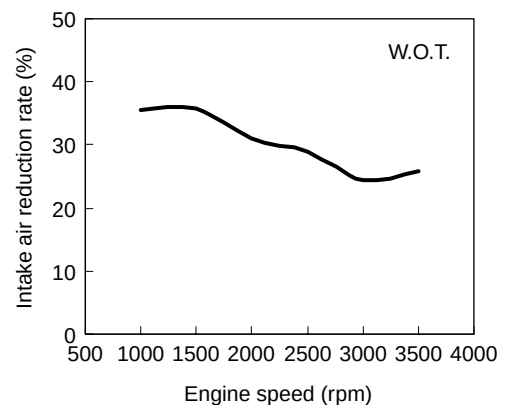


Fig. 6 Intake air reduction rate at inlet valve delayed closure

in the effective compression ratio has reduced the temperature in the cylinders, thus reducing heat loss to the cylinder walls. When the IMEP is increased, at an I.V.C. of 31 deg there is an area in which knocking causes an ignition timing lag and the ISFC increases. At an I.V.C. of 94 deg, however, knocking does not occur and there is no increase in the ISFC, even at wide-open throttle. However, the IMEP declines at wide open throttle with increasing delay in the I.V.C.

4.3. Effectiveness of Delayed Closure Cam in Enhancing Fuel Economy

Figure 8 shows the effect of the I.V.C. on pumping loss, brake-specific fuel consumption (BSFC) and co-variation of the IMEP.

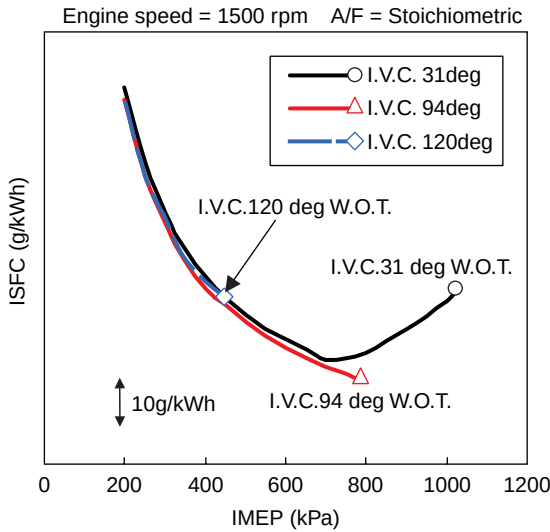


Fig. 7 Influence of I.V.C. on ISFC

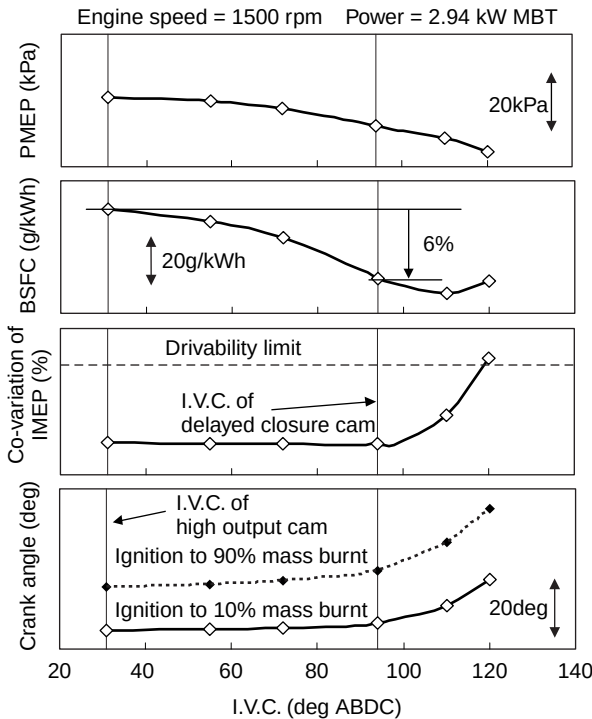


Fig. 8 Influence of I.V.C. on PMEP, BSFC, characteristics of combustion

Pumping loss declines and BSFC decreases with increasing delay in the I.V.C., but after 110 deg there is no further decrease in BSFC, despite the fact that pumping loss continues to decline. This is thought to be because excessive delay in the I.V.C. causes in-cylinder flow attenuation and a reduction in the temperature in the cylinders, resulting in slow initial post-ignition combustion, which cancels the effect of reduced pumping loss⁽¹¹⁾.

Figure 9 shows the results of a simulation of the in-cylinder turbulence energy and fuel distribution in the second half of the compression stroke. Comparing the energy of turbulence around the spark plug for the high output cam (I.V.C. 31 deg) and the delayed closure cam at an I.V.C. of 94 deg, it is clear that the area of highest energy is fairly narrow at 94 deg. At 110 deg, the energy of the turbulence around the plug declines markedly, and the rich area of the fuel distribution is also reduced in size. This is thought to be the cause of rapid rise in IMEP co-variation, which expresses combustion stability during initial and main combustion. These results indicate that an excessive delay in intake valve closure reduces the energy of in-cylinder turbulence at ignition and produces a lean fuel distribution, causing a change in ignition.

Figure 10 shows the rate of reduction of BSFC for the high output cam (I.V.C. 31 deg) when the I.V.C. is varied under load conditions corresponding to those experienced under standard cruising speeds. At 40 km/h, the reduction in BSFC is maximized at an I.V.C. of 98 deg, but at 80 km/h and above, BSFC does not reach bottom, even when the I.V.C. is delayed to 120 deg. This is thought to be because the decline in in-cylinder turbulence and in-cylinder temperature when the I.V.C. is delayed is greater at low vehicle speeds (low engine speeds, low loads) than at high vehicle speeds (high engine speeds, high loads). The I.V.C. for optimum

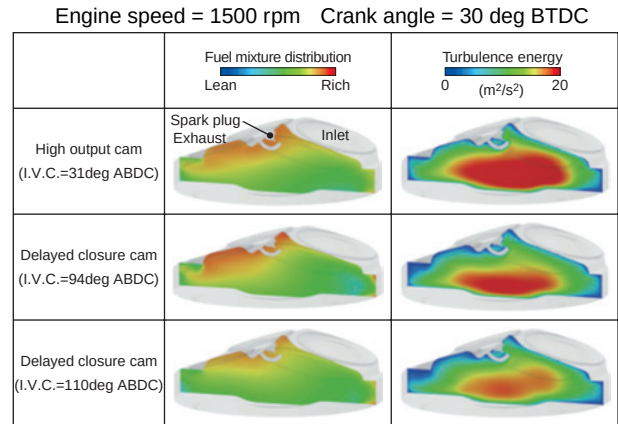


Fig. 9 Comparison of turbulence energy and fuel mixture distribution

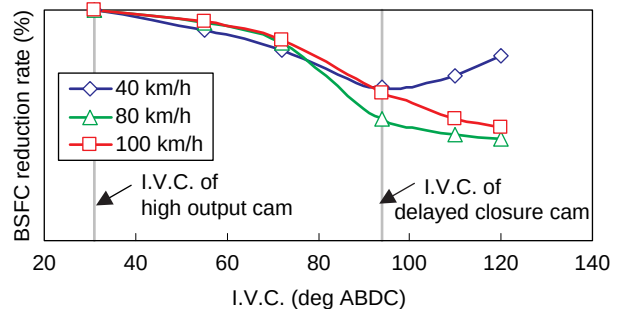


Fig. 10 BSFC reduction rate at cruise condition

BSFC changes as load conditions change. An I.V.C. of 94 deg was selected for the delayed closure cam, taking fuel economy benefits and combustion stability into consideration. This reduced BSFC by 6% against that of the high output cam while ensuring identical combustion stability.

Figure 11 shows co-variation in the IMEP, NO_x emissions and BSFC against the EGR rate. The reduction in fuel consumption with the EGR is higher for the high output cam, but the performance of the delayed closure cam remains superior even when the EGR is introduced.

In terms of combustion characteristics in relation to the EGR rate, the EGR rate limit for stable combustion is lower for the delayed closure cam than for the high output cam, but NO_x emissions for the delayed closure cam at the minimum point of BSFC are identical to those for the high output cam because of the reduction in in-cylinder temperature produced by delayed valve closure.

4.4. Switching Range for Delayed Closure Cam and Enhancement in Fuel Economy

Figure 12 shows a map of the rate of reduction in BSFC for the delayed closure cam against the high output cam. The effectiveness of the delayed closure cam in increasing fuel economy is most apparent at the low engine speed, low load range in which its effectiveness in reducing pumping loss is greatest, but it produces fuel economy benefits across the entire range in which it operates.

In addition to the effect of the delayed closure cam, fuel economy has been further enhanced by the application of technologies to reduce friction, resulting in the achievement of a 17 km/l fuel economy rate in the Japanese 10-15 mode using a vehicle employing the R18A engine.

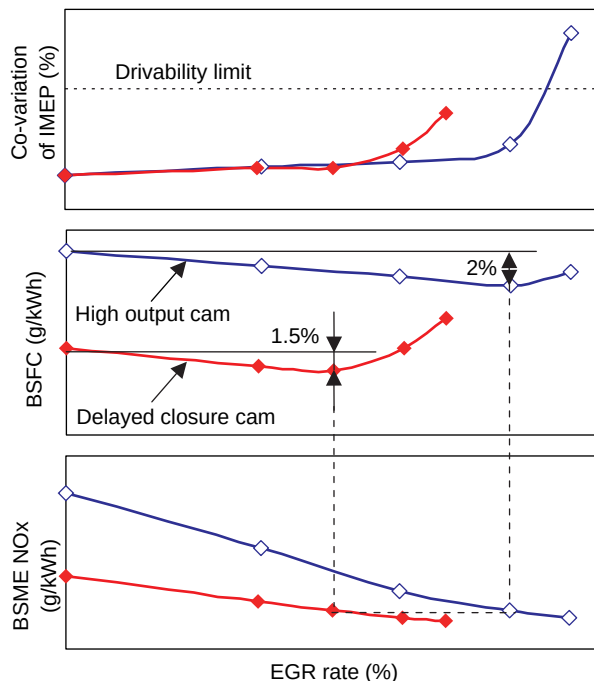


Fig. 11 Comparison of EGR characteristics between conventional valve timing and delayed valve timing

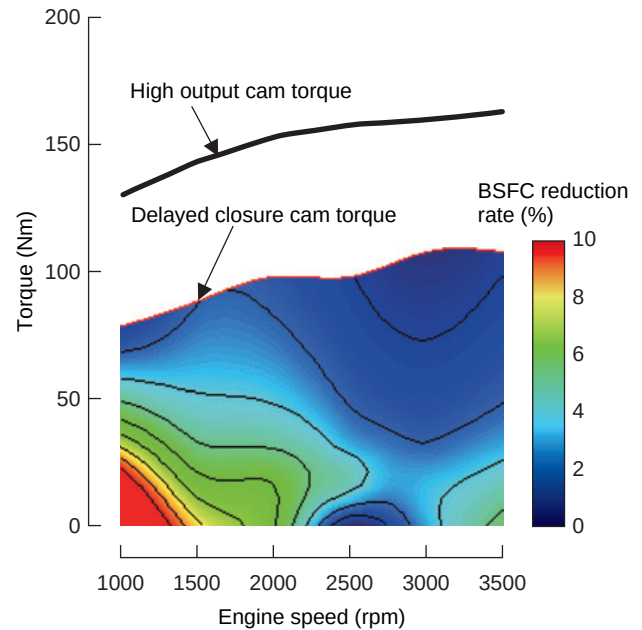


Fig. 12 BSFC reduction rate map (W/O EGR)

5. Conclusion

An i-VTEC mechanism that is able to switch between a high output cam and a cam enabling delayed intake valve closure has been developed. During the development process, the relationship between delay timing and the enhancement of fuel economy was examined. The following are the significant findings of this research:

- (1) Operation of the delayed closure cam produces an over-expanded cycle. At an I.V.C. of 94 deg, BSFC is increased by 6% against that of the high output cam (I.V.C. 31 deg).
- (2) When operation is switched between the high output and delayed closure cams while maintaining the same compression ratio, the enhancement in fuel economy is dependent on the reduction in pumping loss.
- (3) Despite the fact that the delayed closure cam has a lower EGR limit for stable combustion than does the high output cam, their levels of NO_x emissions at the minimum point of BSFC are identical, due to the reduced in-cylinder temperatures produced by delayed closure.

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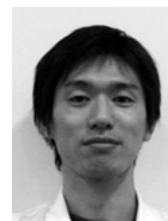
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Friction Analysis by Piston Secondary Motion Simulation in Consideration of Cylinder Block Bore Distortion

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Kinya FUJII*
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ABSTRACT

Piston skirt friction has been considered to be related to cylinder block bore distortion and piston secondary motion, but it has been studied almost exclusively in isolation, and its detailed relationship to the functioning of an actual engine has not been clarified. We used computer simulations to reproduce cylinder block bore distortion and piston secondary motion in combination to try to increase our understanding of their relationship with an actual engine. First, calculated results were compared with measured values to ascertain their validity. Next, the technique was applied to the development of an actual engine, and the engine performance was studied along with the establishment of a method to predict friction. The application of the new simulation technique in engine development resulted in increased reliability and a reduction in friction of approximately 2%, demonstrating the effectiveness of the method.

1. Introduction

Reduction of friction is known to be one basic means of increasing engine fuel economy. In particular, reducing friction in the piston area, where about 25% of engine friction occurs⁽¹⁾, is considered an issue. Factors connected with piston friction include distortion of the cylinder block bore and sliding between the liner and the piston skirt due to the behavior of the piston itself. Recently, there have been reports⁽²⁾⁻⁽⁴⁾ of cases in which computer simulation was used to analyze these factors, and such analysis is also being used in engine development. Accordingly, this research was aimed at reconfirming the accuracy of these simulations and at raising the prediction accuracy by combining cylinder block bore distortion and piston secondary motion. This prediction technology was used in actual engine development, and the results obtained are reported here.

2. Research Aims

Cylinder block bore distortion and piston secondary motion have been investigated before, but these investigations have almost always been carried out separately, so it was not possible to clarify the correlation between specification changes and the detailed performance of an actual engine. Also, the piston is required to have various performance specifications during development, so a method that permitted an overall judgment to be made was needed.

Therefore, computer simulation was used to reproduce cylinder block bore distortion and piston secondary motion in combination. Prediction technology was also developed to permit more accurate judgment of friction reduction and seizing reliability when the shape in the vicinity of the piston skirt was changed.

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3. Calculation Method

3.1. Calculation Flow

Figure 1 shows the calculation flow. ABAQUS was used to perform structural analysis of the distortion and thermal distribution of the cylinder block, and STAR-CD was used to perform computational fluid dynamics⁽⁵⁾ of the thermal heat transfer coefficient distribution in the water jacket to raise the accuracy of the thermal analysis. PISDYN was used to analyze the 3-dimensional behavior of the piston.

3.2. Cylinder Block Distortion Analysis

Figure 2 shows the FEM model used for the cylinder block thermal distribution and distortion analysis. The thermal and distortion analyses are conducted independently of each other. By correlating the results of the thermal analysis to the process following distortion analysis, the distortion when the engine is hot is reproduced.

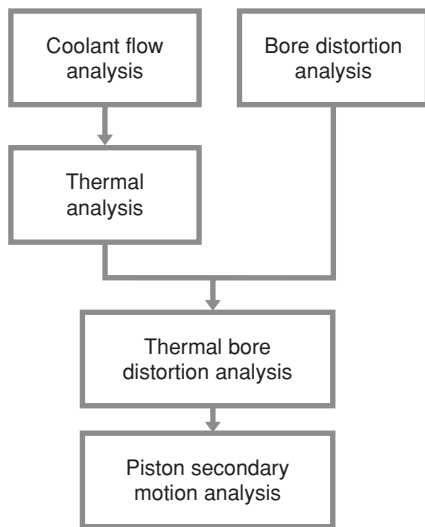


Fig. 1 Calculation flow

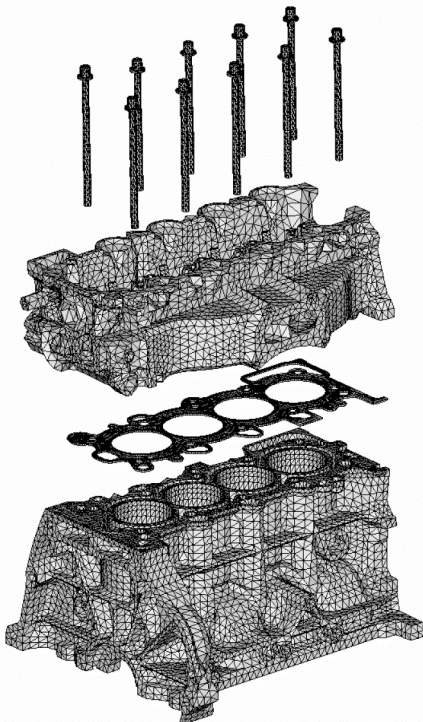


Fig. 2 Cylinder block FEM model

The main input conditions for thermal analysis are the thermal conductivity of each member, the heat transfer coefficient of the surface of the member, and the surrounding temperature. The heat transfer coefficient of the coolant is defined by referring the heat transfer coefficient obtained by computational fluid dynamics to the structural model, as mentioned above. Figure 3 shows the distribution of the heat transfer coefficient obtained from computational fluid dynamics. Regarding other areas, including the liner section, it is necessary to set conditions using measured values or experience, and at present it is desirable to compare these conditions with the measured temperature, and correct them accordingly.

The main input conditions for distortion analysis are Young's modulus, the coefficient of thermal expansion, the assembly bolt force of each member and the spring characteristics of the cylinder head gasket. Regarding Young's modulus and the coefficient of thermal expansion, temperature dependence data is used, while regarding the non-linear spring characteristics of the cylinder head gasket, individual measurement test results are used. These characteristics are shown in Fig. 4.

Figure 5 shows a contour map of the distorted cylinder block when hot.

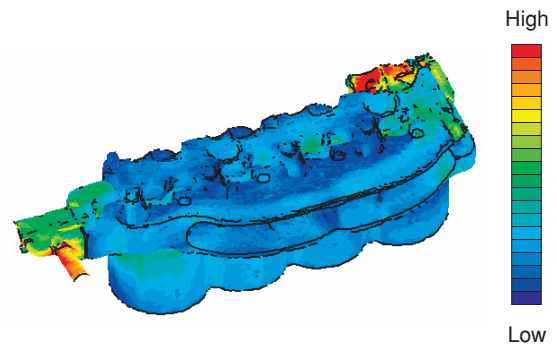


Fig. 3 Distribution of heat transfer coefficient

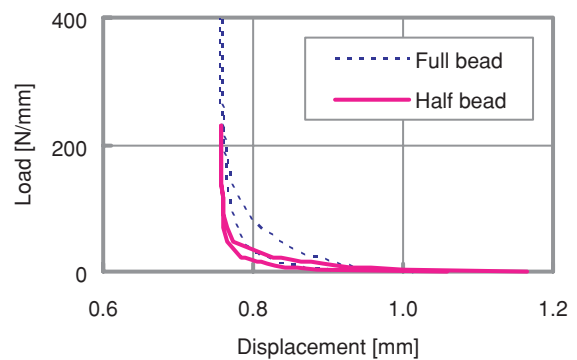


Fig. 4 Gasket characteristic

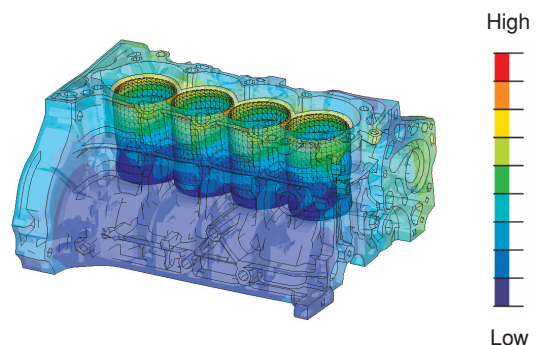


Fig. 5 Cylinder block bore distortion

3.3. 3-dimensional Piston Secondary Motion Analysis

Figure 6 shows an FEM model of a piston. Piston secondary motion analysis is performed using half a cylinder in the program. For the FEM, the piston and cylinder block are prepared, and reduced data is used. Piston temperature is calculated, and a piston model is created, taking into account thermal expansion due to the temperature distribution.

The results of the cylinder block bore distortion analysis are input as numerical data. Oil film characteristics that take into account elasto-hydro dynamics (EHD) are used for the sliding faces of the piston skirt and the liner. The surface roughness R_q is measured and input in order to take into consideration the boundary lubrication condition due to the surface roughness.

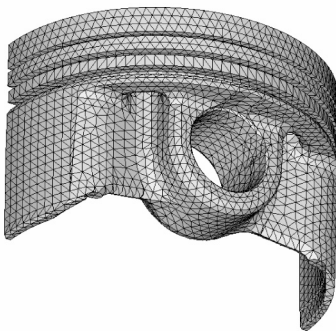


Fig. 6 Piston FEM model

4. Analysis Results and Measurement Results

4.1. Cylinder Block Bore Distortion

Before the distortion analysis results were applied to the performance test of an actual engine, the quality of the results was verified.

Figure 7 indicates cylinder block bore distortion when it is assembled at room temperature, and Fig. 8 indicates cylinder block bore distortion when the cylinder block is heated uniformly in a high temperature chamber. In both cases, the distortion is expressed as a change in diameter in the IN and EX directions. The comparison in a uniformly heated condition is indicated as distortion only corresponding to the temperature change to cancel the error that

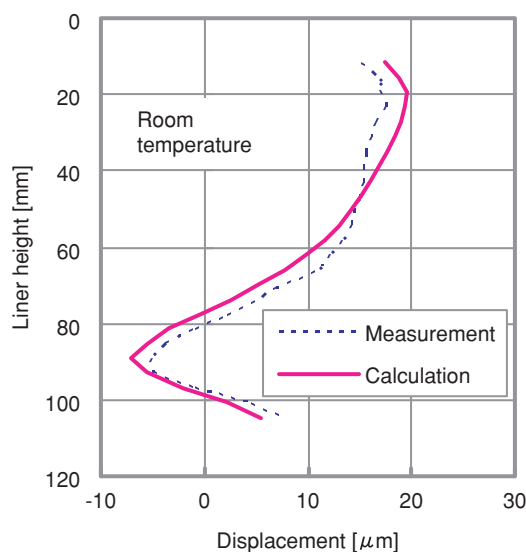


Fig. 7 Bore distortion at room temperature

occurs in the case of room temperature assembly. An inometer was used for measurement, and the comparison was performed using a No.3 bore of an Inline 4-cylinder engine as a representative bore. The results indicate good correlation between analysis and measurement.

Figure 9 shows the temperature distribution measurement points, and Fig. 10 shows a comparison between analysis and measurement. A temperature comparison was performed at two cross-sections located at the liner heights of 9 mm and 89 mm from the upper deck. Thermocouples were used for measurement, and the input conditions were adjusted so that the difference between the analyzed and measured temperatures was within 10 °C.

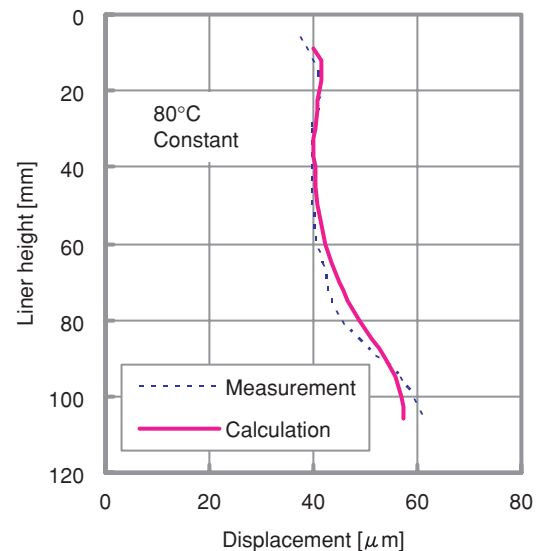


Fig. 8 Bore distortion for temperature difference

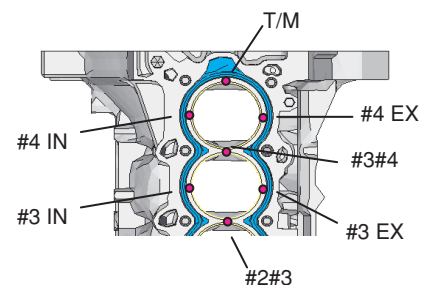


Fig. 9 Temperature measurement point

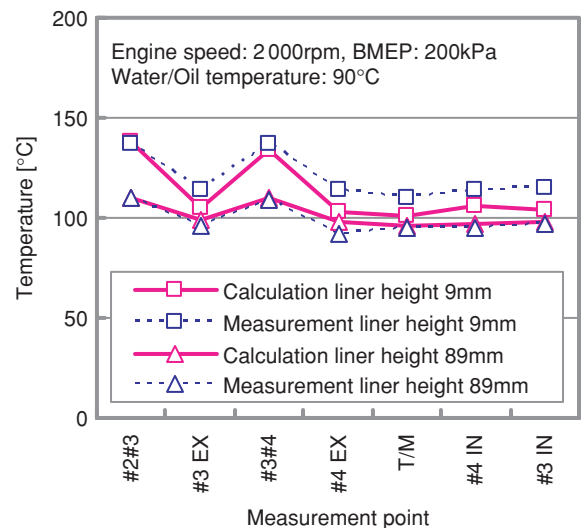


Fig. 10 Temperature distribution of cylinder block

4.2. Piston Secondary Motion

As in the cylinder block bore distortion analysis, a comparison with measured values was carried out to confirm the quality of piston secondary motion analysis. Figure 11 shows a comparison of the analyzed and measured values of translation movement of the piston, and Fig. 12 shows a comparison of the computed and measured values of rotational movement of the piston. Measurement was carried out with gap sensors recessed into the cylinder block. Because of the layout of the sensors, secondary

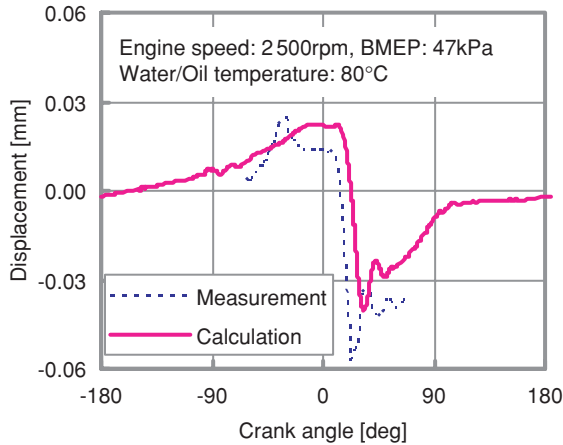


Fig. 11 Translation movement

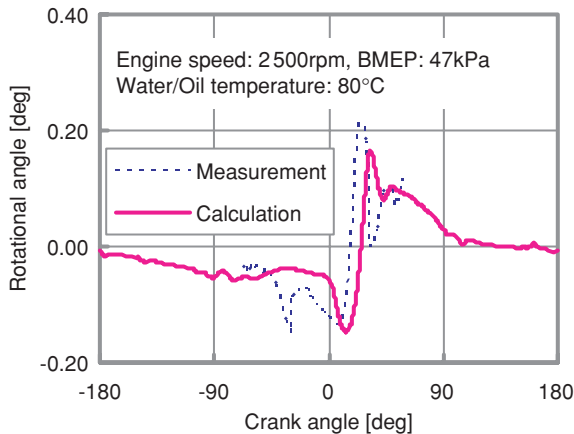


Fig. 12 Rotational movement

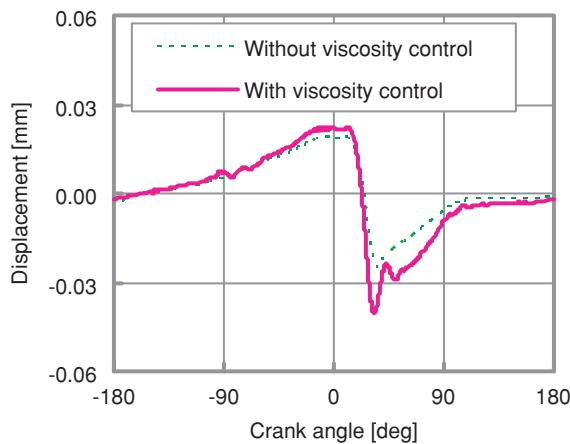
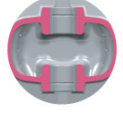


Fig. 13 Comparison of oil viscosity in piston secondary motion

Table1 Specifications of engine and piston

	Previous engine	Newly developed engine
Cylinder configuration	Inline 4-cylinder	Inline 4-cylinder
Engine type	SOHC 2 Valves	SOHC 2 Valves
Ignition system	Dual point sequential ignition	Dual point sequential ignition
Fuel supply system	PGM-FI	PGM-FI
Displacement [cm ³]	1339	1339
Bore x stroke [mm]	73 x 80	73 x 80
Compression ratio	10.8	10.8
Maximum power [kW]	63	70
Maximum torque [Nm]	119	123
Compression height [mm]	26.2	26.2
Skirt width x height [mm]	34 x 29.5	38 x 29.5
Piston weight [g]	207	205
Piston shape		

motion was measured over a range of only about 60 degrees in the vicinity of the TDC.

In the program, the space between the piston skirt and the liner is completely filled with oil, which leads to a tendency to excessive damping. Consequently, in this research, the computed oil viscosity was adjusted, resulting in secondary motion that approximated the measured results. Figure 13 shows the analysis results of the translation movement before and after adjustment of the oil viscosity.

5. Results of Application to an Actual Vehicle

5.1. Basic Specifications of Engine and Piston

As mentioned above, appropriate analytical accuracy was acquired, the effectiveness of changing the piston specification in reducing friction in an engine and increasing reliability was predicted, and the results were compared with the test results.

Table 1 shows the specifications of the engine and piston of a previous engine and those of the newly developed engine. The specifications of the piston were changed twice. In the first change, the shape and material of the piston were changed⁽⁶⁾, and in the second, only the piston form was changed.

Figure 14 shows the stiffness distribution of the piston skirt. Figure 15 shows a comparison of the piston skirt stiffness gradient at the center cross-section of the piston pin. A unit load is assigned to each lattice point of the skirt FEM to obtain the displacement, and the spring value is calculated to express the stiffness. To prevent uneven contact, the development specifications were designed to obtain a softer and more uniform stiffness distribution.

The piston form was first made highly ovalar, aiming at avoiding hard contact along the piston skirt edge. Next, while reducing the hydrodynamic lubrication friction at the oval convexity at the center of the skirt, adjustments were made to prevent the boundary

lubrication friction from increasing.

Within the scope of this study, it was found that the contribution to friction reduction of longitudinal change was not as high as that of the change in ovality.

5.2. Friction Study Results

Table 2 shows a comparison of the analyzed and measured results of the friction reduction rate under motoring conditions subsequent to warm-up. The changes in the shape and material produced a 1.3% reduction in friction, while the change in the piston form produced a 0.7% reduction. Compared to previous specifications, a total friction reduction of 2% was obtained for the engine alone. There was good correlation between analysis and measurement in respect to these values.

Figure 16 shows the pressure distribution for boundary lubrication in the circumferential direction of the piston skirt for

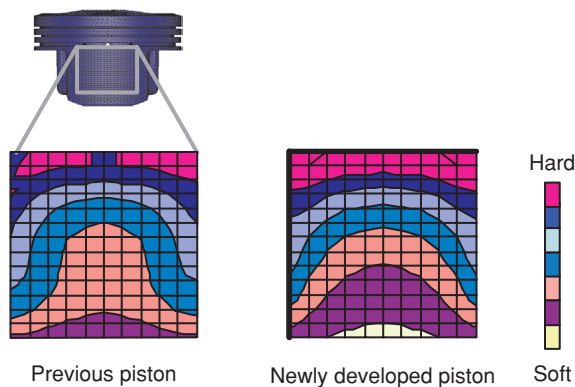


Fig. 14 Piston skirt stiffness

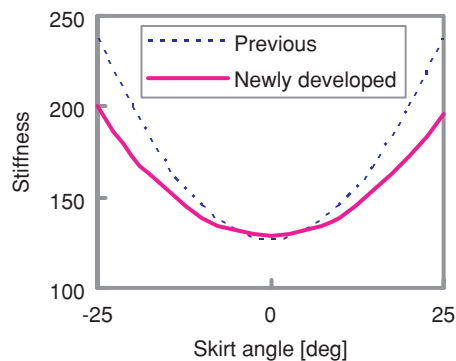


Fig. 15 Piston skirt stiffness in piston pin center section

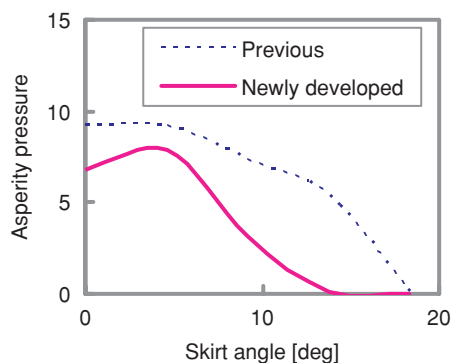


Fig. 16 Asperity pressure

Table 2 Friction reduction rates

	Calculation	Measurement
Piston shape & material	-1.4%	-1.3%
Piston form	-0.7%	-0.7%

the first specification study. The pressure is the integrated value at the cross-section in each circumferential direction. As mentioned above, in the newly developed specifications, the piston stiffness value along the circumferential edge of the piston skirt is low, so the pressure during boundary lubrication is largely reduced along the circumferential edge. From these results, it can be seen that the piston that has a small increase in stiffness along the circumferential edges of the skirt; in other words, it has a more uniform stiffness distribution, and exhibits a lower increase in friction.

Figure 17 shows the contribution to friction reduction of the shape and material (Young's modulus, coefficient of thermal expansion) in the first specification study. The material physical property values were changed in calculations and verified. The contribution due to the abovementioned specification changes were 74% for shape, 22% for Young's modulus and 4% for thermal conductivity.

Figure 18 shows the contribution of the cylinder block bore distortion in the simulation for the second specification change. It can be seen that the friction prediction accuracy is increased by inputting the cylinder block bore distortion.

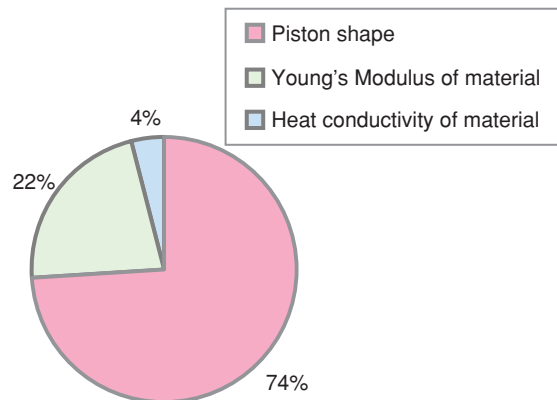


Fig. 17 Influence of piston shape and material

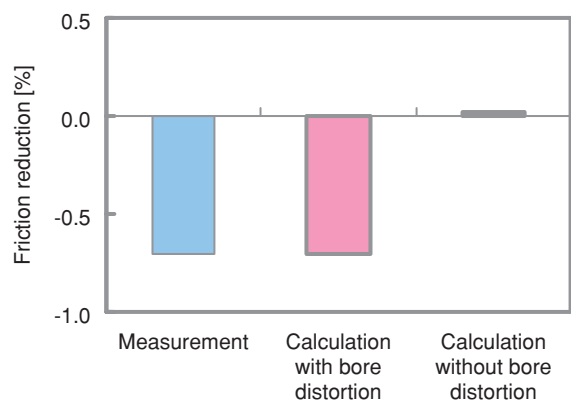


Fig. 18 Influence of bore distortion in friction prediction

5.3. Seizing Study Results

Reliability is evaluated via seizing. The wear load (work rate per contact unit area due to boundary lubrication, face pressure $\times$ velocity) is used as the evaluation index.

Figure 19 shows the comparison results. The study takes into consideration the upper and lower limits of the cylinder block bore distortion based on the trend of the distortion when the temperature of the cylinder block is changed. The error indication in the figure is the wear load at the upper and lower limits of the cylinder block

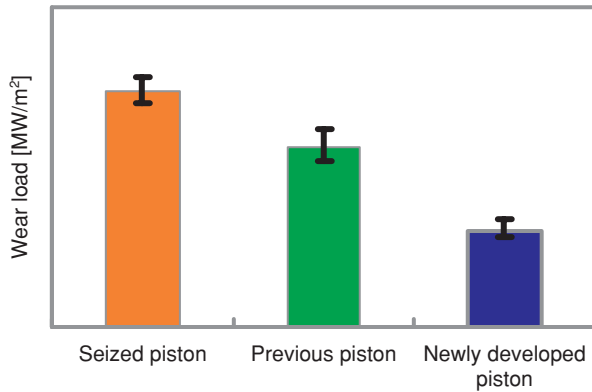


Fig. 19 Comparison of wear load

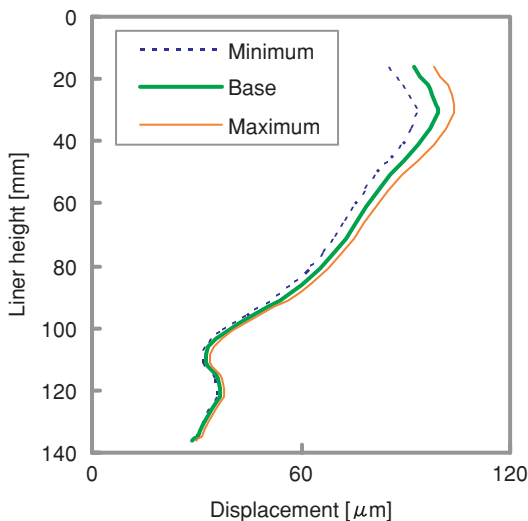


Fig. 20 Bore distortion by temperature difference

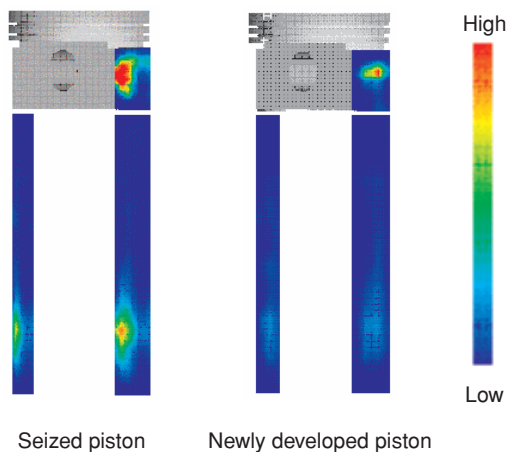


Fig. 21 Distribution of wear load

bore distortion. Figure 20 shows the effect of temperature on the cylinder block bore distortion.

The peak value under previous specifications is lower than that of the specifications under which seizing occurred during an actual engine test, and is even lower in the case of the newly developed specifications.

Figure 21 shows a comparison of the wear load distribution at the piston skirt. It can be seen that the distribution of wear load is narrow. This is because, as in the case of friction, the stiffness of the edge part of the skirt is low, and the ovality of the piston form was increased to prevent a large area of contact on the edge of the piston skirt.

By using the abovementioned specifications, it was possible to complete an actual engine test with no seizing.

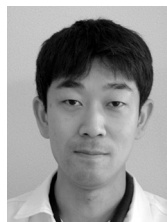
6. Conclusion

- (1) It is possible to carry out a detailed desktop study of friction reduction while judging the trend of durability and reliability.
- (2) A 2% reduction of friction was obtained by using the results of this study in the development of an actual engine.
- (3) Low increase in stiffness at the edge part of the piston skirt and more uniform stiffness distribution have an effect on friction reduction.
- (4) The contribution to friction reduction of the piston under the development specifications was 74% for shape, 22% for Young's modulus and 4% for thermal conductivity.
- (5) The friction reduction effect due to the piston form is 0.7%, and can be increased further by additionally changing the piston form.
- (6) Concerning seizing, an evaluation based on the wear load is possible.
- (7) In the evaluation of friction and seizing, it was necessary to input the cylinder block bore distortion, therefore the evaluation results varied according to the accuracy of the distortion input.

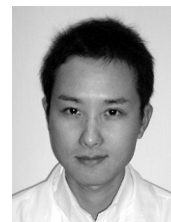
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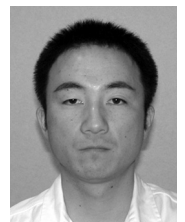
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Effect of Piston Speed around Top Dead Center on Thermal Efficiency

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ABSTRACT

In a reciprocating internal combustion engine, the Otto-cycle allows for maximum thermal efficiency with a same compression ratio, which means complete combustion in a moment at top dead center, which is practically impossible. In view of this fact, it can be considered that a slower transition of the piston speed near top dead center would promote combustion during that period. This, in turn, would increase the in-cylinder pressure and degree of constant volume thereby improving thermal efficiency. To prove this, an engine in which the piston can be displaced moderately to achieve ideal constant volume combustion was prepared and experiments were carried out. As anticipated, while the degree of constant volume increased, the thermal efficiency did not improve due to increased heat loss. Subsequently, further experiments were carried out using a direct injection, stratified charge engine which allows selective reduction of heat loss, and a high thermal efficiency was attained.

1. Introduction

Quick burn is an effective method of increasing the degree of constant volume of an internal combustion engine in order to improve the thermal efficiency. Conventional methods of achieving this involve the use of swirling or turbulence in the cylinder to increase the combustion speed⁽¹⁾, the use of a more compact combustion chamber⁽²⁾, and/or multiple spark plugs⁽³⁾ to reduce the flame propagation distance. In recent years, attention has been focused on the fact that homogeneous charge compression ignition (HCCI) systems are also effective in increasing the thermal efficiency using similar methods^{(4)–(7)}. All of these methods attempt to reduce the time required for combustion to occur, in order to approach the ideal Otto cycle and increase the area enclosed by the P-V diagram. Also, attempts to structurally increase the area of the P-V diagrams have been reported by Miwa et al.⁽⁸⁾.

Here, the authors produced two kinds of prototype engines which employed a conventional crankshaft and connecting rod mechanism. Tests were then performed to see whether or not it was possible to increase the thermal efficiency by slowing the piston speed near top dead center, advancing the combustion during this period, and increasing the degree of constant volume. Also, to compensate for the disadvantages of these engines, a third prototype, direct injection-type engine was subjected to the same test. The results of these tests are described here after.

2. Conventional Engine

In a reciprocating internal combustion engine, the piston decelerates and then stops momentarily at both top and bottom dead center. Because the volume of the combustion chamber stops

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changing at these points, this stop of the piston has the important dual role of completing combustion at top dead center and replacing the gas in the cylinder via the intake and exhaust valves at both top and bottom dead center. In the type of internal combustion engine that is most generally used these days, the piston behavior near top dead center differs from that near bottom dead center. As shown in Fig. 1, it is well-known that the piston displacement when the crankshaft turns through a certain angle is greater at top dead center than at bottom dead center.

Accordingly, it was suggested that if it were possible to make the piston displacement at top dead center equal to that at bottom dead center, the piston displacement in the vicinity of new top dead center would decrease and cause combustion to complete before the volume of the combustion chamber had changed significantly. The subsequent increase in the degree of constant volume would improve the thermal efficiency. This point was verified experimentally.

3. Prototype Engine

3.1. ICVC Engine

3.1.1. ICVC engine specifications

To address the above issue, a prototype engine was prepared by modifying a 4-stroke, air-cooled single cylinder 125 cm³ motorcycle gasoline engine as described below. As shown in Fig. 2, at the bottom of the normal piston is a sub-piston that slides together

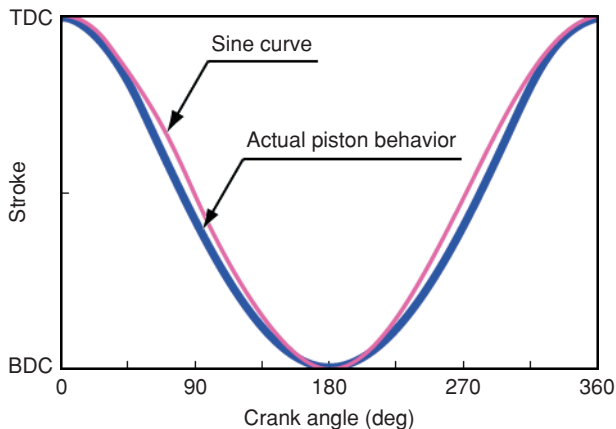


Fig. 1 Actual piston behavior against crank angle

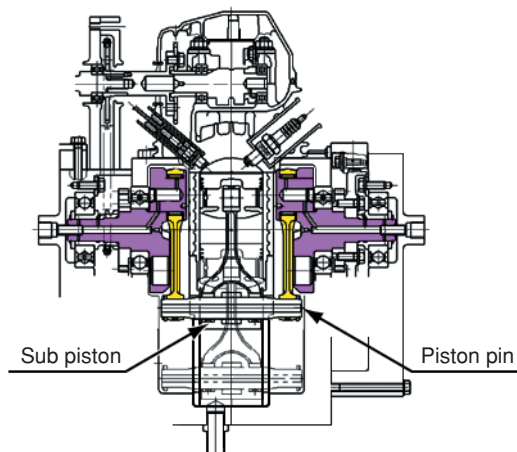


Fig. 2 Ideal constant volume combustion engine

with the normal piston inside the same cylinder. This sub-piston has a long piston pin that passes through the cylinder wall. On both ends of the piston pin are installed the small ends of a pair of connecting rods that can rock freely. The big end of each connecting rod is connected to a crankpin of the divided two pieces of cantilever-type crankshaft. When gas pressure acts on the piston, the connecting rods pull the respective crankpins and rotational force is applied to the crankshaft. The engine was constructed in this way so that combustion would take place at a percentage volume change of the combustion chamber equivalent to that at normal bottom dead center. This type of engine is called an Ideal Constant Volume Combustion (ICVC) engine^{(9), (10)}.

Table 1 shows the main specifications of this engine. The piston, cylinder, cylinder head, connecting rod and some other parts are the same as those used in the base engine. The engine was designed exclusively so that combustion takes place at a position equivalent to bottom dead center of the base engine.

3.1.2. ICVC engine test results

The test was carried out at an engine speed of 3 000 r/min, a partial load resulting in a fuel consumption of 640 g/h, an ignition timing equal to MBT, and stoichiometric air-fuel ratio. The performance of the prototype engine was compared with that of the base engine. Figure 3 shows the results expressed using a P-V diagram. It can be seen that in the case of the ICVC engine, the degree of constant volume has increased, and the P-V diagram is clearly close to the ideal Otto cycle.

Table 1 ICVC engine specifications

Items	Base engine	ICVC engine
Engine type	4-stroke, air cooled, single cylinder	
Bore x Stroke (mm)	52.4 x 57.8	
Displacement (cm ³)	124.6	
Compression ratio	9.35 : 1	9.32 : 1
Connecting rod length (mm)	93.5	

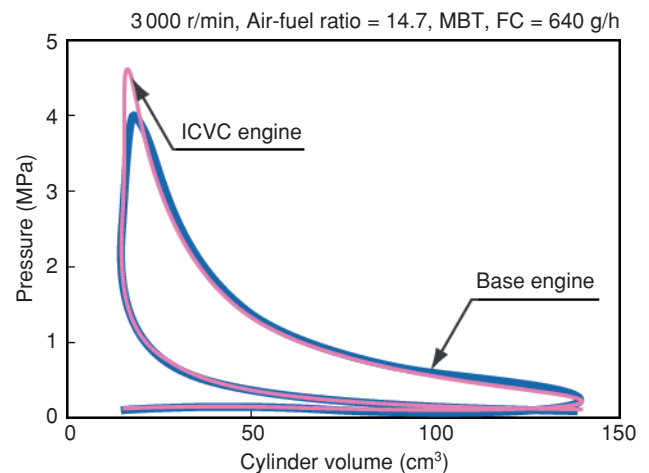


Fig. 3 P-V diagram comparison

Table 2 Test results of ICVC engine

3000 r/min, FC = 640 g/h, Air-fuel ratio = 14.7, θ_{ig} = MBT

Engine	P _{max} (MPa)	Deg. of C.V.	P _{mi} (kPa)	ISFC (g/kWh)
ICVC engine	4.64	0.97	846	243
Base engine	4.02	0.94	893	230

Deg. of C.V. : Degree of constant volume

Table 2 shows the results of calculating the area of the P-V diagram and obtaining the indicated work. It can be seen from the table that the degree of constant volume increased, but despite the fact that the maximum pressure increased by about 15%, the pressure during the expansion stroke fell, and the indicated mean effective pressure fell by 5% causing the thermal efficiency to fall.

3.2. APC Engine

3.2.1. APC engine specifications

To investigate in detail why the thermal efficiency did not improve despite the fact that the degree of constant volume increased, the engine shown in Fig. 4, called an Active Piston Control (APC)^{(9),(10)} engine, with slower piston speed near top dead center was produced.

The crankshaft of the engine has been greatly offset from the center of the cylinder, so that the gas pressure applied to the piston causes the crankshaft to be pulled via the two connecting rods A and B, in the same way as in the ICVC engine, causing the crankshaft to rotate. A control rod is connected to regulate the position where connecting rods A and B are joined together, so that the center of the cylinder does not largely offset from connecting rod A.

As shown in the figure, the engine uses a multiple link mechanism in which the length of the compression stroke differs from that of the expansion stroke. When the crankshaft is rotating counterclockwise, the crank angle required for the intake and expansion stroke is 213 degrees, and that for the compression and exhaust stroke is 147 degrees.

The displacement of the piston of the APC engine is shown in Fig. 5 and the main specifications are shown in Table 3. The engine is a water-cooled, single cylinder gasoline engine that has a 195 cm³ displacement. The compression ratio is 11.5:1, two spark plugs are installed, and a helical intake port is provided, realizing a minimal indicated fuel consumption of 220 g/kWh (at 2 700 r/min, and BMEP300 kPa) during its original operation.

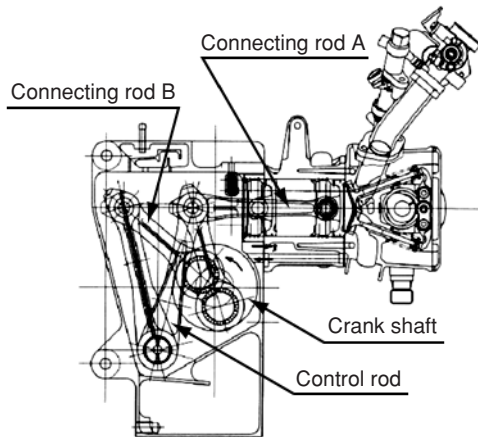


Fig. 4 Active piston control engine

This engine has a camshaft that enables it to be run in reverse. The oil pump and cooling water pump are designed so that the specified quantities of oil and water can be supplied externally at the specified temperature.

3.2.2. APC engine test results

Figure 6 is a P-V diagram which shows a comparison between the results of a test performed on the APC with those of the original engine, for an engine speed of 3 000 r/min, a volumetric efficiency of 34%, a stoichiometric air-fuel ratio, and MBT ignition timing. The crankshaft of the APC engine uses a multiple link mechanism in which the crankshaft is greatly offset from the center of the cylinder. Because an extremely large load acts on each link, the elastic deformation is large and it was found that the crank angle

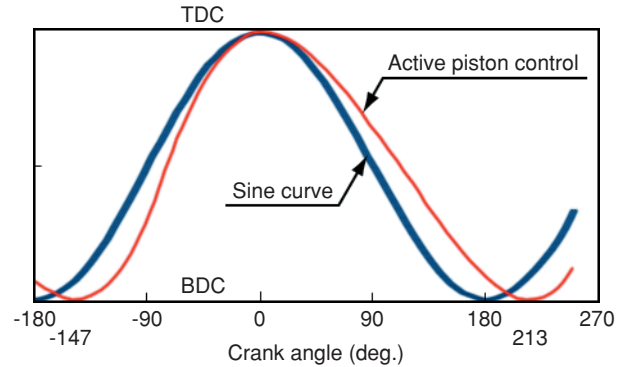


Fig. 5 Piston behavior of APC engine

Table 3 Specifications of APC engine

Engine type	4-stroke, water cooled
Bore x Stroke (mm)	59 x 71.3
Displacement (cm ³)	195
Compression ratio	11.5 : 1
Spark plug	Twin plug

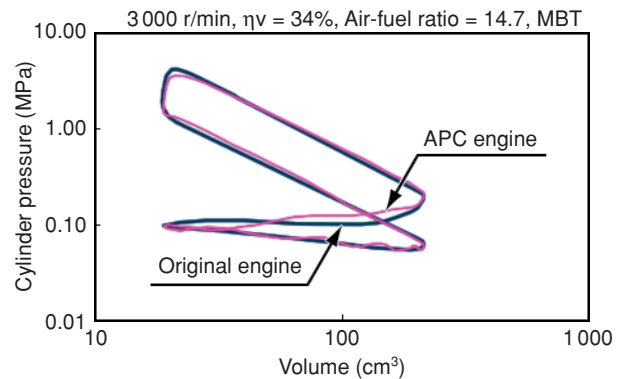


Fig. 6 P-V diagram comparison between APC engine and original engine

Table 4 Test results of APC engine

3 000 r/min, $\eta_v = 34\%$, $\theta_{ig} = \text{MBT}$, Air-fuel ratio = 14.7

Engines	Fuel consumption (g/h)	Exhaust gas		Cooling water (out-in)	
		Temperature (K)	Exhaust loss (kW)	Temperature (K)	Heat loss (kW)
APC engine	490	901	2.40	6.0	1.55
Original engine	494	920	2.47	4.8	1.24

does not determine the piston displacement (combustion chamber volume). For this reason, as shown in Fig. 6, the degree of constant volume of the APC engine did not increase as in the case of the ICVC engine, resulting in a reduction of P_{max} .

Accordingly, it was decided to abandon the idea of obtaining the indicated work from the P-V diagram, and instead to carry out a study from the measured temperature of each part. Table 4 shows the measured results of the fuel consumption, the exhaust gas temperature and the difference in temperature between the cooling water outlet and inlet of the engine at 3 000 r/min and $\eta_v 34\%$. This data is compared with the corresponding figures for the original engine. It can be seen from the table that in the case of the APC engine the exhaust gas temperature has fallen, whereas the water temperature rose, which indicates the amount of heat released to the cooling water is quite large.

On the other hand, it can be considered that because the supplied mixture gas is in the stoichiometric air-fuel ratio, the product of the exhaust gas temperature and the fuel consumption should be proportional to the exhaust loss amount. The results of calculating the heat balance based on the above assumption are shown in Fig. 7.

It is considered that the amount of heat released to the cooling water is equivalent to the heat loss amount, and that the main reason why the thermal efficiency of the ICVC engine did not improve, despite that fact that the degree of constant volume increased, was due to the increased heat loss.

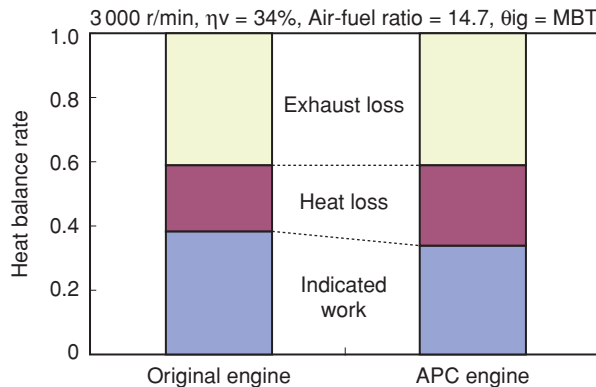


Fig. 7 Heat balance rate of test results

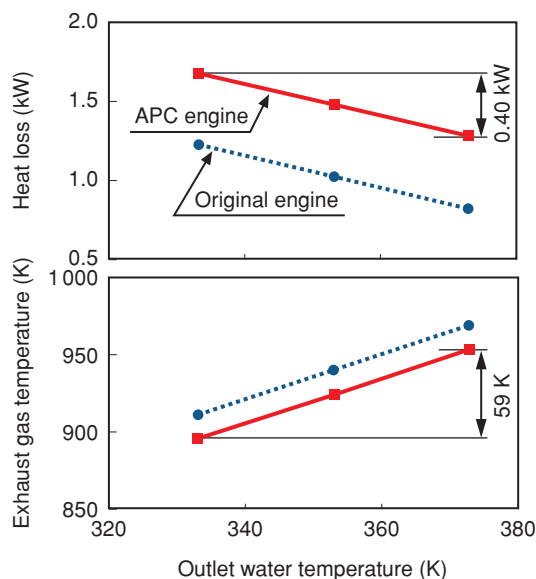


Fig. 8 Heat loss and exhaust gas temperature

To verify this, the cooling water temperature was changed from 60 °C to 100 °C. The exhaust gas temperature at this time and the work ratio obtained by multiplying the difference between the cooling water inlet and outlet temperatures by the cooling water flow rate (heat loss) are shown on the vertical axis of Fig. 8. When the cooling water temperature was raised, the heat loss was reduced by 0.4 kW and the exhaust gas temperature rose by 59 °C. When this value is converted into a work ratio, it corresponds to 0.29 kW, and about 75% of the reduced heat loss is converted into the increase in the exhaust loss. It was found that these figures were the same for both the APC engine and the original engine.

In the final analysis, it was found that reducing the piston speed in the vicinity of top dead center caused the heat loss to increase, and because the sum of the heat loss and the exhaust loss is nearly constant, only the heat loss cannot be selectively reduced.

To further study this relationship, a test was performed with the camshaft reassembled to enable the engine to rotate in reverse. Figure 9 shows the measured values of the temperature difference Δt between the cooling water inlet and outlet temperatures, and the exhaust gas temperature, when the ignition timing was changed through a 10 degree crank angle in the advance and retard directions from MBT to change the heat loss. When the APC engine is running in reverse, the crank angle of the compression stroke is 213 degrees, however the crank angle of the expansion stroke falls to 147 degrees. Consequently, it was predicted that the exhaust gas temperature, which indicates the exhaust loss, would be higher than that of the

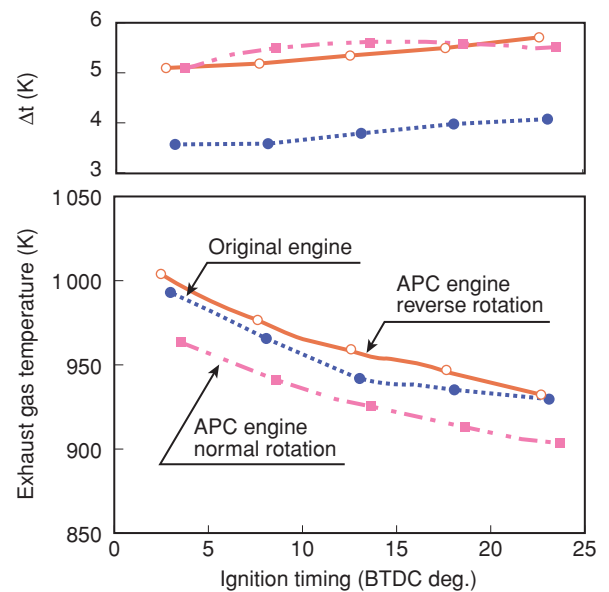


Fig. 9 Heat loss and exhaust gas temperature against ignition timing

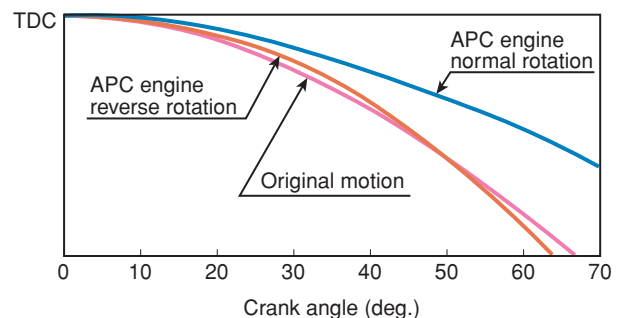


Fig. 10 Detailed figure near TDC including reverse rotation

original engine, and this result was in fact obtained. However, the cooling water inlet and outlet temperature difference was not as small as expected, and the same results were obtained during forward motion as well.

Figure 10 shows an enlarged diagram of the piston displacement in the vicinity of top dead center. In the APC engine, the piston displacement in the vicinity of a 45° crank angle after top dead center is more moderate than that of the original engine, even when the engine is rotating in the reverse direction, so it is considered that the heat lost during this interval probably flowed from the wall of the combustion chamber to the cooling water.

4. Combination with a Direct Injection Engine

4.1. $\lambda = 8$ Engine

4.1.1. $\lambda = 8$ engine specifications

The elastic deformation of each link of an APC engine is large, preventing the position of the piston with respect to the crank angle from being determined. As a result, it was not possible to obtain the indicated work from the P-V diagram. Accordingly, the conventional connecting rod and crank mechanism were used, and the piston speed in the vicinity of top dead center was varied by changing the ratio l between the connecting rod length l and the crankshaft radius r (defined as l/r). By making this value λ as large as possible, it was found that piston displacement versus the crank angle approached a sine curve. Accordingly, a test was carried out on an engine that had a λ of 8, which was considered a realistic value, and carried out an analysis from the P-V diagram. As shown in Fig. 11, the connecting rod of this engine is considerably longer than that of a conventional one.

Figure 12 shows an enlarged diagram of the behavior of the piston in the vicinity of top dead center. The λ of the base engine is 3.34 and the diagram indicates that the displacement characteristics of the test engine are mid-way between a sine curve and the displacement characteristics of the base engine.

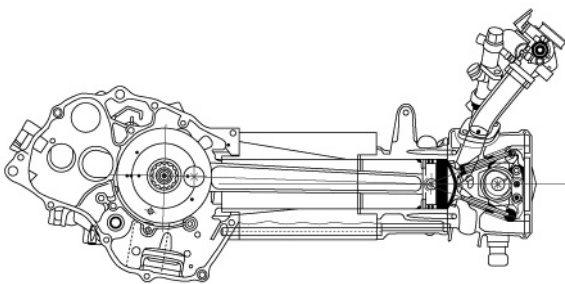


Fig. 11 $\lambda = 8$ engine

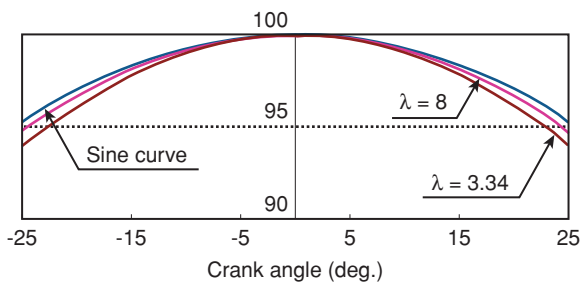


Fig. 12 Piston behavior around top dead center

The bore stroke and specifications of the combustion system are identical to those of the previously mentioned APC engine. This engine was compared with a normal premixed port injection (PI) engine and a direct injection (DI) stratified charge combustion engine. Heat loss can be selectively reduced with the DI stratified charge combustion engine.

4.1.2. $\lambda = 8$ engine test results

The test was carried out at an engine speed of 3 000 r/min and at IMEP of between 0.3 and 0.75 MPa. The results are shown in Fig. 13. Compared to the premixed port injection, the indicated specific fuel consumption of the base DI engine decreased by about 13 to 7 g/kWh. In the case of the engine whose λ is 8, the fuel consumption deteriorated by 8 g/kWh when port injection was used, but decreased by 4 to 8 g/kWh when direct injection was used. This latter case corresponds to the best indicated specific fuel consumption.

4.2. Consideration Concerning Heat Loss and Piston Behavior

The way in which the ratio between the heat loss and the exhaust loss changed was extracted using the respective center values of the data in Fig. 13 and is shown in Fig. 14 as the heat balance rate. The heat loss was obtained as mentioned previously by multiplying the cooling water inlet and outlet temperature difference by the flow rate, and the exhaust loss was obtained by subtracting the sum of the indicated work and heat loss from the input heat amount.

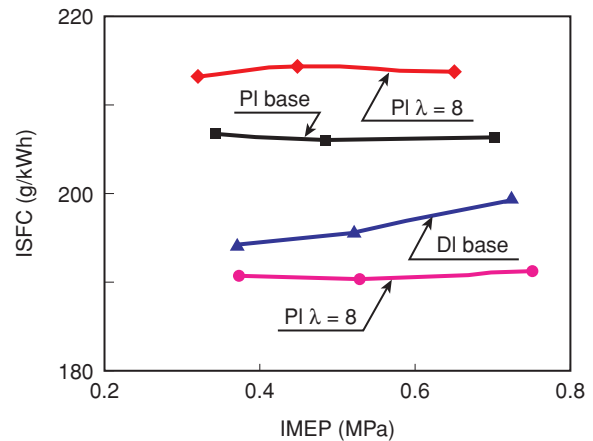


Fig. 13 ISFC comparison between PI and DI engine

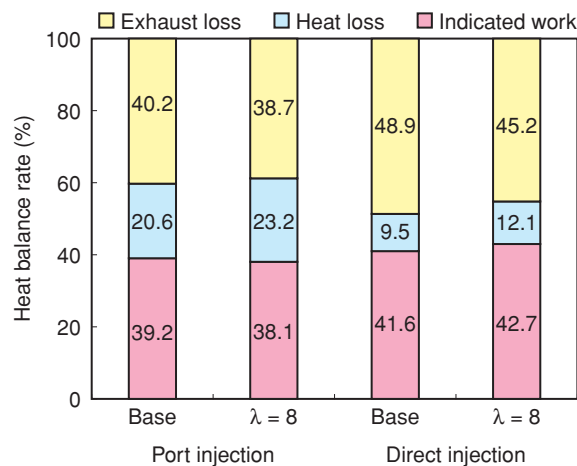


Fig. 14 Heat balance rate

An engine with a large λ had greater heat loss and a reduction in indicated work in the case of port injection. However, when direct injection was used, heat loss was low from the outset, and although the heat loss increased when the λ was raised, the exhaust loss reduction effect resulting from the increased degree of constant volume predominated, resulting in a maximum indicated work of 42.7%.

5. Conclusion

To allow the piston displacement to change moderately around top dead center and increase the degree of constant volume to improve the thermal efficiency, an ICVC engine as well as an APC engine whose compression stroke was of a different length than its expansion stroke were produced and tested. As a result, the following facts were established.

- (1) The heat loss increased and the thermal efficiency did not improve. This was thought to be due to the fact that a high pressure and temperature around top dead center were maintained for a longer period of time.
- (2) By changing the cooling water temperature to reduce heat loss, it was found that the exhaust temperature increased and a mutual trade-off relationship existed between the heat loss and exhaust loss.

Accordingly, to selectively reduce heat loss, the direct injection engine was operated as a stratified charge combustion and a test was carried out using an engine that had a λ of 8. As a result, it was possible to selectively reduce the heat loss alone. Also, the degree of constant volume increased, causing the thermal efficiency to improve by 1.1 point to reach 42.7%, which is the maximum value.

Acknowledgments

The authors would like to take this opportunity to thank Assistant Professor Yasuo Moriyoshi of the Engineering Department of Chiba University for providing valuable advice.

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Simulation of Accessory Drive Belt Behavior Using Multi-body Dynamics

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ABSTRACT

A simulation method was proposed to predict the slippage and transversal vibration of the accessory drive belt. To reproduce these phenomena, the accessory drive belt was represented by a section-by-section model in which the belt was replaced by a finite number of masses and springs using multi-body dynamics simulation. In this model, the belt was able to vibrate in both the direction of advance and the direction perpendicular to it, and a friction contact element was defined between the pulley and the belt. The coefficient of friction was made variable with respect to the slippage speed to enable stick and slippage to be predicted. This method allows for accurate simulation of the amount of belt slippage and the amplitude of transversal vibration, thus enabling the optimum belt layout to be determined at the design stage.

1. Introduction

Recently, the method of driving multiple accessories (pulleys) by passing a single V-ribbed belt around each pulley in a serpentine form has been gaining wider use. This arrangement enables the width of the engine to be reduced, thus contributing to compactness. On the other hand, the load applied to the belt is higher than with the conventional arrangement, and slippage and transversal vibration are likely to occur, producing durability, noise and vibration issues. For this reason, until now, much research has been carried out on methods that predict these issues using complex theoretical equations and simulations^{(1),(2)}. Generally, the dominant factor involved in belt slippage and transversal vibration is the phenomenon of fluctuating belt tension. A belt system has resonance that is determined by rigidity in the longitudinal direction of the belt and by the inertia of each accessory, causing rotational vibration

in the pulleys. This rotational vibration will hereinafter be called longitudinal vibration. There are many cases in which, when the V-ribbed belt of this so-called “serpentine method” is used, the longitudinal vibration matches the fluctuation of the crankshaft’s angular velocity, resulting in large tension fluctuations in normally used areas. For this reason, the one-dimensional longitudinal vibration model has been the main modeling method used in simulations until now. The slippage and transversal vibration between the belt and the pulley has also been estimated from the tension fluctuation obtained using this longitudinal vibration^{(3),(4)}. However, though this method has the merit of reducing computation time, it does not directly predict the extent of slippage or transversal vibration. To solve these issues, simulation technology using a section-by-section model in which the belt is replaced by a finite number of springs was created, as reported here.

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2. Belt Model

The simulation model consists of the accessory drive belt and the crankshaft that drives it.

The belt is divided into a finite number of mass points that are coupled together by beam elements. Each mass point can vibrate not only in the belt's direction of advance, but also perpendicular to the direction of advance. Thus, the belt's transversal vibration, which could not be predicted using a one-dimensional longitudinal vibration model, can now be predicted. This model is called a section-by-section model⁽⁵⁾. Figure 1 shows a pattern diagram of this model.

Figure 2 shows the coupling element of the belt model. Because the filament is more rigid than the rubber, the filament transmits the load. Consequently, in the simulation, adjacent mass points are coupled to each other at the filament's position. The beam elements are used for coupling. As a result, the filament is the neutral surface of the bending deformation, and it is possible to reproduce behavior similar to that of an actual belt. A value identified from the belt-induced vibration test is used as the bending rigidity. In addition, the belt is vulcanized into a cylindrical shape at the time of manufacture. Consequently, because the belt is initially cylindrical, the simulation is also set so that the belt is cylindrical when it is not wrapped around the unloaded pulley.

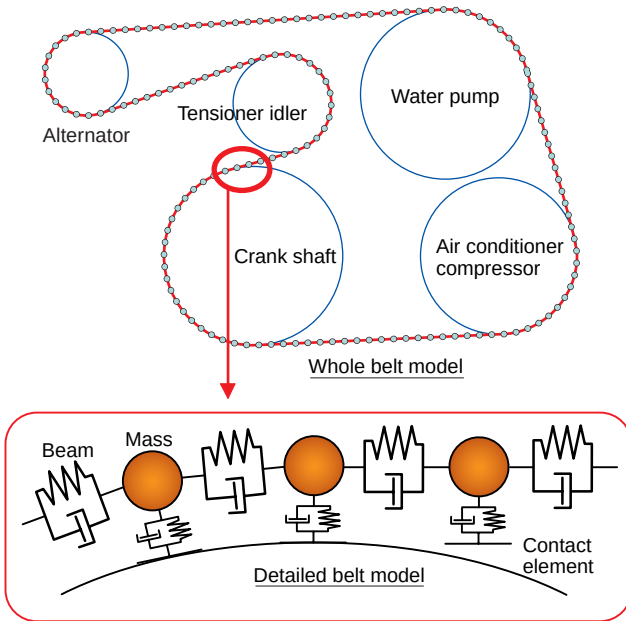


Fig. 1 Section-by-section model of belt

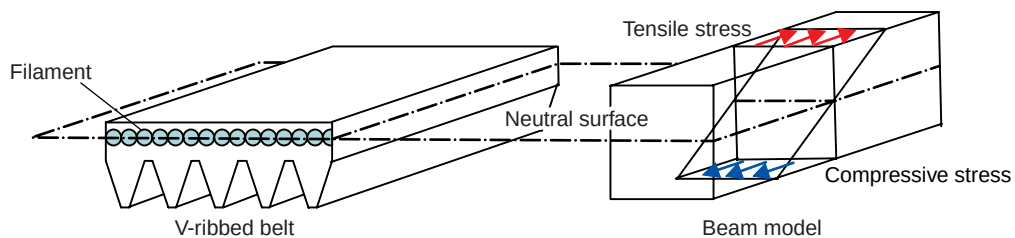


Fig. 2 Neutral surface of V-ribbed belt

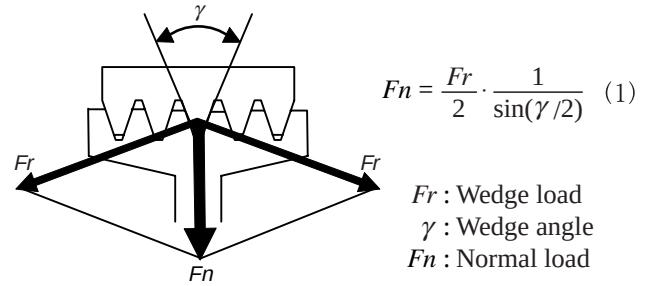


Fig. 3 Wedge load between belt and pulley

Compared to a flat belt, a V-ribbed belt generates high friction force due to the wedge effect when it engages with the pulley as shown in Fig. 3. Therefore, a friction force expressed by equation⁽¹⁾, which takes into account the wedge effect, is also assigned to the simulation. In addition, the coefficient of friction is assumed to be speed-dependent, increasing at low speeds to enable the stick-slip between the belt and the pulley to be reproduced. The coefficient of friction is computed from three parameters using interpolation Eq. (2)⁽⁶⁾. Figure 4 shows an example of a computation equation and the coefficient of friction.

$$\mu = A \cdot \sin \left(2 - \frac{2}{\pi} \cdot \sin^{-1} \left(\frac{B}{100} \right) \right) \cdot \tan^{-1} \left(S \cdot \frac{100}{C} \cdot \left(\tan \frac{\pi^2}{4 \cdot \left(\pi - \sin^{-1} \left(\frac{B}{100} \right) \right)} \right) \right) \right) \quad (2)$$

μ : Friction coefficient

S : Slippage

$A = \mu_{\max}$

$B = \mu_{100}$

$C = S_{\max}$

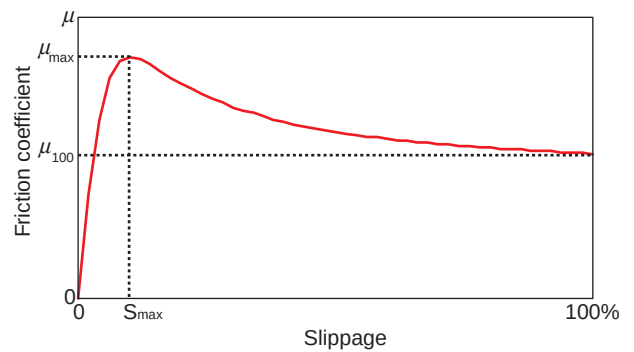


Fig. 4 Friction coefficient

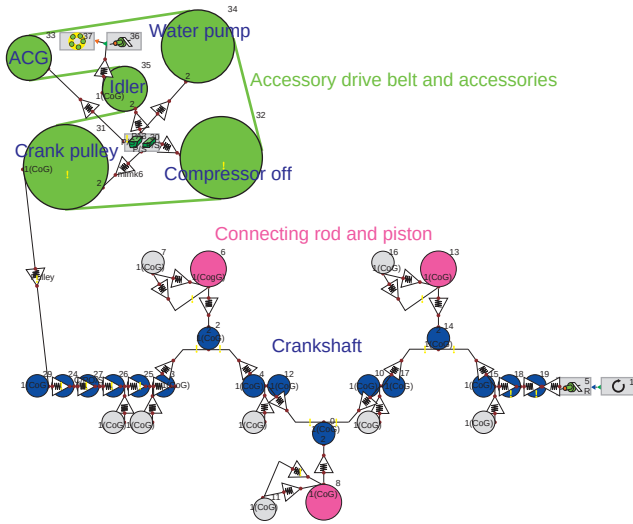


Fig. 5 Simulation model

Thus, the belt in the model is installed on the crankshaft pulley and drives multiple accessory pulleys. Each pulley in the model takes into account the inertia equivalent to each accessory. The load generated by each accessory is defined for each speed. The inertia and load of the pulley that drives the air conditioner compressor vary according to whether the compressor is operating or not. These parameters are also taken into consideration when determining the analysis conditions. To reproduce the crankshaft pulley's fluctuation in angular velocity, a model of the crankshaft that is driven by the cylinder pressure of the engine is created and used. Because fluctuation in the belt tension is caused mainly by fluctuation in the angular velocity of the crankshaft, the crankshaft model is a torsional model that takes into account only rotation. Figure 5 shows the simulation model.

3. Outline of the Test Equipment

The engine used to verify the simulation was a 660 cm³ inline 3-cylinder turbocharged gasoline engine. Figure 6 shows the layout of the installed accessory drive belt. A five-rib V-ribbed belt with a PET filament was used. The driven accessories were the alternator, the water pump and the air conditioner compressor. The belt tension

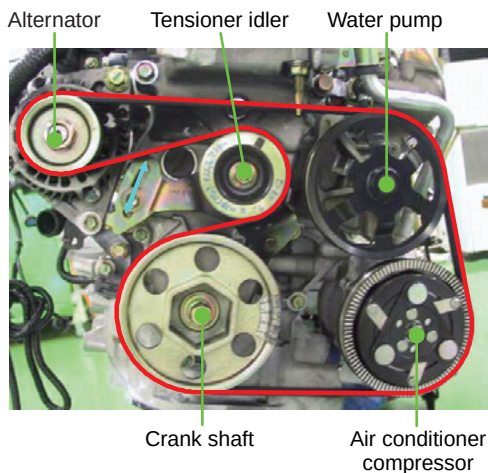


Fig. 6 Accessory drive layout

Table 1 General specifications of test

Engine type	Liquid cooled 3-cylinder turbo charged engine
Flywheel	Single mass flywheel
Crank pulley	Rigid pulley
Displacement	660 cm ³
Accessory drive belt	Serpentine V-ribbed belt
Filament of belt	Polyethylene terephthalate (PET)
Number of ribs	5 ribs
Material of rubber	Alkyl chloro sulfonated polyethylene rubber (A-CSM)
Tensioner	Fixed tensioner

was adjusted by changing the fixing position of the idler. This idler is hereafter called a fixed tensioner. The detailed test specifications are shown in Table 1.

While verifying belt slippage, it was necessary to measure the belt and crankshaft pulley velocities accurately and without making contact. Accordingly, these parameters were measured using a laser Doppler method wherein a laser beam is directed at the object of measurement and the velocity is calculated from the difference in wavelength between the incident and reflected waves. Figure 7 shows the measurement apparatus.

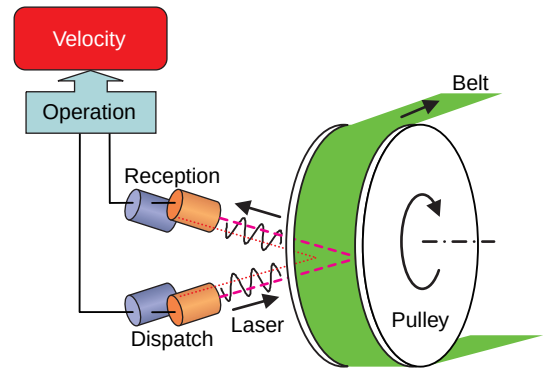


Fig. 7 Measurement of slippage

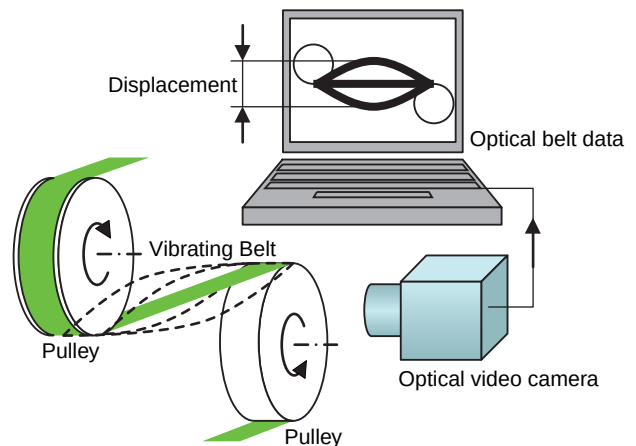


Fig. 8 Measurement of transverse vibration

It was also necessary to measure the displacement of the belt accurately and without making contact during the verification of transversal vibration, so the belt motion was measured using an optical displacement gauge. In this method, a photographed image of the belt was analyzed, and the displacement computed. Figure 8 shows an outline of the measurement apparatus.

4. Results of Verification

In the verification of the simulation, attention was focused on tension fluctuation, the slippage and the transversal vibration of the belt. The values of these items were difficult to predict in the case of a longitudinal vibration model because of their effect on the durability of the belt and because of the noise that was generated. Accordingly, verification was carried out according to the following three methods.

4.1. Belt Tension Fluctuation

Tension fluctuation is thought to be caused by the fluctuating angular velocity of the crank pulley and the resonance of the belt. In the case of the 3-cylinder engine used for the verification, the 1.5th order component was the main component of the fluctuating angular velocity of the crankshaft at low speeds where transversal vibration is liable to occur. This is because the explosion force is larger than the inertial force. Figure 9 shows the results of time series data verification of the fluctuation of angular velocity caused by the crankshaft's torsional vibration, and Fig. 10 shows the results of the analysis of the frequency of this vibration. In addition to this 1.5th order component, a 10.5th order angular velocity fluctuation component due to the 575 Hz torsional intrinsic frequency of the crankshaft was also reproduced. Because this high frequency component has a large effect on belt slippage, the prediction

accuracy must be high. The simulation measures the angular rotational fluctuation, including that in high frequency components, with high accuracy.

Next, attention was focused on the resonance of the belt. The belt resonance consists of longitudinal vibration, which is belt vibration in the direction of advance, and transversal vibration, which is perpendicular to the longitudinal vibration. First, to investigate the belt system's longitudinal vibration resonance, angular velocity fluctuations were applied in steps to the pulley of the crankshaft, and the response was investigated. The stepped angular velocity fluctuations generate impulse angular acceleration. Consequently, the same frequency response results as those of the impulse response were obtained. Figure 11 shows the results of frequency analysis of the angular velocity of the alternator pulley acquired due to acceleration. It can be seen that, of the several longitudinal vibration resonance modes possessed by this belt system, the resonant frequency showing the greatest amplitude is in the vicinity of 75 Hz. This is the primary mode in which shaking mainly by the alternator occurs due to the large equivalent inertia of the alternator. Belt tension fluctuation reaches a peak at a frequency of $75 \times 60 / 1.5 = 3\,000$ rpm, where the resonance coincides with the 1.5th order angular velocity of the crankshaft. Figure 12 shows the maximum value of the fluctuation of belt tension between the alternator and the fixed tensioner versus the rotational speed of the engine. This graph shows that the peak in the tension fluctuation occurs between 3 000 and 3 300 rpm for both the measured and simulated values, thus indicating that the simulation accurately reproduced the actual tension fluctuation. The causes of the peak at 3300 rpm are the closeness of the longitudinal vibration resonance to the primary component of the fluctuating angular velocity of the crankshaft and the closeness of the transversal resonance of 180 Hz between the alternator and the fixed tensioner to the 3rd order component of the fluctuation of crankshaft's angular velocity.

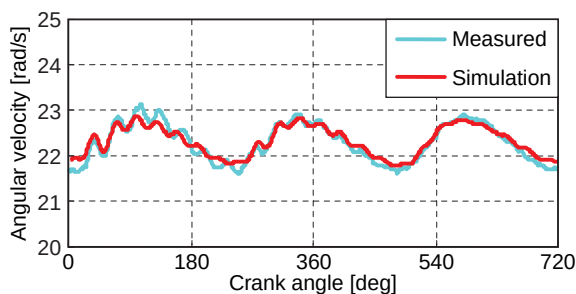


Fig. 9 Measured and simulated crankshaft speed fluctuation

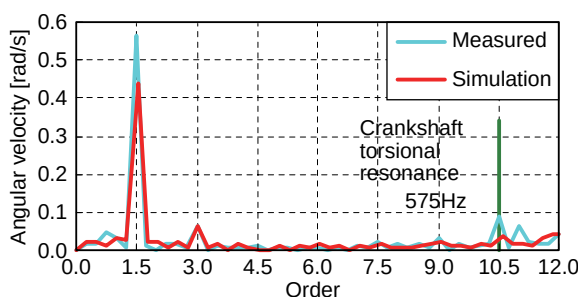


Fig.10 FFT result of crank shaft rotational speed

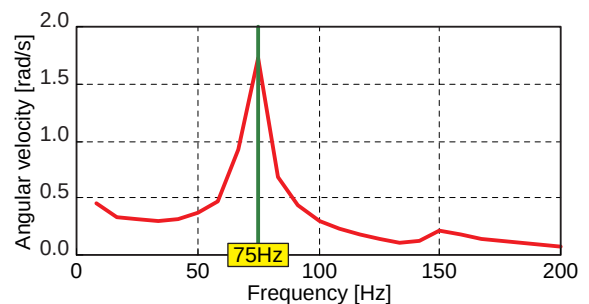


Fig. 11 Step response of angular velocity on alternator

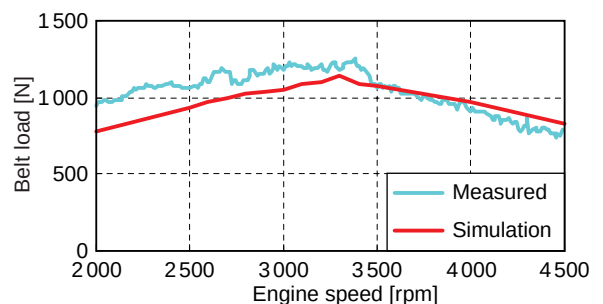


Fig. 12 Maximum value of belt load

4.2. Belt Slippage

Slippage between the belt and the pulley is a factor that has a large effect on the durability of the belt, generating unusual noise.

Excessively large slippage causes belt wear. If the static friction between the belt and the pulley is higher than the dynamic friction, stick-slip of the belt and also unusual noise are liable to occur. Accordingly, attention was focused on the belt slippage and a verification was performed.

The verification was carried out for a belt in which stick-slip occurred and unusual noise was emitted and for a normal belt in which stick-slip did not occur. The comparison was performed at 2 800 rpm, where fluctuations in tension and in angular velocity are large and stick-slip is likely to occur. The results are shown in Fig. 13. The figures on the left are the results for a belt in which stick-slip occurred, and the figures on the right are for a belt in which stick-slip did not occur. In the case of the belt in which stick-slip occurred, an instantaneous deceleration in the belt's angular velocity occurred in the vicinity of maximum speed fluctuation, causing the belt to slip. In the simulation as well, the belt slippage was reproduced at the same timing as that of the test results, and thus, it can be seen that the torsion vibration characteristics of the crankshaft affect stick-slip.

In the case of the belt in which stick-slip did not occur, the small amount of slippage in the vicinity of the point where the fluctuation of rotational speed was at its maximum and the smooth-slipped condition, were accurately reproduced. The coefficient of friction at this time is shown in Fig. 14.

It was possible to reproduce the actual phenomenon in a belt in which stick-slip occurred by applying a high coefficient of friction to the belt at a lower speed compared to that of a belt in which stick-slip does not occur. The coefficient of friction used here was a much higher value than that obtained from the results of general measurement. The conceivable cause of stick-slip lies in the setting

of the friction coefficient. From a computation standpoint, when the sign of the relative speed reverses, it is necessary to reverse the sign of the friction value as well, and when the speed is zero, the coefficient of friction is also zero. For this reason, it is necessary to apply a high friction coefficient at lower speeds to cause sticking to occur when the belt starts moving. On the other hand, in general measurement of the friction coefficient, the fluctuation in speed is less than that of actual stick-slip, and it is conceivable that the change in the friction coefficient at high frequencies is unknown. Consequently, it is considered appropriate when predicting stick-slip to use a friction coefficient that has been inferred so that the simulation agrees with the actual behavior.

4.3. Transversal Vibration

One conceivable cause of belt noise is transversal vibration of the belt. Accordingly, attention was focused on the part of the belt between the alternator and the fixed tensioner, where transversal vibration is liable to occur, and a verification of transversal vibration was carried out.

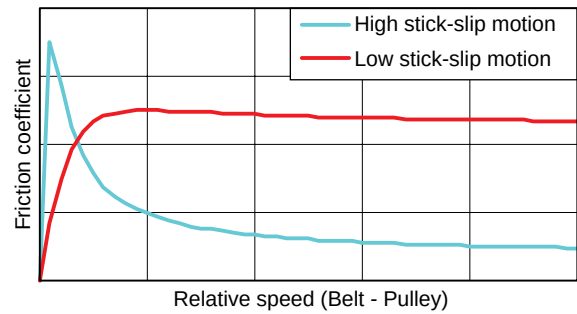


Fig. 14 Slip characteristics for investigated belt designs

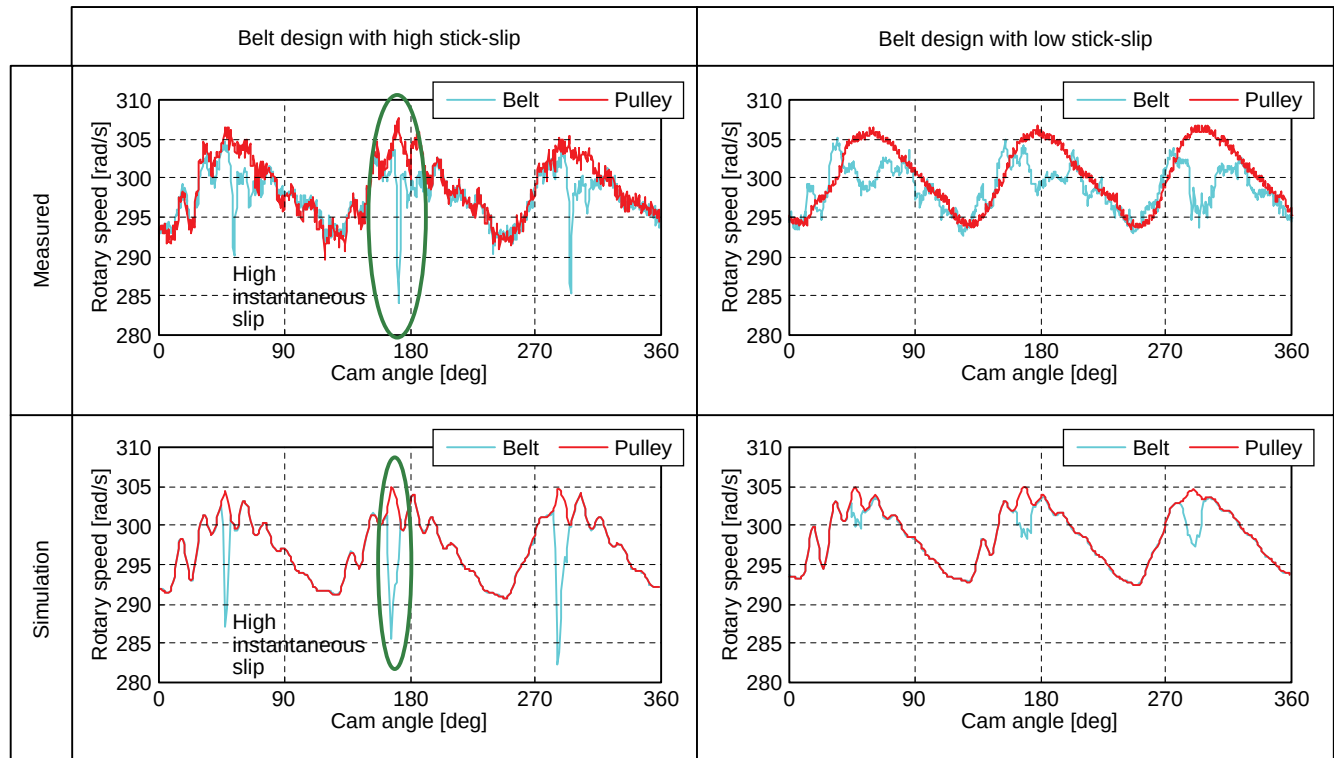


Fig. 13 Stick-slip motion at crankshaft pulley for two different belt designs

It is thought that transversal belt vibration is caused by tension fluctuation and transversal resonance. When the angular velocity fluctuation period of the crankshaft coincides with the belt system's longitudinal resonance, the tension fluctuation increases, and when the duration of this tension fluctuation coincides with that of the transversal vibration, transversal vibration is generated. Here, if the tension fluctuation is large with respect to preloading, the unstable region of the transversal vibration will increase, resulting in the generation of transversal vibration in the belt over a longer period of time. Other conceivable causes of increased transversal vibration include eccentricity of the pulley, friction force when the belt disengages from the pulley, belt inertial force, spring-back due to the belt's bend-rigidity, and engine vibration. Of these, belt inertial force and spring-back due to the belt's bend-rigidity are taken into account in the section-by-section model. Also, in the development of the simulation, predicting engine vibration at the initial stage of design was an important goal. For this reason, engine vibration that cannot be predicted unless the cylinder block layout is determined was not taken into consideration. Figure 15 shows computed results that do take the above factor into account.

At 2 600 rpm and 2 900 rpm, the simulation can accurately reproduce the actual transversal belt vibration. In addition, the results of analyzing the transversal belt vibration frequency at 2 900 rpm are shown in Fig. 16. Because the engine used in the verification was a 3-cylinder type, a $1.5n^{\text{th}}$ order angular velocity fluctuation of the crankshaft occurs. As a result, it can be seen that at 2 900 rpm, the $1.5n^{\text{th}}$ components (n is an integer), namely 72.5 Hz (1.5^{th} order), 145 Hz (3^{rd} order), 217.5 Hz (4.5^{th} order), 290 Hz (6^{th} order), and 362.5 Hz (7.5^{th} order) dominate transversal vibration. The transversal resonance of the moving belt is expressed by Eqs. (3) and (4) below⁽⁷⁾. The preloading during measurement was 300 N, so the transversal vibration frequency of the section of belt between the alternator and the fixed tensioner is $f_0 = 215$ Hz.

$$f_0 = \frac{1}{2L} \left(c_0 - \frac{v^2}{c_0} \right) \quad (3)$$

$$C_0 = \sqrt{\frac{T}{\rho}} \quad (4)$$

f_0 : Natural frequency of transverse vibration

L : Span of belt

v : Velocity of belt

c_0 : Transverse wave propagation speed

T : Tension

ρ : Belt mass per length

It is thought that representative tension fluctuation periods, at which transversal resonance occurs, are $2/3 \cdot f_0 = 143$ Hz, $f_0 = 215$ Hz, and $2 \cdot f_0 = 430$ Hz⁽⁷⁾. In fact, peaks can be observed in the vicinity of 143 Hz and 215 Hz. However, marked peaks can be seen at other frequencies as well. It is thought that this is due to the large tension fluctuation. Figure 17 shows tension fluctuation. As shown in the figure, if the fluctuation component is large compared to the mean tension, the region of transversal resonance will increase, and transversal vibration will occur over a wide frequency range. It is thought that this is the reason why the peak of transversal vibration occurred over a wide period between the 1.5^{th} and the 7.5^{th} orders.

It is thus possible to accurately reproduce the amplitude of the transversal vibration using this simulation model, which takes into account the respective excitation forces and resonances.

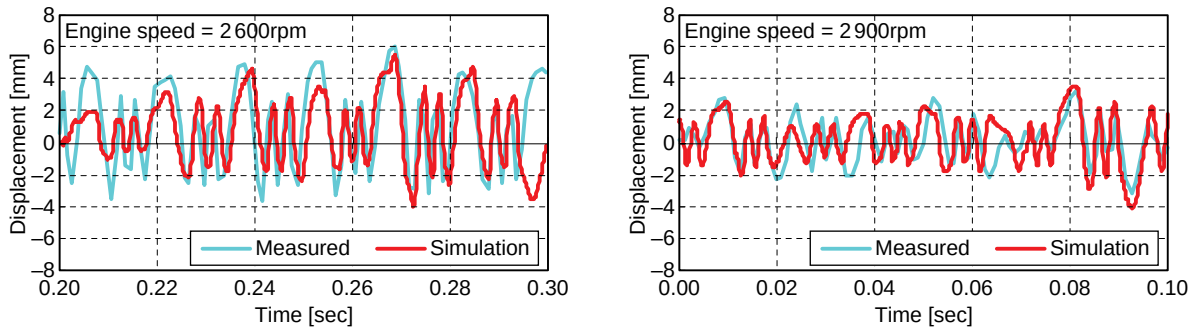


Fig. 15 Transverse belt motion

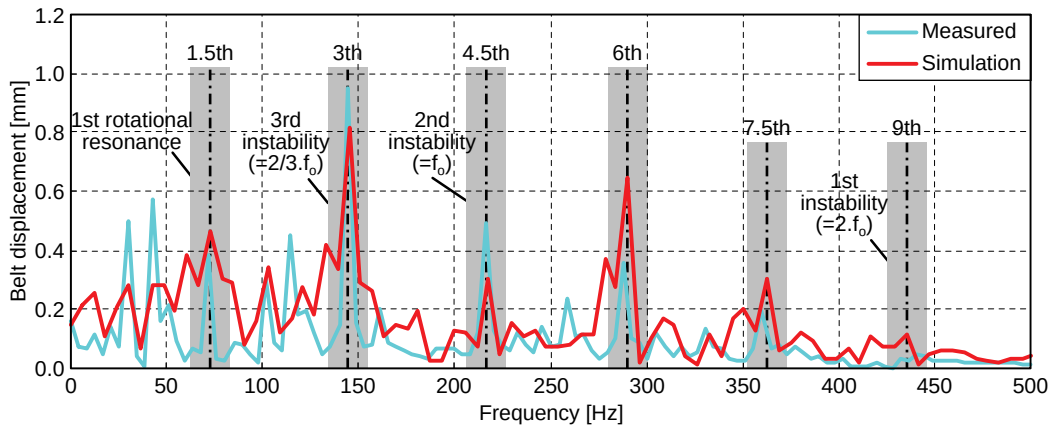


Fig. 16 Frequency contents transverse span vibration at 2 900 rpm (300 N belt pre-load)

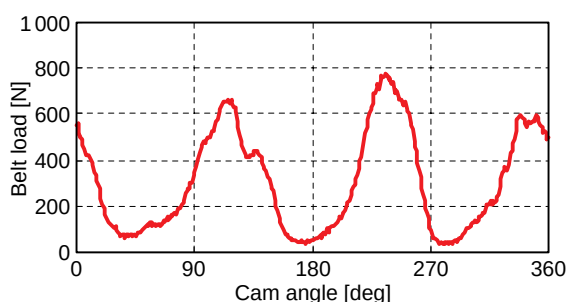


Fig. 17 Belt load between alternator and tensioner

5. Conclusion

The following results were obtained from the simulation using a section-by-section model, and technology for predicting the behavior of the belt was established.

- (1) A V-ribbed belt model, including a crankshaft, was created, enabling accurate reproduction of the fluctuations in angular velocity and belt tension.
- (2) By using a speed-dependent friction coefficient, it was possible to predict a wide range of slippage conditions, including the stick-slip condition.
- (3) By taking into account such factors as the eccentricity of the pulley and the frictional force when the belt starts disengaging from the pulley, it was possible to accurately predict transversal vibration of the V-ribbed belt.

By thus predicting the behavior of the accessory drive belt, it should be possible to establish the optimum belt layout at the design stage.

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Development of Parking Lock System Behavior Analysis

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ABSTRACT

Parking lock can occur when a driver inadvertently puts the automatic transmission lever into park while the vehicle is in motion. Prediction technology was developed to determine specifications that could help prevent the occurrence of parking lock. In order to understand the phenomenon and increase prediction accuracy, gap sensors were used to measure the behavior of the component elements of the parking lock device in an actual vehicle and parking lock was reproduced in a vehicle moving at high speed. In the prediction, multibody dynamics analysis was used to take into account collisions between component elements. The prediction accuracy was increased by identifying the restitution coefficient and the spring damping force. As a result, it was possible to predict the specifications that could help prevent parking lock from occurring while taking into account various layout restrictions at the investigation stage.

1. Introduction

Since automatic transmissions continue to become more compact and more economical, it has become necessary to cope with the reduced diameter, the parts sharing, and the diversification of part layouts in the parking lock device. It is also necessary to determine specifications in a short period of time due to the shorter development period.

Parking lock devices are intended to lock the drive system when the vehicle is stationary, however it is necessary to prevent the drive system from locking when the driver inadvertently shifts the gear into park while the vehicle is in motion. Previously, a qualitative trend of whether locking occurs or not when specifications are changed was already studied and the final specifications were matured by checking them through repetitive tests. But the tests need to be performed on an actual vehicle, and a large number of man-hours are required to check the large number of specifications. So it was necessary to develop a technology that could predict whether locking would occur with changes in specifications.

There is a report about the internal phenomena when a driver puts the lever into park while driving using prediction technology⁽¹⁾.

However, this technology was not intended for reproducing locking, but only for understanding the qualitative trend concerning the collision between the parking gear and the parking pawl.

In the development of the parking lock device, measurement technology for understanding the high-speed behavior of the component elements and prediction technology based on multibody dynamics analysis were developed. As a result, it was possible to quantitatively understand the effect of changing the specifications, enabling more efficient development of the device. The results are described below.

2. Development Goals

2.1. Description of the System

Figure 1 shows the position selector of the automatic transmission. The selector consists of a select lever on the driver side, a control shaft on the transmission side, and a wire joining these two components. The operation of the lever rotates the control shaft via the wire. The control shaft directs the parking lock device that mechanically locks the drive system when the select lever is put in the parking position (hereafter called the “P position”).

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The parking lock device is shown in Fig. 2. Honda adopts a lever-type mechanism on many of its models, as opposed to the cone mechanism used by other manufacturers. The motion of the parking brake lever (hereafter called “lever”) operates the parking pawl (hereafter called “pawl”) via a roller. The pawl engages with the gear, causing the drive system to lock. When the lever is in a position other than P, the pawl is pushed against the rotational control stopper side by the pawl spring. Also, to enable the lever to move to the P position regardless of the phase of the parking gear (hereafter called “gear”) when the vehicle is stationary, the lever is connected to the control shaft via the torsion coil spring as a waiting mechanism, as shown in Fig. 3.

2.2. Putting the Lever in the P Position while the Vehicle is in Motion

If the driver inadvertently puts the lever in the P position while the vehicle is in motion, the pawl of the parking lock device connected directly to the lever will attempt to engage with the rotating gear that is driven by the tires. However, the pawl will be repelled by the gear due to the abovementioned waiting mechanism, preventing engagement and thus preventing parking lock from occurring. Yet, it has been known from tests carried out during the development stage that if the setting of the specifications of the parking lock device is not appropriate, parking lock can occur at relatively high speeds.

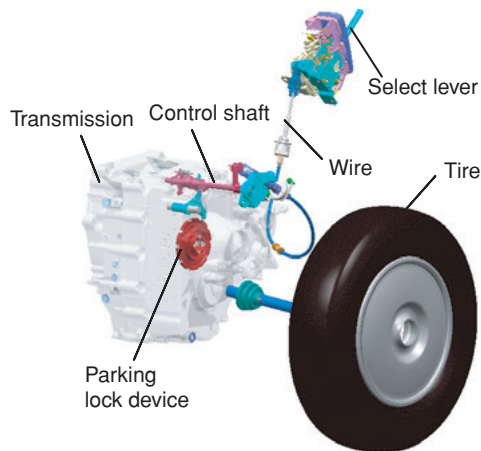


Fig. 1 Position selector

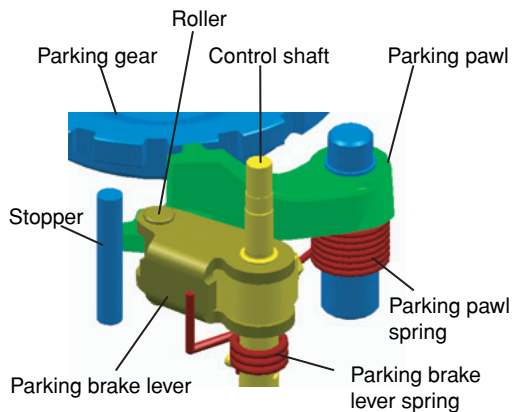


Fig. 2 Components of parking lock device

2.3. Issues Concerning Conventional Development Methods

Previously, the effect of changing the various specifications, including the number of teeth on the gear, the tooth space width, the chamfer angle, the angle and the corner size of the pawl ratchet, and the lever spring load were checked in tests that involved shifting the lever to the P position while the vehicle was in motion. The specifications were determined based on the results, however the effect of changing the various parameters in the parking lock device was not measured quantitatively. These parameters affected each other, so it was necessary to carry out a large number of tests in order to determine the specifications that would satisfy the requirements.

The tests have not been carried out on a bench but on an actual vehicle because of differences in the inertial mass and stiffness of the drive system. The only measurements performed were vehicle speed and acceleration, so it was not practical to evaluate the effectiveness of different specifications.

For this reason, it was necessary to devise a measurement method that would allow the effectiveness of different specifications to be evaluated. Devising the method required the establishment of a measurement technology that could identify the motion of the component elements of the parking lock device inside the transmission of an actual vehicle.

It was also necessary to develop prediction technology in order to cope with the size reduction and sharing of the component parts, along with the increasing compactness and reduced cost of an automatic transmission, the diversification of layout shown in Fig. 4, and the shorter development period.

Accordingly, development was carried out with the following goals.

- (1) Investigation of the mechanism of a parking lock in a moving vehicle
- (2) Development of technology for predicting parking lock in a moving vehicle

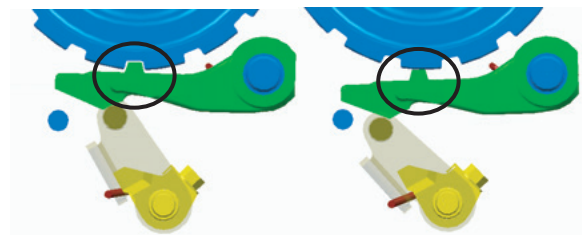


Fig. 3 Position of parking pawl and parking gear

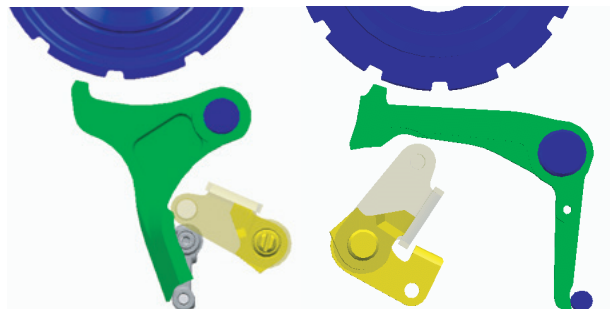


Fig. 4 Variations of parking lock devices

3. Testing and Analysis Technology

A method of measuring the behavior of the component elements of the parking lock device in the transmission of an actual moving vehicle when the lever is put into the P position was developed. As a result, it was possible to reproduce the mechanism that caused parking lock to occur while the vehicle was in motion and also quantitatively understand the effectiveness of each parameter.

3.1. Selection and Layout of the Sensors

The measurement items are the rotational behavior of the pawl, lever, gear, and control shaft, which are component elements of the parking lock device. When the lever is put in the P position while the vehicle is in motion, each element moves at extremely high speed. Therefore, it was necessary to devise a measurement method that responded quickly. This method also needed to permit high-resolution measurement of the rotation angle in order to quantitatively evaluate the engagement of the pawl with the gear. In addition, because measurement is performed on an actual vehicle rather than on a bench, the sensors had to be installed inside the transmission. Given these requirements, miniature gap sensors (eddy current type sensors) that have fast response times and high resolution were used.

Figure 5 shows the layout of the sensors in the transmission. The behavior of the pawl was measured by converting the distance detected by the gap sensors installed in the direction of rotation into an angle. The rotation angle of the lever without a stopper is large and the sensors cannot be installed in the direction of rotation, so the side of the lever was cut to make inclined face. The sensors

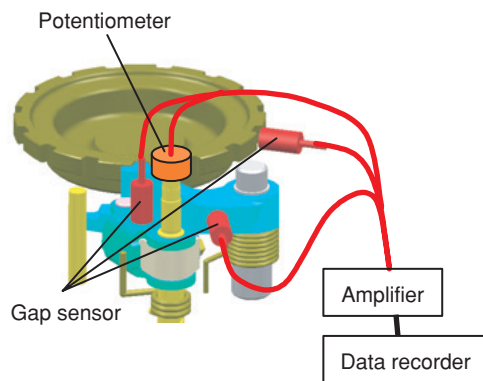


Fig. 5 Sensor layout inside transmission

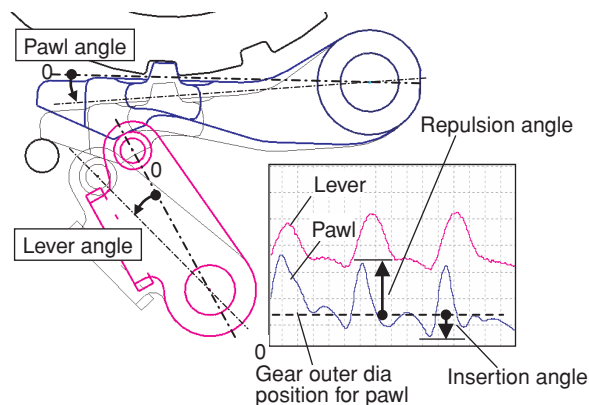


Fig. 6 Definition of pawl and lever angles

were installed in the direction perpendicular to rotation and measurements were performed. The phase of the tooth space of the gear was measured. Because the control shaft passes through the case to the outer side, its rotational angle was measured from outside using a rotary encoder. Figure 6 shows the definition of each angle of rotation and an example of the output. The angle of rotation in the locked condition when the lever is put in the P position is defined as 0.

3.2. Measurement Results

The rotational behavior of the lever and the pawl was measured using this method when the lever was put in the P position while the vehicle was in motion at a specified speed and those behavior after the lever was put in the P position until the vehicle stopped are shown in A of Fig. 7. The pawl and the gear collided with each other several thousand times. It was found that in a certain vehicle speed range (hereafter called "synchronized area"), the degree of engagement of the pawl and repel of the lever are extremely large. The behavior at each vehicle speed range is examined below.

Part B of Fig. 7 indicates the behavior during a period of 0.1 sec in the synchronized area. Rotation of the pawl is restricted between the stopper and the gear, so the pawl goes and returns between them due to being repelled by the gear. The lever is repelled by the pawl without rotational restriction, so the lever rotates largely. Consequently, the rotational behaviors of the lever and pawl are different and the lever is not driven together with the pawl.

Part C of Fig. 7 indicates the behavior during a period of 0.1 sec at a vehicle speed that is lower than the synchronized area. The rotational angle is lower and the motion of the pawl and the lever are the same, so the lever and pawl are driven together without separating from each other.

Figure 8 shows the measurement results of different specifications. Under these specifications, there is not a synchronized area, and the gear and pawl are driven together throughout the entire range.

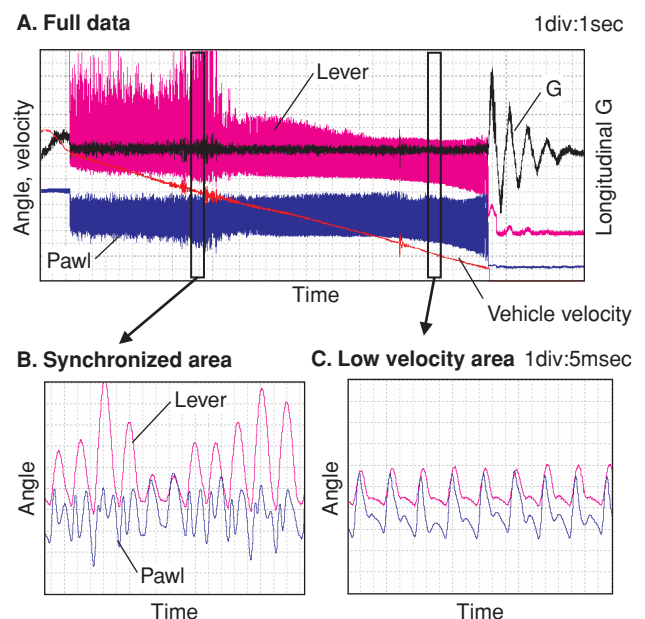


Fig. 7 Example of measurement result

3.3. Examination of the Repelling Behavior

There are two main reasons why the degree of repel differs according to the specifications and vehicle speed. The first is the contact angle of the pawl ratchet against the gear and the contact angle of the pawl against the roller of the lever, which determines the degree of repel. The degree of repel of the pawl depends upon the collision angle and the collision speed with the gear. For example, if the corner of the pawl ratchet collides with the chamfered part of the gear, as shown in A of Fig. 9, the collision angle will be large, therefore the component of the collision speed in the vertical direction at the collision face will be large, so the degree of repel will also be large. Conversely, if the end of the pawl ratchet collides with the chamfered corner of the gear, as shown in B of Fig. 9, the collision angle will be small, therefore the component of the collision speed in the vertical direction at the collision face will be small, so the degree of repel will also be small. Likewise, the degree of repel of the lever also depends upon the collision angle between the pawl and the roller of the lever.

The second reason is the number of tooth spaces, which determines the timing. If the pawl is repelled by the gear and then directly enters the tooth space when it returns to the gear, the degree of engagement will be larger than for the case where the pawl slides against the outer periphery of the gear, resulting in an extremely large degree of repel.

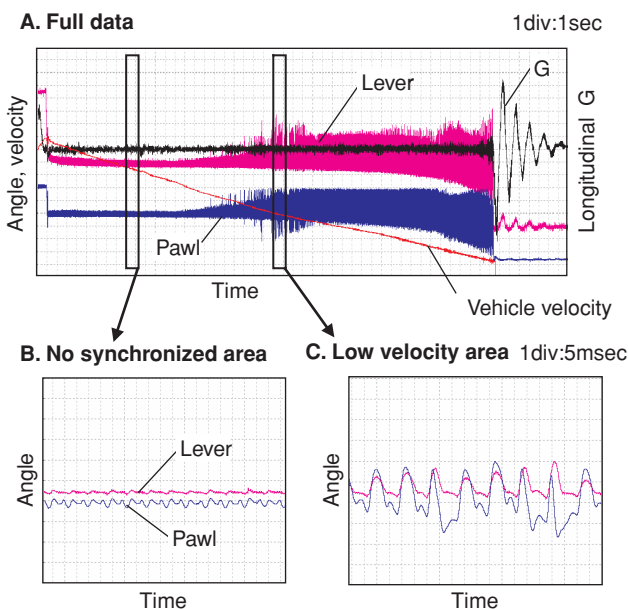


Fig. 8 Measurement result of other variation

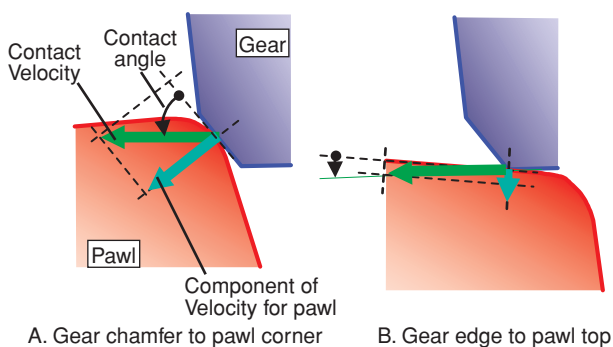


Fig. 9 Contact between gear and pawl

3.4. Mechanism of Parking Lock in a Moving Vehicle

Figure 10 shows the change in the collision angle of the lever with respect to the rotational angle of the pawl (degree of gear engagement). Parking lock occurs when the gear is unable to repel the pawl. This situation occurs if the roller goes past the corner on the lever side of the pawl, causing the collision angle to change greatly and virtually preventing a motion component in the rotational direction of the lever from being generated, as shown in B of Fig. 10.

However, when the vehicle is moving at high speed, the time during which the tooth space remains open for engagement with the pawl is short. So while the lever and the pawl are being driven together, the pawl cannot engage with the gear to the bottom of the tooth space under the push-in force of the lever.

In order to reproduce the mechanism by which the pawl engages with the gear to the abovementioned position in a vehicle traveling at high speed, the internal behavior when parking lock occurred was measured using specifications that were found to facilitate locking during the development process. Figure 11 shows the behavior up to several milliseconds prior to when locking occurs. The broken lines in the figure show timelines of gear tooth space, the figure in the frame of bold line shows behavior of a moment of locking, and behavior of this time is illustrated. When the return cycle of the pawl starts to match the gear tooth space cycle, the

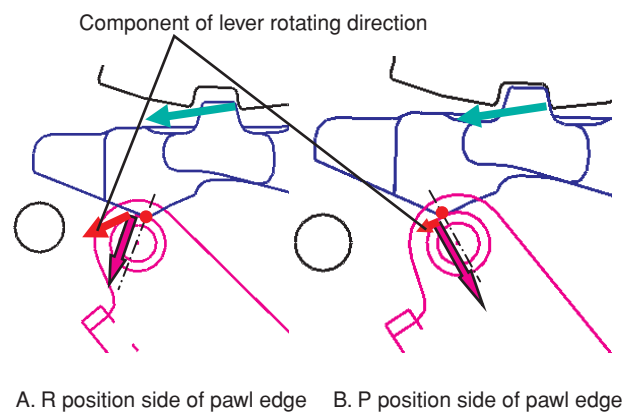


Fig. 10 Relation between lever and pawl

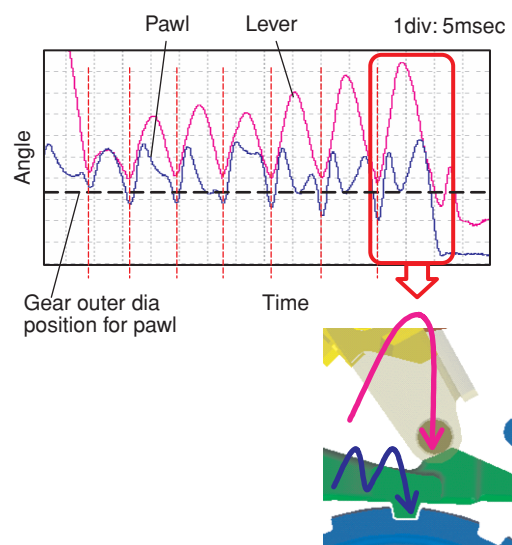


Fig. 11 Parking lock at high speed

degree of repel between the pawl and the lever becomes large. If the return cycle of the lever is also synchronized with this cycle, the pawl will be pushed to the bottom of the gear tooth space at high speed causing parking lock to occur.

As a result of measurement, the factors that greatly affected the mechanism were the rotational speed of the gear, the number of gear teeth, the lever spring load, the collision angle between the gear and the pawl, and the collision angle between the pawl and the lever.

4. Prediction Technology

Based on the characteristics obtained from measurement, a prediction model was constructed by multibody dynamics analysis. This model predicted the occurrence of parking lock when the lever was put in the P position during motion for different specifications.

4.1. Construction of a Prediction Method

From the measurement results obtained in Section 3.2., it was found that internal behavior was extremely complex, and was nonlinear due to the collective effect of the behavior of one component on the others. Consequently, in order to predict the occurrence of parking lock in a moving vehicle, it was necessary to reproduce this behavior taking into account the restriction on rotation and also the collision between the components. Accordingly, multibody dynamics analysis that permitted computation of the collision and rotational behavior at high speed was used.

Figure 12 shows the model used for the prediction. Although it is necessary to take account of the vehicle itself when creating a model for parking lock in a moving vehicle, the contribution analysis enables the model to be simplified to only the vicinity of the parking lock device and the moment of inertia of the drive train.

In the model, collision between each component is taken into account. In order to cope with the various conceivable collision forms, such as corner/face to face/corner, particularly in the case of a collision between the pawl and the gear, contact elements that can compute the contact between curved faces were used. When modeling was carried out, a very small radius was added to pin corners such as chamfered faces of the gear in order to avoid computation errors during the analysis.

Measurement revealed that when the pawl engaged with the gear, torsion was generated in the drive train resulting in a fluctuation of

rotational speed on the gear side. For this reason, the moment of inertia on the gear side was coupled to the moment of inertia on the tire side by a spring that was equivalent to the torsional rigidity of the drive train. The equivalent moment of inertia of the body was added to the tire side.

Similar to measurement, in analysis an initial rotational speed is given to the gear and tires, speed is reduced using a torque, the control shaft is rotated to the P position, and the behavior of parking lock in a moving vehicle is reproduced.

From the predicted results that are computed until the gear stop, the occurrence of parking lock is judged according to whether or not the engagement of the pawl corresponds to the position indicated in Section 3.4. above.

4.2. Increased Prediction Accuracy

4.2.1. Identifying the damping coefficient of the lever spring

The damping generated by the lever spring is due to friction between the spring coils and sliding between the spring and the shaft or lever. Because the damping depends upon velocity, it greatly affects the suppression of repelling behavior in this high-speed mechanism and it is essential to identify this damping in order to carry out predictions. It is necessary to identify this damping at a speed corresponding to the actual repelling condition, but because this speed is very high it is challenging to reproduce the specifications on an individual parking lock device. For this reason, a strain gauge was installed on the control shaft and the torque generated in the lever spring was measured when the lever was moved to the P position while the vehicle was in motion. The spring force computed from the measured angle of rotation and spring rate was subtracted from the abovementioned measured value, enabling the damping force and, in turn, the damping coefficient to be obtained.

4.2.2. Identifying the coefficient of restitution

The coefficient of restitution, which is taken into account in collision behavior, was obtained by performing a collision test on each individual component and reverse-computing the behavior by prediction. The value obtained was lower than the general value and it is supposed that it includes the effect of other parts of the system as well.

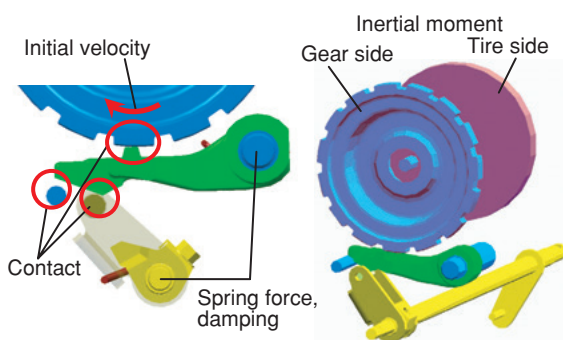


Fig. 12 Simulation model

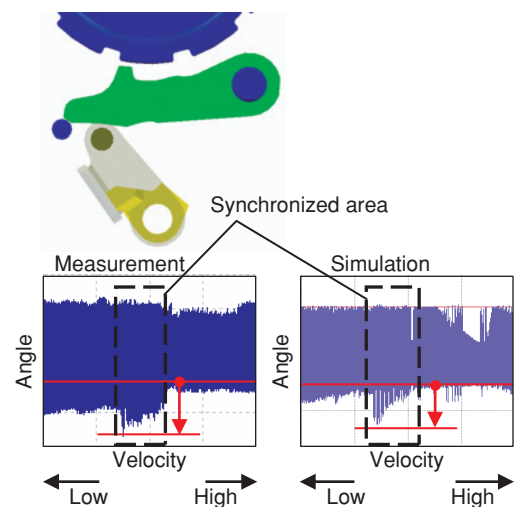


Fig. 13 Comparison between measurement and simulation

4.3. Verification of Prediction Accuracy

As an example, Fig. 13 shows the results of the measured results and the predicted results of the degree of pawl engagement versus vehicle speed. The figure in the frame of bold line shows synchronized area. The predicted engagement of the pawl with the gear and the synchronized velocity correlate well with the measured values.

5. Conclusion

The rotational behavior in the parking lock device of an automobile automatic transmission was made measurable. Based on this, the mechanism of occurrence of parking lock was reproduced and a prediction method was constructed.

The main points concerning this prediction method are as follows.

- (1) It was possible to measure the rotational behavior of the parking lock device using gap sensors in the case where the lever is put in the P position while the vehicle is in motion.
- (2) From the measurement results, it was possible to reproduce the mechanism where parking lock occurs. A high degree of engagement occurs in a vehicle moving at high speed when the cycle of the tooth space of the gear is synchronized with the cycle at which the pawl and lever are repelled by the gear and return.
- (3) A method of predicting parking lock in a moving vehicle was constructed by multibody dynamics analysis and the prediction accuracy was increased by using the damping coefficient of the brake spring obtained from the measurement data.

As a result of the above, it was possible to determine the specifications of the parking device that could help prevent the occurrence of parking lock in a moving vehicle while taking account of the various restrictions on installation.

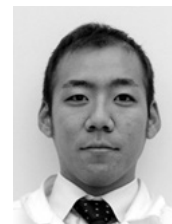
Reference

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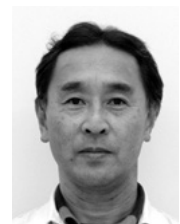
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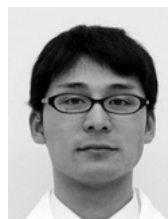
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Study of Radiation Noise for Motorcycle Engines

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ABSTRACT

In reducing the noises emitted by a motorcycle during acceleration, it is very important to reduce the noise radiated from the engine. To reduce engine noise radiation, we have developed an engine-noise-radiation analysis system. In analyzing engine-noise radiation, vibration analysis techniques appropriate for the high frequencies of the engine case and covers, a technique for estimating the excitation force input into the engine, and an analysis technique to assess acoustic output from the measured engine surface vibrations, are required. After overcoming all technical obstacles, our newly-developed system allows assessment of the noise radiation characteristics of motorcycle engines. The above-mentioned analysis method has been applied to reduce noise radiation from an inline four-cylinder motorcycle engine. The reduction of engine noises during acceleration has been confirmed from a modified engine.

1. Introduction

Reduction measures of intake and exhaust noise and drive train system noise in motorcycles have progressed; consequently, the proportion of engine noise in motorcycle noise sources has increased compared to the conventional models. In the future, measures of engine radiation noise reduction are necessary for enhancing quietness. In addition to radiation noises caused by excitation forces from combustion pressure, radiation noise caused by transmission gears is significant because the transmission is united with the crankcase. The frequency range of gear noise is approximately 3 kHz. The vibration mode of the case and cover surface often becomes an issue in the high frequency region.

In order to attenuate the vibration of the case and cover surface, countermeasures have included adding ribs, increasing thickness, and isolating the backside of the cover using a rubber plate pressed by a steel plate.

In conventional acoustic analyses, engine surface vibration velocities were measured using accelerometers and a laser vibrometer. Using general purpose acoustic analysis software, vibration velocity was analyzed to determine which parts have high acoustic radiation energy. The noise contribution rate of each engine component, which had been unknown up to now, can be analyzed in detail. Such analyses, however, have not led to any effective quantitative noise reduction countermeasures. Additionally, the countermeasures were accompanied by increases in weight and costs. For final assessment, it is necessary to quantitatively assess noise radiation by estimating the engine surface vibrations that are not based on actual engine vibration measurements.

As shown in Fig. 1, about 80% of motorcycle engine acoustic energy consists of frequency components more than 1 kHz, while about 50% have frequency components more than 2 kHz. Compared to other noise sources such as intake and exhaust noises, the acoustic energy of a motorcycle engine occupies a very high frequency band.

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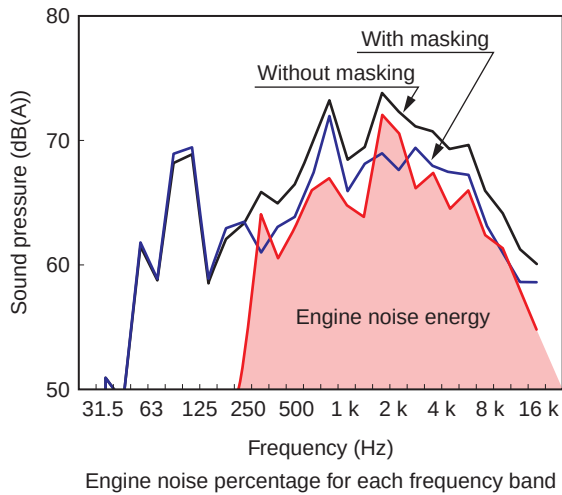


Fig. 1 Engine noise

The modeling method enhancing the analytical accuracy of individual parts and the optimizing method of connector characteristics for each part were applied. The finite element method (FEM) was used to create a structural model for radiation noise analysis. Additionally, it was considered whether the merits of radiation noises could be relatively compared using results that were calculated with the same excitation force applied to two models with modified specifications. As a result, it was found that applying excitation forces with frequency content close to the actual engine frequency characteristic was necessary to compare modified specifications, even in a relative comparison⁽¹⁾. Figure 2 shows the analysis flow of engine noise radiation used in this study.

2. FEM Models of Engine Case and Cover

FEM models of the engine case and cover were generated using the auto meshing functions of CATIA FEM on the CATIA-3D models. An example of the parts group comprising the motorcycle engine is shown in Fig. 3. The model size was determined

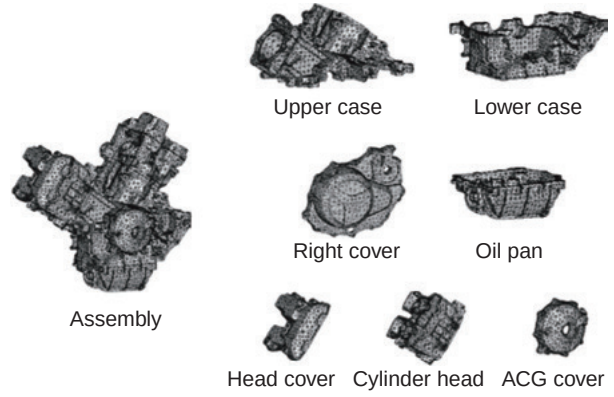


Fig. 3 Parts

(approx. 300,000 nodes) using the calculation resources that were available. The modeling accuracy was verified through the modal analysis of all parts using LMS' Gateway software. With reference to frequency errors, the Modal Assurance Criterion (MAC) value was used as a correlation index of modes. Based on this analysis, the mesh size for each important component and form was determined. For instance, optimization calculations were performed using the Young's modulus of the solid mesh as a parameter to match the calculated results and test results of motorcycle clutch cover. The change in Young's modulus is shown in Fig. 4. This figure suggests that the Young's modulus of the rounded area and the opened area should be reduced. Since the mesh sizes of those areas were found to be large, the overall mesh size and local mesh size were optimized. The change in the transfer function before

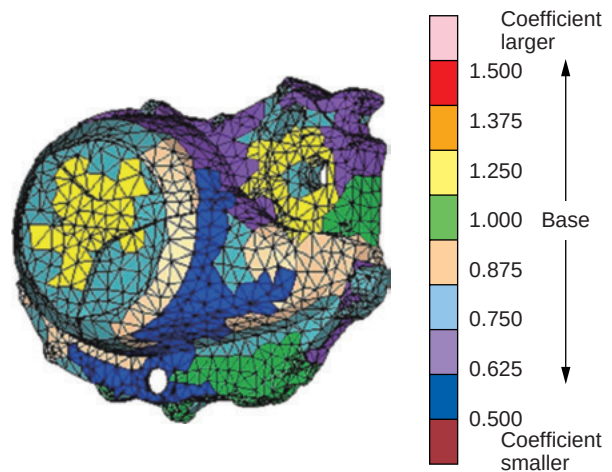


Fig. 4 Coefficient of Young's modulus

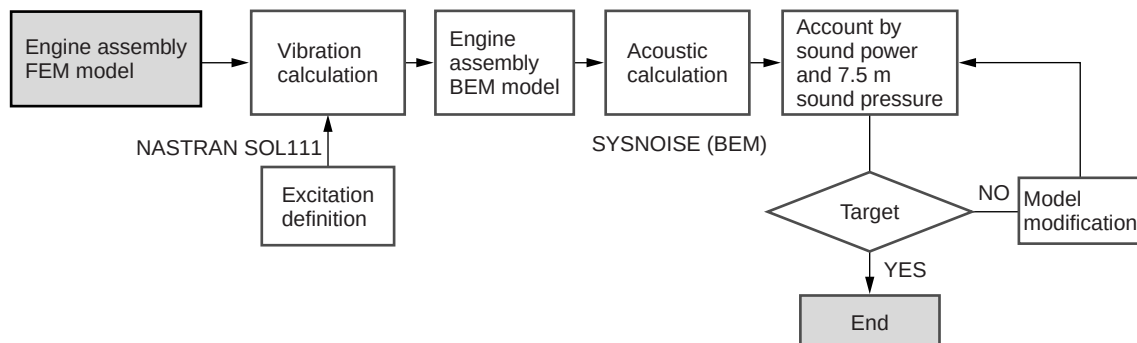


Fig. 2 Analysis flow

and after optimizing the mesh sizes is shown in Fig. 5.

The contact boundary surfaces between each part were calculated previously using GAP analysis. There is also a method of applying springs in parts contacting rigidly⁽²⁾. However, in this method, the meshes of contact boundary surface between each part are not matched. Each part is connected by using the RBE3 elements in NASTRAN at the bolt holes of each part. Each RBE3 element is connected by a six-directional spring element. The spring constant was determined through reference to the modal analysis results of the engine case and cover in their connected conditions. The values are all the same except for the rubber floating part of the cylinder head cover. Figure 6 shows a comparison of the transfer function from calculated results and tested results in the connected condition of the engine case and cover. The MAC value in the connected condition was greater than 0.5 up to a 1.5 kHz. The model accuracy was judged from the level of the transfer function because the mode density significantly increased above 1.5 kHz. Ensuring the accuracy of this analytical model is important because it is also used in identifying excitation forces from the bearing.

3. Estimation of the Engine Surface Acceleration and Excitation Forces from the Bearing during Actual Operation

Conventional techniques for estimating the excitation force acting on the bearing include techniques that estimate forces through the piston and crankshaft from the combustion pressure using EXCITE (AVL Inc.) and estimating the measured stress of bearing. However, the guaranteed accuracy of these techniques is below 1 kHz. Since combustion pressure is not the only cause of engine

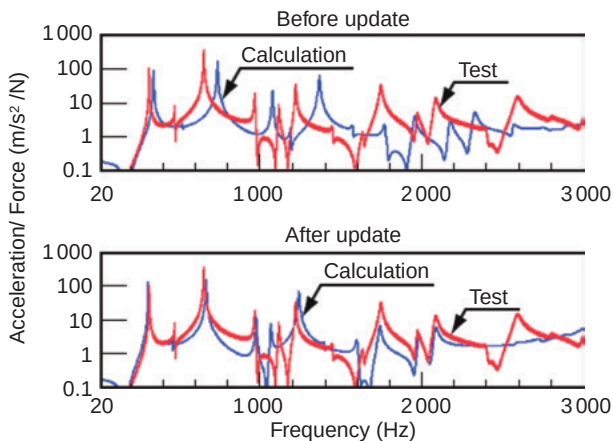


Fig. 5 Transfer function of right cover

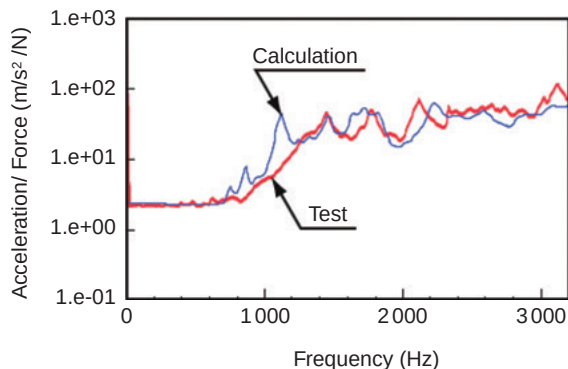


Fig. 6 Transfer function of engine assembly

vibration, these techniques were not used. In this study, vibration measurements were made at multiple points on the actual engine surface. The virtual bearing forces were estimated by the inverse matrix method^{(3), (4)}. The number of bearing excitation points needed to accurately express the vibration of each engine part was studied. It was confirmed that at least three excitation points were needed to accurately express engine behavior. Based experimentally on the number of points that can be measured simultaneously, two virtual bearing excitation points were applied on the crankshaft in addition to two points on the transmission unit. Virtual excitation forces with a total of 12 degrees of freedom were used since four points were taken in three directions. Finally, virtual excitation forces with 12 degrees of freedom were determined using the method of least squares based on the results of simultaneous measurements of 48 degrees of freedom. Figure 7 illustrates the principles. The relationship between transfer function, the excitation force and acceleration response is expressed by Eq. (1). Conversely, the excitation force can also be expressed more clearly as acceleration multiplied by the inverse of the transfer function (Eq. (2)). After determining $12 \times 48 = 576$ transfer functions by structural analysis, 12 excitation forces were determined from accelerations in 48 simultaneous measurements.

$$A = H * F \quad (1)$$

$$F = H^{-1} * A \quad (2)$$

For the inverse matrix calculations, Matlab software (The Math Works, Inc.) was used. Figure 8 shows the estimation technique concept. Only measurement points on the upper and lower cases of engine near the points at which virtual excitation forces act were used. Figure 9 shows an example of the results obtained from the vibration calculations using the excitation forces that were determined. These results are for a V-twin, 1 000-cm³ engine running at approximately 2 800 r/min. It can be observed that the predictions are for a frequency range up to 3 kHz.

4. Results of Verification of Engine Noise Radiation Predictions

From the vibration results found in Section 3, acoustic analysis was performed using a SYSNOISE direct BEM approach. Conversely, engine surface vibration measurements at about 700 points were made during actual operation using a dynamometer. Acoustic analysis was performed using the measured and calculated results on the same model. Calculations could be performed to an accuracy of within 3 dB sound power. Figure 10 shows the results of the analysis verification. Figure 11 shows the comparison of surface sound pressure distributions, expressed as peak sound power. It was determined that both surface sound pressure levels

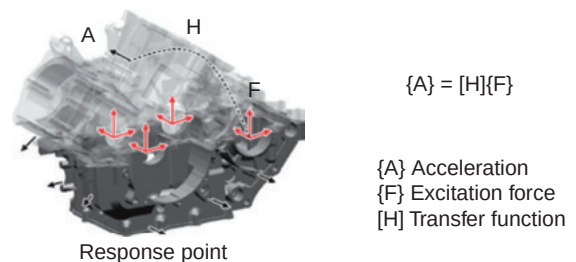
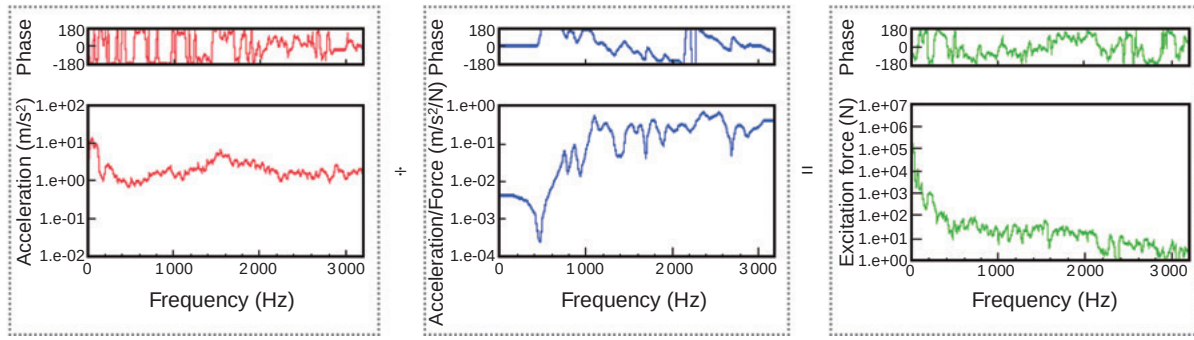


Fig. 7 Definition of excitation force



$$\text{Acceleration (Test)} + \text{Transfer function (CAE)} = \text{Excitation force (Estimation)}$$

Fig. 8 Estimation of excitation force

and distributions were estimated accurately. From the calculations, a significant amount of noise is radiated from the clutch cover and the cam chain chamber.

5. Examples of Actual Motorcycle Applications

A practical application of this analysis method to an in-line four-cylinder 1 000 cm³ engine is shown here. Figure 12 shows the engine 3D model. To identify excitation forces, vibration measurements were made in three directions at 16 points on the upper/lower cases of engine running at approximately 4 900 r/min.

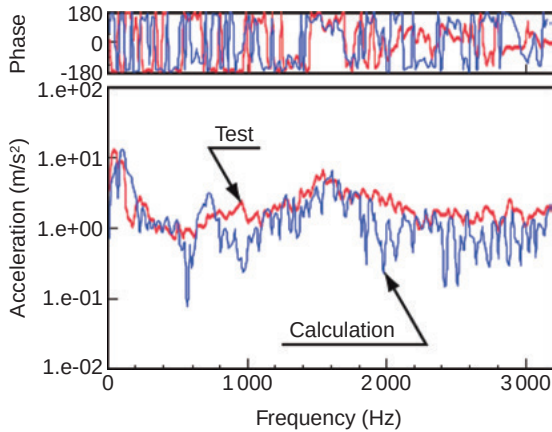


Fig. 9 Comparison of responses between test and calculation

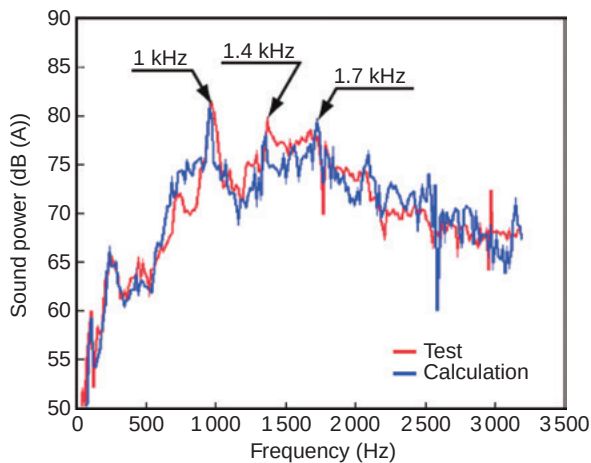


Fig. 10 Comparison of sound powers between test and calculation

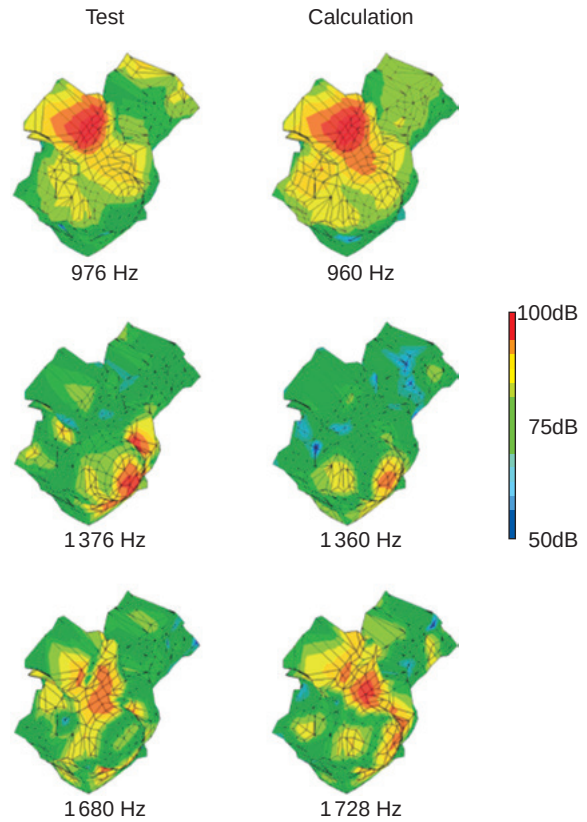


Fig. 11 Comparison of sound pressure distributions between test and calculation

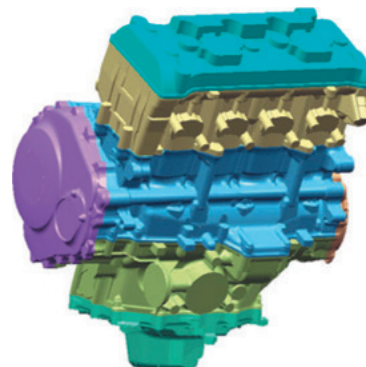


Fig. 12 Engine 3D model

Figure 13 shows the surface acceleration measurements obtained. Figure 14 shows the results of virtual excitation forces that were calculated based on the measured results. As observed from Figs. 13 and 14, the excitation frequencies during gear meshing contributed greatly to vibration, unlike the example of the V-twin 1 000 cm³ engine (Fig. 9). Acoustic analysis was performed for

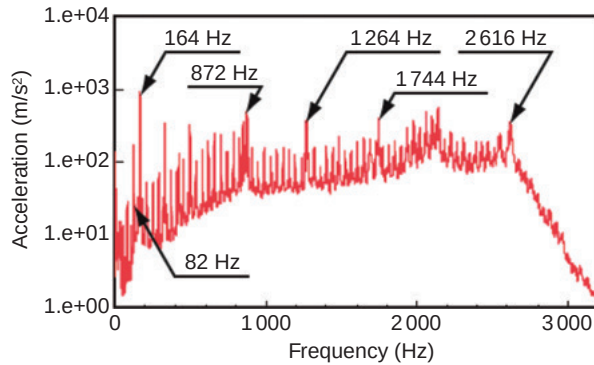


Fig. 13 Measured surface vibration

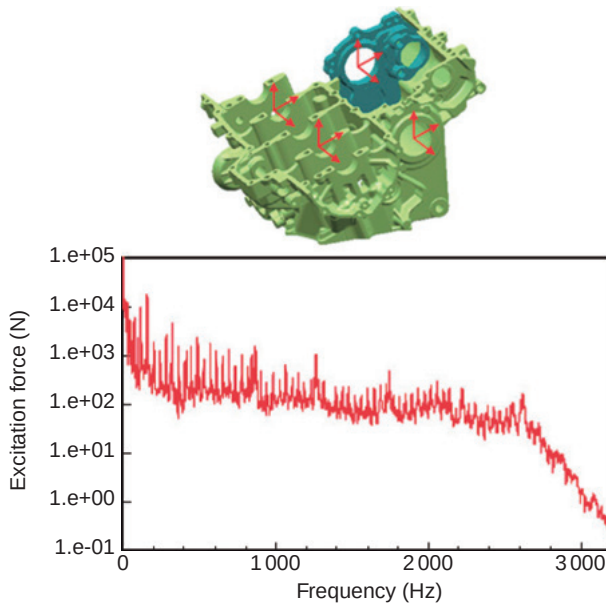


Fig. 14 Estimated excitation force

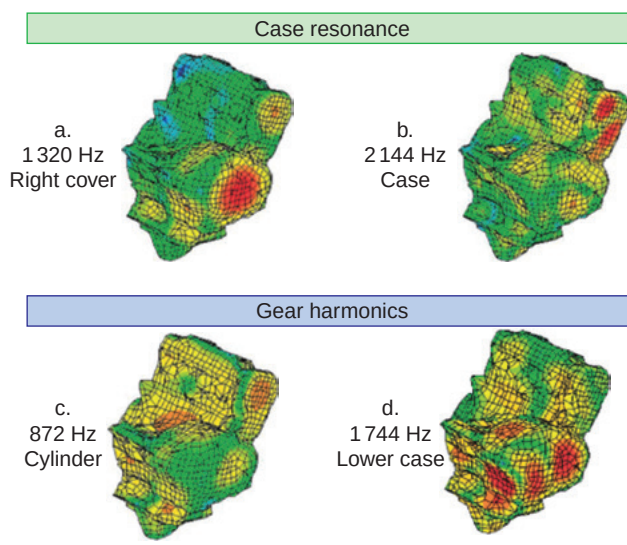


Fig. 15 Calculation result (sound power and surface pressure distribution)

the results mentioned above. Figure 15 shows the surface sound pressure level distribution at peak frequencies of sound power on the engine. This shows that the noise radiation parts vary according to frequencies. Countermeasures for the above were studied (Fig. 16). Ribs were added and board and wall thicknesses were increased. The changes in calculated sound power were plotted as shown in Fig. 17. An approximate 3 dB reduction was obtained in

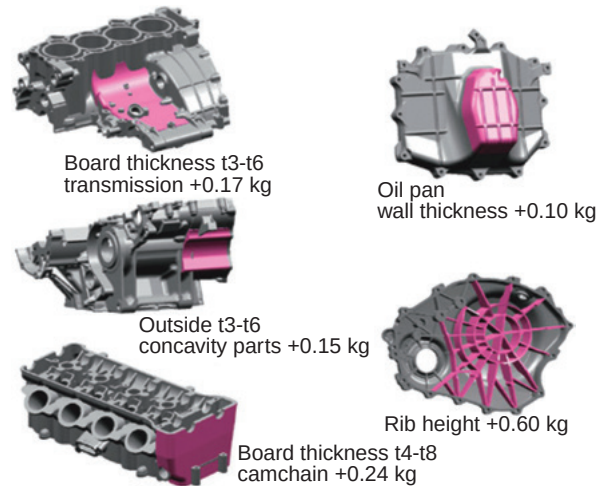


Fig. 16 Modified case and cover

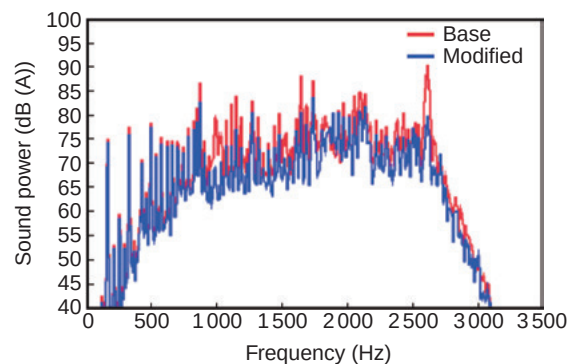
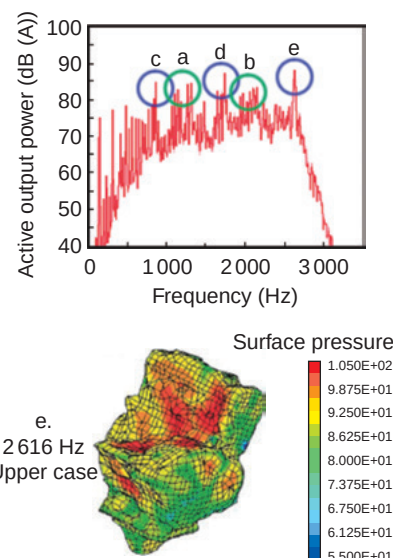


Fig. 17 Comparison of sound powers between base and modified case and cover



the overall sound radiating power. Figure 18 shows the changes in the surface sound pressure distribution before and after adopting the countermeasures. To confirm the effects of these countermeasures practically, a modified engine case and cover was prepared through the use of sand molding. The effect of the countermeasures was confirmed by assembly. Running tests on four different sets of specifications were performed, shown by the circles in Fig. 19. As indicated in Fig. 20, a correlation could be seen between the reduction in calculated sound power and the reduction in the pass-by noise level. The pass-by noise amplitude

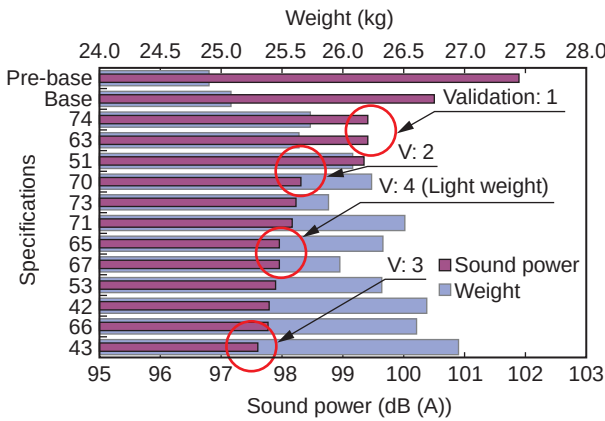


Fig. 19 Sound power versus weight

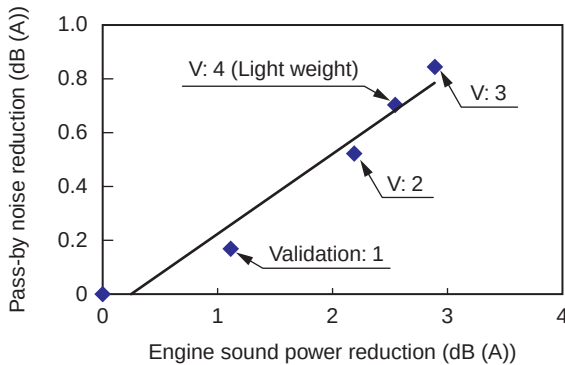


Fig. 20 Effect of engine sound power reduction

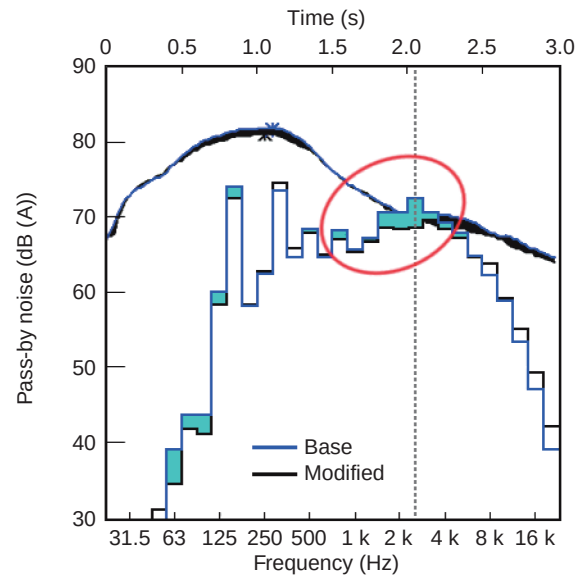


Fig. 21 Pass-by noise

at frequencies caused by engine noise was lowered, as shown in Fig. 21. The pass-by noise was reduced by approximately 0.8 dB. A reasonable result, even when observed from the perspective of the engine's contribution to pass-by noise.

The reduction technique used can be grasped through experience as well as understood through the analyses performed as follows.

- By making sure that there is no coincidence between the resonant frequency of each part and its gear meshing frequency.
- By dispersing the resonant frequencies of various parts such that the peaks do not overlap.
- By increasing the rigidity of each weak point to reduce vibration levels.
- By trying to increase the first resonance frequency as much as possible for gear meshing frequencies below 1 kHz.
- By adding mass for lowering the base response level.

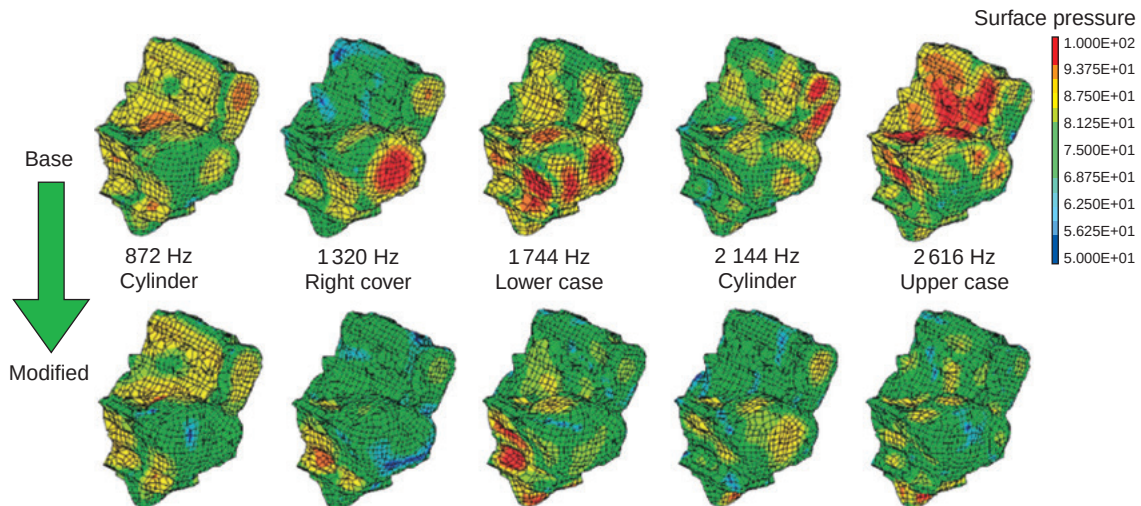


Fig. 18 Comparison of sound pressure distributions between base and modified case and cover

6. Conclusions

- (1) Conventional countermeasures against noise radiation were generally implemented through trial and error by identifying the vibration-producing parts using a stethoscope or accelerometer. However, with the authors' new technique, quantitative judgments on the effect of countermeasures can be made theoretically.
- (2) After preparing modified cases and covers, results verifying pass-by noise reduction were obtained according to the measures adopted. The effectiveness of the accompanying noise radiation analysis was verified.

Simulations were performed using measurement results of bearing forces on the actual engine. In the future, we plan to incorporate parts in the engine, estimate bearing forces that are due to meshing forces from gear transmission errors or combustion pressures, and calculate the noise radiation in the drawing stage.

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Hiroyuki KUWAHARA



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Research on Engine Mount Characteristics Identification of Outboard Motor Using Experimental Modal Analysis

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ABSTRACT

Research was conducted on a method which can directly determine the engine mount dynamic stiffness in an outboard motor to gain a better understanding of the engine mount vibration characteristics of an outboard motor.

First, a method was established to identify the dynamic stiffness of the engine mount in a physical coordinate system using modal parameters acquired from experimental modal analysis. Using the calculation results obtained by the finite element method applied to the outboard motor model, it was revealed that this method has the ability to identify dynamic stiffness with high accuracy.

Next, the method was applied to two outboard motor models, the small BF5 (3.7 kW) and the large BF225 (165 kW). The dynamic stiffness of the engine mount was identified by the results of experimental modal analysis and it was compared to the result obtained by the conventional method using the dynamic stiffness measurement equipment. The results show that this method can identify the engine mount dynamic stiffness more adequately than the conventional method, even when the engine mounts are subjected to loads corresponding to thrust force or even in the case that the stiffness of the parts supporting an outboard motor is low.

1. Introduction

Outboard motor engines range from small single-cylinder 2 kW class engines to large V-type six cylinder engines in excess of 180 kW, and are mounted on various types of boats that differ in weight, shape and structure around the world. The excitation force produced by these engines varies widely for each engine type, while boats also have diverse vibration systems and acoustic fields, so the vibration and noise generated by the compound system of the outboard motor and the boat are almost infinite in variety. This is an important difference from the issue of noise and vibration in products such as motorcycles and cars, where the engine and chassis

are developed as a set, and that difference makes analysis and countermeasure development difficult.

On the other hand, while the absolute value of vibration and noise must be reduced as in motorcycles and cars, with boats there are also strong demands to realize more comfortable noise and vibration. This makes it important not only to reduce the vibration and noise of the outboard motor, but also to study the way of installing the outboard motor for better noise and vibration. Among efforts to reduce outboard motor vibration and optimize the engine mounts, the reduction of outboard motor vibration using a pendulum motion mount method has been reported⁽¹⁾. However, analysis and countermeasures for vibration and noise that include the boat must

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be conducted individually for each case due to the reasons mentioned above. This makes systematic research difficult, and as a result there are few reported cases.

Transfer path analysis is known as a technique for analyzing vibration and noise in complex structural systems such as those in boats^{(2),(3)}. This technique analyzes how force is transferred within an actual structure such as a car. Typical methods for identifying the transmitted force include the dynamic stiffness method, which uses a dynamic spring rate⁽⁴⁾, and the inverse matrix method, which uses an inverse matrix of the transfer function. The dynamic stiffness method directly uses the dynamic stiffness of the physical coordinate system so the effects of changing the structural characteristics and geometry can be easily handled. However, this method is also easily influenced by the stiffness of the mounting frame and other parts that support the outboard motor (outboard motor support parts) and nonlinear characteristics, so it is difficult to achieve accuracy. In contrast, the inverse matrix method tends to be more accurate, but has difficulty handling the effects of changing the structural characteristics and geometry.

A highly accurate transfer path analysis that can easily handle various combinations of outboard motors and boats is required to analyze and implement measures for the vibration and noise of compound systems consisting of outboard motors and boats. For this reason, an attempt was made to experimentally clarify the dynamic characteristics of outboard motor engine mounts⁽⁵⁾. This method consists of performing experimental modal analysis on an outboard motor in the assembled condition, and directly calculating the dynamic stiffness of the physical coordinate system. Establishing this method would make it possible to obtain the dynamic stiffness of engine mounts when the outboard motor's weight and thrust force are applied. This would in turn provide a more accurate understanding of the forces transmitted from the engine to the boat, which would allow more accurate vibration and noise analysis.

2. Engine Mount Dynamic Stiffness Identification Method

Figure 1 shows an outline of the engine mount dynamic stiffness

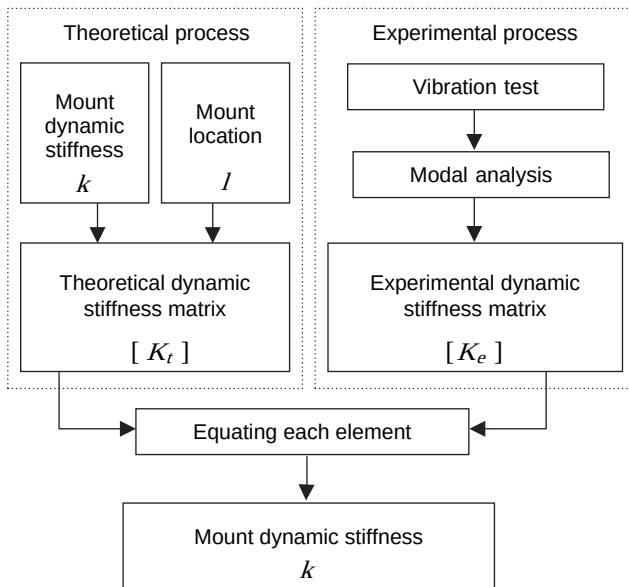


Fig. 1 Outline of engine mount dynamic stiffness identification method

identification method. Here, the dynamic stiffness matrix of the vibration system is calculated both from theoretical equations and through experiments. The dynamic stiffness of each engine mount is then identified by solving the simultaneous equations in which both matrix components are equivalent.

2.1. Calculation of Theoretical Dynamic Stiffness Matrix

The rubber mounts were modeled as shown in Fig. 2. Each rubber piece was divided into the dynamic stiffness for four locations, and the translational stiffness was defined in three directions (k_x, k_y, k_z) for each location. This made it possible to express the rotational stiffness of the mount as translational stiffness.

In addition, both the outboard motor and the mounting frame parts that support the outboard motor are assumed to have a higher stiffness than the rubber mounts. By modeling the parts other than the rubber mounts as a rigid body, the outboard motor supported by the rubber mounts becomes a system with six degrees of freedom. Therefore, the static equilibrium equations can be written as shown in Eq. (1).

$$[K_t] \{u_o\} = \{f\} \quad (1)$$

Where,

$$\{u_o\} = \{x_o \ y_o \ z_o \ \theta_x \ \theta_y \ \theta_z\}^T$$

$$\{f\} = \{f_x \ f_y \ f_z \ T_x \ T_y \ T_z\}^T$$

u_o is the displacement and f is the force.

In addition, the theoretical dynamic stiffness matrix $[K_t]$ can be expressed as shown in Eq. (2).

$$[K_t] = \begin{bmatrix} K_{11} & & & & & \\ 0 & K_{22} & & & & \\ 0 & 0 & K_{33} & & & \\ 0 & K_{42} & K_{43} & K_{44} & & \\ K_{51} & 0 & K_{53} & K_{54} & K_{55} & \\ K_{61} & K_{62} & 0 & K_{64} & K_{65} & K_{66} \end{bmatrix} \quad \text{SYM.} \quad (2)$$

Where,

$$K_{11} = \sum_{i=1}^n k_{xi}, \quad K_{22} = \sum_{i=1}^n k_{yi}, \quad K_{33} = \sum_{i=1}^n k_{zi}$$

$$K_{42} = - \sum_{i=1}^n l_{zi} k_{yi}, \quad K_{43} = \sum_{i=1}^n l_{yi} k_{zi}$$

$$K_{51} = \sum_{i=1}^n l_{zi} k_{xi}, \quad K_{53} = - \sum_{i=1}^n l_{xi} k_{zi}$$

$$K_{61} = - \sum_{i=1}^n l_{yi} k_{xi}, \quad K_{62} = \sum_{i=1}^n l_{xi} k_{yi}$$

$$K_{44} = \sum_{i=1}^n (l_{zi}^2 k_{yi} + l_{yi}^2 k_{zi}), \quad K_{54} = - \sum_{i=1}^n l_{xi} l_{yi} k_{zi}$$

$$K_{55} = \sum_{i=1}^n (l_{zi}^2 k_{xi} + l_{xi}^2 k_{zi}), \quad K_{64} = - \sum_{i=1}^n l_{xi} l_{zi} k_{yi}$$

$$K_{65} = - \sum_{i=1}^n l_{yi} l_{zi} k_{xi}, \quad K_{66} = \sum_{i=1}^n (l_{yi}^2 k_{xi} + l_{xi}^2 k_{yi})$$

Here, K is the dynamic stiffness of the rubber mount, l is the mount position, n is the number of mounts, and the subscript i indicates the mount number. In other words, the theoretical dynamic stiffness matrix $[K_t]$ can be formulated as shown in Eq. (2) by using the rubber mount stiffness k , which is an unknown.

2.2. Calculation of Experimental Dynamic Stiffness Matrix

Next the method for calculating the stiffness matrix $[K_e]$ from the vibration test results is described below.

First, an outboard motor fixed as shown in Fig. 3 is excited, and the natural frequencies and mode shape vectors are obtained for the initial six modes in which the outboard motor vibrates more or less as a rigid body. Assuming the outboard motor to be a rigid body, the mode shape vector for a certain mode can be expressed by three translational components and three rotational components for a total of six degrees of freedom components. Equation (3) shows the relationship between the normalized mode shape vector $\{\phi_i\}$ at the measurement point i and the six degree of freedom components $\{\phi_o\}$ that represent the rigid body movement.

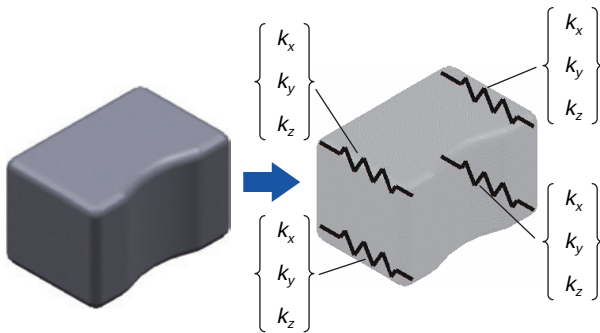


Fig. 2 Modeling of rubber mount



Fig. 3 Outboard motor BF5

$$\{\phi_i\} = [T_i] \{\phi_o\} \quad (3)$$

Where,

$$\{\phi_i\} = \{\phi_{xi} \quad \phi_{yi} \quad \phi_{zi}\}^T$$

$$[T_i] = \begin{bmatrix} 1 & 0 & 0 & 0 & z_i & -y_i \\ 0 & 1 & 0 & -z_i & 0 & x_i \\ 0 & 0 & 1 & y_i & -x_i & 0 \end{bmatrix}$$

$$\{\phi_o\} = \{\phi_{xo} \quad \phi_{yo} \quad \phi_{zo} \quad \phi_{\theta_x} \quad \phi_{\theta_y} \quad \phi_{\theta_z}\}^T$$

Here, $[T_i]$ is the coordinate transformation matrix. x , y and z are the measurement point positions, and i indicates the measurement point number. Equation (3) can be integrated to Eq. (4) for all measurement points. Here, P is the number of measurement points.

$$\{\phi\}_{3P \times 1} = [T]_{3P \times 6} \{\phi_o\}_{6 \times 1} \quad (4)$$

This equation is then transformed to calculate the rigid body mode shape vector $\{\phi_o\}$ as follows.

$$\{\phi_o\} = ([T]^T [T])^{-1} [T]^T \{\phi\} \quad (5)$$

Performing the operations in Eq. (5) for all six modes yields the 6×6 rigid body mode shape vector $[\Phi_o]$.

Finally, the experimental dynamic stiffness matrix $[K_e]$ is calculated from the rigid body mode shape vector $[\Phi_o]$ and the matrix $[\Lambda]$, which is a diagonal matrix of the square of the undamped natural angular frequencies.

$$[K_e] = [\Phi_o^T]^{-1} [\Lambda] [\Phi_o]^{-1} \quad (6)$$

2.3. Calculation of Engine Mount Dynamic Stiffness

Fifteen equations can be obtained by making the non-zero components of the theoretical dynamic stiffness matrix $[K_t]$ in Eq. (2) and the experimental dynamic stiffness matrix $[K_e]$ component in Eq. (6) equal. Rewriting these in the form of simultaneous first-order equations related to the rubber mount dynamic stiffness, which is an unknown, results in the following.

$$[L]\{k\} = \{b\} \quad (7)$$

Where,

$$\{k\} = \{k_{x1} \quad k_{y1} \quad k_{z1} \dots\}^T$$

$$\{b\} = \{K_{e11} \quad K_{e22} \quad K_{e33} \dots\}^T$$

Labeling the number of unknowns as m , the matrix $[L]$ is a $15 \times m$ matrix comprised of the rubber mount position coordinates, $\{k\}$ is a $m \times 1$ row vector comprised of the unknown rubber mount dynamic stiffness, and $\{b\}$ is a 15×1 vector comprised of the Eq. (6) component. In Eq. (7), if m is equal to 15 and $[L]$ has full ranks, the rubber mount dynamic stiffness k can be uniquely calculated.

However, even if m is tentatively set as 15, this matrix $[L]$ may not have full rank in the actual mount layout. In this case, the rubber mount dynamic stiffness is obtained by setting some unknown rubber mount stiffness the same as other ones.

3. Simulations Using an FEM Model

Before applying the newly developed identification method to an actual outboard motor, the validity was checked by applying it to a finite element method (FEM) model consisting of the outboard motor and the engine mounts.

Figure 4 shows the FEM model shape. The outboard motor shape is roughly the same as that of the BF5 outboard motor used for actual motor verification in section 4.1. The rubber mounts support the outboard motor in four upper, lower, right and left locations, and the right and left rubber mounts have the same dynamic stiffness. Eigenvalue analysis and frequency response analysis were performed with the outer sides of the rubber mounts restrained, and the dynamic stiffness of the rubber mounts was identified using the natural frequencies, mode shape vectors and the frequency response obtained at optional points as a result of these analyses.

3.1. Rubber Mount Modeling Method

Each rubber mount is identified as having dynamic stiffness in the X, Y and Z directions, but as described in section 2.3., the matrix $[L]$ may not have full rank depending on the rubber mount layout, so the equations are solved assuming some of the rubber mounts to have the same dynamic stiffness. In this FEM model, this corresponds to the dynamic stiffness of the upper and lower rubber mounts in the Z direction. The upper and lower rubber mount layouts use the same left-right and front-back positions, so there is no relative displacement between the upper and lower rubber mounts in the Z direction. This means that the dynamic stiffness cannot be calculated independently, so the solution was obtained assuming the upper and lower rubber mounts to have the same dynamic stiffness. The same thing could also be said for the dynamic stiffness of the right and left rubber mounts in the Y direction, but the right and left rubber mounts have the same dynamic stiffness in the FEM model. Therefore, there are five rubber mount variables: k_{Ux} , k_{Uy} , k_{Uz} , k_{Lx} , and k_{Ly} .

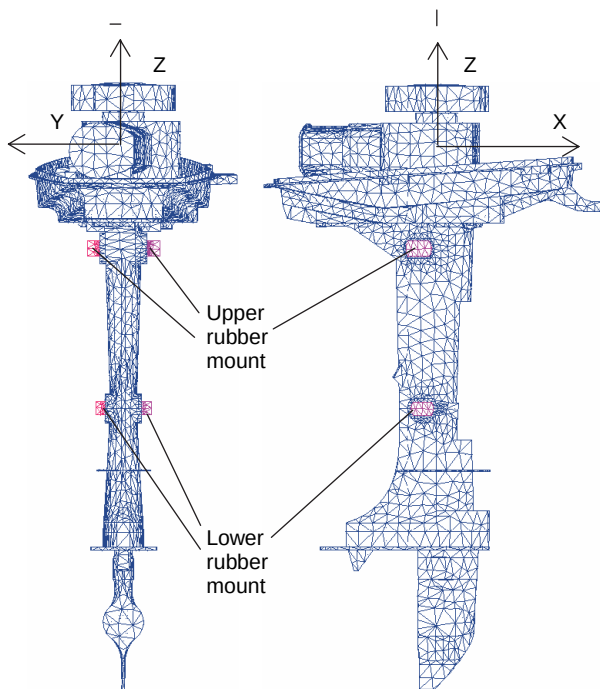


Fig. 4 Schematics of simulation model

Table 1 Results of mount stiffness in simulation study

		Identified	True
Upper mount	k_x	54	54
	k_y	249	261
	k_z	58	49
Lower mount	k_x	72	73
	k_y	351	363
	k_z	58	67

$\times 10^3$ [N/m]

3.2. Identification Results

Table 1 shows the dynamic stiffness of the rubber mounts identified using this method. When evaluated as the total value including the Z direction dynamic stiffness which is assumed to be the same for both the upper and lower rubber mounts, the identified values match closely with the true values, thus indicating that the identification results have sufficient accuracy.

Next, the frequency response function (FRF) at the measurement point was calculated using the obtained dynamic stiffness and the mass characteristics of the outboard motor used in the FEM model to verify the validity of the identification results. Figure 5 shows the FRF at the lower Y direction excitation point of the outboard motor. In this study the upper and lower rubber mounts were identified as having the same dynamic stiffness in the Z direction, and this figure shows that the FRF obtained using these identification values matches the true values closely. This indicates that the dynamic stiffness of the rubber mounts can be obtained using the newly developed method.

4. Application to a Small Outboard Motor

4.1. Structure and Modeling

The newly developed method was first used to identify the dynamic stiffness of the rubber mounts for a small outboard motor (BF5) with a weight of approximately 20 kg.

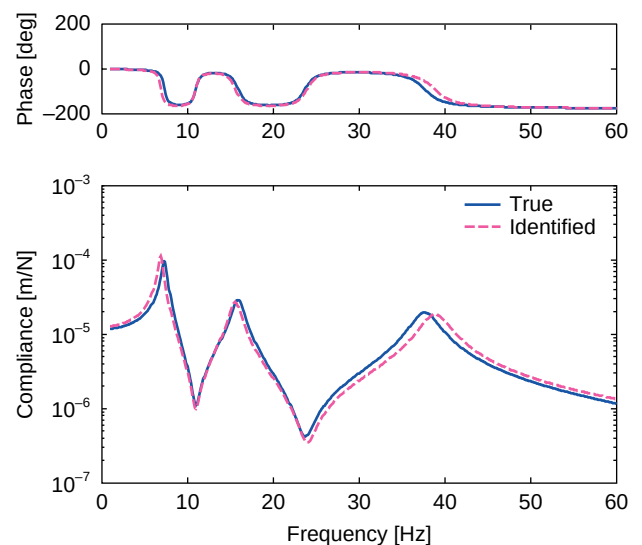


Fig. 5 Fitting of FRFs between identification and FEM model

Figure 3 shows the appearance of this outboard motor, and Fig. 6 shows the rubber mount layout. As seen here, there are two upper rubber mounts and three lower rubber mounts, and the right and left rubber mount layouts are the same. The difference from the FEM model in section 3 above is the addition of the #5 rubber mount in the lower position. This #5 rubber mount is located close to the #3 and #4 rubber mounts, so if the stiffness is obtained independently, the matrix $[L]$ becomes unique, and is thought to result in a large error. Therefore, the #5 rubber mount was included in the #3 and #4 rubber mounts. Like the FEM model, the dynamic stiffness in the Z direction was treated as the same for the upper and lower rubber mounts. Therefore, the five dynamic stiffness values k_{Ux} , k_{Uy} , k_{Uz} , k_{Lx} and k_{Ly} were obtained.

4.2. Identification Results

The outboard motor was fixed as shown in Fig. 3, a vibration by hammering test was performed, and the FRF was measured. Figure 7 shows the natural frequencies for the initial six modes obtained from the measurement results, and here the maximum frequency is approximately 50 Hz. In contrast, the 1st elastic mode of the outboard motor is approximately 130 Hz, so the outboard motor is thought to move more or less as a rigid body in the initial six modes. Therefore, it was judged that the dynamic stiffness of the rubber mounts could be calculated using the newly developed method which assumes the outboard motor to be a rigid body.

Table 2 compares the rubber mount dynamic stiffness identification results obtained using the newly developed method and the discrete rubber mount dynamic stiffness measurement results obtained by the conventional method using dynamic stiffness measurement equipment. Next the FRF at the measurement points was calculated from the dynamic stiffness in the same manner as in section 3.2. to verify the identification accuracy. Figure 8 shows the experimental FRF, the FRF obtained from the identification results and the FRF obtained from the discrete dynamic stiffness measured using the conventional method at the lower excitation point of the outboard motor. This shows that the results obtained using both the newly developed method and the conventional method match fairly closely with the experimental results.

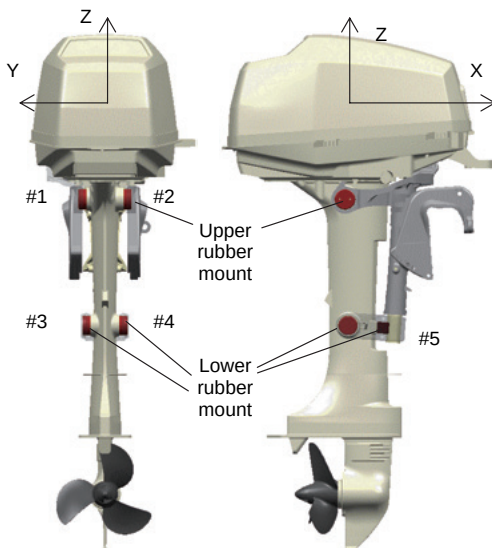


Fig. 6 Schematics of tested outboard motor BF5

Table 2 Results of mount stiffness BF5

		Thrust force		
		0[N]		500[N]
		Identified	Conventional method	Identified
Upper (#1,2)	k_x	52	106	103
	k_y	138	263	181
	k_z	153	101	174
Lower (#3,4)	k_x	414	351	805
	k_y	312	359	432
	k_z	155	211	175

$\times 10^3$ [N/m]

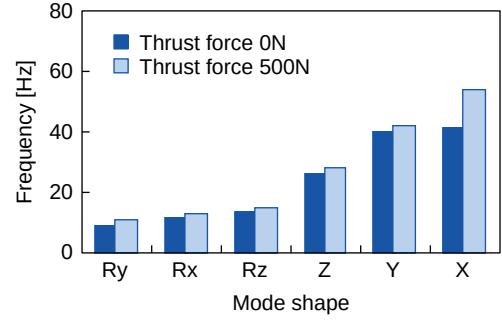


Fig. 7 Natural frequencies

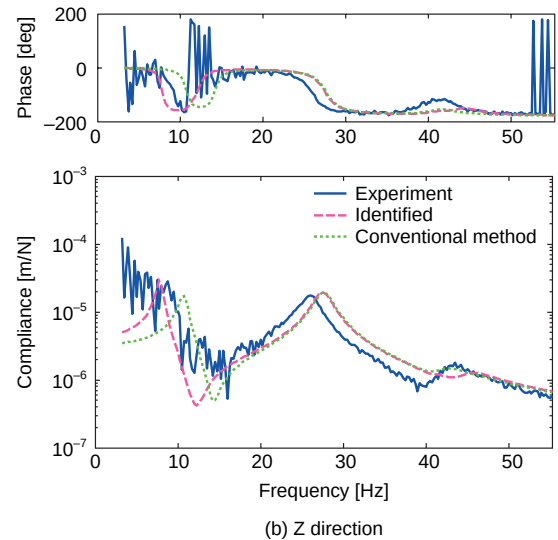
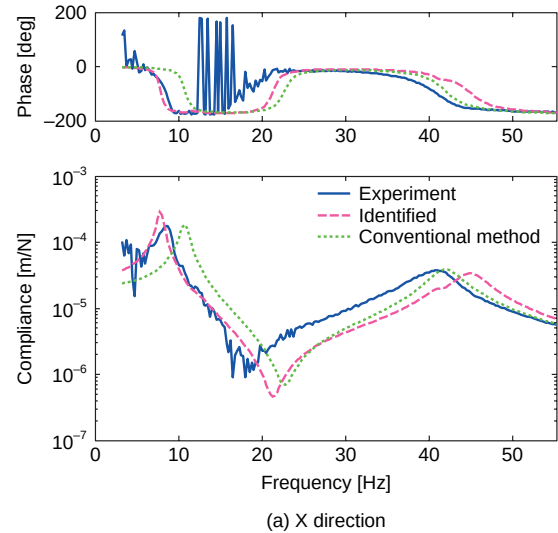


Fig. 8 Comparison of FRFs between identification and experiment

4.3. Change in Rubber Mount Dynamic Stiffness Due to the Application of Thrust Force

As shown above, the newly developed method can identify the dynamic stiffness of the rubber mounts in the condition assembled to an actual outboard motor. This feature can be used to understand changes in the dynamic stiffness of the rubber mounts caused by propeller thrust force, which was formerly not possible. Therefore, excitation tests were performed by applying a static load equivalent to the maximum thrust force (500 N) to the propeller part via a soft rubber piece, and the dynamic stiffness of the rubber mounts was identified.

As shown in Fig. 7, the natural frequency increases somewhat when thrust force is applied. In addition, the dynamic stiffness of the rubber mounts also increases as shown in Table 2. In particular, the dynamic stiffness of the rubber mounts in the X direction changes greatly to approximately double that without thrust force. Next, the FRF was calculated using the identified dynamic stiffness and compared with the experimental values. As shown in Fig. 9, the identification results closely match the experimental values, indicating that changes in the dynamic stiffness due to the application of thrust force are accurately represented.

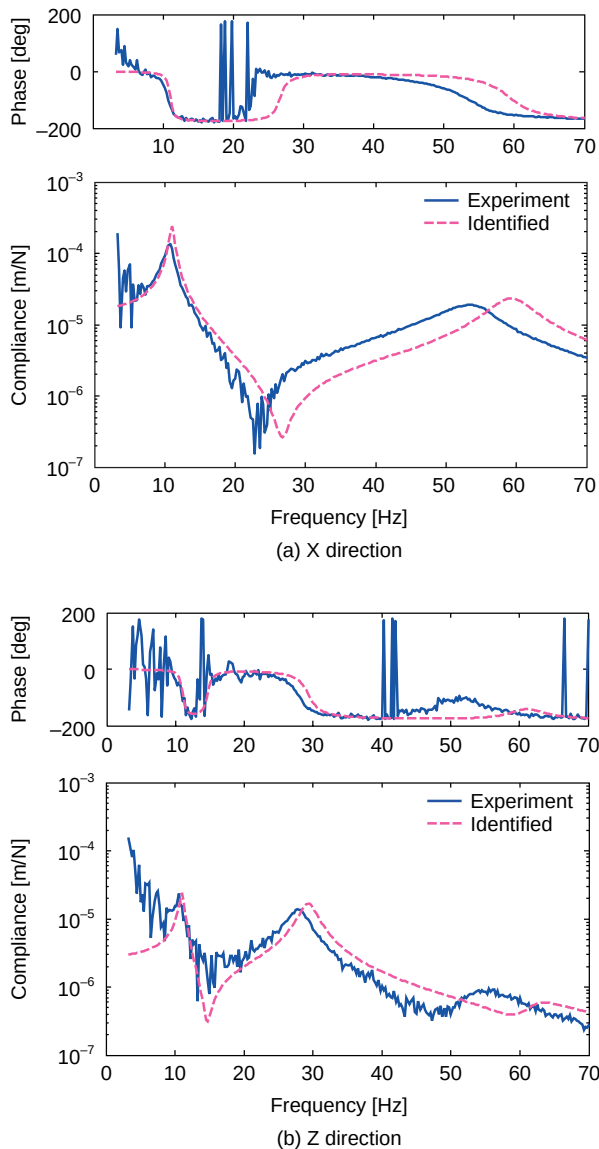


Fig. 9 Comparison of FRFs between identification and experiment

5. Application to a Large Outboard Motor

5.1. Rubber Mount Structure and Modeling

Next, the newly developed method was applied to a large outboard motor with a different size and rubber mount structure to verify the validity once again. The outboard motor used was the BF225, which weighs approximately 270 kg.

Figure 10 shows the outboard motor and the structure of the rubber mount supports used in the experiment. The outboard motor is supported by two upper rubber mounts and two lower rubber mounts that are attached to the mounting frame.

Figure 11 shows the structure of the lower rubber mount. The lower rubber mount deforms as the thrust force increases, and ultimately makes contact with the outboard motor at part A. Therefore, the dynamic stiffness of the lower rubber mount is

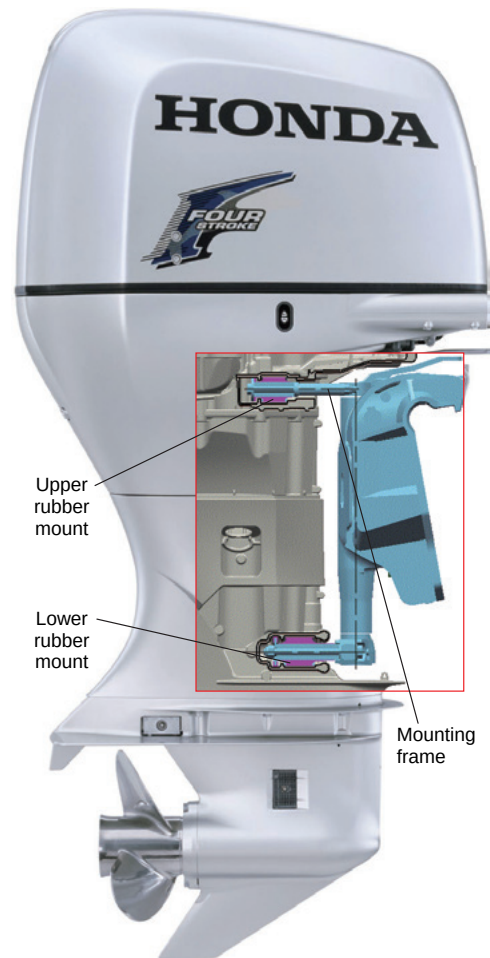


Fig. 10 BF225 side view

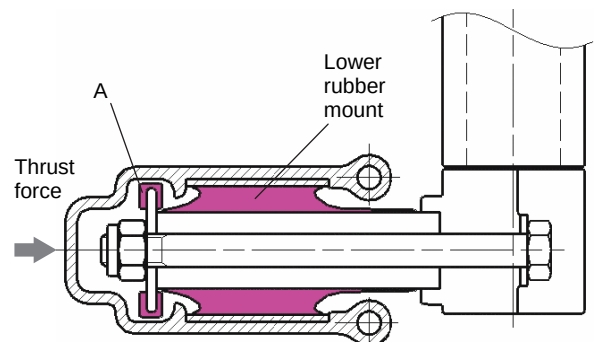


Fig. 11 Section of lower rubber mount

Table 3 Results of mount stiffness BF225

		Thrust force		
		0[N]		8000[N]
		Identified	Conventional method	Identified
Upper (#1,2)	k_x	1550	2280	2940
	k_y	680	14700	850
	k_z	1880	14700	3190
Lower (#3,4)	k_x	1540	1490	15200
	k_y	1100	10930	1690
	k_z	1880	10930	3190

$\times 10^3$ [N/m]

thought to change greatly at the maximum thrust force.

Like the small outboard motor, the dynamic stiffness in the Z direction was treated as the same for the upper and lower rubber mounts. Therefore, the five dynamic stiffness values k_{Ux} , k_{Uy} , k_{Uz} , k_{Lx} , and k_{Ly} were identified.

5.2. Identification Results

Table 3 compares the identification results obtained using the newly developed method and the discrete dynamic stiffness obtained by the conventional method using dynamic stiffness measurement equipment. In this case the identification results obtained by the newly developed method greatly underestimated the dynamic stiffness in the Y and Z directions for both the upper and lower rubber mounts. Next, the FRF at the measurement points were calculated from the dynamic stiffness to verify the identification accuracy, and these results are shown in Fig. 12. In addition, Fig. 13 shows the natural frequencies for the initial six modes in which the outboard motor vibrates more or less as a rigid body. The FRF in the X and Z directions obtained using the conventional method both differ greatly from the experimental results, and the natural frequencies for the X, Y and Rx modes are also completely different. In contrast, the FRF calculated from the identification results are more accurate than the conventional method, and there are no significant differences in the natural frequencies for all modes. Therefore, the dynamic stiffness of the rubber mounts obtained by this identification method is thought to be suited for evaluating and analyzing the effects of vibration on the boat body.

5.3. Identification when Thrust Force is Applied

Figure 14 shows the change in the natural frequencies when the maximum thrust force (8000 N) is applied. The natural frequencies increase overall, becoming approximately double in X mode. Table 3 shows the identified rubber mount dynamic stiffness. Here as well the dynamic stiffness of the rubber mounts increases when thrust force is applied. In particular, the dynamic stiffness of the lower rubber mount in the X direction increases by a factor of approximately ten. This is thought to be due to contact with the outboard motor at part A of the rubber mount as described in section 5.1., and confirms that the rubber mount characteristics change greatly when thrust force is applied.

Next, Fig. 15 shows the results of calculating the FRF at the excitation points using the identified dynamic stiffness. Whereas the FRFs based on the discrete dynamic stiffness values measured using the conventional method differ greatly from the experimental values, the FRFs based on the identification results show slight differences in the natural frequencies, while the FRF tendencies match relatively well overall. In other words, when application of

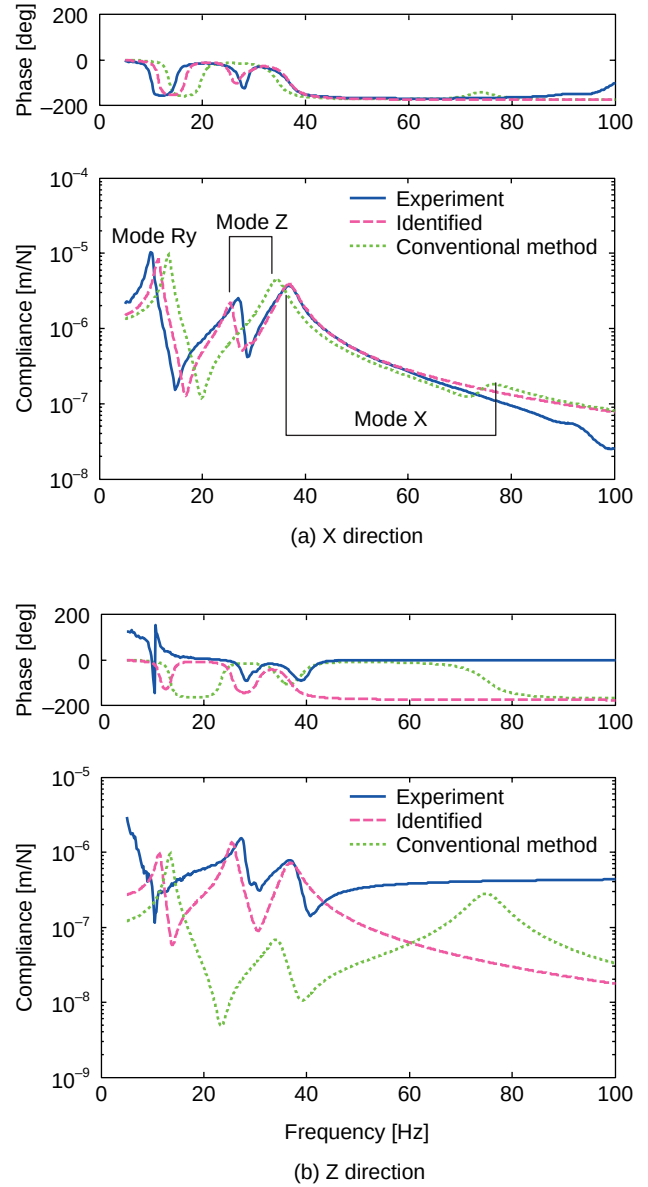


Fig. 12 Comparison of FRFs between identification and experiment

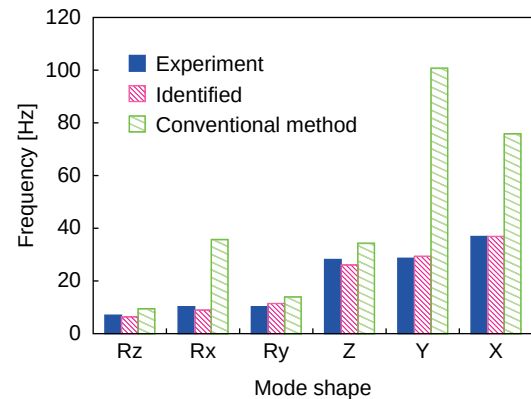


Fig. 13 Natural frequencies

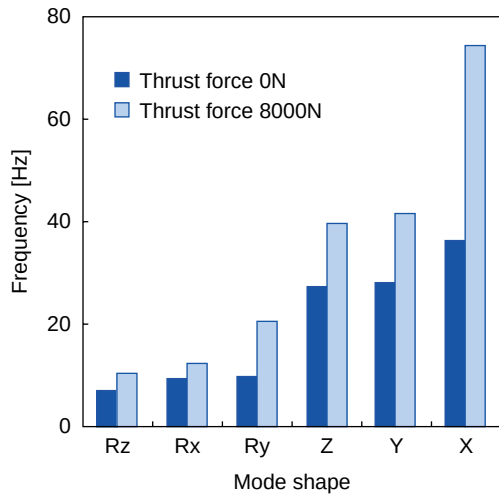
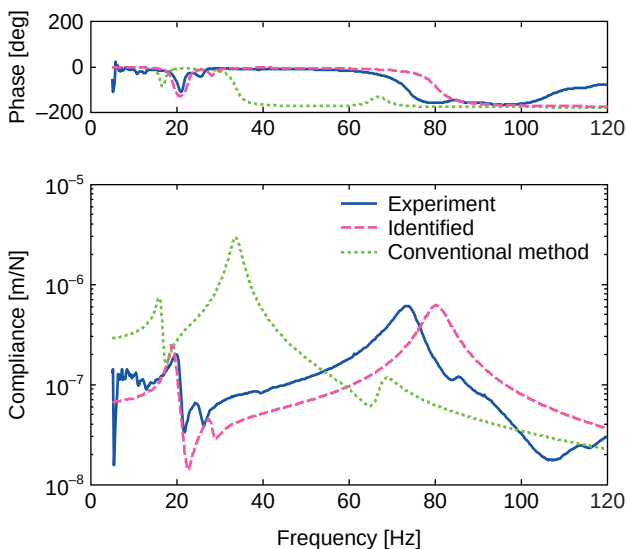
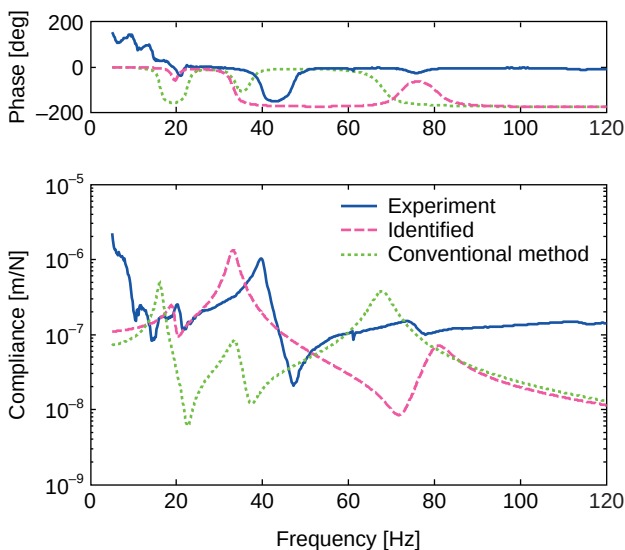


Fig. 14 Natural Frequencies



(a) X direction



(b) Z direction

Fig. 15 Comparison of FRFs between identification and experiment

a load produces significant changes in the rubber mount characteristics, it is considered impossible to accurately analyze vibration using the dynamic stiffness obtained by the conventional method.

5.4. Investigation of Differences from the Conventional Method

The BF225 rubber mounts are cylindrical in shape, so they should be soft in the X direction and hard in other directions as shown by the results of the dynamic stiffness tests. However, the identification results obtained by the newly developed method predict an extremely soft Y direction dynamic stiffness, with a stiffness of approximately 1/20 the actual value for the upper rubber mount in particular. It is difficult to think that this great a difference could result simply from assembly onto an actual outboard motor, so the reasons for the difference were investigated.

This identification method assumes that the outboard motor and the outboard motor support parts have sufficient stiffness and act as rigid bodies in the rubber mount resonance range. The natural frequencies of the initial six modes for the rubber mounts are 80 Hz or less even in the condition with thrust force applied, and the 1st elastic mode of the outboard motor is 200 Hz or more, so this is thought to have little effect. However, the stiffness of the outboard motor support parts had not been confirmed prior to the experiment, so confirmation was carried out after the experiment.

Hammering tests were performed on the outboard motor support parts without the outboard motor and the rubber mounts, and the 1st natural frequency was found to be around 100 Hz. However, in actual operation the weight of the outboard motor is applied, so the natural frequency is even lower. In addition, when the outboard motor support parts were also measured during the hammering tests to identify the dynamic stiffness of the rubber mounts, the outboard motor support parts were found to undergo elastic deformation.

In other words, the stiffness of outboard motor support parts is low compared with the rubber mount stiffness, so the identified dynamic stiffness is thought to be not just the dynamic stiffness of the rubber mounts, but the value also including the stiffness of the outboard motor support parts. A future theme is to calculate the dynamic stiffness of only the rubber mounts without the stiffness of the outboard motor support parts, but as described in section 5.3., the vibration of the outboard motor can be expressed using the identified dynamic stiffness. That is to say, the newly developed method is able to identify the overall stiffness of the rubber mounts and the outboard motor support parts, so it is thought to be more practical than the conventional method.

6. Conclusion

A new method was established by identifying the dynamic stiffness of the engine mounts of outboard motors from the results of experimental modal analysis.

- (1) The validity of the newly developed method was confirmed through simulations using an FEM model.
- (2) Vibration tests were conducted using actual outboard motors, and the dynamic stiffness of the mounts was identified. The frequency response functions (FRF) calculated from the identification results were shown to correlate fairly well with the experimental FRF.
- (3) When the outboard motor support parts have a low stiffness, the conventional method that uses the results of dynamic stiffness tests is unable to accurately estimate the vibration characteristics.

- (4) The newly developed method is superior to the conventional measurement method using dynamic stiffness measurement equipment in the following ways.
- Identification is performed using the results of vibration tests conducted on outboard motors in the assembled condition, which makes it possible to identify the dynamic stiffness of the mounts when the outboard motor's weight and thrust force equivalent to actual operation are applied.
 - The dynamic stiffness is identified for the overall outboard motor support parts including the engine mounts, so identification is possible even when the outboard motor support parts have low stiffness.

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Method of Transfer Path Analysis for Interior Vehicle Sound Using Principal Component Regression Method

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ABSTRACT

This paper describes a time domain transfer path analysis method that can obtain transfer functions and the contributions of structure-borne and airborne sounds in a vehicle's interior. Vibration acceleration and sound pressure signals at the sound sources (reference points) and the sound pressure signal in the cabin (response point) were measured simultaneously to obtain the transfer functions when the test vehicle was driven. Principle component regression method (PCR), which can eliminate both crosstalk among transfer functions and noise in the measured data, was employed to obtain transfer functions accurately. In addition, a normalization procedure was developed to perform the PCR for combined data sets in which both vibration acceleration and sound pressure signals exist.

By applying the above methods, a highly accurate and efficient transfer path analysis method was developed that does not require a hammer test.

1. Introduction

The cause of most noise and vibration in the interior of a vehicle when driven is vibration that originates in the engine and tires, and is transmitted to the interior via multiple transfer paths. To efficiently counteract the most annoying interior noise and vibration, it is essential to identify the transfer paths that make the greatest contribution, and to modify the sources of vibration and transfer characteristics on this basis. In the last decade, transfer path analysis has been discussed as an analytical method enabling this^{(1), (2)}. Transfer path analysis consists of two steps. First, transfer functions are determined for response points for multiple input points of sound and vibration. Second, these transfer functions are combined with forces measured at the input points to generate synthesized response signals. By performing these analyses, the degree of contribution of each transfer path is obtained. Though a previous study⁽³⁾ has demonstrated the method's effectiveness, the conventional method of transfer path analysis^{(1), (2)} is unable to accurately separate the structure-borne and airborne components of vehicle interior sound.

Therefore, it is not sufficient to use the conventional method as an analysis method in order to enhance interior sound quality. In addition, the measurement of transfer functions requires a considerable number of man-hours, and there are cases in which such measurements are unable because of the time restrictions occasioned by current short development times. A simpler method of transfer analysis was therefore required.

This paper discusses the development of a new, practical method of transfer path analysis that can accurately separate structure-borne and airborne sounds, and provides an example of the application of this method to the analysis of interior sound in an accelerating vehicle.

2. New Method of Analysis

In the new method of analysis, when the sound pressure at certain points in the vehicle interior are regarded as response point signals, and the engine, tires, etc. are regarded as sound sources, acceleration vibration at points on major areas of the vehicle framework, e.g.

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the engine mounts and suspension, is considered the reference point signal for structure-borne sound. Sound pressure signals at arbitrary points in the engine chamber, and at areas around the exhaust system with large radiating surfaces, are considered reference point signals for airborne sound. Transfer functions between the response point and each reference point are determined. Vehicle interior sound is synthesized by combining these transfer functions with signals measured at the reference points when the vehicle is operating under load, to synthesize the signals for all the assumed transfer paths. After verifying that this synthesized sound reproduces the actually measured interior sound accurately enough, the contribution of each transfer path is determined. Because vector addition is applied to the signals for each transfer path, it is not possible to discuss the degree of contribution for each path in terms of simple magnitude. Since the sound source distribution is non-uniform, it is difficult to use aerodynamic noise signals as reference points on the basis of a limited number of points. Aerodynamic noise is therefore outside the scope of this method, and measurements are taken using a chassis dynamometer under conditions in which aerodynamic noise will not be generated.

2.1. Reference Point Signals for Structure-borne Sound

Using the conventional transfer path analysis method, it is necessary to excite the reference points and measure the transfer functions. However, in many cases it is difficult to ensure enough space in the engine chamber to allow excitation. In addition, while a sensor attached to the end of a hammer makes it easy to obtain force signals in an excitation test, it is difficult to measure force signals at reference points while a vehicle is being driven. The development of the new method emphasized practicality, and acceleration vibration was therefore adopted as the reference point signal, given that it requires less space for measurement and can be easily measured while the vehicle is being driven. In the transfer path analysis method discussed here, the sound pressure signal at a response point, p , is generated by multiplying the acceleration vibration signals, $\ddot{x}_i$, at n reference points (n being a number sufficient to enable reproduction of p) by their transfer function, h_i , as shown in Eq. (1).

$$p = \sum_i^n \ddot{x}_i \cdot h_i \quad (1)$$

2.2. Calculation Method of Transfer Functions using Principal Component Regression Analysis

Equation (1) cannot be used to calculate multiple transfer functions h_i from an acceleration vibration signal $\ddot{x}_i$ and a sound pressure signal p obtained from a single excitation test.

Accordingly, m ($m \geq n$) arbitrarily selected points are subjected to excitation and the sound pressure signals p_j ($j: j=1,2,3 \dots m$) and acceleration vibration signals $\ddot{x}_{ji}$ for each are measured to generate m linear polynomial equations, as shown in Eq. (2). Equation (2) is expressed as a matrix for convenience of calculation, as shown in Eq. (3), and the transfer function h_i is calculated.

$$\begin{cases} p_1 = \ddot{x}_{11}h_1 + \ddot{x}_{12}h_2 \cdots + \ddot{x}_{1i}h_i \cdots + \ddot{x}_{1n}h_n \\ \vdots \\ p_j = \ddot{x}_{j1}h_1 + \ddot{x}_{j2}h_2 \cdots + \ddot{x}_{ji}h_i \cdots + \ddot{x}_{jn}h_n \\ \vdots \\ p_m = \ddot{x}_{m1}h_1 + \ddot{x}_{m2}h_2 \cdots + \ddot{x}_{mi}h_i \cdots + \ddot{x}_{mn}h_n \end{cases} \quad (2)$$

$$[P] = [\ddot{X}] \cdot [H] \quad (3)$$

P_j , $\ddot{x}_{ji}$ and h_i represent Fourier spectra for their respective frequencies. Calculations are conducted in the same way for all target frequencies.

If the number of excitations, m , and the number of reference points, n , are equalized and the acceleration vibration matrix $[\ddot{X}]$ is made a square matrix, both sides of Eq. (3) can be multiplied by the inverse matrix of $[\ddot{X}]$, $[\ddot{X}]^{-1}$, to enable the transfer function matrix $[H]$ to be calculated, as shown in Eq. (4).

$$[H] = [\ddot{X}]^{-1} \cdot [P] \quad (4)$$

However, when this method is used, the accuracy of the calculation of the transfer function matrix $[H]$ declines markedly when there is noise in the measured acceleration vibration matrix $[\ddot{X}]$ and the sound pressure signal matrix $[P]$, or when there is a high degree of correlation between the acceleration vibration signals of the reference points. The principal component regression analysis⁽⁴⁾ has therefore been used in the new analytic method to provide a solution to these issues.

Using principal component regression analysis, singular value decomposition is first applied to the acceleration vibration matrix $[\ddot{X}]$ measured at each reference point (Eq. (5)), and a principal component matrix, $[T]$, is formulated using Eq. (6). This singular value decomposition completely whitens the correlations between the principal components.

$$[\ddot{X}] = [U] \cdot [S] \cdot [V]^{-1} \quad (5)$$

$$[T] = [\ddot{X}] \cdot [V] = [U] \cdot [S] \quad (6)$$

Next, the singular value matrix, $[S]$, which contains the singular values expressing the magnitude of the principal components in its diagonal elements, is focused on. Principal components that are small in comparison to the other principal components are considered as noise components and discarded. A singular value matrix from which these principal components have been discarded, $[S_r]$, and unitary matrices $[U_r]$ and $[V_r]$ are formulated, and Eq. (7) is used to reformulate a principal component matrix, $[T_r]$, from which the noise components have been removed.

$$[T_r] = [\ddot{X}] \cdot [V_r] = [U_r] \cdot [S_r] \quad (7)$$

Next, the coefficient matrix $[C]$ of $[T_r]$ for the sound pressure signal matrix $[P]$ is determined using Eq. (8).

$$\begin{aligned} [P] &= [T_r] \cdot [C] \\ [T_r]^T [P] &= [T_r]^T \cdot [T_r] \cdot [C] \\ [C] &= ([T_r]^T \cdot [T_r])^{-1} \cdot [T_r]^T [P] \end{aligned} \quad (8)$$

The above equations enable formulation of the coefficient matrix $[V_r]$, which transforms the reference point acceleration vibration signal matrix $[\ddot{X}]$ into the principal component matrix $[T_r]$ from which noise has been removed, and the coefficient matrix $[C]$, which transforms the principal component matrix $[T_r]$ into the response point pressure signal matrix $[P]$. Equation (9) can be used to calculate the transfer function h_i between the acceleration vibration signal (reference point) and the sound pressure signal (response point).

$$\begin{aligned} [H] &= [V_r] \cdot [C] \\ &= [V_r] \cdot ([T_r]^T \cdot [T_r])^{-1} \cdot [T_r]^T \cdot [P] \\ &= [V_r] \cdot [S_r]^{-1} \cdot [U_r]^T \cdot [P] \end{aligned} \quad (9)$$

It is desirable to set the number of excitations m higher than the number of reference points n ($m \gg n$) to exclude the noise component included in the reference points from the principal component analysis and the noise component included in the response points from the regression analysis.

The method discussed here can also be applied to acceleration vibration at response points originating from steering wheel vibration and floor vibration.

2.3. Verification of Transfer Functions Accuracy

Tests were conducted to verify the superiority of the method of calculating transfer functions using the principal component regression analysis, as discussed in Section 2.2.

These tests used a vehicle body that had doors and windows, but was stripped of major interior and external components, including the engine, tires, suspension and seats, as shown in Fig. 1.

As Fig. 2 shows, three-axis accelerometers were positioned at two points, at the left and right extremities of the frame, and a microphone was positioned in the cabin. The two points at which the accelerometers were located were then excited in three directions by hammers to obtain their transfer functions by the conventional method. To calculate their transfer functions using principal component regression analysis, Fourier spectra of the reference and response points during this excitation and of arbitrary excited six points were obtained. Next, two shakers were used at the two excitation points to provide up-down excitation with uncorrelated random signals, and the time domain signals of the reference and response points were obtained.

The data obtained in this way were used to find single input-single output transfer functions and transfer functions generated by principal component regression analysis. Using Finite Impulse Response (FIR) filter processing, these transfer functions were then separately combined with the acceleration vibration signals obtained from each reference point under shaker excitation to produce synthesized sounds, as shown in Fig. 3.

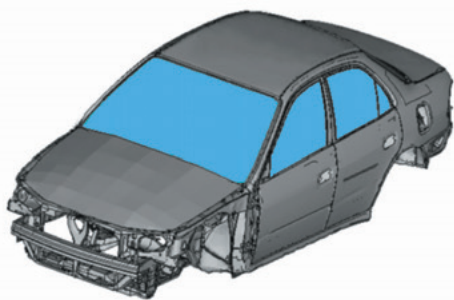


Fig. 1 Trimmed body used in experiment

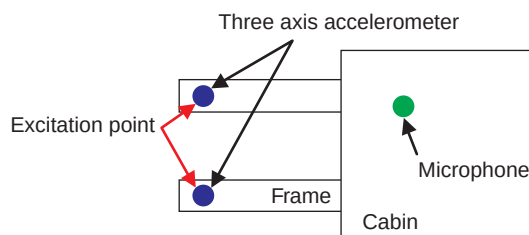


Fig. 2 Measurement and excitation points

Figure 4 shows a comparison of the power spectra of sounds measured at the response points and sounds synthesized using transfer functions obtained by the two different methods. The results show that the transfer functions obtained using the principal component regression analysis are more accurate, demonstrating the superiority of the method.

2.4. Transfer Functions of Airborne Sound

The new method of calculating transfer functions using the principal component regression analysis was next applied to airborne sound, which are thought to be inappropriate to treat as same as structure-borne sound.

As a practical method of calculating transfer functions for airborne sounds, the first proposal was to apply acoustical excitation to the structure, and treat the sound pressure signal measured at an arbitrarily positioned microphone as the reference point, i.e. as the sound source. However, as Fig. 5 shows, when the incidence characteristic of the sound wave changes, the excitation force acting

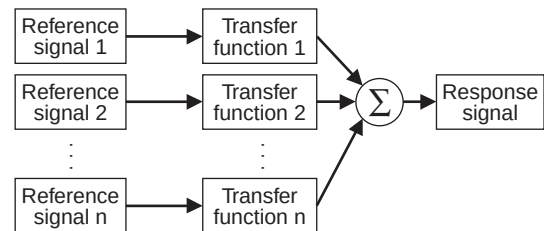
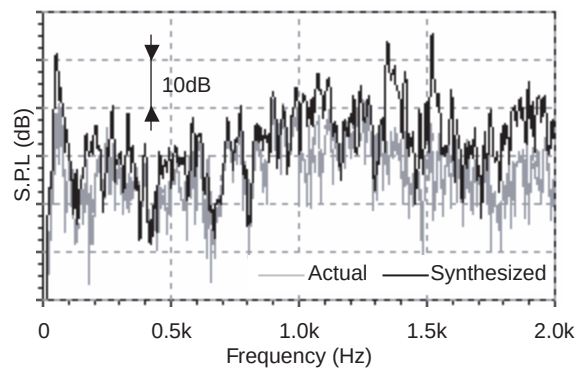
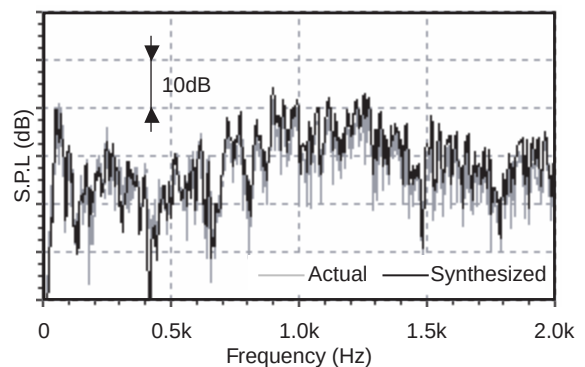


Fig. 3 Path signal synthesis with FIR filter



(a) Without principal component regression



(b) With principal component regression

Fig. 4 Comparison of actual and synthesized signals in averaged spectrum

on the structure changes, even if the sound pressure signal at the reference point is the same and true transfer functions cannot be obtained. To resolve this issue, it would be necessary to conduct excitation tests using a dummy sound source with incidence characteristics identical to the real sound source.

The method finally determined, therefore, was to calculate transfer functions using sound pressure signals measured at reference points during actual vehicle operation.

2.5. Method of Calculating Transfer Functions using Data Obtained during Vehicle Operation

When the transfer functions for airborne sound are calculated using sound pressure signals during actual vehicle operation, it is essential that there is no sound source, other than the reference point. However, this condition cannot be satisfied when a vehicle is in operation, because engine vibration and other factors are constantly generating structure-borne sound at the same time as airborne sound is being produced. Therefore, the acceleration vibration signals measured in operation were used as reference point signals for structure-borne sound, to enable transfer functions for structure-borne and airborne sounds to be simultaneously calculated. The factor that corresponds to the number of excitations m is the number of continuously calculated Fourier spectra. As time elapses, the continuously calculated Fourier spectra are arranged in the order of the columns of the reference signal and response signal matrices. Using this method, the transfer functions that reflect the effect of the engine mount and the bush characteristics affected by the driver wheel torque's reaction force under load could be obtained.

2.6. Normalization Method

To simultaneously calculate transfer functions for structure-borne and airborne sound using the response point sound pressure signal matrix $[P]$, the reference point acceleration vibration signal matrix $[\ddot{X}]$ and the reference point sound pressure signal matrix $[A]$ —all factors based on measurements conducted while the vehicle is being driven—it was considered most effective to conduct the calculations from a matrix combining $[\ddot{X}]$ and $[A]$, as shown in Eq. (10). However, the units of measurement for acceleration vibration (m/s^2) and sound pressure (N/m^2) differ, and therefore cannot be compared in terms of magnitude. That is, if noise is simply discarded from the mixed matrix $[\ddot{X}/A]$ shown as Eq. (10), it is conceivable that one of the signal components, either the acceleration vibration signal or sound pressure signal, would be discarded with the noise component.

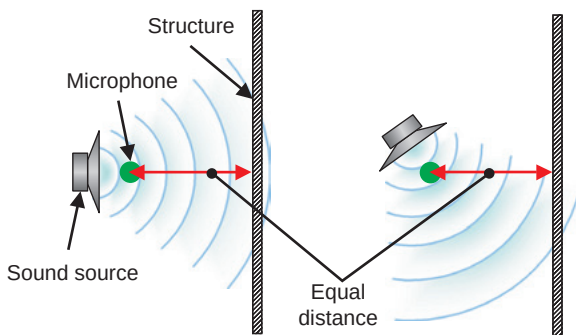


Fig. 5 Acoustic excitation power difference depending on incidence angle

$$[P] = [\ddot{X}/A] \cdot [H] \quad (10)$$

To minimize this possibility, the matrices for acceleration vibration, $[\ddot{X}]$, and sound pressure, $[A]$, are normalized before being combined.

3. Development of a Transfer Path Analysis System

Software was developed combining the ability to generate synthesized sounds and to calculate transfer functions using the principal component regression analysis method. The software also includes supplementary functions, including the ability to verify the validity of the calculated transfer functions and to produce animations of the behavior of each calculated principal component.

4. Test Verification of the New Calculation Method's Effectiveness

The effectiveness of the new calculation method was verified by applying it to the analysis of a front-wheel drive vehicle with an engine displacement of 2.4L.

4.1. Test Method

A complete vehicle was positioned on a chassis dynamometer able to absorb front wheel torque. Measurements were taken in several conditions, with the vehicle in second gear.

With respect to reference points, acceleration vibration measurements were taken in three directions at a total of 21 points, including the side of the engine at the engine mounts, the side of the body, the axle bearings and the exhaust system mounting points, and sound pressure measurements were taken at a total of 14 points including arbitrary points in the engine chamber, in the intake system and around points on exhaust system components that were considered significant contributors to the radiated sound. Two positions, the right and left ear positions of an artificial head microphone for binaural recording, were established as response points. A total of 79 channels of time domain signals were simultaneously measured, and at the same time Fourier spectra were continuously calculated as time elapsed.

4.2. Analysis Conditions

In the vehicle used in testing, front wheel torque reaction force caused the engine to tilt backwards due to acceleration loads and strike against the stopper that controlled engine mount movement, causing the spring constant of the engine mounts to change. Accordingly, the analysis was performed separately for calculating the transfer function between engine-side vibration and the body-side vibration of the engine mounts and for calculating the transfer functions between the body-side vibration of the engine mounts and the interior sound, respectively, as shown in Fig. 6.

4.3. Reproduction Accuracy of Synthesized Sound

Figure 7 shows the spectrograms of measured and synthesized sounds at the left ear of the artificial head microphone. Figure 7(a) shows the spectrogram of the measured vehicle interior sound, and Fig. 7(b) shows that of synthesized sound with acceleration vibration signals from 45 channels and sound pressure signals from 14 channels used as reference points. These results show that the

synthesized sound reproduced the measured sound with a high degree of accuracy.

4.4. Results of Separation of Structure-borne and Airborne Sound

Figure 7(b) shows a synthesis of the signals from all the transfer paths. Also, it is possible to consider each transfer path separately, or to synthesize the transfer paths within a specific delimited range. Figure 8 shows the results of classifying the sound into structure-borne and airborne components. These graphs show that the second-order level for engine speed was comparatively high for airborne

sound, as shown in Fig. 8(b), indicating that some of the transfer paths that create airborne sound contributed significantly.

Figure 9 shows a comparison of averaged spectra for structure-borne and airborne sounds within a specific range of engine speeds, showing that below approximately 500 Hz, structure-borne sound contributed greatly, while above approximately 1 kHz, airborne sound contributed greatly.

As demonstrated above, the new method is able to specify the transfer paths with high contribution from among a large number of transfer paths, and will therefore assist in determining effective countermeasures.

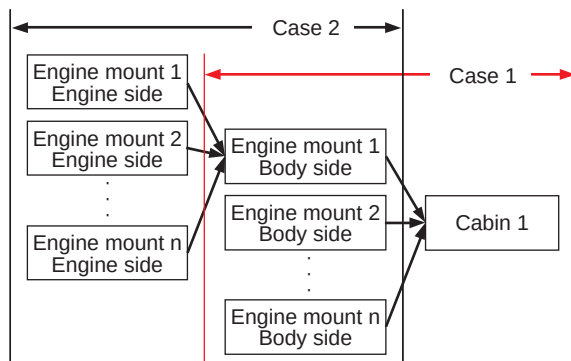


Fig. 6 Example of structural model selection

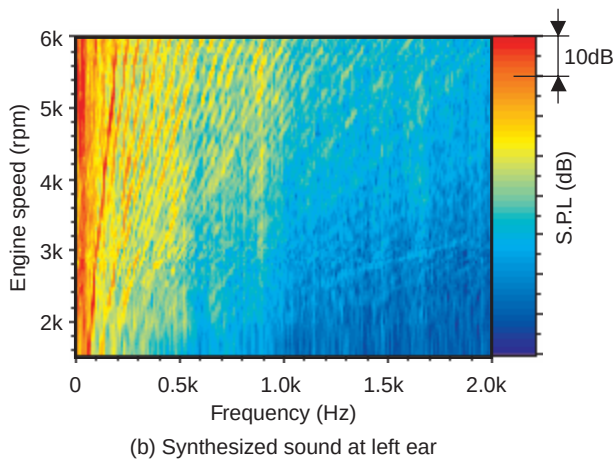
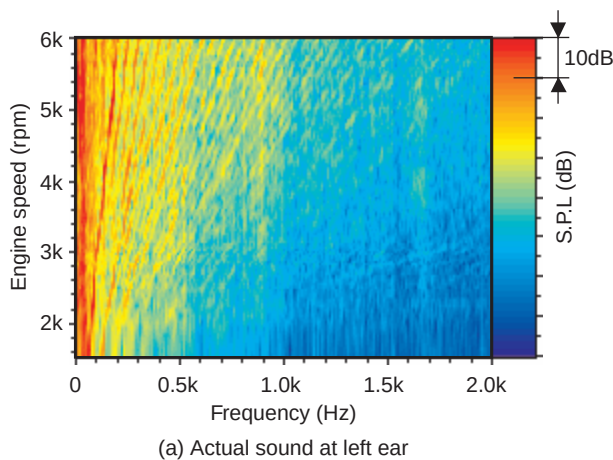


Fig. 7 Comparison between actual and synthesized sounds

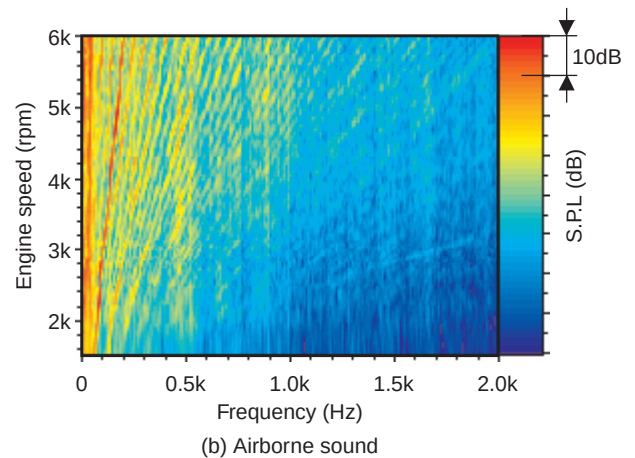
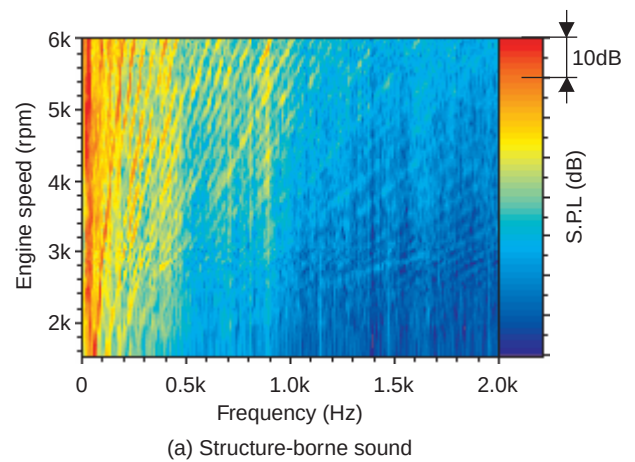


Fig. 8 Comparison between structure-borne and airborne sounds

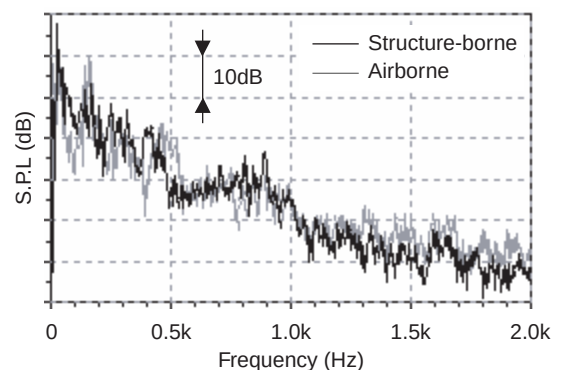


Fig. 9 Comparison between structure-borne and airborne sounds in averaged spectrum

5. Conclusion

A method of transfer path analysis has been developed that uses principal component regression and an original normalization procedure. The new method is able to separate structure-borne and airborne sounds on the basis of data recorded while a vehicle is being driven, with no requirement for an excitation test. Application of the method to an actual vehicle to verify its effectiveness indicated that it was able to identify transfer path characteristics with a high degree of accuracy, and that it was superior to the conventional method in the following respects:

- (1) The highly accurate reproduction of the synthesized sound makes the approach to modifying transfer path characteristics to enhance sound quality more clear.
- (2) There is no need to invest man-hours in performing excitation tests.
- (3) Data can be obtained using only acceleration sensors and microphones, and there are therefore few restrictions on areas to which the analysis can be applied.
- (4) Few restrictions on driving conditions are necessary.

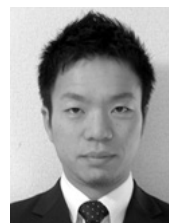
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Study of Interior Sound Preferences at Acceleration

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ABSTRACT

This paper describes car interior sound preferences in general public. Subjective evaluation experiments were performed in Japan, Germany and the United States to investigate the interior sound preferences. Two hundred subjects in each country (600 total) participated in the experiment. As a result, four kinds of preferences were found. A sound preference evaluation system was developed to calculate numeric preference values based on sound stimuli recorded using an artificial head microphone.

Furthermore, subjects' characteristics in each kind of preference were clarified through questionnaires. The results showed that women, older people and people not interested in cars preferred quiet sound, while men, younger people, and people who like cars preferred powerful sound.

1. Introduction

Various researches of sound quality evaluation techniques were conducted for enhancing sound quality in vehicle interiors. As a result of these researches, various metrics have been proposed for auditory stimuli, including loudness and fluctuation intensity⁽¹⁾, and for complex sensations, luxuriousness and sportiness^{(2),(3)}, allowing for a quantitative evaluation of sensations experienced in response to sound. Studies have indicated that even when the sensation experienced is identical, preferences with regard to those sensations can be entirely different⁽³⁾. Also, there is a lack of detailed data on the types of sound preferred by different users.

In the research discussed in this paper, subjective evaluation experiments of interior sound preferences were conducted in Japan, Germany and the United States. Test subjects were asked to fill out a questionnaire on personal information: age, gender, etc. The results were analyzed statistically. Objective numerical values were

calculated for the degree of preference users had for measured interior sounds and the preferences were then classified based on the users' attributes, or personal data.

This paper discusses the results of the analysis, the method used to calculate the degree of user preference, and gives an overview of the preference evaluation system based on this method.

2. Subjective Evaluation Experiments

In the subjective evaluation experiments, interior sound recorded during acceleration was played back via headphones to determine subject preferences.

2.1. Sounds Presented to Subjects

Binaural recordings of interior sounds measured during acceleration in 18 vehicle types marketed in Japan, Europe and the

* Tochigi R&D Center

Table 1 Number of subjects

Sound type	Japanese	German	American
Low engine speed	206	207	202
Middle engine speed	206	207	202
High engine speed	206	207	202

United States were made using artificial head microphones and played for test subjects. Twelve sound types each were selected from low engine speeds (approximately 1 000-3 000 rpm), medium engine speeds (approximately 3 000-5 000 rpm) and high engine speeds (approximately 5 000-7 000 rpm). These recordings were edited down to 2 seconds. An exponential function weighting procedure was used for 0.2 seconds at each end of the signal to control transient sound bits and the recordings were converted into digital audio files so that they could be played back on a computer.

2.2. Test Apparatus

The test apparatus consisted of a computer and a binaural playback system, as shown in Fig. 1. The sounds were played back using an original software program. A maximum of four subjects were able to listen to the sounds at the same time and provide subjective evaluations.

2.3. Test Subjects

The average age and gender of the test subjects were approximately equal to the average age and gender of licensed drivers in the countries in which the tests were conducted. It was predicted that a diverse range of preferences would be indicated, thus making a large pool of test subjects, which was essential to obtain statistically significant results. Approximately 200 test subjects were selected from Japan, Germany and the United States (for a total of about 600). Table 1 shows the number of test subjects organized according to the range of engine speed sounds presented to them in tests.

2.4. Test Method

Sheffe's paired comparison method⁽⁴⁾ was used to provide highly accurate assessments with respect to evaluations and enables the degree of assessment to be determined.



Fig. 1 Experimental apparatus

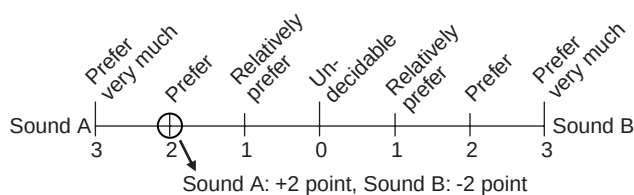


Fig. 2 Answer sheet in Sheffe's paired comparison method

Table 2 Preference value in each comparison

Sound number	No.1	No.2	No.3	No.4	No.5	No.6	No.7	No.8	No.9	No.10	No.11	No.12
No.1		1	1	2	-1	1	-1	-1	1	2	1	1
No.2	-1		-1	2	-2	-2	-2	-2	-1	-2	-2	1
No.3	-1	1		2	-2	-2	-1	-2	-1	-1	-2	1
No.4	-2	-2	-2		-2	-3	-2	-3	-2	-2	-2	-3
No.5	1	2	2	2		-1	1	-2	1	1	2	1
No.6	-1	2	2	3	1		-1	-1	1	-1	2	2
No.7	-1	2	1	2	-1	1		-2	-1	-1	2	1
No.8	1	2	2	3	2	1	2		1	2	2	1
No.9	-1	1	1	2	-1	-1	1	-1		1	2	1
No.10	-2	2	1	2	-1	1	1	-2	-1		0	1
No.11	-1	2	2	2	-2	-2	-2	-2	-2	0		2
No.12	-1	-1	-1	3	-1	-2	-1	-1	-1	-1	-2	
Ave.	-0.6	1.1	0.7	2.3	-0.9	-0.8	-0.5	-1.7	-0.5	-0.2	0.3	0.8

Pairs of sounds were played for subjects over headphones. Each sound pair was repeated twice and the subjects were asked to select their preferred sound and mark a degree of preference on an answer sheet using the scale shown in Fig. 2. This procedure was repeated 66 times for each engine speed range, in random order, for a total of 198 times for each test subject. Approximately 200 subjects were tested in each country, with up to four subjects tested at one time; the tests were therefore conducted approximately 50 times in each country. To reduce the effect of presentation order on the evaluations, the order of the pairs of sounds was changed for each test.

3. Quantification of Sound Preferences

Multivariate analysis was applied to the results of the subjective evaluation experiments for each engine speed range to quantify the test subjects' preferences, which were then classified in relation to specific types of sound and evaluated on the basis of objective numerical values.

3.1. Quantification of Sound Preferences in the Low Engine Speed Range

The results for each test subject were used to produce numerical figures representing their relative degree of preference, as shown in Table 2. Average values for each type of sound presented were calculated using these figures and assigned as numerical representations of the degree of preference expressed by the test subject. Below, these values are termed subjective preference values (SPV).

Figure 3 shows SPVs for all 615 Japanese, German and U.S. test subjects, plotted for the 12 types of sound presented in the tests.

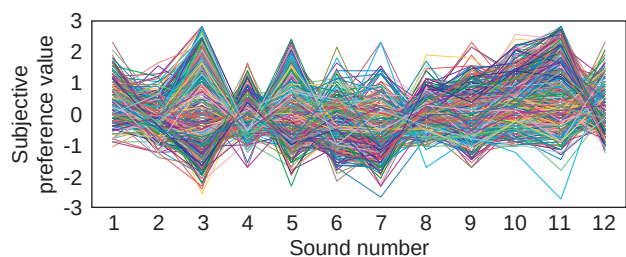


Fig. 3 Preference in each subject

These results indicate that there was significant diversity in the test subjects' individual preferences. Particularly in relation to sounds 3, 7 and 11, for which some subjects indicated an extremely strong preference, while others exhibited an equally strong negative reaction: a significant contrast. User preferences are therefore not uniform. A number of different types of preference can be considered to exist. To elucidate these preference types, cluster analysis⁽⁵⁾ was used to classify test subjects into groups according to the order of their preferences.

As Fig. 4 shows, three clusters were generated at the low engine speed range. This indicates that criteria for preferences do not differ entirely for each individual. Preferences fall mainly into three groups and most individuals can be represented in one of these three groups.

As Fig. 5 shows, Cluster 1 contains the highest proportion of test subjects, at approximately 70%. This indicates that if the sound preferred by the test subjects in Cluster 1 was produced by an actual vehicle, about 70% of vehicle purchasers would like it. Note that statistically insignificant test subjects were included in the "Other" category.

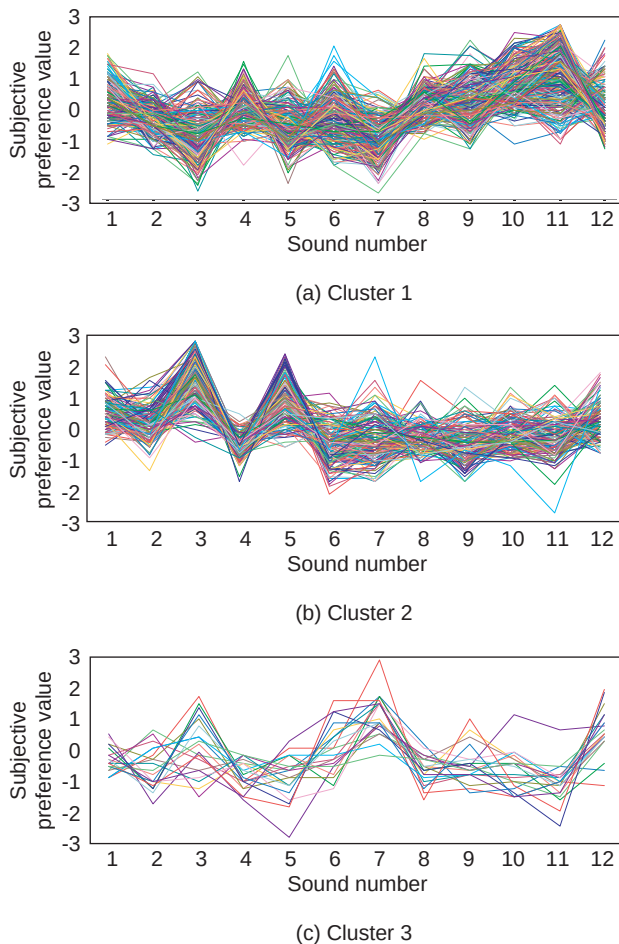


Fig. 4 Subjective preference value in each cluster

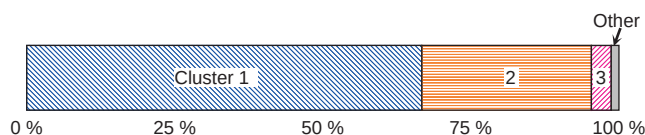


Fig. 5 Preference groupings at low engine speed

Next, multiple regression analysis⁽⁵⁾ was conducted to clarify the types of sound preferred by the test subjects in each cluster. The average subjective preference values of the test subjects in each cluster were employed as dependent variables in the analysis. Values for Honda's sound quality evaluation metrics⁽³⁾ (luxuriousness, smoothness, loudness, sportiness, aliveness and power) were calculated from actual measurements taken using an artificial head microphone and used as the independent variables.

Multiple regression analyses were conducted for each cluster. The results for Cluster 1 are shown in Eq. (1), for Cluster 2 in Eq. (2), and for Cluster 3 in Eq. (3). The terms on the right side of the equations are selected to ensure that the left sides maintain a high degree of correlation with the subjective preferences, which function as dependent variables. The left sides of these equations represent objective preferences, which are termed objective preference values (OPV) below. Each of the terms on the right side of the equations describes the sound characteristics to which the OPVs relate.

$$\begin{aligned} \text{OPV 1 (Cluster 1)} \\ = -5.2 \times \text{Loudness} + 0.4 \times \text{Smoothness} - 1.3 \end{aligned} \quad (1)$$

$$\begin{aligned} \text{OPV 2 (Cluster 2)} \\ = -3.2 \times \text{Loudness} + 1.3 \times \text{Power} - 0.4 \times \text{Aliveness} - 0.7 \end{aligned} \quad (2)$$

$$\begin{aligned} \text{OPV 3 (Cluster 3)} \\ = 3.5 \times \text{Loudness} + 0.8 \end{aligned} \quad (3)$$

The results produced by Eqs. (1)-(3) show that test subjects in Cluster 1 preferred a "quiet, smooth sound," while subjects in Cluster 2 preferred a sound that was "powerful rather than alive," and subjects in Cluster 3 preferred a "loud sound." Figure 6 shows a scatter diagram of the SPVs obtained from the results of the subjective evaluation experiments and the OPVs calculated using Eqs. (1)-(3), and the correlation coefficients between them. High correlation coefficients were obtained for all of the clusters, indicating that the SPVs had been successfully quantified.

3.2. Quantification of Sound Preferences in the Medium Engine Speed Range

In the same way, preferences in the medium engine speed range were quantified by first calculating subjective preference values for each test subject, then conducting cluster and multiple regression analyses.

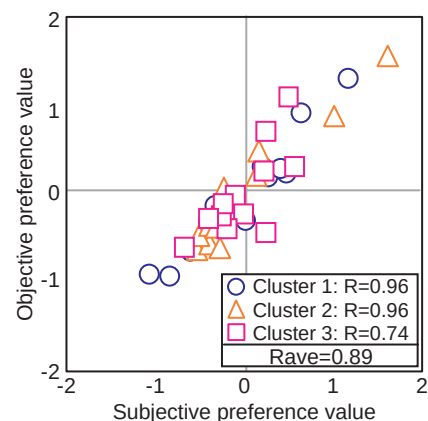


Fig. 6 Correlation between subjective and objective preference values at low engine speed

Cluster analyses of medium engine speed sound preferences generated four cluster types. The ratio of subjects in each cluster is shown in Fig. 7.

The results differ from the results obtained for the low engine speed range. Cluster 1 has the highest ratio, at around 50%, while Cluster 2 also has a high ratio at 40%.

As was the case for the low engine speed range, multiple regression analysis was conducted for each cluster. The results for Clusters 1 through 4 are shown as Eqs. (4) through (7), respectively.

$$\begin{aligned} OPV\ 1\ (Cluster\ 1) \\ = -3.4 \times Loudness + 1.2 \times Smoothness + 0.1 \end{aligned} \quad (4)$$

$$\begin{aligned} OPV\ 2\ (Cluster\ 2) \\ = -1.1 \times Loudness + 0.8 \times Power - 0.8 \times Aliveness \end{aligned} \quad (5)$$

$$\begin{aligned} OPV\ 3\ (Cluster\ 3) \\ = 1.1 \times Loudness + 0.5 \times Aliveness - 0.3 \end{aligned} \quad (6)$$

$$\begin{aligned} OPV\ 4\ (Cluster\ 4) \\ = -3.9 \times Loudness + 0.6 \times Aliveness \end{aligned} \quad (7)$$

The results show that test subjects in Cluster 1 preferred a “quiet, smooth sound,” while subjects in Cluster 2 preferred a “powerful rather than alive sound,” those in Cluster 3 preferred a “loud, alive sound” and those in cluster 4 preferred a “quiet, alive sound.” As Fig. 8 shows, high correlation coefficients between the SPVs and OPVs were also obtained for each cluster in the medium engine speed range.

3.3. Quantification of Sound Preferences in the High Engine Speed Range

As with the low and medium engine speed ranges, preferences in the high engine speed range were quantified by first calculating subjective preference values for each test subject, then conducting

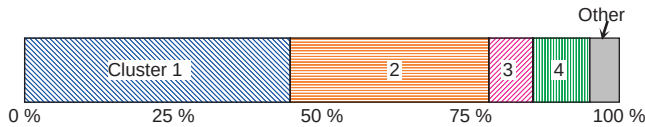


Fig. 7 Preference groupings at middle engine speed

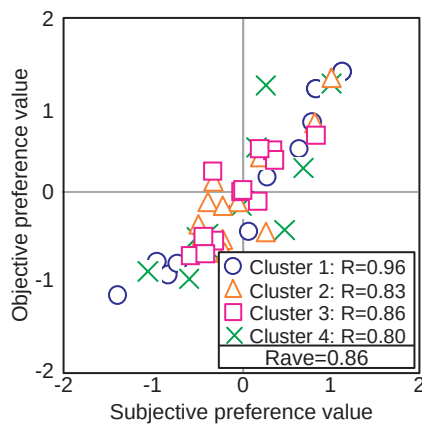


Fig. 8 Correlation between subjective and objective preference values at middle engine speed

cluster and multiple regression analyses.

Cluster analysis of the high engine speed range produced four clusters, the same as for the medium engine speed range. Figure 9 shows the ratio of test subjects in these clusters.

As with the other two engine speed ranges, multiple regression analysis was conducted for each cluster. The results for Clusters 1 through 4 are shown as Eqs. (8) through (11), respectively.

$$\begin{aligned} OPV\ 1\ (Cluster\ 1) \\ = -2.9 \times Loudness + 1.0 \times Smoothness + 1.7 \end{aligned} \quad (8)$$

$$\begin{aligned} OPV\ 2\ (Cluster\ 2) \\ = -5.3 \times Loudness + 2.1 \times Power - 0.4 \times Aliveness - 0.2 \end{aligned} \quad (9)$$

$$\begin{aligned} OPV\ 3\ (Cluster\ 3) \\ = 2.5 \times Loudness - 1.2 \end{aligned} \quad (10)$$

$$\begin{aligned} OPV\ 4\ (Cluster\ 4) \\ = -2.3 \times Loudness + 0.4 \times Smoothness + 0.8 \end{aligned} \quad (11)$$

The results show that subjects in Cluster 1 preferred a “quiet, smooth sound,” subjects in Cluster 2 preferred a “powerful rather than alive sound,” subjects in Cluster 3 preferred a “loud sound” and subjects in Cluster 4 preferred a “quiet, alive sound.” As Fig. 10 shows, high correlation coefficients were obtained between the SPVs and OPVs in each cluster in the high engine speed range.

As discussed above, the analyses conducted indicated three types of preference for interior sound during acceleration at low engine speeds, and four types each for interior sound during acceleration at medium and high engine speeds. The authors theorized that the reason the preference for a “quiet, alive sound” was quantified for the medium and high engine speed ranges, but not for the low engine speed range was that the high-order component, which is essential for the sensation of aliveness, was small at low engine speeds.

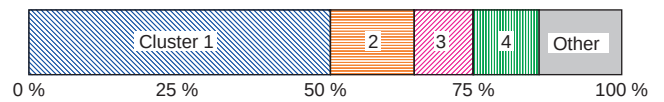


Fig. 9 Preference groupings at high engine speed

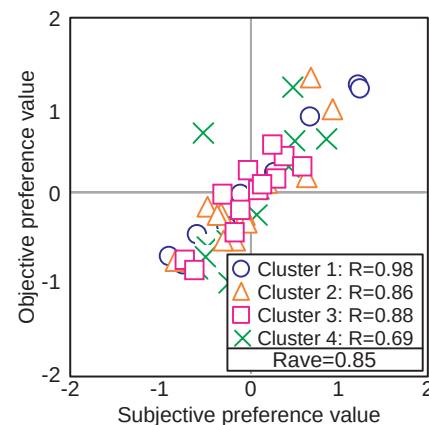


Fig. 10 Correlation between subjective and objective preference values at high engine speed

4. Questionnaire Survey of Test Subjects' Attributes

The subjects who participated in the subjective evaluation experiments were also asked to fill out a questionnaire survey to establish causal relationships between their preferences and their attributes. The questions in the survey concerned the following five areas:

- (1) Personal information
Gender, age, occupation, etc.
- (2) Car-related information
Type of car owned by the subject, level of interest in cars, criteria for choosing cars, etc.
- (3) Flexibility of opinion
Interest in new products, interest in meeting a wide range of people, etc.
- (4) Lifestyle
Degree of emphasis on career, desire to live in the country, etc.
- (5) Hobbies, leisure activities
Travel, sports, etc.

To determine what kind of person preferred what type of sound, the changes in the ratio of subjects in each cluster, i.e. the distribution ratio of subject preferences (termed the preference distribution ratio below) for each attribute was analyzed.

Figure 11 shows the preference distribution ratio for all test subjects before dividing them into attribute groupings. Out of all the engine speed ranges, Cluster 1 has the highest ratio, followed by Cluster 2.

Next, we focused on the interesting question of whether the preference distribution ratio differed between Japan, Germany and the United States, given certain differences in the automotive cultures of the countries. Figure 12 shows the results of the analysis by country. While there are some minor differences between the countries, the results show general similarities.

The rest of this section will analyze the preference distribution ratio with respect to a variety of test subject attributes, particularly in cases in which there is a comparatively high degree of difference in the preference ratio distribution for all test subjects.

Figure 13 shows the preference distribution ratio for each age group. These results indicate that the higher the age of the group, the more the subjects preferred a quiet, smooth sound, and the lower the age of the group, the more the subjects preferred a powerful rather than alive sound. A particularly high number of test subjects under the age of thirty preferred a powerful rather than alive sound in the medium engine speed range.

Figure 14 shows the preference distribution ratio by gender. These results show that a higher number of men preferred a powerful rather than alive sound, while a higher number of women preferred a quiet, smooth sound. A higher proportion of men than women preferred a loud, alive sound.

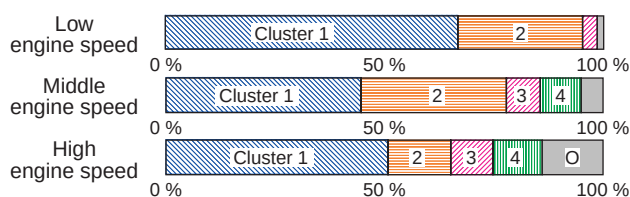


Fig. 11 Preference groupings among all subjects

Figure 15 shows preference organized by degree of interest in cars. These results show that the largest number of people who were interested in cars preferred a powerful rather than alive sound, while the largest number of people who were uninterested in cars preferred a quiet, smooth sound.

In total, the results obtained indicated that larger numbers of older people and women preferred a quiet, smooth sound, while larger numbers of younger people, men and those with an interest in cars preferred a powerful rather than alive sound. The results also indicated that a larger number of men than women preferred a loud, alive sound.

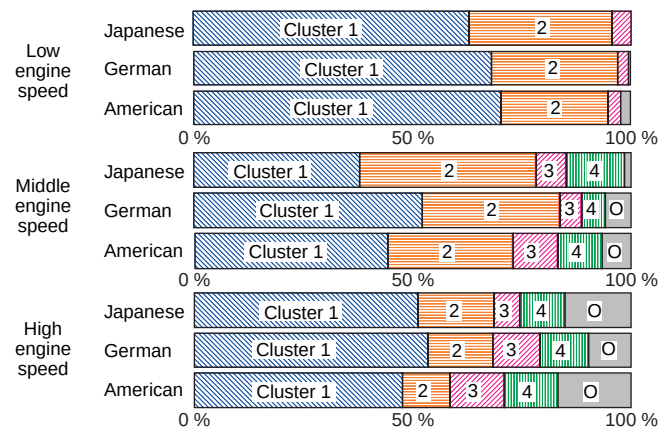


Fig. 12 Preference groupings for each nationality

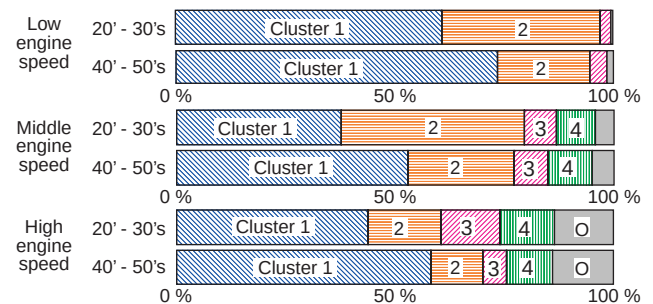


Fig. 13 Preference groupings for each generation

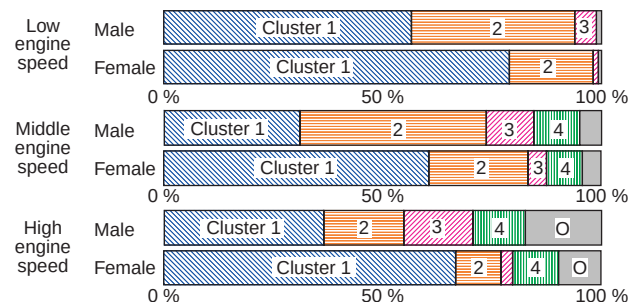


Fig. 14 Preference groupings for each gender

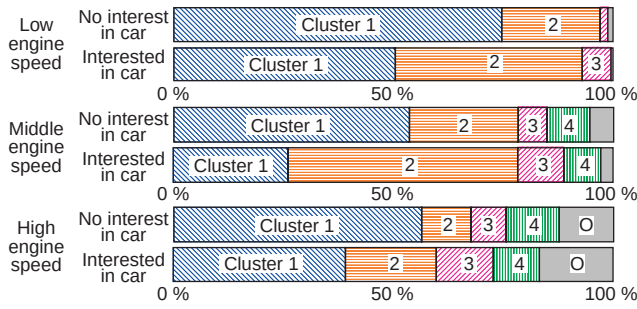


Fig. 15 Preference groupings depending on subject's car interest

5. Development of a System for Evaluation of Objective Preference Value

If the users targeted by a vehicle development concept can be roughly defined, the method discussed in this paper can be used to select the objective preferences that should be focused on in the development process, and determine the makeup of the sound those preferences relate to. A system for evaluating objective preference values was developed to enable efficient evaluations during the product development process. The system numerically processes

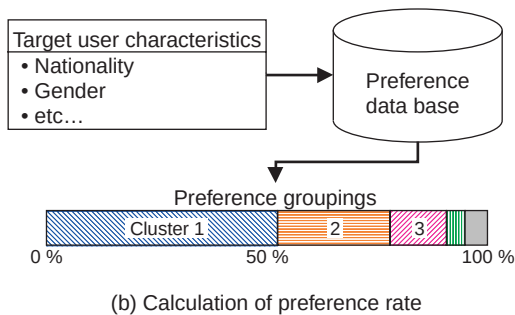
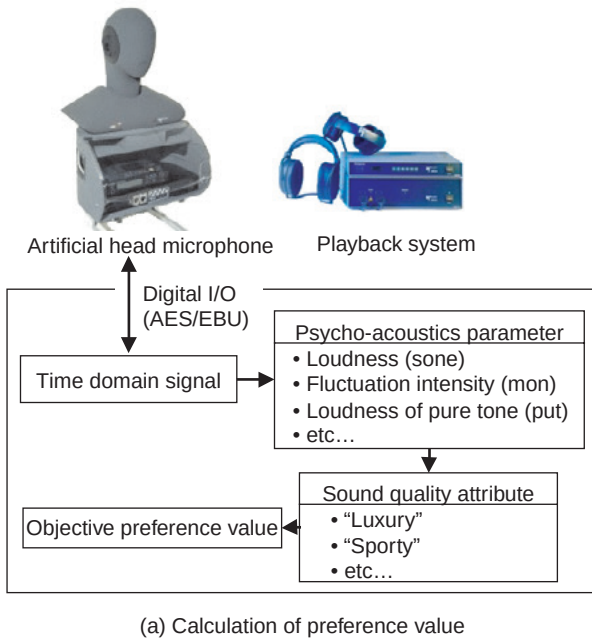


Fig. 16 Preference evaluation system

sound pressure signals recorded by an artificial head microphone to generate metrics for auditory stimuli, including loudness and fluctuation intensity, and for complex sound quality evaluation, including luxuriousness and smoothness. These values allow preferences to be calculated in the low, medium and high engine speed ranges. Figure 16 shows an outline of the system.

As an example, Fig. 17 shows the results of an analysis of the interior sound at full throttle acceleration in the medium engine speed range for two vehicles, A and B, while Fig. 18 shows the preference ratio when the target users are Japanese in their 20s and 30s with an interest in cars.

Based on these results, car B was judged preferable by the larger number of users because of its powerful rather than alive sound.

As this indicates, the system enables objective preference values to be calculated from measured sounds, and simplifies the assessment of whether target users will like the sound.

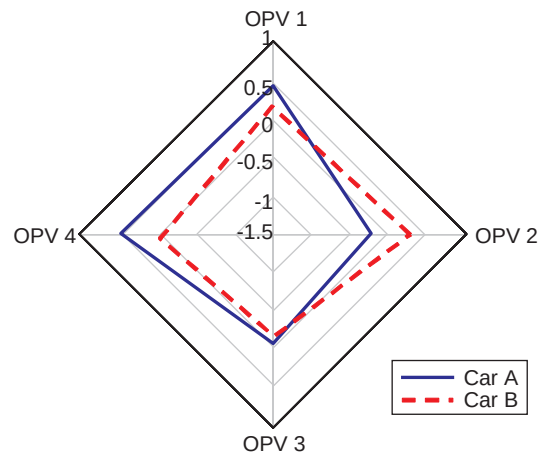


Fig. 17 Calculated preference value

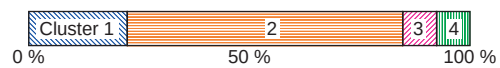


Fig. 18 Preference groupings in selected subject's characteristics

6. Conclusion

The research project discussed in this paper conducted subjective evaluation experiments and questionnaire surveys with the participation of approximately 600 test subjects in Japan, Germany and the United States. The following significant points were determined in this project:

- (1) The research determined three types of interior sound preferences in the low engine speed range, and four types each in the medium and high engine speed ranges.
- (2) It was possible to express the subjective preferences that had been determined as objective sound preference values.
- (3) The attributes of the test subjects, in particular, gender, age and degree of interest in cars, were shown to be factors that broadly divided the subjects' sound preferences.

- (4) A system enabling the evaluation of objective preference values was developed to assist in clarifying measures to modify sound to attain the sound quality preferred by target users.

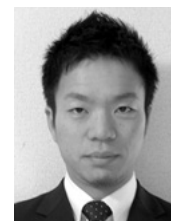
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Kousuke NOUMURA



Junji YOSHIDA

Selection Technique of Sound Absorption Type Acoustic Insulation Material by Applying SEA Modeling Method

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Takayuki DOI*
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ABSTRACT

This paper describes an effective method of developing a soundproof package, which balances both light weight and high noise insulation performance. Since it is necessary to propose the design of sound insulation in the early stages of development, the hybrid statistical energy analysis (SEA) modeling method is applied, which is practical for high frequency analysis. Also an acoustic characteristic estimation technique for multi layer structures is used.

As a result of applying these effective methods, the new 2006 CIVIC has enhanced road noise quietness and decreased weight when compared with the previous model.

1. Introduction

Enhancing quietness of automotive interiors is an important factor in increasing commercial appeal of vehicles. Meanwhile, reducing vehicle weight is an essential factor for achieving fuel efficiency targets. In general, achieving the quietness target by enhancing soundproof performance is incompatible with achieving the weight target. Until now, the investigation of noise treatments had been performed in the middle phase of the vehicle development, when the layout of these components is restricted. Therefore, the investigation for reducing noise in that stage inevitably led to an increase in weight and the difficulty of making effective and efficient countermeasures. In this paper, it is described how the soundproof package is developed using a hybrid SEA modeling method^{(1),(2)} and an acoustic characteristic prediction method of multi-layered porous materials^{(3),(4)} for the purpose of achieving the new CIVIC's goals for quietness and weight. The hybrid SEA modeling method is an analytical simulation technique which is an efficient and quantitatively verifiable method used to evaluate interior noise reduction in the early stage of a vehicle development. In this investigation, we used these technologies to develop a proposal to satisfy total acoustic performance through sound absorption and noise insulation. This paper will discuss details of this investigation. In addition to the noise absorption and insulation

performance, the road noise confirmation results for the actual vehicle are presented.

2. Methodology

2.1. Development Process

The new 2006 CIVIC represented a full model change. The targets were to reduce the weight of the soundproof package and enhance road noise quietness versus the previous model. To achieve these, the following steps for developing light weight soundproof package were applied;

- (1) Carried out a contribution analysis of the previous model using the hybrid SEA modeling method.
- (2) Selected the SEA subsystems which have very high contribution to the interior cavity.
- (3) Set damping loss factors (DLFs) that enabled achievement of the road noise target, but kept coupling loss factors (CLFs) between the interior cavity and the SEA subsystems selected in step (2) the same as the previous model.
- (4) Investigated a soundproof package which is 30% lighter than that of the previous model using the prediction method for acoustic characteristics of multi-layered porous materials, while keeping the DLF & CLF SEA parameters defined in step (3).

* Tochigi R&D Center

2.2. Method of Analysis

This section discusses the hybrid SEA modeling method and the acoustic characteristic prediction method of multi-layered porous materials.

2.2.1. Hybrid SEA modeling method

The hybrid SEA modeling method combines analytical and experimental SEA models for representing the energy flow among the SEA subsystems as accurately as that of an actual vehicle. The model is constructed by the following six steps:

- (1) Create the analytical SEA model
- (2) Define power input subsystems
- (3) Predict dominant power paths between the power input subsystems and the interior cavity subsystem and formulate network diagram of the each dominant power path
- (4) Measure the SEA parameters
- (5) Replace the measured SEA parameters with the analytical ones which are in the model created in step (1)
- (6) Carry out hybrid SEA process

2.2.2. Acoustic characteristic prediction method of multi-layered porous materials

In this development, three-layered noise insulating structure is applied, which contains base steel panel and two layers of felt (see Fig. 1). Applying this structure, it is possible to reduce in total noise insulating material weight since the mass layer is removed. Generally, that would cause acoustic insulation performance to decline. However, by employing high-porosity materials with excellent acoustic absorption characteristics, an overall enhancement in acoustic characteristics can be realized.

A relationship between an acoustic pressure p_i at interface 1 and 2 of the felt 1 and a particle velocity u_i can be expressed as follows:

$$\begin{Bmatrix} p_1 \\ u_1 \end{Bmatrix} = \begin{bmatrix} \cosh(\gamma_1 l_1) & (Z_1) \sinh(\gamma_1 l_1) \\ (1/Z_1) \sinh(\gamma_1 l_1) & \cosh(\gamma_1 l_1) \end{bmatrix} \begin{Bmatrix} p_2 \\ u_2 \end{Bmatrix} \quad (1)$$

In the same way, a relationship between an acoustic pressure p_i at interface 2 and 3 of the felt 2 and a particle velocity u_i can be expressed as follows:

$$\begin{Bmatrix} p_2 \\ u_2 \end{Bmatrix} = \begin{bmatrix} \cosh(\gamma_2 l_2) & (Z_2) \sinh(\gamma_2 l_2) \\ (1/Z_2) \sinh(\gamma_2 l_2) & \cosh(\gamma_2 l_2) \end{bmatrix} \begin{Bmatrix} p_3 \\ u_3 \end{Bmatrix} \quad (2)$$

A relationship between an acoustic pressure p_i at interface 3 and 4 of the base plate and a particle velocity u_i is:

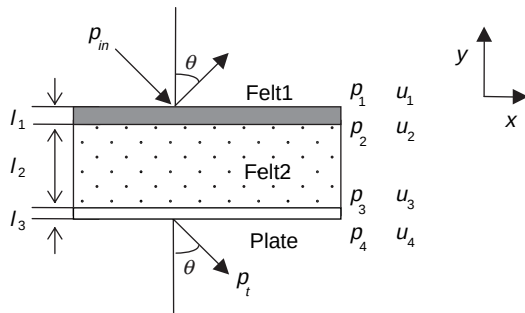


Fig. 1 Detail of three layers of a multilayered material

$$\begin{Bmatrix} p_3 \\ u_3 \end{Bmatrix} = \begin{bmatrix} 1 & Z(3) \\ 0 & 1 \end{bmatrix} \begin{Bmatrix} p_4 \\ u_4 \end{Bmatrix} \quad (3)$$

Equation (4) expresses a relationship between input acoustic pressure p_{in} and interface acoustic pressure p_i , and Eq. (5) shows a relationship between transmitted acoustic pressure p_t and interface acoustic pressure p_i :

$$p_{in} = p_i (Z(1) + \rho C_x) / 2 Z(1) \quad (4)$$

(ρ : Air density; C_x : Velocity of sound in air)

$$p_t = p_i \quad (5)$$

Where Z_i is an impedance of each layer, $Z_{(i)}$ is an acoustic impedance at the surface of each layer, γ_i is a transmission constant of the i th felt layer, and l_i is a thickness of the i th felt layer. Because the felt is fibrous material, the impedance is determined using an equivalent fluid model. An input sound pressure p_{in} and a transmitted sound pressure p_t from Eqs. (1)-(5) can be substituted in Eq. (6). Moreover, with solving Eqs. (7) and (8) a transmission loss is determined.

$$\tau_\theta = \frac{(p_t^2)}{(p_{in}^2)} = |T_r|^2 \quad (6)$$

$$\tau = 2 \int_0^{\theta_{lim}} d\theta |T_r|^2 \sin \theta \cos \theta \quad (7)$$

(θ_{lim} : limit value of angle)

$$TL = 10 \log_{10}(1/\tau) \quad (8)$$

Acoustic absorption is calculated using Eqs. (9)-(11).

$$R = \frac{Z(1) - Z_0}{Z(1) + Z_0} \quad (9)$$

$$\alpha_\theta = 1 - |R|^2 \quad (10)$$

$$\alpha = 2 \int_0^{\theta_{lim}} d\theta \sin \theta \cos \theta \alpha_\theta \quad (11)$$

3. Results of Analysis

3.1. Result of Contribution Analysis with the Hybrid SEA Model

For verifying the accuracy of the hybrid SEA model shown in Section 2.2., predicted and measured values of the interior noise are compared using the previous model CIVIC driving over a road noise evaluation course at 80 km/h. Figure 2 shows the results of this comparison. The difference between predicted and measured road noise is less than ± 3 dB. Therefore, it is safe to say that the hybrid SEA model is accurate enough.

Figure 3 shows the power contribution to the interior cavity from the contiguous subsystems, which is obtained from the hybrid SEA model. According to this graph, sound transmission from the floor including the tunnel and from the trunk have very high contribution to the interior cavity in the frequency range between 800 Hz and 2 kHz. Moreover, the sound leakage around the doors over 2 kHz is significant.

3.2. Selection of Important Parts for CLF

The result of the contribution analysis of Section 3.1. indicates the importance of the CLF of the floor carpet and tunnel, and the DLF of the trunk. Moreover, a countermeasure for sound leakage from the doors is necessary. In terms of the interior cavity soundproof package, the CLF of the floor carpet should be maintained at the same level as that of the previous model, while noise insulating materials are developed to reach the required interior cavity DLF for achieving the road noise target. This paper describes only the development of the interior cavity soundproof package. However, for achieving the overall target, other countermeasures are applied, which involve increasing the DLF of the trunk and enhancing the door seals.

3.3. Determination of Target Interior Cavity DLF to Enable Achievement of Road Noise Targets

Figure 4 shows the target DLF of the interior cavity and the DLF of the previous model. The target DLF of the interior cavity is set assuming that the road noise input power and the CLFs between the interior cavity and contiguous SEA subsystems remain at the same level as those of the previous model. Moreover, the effect from the other countermeasures to interior cavity is considered simultaneously.

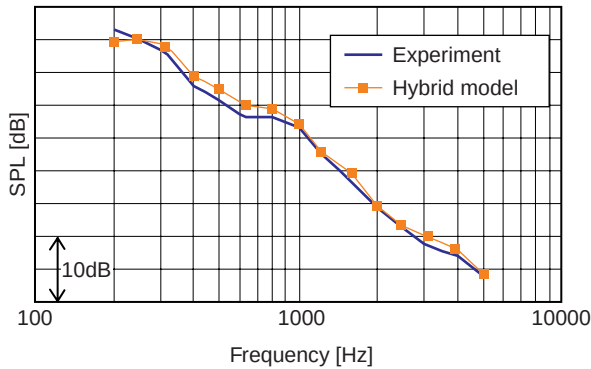


Fig. 2 Interior cavity sound pressure level (Road noise analysis)

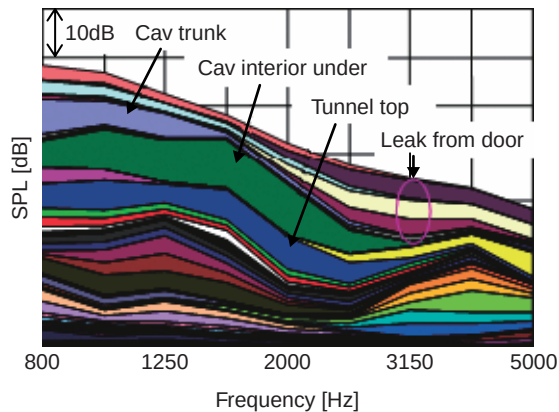


Fig. 3 Power contribution to interior cavity sound pressure level

3.4. Study of Lightweight Soundproof Package

This section will focus on the characteristics of the floor carpet, developed based on the three-layered noise insulating structure shown in Fig. 1 as part of the target to achieve the interior cavity DLF.

3.4.1. Floor carpet DLF and CLF

For achieving the DLF target of the interior cavity, the target absorption coefficients of each sound absorber (α) are reverse-calculated from the sound absorber surface area ratio to the whole interior cavity surface area. According to this result, the target absorption coefficient of the floor carpet is aimed at approximately three times more than that of the previous model over 1 kHz frequency range. However, the target transmission loss of the floor carpet is set at the same level as that of the previous model, since the CLF of the floor carpet had to be maintained. The prediction method for the acoustic characteristics of the multi-layered porous materials (Section 2.2.2.) is applied for determining the following design parameters: Thickness of the porous material, characteristics of the material and most importantly the air flow resistance. This analysis is also done with consideration of the soundproof insulator layout, which is fixed mostly at the early stages of the vehicle development. As a result, the absorption coefficient satisfied the target (see Figs. 5 and 6). Meanwhile, the mass layer elimination causes a 1-2 dB decline in the transmission loss compared to the previous model.

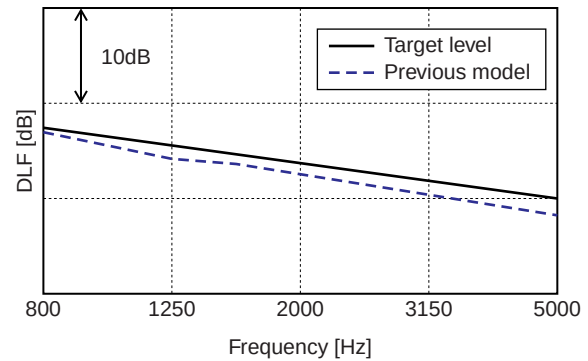


Fig. 4 Target for interior cavity DLF

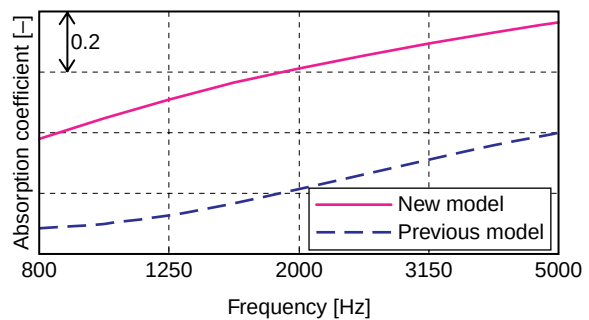


Fig. 5 Comparison of floor carpet absorption between previous and new models

3.4.2. Total acoustic characteristics of floor carpet

For determining how to recover the CLF decline by enhancing the interior cavity DLF, a simple SEA model with three subsystems (interior cavity, floor carpet and under-car cavity) was created (see Fig. 7). Figure 8 shows the difference of the interior cavity sound pressure level between the previous and new models' floor carpet designs. All other parameters are kept the same.

It is clear that the newly developed floor carpet is superior in the overall acoustic characteristic to the previous one.

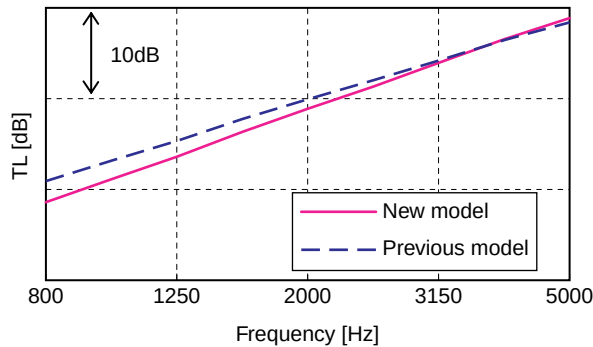


Fig. 6 Comparison of floor carpet transmission loss between previous and new models

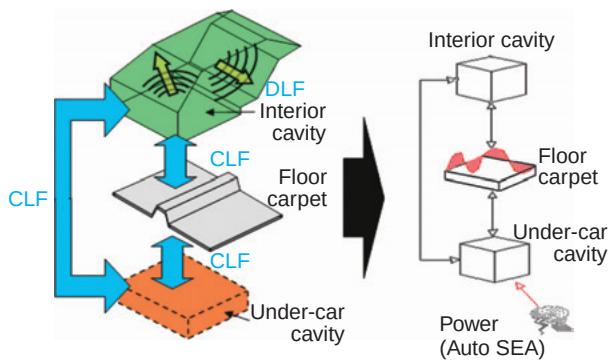


Fig. 7 SEA model (Three subsystems)

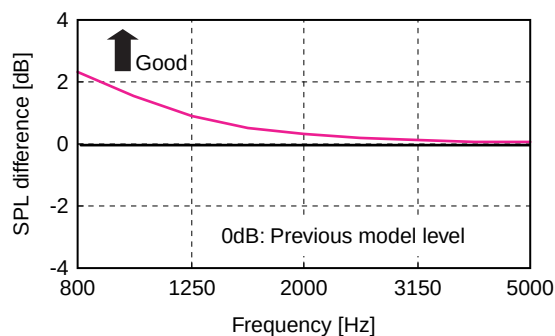


Fig. 8 Effect of new model floor carpet compared to previous model

4. Performance in a Vehicle

In this section, the performance of the vehicle with the newly developed soundproof package is discussed.

4.1. SEA Parameters (CLF and DLF)

The measurements verify that the new interior soundproof package is able to make the SEA parameters achieve the targets. Figure 9 compares the target DLF, which is shown in Fig. 5, with the measured DLF, and it is clear that the measured DLF has achieved the target. Figure 10 shows the floor to interior cavity CLF in the previous model versus the new model. This also indicates the achievement of the CLF target, which means the same level as the previous model was maintained.

4.2. Road Noise Level

Figure 11 shows the comparison for the interior road noise levels between the previous model, the new model with only the new interior soundproof package, and the new model with all noise countermeasures-enhanced door seals and an absorptive trunk package. The test condition is 80 km/h driving on a road noise evaluation course.

The newly developed interior cavity soundproof package reduces the sound pressure level in the high frequency range by approximately 2 dB as compared to the previous model. By applying all noise countermeasures, the reduction of approximately 2-4 dB is accomplished, and the primary target of this project is satisfied. In addition, the total weight of soundproof package has been reduced by more than 30% as compared to the previous model (Fig. 12).

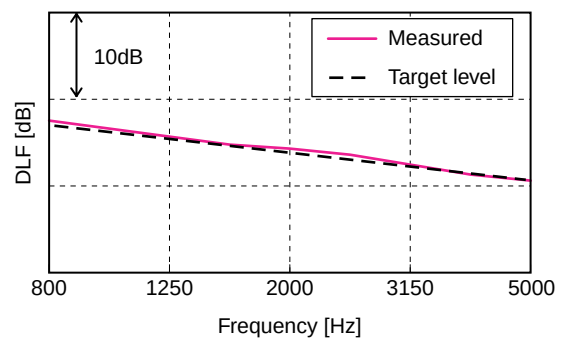


Fig. 9 Comparison of target and measured interior cavity DLF

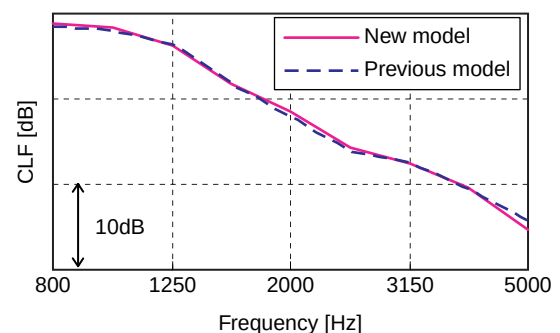


Fig. 10 Comparison of floor to cabin CLF between previous and new models

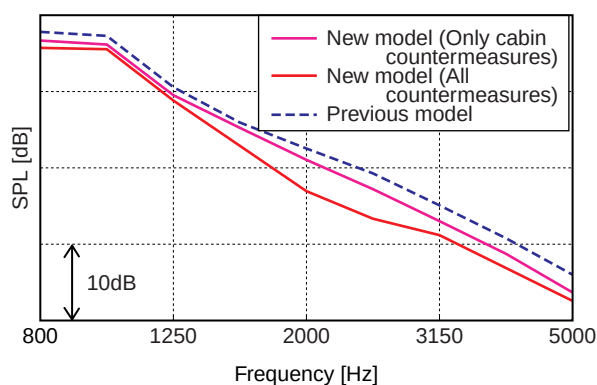


Fig. 11 Interior cavity sound pressure level at 80km/h on rough road

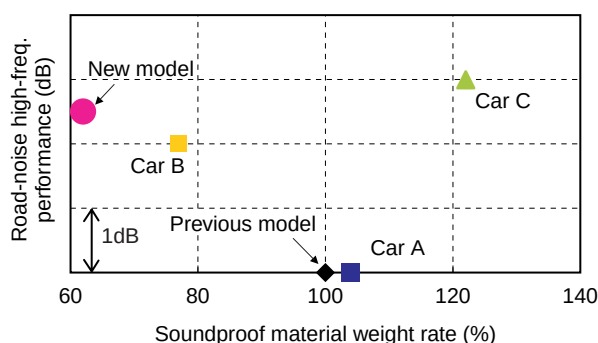


Fig. 12 Hierarchy map (weight rate / road-noise performance)

5. Conclusion

A predominantly sound-absorbing lightweight soundproof package has been developed using the hybrid SEA modeling method and the method of predicting the acoustic characteristics of multi-layered porous materials for the early stages of the development.

- (1) The validity of the hybrid SEA modeling method and the acoustic characteristic prediction technology of the multi-layered porous materials was confirmed.
- (2) A vehicle cabin noise prevention package, including specifications for a floor carpet that achieved total noise characteristics through noise absorption and noise insulation was developed.
- (3) As a result, the target of a 2 dB reduction in the road noise level in the high-frequency range was achieved, and the weight of the soundproof package was reduced by more than 30%.
- (4) These accomplishments enable the new 2006 CIVIC to achieve high performance in weight management and high frequency quietness.

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Analysis of Aluminum Alloy Surface Formed by Rust-preventive Additive Component of Coolant

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ABSTRACT

Coolant protects the engine water jacket against rusting, in other words, it acts as a rust preventive. The rust preventive performance depends upon the type and quantity of the additive.

Automotive coolants used in recent years can be broadly divided into three categories depending on the type of additive used: organic acid coolants, phosphate coolants, and silicate coolants.

In this research project, we studied coolant additives that provided aluminum with corrosion resistance, by combining organic acid coolant with organic additives that have a long heat-resistant life. The coolants were compared by analyzing the aluminum surface after the corrosion condition.

1. Introduction

In recent years, with the progress of rust prevention additive techniques, companies around the world have been applying coolants that have longer replacement intervals. In the 1990s, replacement was generally done every 2 to 3 years, but recently some coolants have recommended replacement periods of 5 years or even longer. This prolonging of coolant replacement life is the result of using an organic rust prevention additive with a long thermal life, a conventional inorganic rust prevention additive, and by optimizing the combination of additives.

As shown in Fig. 1, different regions have adopted different coolant formulations for quite some time^{(1), (2)}. Recently, Europe and North America have primarily used two types of coolants: an organic acid coolant that uses only organic additives or a silicate coolant that uses inorganic silicate.

Japan has used coolants that differ distinctly from those used in Europe and North America. Japan used an amine coolant through the 1980s and during the 1990s. Japan later adopted phosphate coolants that contain a combination of phosphates and organic acid salts.

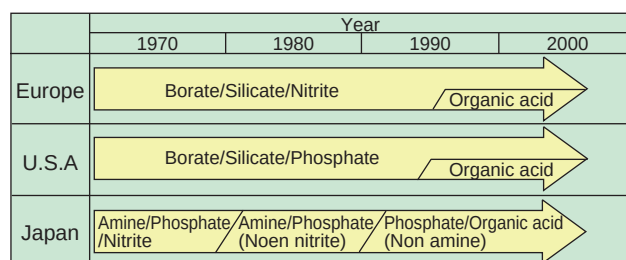


Fig. 1 Coolant formation in each region

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There are some reports that state the use of organic acid coolant causes aluminum erosion-corrosion⁽¹⁾. In order to enhance the anti-erosion and corrosion performance of organic acid coolant, this study was designed to research the rust prevention additive component of coolants.

Furthermore, surface analysis was performed on the aluminum after the corrosion test condition to determine the difference in effects between the types of coolant additives.

2. Test Method

2.1. Investigation of Organic Acid Coolant

When investigating the rust prevention additive of a coolant, it is important to ensure that it can prevent aluminum rust under a lotic environment. In other words, that it has good anti-erosion and corrosion performance⁽¹⁾. Several types of organic acid coolants were created and their aluminum rust prevention performance was evaluated in flow of coolant.

The following items were considered for the selection of the combination of rust prevention additives.

- (1) Organic carboxylate, which has been used as a coolant rust prevention additive before, was selected as the base for the coolant.
- (2) Organic compounds that have chemical structures or functional groups with the potential to prevent rust in aluminum were used with the base coolant (1).

2.1.1. Sample preparation

Table 1 shows the evaluated coolant compositions. The base material for all of the compositions was ethylene glycol. Each formulation was diluted with water to a concentration of 30 volume % and adjusted to a pH of 7.5 - 8.0.

Phosphate coolant and silicate coolant were prepared for comparison.

2.1.2. Test method

A jet erosion-corrosion test apparatus (shown in Fig. 2) was used to evaluate the metal rust prevention performance of the coolants in flow of coolant⁽¹⁾. A metal test piece, regulated by JIS K2234⁽³⁾, was set in the test equipment. After circulating heated coolant for a determined amount of time, the aluminum test samples (AC2A) were checked for appearance and changes in weight.

2.2. Aluminum Surface Analysis after the Jet Erosion-Corrosion Test

A surface analysis was performed on each aluminum test piece after the jet erosion-corrosion test.

2.2.1. Sample and test piece

The aluminum test pieces used in the jet erosion-corrosion test with the three types of coolant below were selected for analysis.

- (1) Organic acid B coolant
- (2) Phosphate coolant
- (3) Silicate coolant

To observe changes in the surface of the test pieces before and after the test, the aluminum test pieces were polished to a mirror-surface finish prior to the test.

2.2.2. Analysis

Aluminum surface analysis methods are shown below.

- Magnified appearance of surface:
Magnified appearance of surface using a microscope

Table 1 Test coolant composition

Test coolant Additive composition	Organic acid coolant						Comparative	
	A	B	C	D	E	F	Phosphate coolant	Silicate coolant
Carboxylate-a	●							
Carboxylate-b	●	●	●	●	●	●	●	●
Carboxylate-c		●	●	●	●	●	●	●
Amino acid			●					
Organic nitride				●				
Organic sulfide					●			
Saccharide						●		
Phosphate							●	
Silicate								●
Triltriazole, etc	●	●	●	●	●	●	●	●
Others		●	●	●	●	●	●	●

- Chemical composition analysis of surface:
Analysis of chemical composition in the direction of surface depth using glow discharge spectroscopy

2.3. Aluminum Surface Changes when Coolant was Replaced

Two types of coolant, with different compositions, were used to examine the affect on the aluminum surface when a coolant with a different additive is used during a jet erosion-corrosion test.

2.3.1. Sample

Phosphate coolant and organic acid B coolant were selected as the test coolants.

2.3.2. Test method

After coolant was circulated for a fixed time in the jet erosion-corrosion test apparatus of Fig. 2, in which a metal test piece has been placed. The coolant was replaced by one with a different composition and further circulated. The external appearance and change in mass of the aluminum after the test were checked.

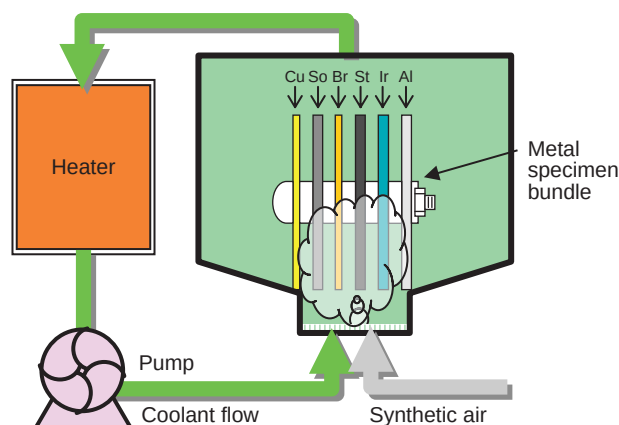


Fig. 2 Jet erosion-corrosion test apparatus

Test 1: Phosphate coolant

The coolant was replaced by organic acid B coolant in the middle of the test.

Test 2: Organic acid B coolant

The coolant was replaced by phosphate coolant in the middle of the test.

3. Results and Considerations

3.1. Investigation of Organic Acid Coolant

Table 2 shows the results of change in mass and external appearance of aluminum after the test. The reduction of mass of the metal test piece is considered to be caused by elution of the metal composition into the coolant. Therefore, coolants with lower amounts of mass reduction are judged to have better rust prevention performance.

Organic acid A coolant mainly uses organic carboxylate as rust prevention additives and its aluminum erosion-corrosion performance was found in previous evaluations^{(1), (4)}. From the test result, it was found that the mass of aluminum was greatly reduced by organic acid A coolant. In contrast, organic acid B coolant, which also includes organic carboxylate, was able to inhibit the mass reduction up to -0.02 mg/cm^2 .

The result of organic acid coolants C through F showed no enhancement of aluminum rust prevention performance, even though new organic materials were used.

On the other hand, the aluminum maintained a shiny silver external appearance similar to the initial appearance. In addition, the mass reduction of aluminum was low and results obtained were more favorable than organic acid formula coolants.

In this study, organic carboxylate and various organic compounds were combined to create organic acid coolants and the coolants were evaluated. Mass reduction was present in all organic acid coolants tested. This research did not lead to finding a coolant that has better aluminum rust prevention performance than phosphate or silicate coolants.

This investigation resulted in a reconfirmation that, although the organic acid coolant can inhibit the aluminum mass reduction, phosphate and silicate coolants are more effective at aluminum rust prevention than organic acid formula coolant.

Table 2 Results of coolant test

		Mass	Appearance
Organic acid coolant	A	-0.83 mg/cm^2	Black
	B	-0.02 mg/cm^2	Black
	C	-0.05 mg/cm^2	Black
	D	-0.08 mg/cm^2	Black
	E	-0.17 mg/cm^2	Black
	F	-0.05 mg/cm^2	Black
Comparative	Phosphate coolant	$+0.01 \text{ mg/cm}^2$	Silver
	Silicate coolant	$+0.01 \text{ mg/cm}^2$	Silver

3.2. Aluminum Surface After the Jet Erosion-Corrosion Test

3.2.1 Appearance of surface under magnification

Figure 3 shows magnified pictures and external appearances of the aluminum test pieces before and after the test. Electron Probe X-ray Micro Analyzer (EPMA) analysis confirmed that the areas on the test samples that look black and grey were precipitates of silicon and copper. These are components of the aluminum alloy.

For phosphate and silicate coolants, there were no significant changes in the external appearance or under magnification of the aluminum surface after testing.

On the other hand, there were significant changes in the external appearance and under magnification of the aluminum surface after testing using the organic acid coolant. The aluminum portion had all changed to black. The silicon that precipitated into rod shapes in the alloy appeared to be white and floating. This is believed to be because the aluminum surface was slightly corroded by flow of

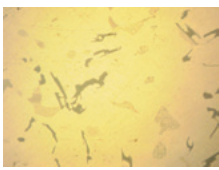
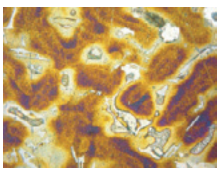
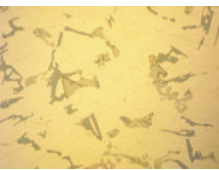
	Reference	After test		
		Organic acid coolant	Phosphate coolant	Silicate coolant
Appearance				
Magnification (x 1200)				
Mass	—	-0.02 mg/cm^2	$+0.01 \text{ mg/cm}^2$	$+0.01 \text{ mg/cm}^2$

Fig. 3 Appearance and mass of aluminum after coolant test

coolant. As for silicon, which is harder than aluminum, it was not corroded and therefore remained intact.

These results provide that the mass reduction of the aluminum test piece for organic acid coolant was caused by slight corrosion of the aluminum surface.

3.2.2 Chemical composition of the surface in the depth direction

Figure 4 shows chemical composition measurement results in the depth direction of the aluminum surface before testing. It is generally known that aluminum forms a very thin, passive, oxide layer on its surface that protects aluminum, in open air, from corrosion⁽⁵⁾. Therefore, the layer of aluminum and oxygen detected on the surface prior to testing is an aluminum oxide layer.

This aluminum oxide layer was maintained on the aluminum surface after testing with phosphate and silicate coolants (Figs. 5 and 6).

Aluminum and oxygen were detected on the surface using the organic acid coolant. However, a thick oxygen layer was also formed, which has different characteristics from that of phosphate or silicate coolants (Fig. 7). From Raman spectral analyses or X-

ray analyses, this thick oxygen layer is considered to be a layer formed primarily of aluminum hydroxide^{(1), (6)}.

The surface under magnification in Fig. 3 shows that the surface layer of aluminum formed by the organic acid coolant was slightly corroded by the flow of the coolant. Therefore, the thick aluminum hydroxide layer formed when using organic acid coolant could peel easily in flow of coolant, compared to the aluminum surface layer found with the phosphate coolant and silicate coolants (Fig. 8).

3.3. Aluminum Surface Changes when Coolant is Replaced

When coolant was replaced, the external appearance of the aluminum changed to brown-red in both tests 1 and 2, and there was also mass reduction seen in each test. In addition, a slight change in color and roughness of the aluminum surface was found when observed under magnification (Fig. 9).

As to the chemical composition in the direction of surface depth, a surface layer with a thick oxygen layer was confirmed. This layer is similar to the one found after testing with only organic acid coolant (Figs. 10 and 11).

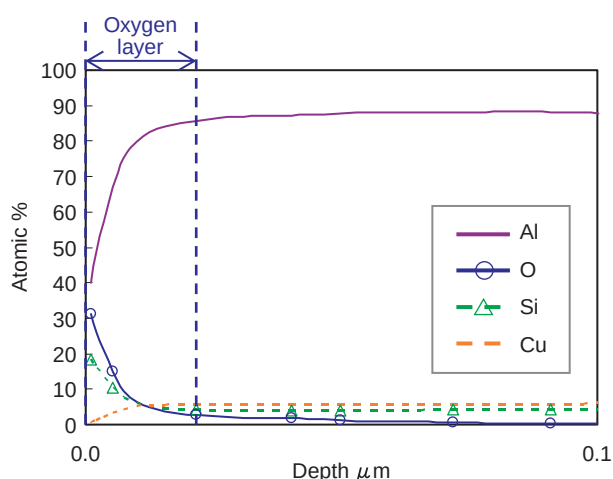


Fig. 4 Depth profile of each component of aluminum surface before coolant test

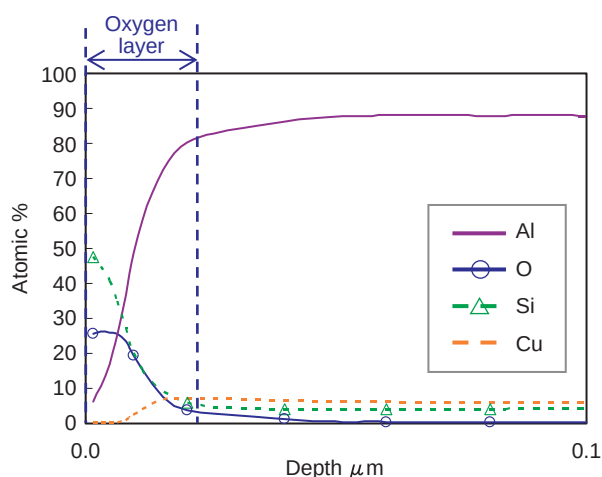


Fig. 6 Depth profile of each component of aluminum surface after silicate coolant test

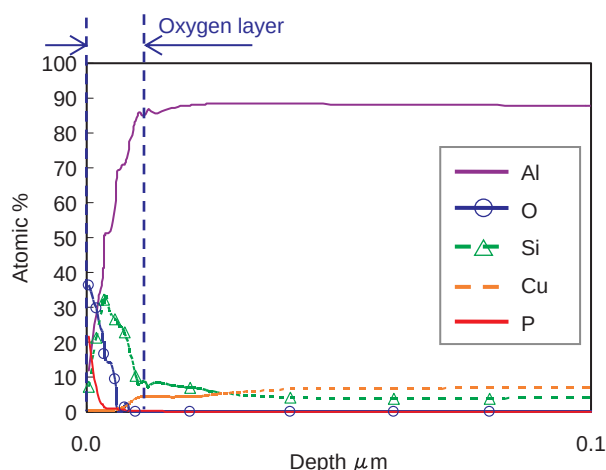


Fig. 5 Depth profile of each component of aluminum surface after phosphate coolant test

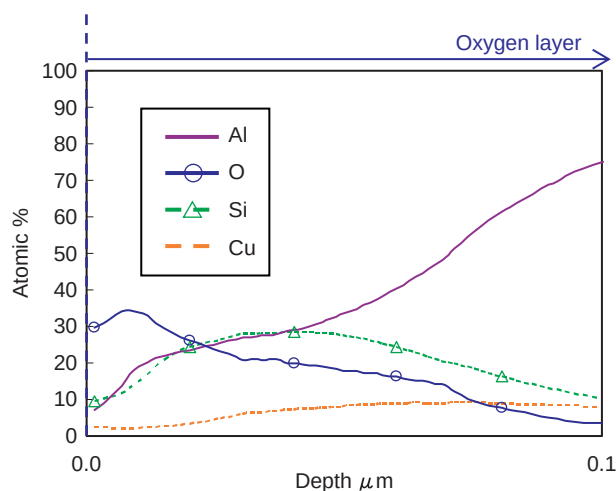


Fig. 7 Depth profile of each component of aluminum surface after organic acid coolant test

From these results, it was found that when phosphate coolant was used and then replaced by the organic acid coolant, the thin aluminum oxide layer on the surface was not maintained and the surface changed to the thick layer similar to that formed by the organic acid coolant.

It was found that when the organic acid coolant was used and then replaced by the phosphate coolant, the surface layer maintained the characteristics formed by the organic acid coolant and was not significantly changed by the phosphate coolant.

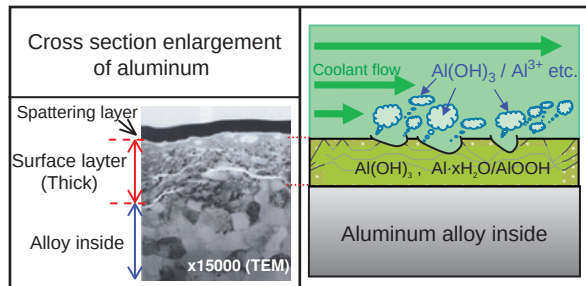


Fig. 8 Aluminum exfoliation mechanism in organic acid coolant flow

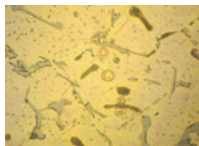
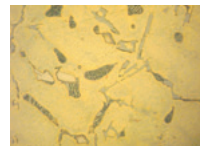
Beginning	Organic acid coolant	Phosphate coolant
Replacement	Phosphate coolant	Organic acid coolant
Appearance		
Magnification (x 1200)		
Mass	-0.06 mg/cm ²	-0.03 mg/cm ²

Fig. 9 Appearance and mass of aluminum after coolant change test

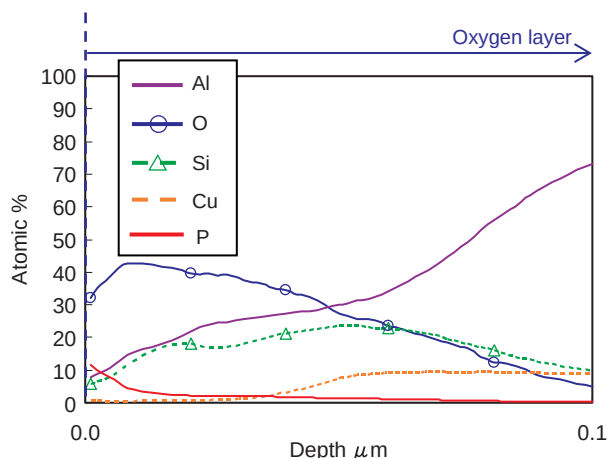


Fig. 10 Depth profile of each component of aluminum surface after coolant change test (phosphate → organic acid)

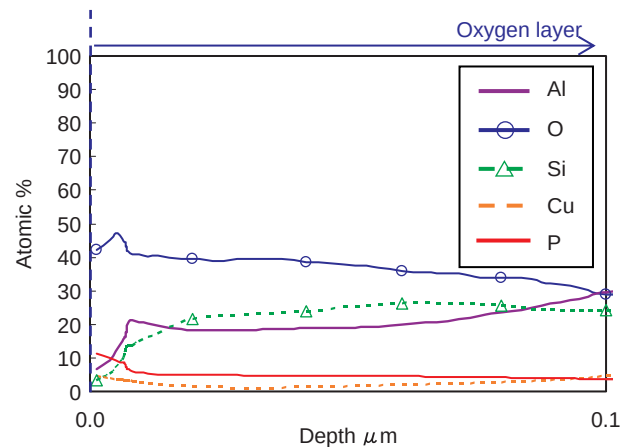


Fig. 11 Depth profile of each component of aluminum surface after coolant change test (organic acid → phosphate)

4. Conclusion

Through investigation of organic acid coolant and aluminum surface analysis, the following results were obtained.

- (1) Organic acid coolant was evaluated for rust prevention performance under a lotic environment, and it was able to inhibit the aluminum mass reduction. However, organic acid coolant could not achieve better results than phosphate or silicate coolants.
- (2) Organic formula coolant forms a thick layer on the surface of the aluminum. However, this layer peels easily in flow of coolant when compared to the aluminum surface layer found after testing with phosphate and silicate coolants.
- (3) When replacing phosphate coolant with organic acid coolant, the surface layer changed to a thick layer similar to that formed by organic acid coolant.
- (4) When replacing organic acid coolant with phosphate coolant, the surface layer maintained the thick layer formed by the organic acid coolant and was not significantly changed by the phosphate coolant.

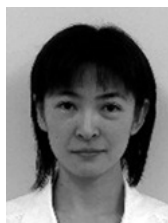
From these results, it can be stated that further research is needed to enhance aluminum anti-erosion and corrosion performance of organic acid coolants.

Lastly, it has been stated that silicate formula coolant has the effect of preventing aluminum from rusting. However, it has been reported that silicate affects the water pump seal⁽¹⁾ and that there are cases where it gels in solution⁽⁷⁾. Furthermore, since the silicate is consumed quickly⁽⁸⁾, it is necessary to consider its life of rust prevention performance.

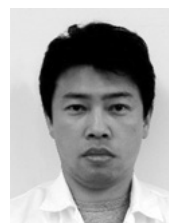
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Ayako OMORI



Mitsuru OSAWA

Development of Tank Shell and Tube Material for Plastic Fuel Tank to Comply with PZEV

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ABSTRACT

A plastic fuel tank system compliant with U.S. PZEV evaporative emission standards was developed and fitted on the new CIVIC-IMA. Techniques for reducing the high permeation that occurs from the tank shell and various tank system components, such as plastic tubes, pump gaskets, and other welded child parts, were studied.

EVOH with 24 mol% ethylene content was chosen for the low-permeation layer of the plastic fuel tank shell, and the fuel permeation rate was reduced to one-fifth that of the conventional tank shell.

Modified PPS was chosen for the low-permeation tubing material. The modified PPS was alloyed with an elastomer to produce a tube with the required flexibility. By using this material, the rate of fuel permeation of the tube alone could be reduced to one-tenth that of conventional tubes.

Additionally, compared to the regulatory value of 54 mg/day for the fuel tank system, a permeation rate of less than 54 mg/day was achieved during measurement using Phase III fuel after adding low permeation parts such as pump gaskets and welded child parts.

1. Introduction

U.S. PZEV evaporative emission standards, which came into force in 2004, are currently the most stringent in the world. These regulations require the fuel permeation rate from all parts of the fuel system to be below 54 mg/day, using Phase II fuel, including methyl tertiary butyl ether (MTBE), as the test solution⁽¹⁾.

However, gasoline additives in the U.S. market have been changing over from MTBE to ethanol, while test solutions have also been changing over to Phase III fuel, including ethanol. The swelling of plastic and rubber materials in ethanol is high compared to MTBE, and the rate of fuel permeation is also higher in ethanol. Thus, the test becomes more stringent in the case of Phase III fuels⁽²⁾.

In view of this background, steel tanks began to be used to clear PZEV emission standards.

Since plastic fuel tanks possess a high degree of freedom in shape and are lighter compared to steel tanks, more than 90% of all Honda vehicles are fitted with plastic fuel tanks. However, since fuel permeation from plastic fuel tanks is greater than it is from steel fuel tanks, technology allowing for compliance with PZEV standards was nonexistent. Therefore, a plastic fuel tank system that could comply with PZEV standards was in demand.

A plastic fuel tank system complying with PZEV evaporative emission standards has been developed, and its development is introduced in this report. The newly developed tank system has been fitted on the new CIVIC-IMA. Figure 1 shows the plastic

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fuel tank system on the new CIVIC-IMA and the parts that incorporate elemental technology for reducing fuel permeation. This completely built car fitted with a plastic fuel tank complies with PZEV standards in SHED tests using Phase III fuel. By using a fuel tank made of plastic, a weight reduction of about 3 kg was achieved, compared with the use of conventional steel fuel tanks.

2. Elemental Technology for Reducing Fuel Permeation

Figure 2 shows the measured results of fuel permeation from the plastic fuel tank system. Phase III fuel was used for the SHED method to measure the permeation. To examine its contribution to permeation, each component was closed by a blank plug, ensuring that no permeation occurred from the closed sections, and the rate of permeation was measured.

From the results, it was clear that the rate of permeation had to be reduced to less than one-third that of the current system to meet the regulatory limit of less than 54 mg/day. The reduction target of the tank shell was set as 1/5, and the tube target was set as 1/10. In addition the numerical target needed for each component to comply with the overall regulatory value was determined.

The five elements of technology that are required to reduce the permeation of fuel tank systems were studied.

As is evident from Fig. 2, these five elements (four components) account for about 90% of the total permeation quantity. Modification of these elements is expected to bring good results in reducing the permeation.

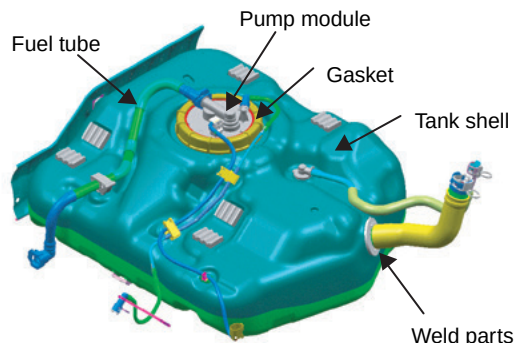


Fig. 1 Fuel tank system

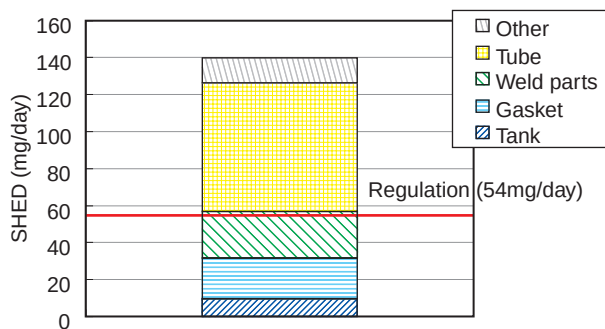


Fig. 2 Evaporative emissions from current fuel tank system

2.1. Tank Shell

The material composition in the fuel permeation barrier layer (hereafter referred to as “barrier layer”) of the plastic fuel tank was modified. In addition the thickness of the barrier layer was changed.

2.2. Plastic Tube Material

The barrier material of the plastic tube was changed to low-permeation polyphenylene sulfide (hereafter referred to as “PPS”), alloyed for higher flexibility in order to satisfy tube functions.

2.3. Reduction in Permeation of Pump Gasket

The pump gasket material was changed from the conventional NBR-PVC material to a fluorine-based material, increasing the initial tightening force and satisfying sealing performance.

2.4. Reduction in Permeation of Welded Child Parts

To restrict the permeation from welded child parts on the tank body, the child part materials were changed, and the welding conditions optimized.

2.5. Integration of Welded Child Parts

Welded child parts that could be integrated in the pump module were integrated, reducing the number of welded child parts, and thereby lowering the rate of permeation.

Among the five elements mentioned above, the new materials for tank shell and plastic tube are described here.

3. Tank Shell

3.1. Configuration of Fuel Tank Layers

As shown in Fig. 3, the plastic fuel tank has a 4-type, 6-layer configuration, and makes use of ethylene vinyl alcohol (hereafter referred to as “EVOH”) in the barrier material.

EVOH is a copolymer of ethylene and vinyl alcohol, and its structure is shown in Fig. 4. By changing component proportions, its permeation characteristics and mechanical properties can be changed⁽³⁾.

3.2. Reduction in Permeation of EVOH

The relationship between the proportion of ethylene in EVOH and the rate of fuel permeation is shown in Fig. 5.

Vinyl alcohol is polar, hindering molecules that try to permeate the tank wall. Thus, as the proportion of vinyl alcohol is increased, the barrier resistance of the EVOH material is enhanced^{(4), (5)}.

The ethylene content of EVOH for the PZEV tank shell was set at 24 mol%, which minimizes permeation to a manageable level and is suitable for the equipment currently used.

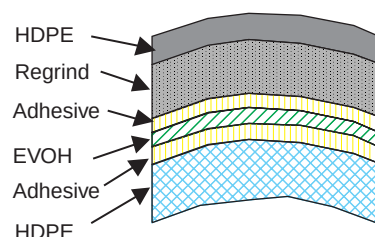


Fig. 3 Structure of fuel tank shell

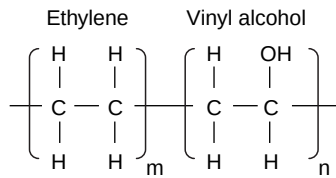


Fig. 4 EVOH structure

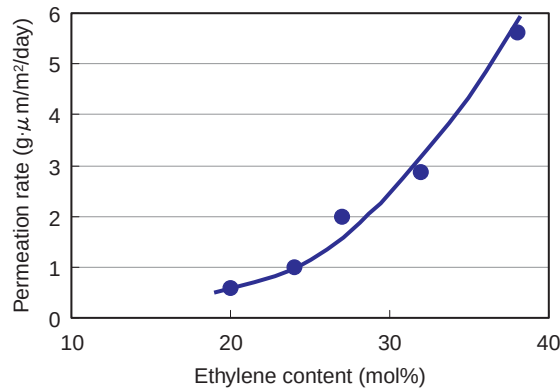


Fig. 5 Ethylene content vs. permeation rate

3.3. Optimization of Layer Configuration

Table 1 shows the basic properties of High Density Polyethylene (hereafter referred to as “HDPE”), the main component of the tank shell, and the basic properties of the barrier layer EVOH. EVOH is a harder material and has 1/50th the elongation as compared to HDPE.

The thickness of EVOH may be simply increased to reduce the rate of permeation from the tank shell, but issues such as impact strength reduction may be anticipated. Accordingly, tests were carried out to study the relationship between the thickness of the EVOH layer and the tensile impact strength of the tank shell. Tests were also carried out to study the relationship between the thickness of the EVOH layer and the rate of permeation. The results of both tests are shown in Fig. 6.

The rate of permeation can be reduced by increasing the EVOH thickness, but if the thickness increases above 4%, the impact value drops below that of the conventional configuration, which is the target value. Therefore, the thickness of the barrier layer of the plastic fuel tank for PZEV was set at 4% so that no reduction in impact strength occurs. If the EVOH thickness is reduced, the tensile impact strength does not change, but the rate of permeation increases. Therefore, the lower limit of the EVOH layer thickness was set at 2.5%.

Figure 7 shows the relationship between the position of the barrier layer and the rate of fuel permeation. It was observed that the highest permeation occurs when the PZEV barrier layer is located on the outside layer of the plastic fuel tank. The rate of permeation lowers as the barrier layer is positioned toward the inside of the tank, but due to restrictions with the current mass-production blow molding machine capability, the barrier layer position could not be located on the inside layer. Its location was finally moved in by 5% from its current location to 35% from the inside.

Table 1 General property of EVOH vs. HDPE

Item	Unit	HDPE	EVOH
Density	g/cm ³	0.95	1.19
Tensile strength	MPa	44	45
Elongation	%	>600	12
Flexural modulus	MPa	950	4770

3.4. Permeation from Tank Shell

Figure 8 shows the effect of the measures described above on the reduction of fuel permeation.

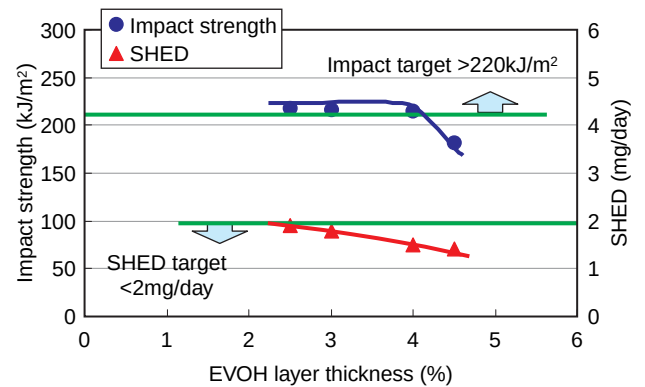


Fig. 6 EVOH layer thickness vs. impact strength and evaporative emissions

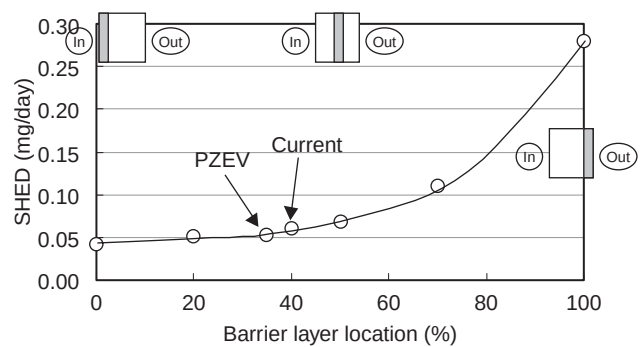


Fig. 7 Barrier layer location vs. evaporative emissions

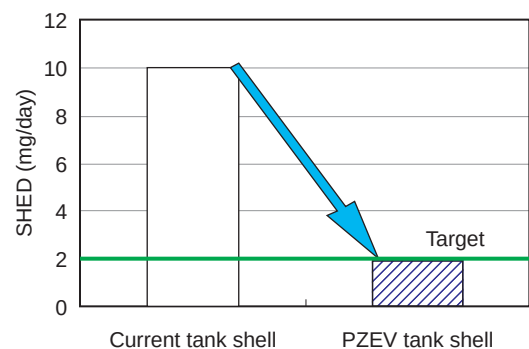
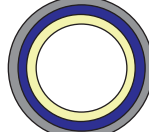
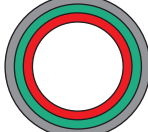
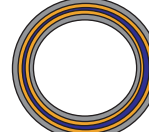


Fig. 8 Tank shell evaporative emissions

Table 2 Monitor material for low permeation tube

	ETFE	PPS	Polyester	Liquid crystal polymer	EVOH
Permeation resistance	Acceptable	Excellent	Excellent	Excellent	Good
Fuel resistance	Excellent	Good	Good	Good	Good
Chemical resistance	Excellent	Good	Acceptable	Good	Acceptable
Heat resistance	Good	Good	Acceptable	Good	No good
Flexibility	Good	Acceptable	Good	No good	Acceptable
Impact resistance	Good	Acceptable	Good	Acceptable	Acceptable
Total	Acceptable	Good	Acceptable	No good	No good

Table 3 Tube structure of present ETFE multi-layer tube and examined 2 types of low permeation tube

		ETFE multi-layer tube	PPS multi-layer tube	Polyester multi-layer tube
Structure				
Barrier layer		ETFE	PPS	Polyester
Number of layers		3	3	5
Layer composition	1 (Inner layer)	ETFE	PPS	PA11
	2	Adhesive	Adhesive	Adhesive
	3	PA11	PA11	Polyester
	4	—	—	Adhesive
	5 (Outer layer)	—	—	PA11

The plastic fuel tank shell permeation was reduced to one-fifth that of the conventional tank, and the target of less than 2 mg/day was achieved.

4. Tube Material

4.1. Selection of Tube Material

Since the target value was set at below 1/10th the conventional value for the plastic fuel tube, a barrier material with an extremely high permeation resistance was required.

Table 2 shows the results of the material evaluation for the barrier layer of the plastic fuel tube. The material evaluation included a comparative study of six parameters including permeation resistance, fuel resistance, chemical resistance, heat resistance, flexibility and impact resistance^{(6) - (9)}.

Since PPS and polyester-based plastic materials provide a strong barrier as well as the heat resistance and flexibility required for the plastic tube application, the performance of these two materials in multi-layer tubes was further evaluated. Liquid crystal polymers have excellent barrier performance but flexibility and cost become issues; therefore, they were not studied for use in multi-layer tubes⁽¹⁰⁾.

PPS has benzene rings in the main chain, and has an inflexible molecular structure, resulting in poor impact resistance and flexibility as compared with conventional ethylene-tetrafluoroethylene (hereafter referred to as “ETFE”). Therefore, a modified PPS alloyed with an elastomer for enhanced flexibility was used as the barrier material. Figure 9 shows its Transmission Electron Microscopy (TEM) photograph. It can be observed that

elastomer regions of about 1 μm are dispersed uniformly as islands in the sea of PPS. Due to the addition of this elastomer, the Izod impact strength at -40°C was enhanced from 25 J/m to 202 J/m.

4.2. Layer Configuration of Multi-layer Plastic Tube

Table 3 shows the layer configuration of a multi-layer plastic tube in which PPS and polyester-based plastic were used as barrier materials.

The three-layer PPS tube was configured as follows: a modified PPS layer with high fuel resistance and barrier capacity as the innermost layer in direct contact with fuel, polyamide 11 (hereafter referred to as “PA11”) as the outermost layer (similar to the conventional ETFE multi-layer tube), and an adhesive layer between the two. The five-layer polyester-based plastic multi-layer tube was configured as follows: PA11 (which is used in a vapor-based single layer tube) as the innermost and outermost layers,

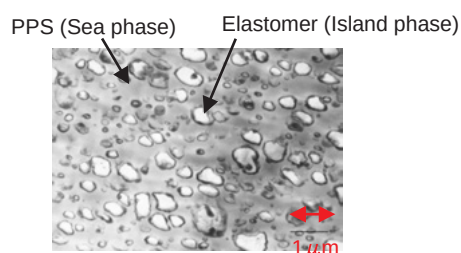


Fig. 9 TEM cross section photograph of PPS tube material

Table 4 Durability test results of multi-layer tube

Item	Tube type	Sour fuel	FUELC/E20	FUELC/M15	Brazil fuel
Cold impact	PPS multi-layer	Good	Good	Good	Good
	Polyester multi-layer	Good	Good	Good	Good
Cold bending	PPS multi-layer	Good	Good	Good	Good
	Polyester multi-layer	No good	No good	Good	Good
Adhesive force	PPS multi-layer	Good	Good	Good	Good
	Polyester multi-layer	No good	Good	Good	Good

polyester-based plastic (forming the barrier layer) as the intermediate layer, and two adhesive layers to bond them.

4.3. Evaluation of Fuel Resistance of Tube Material

Various fuels were used and durability tests were carried out to evaluate the fuel resistance of polyester-based plastic multi-layer and PPS multi-layer tubes. Table 4 shows the fuel types evaluated and the durability test results for each kind of fuel. Tests were carried out at 80 °C, which is the maximum guaranteed temperature on the vehicle body side.

The polyester-based multi-layer tubes developed cracks during low temperature bending tests with sour gasoline. Peeling occurred during adhesive tests using the same construction with sour fuel, and cracks occurred during low temperature bending tests using FUEL C/E20 (ethanol 20%). Figure 10 shows the tube cross section after the tests. Cracks were observed at the interface of the polyester-based plastic and adhesive layer. It was concluded that since polyester resin has ester bonds in the main chains, hydrolysis occurred during fuel immersion, and the material decomposed and deteriorated. From these results, multi-layer tubes with polyester-based plastic as the barrier material were concluded to be unsuitable for use as fuel tubes.

PPS multi-layer tubes satisfied the target performance with respect to all the evaluated parameters for various kinds of fuel. Figure 11 shows the condition of a PPS multi-layer tube cross

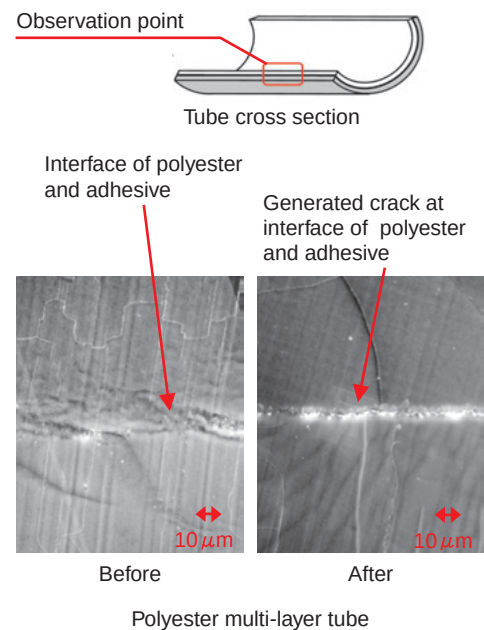


Fig. 10 SEM photograph of polyester multi-layered tube cross section before and after durability test

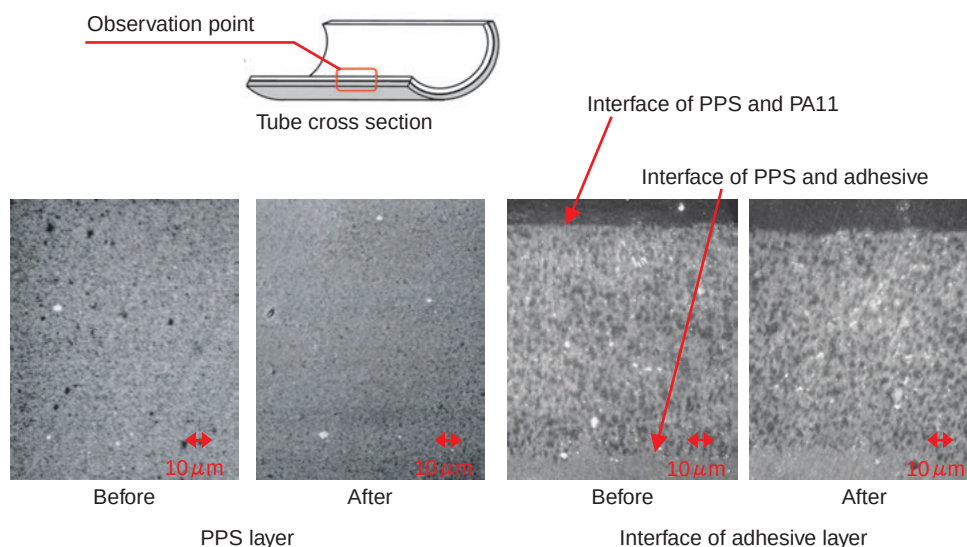


Fig. 11 SEM photograph of PPS multi-layered tube cross section before and after durability test

section. Observation of cross sections after durability tests showed no cracks or deterioration at the interface between the modified PPS resin surface and adhesive layer, confirming adequate fuel resistance and durability. In view of the above, a multi-layer tube with modified PPS as the barrier layer in the low-permeation plastic tube was used.

4.4. Permeation from Tube

Figure 12 shows the results of fuel permeation reduction with the PPS multi-layer tube developed at this time.

The permeation from the plastic tube was reduced to one-tenth that of conventional tubes, and the target of less than 0.6 mg/m/day was achieved.

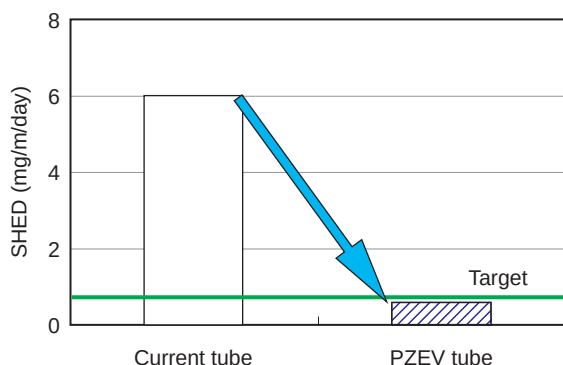


Fig. 12 Tube evaporative emissions

5. Compliance with PZEV Standards

Compliance with PZEV standards has been verified based on the rate of permeation of the fuel system. Test results using the SHED method with Phase III as the test fluid are shown in Fig. 13.

The rate of system permeation was below 54 mg/day as required by PZEV standards.

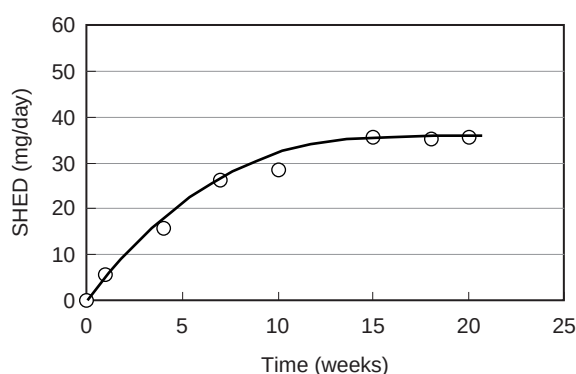


Fig. 13 Fuel system evaporative emissions

6. Conclusions

A plastic fuel tank system complying with PZEV standards was developed and fitted on the new CIVIC-IMA.

- (1) By using the newly developed plastic fuel tank barrier material, a permeation rate of less than 2 mg/day from the tank shell was achieved.

- (2) A modified PPS was used in the plastic tube barrier material and a permeation rate of less than 0.6 mg/m/day was achieved.
- (3) A permeation rate of less than 54 mg/day from the fuel system was achieved in SHED tests using PHASE III fuel.

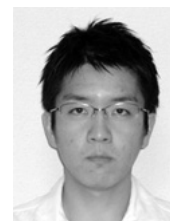
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Development of Super-brilliant Silver Metallic Coating

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ABSTRACT

A technique has been developed to achieve increased brilliance in a silver metallic automotive coating. Super-brilliant coatings have been employed in the past, but the necessity for special pigments and complex paint shop processes has made it difficult to use them in mass production. The new technique exploits characteristics of the paint to achieve optimum orientation of the brilliance pigment. This results in increased brilliance when using conventional pigments and paint shop processes. The new technology produces two times the brilliance of conventional silver metallic paints, and the pigment particles cannot be individually seen, resulting in a superior finish.

1. Introduction

Figure 1 shows recent trends in colors of automotive coatings, highlighting the high level of interest in the marketplace for silver metallic coatings and the steady sales percentage for vehicles of this color. In recent years sales percentages of silver metallic vehicles have increased significantly, indicating an increase in the market demand. Figure 2 shows design changes in the silver

metallic color. From this diagram, it can be seen that design has evolved to emphasize the brilliance and the contrast between brilliant surface areas and shadowed areas. Colors are being developed that approach the appearance of true metals. It is thought that this design evolution has contributed to the strong support that this color has received in the market.

In recent years, an evolved silver metallic color has been used on display models at motor shows and in other demonstrations,

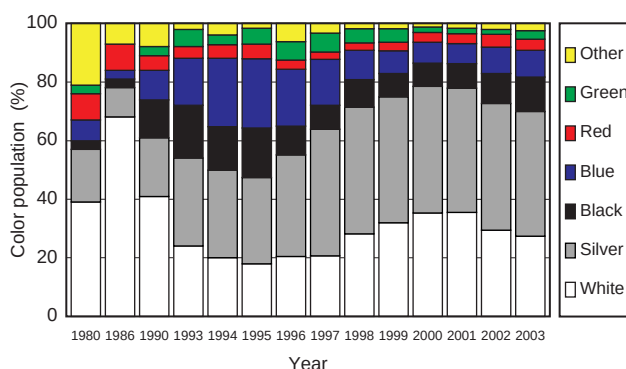


Fig. 1 Domestic (Japan) color trend for automotive bodies

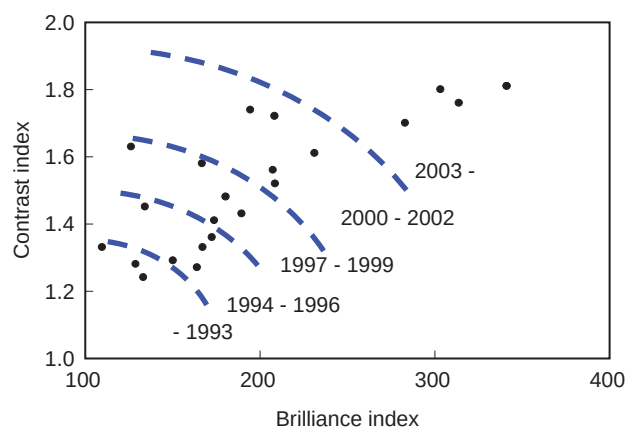


Fig. 2 Evolution of silver metallic color at Honda

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and on a limited number of production vehicles. In addition to the brilliance and high contrast of the coatings, the pigment seams (particle appearance) no longer stand out. However, there are two hurdles with respect to the mass production of brilliant silver metallic coatings with a reduced particle appearance—such coatings require complex coating processes⁽¹⁾ and special pigments^{(2), (3)}.

This report explains a coating system that has enabled the production of a highly brilliant silver metallic color with reduced particle appearance on a mass production line.

2. Current Status and Issues Concerning Production Using Silver Metallic Colors

The coating for automotive bodies is generally applied in four layers, as shown in Fig. 3. The brilliance of the silver metallic color comes from the reflection of light by the aluminum pigment included in the base coat layer. To demonstrate the orientation of the aluminum pigment coating, surface pictures of a standard coating and a coating where aluminum pigment has been oriented are shown in Fig. 4. In this picture, pigment oriented horizontally appears white, while pigment that is not oriented appears grey. It can be seen that there is a random alignment of pigment in the standard coating. The pigment in the other coating is aligned uniformly, has achieved higher brilliance and reduced particle appearance. It can be concluded the random pigment orientation in current coating systems prevents the aluminum pigment from achieving higher brilliance characteristics.

The volume reduction effect in coatings is widely known to achieve uniform orientation of aluminum pigment. After coating, the solvent, which is water, vaporizes, reducing volume. The orientation of scale-shaped pigments is restricted and becomes

uniform (Fig. 5). Increasing the solvent content of the base coat (reducing paint viscosity) results in a condition in which it is easier for the paint to move. Also, the pigment in the paint is in a state where it can move freely, making it hard to obtain a uniform pigment orientation.

Because this tradeoff relationship between volume reduction and paint viscosity exists, generating a uniform pigment orientation has been a challenge.

This report describes a technology that achieves both a volume reduction effect and paint viscosity maintenance through penetration of the solvent to the lower layer.

3. Development of Aluminum Pigment High Orientation Technology

3.1. System Settings

To reduce paint volume and maintain paint viscosity, the base coat, which has conventionally been one layer, has been changed

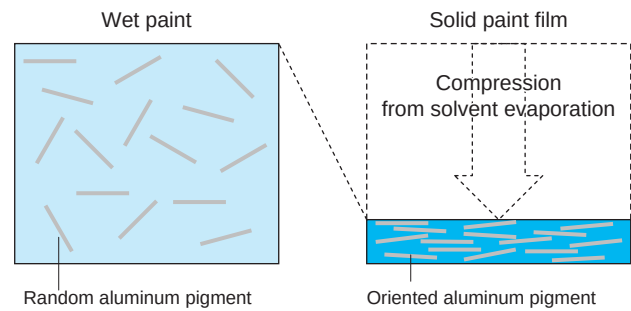


Fig. 5 Effect of shrinkage of paint layer

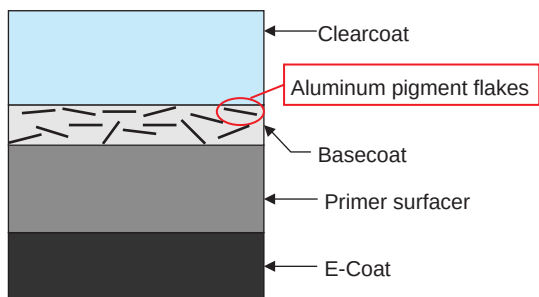


Fig. 3 Standard layer structure of body paint

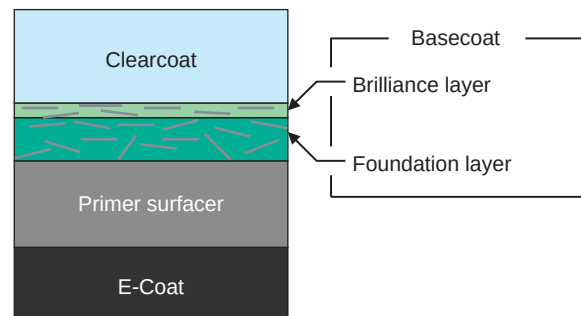


Fig. 6 Layer structure of new basecoat system

	Present silver metallic paint	Pigment with faced orientation
Picture at face		

Fig. 4 Arrangement of aluminum pigment flakes in basecoat

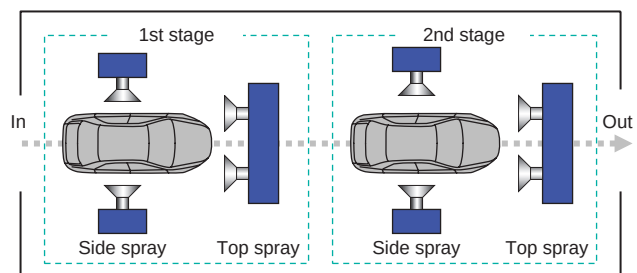


Fig. 7 Paint facility layout in basecoat booth

to two layers with different functions (Fig. 6). In the conventional process, the base coat coating is divided into two stages, as shown in Fig. 7, with the same coating being applied in each stage to ensure coating thickness. Using 2 distinct base coat materials does not decrease productivity. In this system, the base coats applied in the first and second stages are called the foundation base coat and the brilliance base coat, respectively.

The function of the brilliance base coat is to express design. To significantly reduce the paint volume previously described in the aluminum pigment orientation theory, reduction of the solid content in the paint was investigated. To enable the foundation base coat to suppress movement of the aluminum pigment in the brilliance base coat, a method of controlling viscosity was also investigated. A foundation base coat with a high solids content and a brilliance base coat with a low solids content are combined to create a differential of solid content between the two coats. This enables transfer of the water content from the brilliance base coat to the foundation base coat, increasing the solid content of the brilliance base coat. As the viscosity in the brilliance base increases, movement of the aluminum pigment is suppressed.

When considering the theory of this system, the primary paint setting parameters are the foundation base coat solid content and brilliance base coat viscosity. However, as a result of other paint parameters investigated in preliminary testing, it was found that the aluminum pigment content of the brilliance base coat and the difference in solubility parameters between the foundation base coat and the brilliance base coat needed to be further defined. The high orientation of pigment could not be achieved through independent application of the brilliance base coat. The four paint parameters were evaluated using the complete layered system. The results are described below.

3.2. Foundation Base Coat Solid Content

First, the phase transfer phenomenon that accompanies changes of paint solid content in water soluble paint is shown in Fig. 8. While water in the water-rich condition (hydrophilic phase) of the paint is vaporized and the paint solid content is increased before baking, a phase transfer point exists at which the paint condition changes to a solvent-rich condition (hydrophobic phase). Water absorption is considered to be high at this phase transfer point.

To determine the optimal foundation base coat solid content, the amount of water absorbed was measured. As shown in Fig. 9, a capillary tube filled with water was placed in contact with a wet foundation base coating of uniform thickness and the amount of water absorbed was determined. Figure 10 shows the measurement

results for the amount of water absorbed. Contrary to expected results, water was absorbed most efficiently at a solid content of 60%, which is lower than the phase transfer point. It is theorized that at the phase transfer point, the viscosity has increased so much that diffusion of water is inhibited. In addition, the brilliance index after applying the brilliance base, measured at the same time, showed the highest value at the point of maximum water absorption. It was therefore possible to confirm that water transfer, one of the design enhancement theories, was functioning effectively.

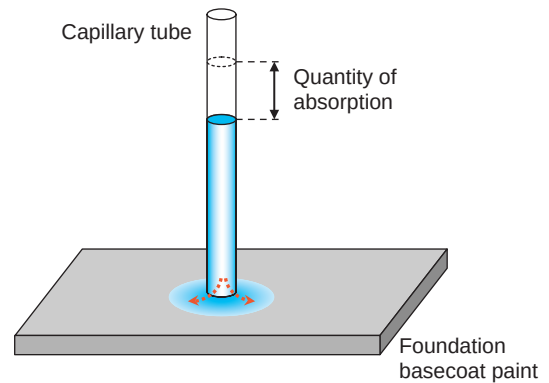


Fig. 9 Measurement of water absorption of foundation basecoat

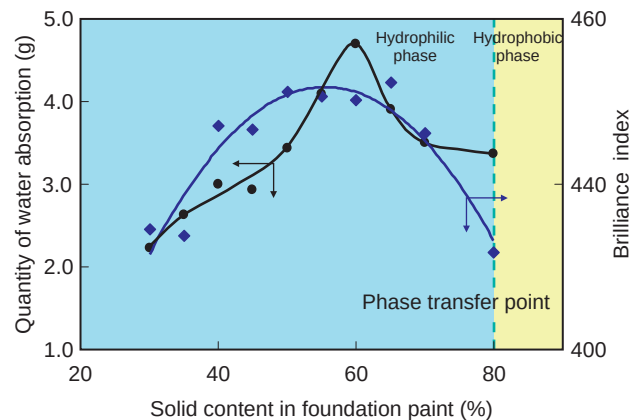


Fig.10 Solid content in foundation paint vs. water absorption and brilliance index

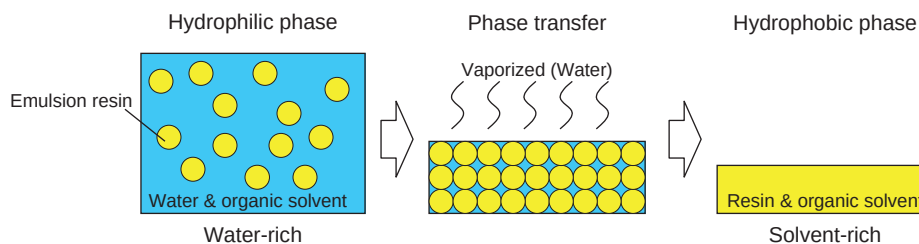


Fig. 8 Phase transfer in water-borne paint

3.3. Brilliance Base Coat Viscosity

Figure 11 shows the measurement results of the viscosity region where movement of the aluminum pigment is suppressed in the brilliance base coating only. It was confirmed that the paint viscosity needed to suppress movement of aluminum pigment was 10 Pa·s or more. The paint solid content was 30% or more. As the optimal paint solid content for the foundation base coat is 60%, to achieve a solid content for the brilliance base coat of 30% or more after absorption of water, it is calculated that the brilliance base coat needs 10% or more solid content. To verify this effect, a viscosity measurement of the brilliance base after application was attempted. Because the whole base coat system is a very thin film of 15 μm thickness, and two types of paint are overlaid, it was hard to measure only the brilliance base coat. Therefore, the brilliance index versus the solid content of the brilliance base coat was measured, and

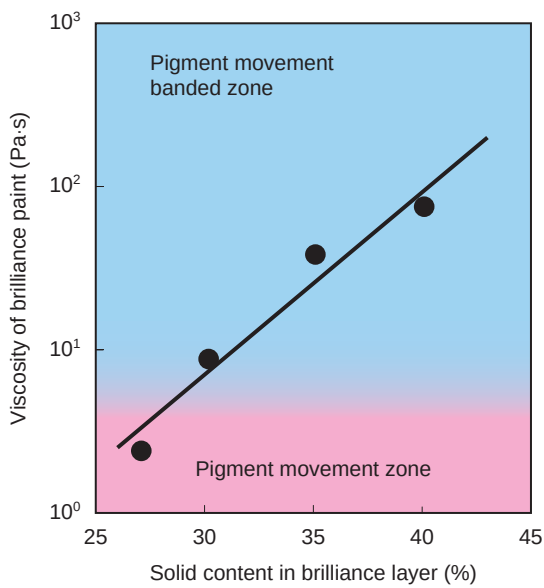


Fig. 11 Solid content in brilliance layer vs. viscosity

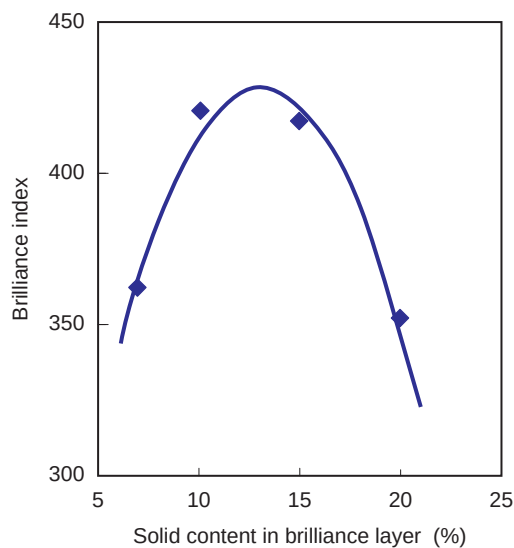


Fig. 12 Solid content in brilliance layer vs. brilliance

Fig. 12 shows the results. It was found that, as expected, a 10% solid content was favorable for orientation of the aluminum pigment. It is thought that the brilliance index decreases because the volume reduction effect is reduced for a brilliance base coat solid content of 15% or higher. The increase in viscosity is insufficient for a brilliance base coat solid content of 10% or less.

3.4. Aluminum Pigment Concentration in the Brilliance Base Coat

To analyze why aluminum pigment concentration has an effect on design, observation pictures of the aluminum pigment orientation for the coating surface were taken (Fig. 13). In these pictures, it can be observed that when the pigment concentration is at a low 10%, the pigment surface coverage is insufficient, leaving gaps. It is thought this causes the foundation base coat to show through, diminishing the design. In the high range of pigment concentration (40%), the surface is covered with pigment. Due to the interference of aluminum pigment particles, the orientation of the pigment is random. Therefore, it is necessary to exclude gaps in the aluminum pigment and randomness of the orientation caused by interference, and it was found that pigment concentration is a factor.

Following confirmation of the optimal pigment concentration in the brilliance base coat, brilliance was measured; however, the change was not significant enough. Figure 14 shows measurement results of contrast values instead. Best results were achieved using a pigment concentration between 15 – 30%. When the aluminum pigment condition is confirmed using an enlarged picture of the surface, existence of gaps are found at a pigment concentration of 20%. However, there is no interference between pigment particles and the condition is favorable.

Pigment content	Picture at face
10%	
20%	
40%	

Fig. 13 Effect of aluminum pigment content

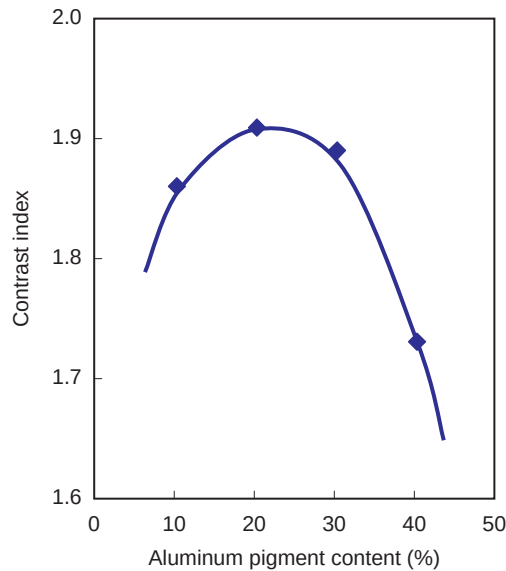


Fig. 14 Effect of aluminum pigment content vs. contrast

3.5. The Difference of Solubility Parameters between the Foundation Base Coat and the Brilliance Base Coat

The solubility parameter is an index that shows the ease of mixing the organic solvents into the coating resins. If these parameters are close, the two materials mix easily. If the values differ significantly, it is not easy for them to mix. When overlaying a foundation base coat with a brilliance base coat that has the same solubility parameter, as shown in Fig. 15, mixture of the layers occurs. The coating system begins to flow thus disturbing the orientation of the aluminum flake particles.

The difference between the foundation base coat and brilliance base coat solubility parameter (ΔSP) was changed and the appearance was confirmed. Figure 16 shows the results. The effect of ΔSP was confirmed as the brilliance index increased with greater ΔSP values. On the other hand, the compatibility between the layers is reduced as ΔSP is increased resulting in an adhesion reduction between the foundation base coat and the brilliance base coat. The result of measuring the adherence between the coatings by cutting the coating indicated that adherence was reduced for areas where ΔSP was greater than 1.0.

To achieve both appearance and adherence of the coating, the ΔSP between the foundation base coat and the brilliance base coat was set to a range of 0.9 – 1.0 by adjusting the solubility parameter for the foundation base.

3.6. Design of the Completed Coating

A silver metallic coating was created by applying the results of this research. Table 1 shows the results confirmed for the design effects. When compared to silver metallic color (satin silver metallic) currently being applied, the frontal surface achieved a brilliance value more than double that of conventional coatings. The contrast value was also increased significantly.

4. Conclusion

A new coating system with enhanced frontal brilliance and contrast compared to conventional coatings, and with reduced pigment particle appearance has been developed by dividing the

Table 1 Comparison of color characteristic values

Item	Color characteristic value	
	Brilliance index	Contrast index
New system silver met.	520	1.87
Satin silver met.	196	1.49

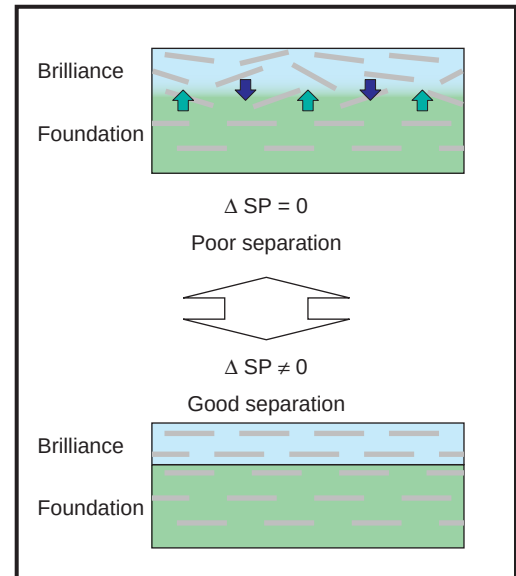


Fig.15 Influence of difference in solubility parameter

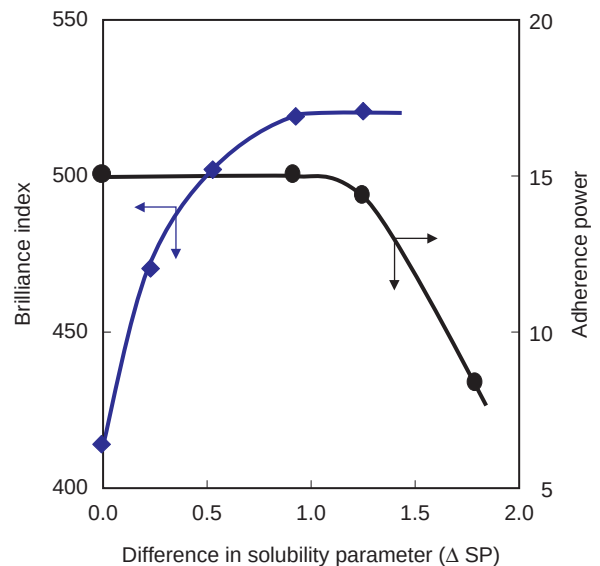


Fig. 16 Influence of difference in solubility parameter vs. brilliance and adherence power

base coat into two layers and adjusting the characteristics for the foundation and brilliance base coats. This new coating system is applied to the new model CIVIC (Fig. 17). The new coating does not require special coating resins, aluminum pigment, and does not slow down the production process. It can be smoothly introduced into a current production line.

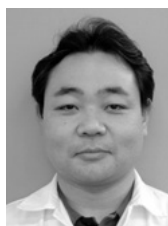


Fig. 17 This System was applied to the New Honda CIVIC (domestic version)

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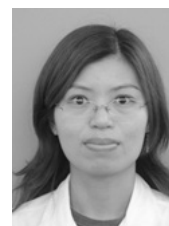
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Development of Low VOC Sealant for Door Hole Seal

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ABSTRACT

A sealer has been developed to reduce the amount of volatile organic compounds (VOC) in automotive cabins. This sealer can be discharged at 60 °C and enhances film retention. During development workability and work safety were taken into account.

Partially cross-linking styrene butadiene rubber (SBR1) has been used to replace butyl rubber as the main component of the sealer. This replacement ensures dischargeability of the sealer, eliminating the need for the solvent additive formerly used to do so. These changes have increased the Thixo index of the material when heated, enabling both dischargeability at 60 °C and film retention to be achieved.

Non-crosslinking styrene butadiene rubber (SBR2) has also been used as a component of the sealer to increase heat resistance.

This new sealer contributes to meeting the standards for the “Voluntary Approach to Vehicle Cabin VOC Reduction.” These standards are slated to be applied to new model passenger cars marketed from April 2007.

1. Introduction

Currently, the most common materials used to seal car door holes are tape, a high-temperature discharge sealer (90 °C to 150 °C)⁽¹⁾, and a normal-temperature discharge sealer with enhanced workability. Recently, however, there have been demands for decreased use of volatile organic compounds (VOC) in vehicle cabins.

For the 2006 model of the Civic (Fig. 1), Honda is using a door panel that opens wide and has excellent manufacturability and maintenance characteristics.

However, because the panel opening is wide, it was necessary to find a way to reduce cabin noise, such as road noise, caused by this large opening. To address the noise issue, a new hole seal was developed that includes a sound insulating sheet. As a result, a new sealer with enhanced retention was required that would correspond to the weight increase of the film.

This report describes the development of a low-VOC sealer, able to be discharged at 60 °C, that maintains the current degree of

workability while providing enhanced film retention at high temperatures.



Fig. 1 Door panel

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2. Development Goals

Sealer used to seal door holes⁽³⁾ prevents rain from entering the vehicle cabin by forming a seal between the door's inner panel and the hole seal. The inside of the door can reach temperatures of almost 70 °C in summer; therefore, in addition to maintaining sealing ability at high temperatures, it must also maintain film retention.

Furthermore, during maintenance of components inside the door panel, the hole seal must be able to be removed; therefore, a material that can be reinstalled after completing this maintenance needed to be used.

In conjunction with development of this new sealer, the following 5 items were defined as development goals.

- (1) No solvents added to the material
- (2) Good retention performance of the hole seal film at high temperatures
- (3) A discharge temperature of 60 °C or less for worker safety
- (4) Can be reinstalled for maintenance
- (5) Favorable workability at the assembly plant

3. Concept for the Basic Specifications of the Sealer Material

Figure 2 shows a design concept for a sealer material that would enable film retention at high temperatures, good discharge performance, and reinstallation. The figure also includes the goal range for each of these characteristics. Characteristics for normal-temperature discharge sealers (Current 1) and high-temperature discharge sealers (Current 2), that are currently in widespread use, are shown for reference.

This figure shows that the materials that are currently used fall within some of the goal ranges, but do not meet all of the goal specifications. This meant that the development concepts could be narrowed down to achieving good dischargeability and retention.

To ensure good retention at high temperatures, the static viscosity of the material (viscosity when shear speed is low) must be high. Additionally, to ensure good discharge performance, the dynamic viscosity (viscosity when shear speed is high) must be low⁽⁴⁾. In other words, it was believed that a high thixotropy index would be

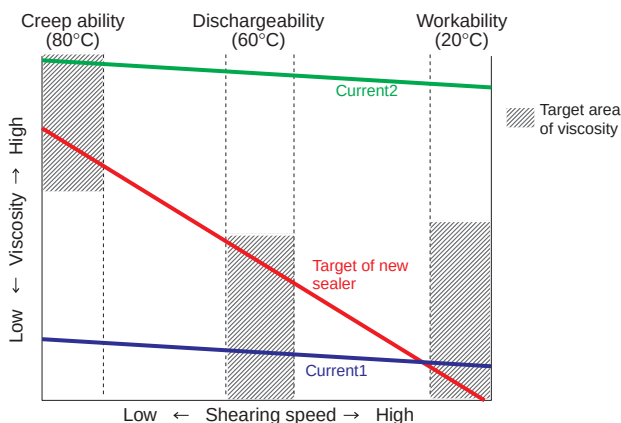


Fig. 2 Concept of viscosity design

needed, in addition to a viscosity that would enable sufficient discharge.

To develop a sealer material easily affected by shear speed, the investigation began by looking at rubber, which is the main component of sealers. In choosing a rubber compound, emphasis was placed on versatility, leading to the selection of butyl rubber (IIR), styrene-butadiene rubber (SBR), and acrylonitrile-butadiene rubber (NBR) for investigation. IIR, which is currently in use, has a relatively linear molecular structure, and the change in viscosity relative to shear speed is low. SBR and NBR, on the other hand, have functional groups at its' terminals and therefore have relatively high polarity⁽⁵⁾. The functional groups at the terminals of these rubber compounds generate an interaction between molecules. Therefore, at low shear speeds high viscosity is achieved and interaction is maintained; while at high shear speeds, interaction is interrupted and viscosity decreases.

Note that to ensure low levels of VOC, essentially none were to be added. Therefore development proceeded without the use of a solvent.

4. Evaluation Method

To evaluate the actual performance of the sealer, viscosity was used to judge dischargeability, a repeated peel test of the film was used to test reinstallation capability, and a high-temperature creep test method, shown in Fig. 3, was used to confirm the high-temperature retention performance.

For the high-temperature creep evaluation, a 180° peel creep resistance test and a shear creep resistance test were conducted. During the shear creep resistance test the load is dispersed over the whole surface. On the other hand, during the 180° creep resistance test there is a higher tendency for stress to be concentrated at the adhered surface, thus providing more severe test conditions. This report only presents the results of the 180° peel creep resistance test.

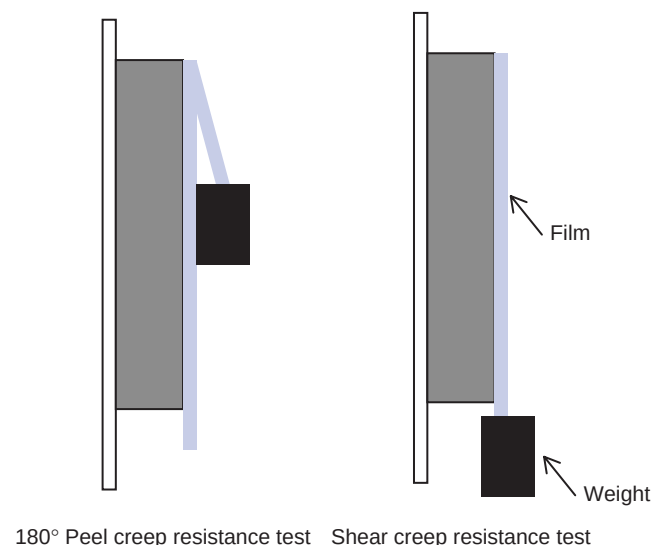


Fig. 3 Creep resistance test method

Table 1 Selection of main rubber

Rubber	180° Peel creep resistance 0.5g, 80°C24h	Viscosity (60°C)	Reinstallation (after heat resistance)
NBR1	5.0mm	60Pa·s	Water leakage Adhesive failure
NBR2	10mm	65Pa·s	Water leakage Cohesive failure
SBR1	1.0mm	59Pa·s	Water leakage Adhesive failure
SBR2	4.0mm	62Pa·s	Water leakage Cohesive failure
IIR	20mm	67Pa·s	Water leakage Cohesive failure

Table 2 Selection of secondary rubber

Rubber	Effect	Result			
		Reinstallation (after heat resistance)	Storage stability (Coefficient of rate of change of viscosity)	Creep ability	Workability
Blank	—	Water leakage Adhesive failure	+30%	1.0mm	Adequate
IIR	Heat resistant	Water leakage Cohesive failure	+5%	2.5mm	Inadequate
NBR2	Heat resistant	Water leakage Cohesive failure	+4%	4.0mm	Adequate
SBR2	Heat resistant	No Water leakage	+5%	1.0mm	Adequate

5. Consideration and Results for the Basic Specifications of the Sealer Material

5.1. Investigation of Achievement of Both Dischargeability and Film Retention at High Temperatures

Table 1 shows results from an investigation that used the three development goals, described previously, to select a main rubber compound.

For NBR and SBR, an investigation was performed of partially heated and cross-linking type (NBR1, SBR1) and non cross-linking type (NBR2, SBR2). For IIR, only non cross-linking types were investigated.

The results showed that SBR1 performed the best. The reason for this is thought to be as follows; to achieve the development goals shown above, a sealer with a higher thixo index than conventional sealers is needed. The SBR1 has localized and dispersed cross-linkings, therefore it is more affected by shear speed, thus having a higher thixo index.

A heat resistance test showed that when only SBR1 was used, cross-linking progressed during the heating process, and reinstallation was no longer feasible; therefore, it was decided to enhance the heat stability of the material.

5.2. Enhancement of Heat Stability

The effect of combining another rubber with the SBR1 main rubber was investigated as a means of enhancing the heat stability of the material.

Table 2 shows the results. Parameters for heat stability, reinstallation characteristics, storage stability and workability were measured. In addition, the initial purpose of film retention at high temperatures was also confirmed.

When combined with non-cross-linking IIR, NBR2, or SBR2, good storage stability results were obtained. This is possibly because combining SBR1 with non-cross-linking rubber compounds reduced the percentage of rubber that was cross-linked overall. Any reduction in the percentage of rubber that was cross-linked using this rubber combination, compared to when the partially cross-linking SBR1 was used alone, could be due to a suppression of the progress of the cross-link reaction caused by heat.

Next, concerning high-temperature creep resistance and water tightness, combinations using either IIR or NBR2 did not have sufficient cohesive strength and therefore did not satisfy water tightness requirements. In addition, compared to SBR2, the amount of creep increased. As a result, it was decided to use a combination of SBR1 and SBR2.

5.3. Investigation of Compositions

The proportional ratios for high-temperature creep resistance and storage stability between SBR1 and SBR2 were investigated. Figure 4 shows the relationship of high-temperature creep resistance and storage stability with respect to the proportion of SBR1, as well as goal ranges for high-temperature creep resistance and storage stability (within the shaded area).

The amount of creep is more favorable with increased amounts of SBR1, but storage stability decreases. This is thought to be because cross-linking, attributed to heat resistance, increased the viscosity when SBR1 was used alone. As the amount of SBR2 increased, creep resistant performance was reduced due to insufficient cohesive strength. Note that the final combination amount was set within the range shown by the shaded area.

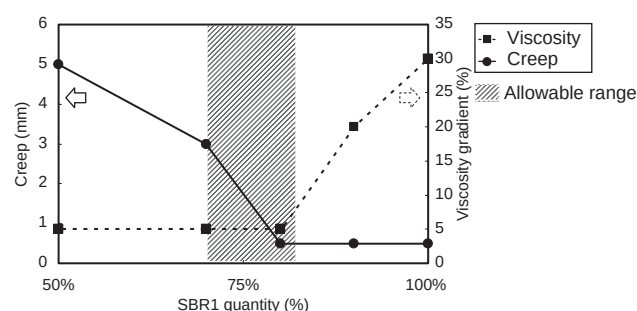


Fig. 4 Effect of SBR1 composition

6. Material Test Results

6.1. Dischargeability

Figure 5 shows viscosity characteristics of the newly developed and conventional sealer materials. The working temperature of the developed material is 60 °C and that of the conventional material is 20 °C.

The viscosity at high shear rates of the developed material is lower than that of the conventional material, and therefore it was expected that dischargeability would be favorable. On the other hand, in the low shear rate region, viscosity is high and therefore creep performance is thought to be enhanced.

As a result of confirming dischargeability using a hand gun and mass production equipment, the discharge performance was found to be favorable.

6.2. High-temperature Creep Resistance Test

As an indicator of film retention, a 180° peel high-temperature creep resistance test was performed, and Fig. 6 shows the results.

The creep resistant performance of the rubber composition

selected from the previously described investigations was found to be better than that of the conventional material. In addition, by combining an inorganic agent, the final material has an increased thixo index at high temperatures, further enhancing creep resistant performance. As a result, during heat load, with respect to the goal of 3 mm or less for the amount of creep, an amount of creep of nearly 0 mm was achieved.

6.3. Door Hole Sealer VOC Measurement Results

Figure 7 shows VOC measurement results for the newly developed and conventional materials. The measurement method was established for each part based on Japan Automobile Manufacturer's Association, Inc. methods^{(6), (7)}, and the data was

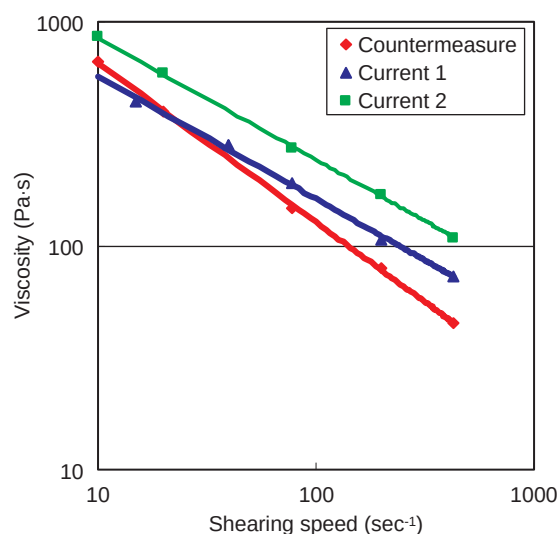


Fig. 5 Shear rate curve

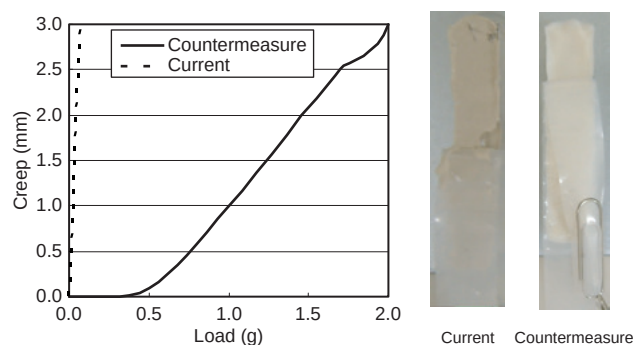


Fig. 6 Creep resistance test result

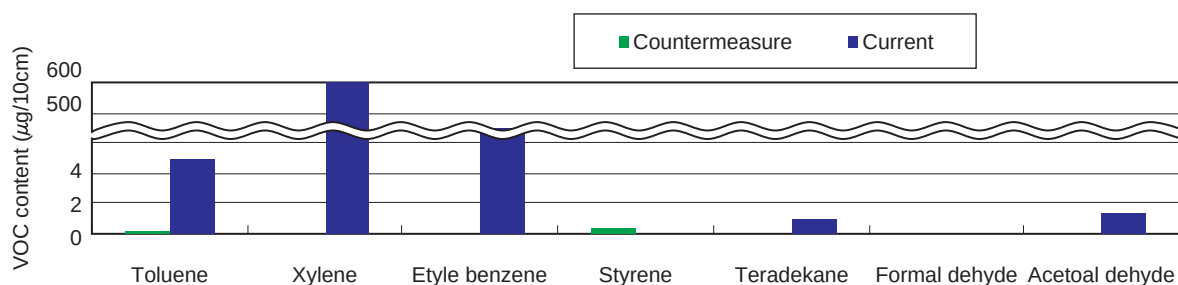


Fig. 7 VOC test result (unit)

evaluated using a Honda-specific method. After the solvent used to adjust viscosity and ensure workability was removed, the amount of xylene and ethyl benzene in the developed material was reduced to less than 1/7000 of that of the conventional material.

7. Results of Tests on an Actual Vehicle

7.1. Compatibility with Actual Vehicle

A heat cycle test was performed on the door of an actual vehicle. The test mode is shown in Fig. 8 and the test results after 5 cycles are shown in Fig. 9. In the previously described high-temperature creep resistance test, the film and sealer were in an unconstrained open system. In this test, the sealer was applied around the entire circumference of the door hole and pasted together with the film, making it a constrained system. Therefore, as the film itself stretches and contracts due to heat cycling, this is a composite test that considers displacement of the film in addition to retention at high temperature. The newly developed sealer material, exhibiting the same behavior as in the component level high-temperature creep resistance test, did not have any creep and it was not affected by the stretching and contracting of the film, leaving no issues to be addressed.

In addition, compared to initial application, there was no significant change in stringing when the film was peeled off after the durability test, and there was no water leakage after reinstallation. This led to the conclusion that there were no issues with regards to maintenance capability.

7.2. Actual Vehicle VOC results

A vehicle was left in an airtight condition for 14 days after manufacturing and the VOC was then measured. The measurement method followed the Japan Automobile Manufacturers

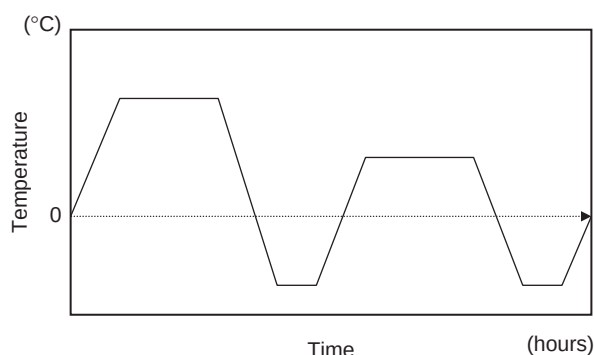


Fig. 8 Heat cycle mode



Fig. 9 Heat cycle test result (door panel)

Association's measurement method. Through the application of the newly developed sealer, concentrations of xylene and ethyl benzene met standard values.

8. Conclusion

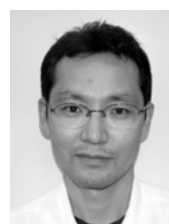
A sealer with low levels of VOC that ensures enhanced retention of the film as well as maintaining the same workability as conventional sealer materials has been developed and the following results were obtained.

- (1) A partially cross-linking SBR rubber compound was selected. Discharge at high temperatures, equivalent to that of conventional materials, was achieved through increasing dependence on shear speed. In addition, the thixo index at high temperatures was enhanced, enabling better retention of the film.
- (2) Investigation of the composition of partially cross-linking and non cross-linking materials enabled the control of cross-link reaction due to heat resistance, while creep resistant characteristics were maintained. This led to enhanced storage stability.
- (3) Due to the fact that solvents were not used in the manufacturing of this new sealant, concentrations of xylene and ethyl benzene inside the vehicle cabin met Japan Automobile Manufacturer Association standards.

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Strength of FSW Joints in Aluminum Extrusions and Simulation

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ABSTRACT

The use of aluminum extrusions joined by friction stir welding (FSW) was examined under the supposition that such a method might produce a vehicle body structure with highly efficient joints that is also lightweight and low cost. The strength of an FSW joint of aluminum was compared with MIG and laser MIG welding in the fabrication of such a structure. Static and dynamic tests were performed in order to evaluate the collision strength of the FSW joint. As a result, it was determined that welding conditions largely determine the dynamic strength performance of the FSW joint. Additionally, modeling of an FSW joint was attempted and simulation methods were devised.

1. Introduction

Due to global warming, air pollution, and rising oil prices, it continues to be important to make lighter vehicle bodies to conserve fuel. One technique for so doing is to adopt structures that join aluminum extrusions with cast or stamped parts. Structures of this kind are already being mass-produced⁽¹⁾. This article examines structures composed of extrusions fabricated with a reduced mold investment and without the use of cast or stamped parts for joints. The use of friction stir welding (FSW) allows for continuous joints, as opposed to the spot joints produced by spot welding, and this makes possible structures with greater joint strength. These structures are also less subject to thermal strain than those produced by MIG welding, making it easier to achieve precision in the vehicle body. In addition, the required investment is lower than laser welding, and the application of FSW is therefore expected to increase in the future, with an emphasis on use in low-volume production vehicles.

Several studies of FSW^{(2)–(4)} have been conducted, and it has begun to be used for some parts in railroad rolling stock and motor vehicles. But there are few reports on the dynamic properties and

strength of lap joints that take into account the application of this method to vehicle frames, including the question of productivity.

Strength tests were therefore performed on the aluminum extrusion frame fabricated with lap joints using FSW, and an evaluation of the FSW joints was conducted. MIG and laser MIG welds that are accessible from one side were selected for comparison purposes. A method for simulating FSW joints was also devised.

2. Test Specimen and Methods

2.1. Shape of Test Specimen and Contents of Tests

The areas of the vehicle body structure to be investigated were the joints of the floor frame, outrigger and side sill on the front section of the vehicle frame, which are typical body structures. These areas not only require static strength; they also demand dynamic strength during collisions. The specific structure is shown in Fig. 1. Aluminum extrusion was applied to the areas in question, and the structure of section closed by FSW joints was investigated. For the testing, test specimens that took load direction in these areas into consideration were used. Static load testing and falling

* Tochigi R&D Center

weight impact testing were carried out, and the joints were evaluated. High-speed tensile testing was also conducted to devise a method of simulating FSW joints, and the resulting dynamic properties were studied.

2.2. Test Materials and Welding Conditions

The parent material for testing was A6063S-T5 aluminum extrusion, a material with a track record in mass-produced space frames⁽¹⁾. The test specimens were fabricated by cutting them from extrusions. The test specimens and stock thickness were selected to match each test. The chemical composition and mechanical properties of the test materials are shown in Tables 1 and 2.

MIG joints and laser MIG joints were selected to provide comparison with the strength of FSW joints. The various welding conditions were decided by quality in terms of external appearance and observation of cross sections using an electron microscope to confirm that there were no flaws. The FSW was performed using a tool with a shoulder diameter of 15 mm. As welding conditions for FSW, MIG and laser MIG welding, welding feed rates of 200, 1500 and 2000 mm/min, respectively, were used. The welding conditions for the test specimens used in the below tests were all identical unless otherwise noted.

2.3. Static Test Method

The static strength test was carried out with test specimens in shapes that simulated portions of a vehicle body structure. Figures 2 and 3 show overviews of the test specimens and test device. A universal testing machine was used for the compression

Table 1 Chemical composition of parent materials A6063S-T5 (%)

Si	Fe	Cu	Mn	Mg	Cr	Zn	Ti	Al
0.40	0.15	0.00	0.03	0.47	0.00	0.00	0.02	Bal.

Table 2 Mechanical properties of parent materials A6063S-T5

Tensile strength [MPa]	Yield stress [MPa]	Elongation [%]
223	202	12

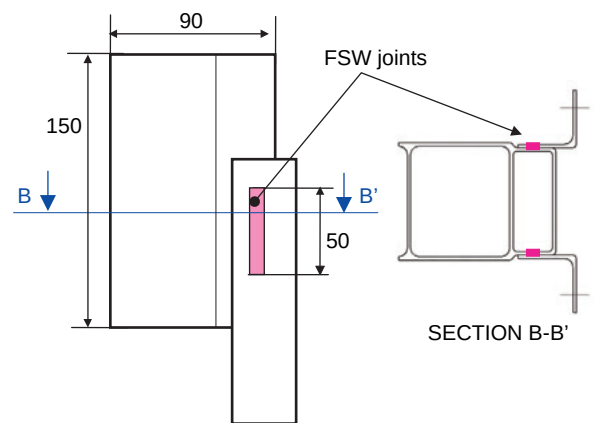


Fig. 2 Test specimen

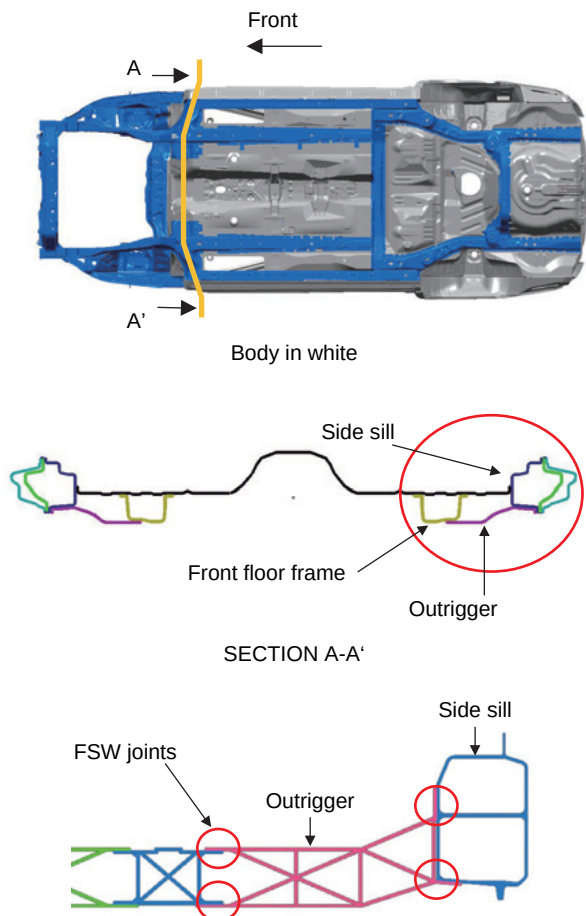


Fig. 1 Body structure with FSW joints

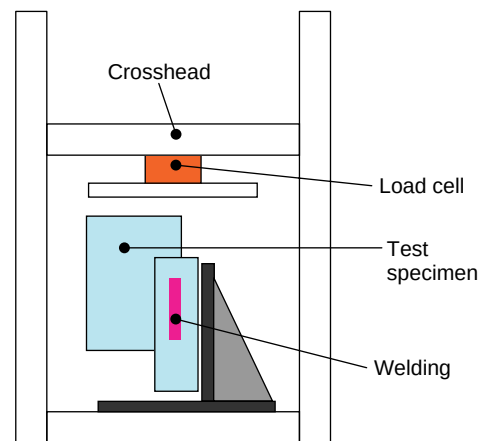
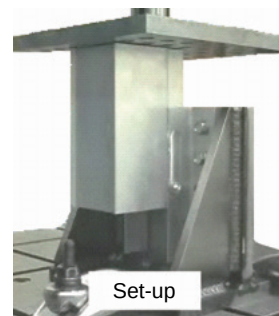


Fig. 3 Device of static load test and set-up

testing device. The load was measured using a load cell attached to the crosshead, and the crosshead displacement value was taken as the stroke. The load centers were positioned at the joint centers on the assumption that the load would be applied there during actual use, and a complex load composed primarily of a shear load was applied to the joints. The test speed was set to 10 mm/min.

The test specimens for FSW and laser MIG were joined with lap joints. In the interests of practicality, the MIG joints were joined by overlapping their flange edges and fillet welding them. All the joints were made a uniform 50 mm in length. The direction of the joints was the direction of the extrusions.

2.4. Dynamic Test Method

The dynamic strength was evaluated by means of a falling weight impact test. Test specimens in shapes simulating portions of a structure, as in the static test, were used so that the test could be compared with the static test. The direction of the load applied to the joints was also the same as in the static test. For the purpose of stability of the joints and the load applied to them, the dynamic test was set up to support the test specimen on both ends instead of just one end, and the length of the joints was extended to 170 mm. Figures 4 and 5 show an overview of the test device and the test specimens. The test was set up to allow a weight to fall freely from

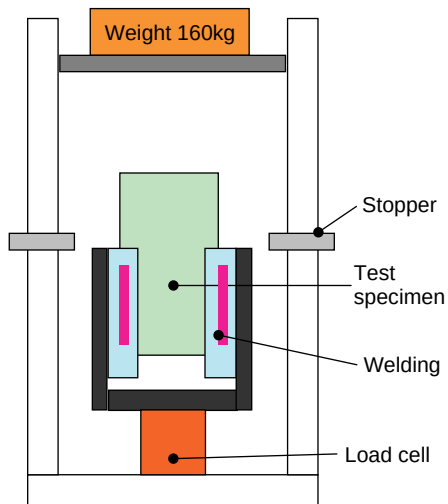


Fig. 4 Device for falling weight impact test

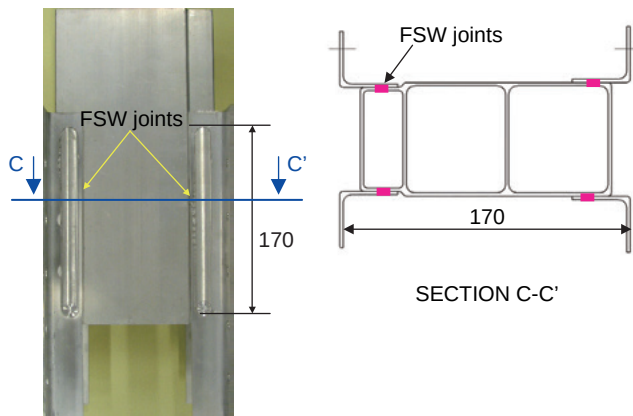


Fig. 5 Test specimen for falling weight impact test

a height of 3.5 m so that the weight would be falling at a rate of 8.3 m/sec at the moment of impact. The load was measured by means of a load cell placed beneath the test specimen, and the stroke was determined by measuring the position of the weight using a laser displacement meter.

3. Test Results

3.1. Static Test Results

Figure 6 shows the static test results. The point in the graph where the load drops abruptly indicates that the joints fractured. The FSW joints and the laser MIG joints fractured at a stroke of 7.5 mm, while the MIG joint fractured at 4.5 mm. The maximum load for the FSW joints was the same as for the MIG joints, and was low at 20% of the laser MIG joints. The length of the stroke up to the point where the FSW joints fractured was the same as for the laser MIG. The test confirmed that the strength of FSW joints is equal to or greater than that of MIG joints.

Figure 7 shows photographs of the joints after they fractured. The fracture pattern in the FSW joints shows that increasing strain in the area of the joints is accompanied by deformation of the L-shaped flanges, and the fractures occur along a straight line that originates at the FSW keyhole. That fracture cross section is shown in Fig. 8. This confirms that the fractured area is at the end point of the unjoined area. The fracture of the laser MIG joints occurred after the deformation of the flange, but the final fracture patterns were not all alike, and instead showed variation. The MIG joints all fractured at the weld bead.

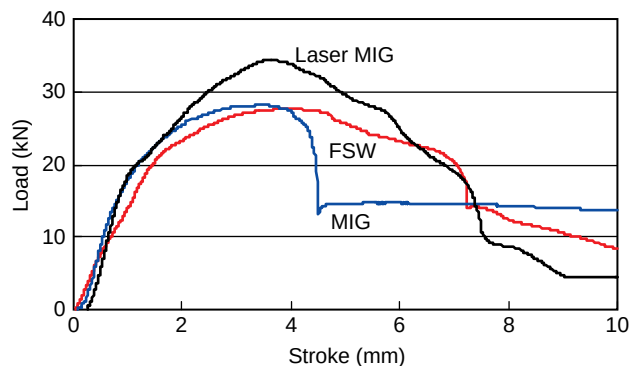


Fig. 6 Stroke – static load for welded joints

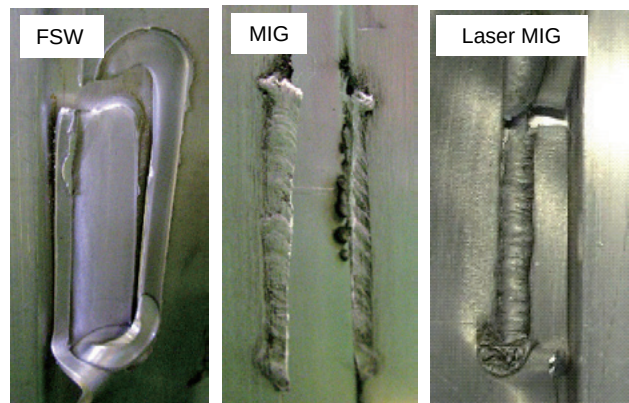


Fig. 7 Welded joints after static load test

3.2. Dynamic Test Results

Figure 9 shows the relationship between load and stroke, and Fig. 10 shows the appearance of fractures in the test specimens. The FSW joints fractured along a straight line at the end point of the unjoined area. The MIG joints fractured in a straight line along the weld bead and the heat-affected zone (HAZ). With the laser MIG joints, the upper portion of the test specimens collapsed, and no fracture occurred in the joints. The fracture patterns in the FSW joints and MIG joints were the same as found in the static test. Because the test specimens collapsed, the load on the laser MIG joints occurred at or after a stroke of 10 mm and fluctuated between 100–200 kN. The maximum load on the MIG joints was 10% lower than that on the laser MIG joints, and the fracture occurred after the stroke reached 15 mm. The maximum load on the FSW joints was the same as on the MIG joints, but the fracture in the joints occurred with fewer strokes than it did in the MIG joints. This result does not agree with the tendency found in the static test, and it indicates that the FSW joints are more easily damaged by impact.

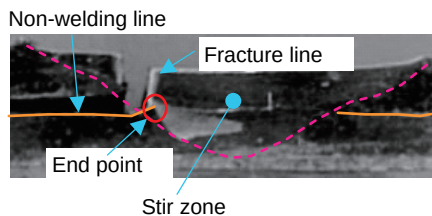


Fig. 8 Section of FSW joint after static load test

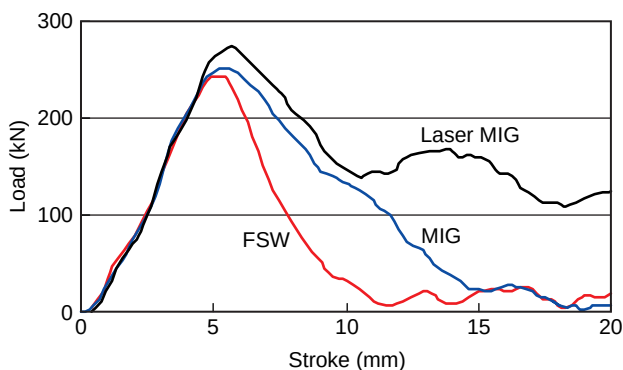


Fig. 9 Stroke-dynamic load for welded joints

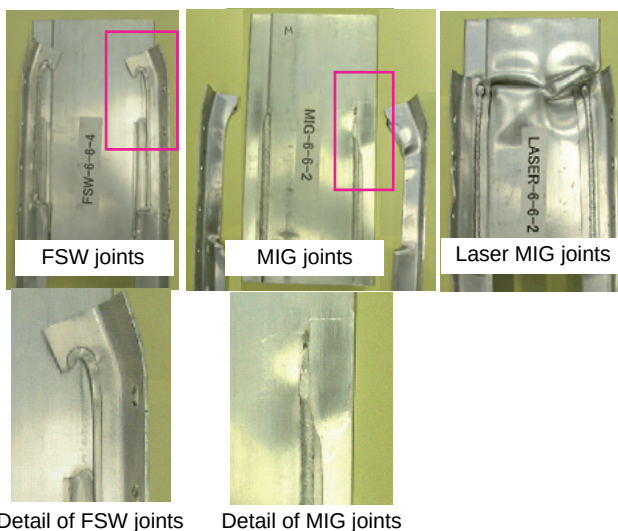


Fig. 10 Specimen after dynamic load test

4. Discussion

4.1. Discussion of Static and Dynamic Strength

The dynamic test results showed that the FSW joints have less dynamic strength than the other joints. Cross sections of the FSW joints were observed to analyze the causes of this weakness. Figure 11 shows the cross section. It is clearly evident that the material in the bottom plate is raised upward in the boundary area between the stir zone and the parent material. This is a phenomenon that occurs because of characteristics that originate in the creation of the FSW. In the case of a lap joint as seen in Fig. 12, the parent materials in the bottom must be stirred together with the parent materials in the top. The boundary area between the upper and bottom plates is raised upward, creating a kind of notch. It is thought that, as a result, the shape effect of the notch is to reduce the dynamic fracture toughness so that the amount of energy absorbed by the joint is decreased. This was proved from the report of an experimental application of the Hopkinson bar method, in which it was found that the dynamic fracture toughness of an aluminum alloy was lower than the static fracture toughness⁽⁶⁾.

4.2. Analysis of Internal Joint Shape

In order to analyze the impact on the joint strength from the raising that occurs when the FSW joint is formed, the FSW welding conditions were changed so as to create joints with less raising. The cross section of a modified FSW joint is shown in Fig. 13. It is clearly evident in this cross section that less raising has occurred in the boundary area. This state of the joint is described in the

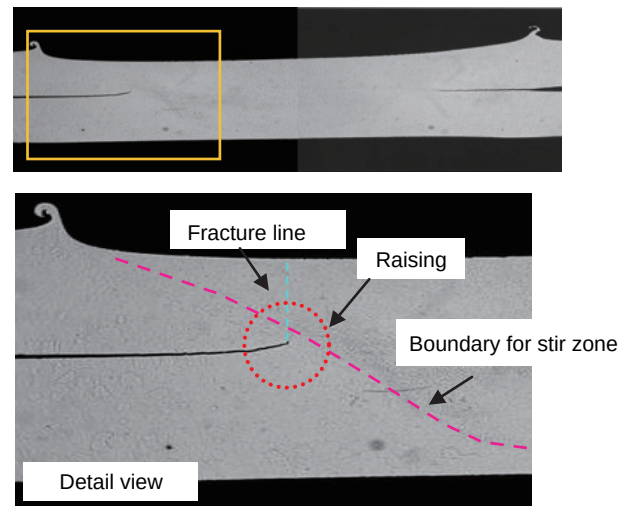


Fig. 11 Section of FSW joint

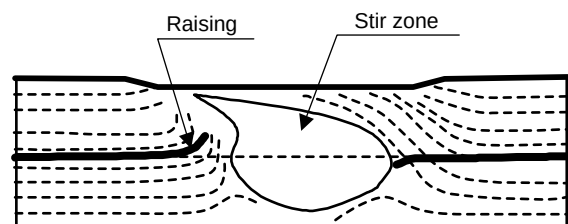


Fig. 12 Principle of FSW

below explanation as being without raising. The static strength test conducted in Section 4 was carried out and the strength with and without raising was compared. The results are shown in Figs. 14 and 15. Compared to the joints with raising, the joints without raising have increased the amount of energy absorbed. This is the result of a change in the fracture pattern. In the case of the joint with raising, one side of the stir zone fractured along a straight line, but in the joint without raising, the fracture occurred on both

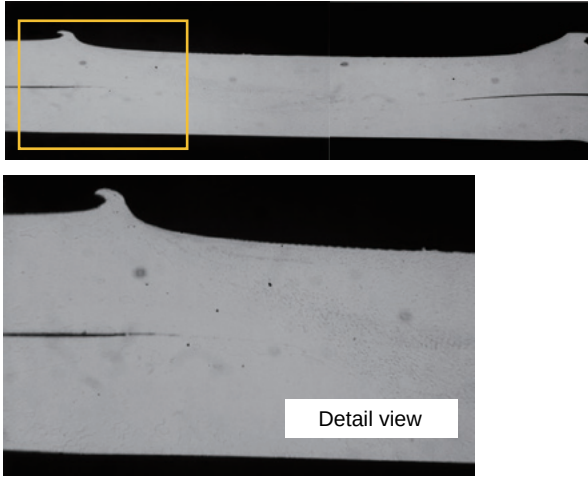


Fig. 13 Section of FSW joint under new conditions

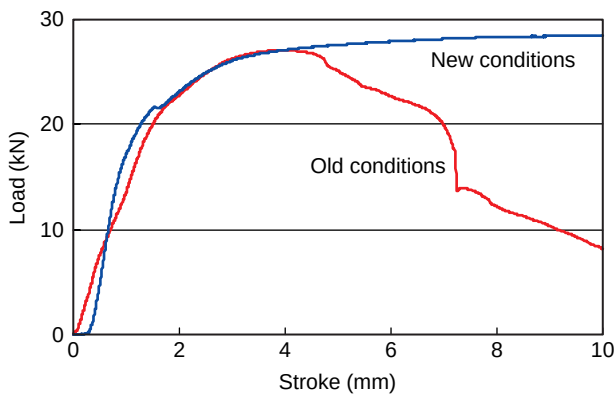


Fig. 14 Comparison of load under new and old conditions

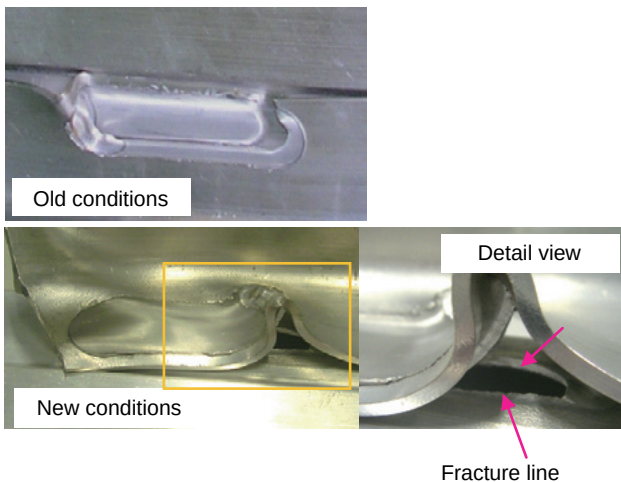


Fig. 15 Comparison of fracture mode of new and old conditions

sides of the stir zone. In this case, the crack did not propagate immediately as it did when the fracture was on just one side. The amount of energy absorbed can be said to have increased as a result. It is clear that the state of raising that occurs according to the welding conditions has the effect of changing the fracture pattern and the load properties.

5. Simulation Model

5.1. High-speed Tensile Test

A high-speed tensile test was carried out to devise a method for simulating FSW joints. The items studied were the tensile strength of the joints and the effect of strain rate on that strength. The test specimens used were 2 mm thick and 5.4 mm wide. The testing was carried out on butt welded joints. The three strain rates of 0.1, 10 and 200/s were used. The local strain in the 20 mm section centered on the joint was also measured to provide a reference for determining the size of the simulation model shell. For the local strain, parallel portions of the test specimens were marked at 1 mm intervals and their elongation was measured.

The tensile strength of the parent material with a strain rate of 200/s tended to be 6–7% higher than at 0.1/s. Figure 16 shows the test results for each joint. The FSW, MIG and laser MIG joints were all like the parent materials in that they did not display any major difference in tensile strength according to the strain rate. The tensile strength of the FSW and laser MIG joints was confirmed to have risen 15%. The fracture areas in the FSW joints were all in an HAZ.

Next the distribution of the local strain was examined. Figure 17 shows the results from measurement of strain distribution in the

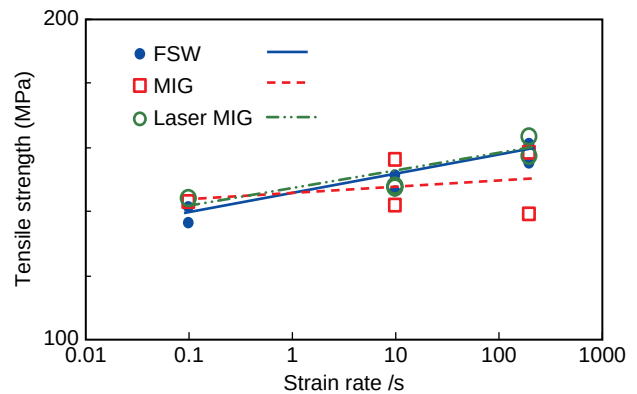


Fig. 16 Tensile strength for strain rate of welded joints

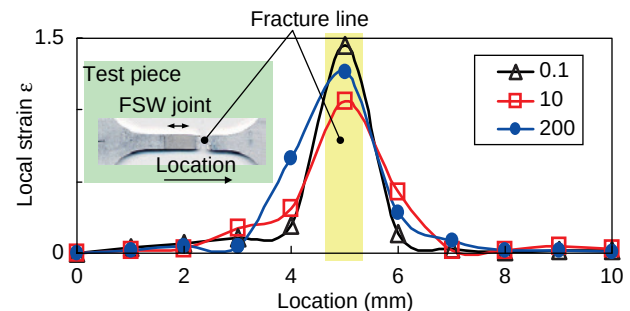


Fig. 17 Local strain for FSW joints under high-speed tensile test

FSW joints. This figure emphasizes the strain in the fractured areas. The lateral axis represents the location of the test specimens. The elongation in 10 mm sections centered on the fractured area in the test specimens increased 19% with a strain rate of 200/s relative to a strain rate of 0.1/s. It was confirmed that greater strain occurred locally within a range of 2–4 mm centered on the fractured area.

5.2. Creation of a Simulation Model for FSW

A simulation model was created from the high-speed tensile test results and FSW joint cross section. Figure 18 shows the conceptual approach for that simulation model. The shell model was created divided into the stir zone (A), the HAZ (B) and the parent material (C), and their respective material characteristics were configured. The reduction that takes place during welding was taken into account when determining the stock thickness. The size of the offset for the thickness of overlapping areas was kept as small as possible to avoid a shape effect. It has already been confirmed that the hole that develops in the terminal area when making an FSW joint does not have any impact on the maximum strength so long as the joint is of a certain length or longer. The modeling of this aspect was therefore omitted.

Figure 19 shows the maximum and minimum values calculated for the various local strain distances (the length of lateral axis sections in Fig. 17) from the local strain quantities in Fig. 17. The maximum values were calculated for strain at the center of the fractured area. The minimum values were calculated for strain at the edge of the fractured area. It can be seen that when the distance was 4 mm, the minimum strain was 0.45 and the maximum strain was 0.65. The shell size was determined from the stock thickness and from the strain distribution in Fig. 17. The pattern of strain

distribution varies according to the strain rate. The strain is concentrated within a distance of 4 mm in any case, however, and it was therefore judged that a shell size of 2–4 mm would be appropriate.

The material constants were identified using the strain rate factor obtained through testing together with the line chart of stress and strain. An isotropic elasto-plastic model was used for the material model. In modeling the aluminum extrusion, the angular edge areas were represented by shells, and the thickness of those areas was set in proportion to their area⁽⁵⁾. In correlating with the tests, the two figures for strain rate factor and ultimate strain were revised. As a result, the ultimate strain for the FSW joint areas of the simulation model came out about 10% lower than the test figures. This strain figure was smaller than the strain range test figures shown in Fig. 19. One conceivable explanation follows.

The strain figure was taken from testing not with lap joints but rather with butt welded joints. Butt welded joints are not affected by the decreased fracture toughness caused by the shape effect of raising as found in the lap joints. Consequently, the results of high-speed tensile tests on lap joints yield lower values for ultimate strain than tests on butt welded joints.

5.3. Results of Calculation

Figures 20 to 23 show a comparison of the static and dynamic test results with the calculation results. They present comparisons of the load-stroke diagram and the fracture patterns after testing. The comparison of testing and calculation shows that the maximum load and energy absorption results are both within 5%. There is a close correlation between the test and the model, and the fracture patterns are also in agreement.

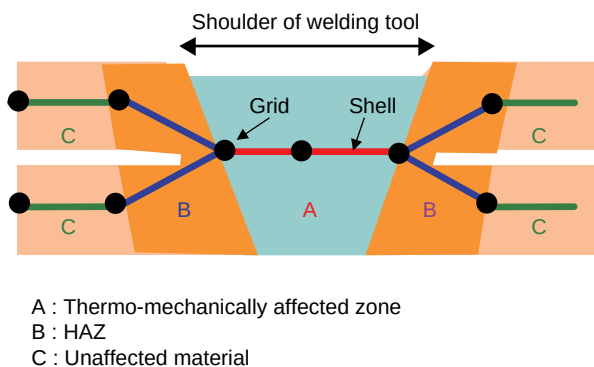


Fig. 18 Simulation model for FSW joints

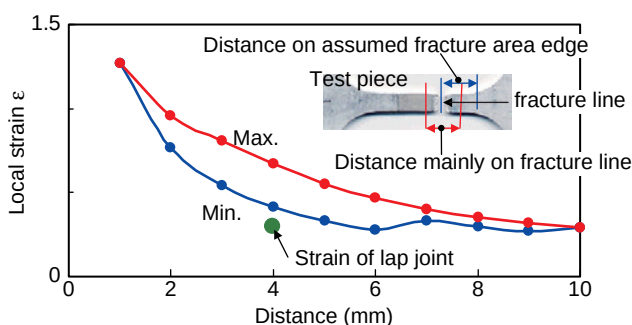


Fig. 19 Local strain distance on assumed fracture area

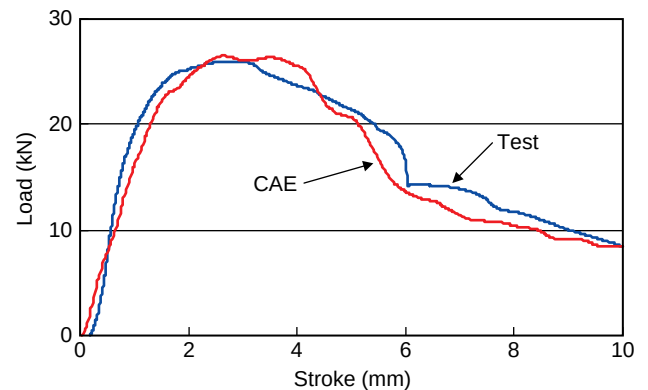


Fig. 20 Comparison of static load of test versus CAE

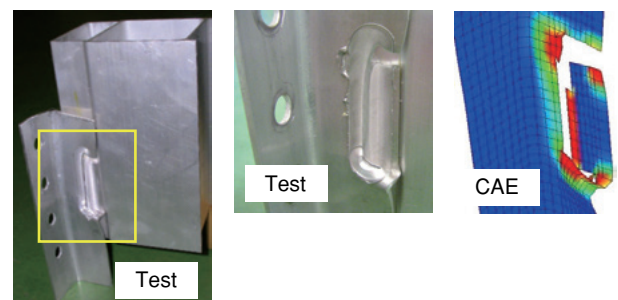


Fig. 21 Comparison of fracture mode of static load test versus CAE

5.4. Verification of Raising in the Simulation

The fracture toughness is not decreased in FSW joints where raising does not occur. It can be projected, therefore, that such joints will experience collapse in the parent material area without any fracture in the joints, just as seen in the laser MIG joints during the falling weight impact test. This was verified in simulation. The results of the analysis of internal joint shape were used for test comparison of the static strength resulting under different welding conditions. This showed that maximum load did not vary while energy absorption increased. It is clear from this result that there is an increase in the allowable strain before fracturing. When only

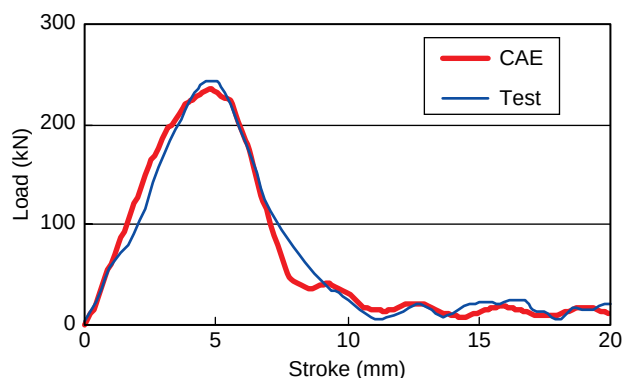


Fig. 22 Comparison of dynamic load of test versus CAE

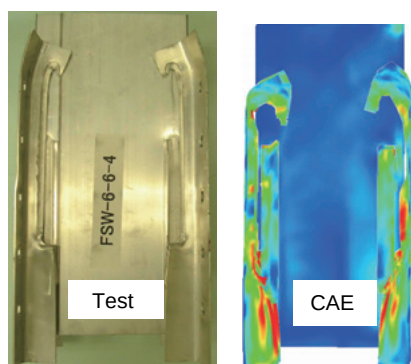


Fig. 23 Comparison of fracture mode of dynamic load test versus CAE

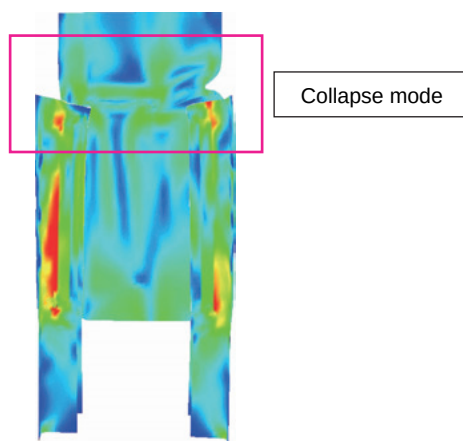


Fig. 24 Result of calculation that assumes non-raising in FSW

the figure for ultimate strain was increased in the simulation by comparison with the ultimate strain identified in the creation of a simulation model for FSW, it was projected that the joints would not fracture when simulating the falling weight test. Instead, there should be a collapse in the parent material area similar to what occurred with the laser MIG joints.

The maximum figure for average strain with a stroke of 4 mm obtained from Fig. 19 was used as the figure for ultimate strain in the simulation. According to these calculation results, the joints do not fracture but collapse does occur in the parent material area. It was evident that performance should be equivalent to that of the laser MIG joints. It can be confirmed from the calculation results in Fig. 24 that collapse occurs in the parent material area. It was confirmed in simulation of FSW joint strength that the welding conditions affect the ultimate strain and have a major effect on dynamic performance.

6. Conclusion

Evaluation was made of the static and dynamic strength of FSW joints on aluminum extrusions and research was conducted into methods for simulating such joints. The following results were obtained:

- (1) The static strength of FSW joint with raising was located between laser MIG joint and MIG joint, and the static strength of FSW joint with reduced raising was equal to that of the laser MIG.
- (2) The dynamic strength of FSW joint with raising was lower than that of the laser MIG joint and MIG joint.
- (3) The raising that occurs in the process of making an FSW reduces dynamic fracture toughness through the shape effect of the notch and has an impact on the fracture pattern and load properties.
- (4) Favorable simulations of static and dynamic behavior were achieved in the modeling of FSW joints by configuring the material characteristics for every area of the joints and by modeling them using shells.
- (5) The dynamic strength of FSW joint without raising was estimated in simulation, and a result equivalent to that of laser MIG joint was achieved.

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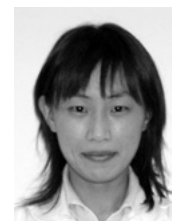
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Development of Resistance Spot Welding Simulation Technology

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ABSTRACT

A technology for simulating resistance spot welding process has been developed that is capable of predicting nugget diameter in automotive steel sheet weldments to determine the specifications of those weldments at an early stage and speed up setting welding conditions. An incrementally coupled analysis was performed for electrical, thermal and mechanical fields using a finite element code. The material properties relevant to this analysis were selected with reference to the phenomena occurring throughout the resistance spot welding process, and they can be predicted by using the chemical compositions and mechanical characteristics. Also, software was developed to aid entry of arbitrary steel sheet combination and welding condition. Verification tests of simulations indicated that nugget diameter was predicted with an accuracy of $\pm 5\%$. We expect that this simulation method will allow us to determine the specifications for automotive welded sheet combination at an earlier stage and reduce the time for setting welding conditions by approximately 45%.

1. Introduction

“Resistance spot welding process” means the welding of metal components by passing an electric current to the contact point of workpieces put between a pair of loaded electrode, and heating the contact point with a temperature higher than melting point locally by using heat generated from electric contact resistance of the interface between the components and Joule heat (resistance heat). Historically, this has been used for most of the welding in automotive manufacturing processes, where productivity and reliability are paramount. In recent years, however, it has become increasingly common to use high-strength steel sheets for body frame components to reduce weight and enhance crashworthiness⁽¹⁾. At the same time, there is increasing variety in the types of sheet

steel combinations welded by resistance spot welding. Thus, more efficient and quick identification of the optimal welding conditions for these sheet combinations is required. Conventionally, tensile tests, sectional inspections and other means have been used to set the welding conditions for such sheet combinations. However, the diversity of welded sheet combinations is increasing with the widening applicability of high-strength steel and the development time for automobiles is shortening. In the future, it will be harder to address these situations using those experimental methods only. Therefore, a technology for simulating resistance spot welding was developed that is capable of predicting nugget diameter in arbitrary automotive steel sheet weldments to determine the specifications of those weldments at an early stage and to speed up setting welding conditions.

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This report describes an outline of the simulation model and a method for defining material properties necessary for predicting nugget diameter. This also describes the development of more user-friendly software, and the benefits this package offers to automobile designers.

2. Setting Goals

Much research has been devoted to simulations of resistance spot welding processes, and there are several software packages on the market that offer such simulations^{(2) - (4)}. However, they are only capable of analyzing a limited number of steel types and sheet combinations because there are few kinds of the necessary material properties which are prepared by default. Therefore, goals for the following topics were designated so that welding conditions for general sheet combinations used in actual vehicles could be more easily set.

- (1) Simulation model: In order to reproduce actual resistance spot welding processes during complicated manufacturing procedures, it is necessary to perform coupled analysis of electrical, thermal and mechanical phenomena. Therefore, the simulation model must allow coupling of electrical, thermal and mechanical fields. At the same time, it must predict phase transformation temperatures and electrical, thermal and mechanical change of material properties in each steel type.
- (2) Applicable steels: Since a variety of steels are used in auto bodies now, from mild to high-strength steel, a key objective was to allow simulation of typical 270 – 980 MPa class steels in arbitrary sheet combinations and welding conditions. Galvaneal steels (GA steels) are used for automotive sheet products because of their corrosion resistance; it was also desirable to allow for the simulation of GA steels similar to uncoating steels.
- (3) Simulation accuracy: In consideration of joint strength of weldment, the nugget diameter was selected as the parameter for accuracy in this simulation. Generally, the optimal welding conditions for an automotive welded sheet combination are chosen on the basis of the range of electrical current, between the minimum current providing the necessary nugget diameter and the current that causes spatter. Figure 1 shows the relationship between welding current in two sheet combination with identical thickness and nugget diameter; in the low-current region, small differences

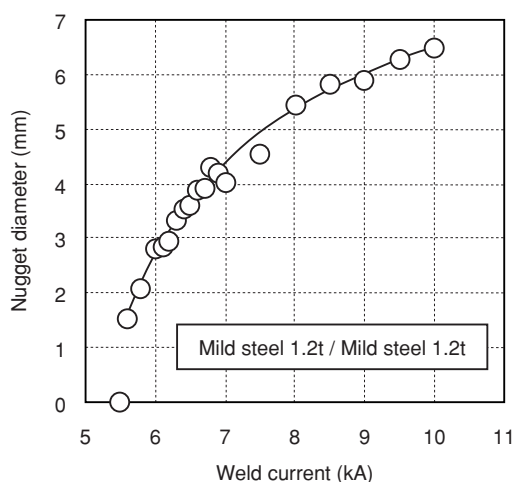


Fig. 1 Relationship between nugget and weld current

in welding current result in large changes in nugget diameter and there is considerable scatter in the measured diameters under otherwise apparently identical conditions. Accordingly, to verify the accuracy of this model, the nugget diameter range for this model was set between the necessary diameter and the diameter generated at the current that causes spattering.

The desired accuracy in prediction of nugget diameter was set at $\pm 5\%$, in consideration of the actual scatter in nugget diameter.

3. Simulation Model and Material Properties

3.1. Outline of Simulation Model

The 2-dimensional axially-symmetric finite element model (FEM) was used to simulate a pair of electrodes and the workpiece, consisting of two or more steel sheets (Fig. 2).

Electrical contact resistance in the interface is the dominant factor in the initial rise in temperature during resistance spot welding current; the generated heat generally depends on the temperature and pressure in the contacting interface⁽⁵⁾. These interface conditions change from moment to moment with the thermal conditions in the steel at the interface. As the materials are heated to higher temperatures, a solid/liquid phase change occurs, or a phase transformation in which the crystal structure of the steel changes, resulting in changes in mechanical characteristics, coefficient of liner expansion, or other material properties. Therefore, a simulation of resistance spot welding processes, including thermal changes of material properties and deformation processes accompanying temperature, must be based on a coupled analysis that accounts for metallurgical behavior. The framework presented in this report is illustrated in Fig. 3. It is based on a commercial nonlinear finite element code, and incorporates both a coupled electrical-thermal analysis for predicting temperature changes in steel sheets during current flow and a coupled thermal-mechanical analysis for predicting changes in mechanical characteristics with temperature rise in steel sheets. These analyses are repeated alternately with adequate small intervals in an incrementally coupled analysis method during the welding cycle, while considering changes in the material properties⁽⁶⁾. The electrical-thermal analysis is solved with contact information, such as contact area and pressure distribution, obtained from the previous thermal-mechanical analysis. Thermal history obtained from the electrical-thermal

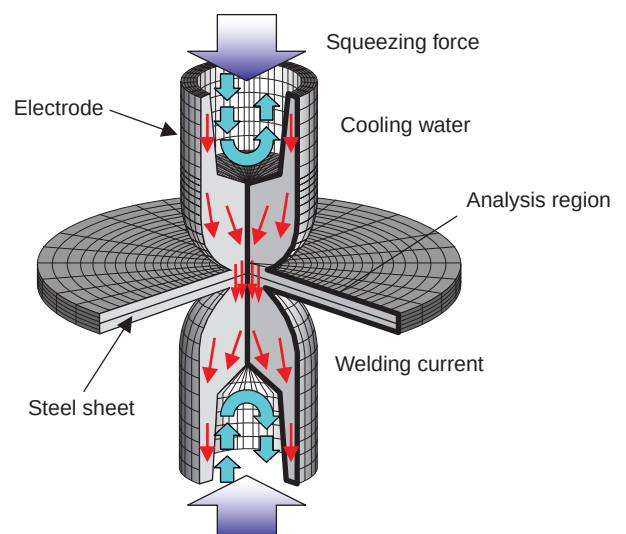


Fig. 2 Simulation model

analysis is then used as the input condition of the next thermal-mechanical analysis.

3.2. Material Properties

It was necessary to consider the changes in material properties of all applicable steels during the resistance spot welding process in order to obtain results that met the desired accuracy. Therefore the first investigation was into how to select and define the material properties necessary for this kind of simulation.

3.2.1. Selection of required material properties

As shown in Fig. 4, the resistance spot welding process was divided into three processes, (1) squeezing and turning on the current, (2) heating/melting, and (3) cooling, and the required material properties affecting each phenomenon were identified.

(1) Squeezing and turning on the current Resistance spot welding is initiated by putting workpieces between a pair of loaded fixed shaped electrode and activating the electrical current flow. Once current begins to flow, the material temperature begins to rise due to heating generated by electrical contact resistance and Joule heating. This changes the contact area and the contact pressure, important parameters in resistance spot welding heating behavior discussed below. Since these changes are caused by thermal distortion of the sheet with the temperature rise, it is necessary to define the changes in the material properties of the

steel, i.e., mechanical characteristics and the coefficient of liner expansion. Therefore, the stress-strain relationship (S-S property), accounting for temperature dependence, and the coefficient of liner expansion were selected for the required material properties during squeezing and turning on the current.

(2) Heating/temperature rise Heating during resistance spot welding process is caused by both electrical contact resistance and Joule heating.

Several models have been presented in the literature for contact resistance in the initial stage of current flow. This simulation employed the contact resistance model suggested by Babu et al⁽⁷⁾:

$$R_{CA} = (\rho_1 + \rho_2) \times \left[\frac{1}{4} \sqrt{\frac{\pi \sigma_y}{\eta P}} + \frac{3\pi}{16 \sqrt{\eta}} \right] \quad (1)$$

(R_{CA} , electrical contact resistance; ρ_1, ρ_2 , electrical resistivities of sheets 1 and 2; σ_y , yield stress; P , contact pressure; η , function of contact pressure P)

Joule heating occurring during exposure of a metal of a given resistance to a current over a given time can be expressed as in Eq. (2)⁽⁸⁾:

$$Q = R \times I^2 \times t \quad (2)$$

(Q , Joule heat; R , resistance; I , current; t , time)

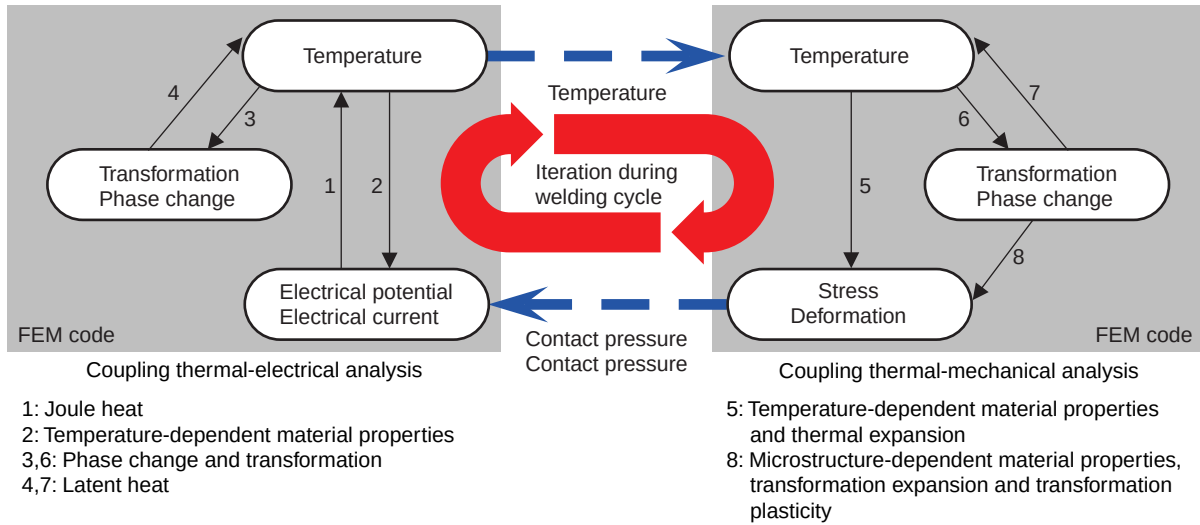


Fig. 3 Schematic illustration of the incrementally coupled electrical-thermal-mechanical analysis procedure

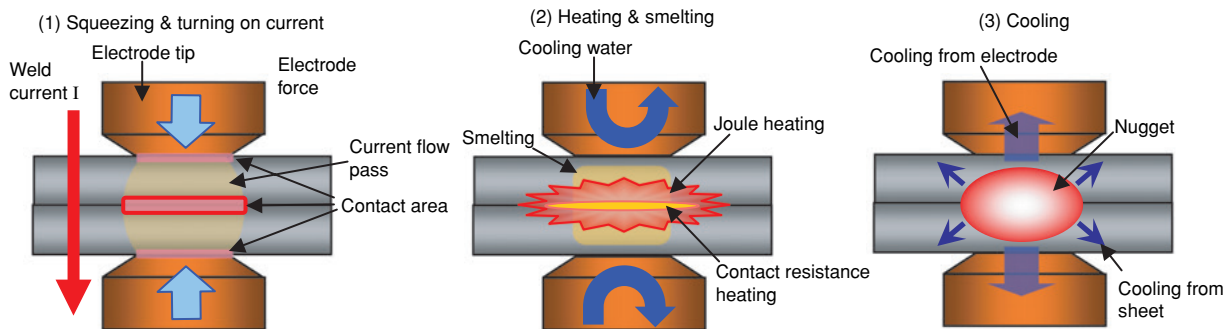


Fig. 4 Schematic explanation of the SPOT welding process

If Eq. (2) is rewritten in terms of electrical resistivity, which is one of the material properties of steel, and current density, which changes with rise in steel temperature, the Joule heat density can be expressed by Eq. (3):

$$q_v = \rho \times \delta^2 \times t \quad (3)$$

$$\rho = R \times \left(\frac{S}{l} \right), \quad \delta = \frac{l}{S}$$

(q_v , Joule heat per unit time, unit volume; ρ , electrical resistivity; δ , current density; t , welding time; S , contact area diameter; l , thickness)

As Eq. (4) shows, the relationship in resistance spot welding between heat generated during the welding time Δt and the temperature rise in the sheet steel is determined by the heat generated and the relationship between the specific heat of the material and its density.

$$\Delta\theta = \frac{q_v \Delta t}{(c \times D)} \quad (4)$$

($\Delta\theta$, temperature rise; c , specific heat; D , density)

Accordingly, because of the heat generated by electrical contact resistance, calculated in Eq. (1), the Joule heating calculated in Eq. (3) and the relationship between generated heat and temperature rise, shown in Eq. (4), the required material properties for heating and temperature rise phenomenon are electrical resistivity, yield stress, specific heat and density.

(3) Cooling Fourier's principle, which describes a material body with temperature differences, predicts the amount of heat conducted over a unit time. As Eq. 5 shows, the principle depends on the material's thermal conductivity and temperature gradient.

$$q = -\lambda \cdot \text{grad}(T) \quad (5)$$

(q , heat flow rate; λ , thermal conductivity; $\text{grad}(T)$, temperature gradient)

Cooling by heat conduction in the workpiece and the electrode occurs simultaneously with heating generated by turning on the current during the resistance spot welding process. Therefore, thermal conductivity was selected as the required material property for the cooling process.

In view of the above results, six material properties were chosen for resistance spot welding process simulation: the stress and strain (S-S) property, the coefficient of linear expansion, electrical resistivity, specific heat, density and thermal conductivity.

3.2.2. Defining material properties

The thermal dependence of all the material properties listed in the previous section needs to be found, but measuring these parameters for all steel types would be an extremely complicated and time-consuming process. Therefore, it seemed necessary to estimate the material properties of each steel by using its mechanical characteristics at room temperature and its chemical compositions, which are relatively easily obtained, and other properties.

(1) S-S property It is important to account for temperature dependence of the properties of S-S. An approximation equation was investigated to use for standardized, simple definition of the properties.

At first, to understand the temperature dependence, changes in S-S property with temperature were measured. Figure 5-1 shows measured results in the S-S curve for mild steel and high-strength steel. Both steels showed declining strength with higher

temperatures, and there was almost no difference between the two at 800 °C. As a reason for this phenomenon, it is assumed that the effects of the chemical compositions and microstructures which keep the S-S properties at room temperature practically disappears

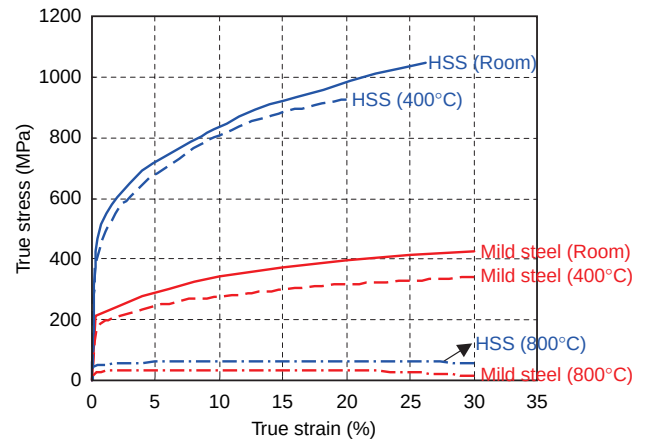


Fig. 5-1 S-S curve of mild steel and high-strength steel (HSS)

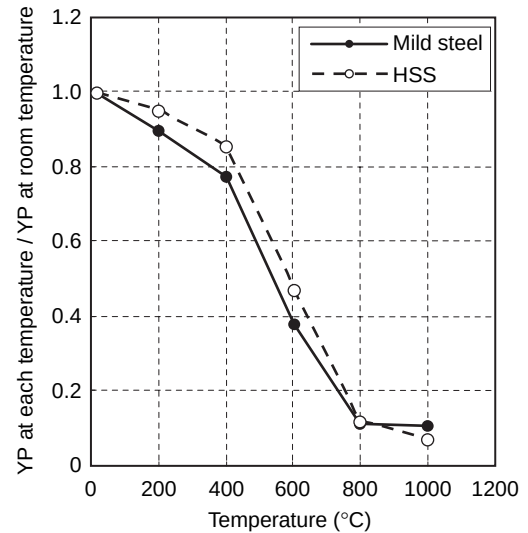


Fig. 5-2 Dependence of 0.2% proof stress on temperature

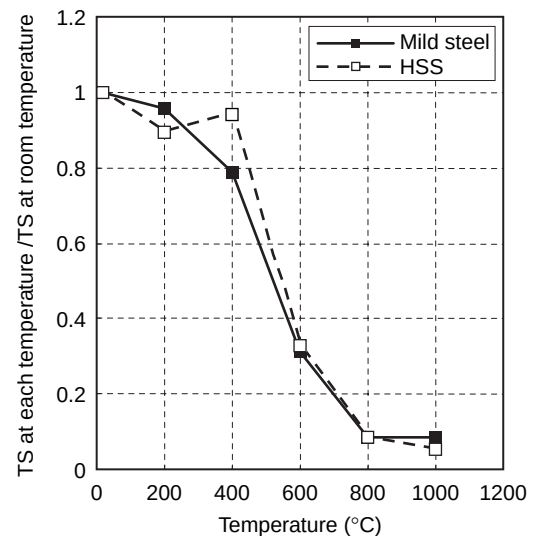


Fig. 5-3 Dependence of tensile strength on temperature

by grain growth and phase transformation in high temperature area.

Next, the temperature dependence of the tensile strength and yield strength were examined to standardize the temperature dependence of S-S property. As Figs. 5-2 and 5-3 show, both strengths were well represented by functions of temperature.

Thus an approximation equation was constructed to obtain the temperature dependence of the S-S property using the yield and tensile strengths at room temperature and the temperature as variables, allowing above property values at room temperature only to be inserted into the basic equation to define the S-S property at high temperatures.

(2) Thermal conductivity Generally, most of thermal conduction of a metal is represented by the propagation of heat for free electrons. The thermal conductivity is affected by the thermal vibration of metal atoms and by the presence of alloying elements. Accordingly, the chemical compositions and temperature were selected as the variables, employing Miettinen's formula⁽⁹⁾, which allows different expressions for different crystal structures. The results of the investigation are shown in Fig. 6-1. The measured

values for mild steel were nearly the same as the calculated values, but the thermal conductivity in high-strength steel diverged greatly from the measurements below 800 °C.

The authors then considered that the error might have been due to the influence of alloying elements not included as terms in Miettinen's formula. An approximation equation for the thermal conductivity was constructed that included such influences. Figure 6-2 shows the verification results of that equation.

The new equation showed a good correlation with the measurements for all types of steel tested. It is now possible to define thermal conductivity simply by inputting the chemical composition and the temperature.

(3) Electrical resistivity Since thermal conduction by free electrons is the dominant factor in metals, it has been shown that the reciprocal relationship between thermal conductivity and electrical resistivity shown in Eq. (7) holds; this is called the Wiedemann-Franz Law (W-F Law)⁽¹⁰⁾.

$$\lambda \rho = L \cdot (T + 273) \quad (7)$$

$$L = 2.45 \times 10^{-8}$$

(λ , thermal conductivity; ρ , electrical resistivity; T , temperature; L , Lorentz number)

Thus, the W-F Law was investigated for its utility in formulating an approximation equation for electrical resistivity. Figure 7-1 shows the results. The W-F Law somewhat underestimated the electrical resistivity, especially in high-strength steels, and provided no clear difference between high-strength and mild steels. This was probably because under the W-F Law, dependence on alloying elements, which have effects on the movement of free electrons, is thought to be similar for both thermal conductivity and electrical resistivity. However, this is not necessarily true in steel sheet.

Therefore, the authors considered that other influential factors could be the presence of allowing elements in the steel sheet and the thermal vibration of atoms of the base material, since electrical resistance in metals occurs due to collisions of the free electrons with the atoms making up the metallic crystal. The allowing elements are chemical composition values of the steel sheet that can be expressed as C, Si, Mn, etc., and the thermal vibration of the base material atoms can be expressed as a function of

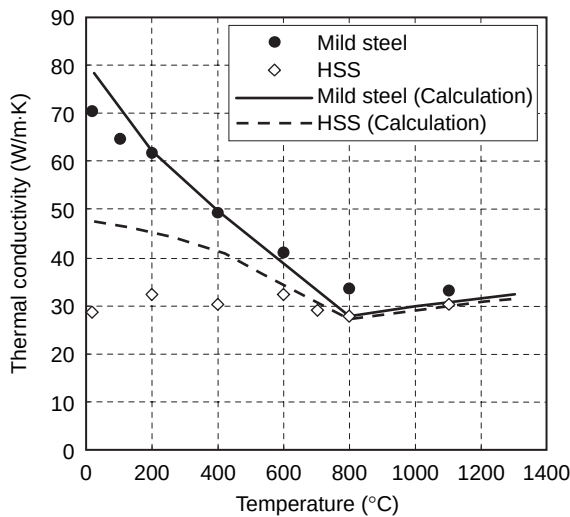


Fig. 6-1 Comparison of thermal conductivity between measurements and predictions by Miettinen's formula

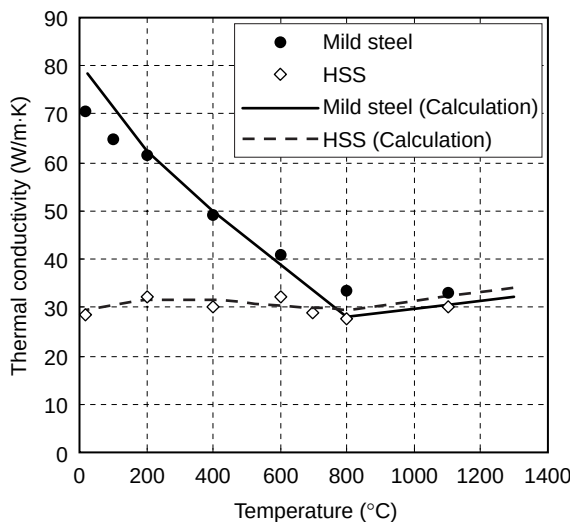


Fig. 6-2 Comparison of thermal conductivity between measurements and predictions by modified Miettinen's formula

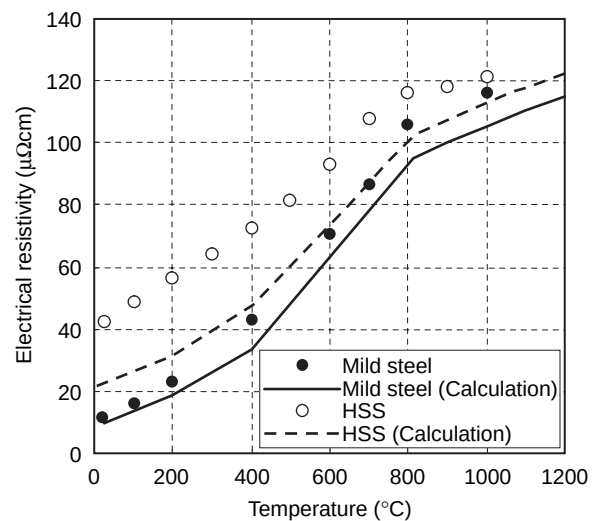


Fig. 7-1 Comparison of electrical resistivity between measurements and predictions by W-F Law

temperature. Accordingly, an approximation equation was constructed that was based on actual measured values and could express the differences between steel types. The results of the verification experiments using the equation are shown in Fig. 7-2. These had a better correlation with the measured values than the expression derived from the W-F Law and clearly distinguished between different steel types.

(4) **Specific heat** There are no organized data about the specific heats of different steels, leaving variations between steel types unclear, so changes in the specific heat with temperature were measured in various steels (Fig. 8). No differences between the types were found, so this parameter has been defined with identical values for every steel type.

(5) **Density and coefficient of linear expansion** The equation proposed by Jablonka et al.⁽¹¹⁾ is well established for density; it uses the carbon content and the temperature as parameters. However, the characteristics of steel sheets used in the automobiles are changed by adding C and numerous other elements as well as Si, Mn, so Miettinen's formula⁽⁹⁾ (Eqs. (8-1) – (8-3)), obtained by taking into account the influence of Si, Mn, Cr and Mo on Jablonka et al.'s expression, were employed.

(austenite)

$$\rho_r = 8099.79 - 0.506T + (-118.26 + 0.00739T)C_c - 68.24C_{Si} - 6.01C_{Mn} + 12.45C_{Mo} + (-7.59 + 3.422 \cdot 10^{-3}T - 5.388 \cdot 10^{-7}T^2 - 0.014271C_{Cr})C_{Cr} + (1.54 + 2.267 \cdot 10^{-3}T - 11.26 \cdot 10^{-7}T^2 + 0.062642C_{Ni})C_{Ni} \quad (8-1)$$

(ferrite)

$$\rho_a = 7875.96 - 0.297T - 5.62 \cdot 10^{-5}T^2 - 206.35 + 0.00778T + 1.472 \cdot 10^{-6}T^2 C_c - 36.86C_{Si} - 7.24C_{Mn} + 30.78C_{Mo} + (-8.58 + 1.229 \cdot 10^{-3}T + 0.852 \cdot 10^{-7}T^2 + 0.018376C_{Cr})C_{Cr} + (-0.22 - 0.470 \cdot 10^{-3}T - 1.855 \cdot 10^{-7}T^2 + 0.104608C_{Ni})C_{Ni} \quad (8-2)$$

(cementite)

$$\rho_{cc} = 7686.45 - 0.0663T - 3.12 \cdot 10^{-4}T^2 \quad (8-3)$$

The coefficient of linear expansion can be obtained from changes in density using Eq. (9). There is no need to measure the relationship between temperature and elongation using the Formaster tester. This method was used because the density found in Eq. (8) could be inserted in Eq. (9) to take into account the influence of chemical constituents in the steel sheet.

$$\alpha(T) = \frac{1}{T - T_R} \left[3 \sqrt[3]{\frac{\rho(T_R)}{\rho(T)}} - 1 \right] \quad (9)$$

($\alpha(T)$, coefficient of linear expansion; T , arbitrary temperature; T_R , room temperature; $\rho(T)$, density at arbitrary temperature T ; $\rho(T_R)$, density at room temperature)

3.3. Influence of Coating Layer

Coating steel is used to confer protection against corrosion in automobiles, so it is necessary to include the effect of weldment coating in this simulation method.

Generally, a higher welding current is necessary for a coated steel than in an uncoated one to obtain any desired nugget diameter. This is mainly attributed to the fact that melting of coating occurs before melting of base metal due to the melting point of coating being lower than that of the steel. This enlarges the area turning on the electrical current, so Joule heating is spread over a greater volume of material, reducing the temperature gain. The simulation was programmed to reflect the expansion in the diameter of the area turning on the electrical current. As shown in Fig. 9, the simulation accurately reproduced the actual measured values in

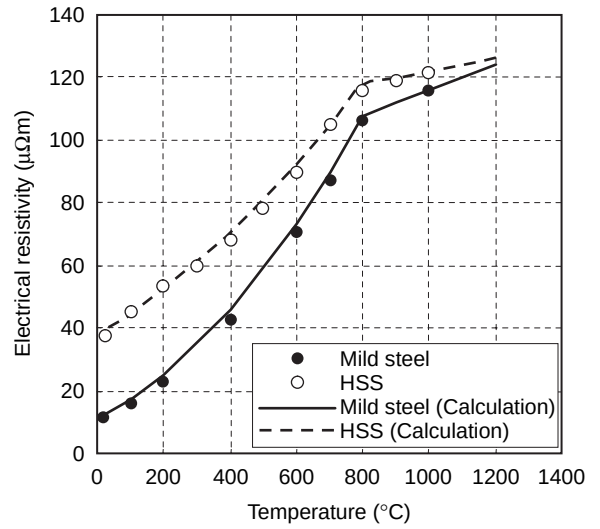


Fig. 7-2 Comparison of electrical resistivity between measurements and predictions using our modified equations

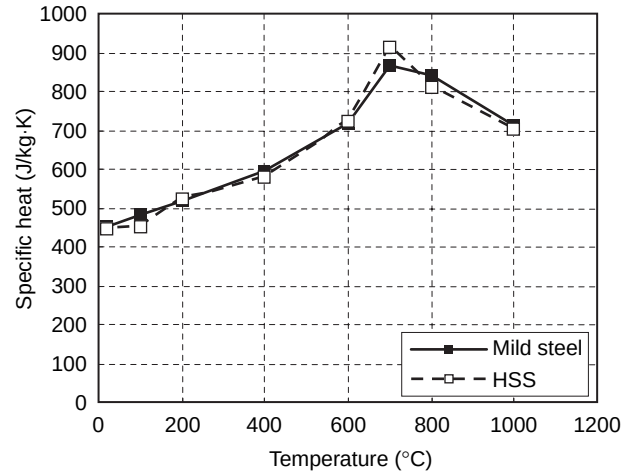


Fig. 8 Relationship between temperature and actual specific heat

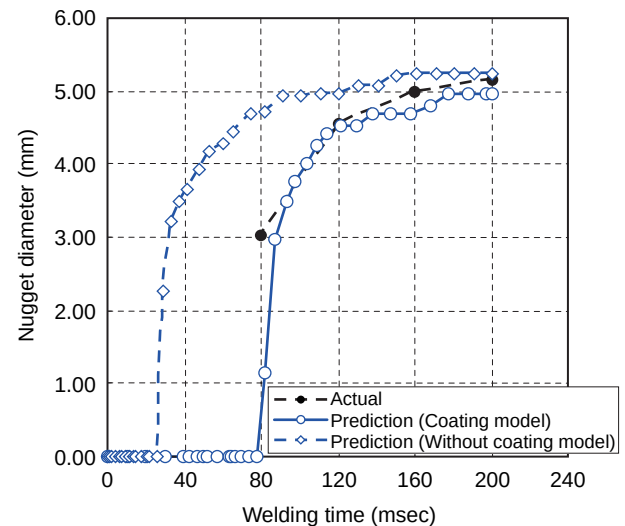


Fig. 9 Verification result of coating model

Table 1 Material properties of two-sheet combination

Sheet combination	Material	t mm	Mechanical properties		
			YP MPa	TS MPa	EI MPa
	HSS	1.6	432	811	27
	HSS	1.6	432	811	27

Table 2 Material properties of three-sheet combination

Sheet combination	Material	t mm	Mechanical properties		
			YP MPa	TS MPa	EI MPa
	Mild steel	0.7	138	305	49
	HSS	1.6	463	832	26
	HSS	2.0	568	631	18

the initial stage of nugget diameter. Sheet combinations of coated steel can be analyzed as well as those of uncoated steel with this model.

4. Development of Software

In order to execute this simulation, the user must create the model, set the boundary conditions and enter material properties and welding conditions for the arbitrary weldment sheet combination and electrode shape. Academic and practical expertise in FEM analysis, metallurgy and other areas is necessary to directly create the entry file for analysis in a commercial nonlinear FEM code, so a user-friendly interface was developed to ease use of this software.

It has been simplified so that anyone can use it; users need no special expertise to input weldment information, such as steel type and sheet thickness, or welding conditions such as pressure, welding time and welding current, to carry out an analysis. This software will be extremely effective in speeding up the process of analyzing weldments.

5. Verifying Simulation Accuracy

This software was used to simulate two assemblies, one of two sheets with the same thickness (Table 1), and the other of three sheets of differing thickness (Table 2). The accuracy of the simulations was examined and the results are provided in Figs. 10, 11-1 and 11-2. Both cases showed good correlation between the calculated values and the measured values. The nugget diameter was predicted with good accuracy.

The above results indicate that this simulation method accurately predicts nugget diameters in automotive sheet combinations in the initial stage. It also satisfies the objective of accuracy within $\pm 5\%$.

6. Effectiveness in Application

Two of the benefits expected in applying this simulation method in vehicle body development are the ability to do early investigations of welded sheet combinations and to speed the determination of welding conditions. At present, specifications and welding conditions are determined on an experimental basis using tensile tests and sectional macro inspections, which require a lot of time. This simulation allows the numerical calculation of

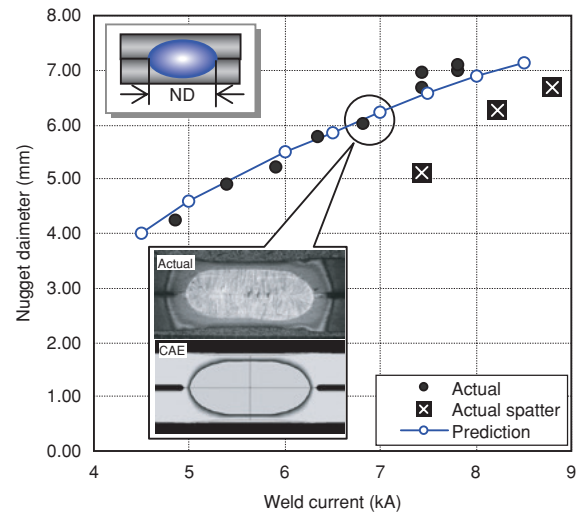


Fig. 10 Comparison of actual nugget diameter and predicted nugget diameter (two-sheet combination)

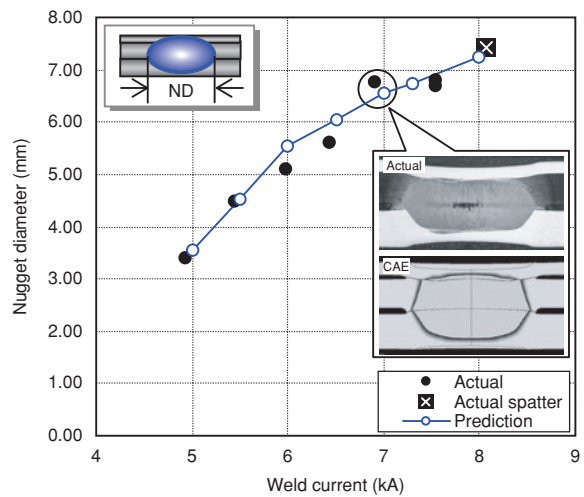


Fig. 11-1 Relationship between welding current and nugget diameter in actual sheet combination

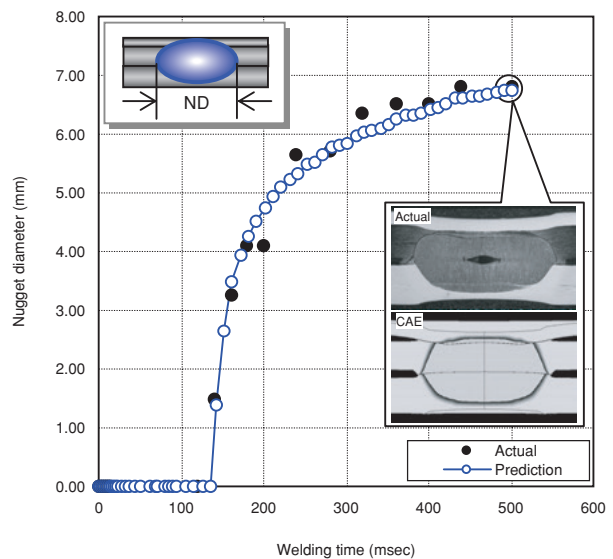


Fig. 11-2 Relationship between welding time and nugget diameter in actual sheet combination

weldability of automotive welded sheet combinations on the PC and early investigation of specifications. It may also allow for eliminating measurements of nugget diameters during sectional inspections, which would result in up to a 45% reduction in time.

7. Conclusion

A technology for simulating resistance spot welding using incrementally coupled analysis of electrical, thermal and mechanical fields has been developed to predict resistance spot welding conditions for automotive welded sheet combinations.

- (1) The required material properties for incrementally coupled analysis of electrical, thermal and mechanical fields were selected. A model equation capable of predicting the material properties based on chemical components and mechanical characteristics was constructed.
- (2) The simulation model was able to simulate the influence of galvanneal coating.
- (3) Software was developed allowing easy data input of arbitrary steel sheet weldments and welding conditions. No specialized expertise is necessary to quickly obtain analytical results.
- (4) The simulation model met the accuracy goal for predicting nugget generation processes in automotive weldments.
- (5) As a result of using this simulation, it is possible to investigate weldment specifications on the PC, and a total time savings of about 45% is anticipated for setting welding conditions.

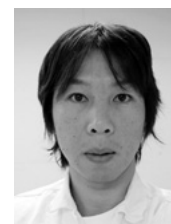
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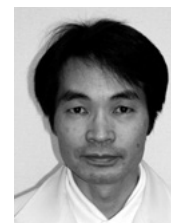
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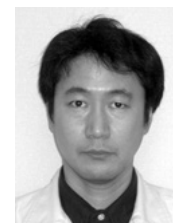
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Development of Metal Honeycomb Substrate for Small Two-wheeled Vehicle Catalysts

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ABSTRACT

With the growing concern on protecting the global environment, exhaust emission regulations are becoming more stringent than ever, not only in advanced countries, but also in developing countries. To purify exhaust gases, the use of a metal-honeycomb catalyst is effective. A metal substrate for a honeycomb catalyst has been developed recently to use in small motorcycles used by a large number of people in developing countries. As part of this development, we have established a new assessment method for the CAE analysis of thermal stress and the hot-cold cycle and vibration-durability tests. Making full use of the said techniques, a low-cost metal-honeycomb substrate featuring vibration resistance and hot-cold cycle durability has been developed.

1. Introduction

In recent years, exhaust gas emission regulations have become increasingly stringent, mainly in advanced countries, and this has spurred moves to strengthen regulations in developing countries as well.

To clean exhaust gas, countermeasures such as the development of new engine technology and devices, different carburetor settings, and FI⁽¹⁾ are being adopted. However, along with demands for increasingly effective purification performance, catalysts using a honeycomb substrate (hereinafter called “honeycomb catalysts”) are becoming indispensable, and in fact are now commonly used in automobiles.

The majority of motorcycles manufactured are used as the main mode of transportation in developing countries, especially in India, China and the ASEAN region, and demand is great for an inexpensive honeycomb for installation on motorcycles intended for use in those countries. The component elements of a honeycomb

catalyst are divided into the catalyst coating, called a load, which cleans the exhaust gas, and the honeycomb substrate, which supports the coating. Of these elements, the inexpensive ceramic honeycomb substrate, which is the main type used in automobiles, is not practical in most models of motorcycles from the viewpoint of vibration strength, as they lack the characteristics required to withstand the vibration inherent in a motorcycle. For this reason, a metal honeycomb substrate is used for motorcycles, even though it is more expensive. To increase the range of application of the honeycomb catalysts for small motorcycles intended for use in developing countries, it is essential that the cost of the metal honeycomb substrate be reduced without loss of durability.

Through this research, metal honeycomb substrates for small motorcycles that were inexpensive but highly durable were developed. These substrates are shown in Fig. 1. This paper describes these metal honeycombs for small motorcycles (Fig. 2), which were used on models intended to meet the 2005 Indian emission regulations, and the methods used to develop them.

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Fig. 1 Metal honeycomb substrates

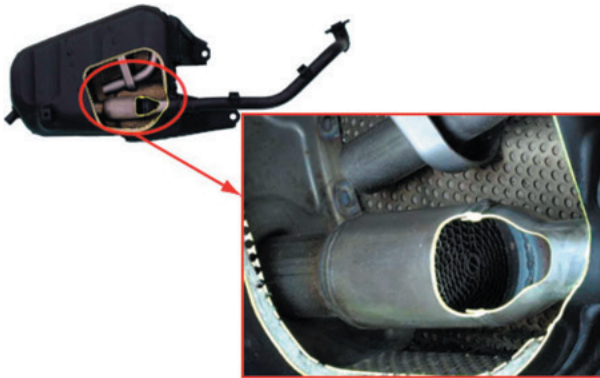


Fig. 2 Outline view of muffler with metal-honeycomb catalyst on small sized two-wheeled vehicle for India model

2. Background of Development

A metal honeycomb substrate (hereinafter called a “metal substrate”) is manufactured as follows: A ferrite-based stainless steel flat foil and a corrugated foil are put together and rolled up to form a honeycomb core. The honeycomb core is inserted into an outer jacket made of ferrite-based stainless steel. Figure 3 shows the flat foils and the corrugated foil used in the honeycomb core.

The flat foil, corrugated foil and outer jacket are bonded to each other by brazing, diffusion bonding, laser welding⁽²⁾⁻⁽⁴⁾, or some other method. Currently, the main method is brazing, using an Ni-based filler metal (shown in Fig. 4) that permits a high degree of bonding freedom.

Metal substrates have previously been used in small motorcycles in certain regions such as Thailand, but in most cases their specifications were developed at the beginning of the 1990s, which was the peak of the small 2-stroke engine era. Compared to a 4-stroke engine of the same displacement, the reciprocating parts of a 2-stroke engine, such as the piston and connecting rod, were heavy, so engine vibration was strong. Also, such 2-stroke engine motorcycles were generally inexpensive by design, so some of these machines did not even have a balancer shaft. For this reason, the vibration acceleration generated by the engine vibration was extremely high, so high mechanical durability was demanded. The metal substrate was designed to be durable enough to prevent the honeycomb core from dropping out due to fatigue resulting from the vibration forces *G* (hereinafter called “vibration *G*”) generated by a motorcycle, as shown in a and b of Fig. 5. The method used



Fig. 3 Flat foil and corrugated foil

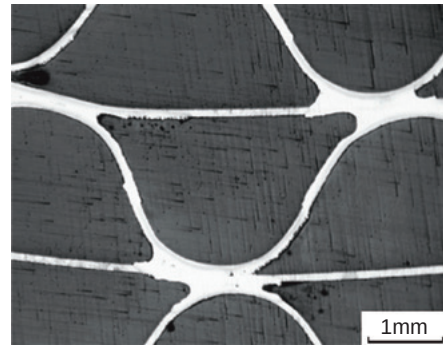


Fig. 4 Microstructure of cross section for Ni brazing

to obtain this durability involved increasing the bond area using Ni brazing filler metal to enhance the rigidity (Fig. 5-c).

However, the Ni brazing filler metal itself is expensive, so the question of how to minimize the amount of Ni brazing filler metal was of technical concern.

Recently, however, because of demands for clean exhaust gas, an increasing number of motorcycles are being fitted with 4-stroke engines⁽⁵⁾, which produce cleaner emissions than 2-stroke engines. Along with this, the required characteristics of the metal substrate have changed. The methods of mounting the engine and fixing the muffler are different from that of automobiles, so the required vibration *G* remains high, although the generated vibration *G* is about one half of that of a 2-stroke engine. The increase in the temperature of the exhaust gas due to the use of a 4-stroke engine causes a corresponding increase in the rate of temperature rise inside the honeycomb, which has brought about demands for higher hot-cold cycle durability. This situation is shown in Figs. 6 and 7 and also Table 1.

As mentioned above, the necessary characteristics for a metal substrate intended for a small motorcycle are as follows:

- Good vibration durability and hot-cold cycle durability
- Low cost

Knowledge about the brazing area and metal substrate properties determined from past developments are summarized in Table 2.

If the brazing area is taken as a parameter, it can be seen that the requirement of vibration durability is incompatible with the requirements of hot-cold cycle durability and cost. To satisfy all of these requirements, it is important to use the minimum amount of brazing filler metal necessary, and to use it as efficiently as possible. To enhance hot-cold cycle durability, it is important to minimize the thermal stress generated when the temperature of the metal substrate changes. The key to achieving this depends upon where inside the honeycomb, and also where between the honeycomb core and the outer jacket, brazing is carried out.

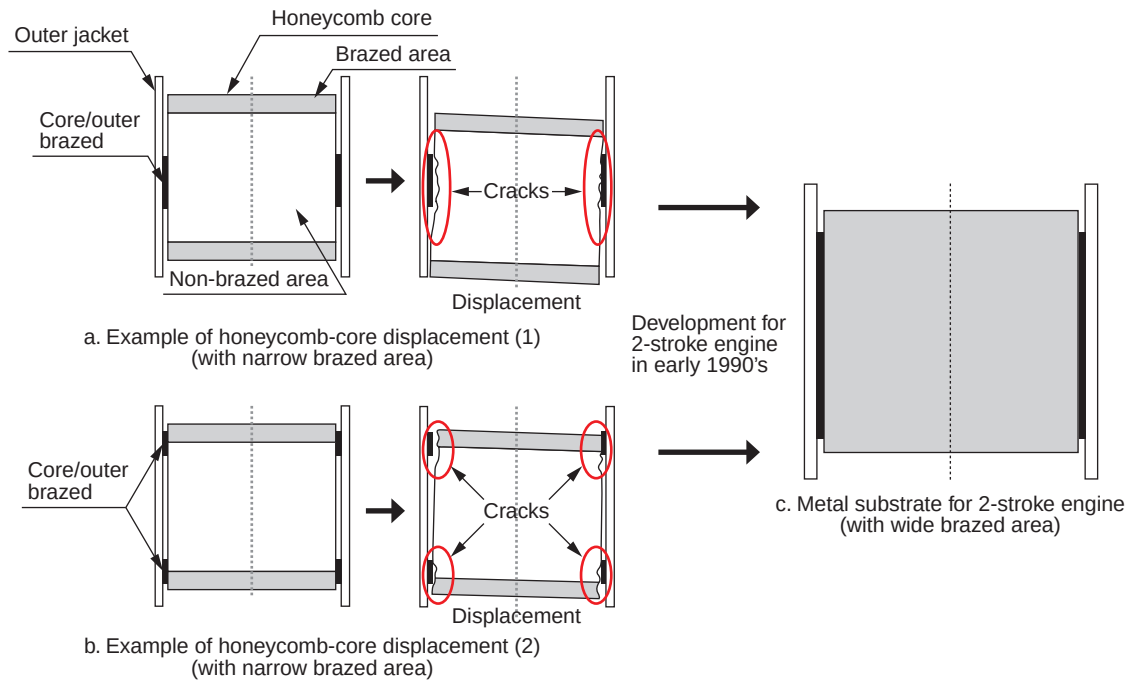


Fig. 5 Brazing joint structures of metal substrates

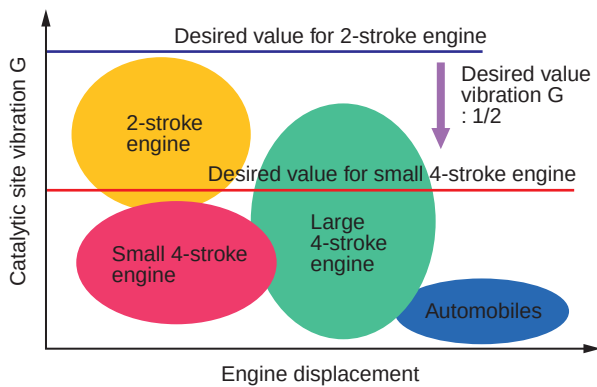


Fig. 6 Positioning of small size 4-stroke engine

Table 1 Comparison and demand characteristic of honeycomb catalyst load of 2-stroke engine and 4-stroke engine

Items	Comparison	Requirement characteristic
Highest temperature (See Fig. 7 area: A)	2-stroke >> 4-stroke	Vibration durability (High temperature fatigue strength)
Heating rate (See Fig. 7 area: B)	2-stroke << 4-stroke	Heat cycle durability

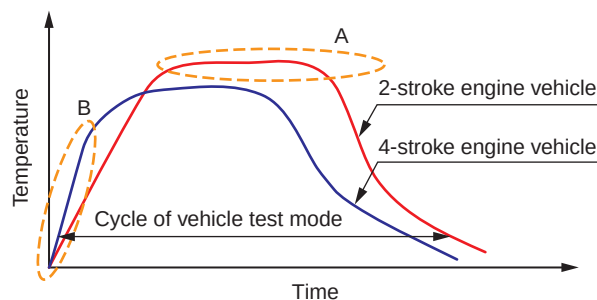


Fig. 7 Comparison of catalytic center temperature profiles during the endurance vehicle test mode

Table 2 Relationship between brazed area and properties of metal honeycomb substrates

Property	Brazing area Wide ↔ Narrow
Durability of vibration strength heat cycle resistant	Good ↔ Bad Bad ↔ Good
Cost	High ↔ Low

- Minimum brazing area necessary to enable the honeycomb to withstand the vibration G of a small motorcycle
- Optimization of the brazing locations to satisfy hot-cold cycle durability.

Also, to realize further cost reduction, it is necessary not only to reduce the amount of Ni brazing filler metal, but also to take into account productivity as well. To this end, a “simplified brazing design,” which obviates the need for a complicated production line, was adopted.

3. Contents of Development

3.1. Basic Concepts

The development concepts based on the model characteristics of small motorcycles are as follows:

3.2. Technical Methods of Specification Development

The technical methods used to satisfy the abovementioned concepts are as follows:

- Hot-cold cycle durability: Thermal stress CAE
- Vibration durability and hot-cold cycle durability: Unit durability test for new development

Hot-cold cycle durability of the metal substrate refers to the thermal fatigue life of the honeycomb core. CAE evaluation has recently been used to evaluate the thermal fatigue properties of exhaust system parts, particularly related to the thinner walls^{(6), (7)} of the exhaust manifold of an automobile. However, there are very few examples of a CAE evaluation being applied to a metal substrate. Also, because the phenomenon of thermal fatigue occurs in the plastic region, the few examples where CAE has been applied involved the implementation of Marc. or non-linear structure analysis⁽⁸⁾ computation which requires several days to complete. This method cannot be considered adequate from a practical viewpoint. In developing the new honeycomb substrate, general linear structural analysis was used, and a method of analysis in which the models, boundary conditions, and so on were optimized and used with thermal stress taken as an evaluation parameter. Figure 8 shows an actual example of heat transfer analysis using the model employed in this development.

Normally, when fatigue life assessment in the plastic region is carried out, it is not feasible to use stress as an evaluation parameter. However, a honeycomb metal substrate with conventional specifications has a durability level such that the core would not drop out, so there was no need to perform an absolute life assessment in this development project. Because the aim was to optimize the brazing locations to reduce thermal stress further, it was judged that thermal stress could be used as a relative evaluation parameter. As a result, it was possible to carry out screening in only a few hours, in contrast to the conventional evaluations in which prototypes with different specifications were actually made, and durability verification tests were carried out on an actual vehicle.

Concerning vibration durability, an apparatus was fabricated for a unit durability test in which vibration was applied while heating and cooling gases were alternatively pumped across the honeycomb substrate (this test is hereinafter called a hot-cold cycle vibration durability test), establishing a hot-cold cycle vibration durability mode that was correlated to an actual vehicle. Figure 9 shows an outline of this test apparatus. In this way, the appropriateness of the computed specifications from the viewpoints of vibration durability and hot-cold cycle durability were evaluated.

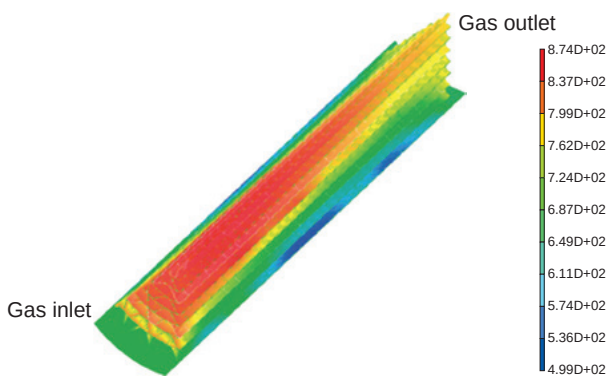


Fig. 8 CAE model; result of heat transfer analysis

3.3. Configuration of the Metal Substrate

As mentioned above, the component elements of a metal substrate can be classified into a honeycomb, an outer jacket and the brazing filler metal that joins them together.

For the honeycomb material, 20Cr-5Al ferrite-based stainless steel foil of 100 μm thickness was used because of its oxidation resistance and strength. Also, for the material of the outer jacket, Mo-added 17Cr-based stainless steel sheet (equivalent to JIS SUS436L) of 1.0 mm thickness was used because of its oxidation resistance.

Material that was equivalent to JIS BNi-5 was selected as the brazing filler metal. These materials are shown in Table 3.

3.4. Setting the Minimum Brazing Area

As shown in Fig. 10, the fracture load was measured for different brazing areas. Random variations of load and also the failure form were taken into account, and the brazing area that satisfied the specified load value was selected. This set value was no more than half of that used in the conventional 2-stroke engine specifications.

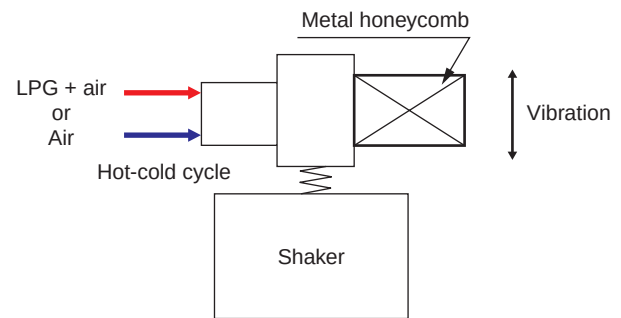


Fig. 9 Facility image of unit durability tests

Table 3 Composition of metal honeycomb substrate

Items	Materials
Honeycomb material	20Cr-5Al-REM (YUS205M1)
Outer jacket material	17Cr-Mo-LC (JIS SUS436L)
Brazing metal	Ni brazing filler metal (JIS BNi-5)

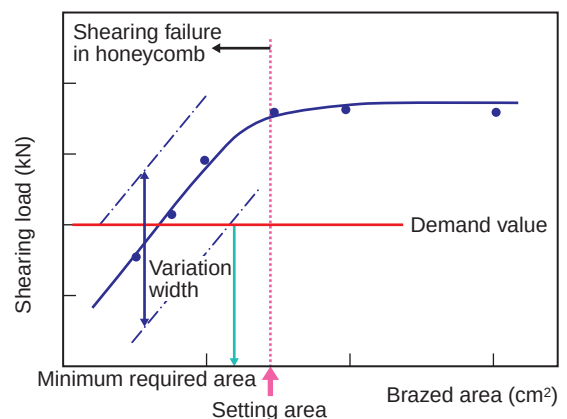


Fig. 10 Optimization of brazing area

3.5. Optimization of Brazing Positions

A small motorcycle was used, and the exhaust gas temperature and detailed temperature distribution inside the metal substrate were measured. The resulting values were assigned to the CAE model, and a CAE analysis was performed in the sequence of heat transfer analysis and thermal stress analysis.

The following three schemes were studied using as a standard the necessary bonding area obtained from the conventional metal substrate for a 2-stroke engine and the shearing load value.

Scheme A: Honeycomb core, and between the honeycomb core and the outer jacket (both brazed at the exhaust gas inlet side)

Scheme B: Honeycomb core, and between the honeycomb core and the outer jacket (both brazed at the exhaust gas outlet side)

Scheme C: Honeycomb core (brazed at the exhaust gas inlet side), and between the honeycomb core and the outer jacket (brazed at the exhaust gas outlet side)

From the CAE analysis results, scheme B, which generates the lowest thermal stress, was selected. Figure 11 shows an example of the actual analysis.

It was confirmed that by using the brazing specifications of Scheme B, the stress value at the outermost periphery of the honeycomb core, which is the weakest part of the metal substrate, was no more than about one half that of the stress value under conventional specifications (Fig. 12).

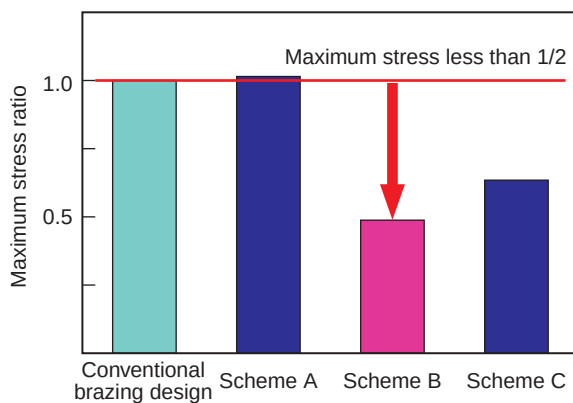


Fig. 12 Comparison of CAE results

3.6. Hot-cold Cycle Vibration Durability Unit Test Results

Based on the brazing position determined by CAE, the brazing width, amount of brazing material used, and other factors were optimized, and a test metal honeycomb was made. Table 4 shows the specifications of the test metal honeycomb.

A hot-cold cycle vibration durability test was carried out using this metal substrate, and the vibration durability and hot-cold cycle durability were evaluated. Figure 13 shows the external appearance of the metal substrate after the durability test. It can be seen that the core has neither dropped out nor been damaged due to the vibration G, and also that the cells in the vicinity of the outermost periphery of the core are neither deformed nor cracked, thus indicating conformity with the design scheme.

The test metal honeycomb was compared with the conventional honeycomb, and it was confirmed that the static shearing load after the durability test was about 2.5 times higher, as shown in Fig. 14.

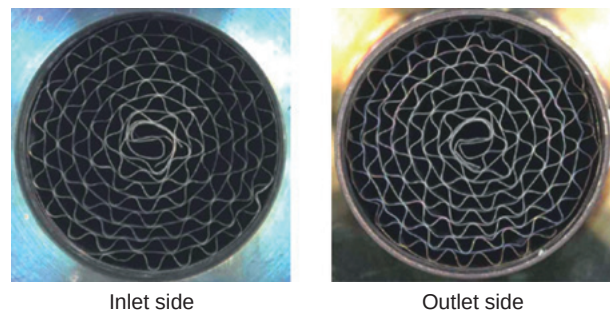


Fig. 13 Photograph of after unit durability test

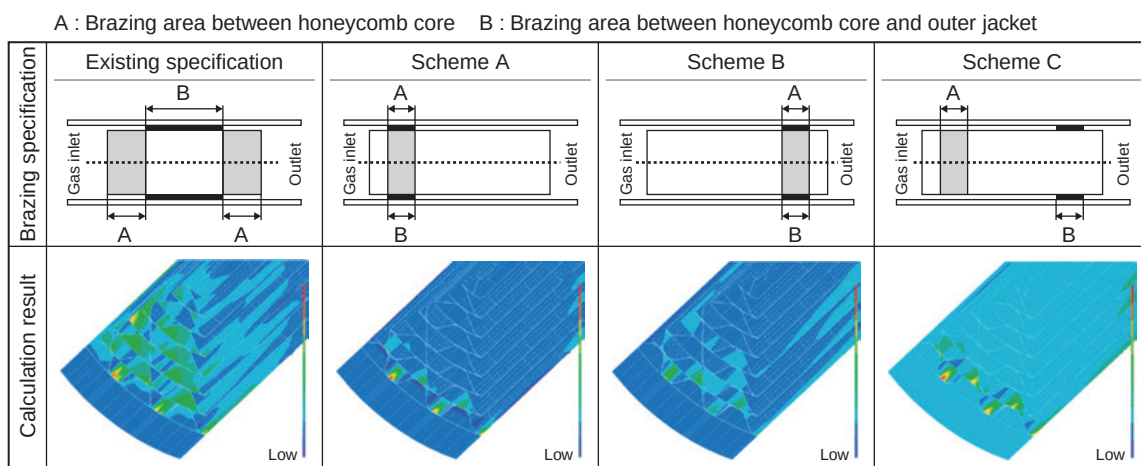


Fig. 11 CAE results of thermal stress analysis

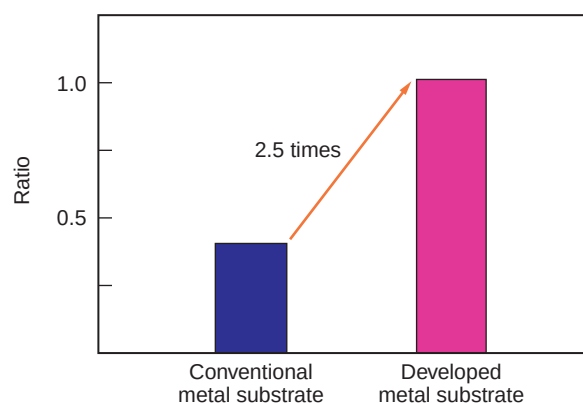


Fig. 14 Comparison of shearing load after unit durability tests

3.7. Evaluation of Durability in an Actual Vehicle

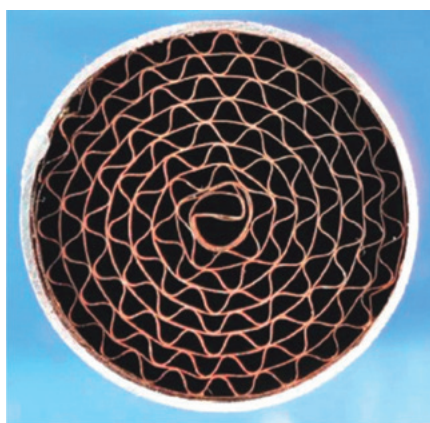
To make an overall evaluation of the function of the metal substrate as a component, the metal substrate indicated in Table 4 was coated with a catalyst, and subjected to a test in an actual vehicle.

(1) Mechanical durability

For the test vehicle, a model that generated a maximum vibration G roughly equal to the required vibration G was selected. After a durability test was performed with the metal substrate installed in a running vehicle, it was found that, just as in the unit test, the cells in the vicinity of the outermost periphery of the core were neither deformed nor cracked, thus indicating that the substrate had adequate durability (Fig. 15).

Table 4 Specifications of test metal honeycomb

Items	Specifications
Cell density	100 cpsi
Honeycomb core diameter	40 mm
Honeycomb core length	90 mm



Inlet side

Fig. 15 Photograph of after vehicle durability test

(2) Exhaust gas purification performance

To evaluate the deterioration of the exhaust gas purification performance, which includes the adhesion of the catalyst coating, the mounting construction of the exhaust pipe on which the metal honeycomb catalyst in the test vehicle was installed was derived, and an acceleration test using an actual machine was performed on the bench under conditions such that a vibration G of about twice that of the required vibration G was generated. It was confirmed that good adhesion of the catalyst coating layer still remained after the acceleration test. Also, as shown in Fig. 16, the substrate had exhaust gas purification performance at least equal to that of a conventional substrate.

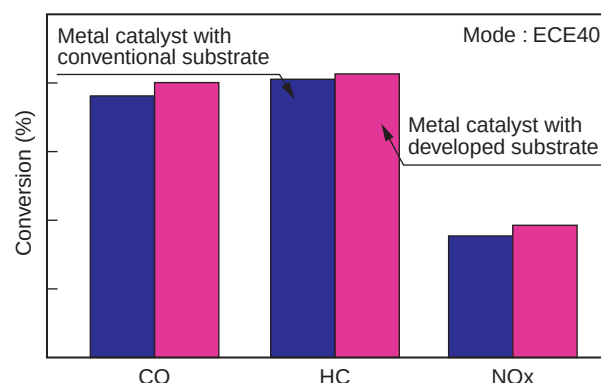


Fig. 16 Comparison of CO/HC/NOx conversion after vehicle durability tests

4. Conclusion

In the development of the metal substrate, methods of evaluating the thermal stress by CAE and hot-cold cycle vibration durability unit test were established. By using these methods, an inexpensive metal substrate that had both high vibration and hot-cold durability was developed.

- (1) The shearing load that would enable the substrate to withstand the vibration G produced by a small 4-stroke engine was set, and the necessary brazing area was determined. Compared to a conventional substrate, it was possible to reduce the brazing area to one half or less. In other words, it was possible to reduce the cost of the brazing material to one half or less.
- (2) To increase hot-cold cycle durability, a newly determined CAE method for estimating thermal stress in the honeycomb core was devised, aimed at optimizing the brazing position.
- (3) As a result of the newly established hot-cold cycle vibration unit test, vibration and hot-cold durability were evaluated, and it was confirmed that the static shearing load after the durability test was 2.5 times higher than that of a conventional substrate.

The developed metal honeycomb substrate for use in small motorcycles was first put into application in 2005 models destined for India. They are scheduled to be installed in future small motorcycles destined mainly for developing countries. Also, the evaluation method devised while developing the metal honeycomb substrate is scheduled to be used as an effective evaluation method in the diversification of specifications of metal honeycomb substrates, which would be expected from now on to achieve additional exhaust gas purification performance.

Acknowledgements

The authors wish to take this opportunity to express their gratitude to all concerned at Goshi Giken Co., Ltd.

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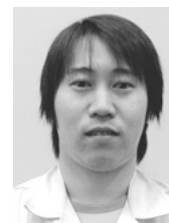
■ Authors ■



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Technical Papers Submitted to Out-of-company: July, 2005 to December, 2005

(English Papers only)

Date	Conference	Title	Author	Organization
2005.7.4-8	IFAC (International Federation of Automatic Control)/16th IFAC World Congress (Prague, Czech)	Development of Obstacle Avoidance System using Co-operative Control	Yukihiro Fujiwara Yasushi Shoda Shuichi Adachi	Tochigi R&D Center Tochigi R&D Center Utsunomiya University
2005.7.17-22	ASME with the participation of the American Institute of Chemical Engineers/2005 ASME Heat Transfer Summer Conference (San Francisco, USA)	Infrared radiation from dry and wet porous silicon	Kazuyoshi Akutsu Masahiro Hirota Masayoshi Kitada Hajime Hirata Yuko Amo	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Yamagata University
		Large-Scale Quantum Chemical Molecular Dynamics Simulations on the Formation Dynamics of Hydrogen by the Chemical Reactions of Water	Masahiro Hirota Kazuyoshi Akutsu Masayoshi Kitada Hajime Hirata Momoji Kubo Ewa Broclawik Akira Miyamoto Yumiko Sasaki Keiko Chiba Pei Qiang Abdul Rajjak Hideyuki Tsuboi Michihisa Koyama	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tohoku University Tohoku University Tohoku University Tohoku University Tohoku University Tohoku University Tohoku University Tohoku University Tohoku University
2005.7.30	JSAE/Review of Automotive Engineering Vol.26, No.3	Research on Technique for Noise Source Analysis of Pass-by Noise Using Inverse Matrix Method	Yukihiko Hanzawa Jyun Kitahara Hiroshi Nakagawa Hirofumi Nakajima	Tochigi R&D Center Tochigi R&D Center Nittobo Acoustic Engineering Co., Ltd. Nittobo Acoustic Engineering Co., Ltd.
		Development of 590/780 MPa Grade High-Ductility Steel Sheets	Takahiro Tanae Katsuyoshi Kaneko Yoshihisa Takada Atsushi Itami	Tochigi R&D Center HONDA R&D EUROPE (U.K.) LTD. Nippon Steel Corporation Nippon Steel Corporation
		Development of Aluminum Subframe Using Hot Bulging and Vacuum Die Casting	Fumiaki Fukuchi Tsutomu Ogawa Izuru Hori Takanori Yahaba Hiroshi Akiyama	Tochigi R&D Center Tochigi R&D Center Honda Engineering Co., Ltd. Tochigi R&D Center HONDA R&D AMERCAS, INC. OHIO CENTER
		Development of New Intelligent Tourer 2004 Model INSPIRE	Shigehiro Yasuki	Tochigi R&D Center
2005.8.1	MTZ Volume 66 7-8/2005	Development of the Diesel Engine Series i-CTDi by Honda	Kenichi Nagahiro Tomoya Abe Kenichi Okawara Masakazu Yamazaki Ikuro Hara	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
	Auto Technology Volume No.4 August 2005	Honda's 2.2-Litre 4-Cylinder i-CTDi Engine	Ikuro Hara Isao Kaneko Kenji Fujiki Masami Ohshima	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
2005.8.2-6	IEEE/IROS2005 (International Conference on Intelligent Robots and Systems) (Edmonton, Alberta, Canada)	Analysis of humanoid appearance in human-robot interaction	Takayuki Kanda Takahiro Miyashita Taku Osada Yuji Haikawa Hiroshi Ishiguro	ATR Intelligent Robotics and Communication Labs. ATR Intelligent Robotics and Communication Labs. Wako Research Center Wako Research Center ATR Intelligent Robotics and Communication Labs.
2005.8.22-24	KSAE/IPC-13 (The 13th International Pacific Conference on Automotive Engineering) (Kyongju, Korea)	Application of Beamforming Method to Identification of Sources of Vehicle Pass-by Noise	Yukihiko Hanzawa	Tochigi R&D Center

		Validation and Application of Aero-acoustic Noise Measurement of Vehicle using Surface Microphone in Wind Tunnel	Masaki Tsuchiya Tsuyoshi Yamashita Niels V. Bogholm Toshikazu Sato Masaki Kimura	Tochigi R&D Center Tochigi R&D Center Bruel & Kjaer Bruel & Kjaer Division of Spectris Co. Ltd., Japan Bruel & Kjaer Division of Spectris Co. Ltd., Japan
		Study of Harshness Noise Reduction Technique	Taro Yamashita	Tochigi R&D Center
		Ignition and Combustion Modeling with G-Equation in Spark Ignition Engines	Xiaofeng Yang Toshihiro Ohashi Toru Takabayashi Shunichi Kubota Hideki Katsuyama Yasuhiro Urata	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Research of low tension ISG Belt-drive System for Idling Stop System	Kazuma Okuda Tadashi Fujiwara Youichi Nakahara Yasuyuki Komatsu Shigeru Aoki	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Development of ACCORD Hybrid System	Koichiro Takemasa Masaaki Kaizuka Toshitaka Imai Susumu Ishikawa Shinji Yasuno Satoshi Watanabe	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Research on the Turnover Fuel cut Valve in the Fuel Injection System of a Two-Wheeled Small Vehicle	Toichiro Hikichi Toshio Yamamoto	Asaka R&D Center Asaka R&D Center
		Research on Ultra Low Emission Technology for Large Displacement Motorcycles	Tomoya Kono Hiroaki Miyata Mamoru Uraki Ryutaro Yamazaki	Asaka R&D Center Asaka R&D Center Asaka R&D Center Asaka R&D Center
2005.8.29-9.2	International Association for Vehicle System Dynamics/XIX IAVSD Symposium (Milano, Italy)	Simultaneous Robust Optimization of Suspension and Active Control System of Road Vehicles for Handling Improvement	Shinya Nohtomi Hiroyuki Urabe Kazuyuki Okada Shinichiro Horiuchi	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Nihon University
2005.9.2-5	IEEE (The Institute of Electrical and Electronics Engineers, Inc.)/2005 IEEE Congress on Evolutionary Computation (Edinburgh, UK)	A New Approach to Dynamics Analysis of Genetic Algorithms without Selection	Tatsuya Okabe Yaochu Jin Bernhard Sendhoff	Wako Research Center Honda Research Institute Europe GmbH Honda Research Institute Europe GmbH
		Theoretical Comparison of Genetic Algorithms and Evolution Strategies - Dynamics of Offspring Generation -	Tatsuya Okabe Yaochu Jin Bernhard Sendhoff	Wako Research Center Honda Research Institute Europe GmbH Honda Research Institute Europe GmbH
2005.9.8-9	AVL/17th International AVL Engine & Environment Conference (Graz, Austria)	How Up-to-Date Control Method Give the Improvement for Powertrain	Yasuyuki Sando	Tochigi R&D Center
2005.9.12-14	IEEE (The Institute of Electrical and Electronics Engineers, Inc.) Microwave Theory and Techniques, Antennas and Propagations/2005 Workshop on Computational Electromagnetics in Time-Domain (CEM-TD 2005) (Atlanta, GA, USA)	A New Hybrid Method for Analysing Scattering Problems	Masatoshi Harumatsu	Wako Research Center

2005.9.12-15	Japanese Society of Human Environment System/The 3rd International Conference on Human-Environment System (ICHES'05) (Tokyo, Japan)	New system for the analysis of thermal judgments: multi-point measurement of skin temperatures, thermal sensations, thermal affect, and their joint visualization.	Hidegori Esaki Naoki Osawa Mayumi Nakamura Kazuyuki Kanosue Kei Nagashima Akiko Kobayashi Aki Konishi Sho Yasuhara	Tochigi R&D Center Tochigi R&D Center Waseda University Graduate School Waseda University Waseda University Waseda University Waseda University
2005.9.13-14	Technische Universitaet Dresden IFTE/Technische Universitaet Dresden IFTE/10. Tagung Zahnriemengetriebe (Dresden, Germany)	Simulation of belt system dynamics using a multi-body approach	Hiroshi Takagishi Hiroshi Yoneguchi Martin Sopouch Ingolf Thiele	Tochigi R&D Center Tochigi R&D Center AVL AVL
2005.9.14	Carbon Nanotube and Nanofiber Materials and Applications Workshop/Midwest Chapter of the SAMPE (Society for the Advancement of Material and Process Engineering) (Kettering, OH, USA)	The study of plasma coated carbon nanotube as composite application	Kazuhiro Miura Toshiyuki Ohashi	Wako Research Center Honda Research Institute USA Inc.
2005.9.18-20	SAE/Homogeneous Charge Compression Ignition (HCCI) Symposium (Lund, Sweden)	Experimental Studies on the Auto-ignition Combustion for the Petrol Engines	Yoichi Ishibashi	Asaka R&D Center
2005.9.21	JSCES (The Japan Society for Computational Engineering and Science) (Tokyo, Japan)	A New Approach to Dynamics Analysis of Genetic Algorithms without Selection	Tatsuya Okabe Yaochu Jin Bernhard Sendhoff	Wako Research Center Honda Research Institute Europe GmbH Honda Research Institute Europe GmbH
2005.9.24-28	ASME 2005 International Design Engineering Technical Conferences (California, USA)	Sound-Radiation Analysis for Boat Hull Based on Hammering Test and BEM	Taro Tomatsu Tetsuya Otsuka Masaaki Okuma Takeshi Okada Tetsuro Ikeno Kazuyuki Shiomi	Tokyo Institute of Technology Tokyo Institute of Technology Tokyo Institute of Technology Asakahigashi R&D Center Asakahigashi R&D Center Asakahigashi R&D Center
		A Method to Identify the Stiffness of Engine Mounts using Experimental Modal Analysis	Taro Tomatsu Takeshi Okada Tetsuro Ikeno Kazuyuki Shiomi Masaaki Okuma	Tokyo Institute of Technology Asakahigashi R&D Center Asakahigashi R&D Center Asakahigashi R&D Center Tokyo Institute of Technology
2005.9.27-29	International Council for Powertrain Engineering & Management/GPC2005 (Global Powertrain Congress) (AnnArbor, USA)	Next-Generation Fuel Cell Stack for Honda FCX	Jun Kondo Masajiro Inoue Nobuhiro Saito	Wako Research Center Wako Research Center Wako Research Center
2005.10.1	JSAE/Review of Automotive Engineering Vol.26, No.4	Development of New Honda LEGEND	Junichi Fukuda Masahiro Watanabe Masafumi Takita Takefumi Nakamura Yasunori Oku Shoichi Nemoto	Tochigi R&D Center Wako R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Optimization of Combustion Chamber for Direct Injection Gasoline Engine Employing Center Injection System	Shinya Abe Naoaki Takeda Shigeru Akimoto Naohiro Ishikawa Hiromi Matsuura	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
2005.10.2-6	IEEE/IEEE IAS 2005 (China)	Influence of PM Flux Variation and Magnetic Saturation on Sensorless Control for Interior Permanent Magnet Synchronous Motors	Nobuyuki Imai Shigeo Morimoto Masayuki Sanada Yoji Takeda	Tochigi R&D Center Osaka Prefecture University Osaka Prefecture University Osaka Prefecture University

2005.10.4-6	ELSEVIER/Grove Fuel Cell Symposium (London, UK)	Fuel Cell development for automobile application	Yuji Kawaguchi	Wako Research Center
2005.10.5-6	Institute of Automotive Engineering Aachen/Aachen University of Technology/14th Aachen Colloquium "Automobile and Engine Technology" (Aachen, Germany)	Development of Low Emission Diesel Engine with DPF System	Isao Kaneko Isao Chiba Ryuuta Matsui Ryuuji Matsukado Yoshinori Ishihara	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
2005.10.12-14	JSAE & SAE/SETC '05 (2005 Small Engine Technology Conference and Exhibition) (Bangkok, Thailand)	Development of Electronically Controlled Belt-type CVT for Motorcycles	Michio Asumi Hideo Ishikawa Kazuhiko Mori Naotada Matsudaira	Asaka R&D Center Asaka R&D Center Asaka R&D Center Asaka R&D Center
		Fuel Injection System of Air-cooled Engines for Small Displacement Motorcycles	Katsunori Komuro Katsuichi Yagisawa Shunji Akamatsu Akira Hayashi Minoru Ueda	Asaka R&D Center Asaka R&D Center Asaka R&D Center Asaka R&D Center Asaka R&D Center
		Development of High-Pressure Die-Casting Simulation and Verification in Motorcycle Parts	Masahiko Nakagawa Koichi Chiba Eitaro Koya	Asaka R&D Center Asaka R&D Center Asaka R&D Center
		Research of an Electronic Controlled Fuel Supply System for a Diesel Engine for Small Displacement Two-wheeled Vehicles	Yoshihide Nanba Yoshihiro Takada Tomoo Shiozaki	Asaka R&D Center Asaka R&D Center Asaka R&D Center
		Study of CNG Fueled Two-Wheeled Vehicle with Electronically Controlled Gas Injection System	Takashi Tsutsumizaki Masahiro Kuroki	Asaka R&D Center Asaka R&D Center
		Effect of Active Piston-Movement Control on Thermal Efficiency in Different Heat Release Profiles	Yasuo Moriyoshi Masaki Sano Seung-hwan Choi Masatoshi Suzuki Satoshi Iijima	Chiba University Chiba University Chiba University Asaka R&D Center Asaka R&D Center
		Development of Pb-Free, Free-Cutting Microalloyed Steel With High Toughness for Crankshafts of Small Multipurpose Engines	Shingo Shimizu Yuzuru Tahara	Asakahigashi R&D Center Asakahigashi R&D Center
		Study of Sound Isolation Structure for Engine Generators	Katsumi Maruyama Toru Fukuda Kazuhiro Takemura Shinichi Suzuki	Asakahigashi R&D Center Asakahigashi R&D Center Asakahigashi R&D Center Asakahigashi R&D Center
		Research on Prediction of Cooling Air Flow in Small Air-Cooled Utility Engines	Takashi Kushibiki Toshiro Kiura Kazuyuki Shiomi	Asakahigashi R&D Center Asakahigashi R&D Center Asakahigashi R&D Center
2005.10.23-27	361.WE-Heraeus-Seminar on Hydrogen Storage with Novel Nanomaterials/Wilhelm und Else Heraeus-Stiftung (WE-Heraeus Foundation), H-Workshop 2005 (Bad Honnef, Germany)	Hydrogen storage properties of noble metal clusters	Hajime Goto Terumi Furuta	Wako Research Center Wako Research Center
2005.10.25-27	Automotive Circle International/EURO CAR BODY 2005 (Frankfurt, Germany)	The new Honda Civic car body	Hironao Ito Katsumi Semba Shigeto Yasuhara Hirokuni Kumekawa Takashi Minegishi Kenji Ohkura Hidekazu Shono Masayuki Fukumori	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Honda Engineering Co., Ltd.

2005.10.31	The Electrochemical Society, Inc./Journal of the Electrochemical Society	Advanced mesophase carbons for Electrochemical Capacitor	Takeshi Fujino Minoru Noguchi	Wako Research Center Wako Research Center
2005.11.5-11	ASME/2005 ASME International Mechanical Engineering Congress and Exposition (Florida, USA)	Simulation of Fracture Behavior of Elastomer Modified Polypropylene Based on Elastoviscoplastic Constitutive Equation with Craze and Tensile Softening Law	Hiroyuki Mae	Tochigi R&D Center
2005.11.8	ITS AMERICA/ITS World Congress 2005 (San Francisco, USA)	Intelligent Night Vision System	Takayuki Tsuji Hideki Hashimoto	Tochigi R&D Center Tochigi R&D Center
2005.11.9-10	HART ENERGY PUBLISHING/WORLD FUELS CONFERENCE ASIA 2005 (Beijing, China)	ENERGY DIVERSITY AND OPTIONS FOR AUTOMOBILE	Kaoru Horie	Tochigi R&D Center
2005.11.10	Aluminum Association Auto & Light Truck Group/Automotive Aluminum Technology Forum (Detroit, USA)	Development of Aluminum Application Technologies for Automotive Components	Katsuhiko Shibata	Tochigi R&D Center
2005.11.14-18	Fuel Cell Seminar Organizing Committee/Fuel Cell Seminar 2005 (Palm-springs, CA, USA)	Next-Generation Fuel Cell Stack for Honda FCX	Yosuke Fujii Masajiro Inoue Nobuhiro Saito	Wako Research Center Wako Research Center Wako Research Center
		Research of Home Energy Station	Ikuya Yamashita Takashi Arai	Wako Research Center Wako Research Center
2005.11.15	AAAM, ICADTS, IRCOB, ITMA/Traffic Injury Prevention	INFLUENCE OF PEDIATRIC AGE ON THE INCIDENCE OF BODY REGION INJURY AND OVERALL INJURY SEVERITY IN CHILD PEDESTRIAN CASUALTIES	B. Johan Ivarsson Jeff R. Crandall Masayoshi Okamoto	University of Virginia University of Virginia Tochigi R&D Center
2005.11.16-18	The Electrochemical Society of Japan/The 46th Battery Symposium (Aichi, Japan)	Development of NiMH Battery for HEV	Masahito Mitsumori Katsuya Inoue Tsuyoshi Matsuda	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
2005.12.5-7	Redox Engineering, LLC/THE 15th INTERNATIONAL SEMINAR ON DOUBLE LAYER CAPACITORS AND HYBRID ENERGY STORAGE DEVICES (Florida, USA)	Characterization of advanced mesophase carbon using novel mass production methods	Takeshi Fujino Byoungju Lee Shigeki Oyama Minoru Noguchi	Wako Research Center Tochigi R&D Center Wako Research Center Wako Research Center
2005.12.5-9	CBMM/International Symposium on Nb Containing Steel Sheet for Automotive Application (Brazil)	The weight reduction of automobile by High Strength Steel	Yuichi Azuma	Tochigi R&D Center
2005.12.20-22	JSAE/18th Internal Combustion Engine Symposium (International) (Jeju, Korea)	Two-Stroke Engines as Means of Improving Vehicle Fuel Economy-A Large Scooter Application of the Pneumatic Direct Injection-AR Engine	Kenji Nishida Takahiro Kimijima	Asaka R&D Center Asaka R&D Center

Editor's Postscript

Our fourth gathering for the presentation of technical papers took place at the Tochigi R&D Center on January 11—so early in 2006 that the New Year's fog had not yet completely lifted. In addition to the usual presentation of papers, there was also a special lecture given this time by Executive Chief Engineer Sugiyama, who has served as chair of the Technical Paper Committee for the past four years. He began his talk by showing a slide of an old experimental research report, which he mentioned was the starting point of these paper presentations. He closed by saying that our "Three Joys" as research center engineers should be the joy of doing research, the joy of development and the joy of giving presentations, and that we should take every advantage of these paper presentations as occasions that allow us to experience the joy of giving presentations. *The Honda R&D Technical Review* is now entering its 18th year since the inaugural issue, and I can see that many of us have utilized the publication as a forum for the joy of giving presentations. Our authors, of course, have included people who have had to carry on product development while writing articles concurrently, and I'm sure that they've put forth a great deal of effort in doing so. I believe, however, that they've gained a correspondingly greater sense of accomplishment from this.

Recording the results of research in written form is of course one of the joys of being an engineer. It is also an extremely important means for passing on knowledge and technology. However, it seems to me that people have been setting things down in writing less and less in recent years. Up to around a decade ago, people were putting a great deal of time and effort into crafting their experimental research reports into publications for other people to read, composing them with the purpose, conclusion, process by which the conclusion was reached, results, and observations, in that order. Advances in personal computers have brought considerable changes in how people communicate information to others. Pictorial and graphic presentations present things to us visually, and when we listen to the narrations at the same time, we can understand what is being said at that time. The information is not recorded in written form, however, so if we look back at it later, or if we are shown just the materials without any explanation, then we might not know what it means. I expect that quite a few of us have had that experience, and in this sense, I find this *Honda Technical Review* fulfills an extremely significant function.

On this occasion, I'd like to convey the great joy I feel in having been involved in editing *the Honda Technical Review*. At the same time, I sincerely hope that *Honda Technical Review* will continue being the wonderful form of help in passing on Honda technology and knowledge that it is, and in giving rise to the creation of new things.

(Baba)

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