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Technical Review



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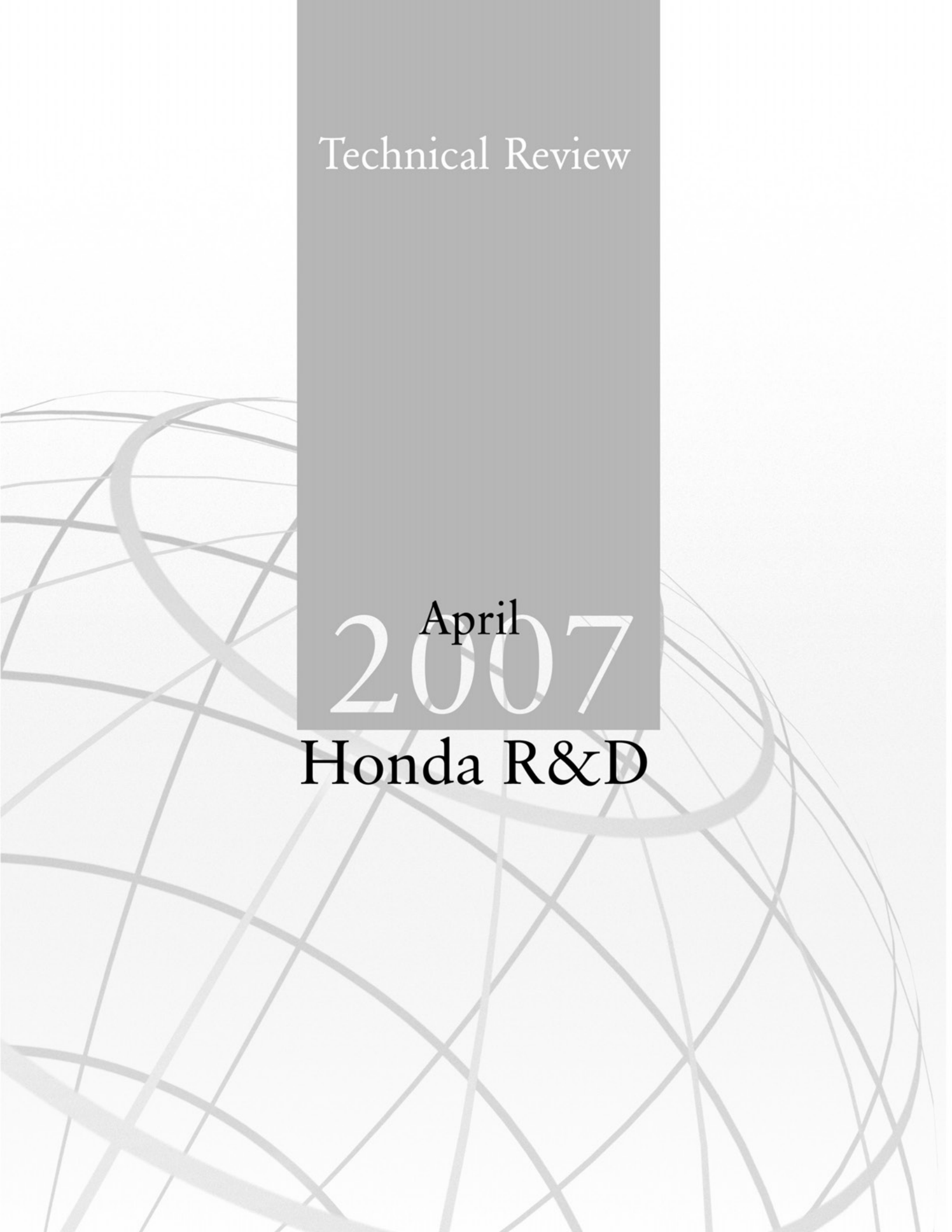
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Style and Design for New CR-V

Yoshiki HONMA*

Shoji FUJIMAKI**

ABSTRACT

The CR-V has been redesigned under a new concept—"Premium Smart SUV"—combining the toughness of an SUV with the quality and sophistication of a passenger car, all without sacrificing the functionality of the classic CR-V.

To achieve this, we have drastically redesigned the exterior of the new CR-V based on the concept "Dynamic & Emotional," giving it a combination of dynamic sporty forward cabin and a rugged lower body that emphasizes large wheels. The concept for the interior of the redesigned CR-V has been given the name "Quality & Toughness," and like the exterior aims to provide the vehicle with a tough, functional and driver-friendly SUV design with the refined space of a safe, exhilarating passenger car.

By optimally merging these concepts to build the newly designed CR-V, we have created an SUV that is fresh and upscale.

1. Introduction

With the unveiling of the CR-V in 1995, the radical new concept of a "light off-road vehicle" was introduced to the market. The idea of combining the best of a passenger car with that of an off-road vehicle has proved popular with car owners, particularly among young families in Japan. The CR-V is now sold in more than 160 countries and is on its way to becoming a world-wide hit alongside the CIVIC and ACCORD.

By conducting surveys in the United States (currently our main market) prior to developing the new CR-V, we found that owners of previous CR-V models were very fond of the vehicle's functional features, and many CR-V owners were thoroughly satisfied with their choice of vehicle.

Additional surveys were conducted, including customers of other vehicles to understand how drivers actually used their CR-Vs, how they look while driving in the city or on the freeway and other concerns. We were able to get a real sense of what drivers might want or need in a new SUV designed to change with the times. This lead us to take the next step to enhance the usability and functionality our CR-V owners have come to expect since the first generation rolled on to the showroom floor and to create a design that portrays the enhanced strength and

dynamic performance of an SUV.

Furthermore, while it is an SUV, the new CR-V also possesses an emotional dimension equal to that of any high-quality sedan. It is the high-dimensional fusion of these conflicting elements that is the real value of this new CR-V; the realization of the "Premium Smart SUV" concept.

2. Design Goals

We aimed to develop a next-generation compact SUV that incorporates the quality of a passenger car, while maintaining at its core the toughness of an SUV. With the overall concept being a "Premium Smart SUV," our goal was to design a CR-V that appeared active and sleek in any situation, portraying the driver as adults who think outside the box.

The exterior design concept of "Dynamic & Emotional" made this possible by bringing together the sporty, sleek cabin and the sturdy, dynamic lower body.

For the interior design of the new CR-V, we turned to the theme of "Quality & Toughness". We created a simple, high-quality, comfortable design built around a hardy image created by using an exposed frame and a functional design configuration that expresses an air of safety.

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3. Package Layout

The fundamental idea behind the new CR-V package was to begin with the previous model's compact body size and spacious, functional cabin, which have proved popular with drivers worldwide. We then added to the new CR-V's "DNA" such basic dimensions as an eye point that provides a feeling of reassurance and a hip point that makes it easier to get in and out of the vehicle.

Inside, there is ample room for four adults and plenty of space for their luggage. For the passenger car dimension, we designed a stylish space with an exhilarating and high-quality, emotional forward cabin. For the SUV dimension, we brought together the iconic large wheels and tough lower body, resulting in a new SUV frame that embodies the concept "Premium Smart SUV." (Fig. 1)

To make it more driver-friendly, we put a great deal of thought into the tiniest of details. The new CR-V adds such items as a double-deck cargo shelf that makes it easy to adjust the luggage space, a tailgate with reduced torque, and lower side door steps that make it more comfortable when getting in and out of the vehicle. These features realize the totally out-of-the-ordinary in your everyday life, with sleeker functionality, emotionalism and dynamic sturdiness all in one effective package.

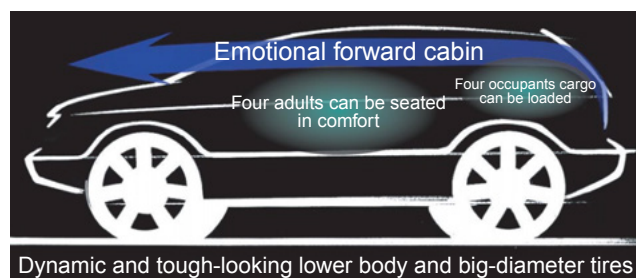


Fig. 1 Package concept

4. Exterior Design

We created the exterior of the new CR-V as an urban, refined "Dynamic & Emotional" design, without sacrificing the functional quality of the package of previous CR-V models.

By changing the way the tailgate opens and closes, we were able to give the new CR-V ample storage space. We also tilted the rear window, and this, combined with bringing the windscreen forward, resulted in a sporty, dynamic look for the cabin. We have done away with the quarter pillar and given the CR-V dynamic and personal one-motion side window graphics with added premium chrome-plated trim. (Fig. 2)

In addition to the increased presence created by using

larger wheels, the supple, large projecting front and rear fenders serve to emphasize the strength of the lower body. Together with the tough-looking lower garnish, these wheels and fenders give the new CR-V a sense of stability. (Fig. 3)

In the new design, we gave the front of the vehicle a look that makes it instantly recognizable as a CR-V. We accomplished this by fusing the upper and lower halves of the grill with the large headlight array, which results in an impression of sturdy strength.

For the back of the vehicle, we continued the use of vertical combination lights — the mark of a classic CR-V. We made the rear appear rugged with a three-dimensional protrusion symbolizing a rear spare tire, giving the new CR-V its unique rear design. (Fig. 4)



Fig. 2 Side view



Fig. 3 Dynamic emotional



Fig. 4 Final rendering

5. Interior Design

With the enhanced off-road rugged functionality of an SUV and the urban sophistication of a passenger car, we aimed to create an interior design that was upscale and suitable for any situation through the concept of “Quality & Toughness.”

5.1. Quality & Toughness

The core components of the interior design incorporate a structural motif that creates the sense of toughness and hardness that are the mark of an SUV.

A pillar-like solidity is created by the center instrument panel, that includes such controls as the audio components, air conditioning and gear shift, and defrosters and vents on each side. The main components, including the steering wheel, door grip and front seat-backs, also have been designed using this motif. (Fig. 5)

On the Quality side, previously “hard” components were given a feel of quality and softness, thereby merging rugged functionality and high-class sophistication. We were able to achieve this interior concept of “Quality & Toughness” through user-friendly features, such as the double-deck cargo shelf. (Fig. 6)



Fig. 5 QUALITY TOUGHNESS

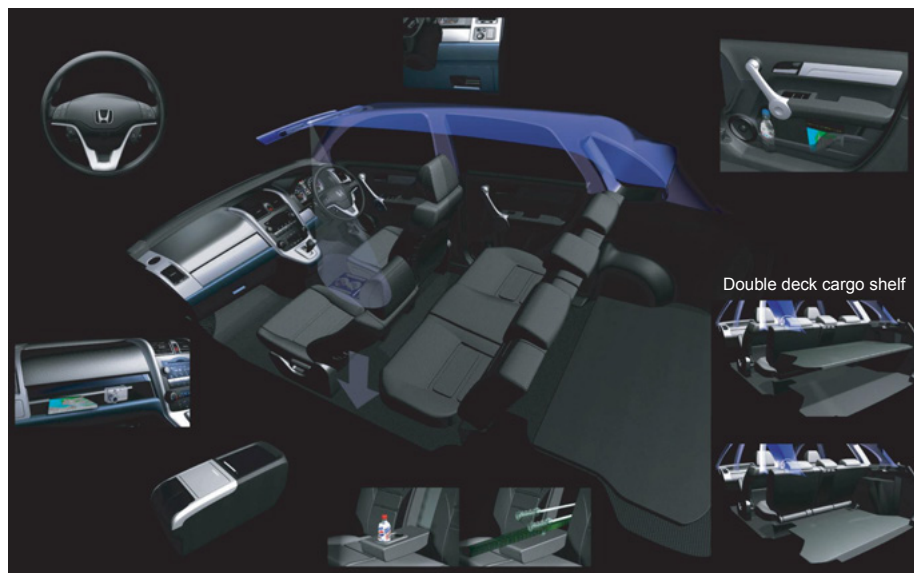


Fig. 6 Interior utility

6. Color Design

6.1. Exterior

To create colors that reflect our overall concept of a “Premium Smart SUV,” we developed the Premium Smart Color palette which makes the CR-V appear stylish in any situation.

To effectively convey the tough image and rugged surface exterior of an SUV, we developed Whistler Silver M especially for the new CR-V. This color contains metallic particles much larger than normal, giving it its dynamic sophistication.

In addition to the new color, the new CR-V will be available in five existing colors. (Fig. 7)

6.2. Interior

Developed with the same theme as that of the exterior, we aimed to combine the functionality of an SUV with the distinction of a passenger car in the interior of the new CR-V.

For the seats we used embossed tricote fabric, coordinating the interior with our new Indigo Black — the

interior theme color (Fig. 8a). The unique color presents an image of active functionality and urban sophistication, resulting in a cabin space that inspires a feeling of cool and confidence.

The CR-V will also be available with Ivory leather which creates a comfortable, relaxing space that projects a high-quality premium interior. We chose these two color variations as the expression of “Premium Smart.” (Fig. 8b)

7. Conclusion

The members of the CR-V design team have worked together to bring you this “Premium Smart SUV,” based on the overall concept of a vehicle that can be used for everyday driving or out-of-the-ordinary adventures. The new CR-V portrays the drivers as adults who think outside the box, no matter the situation. Despite the seemingly contradicting elements, we have made a higher quality CR-V that is sleeker looking and more emotionally satisfying and functional.

As we brought the new CR-V to fruition, the

Image not displayed for copyright reasons.

Fig. 7 Exterior color

BLACK& INDIGO BLACK



Fig. 8 (a) Interior color

BLACK & IVORY



Fig. 8 (b) Interior color

development team members discussed a variety of ideas and had many disagreements as they worked toward their goal. We can say with confidence that this has resulted in an appealing third-generation CR-V that is better designed and has enhanced driving performance, and of course has made significant advancements in usability. We hope that drivers will appreciate the full extent of the attractiveness of the new CR-V, and will feel satisfaction with the sense of excitement we have brought to this CR-V.

We hope this new CR-V, like its predecessors, will be used in a variety of ways, making drivers happy around the world.

■ Author ■



Yoshiki HONMA



Shoji FUJIMAKI

Styling and Design of New ASIMO

Takeshi KOSHIISHI*

ABSTRACT

The new ASIMO has been developed to enhance the robot's ability to function autonomously in real-life environments (offices, etc.) and its physical capability. The design concept for the new model is "Advanced functions for increased usefulness." The orientation of the development was decided while comparing different prototypes constructed to focus on "advancement" and "friendliness" respectively with the current ASIMO. Designers and engineers collaborated from the initial stages of development in design work using three-dimensional data. The fine-tuning ASIMO's dynamic status using computer graphics (CG), and the robot's presence using mockups in actual spaces, have enabled the achievement of a superb "harmony" between design and mechanical features. The resulting design enables the incorporation of new functions while retaining ASIMO's "friendly" image.

1. Introduction

Honda has long engaged in research on humanoid robots, which present new values by functioning usefully in society through coexistence and cooperation with people, in its attempt to innovate forms of mobility. The result of this effort was the arrival of ASIMO in late October 2000^{(1), (2)}, and since then Honda has provided venues for people in Japan and worldwide to interact with this robot through various activities.

The new ASIMO, announced in 2005, was developed to enhance the robot's ability to work autonomously in real-life environments, such as offices, and to enhance its physical capabilities⁽³⁾. The new ASIMO contains sensors in its lower torso, waist, the front and back of the head, and its ears to recognize its surrounding environment and the positions of people. New joints have been added in its waist, wrists, thumbs, and neck to enhance its physical capabilities and expressiveness, and the elbows have a larger range of motion. In addition to these new mechanisms, designers were asked to make its form more beautiful by reconciling major changes such as increasing the height of the robot to enhance its presence in real-life environments. This article introduces the new ASIMO's design concept and process.

2. ASIMO Design Concept

A humanoid robot, one with a human-like form, is unlikely to receive sympathy from people if its design is too radical. It is also unlikely to gain acceptance on the basis of familiar looks alone. Ideally, the robot should strike a fine balance between the qualities of being "advanced" and "friendly," and it should keep "a comfortable distance" from the human form, not being too close or too far. Accordingly, the basic concept of ASIMO's design is to harmonize its "advanced" and "friendly" qualities.

The design of the earlier ASIMO was based on the concept of being "more familiar," so that humanoid robots could start to come out of the laboratory, step into the world, and be accepted as a new industrial product by many people.

If the new ASIMO design were to change this image, people would miss the previous ASIMO. This is the issue in changing a humanoid robot model.

The new ASIMO was designed for realizing new functions on the concept of "advancing functions that make the robot more useful around people" while retaining the image of the previous ASIMO.

Figure 1 shows an overview of ASIMO's evolution and design.

* Fundamental Technology Research Center

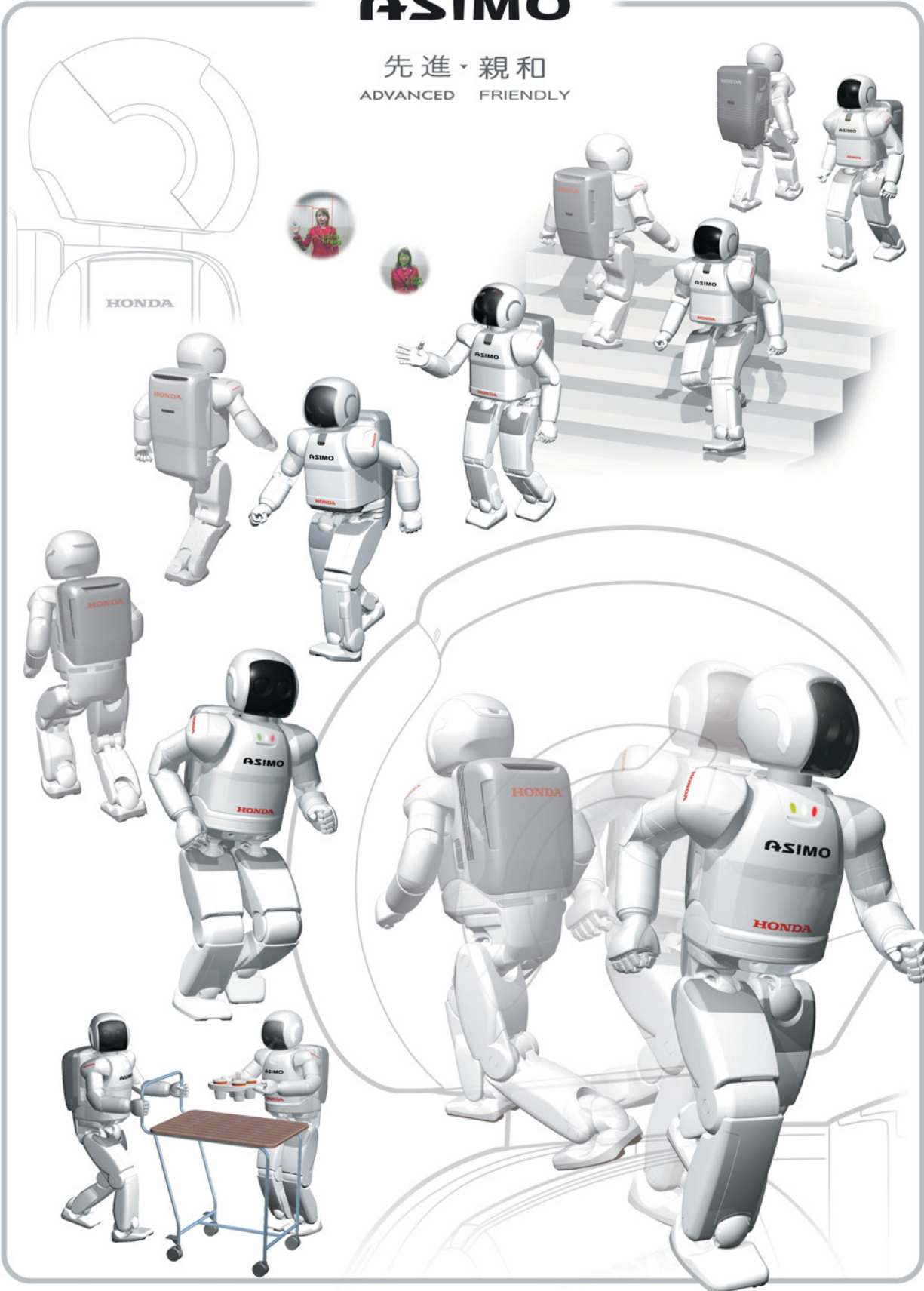


Fig. 1 Evolution of styling and design of ASIMC

3. Design Process

3.1. Design Techniques

Many unique humanoid issues are involved in achieving a humanoid robot, such as safety considerations in light of the many moving parts and avoidance of a sense of incongruity on appearance when the robot moves. Function is directly related to the form, a fact that necessitated the optimal placement of sensors and the addition of joints to increase the range of motion. Design plays a critical role in solving these issues, so designers and engineers worked closely together from the earliest stages of development and did design work with a common tool, 3D data.

Though there are conflicts between designers who seek aesthetic form and engineers who pursue innovative technology, the use of 3D data helped the two to share ideas concretely and mutually to enhance their work. In addition, because the designers were able to operate the 3D software directly, they could focus on the robot's aesthetics, not only in its static status but also its dynamic status, and could simultaneously work with each line and surface. Repeatedly making and unmaking the robot, the designers were able to create a sophisticated design. Designers ran highly precise simulations of the completed image through computer graphics (CG) rendering and animation (Fig. 2). The current computer technology, however, is still unable to test for the sense of presence a robot has when it is in the same room with people, so mockups were produced to make up for this. By putting a mockup in proximity to people, it can be checked how the robot will be viewed and sensed in these situations.

3.2. Goals of Design Study

Both CG and mockups were used to conduct design studies whose goal was to generate ideas for new design possibilities and new mechanism ideas such as equipping the robot with sensors and increasing the number of joints.

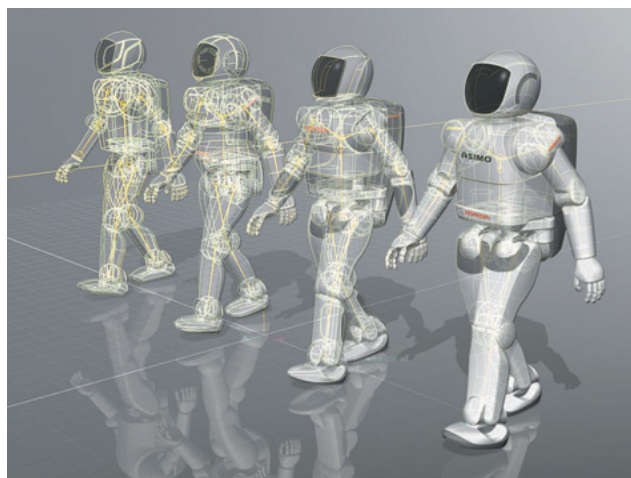


Fig. 2 Rendering and animation by computer graphics (CG)

Figure 3 illustrates the design studies in relation to the previous ASIMO. Taking the previous ASIMO with its “harmonization” of “advanced” and “friendly” qualities as the starting point, the balance was shifted and two design studies were conducted for comparison purposes: Design Study A, which focused on the “advanced” look (Fig. 3 (a)), and Design Study B, with an emphasis on the “friendly” look (Fig. 3 (b)).

The goals of each study were as follows.

(1) Goal of Design Study A

The goal was to express a humanoid robot with high functionality by focusing on an advanced image. The form of the head was based on a full face helmet, with sensors on the surface and illuminated ears and mouth. Joints were made spherical so that there would be little gap in the outer form during movement. Flexible material was used to enhance safety while maximizing the range of motion.

(2) Goal of Design Study B

The goal was to give the robot a face to focus on friendliness and enhance the sense of communication. Joints were added to the neck, elbows, wrists and waist to increase the angle of available motion, which enabled the robot to make various motions.

3.3. Design Clinic Results

Design clinics were held to obtain the opinions of non-designated subjects and thereby determine how well the goals of the two design proposals would be conveyed to people and how people would understand the relationship with the previous ASIMO. Mockups of the previous ASIMO (Fig. 4), Design Study A (Fig. 5), and Design Study B (Fig. 6) were presented, as were CG animations of each, and subjects responded in questionnaires and interviews as follows.

(1) The previous ASIMO

The previous ASIMO was likable, with a cute and friendly image. Subjects looked forward to seeing how this ASIMO would turn out in the future. They felt that it could be evolved by enhancing individual parts, such as the heavy-looking back.

(2) Design Study A

The impressions of the robot were that of a high performance robot that would get its work done steadily; that it seemed futuristic but not warm; and that it looked like a completely different humanoid robot, with little connection to the previous ASIMO. The new mechanisms seemed a little incongruous.

(3) Design Study B

This robot conveyed a higher sense of communication. However, its expression seemed rather simple. This robot could be teamed together with the previous ASIMO



Fig. 3 Relation of design studies with previous ASIMO [(a) : advanced design, (b) : friendly design]



Fig. 4 Previous ASIMO (Mockup)

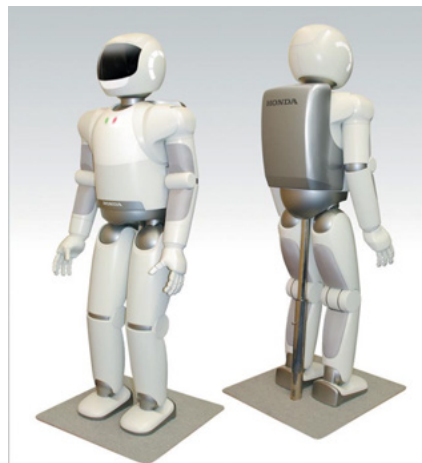


Fig. 5 Design study A (Mockup)



Fig. 6 Design study B (Mockup)

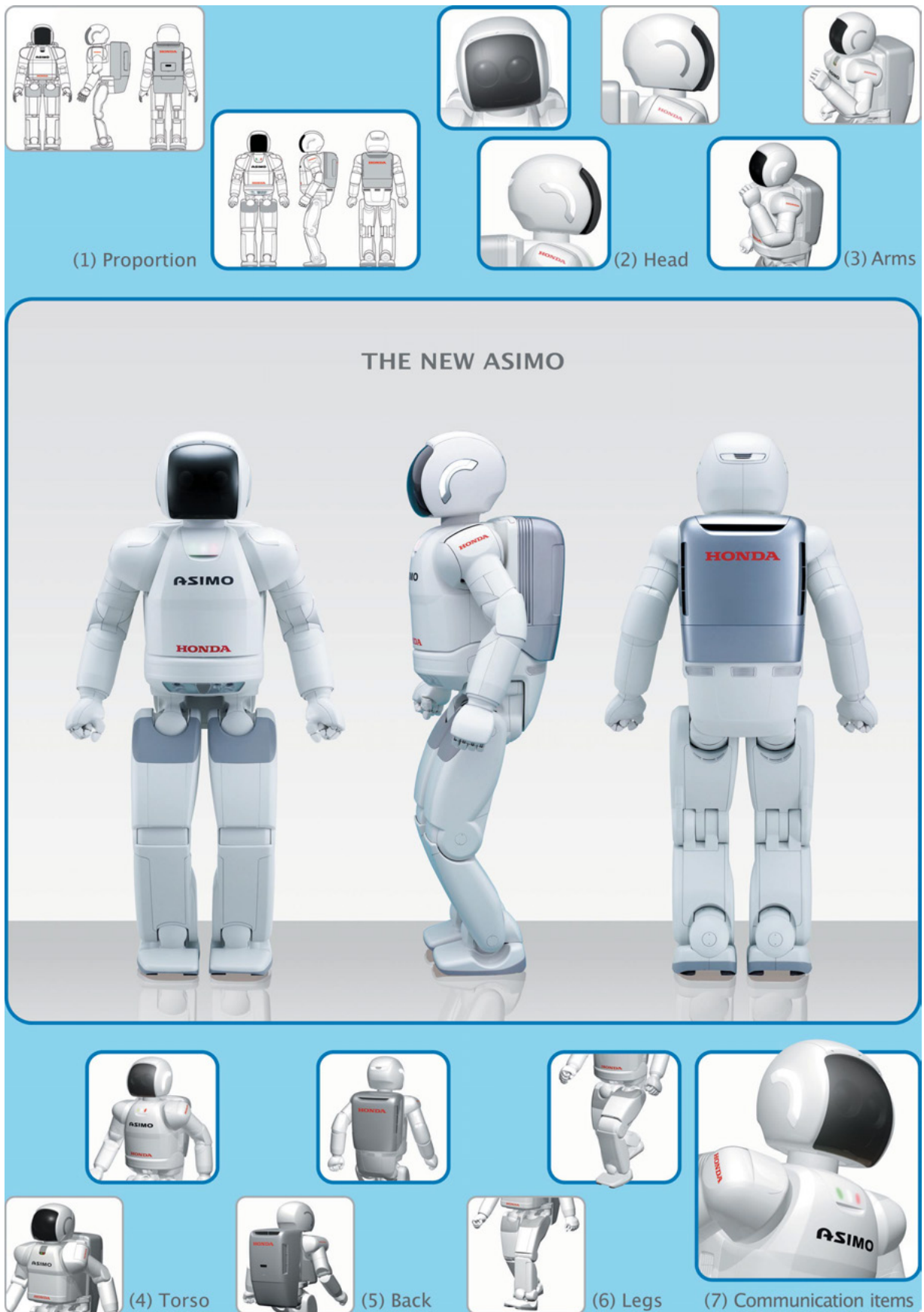


Fig. 7 Overall final design and design of parts (Blue frame : new ASIMO ; Gray frame : previous ASIMO)

without seeming out of place, but it seemed like just a variation of the original, e.g., like a brother. The new mechanisms did not seem incongruous.

In summary, the subjects' opinions indicate that they understood the designs as the designers intended, and that the functional simple design of the previous ASIMO had received the sympathy of the subjects.

3.4. Deciding on the Final Design

The new ASIMO was designed with respect to the following three perspectives by incorporating the new mechanisms investigated in the design studies while using the image of the previous ASIMO.

1. The new ASIMO would express "ease of communication" by using LEDs for illumination and integrating a head tilting motion into the design.
2. It would express "gentleness" with rounded surfaces and crisp lines basically.
3. It would express "good motion" with a functional and strong form.

Figure 7 shows a comparison of the previous ASIMO and the final design of the new ASIMO. Details are shown below.

(1) Proportion

The height of the previous ASIMO was 120 cm, which was considered to be the minimum size to be useful when occupying space with a human, and it had a six head-length body proportion, similar to that of a child of about the same height. The new ASIMO is 130 cm tall and has an approximately 6.3 head-length body proportion to accommodate the new mechanisms and to enhance its presence. The new design also attempted to enhance the robot's posture by changing the locations of the axes of the neck and arms. It also expressed "gentleness" with rounded surfaces and crisp lines all over the body so that it could be active in proximity to people.

(2) Head

Because the head has a particularly strong impact on the overall image, several proposals were developed before the following design was finally chosen.

The size deemed appropriate at first on the 3D data appeared smaller than expected in the mockup form, resulting in an impression of less "friendliness." Therefore the size was determined after adjusting the balance.

The head was rounded overall to increase its "friendliness," and sensors were built into the front, back, and ear areas. A joint was added to the neck, enabling the robot to better express itself by tilting the head. To do this without touching the chest or back, the lower surface was rounded. To express the image of "running," the angle of the top of the head and lines on the sides were used to recall hair streaming in the wind.

The eyes are important in designing humanoid robots. A series of lenses were tested so that a modest amount of light would reflect, projecting "pupils" that would make the robot look friendlier.

(3) Arms

The shoulders were made compact and nearly spherical to minimize gaps in the outer form when the arms moved. Due to the increased angle of available motions of the elbows, a soft material was placed inside, and the outside was formed to reduce gaps during movement. Whereas the previous ASIMO had cylindrical wrists, the new model has spherical wrists to accommodate the increasing degrees of freedom. Each finger was modified rounder to give a gentle touch when shaking hands.

(4) Torso

The torso of the new ASIMO has a rotating mechanism to enhance its physical capabilities. As a result, an angular torso like that of the previous ASIMO would display an unevenness of angles when turning, so the rounded form studied in Design Study B, having little unevenness between the upper and lower parts when turning, was used as a basis. Sensors for recognizing the surrounding environment were arranged in the lower part to have a functional look to demonstrate this performance.

(5) Back

Due to the changes to the internal structure and a more compact battery, the back of the robot was modified smaller to enhance a sense of unity with the torso. Moreover, the structure was made to be a single piece with the changeable battery, and a handle for battery removal and installation and an operating switch panel were built into the upper lid for ease of use.

The waist was made into a separate component by making the back smaller and giving the torso a rotating mechanism. This is a part that is highly different from the previous ASIMO. Repeated studies were performed not to generate any sense of incongruity and the waist was designed to be functional with consideration given to the placement of sensors and insertion into the back of the legs.

(6) Legs

At first the legs were designed to be the same length as those of the previous ASIMO, and then the proportions were modified by making the thighs and shins longer so that the torso would not appear too long. The bumps of the knees were reduced to enhance safety, and the hem was flared to express strong legs that are ready to "run."

(7) Communication items

Humans associate lights and the way in which things are illuminated with a sense of life. To make effective use of this association, the new ASIMO's ears and chest are

illuminated. The ears clearly illuminate, but the chest was given an illumination design tested in Design Study A: when illuminated, the chest shines gently, projecting a sense of having a core, and when not illuminated, it reduces its sense of presence.

4. Conclusion

The process of developing a design for the new ASIMO was designed to achieve a new level of “harmony.” This endeavor included design clinics with models focusing on, respectively, “advanced” and “friendly” qualities, and the process furthermore sought out points of agreement between designers and engineers, who shared 3D data from the beginning of the development and enhanced each other’s ideas. At the same time, the design considered both the robot’s static and dynamic status, and expressed “ease of communication,” “gentleness,” and “good motion.” As a result, the robot was designed to present the concept, “advancing those functions that make the robot more useful around people” while retaining the “cute and friendly” image of the previous ASIMO. This design was intended to receive sympathy from people.

5. Closing

Humanoid robots continue to inspire hopes and interests among people. And the dream spreads endlessly. The drawings that children around the world send to Honda, and the fascination with which people look at ASIMO, are testament to this. While it is easy to dream up a robot in the imagination, actually building one has many challenging issues. However, with its goal of building a robot to coexist with humans, Honda is committed to achieving this goal one step at a time.

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■ Author ■



Takeshi KOSHIISHI

Development of 1.8L Flexible Fuel Vehicle System for 2007 Model Year CIVIC for Brazil

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Hideyuki OKI*

Yoichi GORYOZONO*

ABSTRACT

Honda has developed a 1.8L VTEC 4-cylinder engine for a Flexible Fuel Vehicle (FFV) that meets the needs of the Brazilian market. This engine has a function that enables it to be run on either gasoline or ethanol, or on both fuels mixed in any ratio. Engine specifications that are suitable for Brazilian fuel were selected by optimizing the valve timing using VTEC and changing the compression ratio. For the fuel system parts, materials with ethanol corrosion resistance were chosen and applied to the Brazilian market. In this paper, the performance achieved by this engine and the applied technology are described.

1. Introduction

There has been growing demand in recent years for reduced CO₂ emissions to address the issue of global warming. Consequently, the use of biofuels, which are renewable, is capturing increasing attention. As alternatives to gasoline, there are bio-ethanol fuels made from sugar cane, corn, etc., and as alternatives to diesel, there are bio-diesel fuels made from soybean, rapeseed, and so on. Raw material for the bio-ethanol used in Brazil and some other places comes from sugar cane or similar plant life. These plants absorb CO₂ as they grow. The CO₂ emission that results from their use as motor vehicle fuel, therefore, is treated as zero. Biofuels are gaining attention as an alternative to petroleum, and at the same time, as a countermeasure against global warming.

The volume of sugar cane produced in Brazil is high, and there are numerous bio-ethanol manufacturing plants. National policy has, so far, dictated the sale of bio-ethanol fuels. The gasoline that is sold in Brazil needs to have a mixture of 20–25% ethanol (hereafter referred to as gasohol) but fuel composed only of ethanol (hereafter referred to as E 100) is also for sale.

Honda has been selling the CIVIC GX^{(1), (2), (3)} since 1998, which is an alternative fuel vehicle running on natural gas. The new model CIVIC⁽⁴⁾ that went on sale in 2005 has been used as the base vehicle for a flex-fuel engine that was developed for the CIVIC FFV. This paper describes the technology adopted for the CIVIC FFV.

2. Development Goals

In Brazil, the CIVIC with this engine installed is a key model, and it sells at a level second to the Fit. The following development goals were defined in order to meet user needs as a vehicle capable of using both gasohol and ethanol, and to support the diversification of energy sources:

- (1) That there be flexibility in selecting the fuel to be used.
- (2) That the vehicle offers guaranteed driving performance that is not determined by the fuel used.
- (3) That the vehicle offers commercial product appeal in terms of refueling.

3. Fuel Properties

Table 1 shows the typical properties of the main fuels. The commonly distributed fuels in Brazil include gasohol and E 100. One general advantage of ethanol as a fuel for internal combustion engines is its high octane value (RON 110). This makes it possible to increase the compression ratio, so that the thermal efficiency can be enhanced. On the other hand, there are various issues such as the corrosion of fuel system parts, the deterioration of rubber and resin materials, and the reduction in fuel economy due to the lower heating value. Also, there are issues with low temperature starting and drivability because of the lower vapor pressure.

* Automobile R&D Center

Table 1 Fuel properties

	E0	E100
Heating value (MJ/Air kg)	2.95	2.99
Octane number RON/MON	93/81	110/92
Stoichiometric air fuel ratio	14.6	9.0
Latent heat of evaporation (kJ/Air kg)	30.2	101
Boiling point (°C)	25-200	78
Theoretical air fuel ratio (Volume after carburetion)	43.1	14.3
Ratio of the number of moles of combustion before and after	1.07	1.14

Table 2 Engine specifications

	2006 Gasoline engine	2007 FFV engine
Fuel	E20-E25	E20-E100
Cylinder configuration	In-line 4-cylinder	In-line 4-cylinder
Bore × stroke (mm)	81 × 87.3	81 × 87.3
Displacement (cm ³)	1799	1799
Compression ratio	10.5 : 1	11.5 : 1
Valve train	SOHC i-VTEC Inlet delayed closure	SOHC i-VTEC 1 intake valve inactive
Number of valves	4 per cylinder	4 per cylinder
Valve diameter (mm)	In.	32
	Ex.	26

4. Engine Overview and Main Specifications

4.1. Engine

Figure 1 shows the appearance of the R18A FFV engine. Table 2 shows the major specifications of this engine in comparison with the gasoline engine it was based upon. Various enhancements were made with respect to the 2006 model of the gasoline engine, including the following:

- (1) A higher compression ratio suited to the fuel properties was applied.
- (2) To ensure low temperature starting, sub fuel injection devices were added on the intake manifold, and the intake manifold material was changed to aluminum.
- (3) To enhance low temperature drivability, the valve timing was changed.
- (4) To prevent valve recession when using ethanol, the valve and valve seat materials were modified.

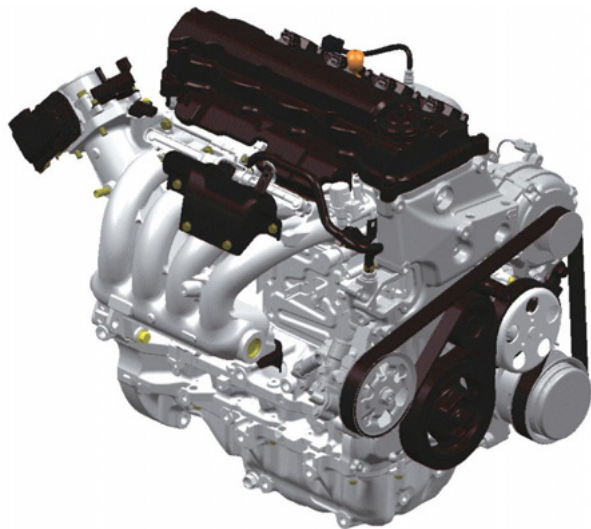


Fig. 1 R18A FFV engine

4.2. Fuel Supply System

Figure 2 shows the layout of the FFV fuel tank and fuel supply system. The fuel supply to the engine consists of the main fuel system, which supplies fuel from the main fuel tank, and the sub fuel system, which supplies fuel from the sub fuel tank. Figure 3 shows the main fuel

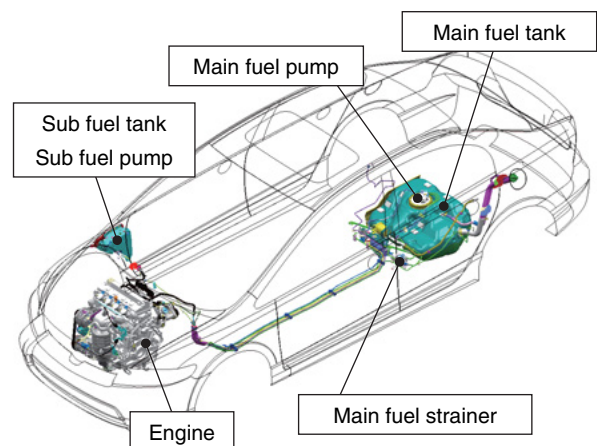


Fig. 2 Overview of CIVIC FFV fuel supply system

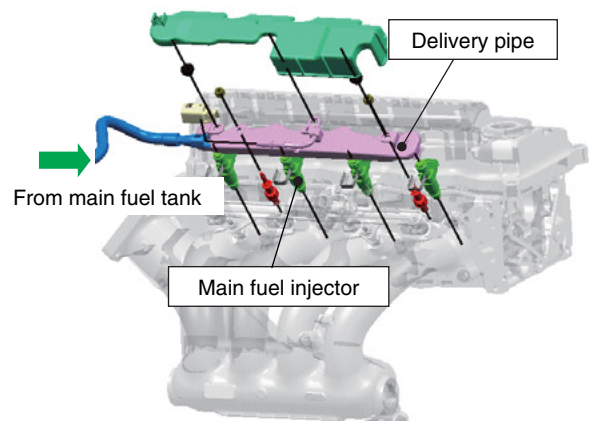


Fig. 3 Main fuel supply system

supply system. The delivery pipe material was changed to SUS 304 to enhance its resistance to corrosion by ethanol. This delivery pipe and the FFV main injector shown in Fig. 4 construct the system. Figure 5 shows the sub fuel system, which consists of the sub delivery pipe, sub fuel injector, and solenoid valve installed to assist with low temperature starting.

The fuel tank was adapted from the tank developed for the 2006 model gasoline vehicles. A fuel pump module specifically for use with E 100 was added. Although the conventional fuel strainer was integrated into the fuel pump module, it was changed to an external strainer at the fuel tank for easier maintenance and replacement.

An auxiliary fuel tank (hereafter referred to as the sub tank) for use in starting the engine is located under the right front fender. The sub tank fuel fill door was placed in the right fender to enhance product appeal when refueling. It is harder to start the engine at low temperatures when using E 100 or high-concentration ethanol as fuel. Therefore, the gasohol in the sub tank is supplied to the engine during startup to assist the startup process. The fuel in the sub tank is only used when

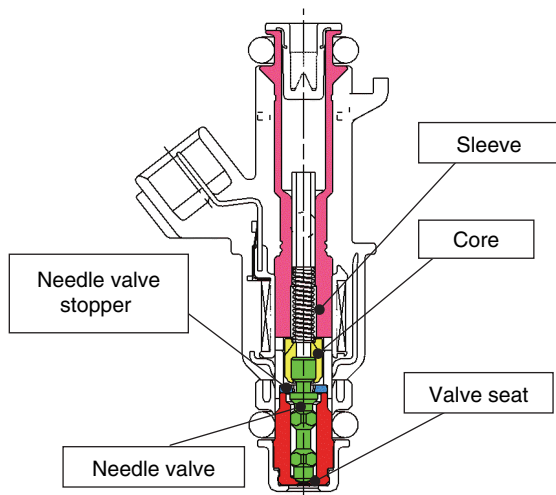


Fig. 4 Fuel injector for FFV

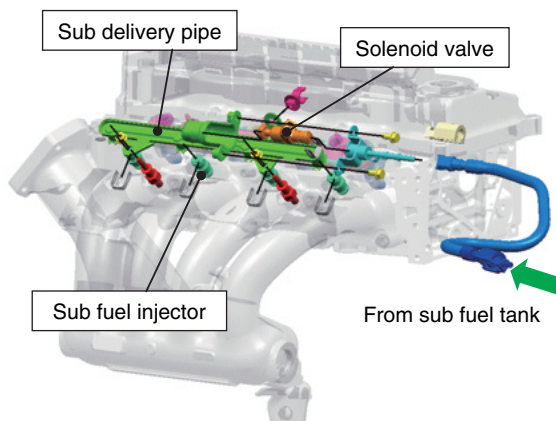


Fig. 5 Sub fuel supply system

Table 3 Material for main fuel system

Parts		Gasoline engine	FFV engine
Fuel injector	Valve seat	SUS440C	ATS34 Passivation film
	Needle valve	SUS440C	ATS34 Passivation film
	Needle valve stopper	SUS440C	SUS440C Passivation film
	Core	PB	K-M62F Passivation film
	Sleeve	K-M35FL	K-M62F
Delivery pipe		JSH270C Electroless Ni-P plating	SUS304L

starting the engine at low temperatures. The sub tank was made with a capacity of 700 cm³, a size that was chosen after taking into account the rate of fuel consumption and the frequency of refueling. The fuel pump module in the sub tank was adapted from modules used in gasoline vehicles.

4.2.1. Main fuel supply system

The main fuel system has highly concentrated ethanol or E 100 supplied from the main fuel tank. Materials with enhanced corrosion and wear resistance were adopted for use in the delivery pipe and injectors.

As shown in Fig. 3, fuel from the main fuel tank is supplied to each cylinder through the delivery pipe and injectors. The material for the delivery pipe was changed to SUS 304, which enhances its resistance to corrosion from ethanol.

Figure 4 shows the structure of the main injector. These differ from the injectors for use with gasoline in that different materials were used in each part and efforts were made to enhance resistance to corrosion and wear. Table 3 presents a comparison of materials used for the main injectors and delivery pipe in this engine and in a gasoline engine.

4.2.2. Sub fuel supply system

Low temperature starting performance declines when using highly concentrated ethanol. Therefore, gasohol is supplied from the sub tank to assist during engine startup.

Figure 5 shows the sub fuel system used for starting that is attached to the intake manifold. Figure 6 shows the cross-section of a sub fuel injector installed on the intake manifold. The system consists of the sub fuel injectors, the sub delivery pipe, and the solenoid valve installed on the aluminum intake manifold. Gasohol is supplied from the sub fuel tank when the engine is using high concentrations of ethanol and is doing a low-temperature start. The sub delivery pipe is filled up with

gasohol through the solenoid valve, and the sub fuel injectors installed in each cylinder supply the fuel to the cylinders. The mechanism is configured so that when it determines that the engine has started, it will stop the supply of fuel from the sub tank and close the solenoid valve, thus cutting that fuel off.

When the engine is starting in a warm condition or the fuel in the main fuel tank is gasohol, the engine can be started with the main supply of fuel alone. When this is the case, the sub fuel is not supplied to the engine.

4.3. Output characteristics

4.3.1. Characteristics during combustion of ethanol

A comparison of gasoline with ethanol, when used as an internal combustion engine fuel, is shown in Table 1.

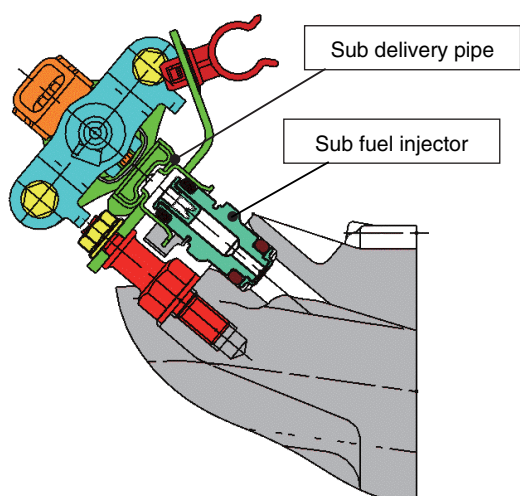


Fig. 6 Side view of sub fuel injector

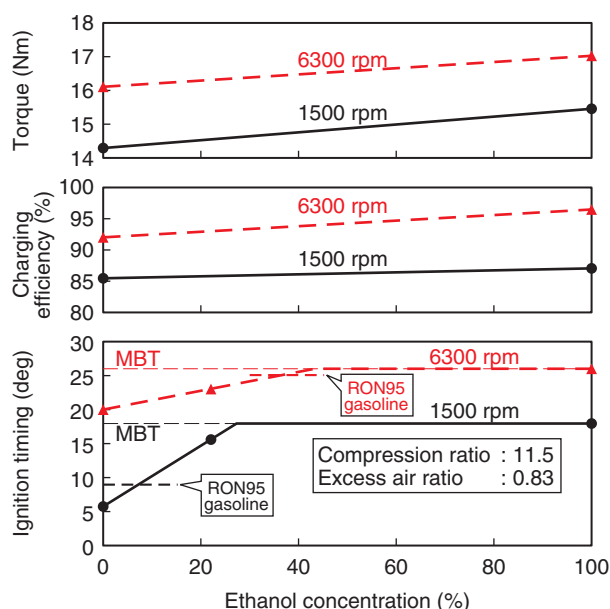


Fig. 7 Relation of ethanol concentration to ignition timing, charging efficiency and engine torque

Figure 7 shows the relationship of ethanol concentration to the setting of ignition timing, charging efficiency, and engine torque. Test results confirmed the following points:

- (1) There is virtually no difference in the MBT at wide open throttle.
- (2) RON enhancement makes it possible to advance the ignition timing (effect is greater at low speeds).
- (3) The increase in latent heat evaporation yields an increase in charging efficiency (effect is greater at high speeds)
- (4) Greater power is achieved by advancing the ignition timing and enhancing charging efficiency (effect is greater at low speeds).

It was confirmed that increasing the concentration of ethanol raises the RON and makes it possible to advance the ignition timing. E 22 has the lowest RON in the Brazilian market, and it can be inferred to have a RON higher than RON 95 gasoline at low speeds. At high speeds, however, it can be inferred to have a lower RON than RON 95 gasoline. The enhancement in charging efficiency from latent heat evaporation is thought to have a greater effect at higher speeds, where the volume of fuel injected per unit of time is greater.

Figure 8 shows the relationship between ethanol and gasoline under the same conditions for cylinder pressure and temperature in the high speed range. Even though cylinder pressure rises, the increase in the number of moles following combustion results in a decrease in cylinder temperature. Table 4 shows the comparison of output power, maximum cylinder pressure, and maximum cylinder temperature between ethanol and gasoline operations. The power is approximately 4% greater, but the maximum cylinder temperature is lower.

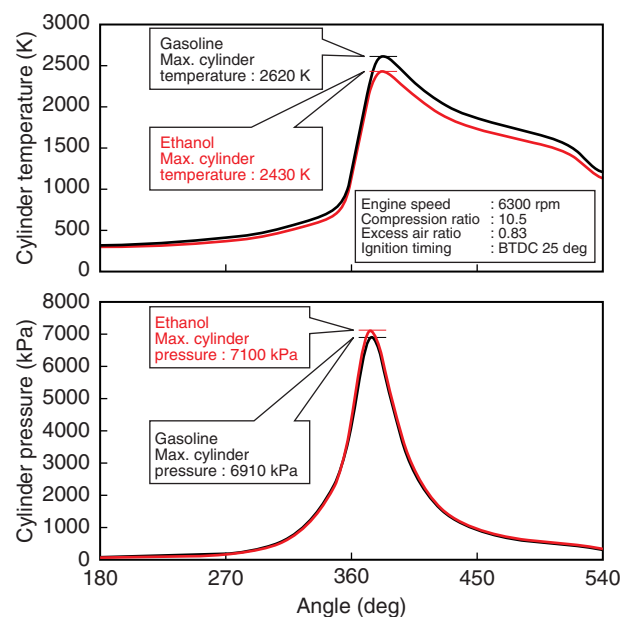


Fig. 8 Comparison of cylinder pressure and temperature for ethanol and gasoline

Table 4 Comparison of engine power for ethanol and gasoline

	Ethanol	Gasoline
Output power (kW)	107.6	103.6
Max. cylinder pressure (kPa)	7100	6910
Max. cylinder temperature (K)	2430	2620

Table 5 Engine power

	Ethanol	Ethanol 22% Gasoline 78%
Max. power (kW/rpm)	103/6200	102/6200
Max. torque (Nm/rpm)	174/4300	172/5000

4.3.2. Engine performance at wide open throttle

The characteristics of this engine are as follows:

- (1) Greater charging efficiency in the low speed range, as well as higher flow velocity, yielding operation on a single intake valve with enhanced brake specific fuel consumption (BSFC) under conditions of low water temperature combustion and partial load
- (2) High power output operation on two intake valves in the high speed range
- (3) A compression ratio that takes into account vaporizability when using 100% ethanol and knocking properties at the lower limit of ethanol concentration

Figure 9 and Table 5 show the engine output at upper and lower limits of the ethanol concentration.

4.4. Engine Performance at Low Water Temperature

4.4.1. Characteristics of ethanol at low temperatures

Issues associated with use of ethanol as motor vehicle fuel include low saturated vapor pressure at low temperatures resulting in poorer startability. Gasoline contains components with low boiling points, such as butane, that provide adequate vapor pressure even at low temperatures. With ethanol, the vapor pressure needed for combustion at low temperatures can not be attained. Increasing the amount of injection is ineffective because ethanol does not contain other components with low

boiling points such as in gasoline. It is important to take these characteristics of ethanol into account and select specifications that are suited to the FFV.

The basic specifications that are suited to the FFV were examined. With charging efficiency taken into account, the pressure and temperature after compression can be shown as follows:

$$P_C = P_0 (\eta_v \varepsilon)^\kappa \quad (1)$$

$$T_C = T_0 (\eta_v \varepsilon)^\kappa \quad (2)$$

P_C : Cylinder pressure after compression

P_0 : Intake manifold pressure

T_C : Cylinder temperature after compression

T_0 : Intake air temperature

η_v : Charging efficiency

ε : Compression ratio

κ : Specific heat ratio

The saturated vapor pressure rises sharply relative to temperature, with these results:

- (1) Increased charging efficiency (low speeds)
- (2) Higher compression ratio

These raise the cylinder temperature, increasing the vaporizability of the ethanol under low ambient temperature conditions. Putting one intake valve at rest is also an effective method of raising the flow velocity of the air-fuel mixture and increasing the vaporizability of ethanol, because of increasing contact by the ethanol with the intake air and cylinder wall.

The compression ratio for this engine was determined to accommodate different ethanol contents ranging from 20 to 100%. A VTEC mechanism was adopted to operate the valve timing, which determines charging efficiency, so that the system could switch back and forth between:

- (1) Operation with one intake valve, for starting the engine and for low speeds
- (2) Operation with two intake valves, for high speeds

This has achieved both driveability and adequate top power output.

4.4.2. Combustion characteristics of ethanol at low temperatures

The second issue arising from the use of ethanol as motor vehicle fuel includes starting the engine at low ambient temperatures and high-load operation (acceleration), when making a standing start immediately after startup. Figure 10 shows the air-fuel ratio

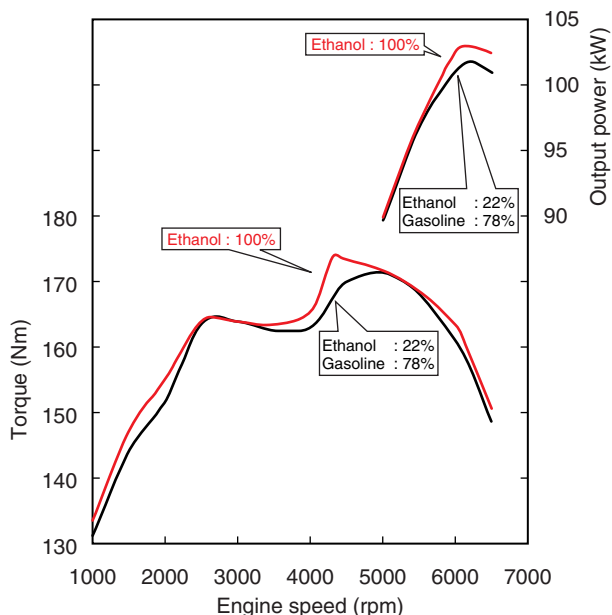


Fig. 9 Engine performance

Table 6 Comparison of MBT after cold start and after warm-up

Intake pressure (kPa)	MBT BTDC (deg)	
	Tw=80°C Ta=25°C	Tw=-10°C Ta=-10°C
-60	25	30
-13.3	10	25

characteristics with low intake air temperatures and low water temperature. The intake pressure of -60 kPa represents the idling state immediately after startup, while the intake pressure of -13 kPa represents the high-load state when starting. In high-load operation using ethanol, the total hydrocarbon concentration (THC) is high. This is thought to be because the intake air temperature and water temperature are low, so that the large amounts of ethanol involved in high-load operation cannot be vaporized. Ordinarily, the concentration of CO₂ is at a maximum with the theoretical air-fuel ratio. Ethanol that is not vaporized is not involved in combustion, however, so that λ reaches its maximum at around 0.9. Table 6 shows the relationship of MBT to intake pressure, water temperature, and intake air temperature. The MBT varied with the load, and the timing was advanced farther under higher loads. The reason is that high loads involve the injection of large amounts of fuel, and the resulting lower vaporizability means that ignition should be advanced in order to obtain good combustion.

4.4.3. Transient combustion characteristics

One of the challenging conditions for driving with

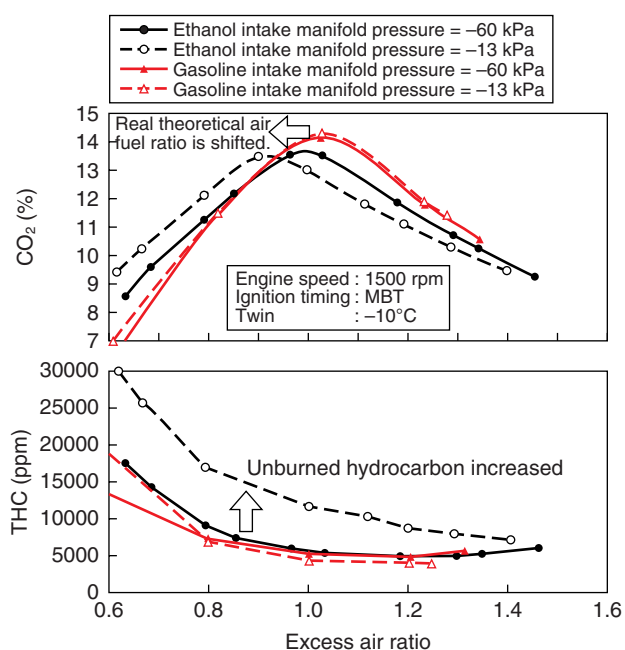


Fig. 10 Characteristics of excess air ratio at low temperature

ethanol as the fuel is when high-load operation follows immediately after engine start. The torque was measured at the transition to high-load operation to serve as an index for evaluation. Figure 11 shows the torque characteristics during the transition when partial load, immediately after startup, is suddenly changed to high-load operation. Figure 12 shows the torque characteristics of the steady state in the case of high-load operation immediately after the engine is started. When operating with two intake valves, misfires take place under high-load operation, even with a cooling water temperature up to 13 °C.

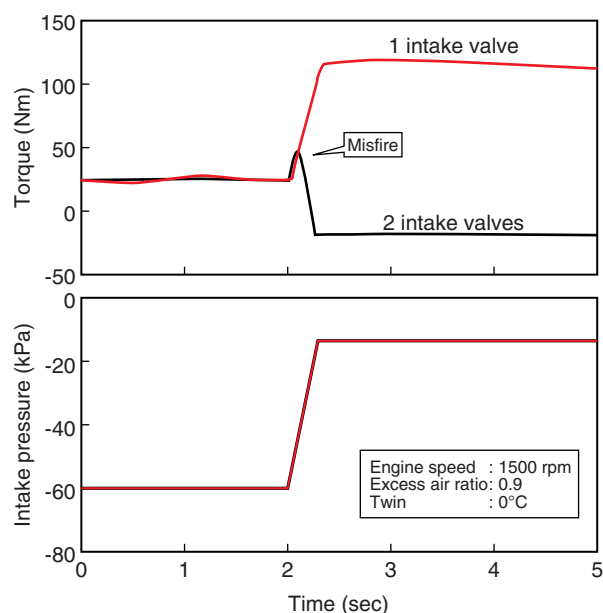


Fig. 11 Comparison of torque during acceleration between 1 intake valve and 2 intake valves

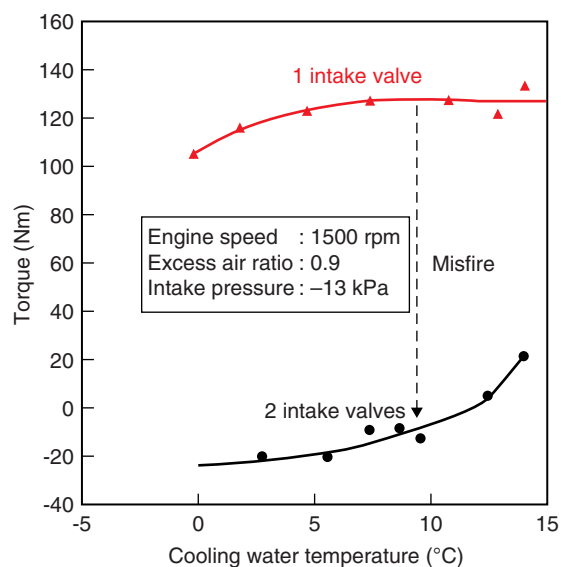


Fig. 12 Comparison of torque during acceleration between 1 intake valve and 2 intake valves

When driving with one intake valve, good combustion can be obtained, even with a cooling water temperature of 0 °C, which corresponds to the condition immediately after startup with the lowest ambient temperature experienced in Brazil. The reason is thought to be that the increased flow velocity and the generation of swirls increase the contact frequency of fuel, intake air, and cylinder wall. This then enhances vaporization.

5. Fuel Control

5.1. Main Fuel Control

The fuel control specifications were modified for the FFV according to the fuel properties as they differ from the properties of fuel used in a gasoline engine.

5.1.1. Ethanol concentration learning

As in the case of the gasoline engine, this engine uses a Universal Exhaust Gas Oxygen sensor (hereafter UEGO sensor) to operate with stoichiometric feedback (hereafter F/B). Ethanol and gasohol, however, have different theoretical air-fuel ratios, so that greater amounts of fuel are required when ethanol is used than when gasohol is used. The increase in the amount of required fuel is detected by the UEGO sensor, and the system then estimates the ethanol concentration of the fuel in the main tank.

The supply of fuel sometimes involves the supply of fuel that is different from the fuel presently in use. This means that the output of the fuel level sensor in the fuel tank is used to determine when fueling takes place. At that time, the system relearns the ethanol concentration.

5.1.2. Fuel compensation for ethanol concentration

Ethanol has a higher boiling point and lower vapor pressure than gasohol, and therefore does not vaporize as readily at low temperatures. Consequently, combustion at low water temperature requires a larger amount of fuel than is accounted for just by the difference between the theoretical air-fuel ratio of ethanol and that of gasohol. The system therefore selects a fuel volume increase factor for start-up that corresponds to the ethanol concentration learning value, and implements a wall-wetting fuel compensation control according to the ethanol concentration.

5.1.3. A/F F/B factor active limits

Virtually the only factor that influences the output of the UEGO sensor during stoichiometric operation of a gasoline engine is variation in the precision of the component parts. There is no need to set a wide margin for F/B. In addition, the FFV conducts stoichiometric F/B using fuels with a wide variation in their theoretical air-fuel ratio, so the F/B margin has to be set wide. However, simply setting a wide spread between upper and lower F/B limits of the gasoline vehicle may promote unnecessary

Table 7 Emission standard

Stage	CO	NMHC	(g/km)
			NOx
PROCONVE L-4	2.0	0.16	0.25
PROCONVE L-5	2.0	0.05	0.12

fuel F/B. Consequently, the initial values for the upper and lower limits are set as for a gasoline vehicle, and the system configured so that when the F/B factor reaches the limit value, it judges that a different fuel is now being used, and gradually widens the limits. This achieves a combination of both positive fuel F/B performance and fuel switchover learning performance.

5.2. Sub Fuel Control

When the ambient temperature is low and a high concentration of ethanol is being used, the startup time can be prolonged or startup may not occur. As a result, when the water temperature at startup does not rise above a set temperature and the ethanol concentration is above a set value, the sub fuel from the sub tank is injected into the inlet port to assist in engine startup. The system is configured to implement control in that manner, and thus provides good starting performance across a wide range of temperatures, regardless of the fuel being used.

6. Brazilian Exhaust Gas Regulations and Low Emission Technology

6.1. Brazilian Exhaust Gas Regulations

Table 7 shows the PROCONVE L-4 regulations that were implemented starting on January 1, 2005. It also shows the PROCONVE L-5 regulations, which are slated to be adopted from 2009, for reference. This engine complies with PROCONVE L-4. A distinctive requirement of legal statutes in Brazil is that they mandate satisfaction of the above regulation values for the three types of test fuel shown below.

- (1) AEHC (hydrated ethyl alcohol fuel)
- (2) E-22 gasoline (a mixture of 78% standard gasoline and 22% anhydrous ethanol) (v/v)
- (3) A compound of 50% standard E-22 gasoline and 50% standard AECH (v/v)

The FFV is capable of using any mixture of fuel. That is why this legal requirement was established.

6.2. Exhaust Gas Treatment System

Figure 13 shows the results achieved in meeting emissions requirements. An EGR valve was removed from the exhaust system for the base gasoline vehicle. Items (1)-(3) in the figure show the various test fuels mentioned above. When the concentration of ethanol is higher, larger amounts of fuel, relative to gasoline, are required when starting and immediately after starting the engine. This is why the volume of NMHC emissions tends to grow larger.

7. Afterword

The FFV engine developed in this project was mounted in the CIVIC to be sold in Brazil. Equipped with VTEC and sophisticated fuel control, the vehicle achieves both compatibility with diverse fuels and superior emission performance. The development goals that were initially set for the project have been met.

This development project has established basic technology for satisfying emissions controls, which will be made more stringent in the future, as well as for ethanol fuel mixtures, which are being adopted worldwide at an increasingly rapid pace. Additional enhancements will be incorporated in the future with the goal of developing a highly efficient power train that is capable of satisfying the demands of different regions and different times.

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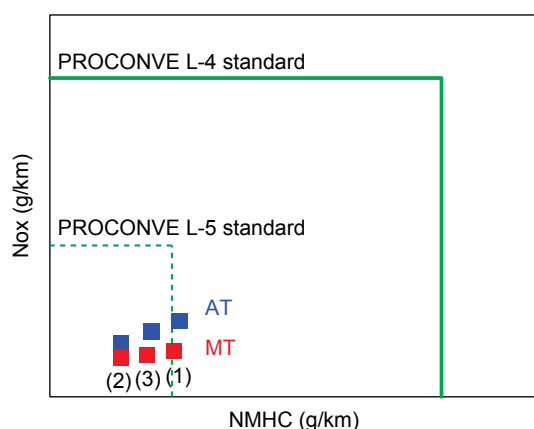
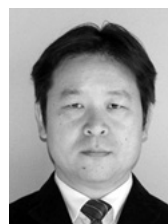


Fig. 13 Achievement of exhaust emission

■ Author ■



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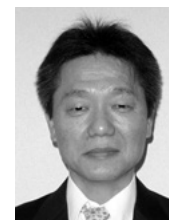
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Development of New BF90 Outboard Motor

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ABSTRACT

Honda has developed an all-new mid-sized outboard motor in the 4-stroke outboard motor series, the new BF90. Development of the new BF90 stressed enhancements to operating performance and fuel economy.

The new 2006 BF90 uses a SOHC 2-stage VTEC and a dual independent circuit control cooling system. With refinements to the intake system general specifications, including the intake manifold and air silencer, this outboard motor achieves a 13% higher specific power (kW/L) than the current BF90. Fully considering the ways in which outboard motors are used, an open-loop lean burn control system, which uses oxygen sensors, was adopted, raising fuel economy by 20% over the current BF90. At the same time, the motor complies with 2008 CARB emission regulations and the 2006 European Recreational Craft Directive's regulations.

Modifications to the gear case shape have reduced water resistance. This, combined with a reduction in total motor weight, has enhanced operating performance. When used with an aluminum fishing boat, this new BF90 motor achieves 0-200 m acceleration 0.9 seconds faster than in the current motor, and a maximum speed that is 4.6 km/h higher.

1. Introduction

Mid-sized outboard motors (above 37 kW [50 hp] and below 74 kW [100 hp]) account for approximately 18% of all outboard motors in the world market (FY05; Honda research), totaling approximately 119 000 units. With an increasing awareness of fuel economy, comfort, and environmental conservation, the market for this class is becoming stronger, together with an overall market expansion.

In 1995, Honda launched the carburetor model BF75/90. Now, however, Honda has developed a new fuel injection (referred to as FI hereafter) model BF90, which surpasses the current BF90 in terms of being both economical and environmentally friendlier. It also provides enhanced operating performance through higher power and a reduction in operating resistance.

The external design of the new motor is a successor to the Wing Form, based on the integrated lines found in the BF225 and BF150, completing the design lineup of Honda's mid-sized and large outboard motors. Figure 1 is a photograph of the BF90.



Fig. 1 External view of new BF90

* Power Products R&D Center

Table 1 Specifications

Code name	New BF90 L-type
Overall length (mm)	740
Overall width (mm)	440
Overall height (mm)	1570
Transom height (mm)	537
Dry mass (kg)	165
Engine type	4-stroke L-type 4-cylinder
Displacement (cm ³)	1496
Bore (mm) × stroke (mm)	73 × 89.4
Compression ratio	9.7:1
Valve train	SOHC VTEC 16 Valves
Valve number	4/cylinder
Maximum power (kW/rpm)	66.2/5800 (90PS/5800rpm)
Maximum torque (N·m)	127
Recommended engine speed range (rpm)	5300 - 6300
Fuel supply system	Programmed Fuel Injection
Ignition system	Programmed Ignition (with Plug Top Coil)
Charge output (A)	35
Cooling system	Water cooling (with 2 thermostats)
Exhaust system	Through propeller hub

2. Main Specifications and Overview

The main specifications of the new BF90 are given in Table 1. The engine uses the 1.5 L in-line 4-cylinder engine⁽¹⁾ used in the 2005 model FIT as a base. A 4-valve SOHC system, which features a 2-rocker arm VTEC mechanism⁽²⁾ on the intake side only, is used for the new valve system.

3. Enhancing Engine Performance

3.1. Increasing Engine Power

The new BF90, in a first for an outboard engine, uses an SOHC 2-stage VTEC engine with a shared rocker shaft for both intake and exhaust. In addition, as a result of an air fuel ratio tuning, adoption of a knock control, and the loss reduction in the fully equipped state by using air intake refinements and the dual independent circuit control cooling system⁽³⁾ adopted on the BF150, higher output was achieved across the entire range. Though the displacement was reduced to 1496 cm³ from that of the current BF90's 1590 cm³ (Fig. 2).

3.2. Enhancing Fuel Economy

3.2.1. Lower fuel consumption achieved through lean burn

The new BF90, like the BF150, implements lean burn control to achieve lower fuel consumption. The BF150 was the first outboard motor to use lean burn control⁽³⁾

with linear air fuel ratio (LAF) sensors. In the BF90, an open-loop lean burn control system with an oxygen sensor was used to reduce cost. As a result, fuel economy was enhanced by 20% over that of the current BF90 (Fig. 3).

Moreover, by making the air fuel ratio (referred to as A/F hereafter) a lean 18:1 in the frequently used cruising speed range, an advantage was gained over other companies' motors in terms of practical fuel economy. Figure 4 is a graph of fuel consumption during operation.

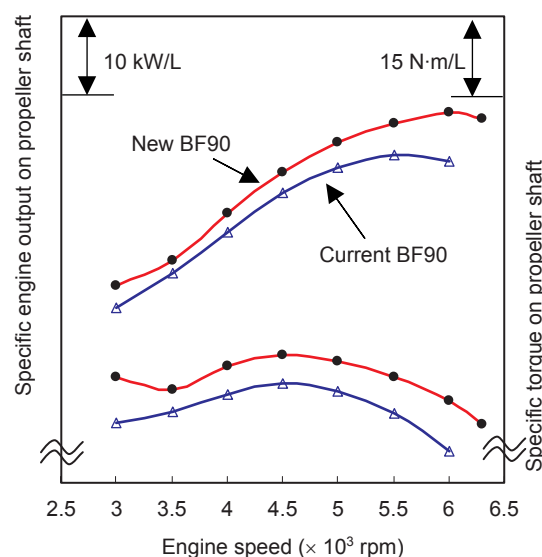


Fig. 2 Specific power and torque of BF90

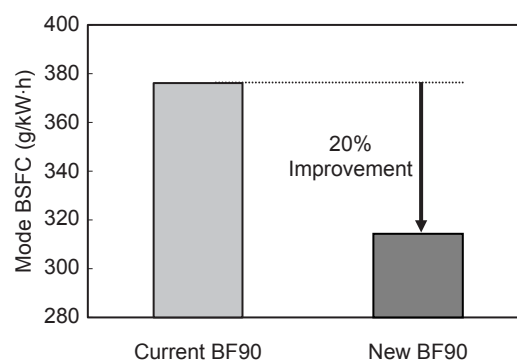


Fig. 3 BSFC in emission measuring mode

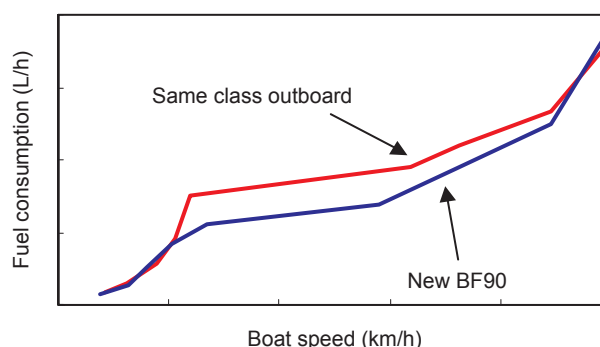


Fig. 4 Fuel consumption at cruising speed

3.2.2. Lean burn control using oxygen sensors

The new BF90 uses an oxygen (referred to as O_2) sensor instead of the LAF sensor used in the BF150. LAF sensors and O_2 sensors have difference in sensor output characteristics. Figure 5 shows these output characteristics. LAF sensors can detect A/F over a wide range, using sensor control circuits, including attached microcomputers. In contrast, O_2 sensors show an abrupt change in electromotive force of the sensor element at the stoichiometric air fuel ratio boundary, so it is only possible to judge whether the ratio is rich or lean. With an O_2 sensor, it is impossible to correct the fuel injection amount using feedback control.

Therefore, for the new BF90, a basic injection quantity map has been prepared for the stoichiometric air-fuel ratio aims, with engine speeds and load (intake manifold pressure) as the parameters. An open-loop lean burn control is used that determines the fuel injection amount at the time of lean burn by multiplying it by the target lean coefficient (equivalence ratio). This makes it possible to have lean burn operation without the use of expensive LAF sensors.

However, with this method, small errors in the basic injection quantity map lead directly to mismatches in the air-fuel ratio during lean burn operation, so accuracy of this map is extremely important. To ensure this accuracy, the following controls are carried out:

In the operation range assuming lean burn operation, once a feedback operation is carried out with the stoichiometric air-fuel ratio, the basic injection quantity map is corrected based on the oxygen sensor output, and the corrected amount is calculated. The correction values are recorded in nonvolatile memory as learned values for each of the ranges when the operating range is divided into eight segments in the engine speed direction. These values are saved, even after the engine stops.

The reason the operating range is not divided in the load direction is derived from the unique outboard motor cruising load characteristics. Unlike motorcycles or automobiles, for outboard motors without a gear shift device, when the boat and propeller used are determined, the relationship between engine speed and load is almost

uniquely determined, as shown in Fig. 6. That is to say, a load at a particular engine speed does not change much during steady operation, so it was determined that optimizing learning values for load direction would be unnecessary.

In this way, by segmenting the correction value for each operation range and optimizing, the accuracy of the basic injection amount map is enhanced.

The learning values determined in this way are used to correct the basic injection quantity map during lean burn operation. Corrected-value calculations involve establishing representative engine speeds within each learning range, with the learning values then determined by linear interpolation at the representative points.

The conditions for shifting to lean burn operation are that engine warm-up is complete, that it is within the range subject to lean burn operation, that there is steady operation, and that there has been an update to learned values in the learned range divided into 8 according to engine speed.

In a certain learning range, if a learning value of the basic injection quantity map correction amount moves from the updated learning range to another learning range in which the learning value has not yet been updated, lean burn operation stops and operation is carried out with the stoichiometric air-fuel ratio. After the learning values are updated, the unit shifts again to lean burn operation. In addition, if the learned values are updated in all lean burn ranges, even if there are learning range changes during lean burn operation, the learning values in each range are used, and lean burn operation continues.

With the learning value and lean burn operation condition settings as described above, the air-fuel ratio (A/F) deviation during lean burn operation can be kept to $A/F = 18 \pm 0.5$, which is also favorable in terms of drivability during lean burn operation.

The new BF90 exhaust gas emissions conform to final year (2008) requirements stipulated under the CARB marine emissions regulations, with the motor achieving an ULTRA-LOW emissions rating. In addition, it also conforms to the European Recreational Craft Directive's stipulated values, in force since 2006.

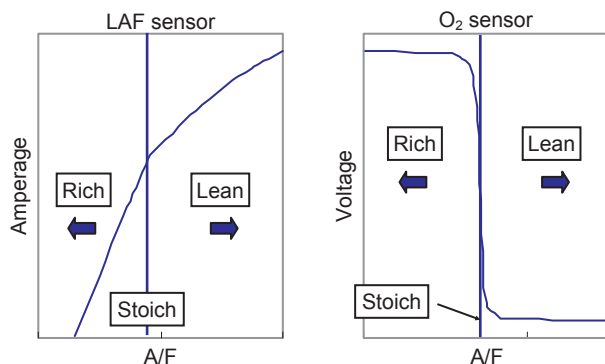


Fig. 5 A/F ratio sensor output

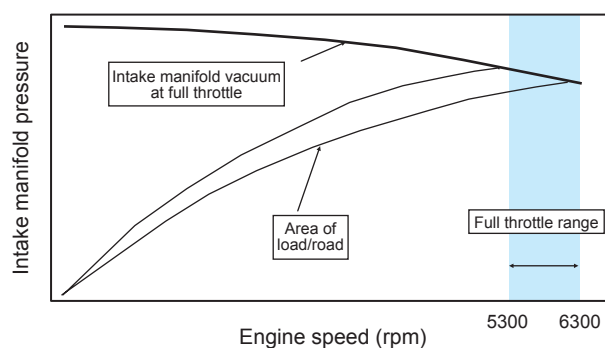


Fig. 6 Intake manifold pressure

4. Enhancing Operating Performance

4.1. Speed Increases due to Gear Case Shape Modifications

The underwater gear case parts comprise the gear box section, the parts connecting the gear box to the frame (body section hereafter), and an anti-ventilation plate provided to maintain pressure on the top of the propeller. Figure 7 shows the structure of the gear case. Making these parts smaller reduces the water resistance of the immersible parts, but ensuring the exhaust path and the durability of the gear need to be considered, and so making them extremely small is difficult. The following two items were the main focus when working to increase maximum speed both in the actual units and in simulations:

- (1) Modifications to the anti-ventilation plate, to reduce water resistance
- (2) Modifications to the gear box section and curve radius shape on the front of the body section, to enhance propeller aeration toughness by mitigating surface pressure reduction

The shape of the new BF90 gear case anti-ventilation plate was modified to make the case as small as possible compared to the current BF90 with no drop in aeration toughness when accelerating or under high load. This resulted in a maximum speed increase of approximately 0.8 km/h. In addition, a maximum speed increase of approximately 0.9 km/h was also achieved due to modifications to the water screen and screen rear shape, and tapering of the shape at the front and rear of the gear box section.

In comparison with the current BF90, a simulation of the pressure distribution on the surface of the new BF90 gear case showed mitigation of pressure reduction, due to enlargement of the gear box and front surface curve radius stop area. Figure 8 shows gear case surface pressure distribution. Changing the surface curve radius of the front of the body section from 250 mm to 500 mm leads to a speed increase of approximately 0.9 km/h. Figure 9

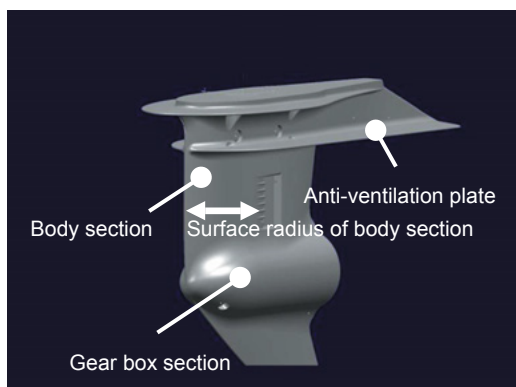


Fig. 7 Structure of gear case

shows the maximum speed according to differences in the front surface curve radius of the body section. These gear case shape modifications provided a maximum speed increase of 2.6 km/h over the current BF90.

4.2. Technologies to Reduce Weight

Through the use of a neodymium magnet ACG and by integrating the exhaust manifold into the cylinder head, the dry weight has been reduced by 7 kg compared to the current BF90.

4.2.1. Neodymium magnet ACG

To achieve a lightweight and compact unit that also has high charge power, the new BF90 uses a neodymium magnet ACG that is attached to the crankshaft instead of the belt driven alternators that are fitted on the large FI Honda outboard motors (115 hp or more).

Compared to the conventional belt-driven alternator's charge system, this ACG weighs 20% less. Also, the total number of parts has been dramatically reduced by eliminating the belt drive mechanism.

Moreover, by having the ACG stator coils and rotor closely attached to the engine, the height was reduced by approximately 10% as compared to conventional multi-pole magnet ACGs. To adapt the neodymium magnet ACG for use in an outboard motor environment, an air intake port was added to the ACG rotor, creating a structure that cools the ACG stator coils and rotor magnet. In addition, when considering its use at sea, the

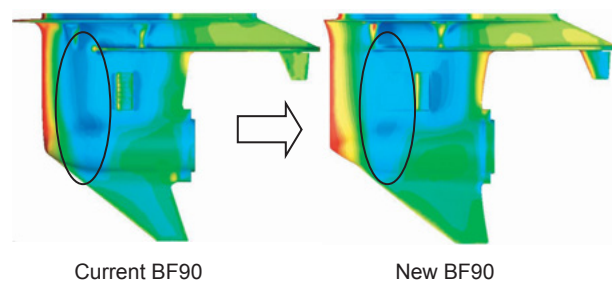


Fig. 8 Pressure distribution of gear case

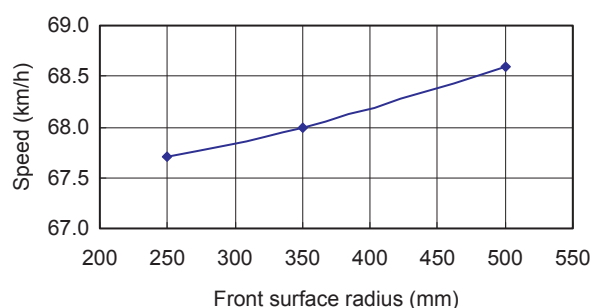


Fig. 9 Maximum speed according to front surface radius

specifications fully consider the corrosion resistance of the neodymium magnet. Figure 10 shows a photograph of the ACG.



Fig. 10 New BF90 ACG

4.2.2. Cylinder head with integrated exhaust manifold

Conventionally, the structure around the exhaust pipe is of two types – a separately fitted exhaust manifold, or a passage in the cylinder block. In contrast, the new BF90 creates an exhaust manifold using the cylinder head alone, thereby reducing the weight of the unit by 3 kg when compared to the conventional unit.

With this new cylinder head design, a dual independent circuit-control cooling system, water-cooled regulator, and cooling routes free from external pipes were also used, thereby achieving both high power and compact size.

Figure 12 shows the cooling routes.

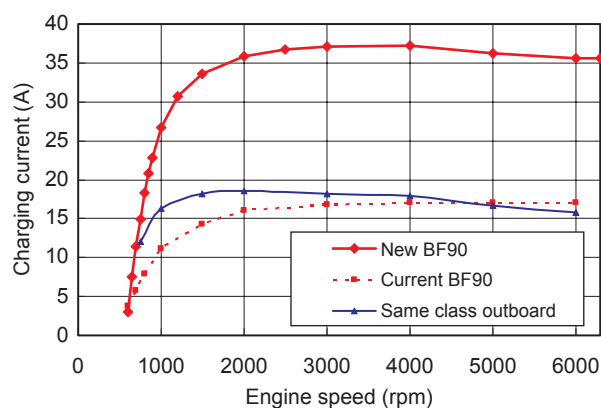


Fig. 11 Comparison of charging performance

4.2.3. ECU

The ECU, developed specifically for this new BF90, uses an Interstitial Via Hole (IVH) 4-layer circuit board to enhance density. The effective circuit connection surface area is increased by the use of non-penetrated inner via. By having circuitry pat on the via, the circuit board surface area was reduced by 45%. In combination with space-saving connectors, the case was made smaller, thereby reducing the injected resin reinforcements, making the weight of the ECU parts at least 40% lighter than ECU parts for the BF150's conventional in-line 4-cylinder engine. Figure 13 shows a photograph of the ECUs.

The engine control system was fitted with programmed fuel injection (FI) developed for outboard motors larger than the BF115. With the aim of achieving both excellent fuel economy and superior emissions performance, oxygen sensors and resonance knock sensors were adopted, the latter to detect engine knocking.

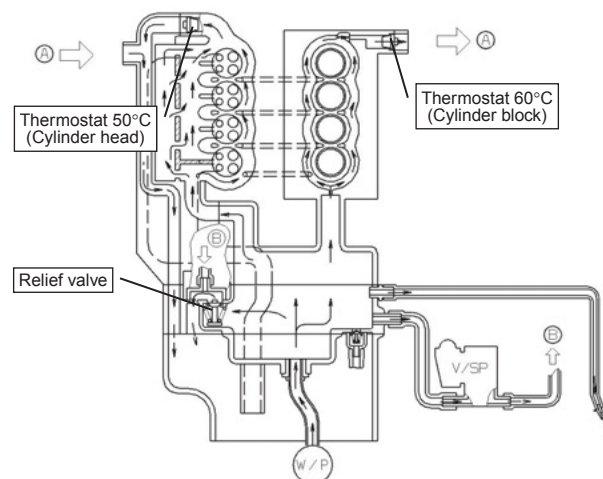


Fig. 12 Cooling system of new BF90

5. Conclusion

In the development of the new BF90, efforts were made to achieve both excellent fuel economy and superior operating performance. The following results were achieved, in comparison with the current BF90.

- (1) A 13% higher specific power.
- (2) A 20% enhancement in fuel economy.
- (3) In terms of acceleration performance, 0 to 200 m acceleration was reduced by 0.9 sec.
- (4) A 4.6 km/h higher maximum speed.

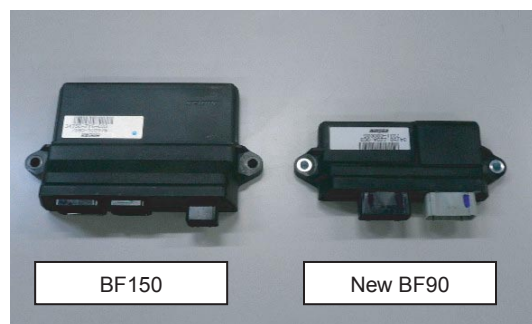


Fig. 13 New BF90 ECU

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■ Author ■



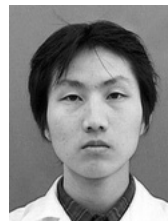
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Development of Super Handling-All Wheel Drive System for SUV

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ABSTRACT

A rear differential unit suitable for SUV vehicles was developed based on the super handling – all wheel drive (SH-AWD) unit that was mass-produced for the LEGEND. This newly developed rear differential unit has a higher torque capacity and is smaller and lighter than the previous SH-AWD. Redefining the demanded performance of the rear differential unit, according to the characteristics of the vehicle in which it is to be used, resulted in a balance of increased maximum torque and weight reduction. To achieve the development goals, selection of the optimal ratio (speed-increasing ratio), a clutch capable of high μ with low abrasion, and an actuator that optimally control the new clutch were mentioned.

1. Introduction

The SH-AWD system, mass-produced for the LEGEND in 2004, enhanced cornering performance while reducing the burden on the driver by the four-wheel drive system.

Now, the SH-AWD system is also mounted in the RDX and MDX released in 2006, and these models use a newly developed rear differential unit.

This newly developed rear differential unit is based on the torque control structure by direct electromagnetic clutch which is the main feature of the SH-AWD system. The performance demanded from the 4WD system has been set with consideration of the applied vehicle's characteristics. Also, the maximum output torque has been increased and the weight and size have been reduced by using the optimum settings for each element.

This paper mainly presents an overview of the newly developed rear differential unit and its major technologies.

2. Development Goals

The goals for this new rear differential unit development were set as follows.

- Enhance cornering performance (relative to the previous vehicle) based on free control of the four-wheel drive force
- Simultaneous achievement of higher output, smaller size, and lighter weight

Development was conducted with a focus on the following items as technical elements for realizing the above goals:

- (1) Speed-increasing ratio settings that match the vehicle's characteristics
- (2) A new clutch achieving both the high μ and low abrasion
- (3) Actuator settings suited to the new clutch

3. Basic Structure and Specifications

Figure 1 shows the unit appearance, Fig. 2 shows the main cross-section view.

The overall structure closely resembles the VTM-4 (Variable Torque Management® 4-Wheel Drive System) shown in Fig. 3, which was mounted in the first-generation of the MDX, PILOT, and RIDGELINE, but the double-reduction configuration using a hypoid gear and

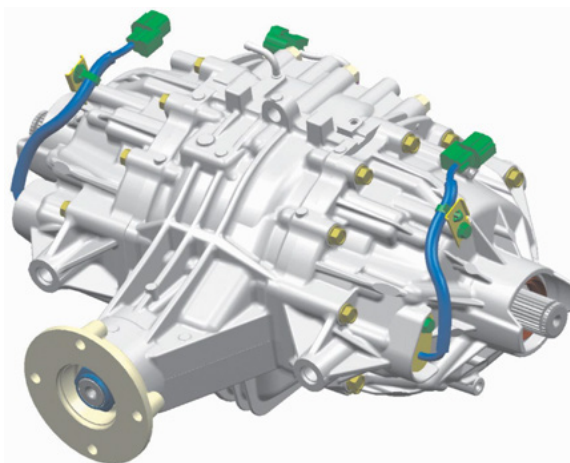


Fig. 1 View of newly developed rear differential unit

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planetary gear is based on the SH-AWD.

A hypoid gear, set to a dedicated ratio, and a lubricating pump are located in the center section. The case configuration has a back and front two-block construction similar to that of the VTM-4, and the case is also produced by the same high-pressure die-cast manufacturing method.

The difference in the center section from the VTM-4 is that the hypoid ring gear mount has been changed from a ball bearing to a tapered roller bearing. This was done to increase the gear mounting stiffness and to reduce weight by reducing the bearing size.

Figures 4 and 6 show the developed rear differential unit structure, and Fig. 5 shows the VTM-4 structure.

A bearing set plate and M8 flange bolt, that secures this plate to the case, have been added for the developed rear differential unit.

An axial load (called a preload) is needed to make the tapered roller bearing function properly. However, when using a front and back divided case, a device (jig) that

compresses the bearing is needed during assembly. Therefore, the bearing set plate structure makes it possible to apply a preload after case coupling. Therefore, mounting stiffness can be increased while ensuring ease of assembly.

The right and left clutch control sections have a configuration similar to that of the previous SH-AWD. (Fig. 7) The control sections are comprised of the following three elements:

- (1) Planetary gear assembly
- (2) Multiplate wet clutch
- (3) Electromagnetic solenoid

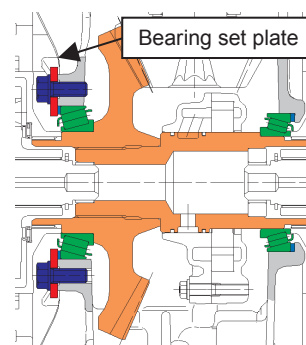


Fig. 4 Developed structure

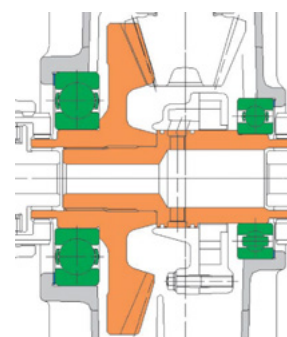


Fig. 5 Previous structure

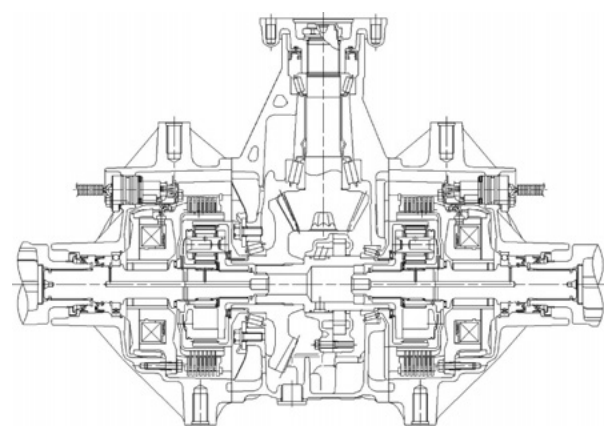


Fig. 2 Cross-section view of newly developed rear differential unit for RDX and MDX

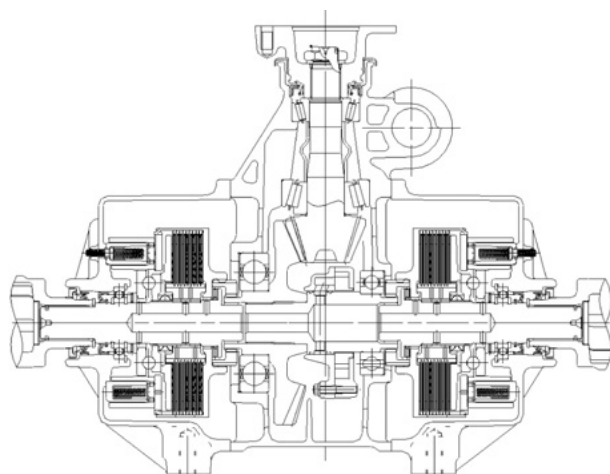


Fig. 3 Cross-section view of previous rear differential unit VTM-4

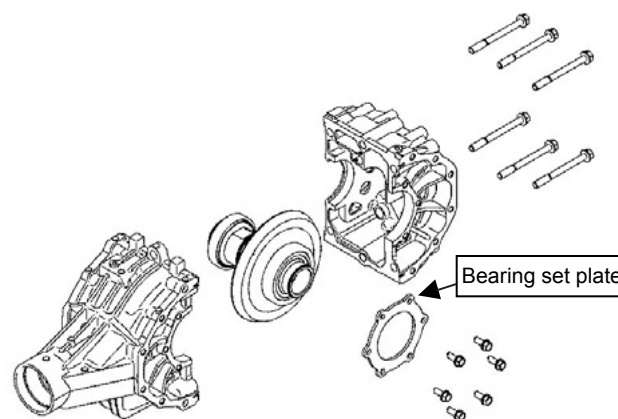


Fig. 6 Developed structure

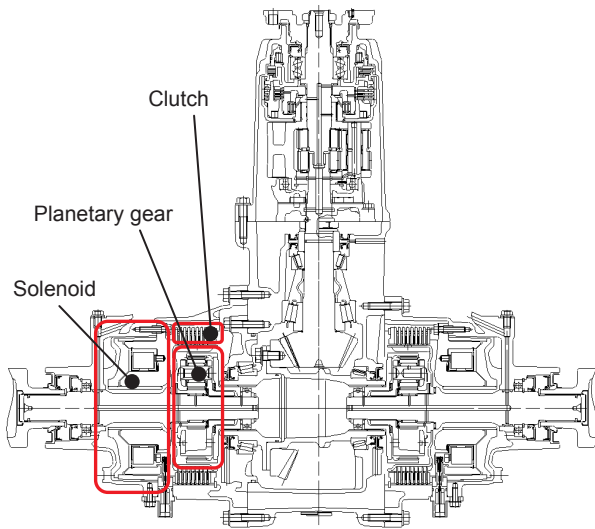


Fig. 7 Cross-section view of rear differential unit SH-AWD for LEGEND

The planetary assembly has three elements: a ring gear, a carrier, and a sun gear. Torque is input from the ring gear and output from the carrier. The output torque can be controlled freely, from zero to the maximum output, by controlling the torque of the remaining element, the sun gear. This mechanism itself is the same as the one that was developed for the previous SH-AWD, but everything except the planetary ratio setting has been newly designed to reduce the size and weight.

This has increased the maximum output torque compared to those of the previous VTM-4 and the SH-AWD.

4. Speed-increasing Ratio Setting

The target application vehicle was determined to be the RDX and MDX in this system development.

These models both fall under the SUV category and are subject to a strong demand for driving performance equivalent to that of a sports sedan as the concept for these vehicles.

However, SUVs generally have a higher center of gravity than sedans, and it is not easy to increase cornering performance. Therefore, the SH-AWD system is positioned as one measure for enhancing cornering performance.

The SH-AWD system must set the internal revolutions of the rear differential unit higher than the front wheel revolutions. This is to apply torque in the drive direction to the rear outer wheel when the turning path of the rear outer wheel is larger than the turning radius at the center of the front wheels⁽¹⁾.

The speed-increasing ratio setting is an important factor in the SH-AWD system, so this was investigated in various driving modes. The results showed that

sufficient merit in cornering performance can be obtained if the rear outer wheel can be driven at a turning radius of 30 m or more. Based on these results, this development set a constant speed-increasing ratio of 1.7%, and eliminated the speed-increasing gear section applied to the previous SH-AWD to perform two-speed shifting. The reason for this setting is that based on the above-mentioned SUV characteristics, the front wheel slip ratio was predicted to increase as the steering angle increased, and this was predicted to increase the actual speed-increasing ratio. The front wheel slip ratio was estimated at 0.8%, and theoretical investigations using this figure, predicted that sufficient performance could be obtained at a speed-increasing ratio of 1.7%. Figure 8 shows the investigation results, and indicates that the rear outer wheel can be driven at a turning radius of 28.4 m or more given a front wheel slip ratio of 0.8%. This supports the requirement confirmed by the above investigation that the rear outer wheel be driven at a turning radius of 30 m or more.

Figure 9 shows these verification results.

Performance was confirmed by actual driving tests over normal paved roads and snow surfaces. The results showed that a front wheel slip of 0.8% or more occurred.

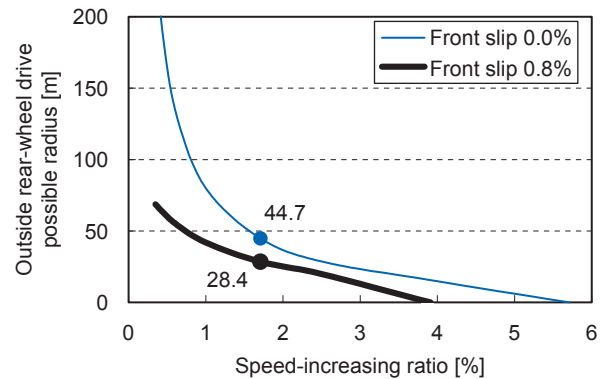


Fig. 8 Relation between speed-increasing ratio and drive possible radius

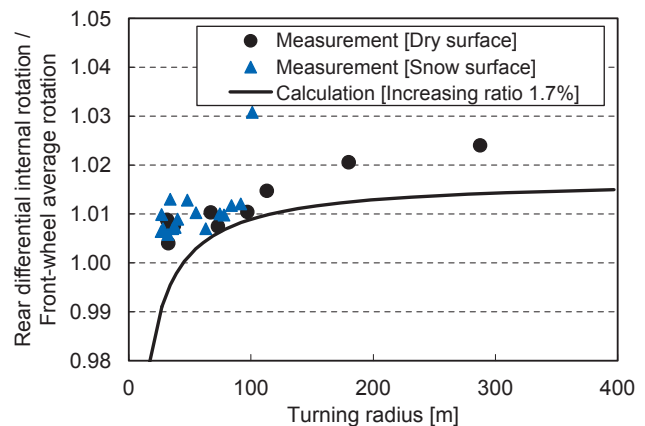


Fig. 9 Measurement result of rear differential internal rotation

The figure was used in the theoretical calculations, and this indicates that the rear outer wheel could be sufficiently driven even at a turning radius of 30 m or less. In this manner, setting the speed-increasing ratio with a focus on the vehicle characteristics made it possible to realize sufficient lateral drive force distribution effects without two-speed shifting by a speed-increasing gear.

As shown above, the SH-AWD speed-increasing ratio is an important factor in determining cornering performance. In addition, it also greatly affects the clutch abrasion described in Section 5, so this must also be taken into account.

5. High μ , Low Abrasion Clutch

The μ value and characteristics of the clutch are pointed out as the element to determine the torque capacity (maximum-output torque) of the rear differential unit. The SH-AWD system clutch constantly performs slip control, so in addition to a high absolute value for μ , low abrasion is also required. This abrasion amount affects the shape of the actuator (hereafter “electromagnetic solenoid”) described in Section 6.

This development not only reviewed the materials of the clutch developed for the previous SH-AWD, but also changed the manufacturing method. This simultaneously achieved both a higher μ value and low abrasion. The specific changes are as follows.

- (1) The facing materials were changed.
- (2) The molding process was changed.

Figure 10 shows the measured μ characteristic values for different slip revolutions. This shows that the μ value increased compared to that of the previous clutch. The important point here is that the μ value increases together with the revolutions. When this characteristic is lost, issues such as vibration and abnormal noise can occur during slip control.

A higher μ value was achieved by optimizing the percentages of the raw material mixture that comprise the facing of the clutch and by increasing the surface smoothness. Increasing the μ value made it possible to

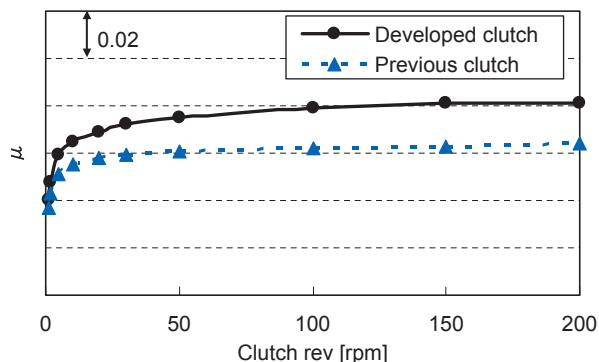


Fig. 10 Clutch μ characteristic

reduce the number of clutch plates, which also reduced the weight and cost.

Abrasion reduction was achieved by changing the manufacturing method to increase the hardness of the friction materials. Figure 11 shows the initial abrasion immediately after the start of use and the abrasion after use equivalent to 240 000 km. Compared to the previous clutch, the newly developed clutch reduced both initial abrasion and durability abrasion.

The clutch-absorbed energy was used to predict the lifetime clutch abrasion. This can be calculated from the output torque and clutch revolutions while running. The absorbed energy when running in a certain mode was converted to the amount of lifetime and was used as the basis for the lifetime abrasion setting.

In addition, abrasion is greatly influenced by the speed increasing ratio described in Section 4, so investigations were also simultaneously conducted from that aspect. Figure 12 shows the results of those theoretical investigations.

The absorbed energy and the speed increasing ratio basically have a proportional relation. However, this relation changes according to the amount of lateral

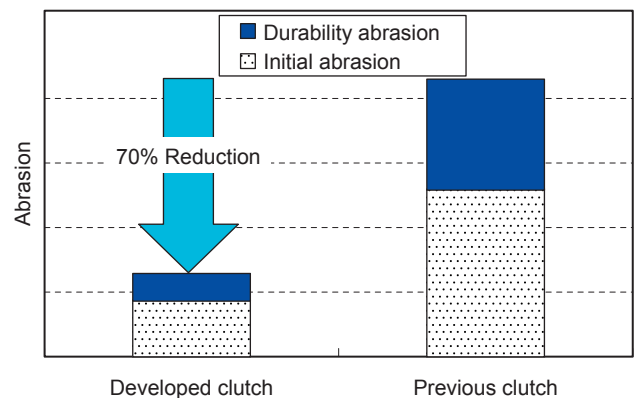


Fig. 11 Abrasion comparison

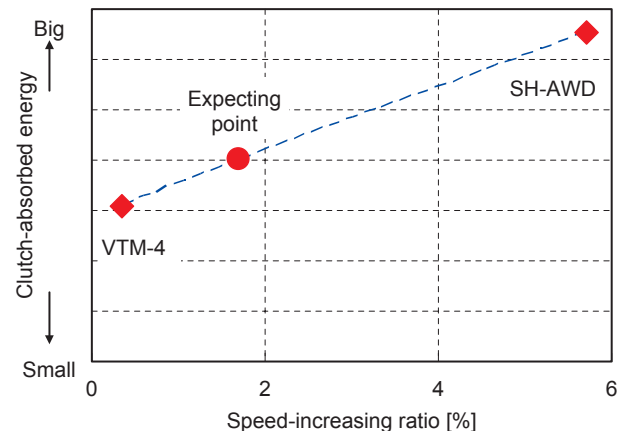


Fig. 12 Relation between speed-increasing ratio and clutch-absorbed energy

distribution control required from the viewpoint of maneuvering stability, so coordination with the vehicle control side is necessary.

6. Actuator Setting

The actuator (hereafter “electromagnetic solenoid”) that controls the clutch uses a flat gap, which reduces both the size and weight by optimizing the speed increasing ratio setting and reducing clutch abrasion. The specific changes are as follows.

- (1) The air gap shape was changed (from tapered to flat).
- (2) The number of main coil windings was changed.
- (3) The number of search coil windings was changed.

The direct electromagnetic clutch, which is a feature of the SH-AWD system, has a structure that controls torque by applying an electromagnetic solenoid attraction force directly to the clutch. The attraction force is inversely proportional to the square of the air gap, so setting a smaller air gap is an effective means of reducing the size and weight, but the air gap cannot be set smaller than the lifetime clutch abrasion. That is to say, the electromagnetic solenoid requires an air gap to generate attraction force, so the attraction force naturally cannot be generated if the clutch becomes worn to the point that the air gap disappears (the yoke and the armature contact each other). This shows that the air gap setting and the clutch abrasion are closely interconnected.

This development reduced the lifetime air gap variation by setting the optimum speed increasing ratio and developing a low abrasion clutch. As a result, the shape of the electromagnetic solenoid air gap was changed from a tapered shape to a flat shape, which resulted in a significant reduction in size. Figure 13 shows a cross section of the developed and previous electromagnetic solenoids at the same scale. The red bold lines in the figure indicate the air gap.

As mentioned previously, the characteristics of the electromagnetic solenoid are such that the attraction force changes with the air gap. However, slip control is constantly performed, so the clutch thickness that

determines the air gap experiences abrasion, albeit at an infinitesimal rate. This means that the air gap changes, so the attraction force and thus the clutch torque also change. The magnetic-flux detection system detects this change and corrects the current value applied to the electromagnetic solenoid⁽²⁾. The magnetic-flux detection system used in this development is based on the previous SH-AWD, but changing to a flat air gap made it necessary to enhance the magnetic-flux detection accuracy. Both the number of search coil windings and the detection current were increased, resulting in a magnetic-flux detection accuracy increase of 20%. The control performance equivalent to the previous actuator was achieved by these measures.

7. Results

The following results were obtained with respect to the development goals of “enhancing cornering performance” and “the simultaneous achievement of higher output, smaller size, and lighter weight.”

7.1. Enhanced Cornering Performance

The performance of a vehicle equipped with the newly developed rear differential unit and a first-generation MDX were compared on a normal dry road, snow-packed road, and dirt road surfaces. The results show that the development concept of “enhanced cornering performance” was achieved (Fig. 14).

7.2. Simultaneous Achievement of Higher Output, Smaller Size, and Lighter Weight

Figure 15 compares the maximum output. The newly developed unit achieved 10% higher output compared to the VTM-4 mounted in the previous vehicle.

Figure 16 compares the weight. Here as well, the newly developed unit achieved a weight reduction of 5% compared to the VTM-4.

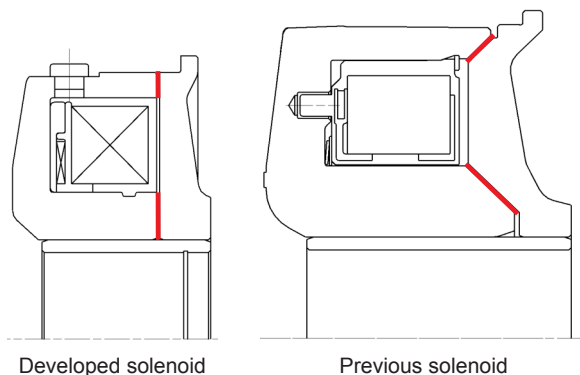


Fig. 13 Cross-section of electromagnetic solenoid

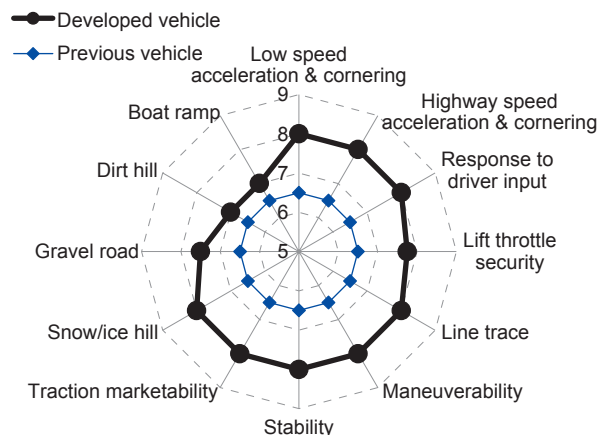


Fig. 14 Evaluation value

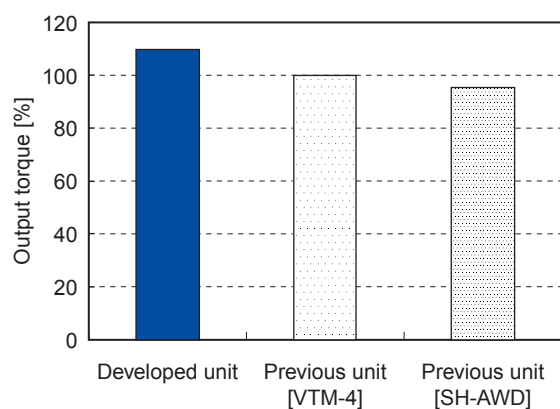


Fig. 15 Output torque comparison

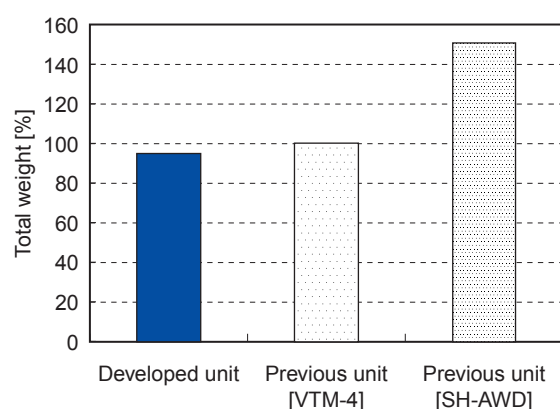


Fig. 16 Weight comparison

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8. Conclusion

- (1) Focusing on the vehicle characteristics and the tire slip rate, the turning radius at which the rear outer wheel can be driven was predicted to be 28.4 m at a speed-increasing ratio of 1.7%. In addition, the accuracy of this figure was verified through actual driving tests.
- (2) The optimum settings were made for an SUV vehicle based on free control of the four-wheel driving force developed for the LEGEND. This enhanced performance over all operating ranges compared to the previous vehicle.
- (3) Capacity increasing measures centering on the newly developed clutch increased the output by 10% compared to the rear differential unit mounted in the previous vehicle.
- (4) By setting the optimum speed-increasing ratio to reduce the clutch-absorbed energy, the electromagnetic solenoid weight was reduced. This achieved a 5% weight reduction compared to the previous rear differential unit.

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ABSTRACT

A POP-UP HOOD SYSTEM has been developed to help reduce the severity of head injuries to pedestrians in pedestrian-to-automobile accidents. The system operates by lifting the rear portion of the hood upon vehicle collision with a pedestrian, therefore increasing the clearance between the hood and the solid components underneath the hood, such as the engine. This system, which is being used on the new LEGEND model, enables the automobile to have a lower styled hood and thus a sporty exterior design, while enhancing safety features in line with the European pedestrian protection regulations that took effect in 2005.

1. Introduction

Awareness of pedestrian protection continues to increase as evidenced by new regulations in effect in Japan and Europe since 2005. Honda has been a leader in pedestrian protection, using for example a new vehicle body construction launched to the market in 1998 that helps reduce harm to pedestrians. It has also developed the new POP-UP HOOD SYSTEM as a way of further helping reduce injury to the head of pedestrians. This system, which is being used on the new LEGEND model, enables the automobile to have a lower styled hood and thus a sporty exterior design while enhancing safety features in line with European pedestrian protection regulations. The following paper provides details of the technology associated with the development of the POP-UP HOOD SYSTEM.

2. Development Goals

In the majority of accidents that are fatal to pedestrians, the pedestrian is fatally injured after being struck by the front of the vehicle and hitting his or her head on the hood⁽¹⁾(Fig. 1).

To help mitigate the severity of the impact to the pedestrian's head during collision, Honda has developed a system for the LEGEND that lifts the rear portion of the hood upon collision between a vehicle and a pedestrian. This in turn provides more clearance between the hood and the solid components underneath the hood, such as the engine. The goals of this development were as follows:

- (1) Achieve rapid system operation
- (2) Achieve precision collision sensing
- (3) Achieve pedestrian safety performance in line with European regulations

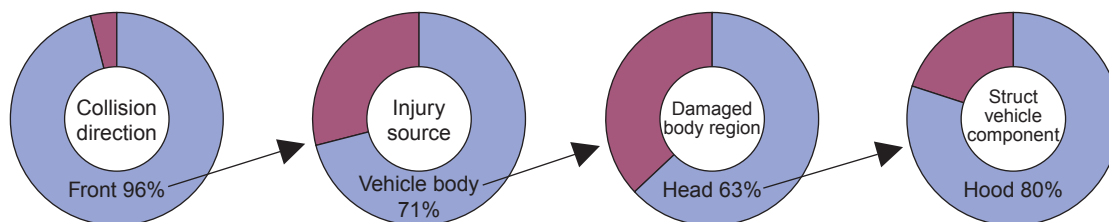


Fig. 1 Data of fatal pedestrian accidents

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3. System Overview

3.1. System Construction

The system configuration is shown in Fig. 2. The system consists of pedestrian crash sensor that detects collisions with pedestrians, wheel sensors to detect vehicle speed, an ECU for control, pyrotechnic actuators to lift the hood, hinges, and the hood.

The system's flow of operation is shown in Fig. 3. The operation of the system is controlled by inputs from both the pedestrian crash sensor and wheel sensors with thresholds set to match pedestrian impacts at a pre-defined speed or greater. Then, the pyrotechnic actuators lift the hood which deform the hinges. After this operation the position of the hood is maintained with the pyrotechnic actuators' lock mechanisms.

The distance that the hood raises is approximately 90 mm at the back end which is intended to not interfere with

the driver's vision during and after the system has been activated.

3.2. Pedestrian Crash Sensor

As Fig. 2 shows, a pedestrian crash sensor consists of an absorber supported by the bumper beam, which absorbs the energy of the collision with the pedestrian's legs and accelerometers. The absorber has three uni-axial accelerometers mounted on it.

These accelerometers are electrostatic capacitance type with environmental durability and reliability.

The ECU algorithm for operation is shown in Fig. 4. Accelerometer data are transmitted to the POP-UP HOOD SYSTEM ECU where calculations are performed and the results are compared to a preset threshold value. If the threshold value is surpassed, an electric current is sent to the pyrotechnic actuators for activation.

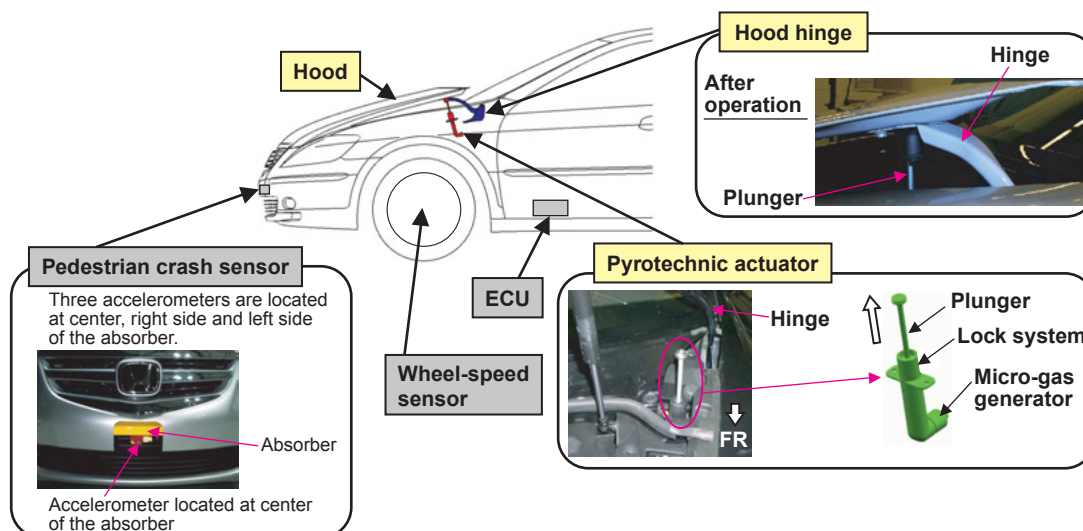


Fig. 2 System configuration

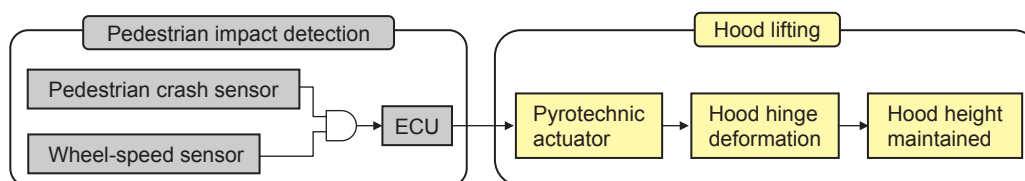


Fig. 3 Block diagram

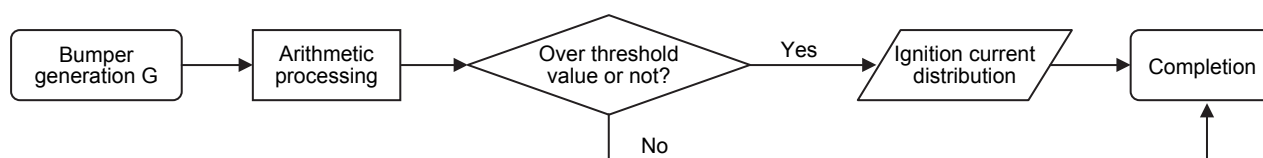


Fig. 4 Detection algorithm

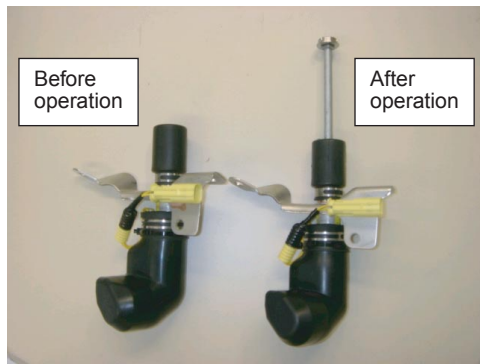


Fig. 5 Pyrotechnic actuator

3.3. Pyrotechnic Actuators

A pyrotechnic actuator is shown in Fig. 5. One pyrotechnic actuator is located under each of the vehicle's right and left hinges. As the pedestrian is struck by the vehicle, the pyrotechnic actuators activate and lift the hood prior to the pedestrian's head contact to the hood.

4. Performance confirmation

4.1. Confirmation of Instant System Operation

The system operation performance was confirmed in impact tests using pedestrian dummies (Fig. 6). The dummies tested included average American male adult size (175 cm tall, which is called AM50 henceforth and is a standard size crash dummy), and an average six-year-old child size (120 cm tall, which is called C6Y henceforth). The dummies simulate pedestrians struck by the vehicle from the side while crossing a road⁽²⁾. The tests were performed at a collision speed of 40 km/h with the pedestrian struck at the front of the vehicle along its mid-

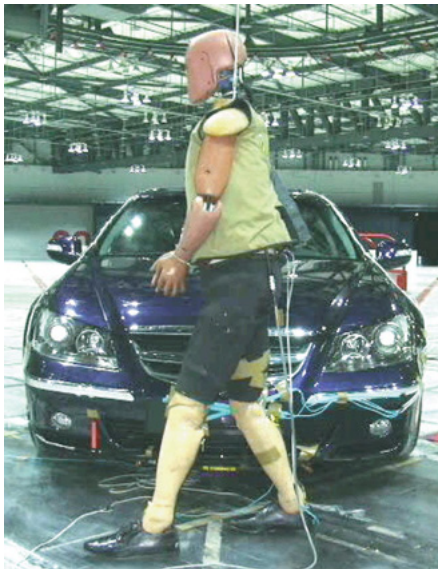


Fig. 6 Setting up of AM50 pedestrian dummy test

plane. Test results are shown in Fig. 7. The test confirmed that the system operation was completed quickly and prior to the head strike to the hood for both the C6Y and the AM50.

4.2. Precision Collision Sensing Confirmation

Sensing performance was confirmed with pedestrian dummy tests. The tests were performed at a collision speed of 40 km/h using the C6Y dummy which is lightweight and gives less input to the bumper, and therefore is more challenging to detect. Figure 8 gives representative test results. The tests confirmed that the sensors detected collision with the pedestrian across a wide area of the bumper quickly.

4.3. Confirmation of Pedestrian Safety Performance in Line with European Regulations

New pedestrian protection regulations took effect in Europe in October 2005. The tests involve the impact of pedestrian representative heads and legs with a vehicle and applicable injury criteria applied (Fig. 9). The head impact test is performed at an impact speed of 35 km/h.

The head impactor test for the European pedestrian protection regulations is shown in Fig. 10. The test results

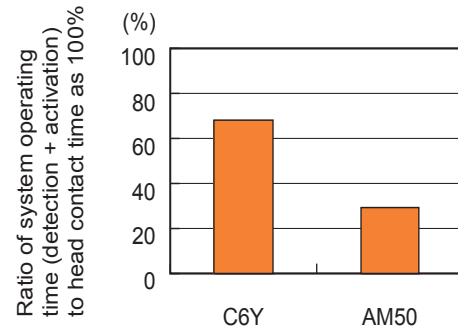


Fig. 7 Results of pedestrian dummy test struck at center of vehicle

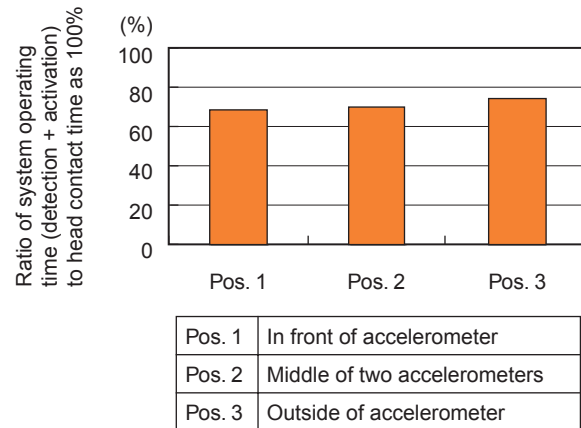


Fig. 8 Results of C6Y pedestrian dummy collision test at various crash positions

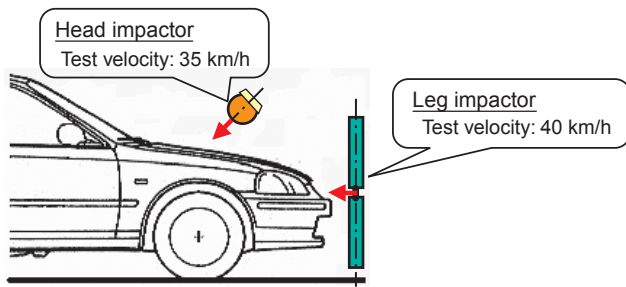


Fig. 9 European regulation test

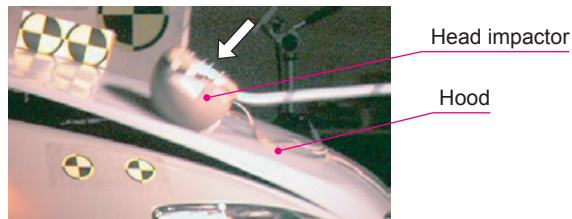


Fig. 10 Head impactor test with POP-UP HOOD

confirmed the Head Injury Criterion (HIC) did not exceed 1000 over two-thirds of the head impactor test area and 2000 for the remaining one-third of the head impactor test area.

5. Conclusion

A POP-UP HOOD SYSTEM has been developed to help reduce the severity of head injuries to pedestrians in pedestrian-to-automobile accidents. The system incorporates the following performance characteristics:

- (1) Using pedestrian crash sensor and pyrotechnic actuators, the system operation was completed rapidly and prior to the time the head struck the hood.
- (2) Precision collision sensing was achieved to detect collision with the pedestrian rapidly across a wide area of the bumper.
- (3) Pedestrian safety performance in line with new European regulations was achieved with a design that lifts the rear portion of the hood and provides more clearance between the hood and the solid components underneath the hood, such as the engine.

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Development of Active Damper System using Magnetorheological Dampers for 2007 Model Year Acura MDX

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ABSTRACT

This paper describes the development of magnetorheological (MR) dampers and their application to the 2007 model year MDX. MR dampers have been used in the market on various vehicles since 2001. They use a special oil-based fluid (Magnetorheological fluid, MRF) that contains small iron particles (1–10 μm in size) and a controllable electromagnetic piston to allow a wide range of damping forces. The system provides the customer a benefit over conventional dampers by using its wide range of available damping force to enhance body control while simultaneously providing outstanding ride comfort. This technology was implemented on the 2007 model year MDX under the name Active Damper System (ADS). The ADS system uses Honda-exclusive control logic that precisely controls damping force via skyhook, roll, pitch, and several other algorithms. The result is a vehicle with a higher level of both ride and handling performance that leaves all of our customers more satisfied.

1. Introduction

Magnetorheological (MR) dampers have been in the automotive market since 2001. The key component to MR dampers is a special Magnetorheological fluid (MRF) that consists of damper oil with approximately 20% pure iron particles that are 1-10 μm in size. Some of the early shortcomings of MRF that included sedimentation and long-term viscosity variation have been completely resolved by special additives being incorporated in the fluid⁽¹⁾. Honda investigated ADS technology and enhanced its overall vehicle performance and durability through electrical, mechanical, and control design.

The base structure of the MR damper is similar to a monotube damper except that the piston valve is replaced with an electromagnetic piston core that is then connected to the Electronic Control Unit (ECU) via an external electrical connector. This connection structure has been the detriment of MR dampers in the past and has been enhanced during the development. Changes were made to the damper's internal structure to increase maximum damping force and to its external structure to increase robustness to electrical connectivity and oil sealability.

overall sprung mass motions without compromise. In the end, two distinct settings of ADS were developed to accentuate the uniqueness of the system. Sport mode was developed to maintain the ride levels of the base 2007 model year MDX with conventional dampers and enhance handling to best-in-class levels. Comfort mode targeted base 2007 model year MDX levels of handling and class-leading levels of ride comfort. Figure 1 describes this concept.

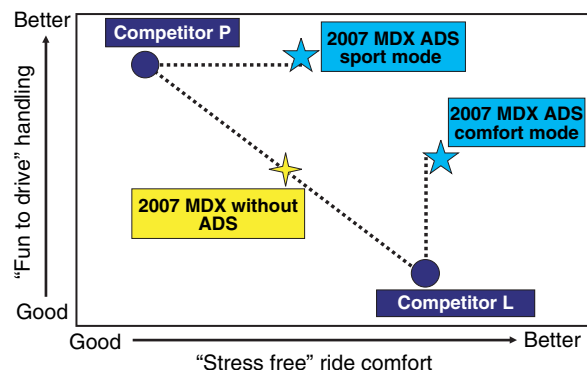


Fig. 1 Active Damper System targets

The target of ADS was to enhance ride, handling, and

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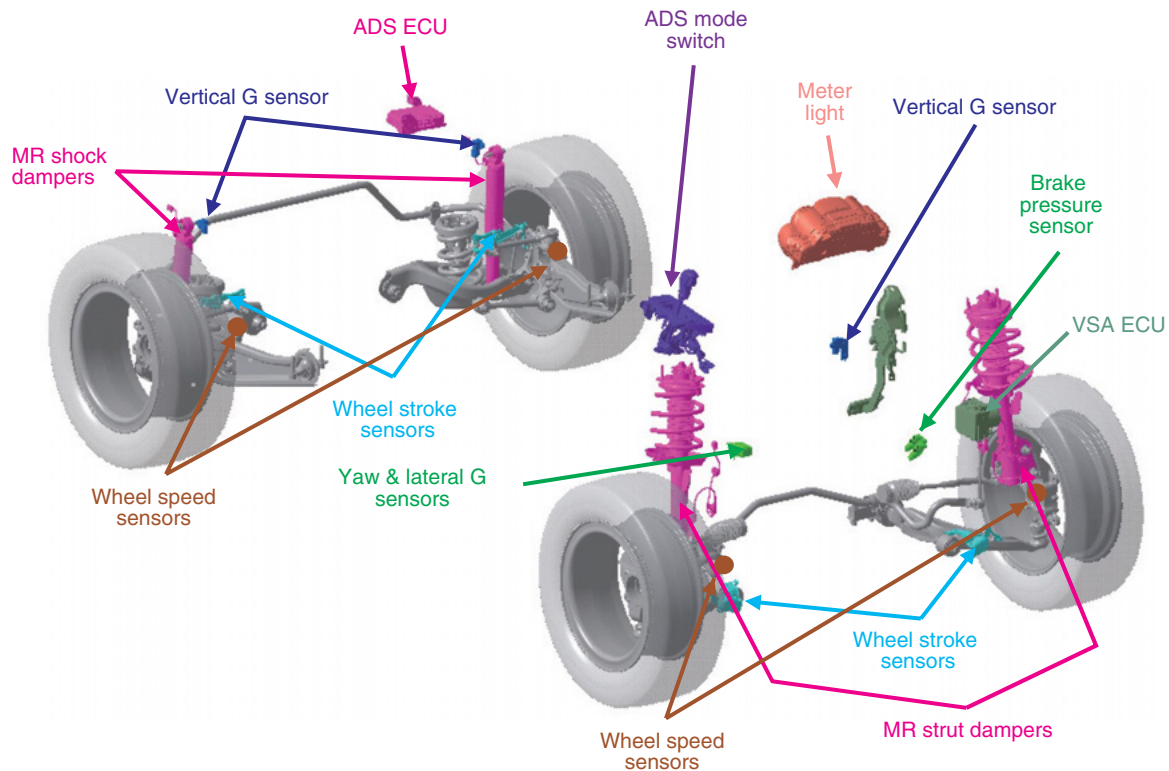


Fig. 2 Active Damper System components

2. System Overview

ADS consists of four wheel stroke sensors, three vertical acceleration sensors (G sensors), a combined yaw and lateral acceleration sensor, four MR dampers, and an ECU as visualized in Fig. 2. In addition, ADS communicates with the VSA system to determine vehicle speed and brake pressure. The output of each sensor is used by the control logic in the ECU to precisely determine the appropriate damping force required at each MR damper in order to achieve the desired level of lift, roll, pitch, and road input isolation for the vehicle.

3. MR Damper Construction

The 2007 model year MDX suspension is set up with front strut-type suspension and rear multi-link trailing arm-type suspension. In order to package the monotube structure of the MR damper in a strut, an inverted monotube strut is used. The rear MR damper uses a monotube shock. Figure 3 shows the details of the MR damper internal structure.

Unlike a conventional oil-based damper, the MR damper piston does not utilize a valve structure of deflecting disks and springs. Instead, the MR damper has a simple structure consisting of a piston outer ring, an electromagnetic piston core and two piston plates which set the piston core concentric with the piston ring making

a constant annular gap. It is this annular (radial) gap that defines the damping force characteristics of the MR damper. These structures are shown in Fig. 4.

Reference⁽³⁾ detail how an MR damper can vary the damping force versus velocity characteristics by applying

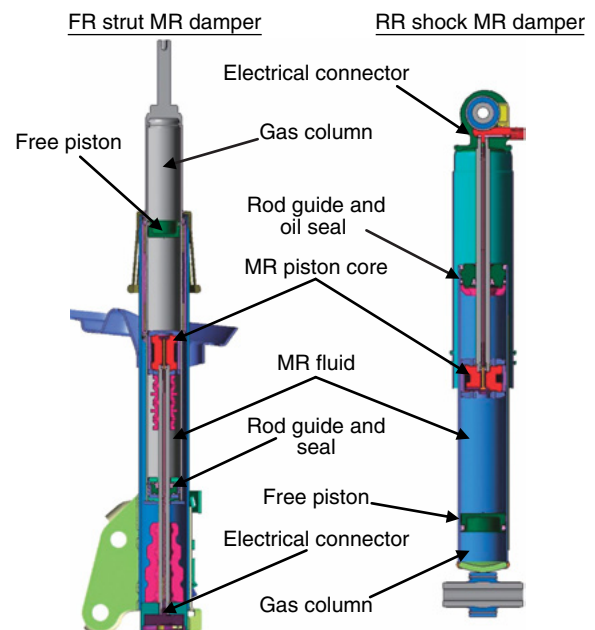


Fig. 3 MR damper construction

a range of 0 to 5 Amps of electrical current to the electromagnetic piston core of the damper. When the damper is in its off state (no current applied), MRF flows through the annular gap in a laminar state. When current is applied (on-state), the electromagnet builds up a magnetic field and creates magnetic bonds with the iron particles aligning within the annular gap. This effectively increases the viscosity of the fluid moving through the

orifice, increasing the amount of damping force the damper produces. Higher levels of current will further increase the strength of the bonds, raising the fluid shear force and subsequent damping force. The shearing occurs in the walls of the annular gap where the stress caused by the motion of the damper is greater than the yield stress created by the magnetic bonds. The area between the shear walls flows as a plug of magnetically bonded iron particles. This is called the MR effect and can be seen in Fig. 5. The MR effect only occurs at the ends of the piston where the magnetic flux vector is perpendicular to the annular gap. Varying the current applied to the electromagnetic piston core varies the flux.

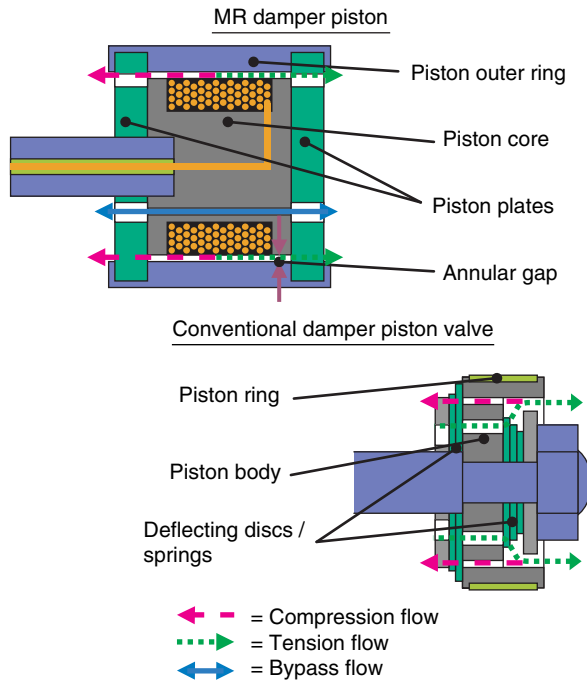


Fig. 4 Conventional damper piston valve versus MR damper piston

The ability to change damping characteristics on command is what separates the MR damper from a conventional damper. An MR damper responds more quickly than other semi-active and active damper systems with a response time of less than 5 milliseconds as compared to 10 milliseconds or more for other controlled systems and conventional dampers. The wide range of damping force available coupled with quick response time allows the vehicle to achieve simultaneous good ride and handling without the compromise that must be made when tuning the single damping force characteristic curve of a conventional damper. Figure 6 shows a comparison of a conventional damping force curve to that of an MR damper.

Within this characteristic curve, an MR damper can be tuned by two methods. The annular gap previously described can be varied. A smaller gap will increase the peak damping force available; conversely, a larger gap

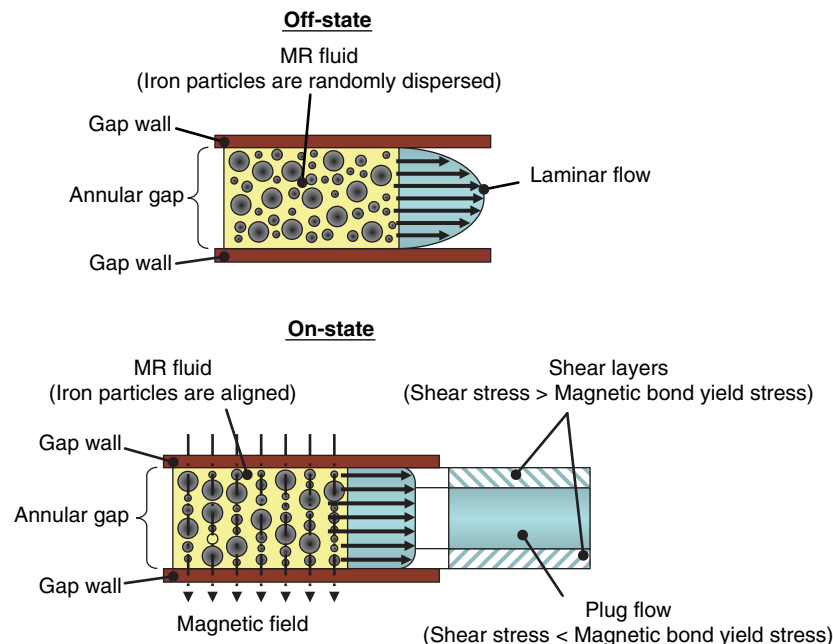


Fig. 5 MR fluid off- and on-state flow

will decrease the peak damping force. Within the piston there is also flow through an orifice inside the piston that does not see the magnetic field. This flow is called bypass flow. Varying the size of this orifice modifies the slope at which the damping force curve reaches peak damping force and off-state damping force characteristics.

Increased damping force was required for the 2007 model year MDX program over the current available capability of MR strut-type dampers in order to impart the SUV Platform with the desired level of roll control. Having already reduced the bypass lower than any other MR damper in production or test, the internal piston core bypass holes were removed, effectively filling them with the base steel of the core. This allowed more magnetic flux through the core and thus higher levels of damping.

The electrical connectors were also a focus for enhancement. On-vehicle connector vibrations were measured and modifications were made to the connector structure in order to enhance sealability in these conditions. The strut connector added three bolts and molded-in sleeves in order to maintain seal compression

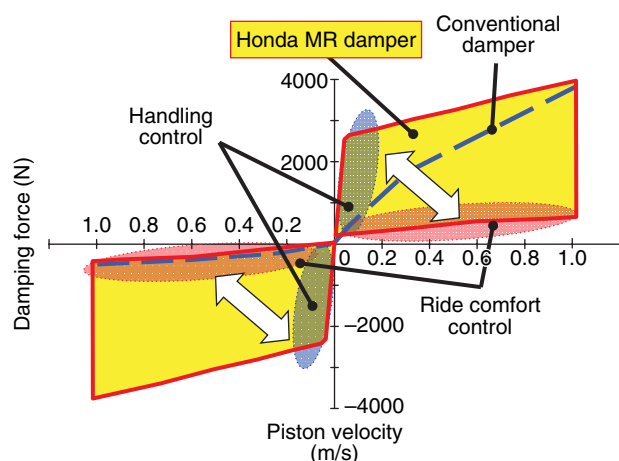


Fig. 6 Typical MR damper force versus velocity curve

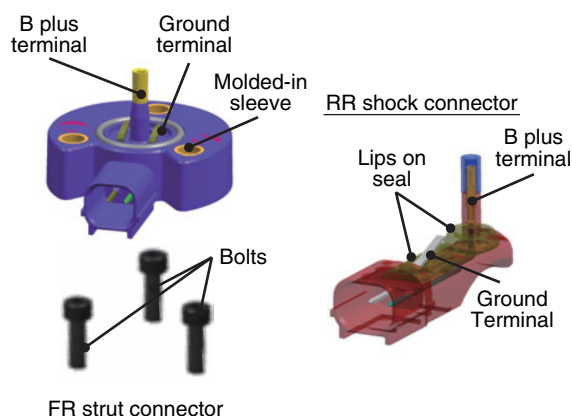


Fig. 7 Honda strut and shock MR damper connectors

over life. The rear connector seal structure was enhanced by adding lips and separating the ground and positive terminal seal structures. These changes are highlighted in Fig. 7.

4. ADS Control Logic

The basic theory for flat ride control of ADS control logic is skyhook suspension control. Skyhook control theory is visually depicted in Fig. 8. This well-known suspension concept applies damping force in a model that treats the sky as a fixed body with the damper C_1 between the sky and the sprung mass of the vehicle. The excitation force comes from the road surface through the tire with stiffness K_1 , the unsprung mass, and the vehicle suspension of stiffness K_2 and damping C_2 . Thus, the sprung mass motion can be controlled by the damping force provided by the damper connecting it and the fixed sky. In ADS, this control logic is taken to further understanding where the damping force can be controlled C_s in order to reduce sprung mass movements and thus flatten the ride of the vehicle for the customer.

In addition to skyhook control, ADS implements roll control logic, pitch control logic, and unsprung mass control logic. Reference⁽²⁾ describes.

the unique feature of the roll control logic in that it takes into account ride isolation while reacting to vehicle yaw and lateral acceleration response to firm-up the dampers and counteract roll. ADS implements this concept as a method that targets a roll control proportional to the lateral acceleration differential value of the vehicle. Figure 9 shows the block diagram of ADS control logic.

The vehicle inputs are converted to damper current values by ADS control logic in the following procedure:

- (1) Vehicle sensor inputs are interpreted by the body control logic (skyhook, pitch, and roll) after the derivative and integral processes have been completed on the sensor outputs

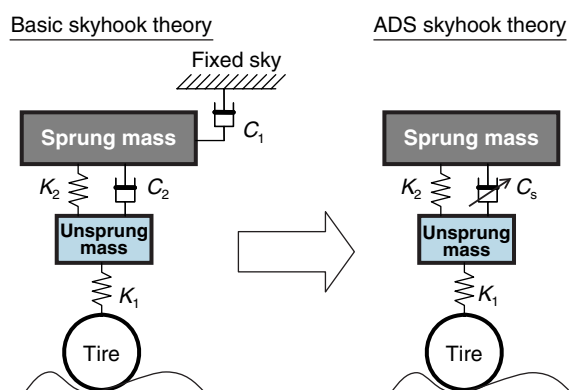


Fig. 8 Skyhook suspension control theory

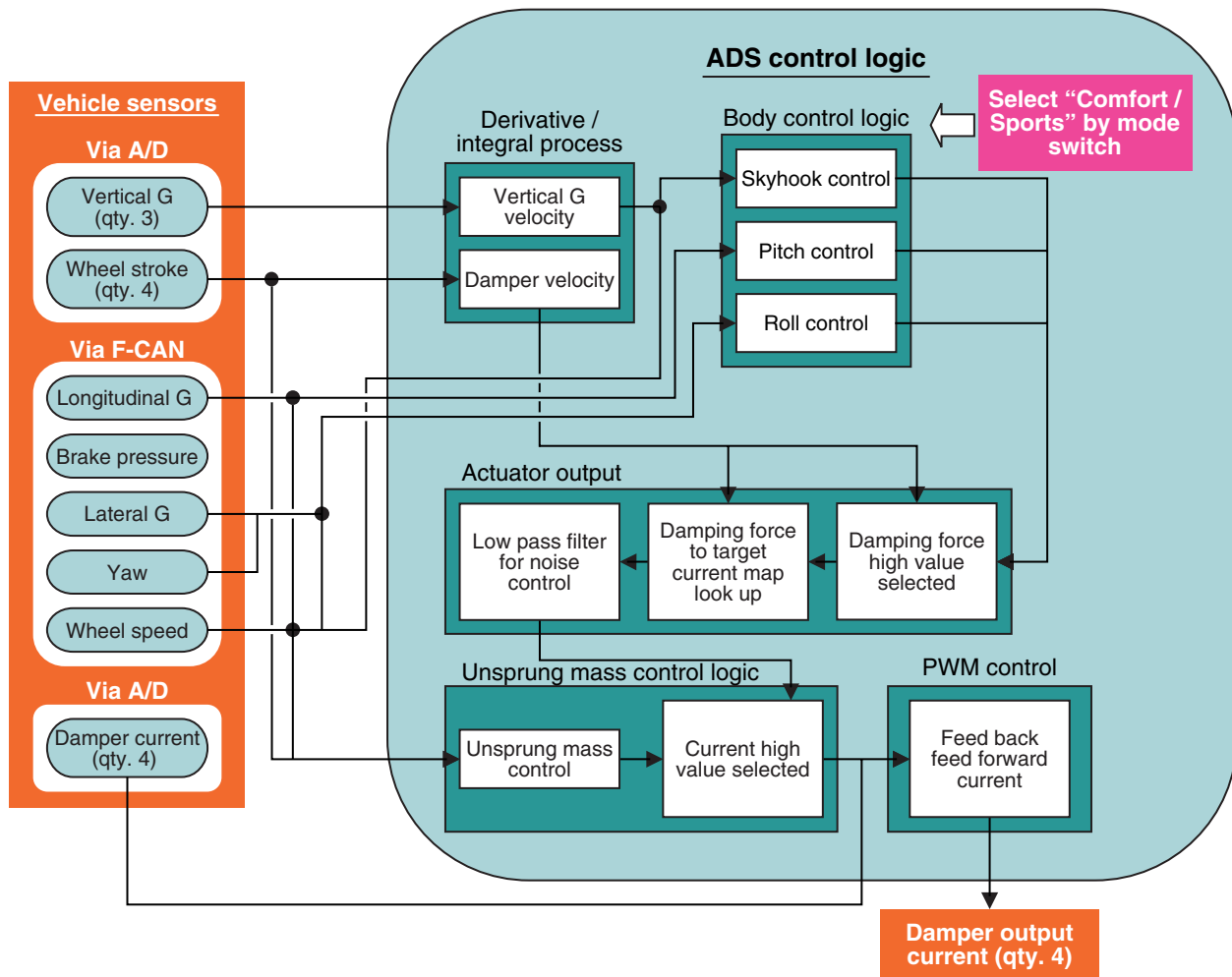


Fig. 9 Control logic block diagram

- (2) The highest damping force is selected from one of the three body control algorithms
- (3) Damper target current is looked up in the damping force map to match the high-selected damping force from the body control logic
- (4) The target current is then run through a low pass filter for noise control
- (5) If required, unsprung mass control logic adds current to the target current in order to modulate aftershake and other unsprung mass incidents
- (6) The target current is then compared to actual damper current and an adjusted output current is given to each damper

5. Vehicle Performance

The large range of damping force allowed by the MR dampers combined with the derivative control-based algorithm allow for a very wide range of ride and handling tunability. In order to appeal to the broadest possible range in consumer tastes, two driver-selectable damper modes were developed: "Sport and Comfort."

The goal of the selectable mode is to allow the driver to tailor the ride comfort and handling of the vehicle to their mood or driving condition and broaden the overall appeal of the 2007 model year MDX by reducing the ride-handling tradeoff that occurs with conventional damper tuning. Changing between the modes is accomplished using a two-position switch mounted on the console. The ADS operating mode is relayed to the driver via an indicator on the mode switch as well as through a message that is displayed on the combi-meter MID display for a short time, each time the ADS mode is changed.

Comfort mode and Sport mode operate using the same sensor inputs, algorithm, and damping force look-up tables. The ride and handling differentiation comes from higher Sport mode gain settings for the roll, pitch, and skyhook portions of the algorithm. These higher gain levels result in a higher commanded current for a given maneuver in Sport mode versus Comfort mode. The higher commanded current gives higher damping forces, yielding highly controlled roll, pitch, and heave motions. The highly controlled body motions in Sport mode have

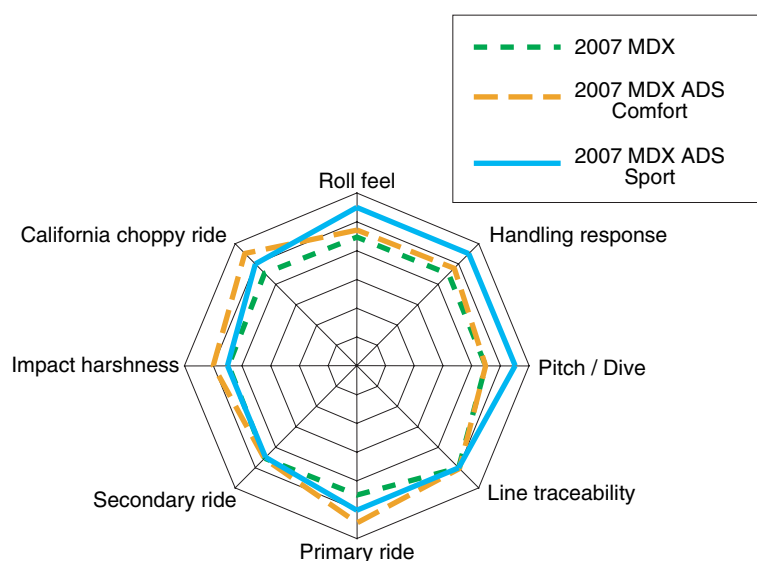


Fig. 10 ADS vehicle performance achievement

the added benefit of enhancing load transfer during transient maneuvers. Faster load transfer results in less delay in tire force build up, helping the vehicle to respond more quickly to driver steering and brake inputs. The net result is that the vehicle has a more nimble, agile feel in Sport mode versus Comfort mode. The higher damping forces also result in a small decrease in bump isolation, though ride levels are still significantly better than that of a conventional non-MR damper sport suspension. Decreased isolation actually has some positive benefit in that it differentiates the Comfort and Sport modes and provides the driver with an increased, but not annoying, level of tactile feel. Overall achievement levels of both the Sport and Comfort modes can be seen in Fig. 10 as referenced to the base conventional damper.

The Comfort mode uses the same algorithm as the Sport mode but lowers the gains and resulting damping force to allow the sprung mass to have a more relaxed, isolated road feel. Roll, pitch, and vehicle heave motions are tuned to be closer to that of the conventional suspension vehicle and the result is a level of ride isolation that exceeds the capability of a conventional mechanical damper. To take full advantage of the damper capability and maximize Comfort mode ride isolation, the spring rates on the MR equipped vehicle were decreased by about 18% versus the conventional damper springs.

6. Conclusion

Having focused on the limits of MR damper capability, extensive investigation, testing, and tuning have provided the following results:

- (1) A unique MR damper with enhanced damping force capability and durability through changes to the piston

core, electrical connectors, and sealing systems.

- (2) Combined with the enhanced MR damper and a two-gain mode algorithm, a vehicle that can exhibit traits of both a ride comfort-tuned and a sport handling-tuned suspension without typical tradeoffs.

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Sensor Calibration Technique and Check Method for Active Damper System

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ABSTRACT

An active damper system is an electronically controlled damping force suspension system. Technology for calibrating the sensor of this system and a method of checking its operation were developed. The wheel stroke position is calibrated when the wheels are moved up and down by driving the vehicle over a bump, and the operation of the sensors is checked simultaneously. The sensors were calibrated and their operations were checked within a few seconds without the need of any dedicated equipment.

1. Introduction

An active damper system (ADS) is an electronically controlled damping force suspension system, which satisfies riding comfort and also keeps the flat feeling of the vehicle posture (Fig. 1).

Generally, calibration of the sensors is necessary for an electronically controlled system after the vehicle is assembled. Also, the system operation is checked before shipment to help preventing a system malfunction from occurring after shipment.

However, if large equipment is used to calibrate the sensors and check the operation of the system, the manufacturing cost will be higher because the working time for one vehicle tends to be longer.

As same as other electronically controlled systems, it is necessary for the ADS that the sensors should be calibrated and their operation checked. Technology for completing this in a short time, without using any dedicated equipment, was developed and is described in this paper.

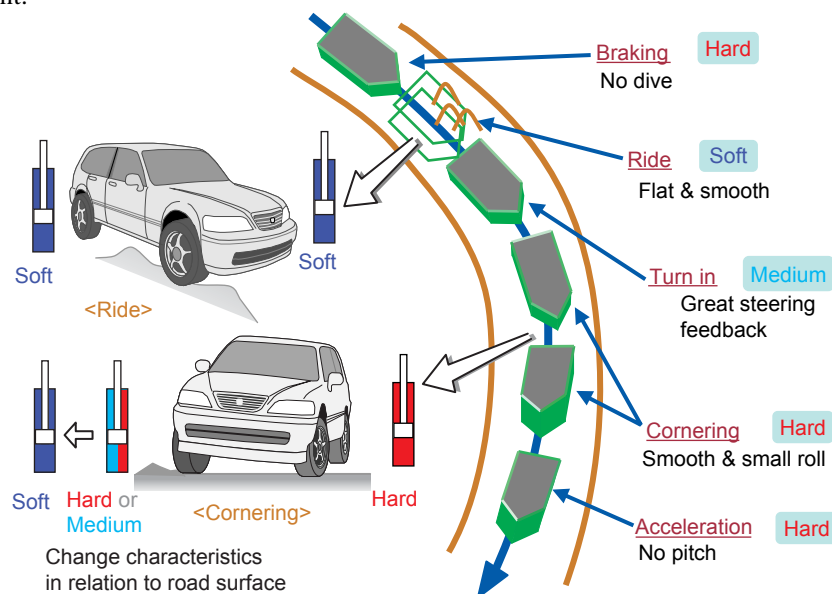


Fig. 1 Active damper system overview

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2. Outline of the ADS

The ADS controls the damping force of the suspension damper according to the differential of the longitudinal acceleration of the vehicle, and suppresses the pitching phenomenon whereby the front of the vehicle falls during braking and rises during acceleration. Likewise, the damping force of the damper is controlled according to the differential of the lateral acceleration of the vehicle, and suppresses the change in posture whereby the vehicle leans when turning or changing lanes⁽¹⁾. The ADS also reduces the vehicle vertical velocity that is generated by an uneven road surface.

When controlling the damping force of the damper, it is necessary to accurately detect the tension and compression velocity of the suspension, that is, the time change of the wheel stroke position. Enhancement of the mounting accuracy of the stroke sensors that detect the wheel stroke and suspension parts was studied to increase the damping force control performance.

The stroke sensor is a rotation angle sensor that detects the wheel stroke position as a link angle, converts the angle into voltage, and outputs it. To accurately detect the wheel stroke position, it was decided that after assembling the vehicle at the factory, the output from the stroke sensors at the position where the suspension is fully tensioned is calibrated.

Also, in the case of an electronically controlled system such as an ADS, the system is checked to ensure that it is operating normally, after the vehicle is assembled. The operation of the system of the vertical acceleration sensors and stroke sensors shown in Fig. 2 can be checked when the vehicle is moving up and down. Accordingly, by moving the vehicle up and down, the operation check of the vertical acceleration sensors and stroke sensors in conjunction with the calibration of the stroke sensors was studied.

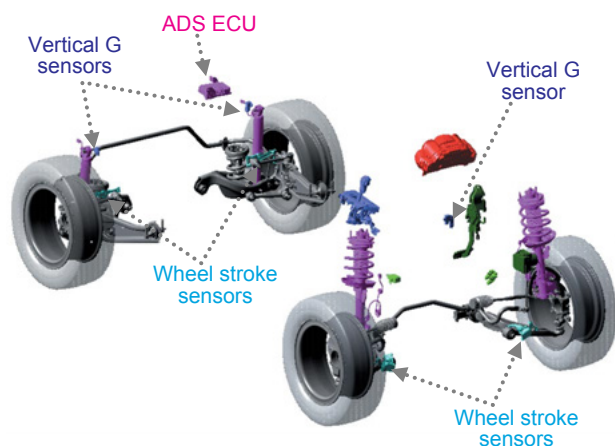


Fig. 2 ADS ECU and sensors

3. Stroke Sensor Calibration Technology

The following three methods of tensioning the suspension are used to calibrate the stroke sensors.

- (1) Raise the jack-up point underneath the vehicle.
It is necessary to use a hydraulically operated lift equipment of the type used for replacing four tires simultaneously. The vehicle is raised slowly, so checking the operation of the vertical acceleration sensors is a challenging task.
- (2) Use an excitation equipment that lifts up the wheels sharply. As the equipment moves the wheels up and down stably according to a program, the stroke sensors can be calibrated and the operation of the vertical acceleration sensors can be checked simultaneously. However, the equipment is expensive.
- (3) Drive the vehicle over a bump to move the wheels up and down. No additional equipment is necessary for this method and so it is inexpensive; the operation of the vertical acceleration sensors can also be checked simultaneously.

By comparing the above methods, it was decided to study the method of passing over a bump, which does not call for any equipment. In this concrete method, when the vehicle is driven over a bump, the wheel stroke of the suspension is fully tensioned, and the ADS electronic control unit memorizes the position where the suspension is maximally tensioned. Considering the situation that the workers drive a vehicle in various ways, the suspension should be stably tensioned at a wide range of vehicle speeds to establish this bump passing method.

3.1. Simulation Study

Figure 4 shows the results of simulating the wheel stroke change in this case that only a single frequency is contained in the height transition of the road surface when the vehicle is passing over a bump as shown in Fig. 3.

In the case of a single frequency in the height transition of the road surface, the range of the wheel stroke suspension displacement changes according to the vehicle speed. The bump to maintain the constant range of the suspension displacement at a wide range of vehicle speed is ideal. If the frequency contained in the bump height transition coincides with the sprung mass resonance frequency (1.3 Hz in the example described here), the height transition on the sprung mass and the wheel stroke change of the suspension can be stabilized to constant values with respect to the bump height. Consequently, a sweeping frequency bump (Fig. 5) where frequencies from lower than the sprung mass resonance frequency (1.3 Hz) to higher were swept and included in the bump height transition was devised. It was confirmed by simulation that the sweeping frequency bump was able to tension the suspension stably at a wide range of vehicle speeds compared to the case where the bump has only a single height frequency (Fig. 6).

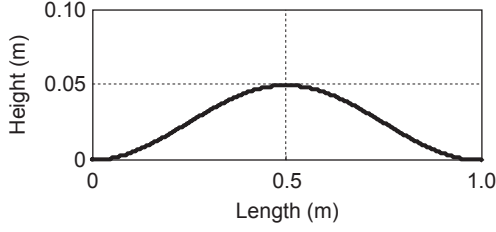


Fig. 3 Single frequency bump shape

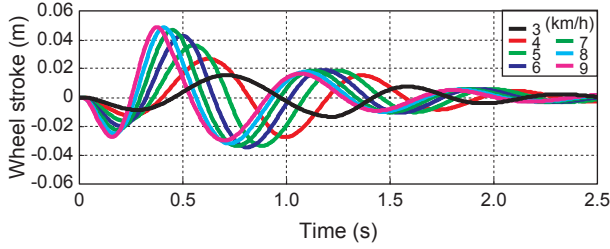


Fig. 4 Wheel stroke through single frequency bump (Simulation result)

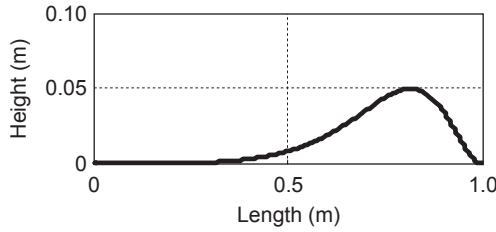


Fig. 5 Sweeping frequency bump shape

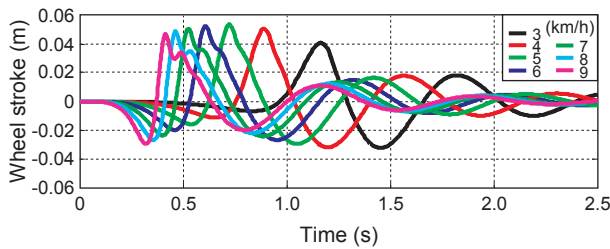


Fig. 6 Wheel stroke through sweeping frequency bump (Simulation result)

3.2. Study Using Actual Vehicle

A sweeping frequency bump was fabricated. And Fig. 7 shows the calibration results of the positions where the wheel stroke of the suspension is fully tensioned using the ADS electronic control unit as changing the vehicle speed when passing over the bump.

The numerical values measured when the vehicle is jacked up to tension the wheel stroke were almost equal to the numerical values calibrated by the electronic control unit when the vehicle was passing over a bump in the vehicle speed range between 7 and 15 km/h.

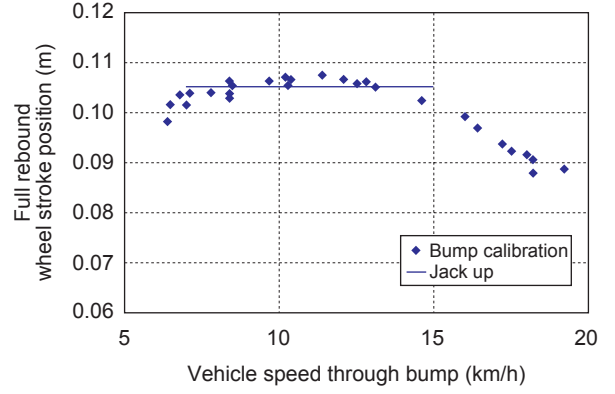


Fig. 7 Full rebound calibration result through sweeping frequency bump

The necessary time to pass over the bump is within a few seconds, and the stroke sensors can be calibrated in a short time. This method also has the advantage of enabling the operation of the stroke and vertical acceleration sensors to be checked, as described in the following section.

4. Sensor Operation Check Method

To confirm that the sensors are functioning normally, a mutual comparison of the stroke sensors and vertical acceleration sensors was performed when the vehicle passed over the bump. The wheel stroke and vertical acceleration were converted into sprung mass vertical velocity which is the same physical quantity (Fig. 8). The band-pass filter is effective in extracting the sprung mass resonance frequency and canceling the offset value of the sensors.

If the frequency contained in the height transition of the road surface when the vehicle is passing over the bump is lower than the sprung mass resonance frequency, the sprung mass vertical velocity is generated, but stroking ceases to occur. And at frequencies that are higher than the sprung mass resonance frequency, stroking occurs, but the sprung mass vertical velocity decreases. In order to move both the stroke displacement and sprung mass height transition with good balance, the bump should be set to

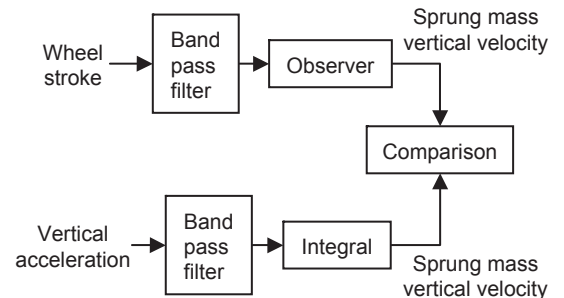


Fig. 8 Sensor comparison checking method

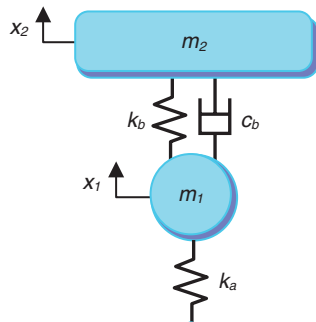
move the tires up and down at the sprung mass resonance frequency. A sweeping frequency bump enables the sprung mass resonance frequency to be included in the bump height transition at a wide range of vehicle speeds, so it is suitable in performing a mutual comparison of the stroke sensors and vertical acceleration sensors.

The sprung mass vertical velocity is obtained by integrating the vertical acceleration. To obtain the sprung mass vertical velocity from the stroke sensors, an observer of modern control theory (Fig. 10) was configured in the states where x_1 and the first order differential of x_1 , and x_2 and the first order differential of x_2 with respect to the suspension model (Fig. 9) which consists of the sprung mass and unsprung mass⁽²⁾.

The equations of motion for the suspension model are Eqs. (1) and (2).

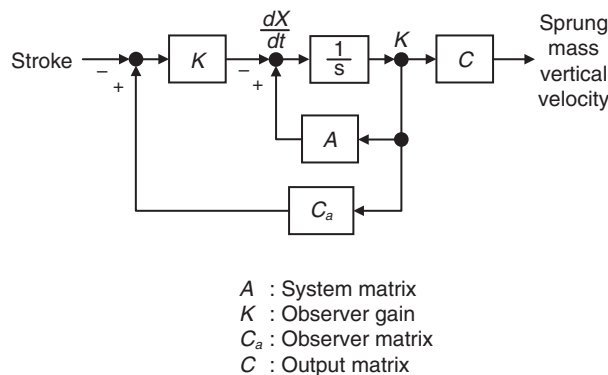
$$m_1 d^2 x_1 / dt^2 = k_b (x_2 - x_1) - k_a x_1 + c_b (dx_2 / dt - dx_1 / dt) \quad (1)$$

$$m_2 d^2 x_2 / dt^2 = k_b (x_1 - x_2) + c_b (dx_1 / dt - dx_2 / dt) \quad (2)$$



m_1 : Unsprung mass
 m_2 : Sprung mass
 k_a : Tire vertical stiffness
 k_b : Suspension stiffness
 c_b : Suspension damping coefficient
 x_1 : Unsprung mass vertical position
 x_2 : Sprung mass vertical position

Fig. 9 Suspension model



A : System matrix
 K : Observer gain
 C_a : Observer matrix
 C : Output matrix

Fig. 10 Observer

Figure 11 shows the results of that the electronic control unit of ADS calculates the vertical velocity of the vehicle from the output signals of the vertical acceleration sensors and stroke sensors when the vehicle passes over the bump.

Malfunctions in the sensor output gain can be found by performing a mutual comparison of the vertical velocity of the vehicle. Figure 12 shows the calculated results of the vehicle vertical velocity in the case where the gain dropped due to a breakdown in the signal output circuit of the stroke sensor.

Malfunctions in the sensor output gain can be judged by the four vertical velocities of the right and left stroke sensors and right and left vertical acceleration sensors. The right and left wheels pass over the bump at the same time, so the four vertical velocities should be the same. Compare the time integral of the absolute value of the vertical velocity obtained from the right stroke sensor whose output gain has fallen with the mean value of the time integral of the absolute values of the vertical velocity obtained from the other three sensors to judge the gain malfunction. When two sensors are compared for one wheel, malfunctions in the output gain can be judged, however, the malfunctioning sensor cannot be determined. The four sensors are compared by the right and left

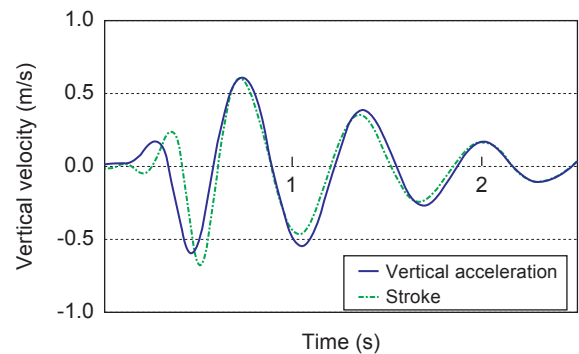


Fig. 11 Sprung mass vertical velocity comparison through bump

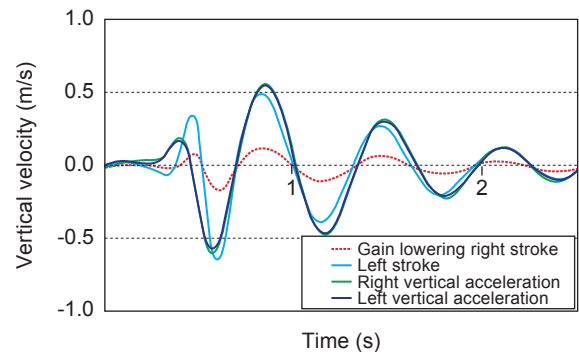


Fig. 12 Sensor gain lowering in sprung mass vertical velocity comparison

wheels, therefore, the sensor with the malfunctioning output gain can be determined by majority rule.

As a result of the above mutual sensor comparison technology, the operation of the sensors can be checked quantitatively without relying on the sensory evaluation by the workers.

5. Case of Application

A case in which this technology was used in the actual process is described here. The worker drives the vehicle and stops momentarily before the bump (Fig. 13). Then the worker starts the inspection using a shipping inspection tester (Fig. 14) and passes over the bump. The electronic control unit of the ADS memorizes the position where the wheel stroke is fully tensioned, simultaneously compares the stroke sensors and vertical acceleration sensors mutually, and sends the results thereof to the shipping inspection tester. Only the bump and software of the electronic control unit are enough to calibrate the stroke sensors and to mutually compare the sensors, so it is possible to dispense with dedicated equipment in this process.

By changing the length of the bump, the speed to pass over the bump can be adjusted. The passing speed of the vehicle driven by a worker at the bump location should be investigated, and the length of the bump should be



Fig. 13 Sweeping frequency bump in vehicle build plant



Fig. 14 Shipping inspection tester

adjusted accordingly. Honda of Canada Manufacturing, which established this process, set the bump passing speed to between 7 and 15 km/h. To reduce the bump passing speed, shorten the length of the bump. As an example, the length of the bump shortened by 40% reduced the passing speed to between 4 and 9 km/h.

6. Conclusion

Sensor calibration technology and a method of checking sensor operation were developed for vehicles equipped with an active damper system.

- (1) In the applied case of this technology, the worker drives the vehicle over a sweeping frequency bump. When the vehicle passes over the bump, the electronic control unit of the ADS runs a software program to calibrate the sensors and check the sensor operation.
- (2) The suspension was tensioned stably at a wide range of bump passing vehicle speeds. The output of the stroke sensor at the position where the suspension was fully tensioned was calibrated, thus the damping force control performance was enhanced.
- (3) When the vehicle was passing over the bump, the outputs of the stroke sensors and vertical acceleration sensors were converted into the sprung mass vertical velocity which is the same physical quantity, and mutually compared. The operation of the sensors can be checked quantitatively without relying on the sensory evaluation of the workers.
- (4) The sensor calibration and the sensor operation check were completed in a few seconds without the need of any additional large equipment.

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Road Conditions and Engine Mount Displacement of Sao Paulo and Boston

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ABSTRACT

This paper describes the displacement of the engine mount when a vehicle was driven in Sao Paulo, Brazil, and in Boston, U.S.A. The both areas were characterized by large displacement inputs to the engine mounts. In Sao Paulo, the input in the normal speed range was larger than in Boston, and the frequency was about ten times as high. Also, a large peak that cannot be measured in other regions was found here.

This peak was due to the uneven roads and steep slopes in Sao Paulo.

1. Introduction

In addition to supporting the engine as a rubber vibration isolator, the engine mount is an important component that reduces the transmission of engine vibration to the vehicle body.

In the development stage, it is important to design sufficiently durable rubber and bracket after assessing the input on the mount so that the vibration isolator characteristics can perform well over an extended time. In particular, larger input values are important when designing strength and durability.

On the other hand, it has been determined that road conditions in different areas of the world have their own characteristic tendencies in displacement input on the engine mount. Therefore, it was decided to measure engine mount displacement on an actual vehicle in Sao Paulo, Brazil, a region with particularly larger input, and Boston, which has relatively larger input in the USA. The displacement measurement results were used to examine how the road conditions affect the displacement behavior.

2. Road Conditions

This section describes the road conditions in Sao Paulo and Boston. Generally, road surfaces in Sao Paulo are rough with the following characteristics.

- (1) There are many brick paved steep slopes with many

- intersections and frequent parking on slopes (Fig. 1(a)).
- (2) Residential roads are a mixture of brick and asphalt paving with deep undulations (Fig. 1(b)).
- (3) There are many wide drainage ditches with large and steep V shape (Fig. 1(c) and (d)).

Boston, which has rough road surfaces relative to other places within the U.S., has the following characteristics.

- (1) Many slightly distorted asphalt road surfaces (Fig. 2(a)).
- (2) Deep indentations due to many manholes (Fig. 2(b)).
- (3) Many road surfaces with deep undulations (Fig. 2(c)).
- (4) Streetcar tracks on road surfaces (Fig. 2(d)).

3. Displacement Measurement Method

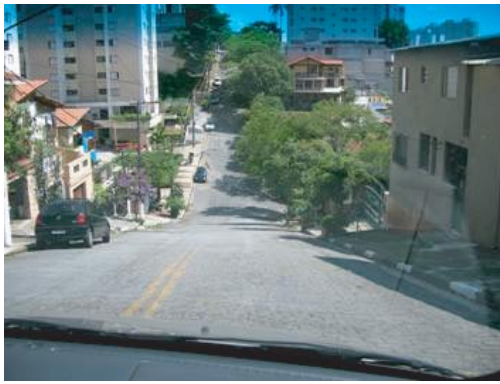
Displacement measurements were taken on routes of regular roads in urban, central areas, with one circuit measuring a few dozen kilometers in Sao Paulo and Boston. Highway driving was not included, as the routes were based on urban driving routes.

One of the methods of measuring engine mount input is to attach a strain gauge to the installation bracket. However, accurate measurements from low to high amplitude, behavior typical of rubber, are challenging to obtain by this method.

Therefore, to measure the input as directly as possible, potentiometer-type linear displacement transducers were installed (Fig. 3) near the bracket joints for displacement in both the up and down and front and rear directions to

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(a) Brick road steep slope



(b) Mixture of Brick road and asphalt paving with high waviness



(c) Large and wide drainage ditch with steep pitch



(d) Road surface at a drainage ditch in intersection

Fig. 1 Unique road surfaces in Sao Paulo



(a) Slightly distorted asphalt road



(b) Many indentations on road surface by manholes



(c) Undulating road surface



(d) Existence of streetcar track

Fig. 2 Unique road surfaces in Boston

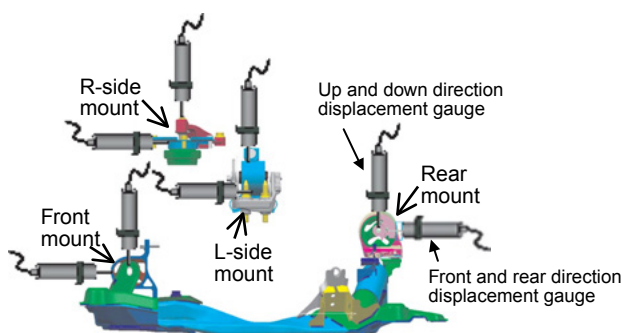


Fig. 3 Installation of engine mount displacement gauges

be measured, and the transducers were then synchronized. From this, the displacement that was generated by the input and obtained at the mount and bracket joint was considered the actual displacement, enabling displacement behavior in two dimensions to be assessed.

The displacement values measured with the displacement transducers were converted to load using the displacement-load curve that had already been measured for the mount, and was used in the actual development.

A 1.7L 4AT medium-size test vehicle was used for the measurements. The same test car was transported to the test locations to reduce individual vehicle variations and measurement driving tests were performed without making changes to, for example, the suspension settings.

4. Measurement Results

Figure 4 shows the displacement locus chart of each mount point when driving with the traffic flow on an urban route. The input direction of the mount displacement in San Paulo is about the same as that in Boston. In addition, it was discovered that the displacement of the rear mount was especially large, even though the frequency was low (Fig. 5).

When the driving conditions that generated this particularly larger displacement were analyzed a momentary tire skid was assumed to have been generated when starting the vehicle on a steep slope, and that the drivetrain generated momentary resonance and larger displacement in response.

In Sao Paulo, there are many intersections on slopes where drivers often have to stop. In addition, the asphalt has come off in some areas of the road surface, and other areas have brick roads. Furthermore, when there is insufficient load on the front drive wheels on steep slopes, a momentary tire skid may be generated (Fig. 6).

Meanwhile, the input examination for each route has clarified that this larger input is not generated even when passing over large gaps in the road in Sao Paulo, such as over regular drainage ditches.

Then, the same measurements performed with a medium-size vehicle were made with a small-size vehicle to check the variation between vehicles for larger input generation when starting on a steep slope. When a momentary skid was generated on the same steep slope on an urban route, the results showed the same characteristic peak for the mount displacement as shown

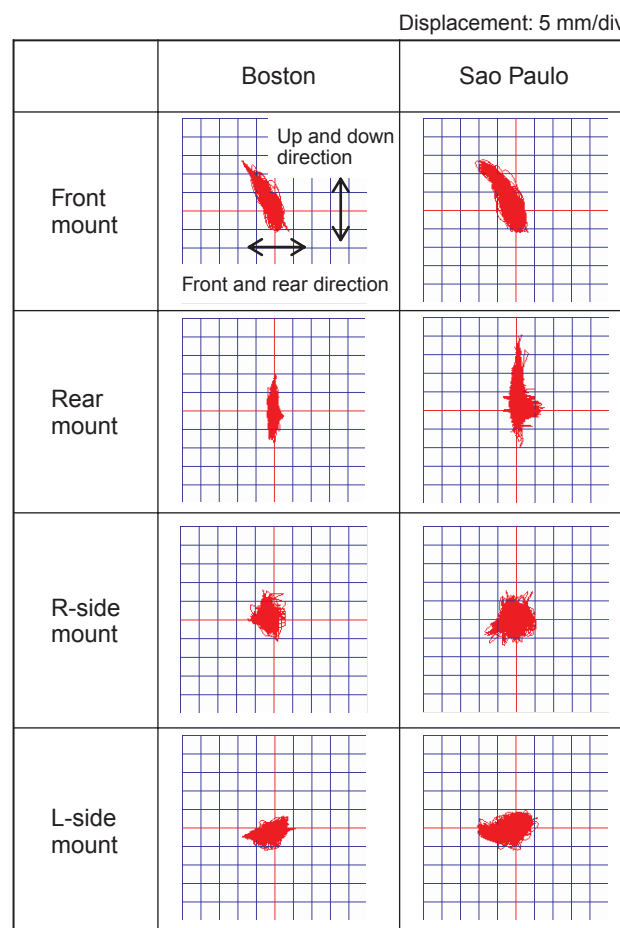


Fig. 4 Comparison of displacement between Sao Paulo and Boston

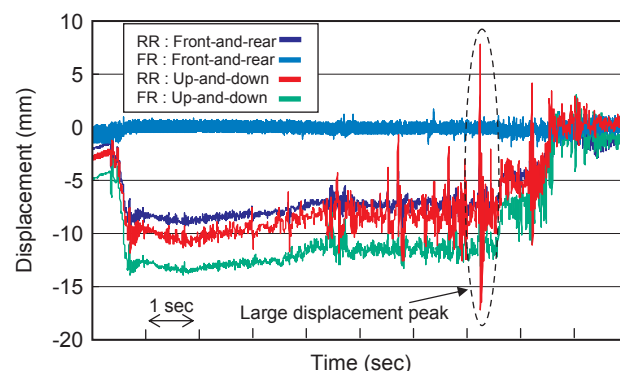


Fig. 5 Displacement of front and rear mount during starting with squeal on steep slope in Sao Paulo

in Fig. 5. However, the mount with the peak in the small-size vehicle was not the RR mount as in the medium-size vehicle, but the largest input was shown by the L-side mount (Fig. 7). This is likely to be because of the variations in the characteristic values of the spring-mass system due to differences in the vehicles types and layouts of the engine, drive system and engine mount.

While driving for the measurements in Sao Paulo the driver felt the vehicle shaking a lot because of the rough and undulating road surface conditions.

Therefore, the displacement frequencies of each mount were compared as follows: to perform the statistical distribution analysis for the displacement input the figures were multiplied by the number of circuits required if a total of 100 000 km were to be driven on the urban route, which had a distance of a few dozen kilometers. As a result of the comparison of all the displacement frequencies, the frequency in the Sao Paulo region was determined to be about 10 times higher than that in the Boston region (Fig. 8, Fig. 9).

In addition the larger displacement of approximately -18 mm, for the rear mount was generated by a skid while

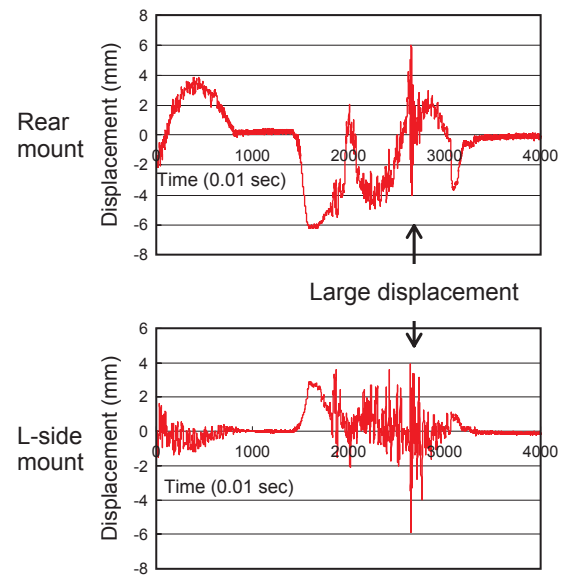


Fig. 7 Small car displacement of rear mount and L-side mount during starting with squeal on steep slope (up-and-down direction)



Example of road surface where squeal occurs in irregular section

Fig. 6 Typical steep slope in Sao Paulo

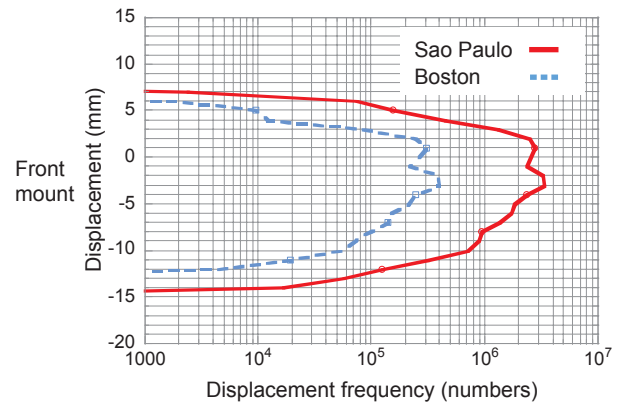


Fig. 8 Front mount comparison of displacement frequency for 100000 km between Sao Paulo and Boston (up-and-down direction)

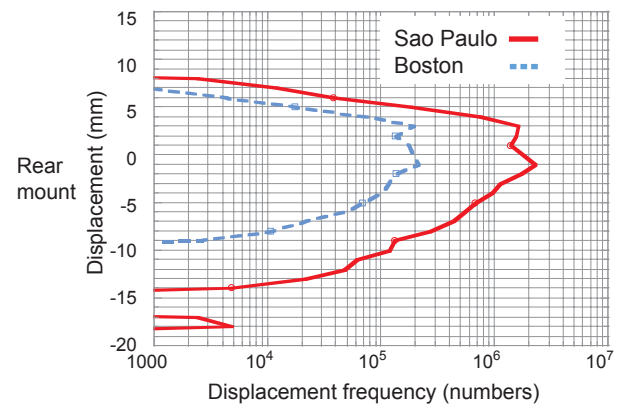


Fig. 9 Rear mount comparison of displacement frequency for 100000 km between Sao Paulo and Boston (up-and-down direction)

driving on a steep slope, however, the occurrence frequency was lower.

The following reasons may help clarify why the input frequency in Sao Paulo is higher than that in Boston.

- (1) Many locations of exposed brick road exist.
- (2) At some locations, the asphalt paving on the brick road has come off or asphalt paving is incomplete, so the road is an uneven mixture of brick road and asphalt. The many inputs are generated when passing over the boundaries between these areas.
- (3) Drainage ditches are everywhere and the vehicles often enter the ditches at normal speed, generating a larger input.
- (4) There are many intersections at the end of downhill slopes, at which the vehicles are likely to contact the ground because of slope differences between the road and drainage ditches.

5. Conclusion

It was determined that larger input existed for engine mounts in the market due to road surface influences when driving.

In particular, in the Sao Paulo region, resonance was found to occur between the engine and drivetrain due to tire skids when starting on steep slopes, causing some mounts to generate a larger displacement.

Durability is an important factor in the design of an engine mount, so larger inputs should be incorporated into the measurement data. For this reason, it is important to assess actual inputs in the market.

Different vibration test conditions in the unit durability evaluation of mount units were set for different regions during development, including the results of this examination. Particularly for Brazil, in addition to the conventional vibration test conditions, vibration test conditions to match input equivalent to that when starting on steep slopes were set as needed.

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Research on Extended Expansion General-Purpose Engine (Part 2) – Heat Release and Brake Performance –

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ABSTRACT

The combustion and efficiency of an extended expansion general-purpose engine employing a multiple linkage system was compared to that of a conventional engine consisting of the same component parts with the exception of the linkage system.

The results show that this extended expansion increases indicated thermal efficiency from 29.6% to 33.2%. In addition, as the piston speed near the compression top dead center of this engine is slow, mass fraction burned depends upon the change in the cylinder volume with respect to the capacity of the combustion chamber during the combustion period. The change of the minimum spark advance for best torque (MBT) timing relative to load is smaller than that of the conventional engine. Brake thermal efficiency of the test engine was increased by approximately 13%, from 20.1% for the conventional engine to 22.7%, under the EPA mode with fixed ignition timing operation.

1. Introduction

As concerns mount over global warming and the depletion of fossil fuels, fuel costs are also rising due to the recent surge in oil prices, and as a result, demands are growing for enhanced thermal efficiency, even for work equipment with general-purpose engines. The authors previously reported⁽¹⁾ the theory and overview of the mechanism of an extended expansion engine that realizes an Atkinson cycle through the use of a piston-crank system with a multiple linkage system as shown in Fig. 1. In that study, an actual prototype engine was manufactured, and the indicated thermal efficiency was compared with that of mass-produced engines in the same power class. However, the bore, spark plugs and valve arrangement of the mass-produced engines used for comparison differ from those of the test engine. The difference in the indicated thermal efficiency therefore included the difference due to these factors, in addition to the effects of extended expansion. The valve train also differs, which is thought to produce a difference in mechanical loss, so a comparative evaluation of brake performance was not achieved.

For this study, a prototype conventional engine with the same structure and combustion chamber shape as the

test engine was manufactured, and the performance of this conventional engine was compared with a multiple linkage extended expansion engine under roughly the same conditions to verify the effects of extended expansion in

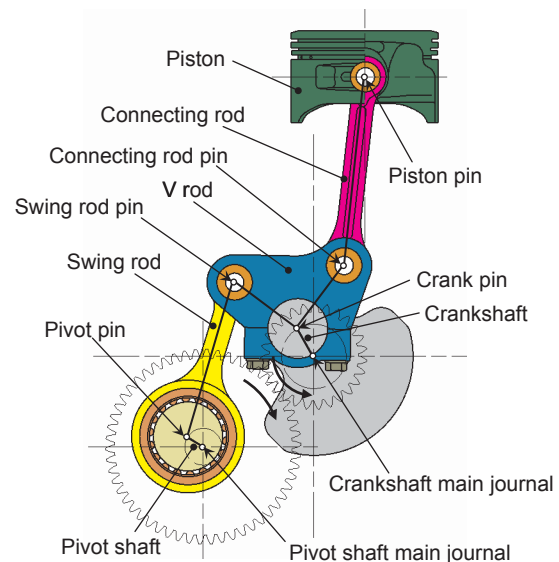


Fig. 1 Configuration of multi-link system

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increasing the indicated thermal efficiency. In addition, this paper also reports the brake performance of the test engine.

2. Test Method

2.1. Test Engine Specifications

Figure 2 shows an external view of the previously reported multiple linkage extended expansion engine, and Table 1 lists the main specifications of this test engine and of the prototype conventional engine used for comparison.

The comparison assumed the case with an extended expansion for the conventional engine, and the stroke volume of the conventional engine was therefore set for the maximum output power to be roughly equal for both engines.

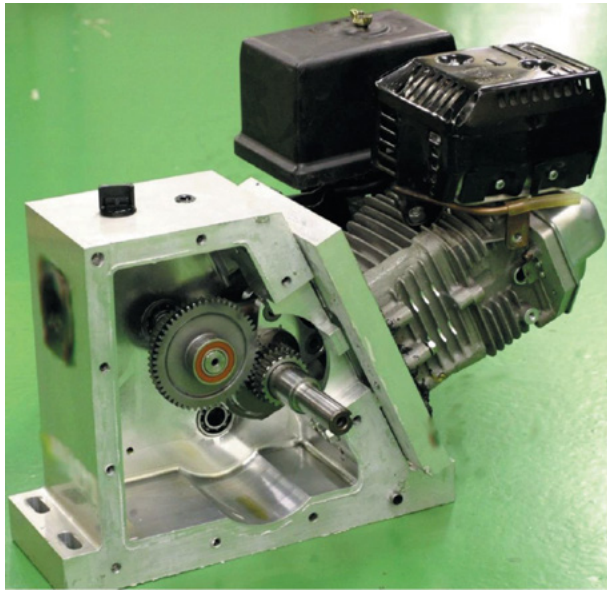


Fig. 2 External view of test engine

Table 1 Engine specifications

		Extended expansion	Conventional
Engine type		Air-cooled 4 stroke OHC single-cylinder spark-ignition	
Bore (mm)		64	64
Stroke (mm)	Intake and compression	42.1	51.0
	Expansion and exhaust	63.3	
Displacement (cm ³)	Intake and compression	135	164
	Expansion and exhaust	204	
Combustion chamber volume (cm ³)		18.1	21.9
Compression ratio		8.5	8.5
Expansion ratio		12.3	
Fuel		Gasoline	
Air-fuel ratio		12.5	

Both engines had the same bore and combustion chamber shape, and the combustion chamber volume was adjusted to match the compression ratios by inserting shims into the bottom of the cylinders. Identical parts were used for all structural parts, such as the cylinder head, spark plugs, inlet and exhaust valves, valve train, pistons and cylinder block, as well as for all intake and exhaust system auxiliary parts, such as the air cleaner, carburetor and muffler.

In addition, the test extended expansion engine uses a multiple linkage system, so the crank angle between the top and bottom dead center positions is not precisely 180° (Fig. 3). Therefore, the exhaust valve open timing and inlet valve close timing were set using the crank angle at the bottom dead center point of each piston as the reference for $\Delta\theta$ to be the same for both intake and exhaust as shown in Fig. 4.

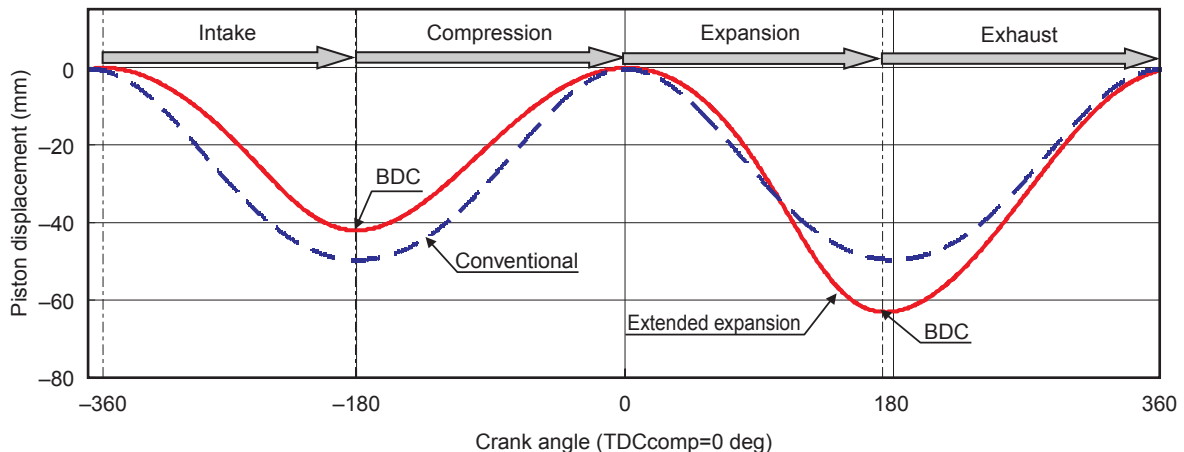


Fig. 3 Piston displacement of test engine

2.2. Mean Effective Pressure of Extended Expansion Engine

The mean effective pressure is generally used to compare the performance of engines with different stroke volumes. However, the mean effective pressure is generally defined under the condition that the each stroke volume is equal as expressed by Eqs. (1) and (2).

$$IMEP_{net} = \frac{W_i}{V_d} \quad (1)$$

$$BMEP = \frac{W_e}{V_d} \quad (2)$$

Here, W_i represents the indicated work, W_e the brake work, and V_d the stroke volume.

This study compared engines with different stroke volumes for each stroke, so the indicated mean effective pressure ($IMEP_{net}$) and the brake mean effective pressure ($BMEP$) were defined as follows.

$$IMEP_{net} = \frac{W_i}{\oint |dV|} \quad (3)$$

$$BMEP = \frac{W_e}{\oint |dV|} \quad (4)$$

In Eq. (3), the indicated work W_i can be obtained from the P-V diagram area as shown below in the normal manner even during the extended expansion cycle^{(2), (3)}. (Fig. 5)

$$W_i = \oint p dV \quad (5)$$

W_i here represents the indicated work including the pumping loss as shown in Fig. 5.

2.3. Test Conditions

The test shown below was performed in conformance with the EPA exhaust emission control⁽⁴⁾ operating mode prescribed as the typical operating mode for general-

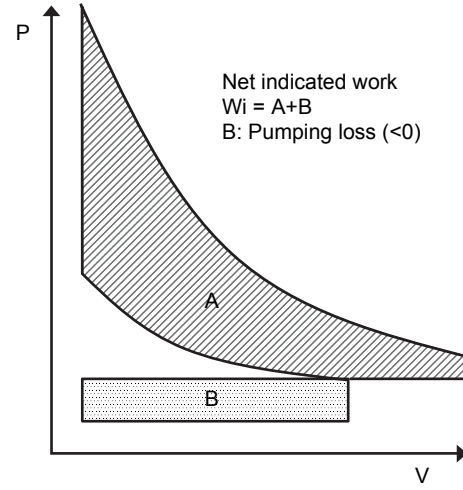


Fig. 5 Definition of net indicated work

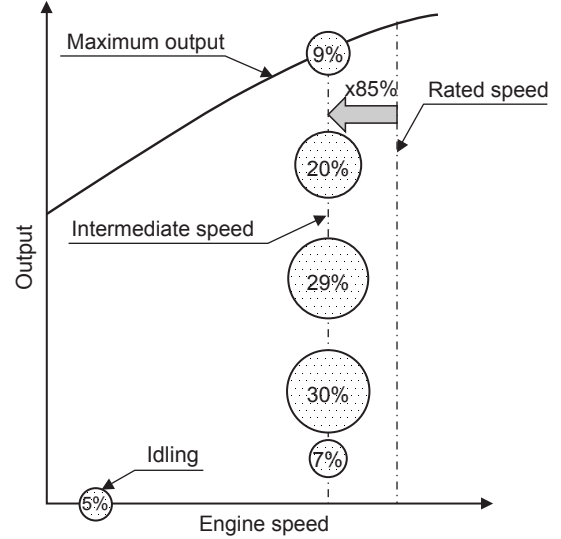


Fig. 6 Weighting factor of test cycle (subpart E of 40 CFR part 90)

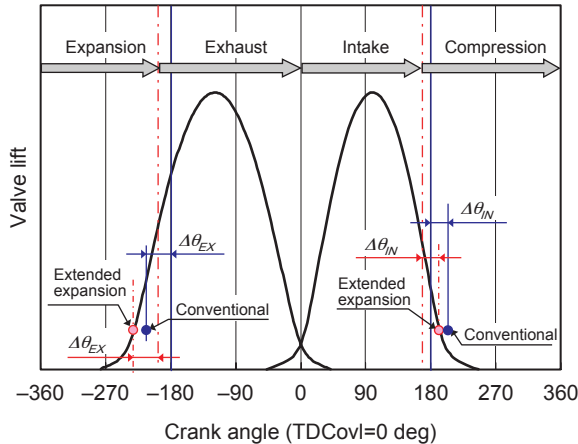


Fig. 4 Valve lift curve

purpose engines. This operating mode consists of six test cycles with different engine speeds and loads as specified in each circle in Fig. 6. The output, fuel flow and cylinder pressure were compared at an engine speed of 3060 rpm, which is prescribed as 85% of the rated speed. The stroke work was calculated by measuring the cylinder pressure using piezoelectric pressure sensors (Kistler 6041A) mounted on the cylinder head. Mode thermal efficiency was calculated by taking the weighted average of the results for each test mode using the weighting factor represented by the size of the circles in Fig. 6.

3. Test Results

3.1. Enhanced Indicated Thermal Efficiency

The indicated work and loss of each stroke during operation with wide open throttle (WOT) were compared

Table 2 Measured work and loss of each cycle

		Extended expansion	Conventional
Work or loss of each stroke (J/stroke)	Intake	-1.2	-3.6
	Compression	-34.5	-41.2
	Expansion	186.3	190.4
	Exhaust	-4.5	-2.9
Total work (J/cycle)		146.1	142.7
Supplied Heat (J/cycle)		439.5	481.8
Indicated efficiency (net)		33.2%	29.6%

for the test engine and conventional engine to verify the effects of extended expansion in increasing the indicated thermal efficiency. Table 2 shows the work per cycle, which is the total of the indicated work and loss of each stroke, the amount of heat supplied per cycle as obtained from the amount of fuel consumed, and the indicated thermal efficiency.

The total amount of work is approximately the same for both engines, but less heat is supplied to the test engine. This indicates that an equal amount of indicated work is obtained from less fuel. The indicated thermal efficiency of the test engine is 33.2%, which is an increase of 3.6 points over the 29.6% for the conventional engine. This corresponds to an enhancement of approximately 12% compared to the conventional engine.

Focusing on the loss within the cycle, the exhaust stroke loss of the test engine increased by 1.6 J compared to that of the conventional engine, but the intake stroke and compression stroke loss were 9.1 J lower than those of the conventional engine. This is thought to be due to the following reasons. The expansion stroke of the test engine is longer than that of the conventional engine and

the exhaust stroke loss increases, but the expansion ratio is higher, resulting in a lower exhaust pressure and smaller increase in loss. On the other hand, the test engine has shorter intake and compression stroke lengths, so the loss during both strokes is reduced. This led to the result where the total loss for the test engine was lower than that of the conventional engine.

The pumping loss of both engines was compared using the pumping mean effective pressure (PMEP) to evaluate the intake stroke and exhaust stroke loss at each load. As shown in Fig. 7, the PMEP of the test engine decreases over the entire load range, indicating that extended expansion effectively reduces loss, even in the lower load range where throttling loss occurs due to the throttle valve.

3.2. Combustion Characteristics of Multiple Linkage Engines

There are a number of reports concerning the relationship between piston speed and combustion characteristics of reciprocating engines⁽⁵⁾⁻⁽⁷⁾, such as on the degree of constant volume.

The authors' previously reported research compared the cylinder pressure of a multiple linkage extended expansion engine and conventional engines using numerical simulations. The multiple linkage engine was predicted to have a higher maximum cylinder pressure (P_{max}) than the conventional engine, and this was said to be due to the change in piston speed near the top dead center⁽¹⁾. Figure 8 compares the volumetric change inside the cylinder near the compression top dead center for the multiple linkage engine and conventional engine. The horizontal axis shows the crank angle, and the vertical axis shows the ratio of the cylinder volume to the combustion chamber volume (volumetric ratio). The multiple linkage engine exhibits slower volumetric change, so assuming identical heat release relative to the crank

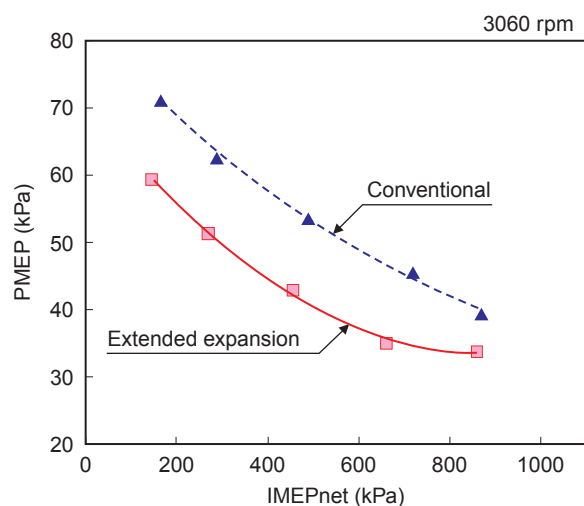


Fig. 7 Pumping mean effective pressure

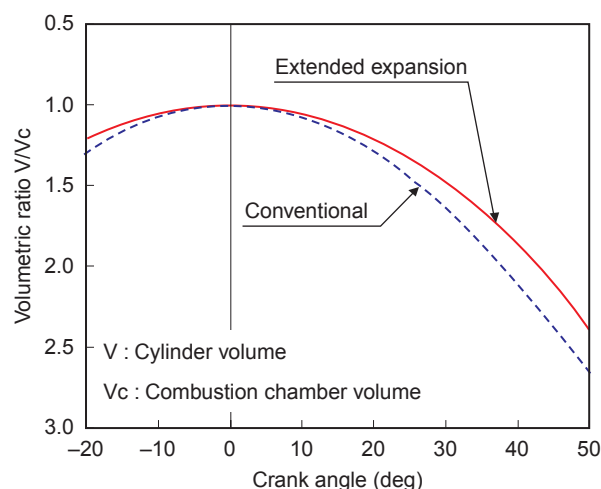


Fig. 8 Volumetric change during combustion

angle, a larger P_{max} value can be expected. Figure 9 shows the change in the cylinder pressure measured during operation with WOT at minimum spark advance for best torque (MBT) timing. Contrary to expectations, however, the P_{max} value for the multiple linkage engine was lower than that of the conventional engine.

The mass fraction burned (MFB) relative to the crank angle was compared to investigate this discrepancy. (Fig. 10) Compared to the conventional engine, the multiple linkage engine exhibited slower combustion from 10% mass fraction burned and onward, so the main combustion period (10 to 90% MFB) was 8° longer and the heat release of the two engines differed. On the other hand, comparing the mass fraction burned of both engines over

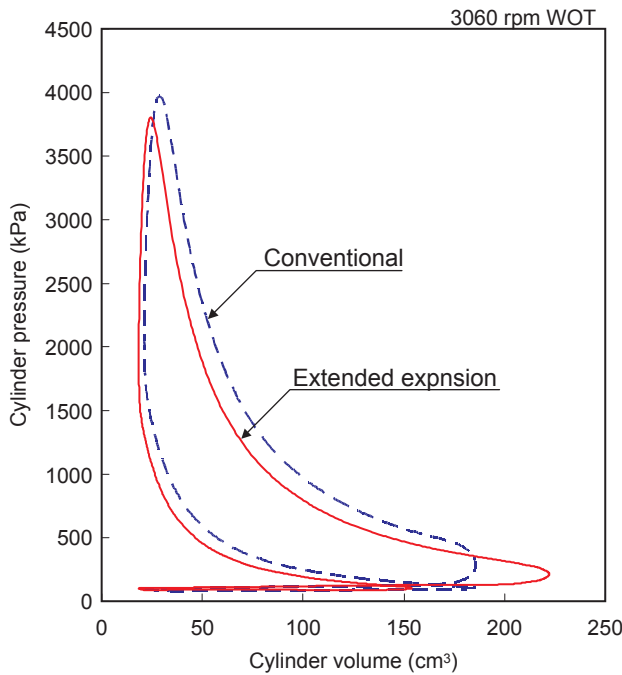


Fig. 9 Measured P-V diagram

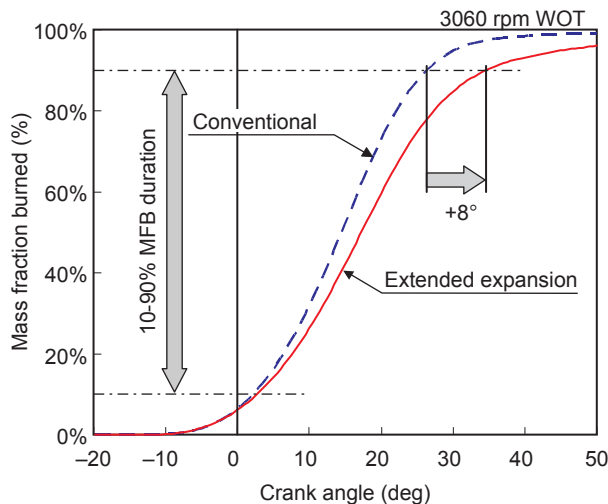


Fig. 10 MFB versus crank angle

the same period relative to the change in the above-mentioned volumetric ratio (Fig. 11) showed that the tendencies match over most of the main combustion period.

Based on these results, the assumption previously reported does not hold, and the actual heat release can be judged to be highly dependent on the volumetric ratio.

3.3. Ignition Timing Characteristics

The fact that heat release is governed more by the volumetric ratio than the crank angle means that the ignition timing characteristics that serve as the crank angle reference also differ from the conventional engine. Figure 12 compares the output at WOT when the ignition timing is changed relative to the MBT timing. According to this diagram, when the ignition timing is changed, the change in output is less for the multiple linkage engine than for the conventional engine.

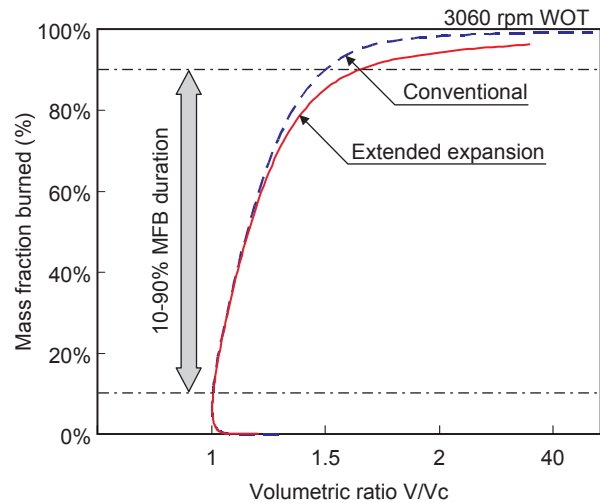


Fig. 11 MFB versus volumetric change

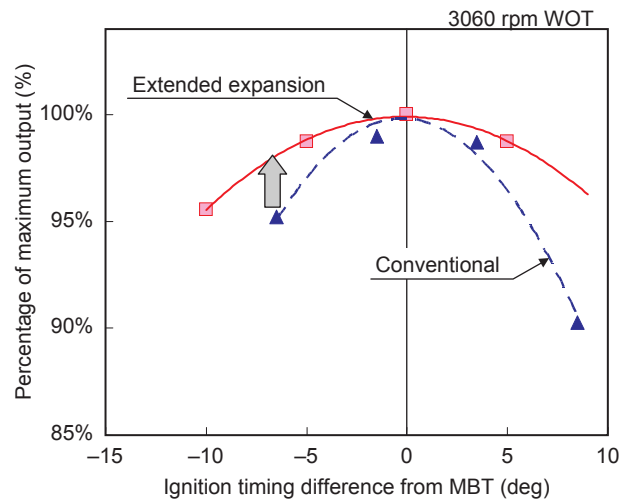


Fig. 12 Ignition timing characteristic at WOT

In addition, Fig. 13 shows the change in the ignition timing characteristics when the load is varied. The horizontal axis shows the load percentage, and the vertical axis shows the difference in MBT at each load from the MBT at WOT (MBT_{WOT}). The combustion period generally becomes longer during operation at lower loads, so the MBT timing advances and the difference relative to MBT_{WOT} becomes larger as the load becomes lower. However, this diagram shows that this tendency is less for the multiple linkage engine than that for the conventional engine.

The fact that the MBT timing at lower loads does not change much from when the throttle is wide open means that enhanced thermal efficiency at lower loads can be expected for general-purpose engines that do not usually have an ignition timing control device to simplify the ignition system. To examine this, the indicated thermal efficiency was compared at a 25% load (Fig. 14), which has the largest test cycle weighting factor of 30% within the EPA mode (Fig. 6). When operated with the ignition

timing fixed to the MBT_{WOT} for each load, the test engine efficiency was 2.1 points higher than that of the conventional engine. In contrast, when operated with the variable ignition timing of the MBT for each load, the indicated thermal efficiency increased by only 1.6 points. This confirmed that a multiple linkage engine using this system has combustion characteristics suitable for general-purpose engines that are operated with fixed ignition timing.

3.4. Brake Thermal Efficiency and Thermal Efficiency in EPA Mode

Figure 15 compares the relationship between the brake mean effective pressure and brake thermal efficiency for the test engine and conventional engine. The brake thermal efficiency of the extended expansion engine, operated with the ignition timing fixed to the MBT_{WOT} , is higher than that of the conventional engine, even in the lower load range. Figure 16 shows the brake thermal efficiency for the EPA mode. The mode thermal efficiency for fixed ignition timing operation is 22.7%, which is 2.6

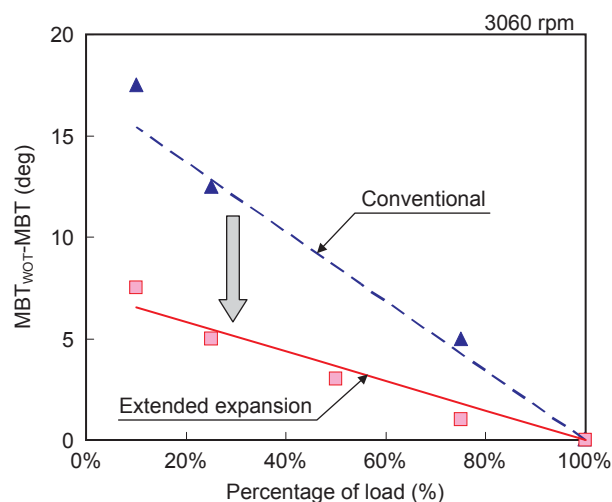


Fig. 13 MBT difference from MBT_{WOT}

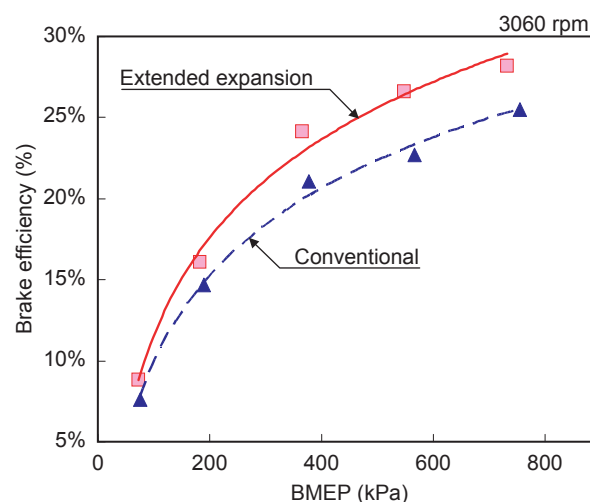


Fig. 15 Brake efficiency ($\theta_{ig}=MBT_{WOT}$)

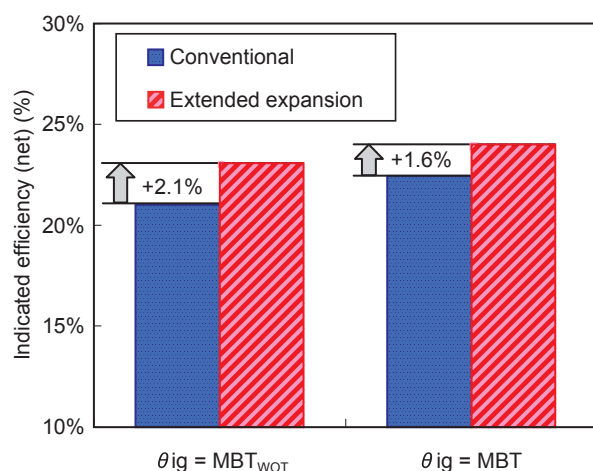


Fig. 14 Indicated efficiency (net) at 25% load

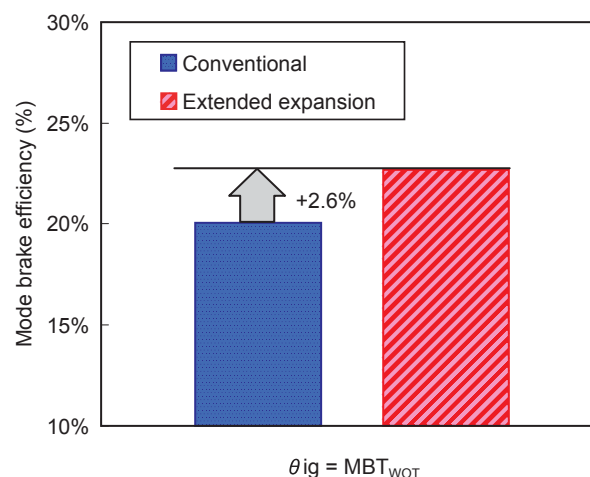


Fig. 16 Mode brake efficiency comparison

points higher than the 20.1% for the conventional engine. This corresponds to an increase in fuel economy of approximately 13%. These results showed that the test engine has better thermal efficiency than that of the conventional engine in these cases, and that the increase in the indicated thermal efficiency due to extended expansion leads to an increase in the brake thermal efficiency.

4. Conclusion

The performance of an extended expansion engine employing a multiple linkage system was compared to that of a conventional engine consisting of the same component parts with the exception of the linkage system, and the following results were obtained.

- (1) The indicated thermal efficiency of the test engine with WOT was 33.2%, which was an increase of 3.6 points over the 29.6% for the conventional engine. In addition to the increase in indicated thermal efficiency due to the increased expansion ratio, this was also due to a reduction of the intake stroke loss.
- (2) The heat release of the test engine depends on the change in cylinder volume during the combustion period relative to the combustion chamber volume. The piston speed near the compression top dead center of this engine is slow, so the heat release of the test engine relative to the crank angle is slower than that for the conventional engine. But the heat release relative to the cylinder volumetric ratio matches closely with the conventional engine over the main combustion period.
- (3) The change in MBT timing relative to the load is smaller for the test engine than that of the conventional engine. As a result, the mode thermal efficiency during fixed ignition timing operation under the EPA mode increased by 2.6 points from 20.1% in the conventional engine to 22.7%. This corresponds to an increase in fuel economy of approximately 13%.

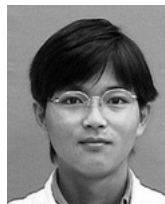
As mentioned above, extended expansion general-purpose engines that employ a multiple linkage system are a technology that can contribute to increasing the efficiency of general-purpose engines. However, a number of issues for practical application remain, such as reducing the reciprocating part vibration that differs from that of conventional engines, establishing a strong and rigid design for each linkage component, and selecting weight reduction methods, materials and manufacturing methods based on these requirements. The authors intend to continue efforts to overcome these issues with the aim of further reducing the environmental load of general-purpose engines that are used in a wide range of work equipment.

Note that Fig. 6 on page 84 of the previous Technical Review⁽¹⁾ “Research on Extended Expansion General-Purpose Engine” contained an error. “ $\kappa = 3.1$ ” should be “ $\kappa = 1.3$ ”.

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Hibiki KOGA



Sei WATANABE

Study of Ignition Timing Control of Gasoline HCCI Engine

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ABSTRACT

Homogeneous Charge Compression Ignition (HCCI) combustion is a method of combustion in which ignition occurs without being dependent on the ignition of a spark plug, unlike flame spread combustion. There are many parameters involved in ignition, and it is a challenge to control the ignition timing to an optimum value. Accordingly, a two-part HCCI ignition timing simulation model was constructed, with one part corresponding to the exhaust and intake strokes that determine the internal EGR and air intake amount, and the other part corresponding to the compression stroke during heat release by a chemical reaction. The results obtained using this model agreed with the performance test results. From the ignition timing simulation model, a control algorithm was developed and applied to a 4-cylinder engine fitted with an electromagnetic valve train, and the ability to control the ignition timing to the value corresponding to the demand torque in HCCI combustion was confirmed.

1. Introduction

As concerns over environmental issues have increased, the reduction of CO₂ emissions has become an important issue for the automotive industry. Homogeneous charge compression ignition (HCCI) is a combustion method that enables multi-point ignition of a homogeneous lean mixture. The rapid combustion achieved in this method ensures a high degree of constant volume, pumping loss reduction, and higher brake thermal efficiency. In addition, low temperature combustion results in a level of nitrogen oxide (NO_x) emissions of only several ppm⁽¹⁾⁻⁽⁵⁾. Given these characteristics, a considerable amount of research is presently being conducted on this combustion method.

The HCCI is a self-ignition method different from those that initiate flame spread by means of spark plug ignition. A large number of parameters are involved in achieving conditions for the initiation of self-ignition, and optimum control of ignition timing to maximize the effect of combustion in terms of generating demand torque is therefore a significant technological issue when utilizing HCCI.

2. Research Goals

The research discussed in this paper focuses on the

role of the different engine strokes in HCCI combustion. An ignition timing simulation model was formulated by the parameters for ignition timing control clarified in HCCI tests using a single-cylinder engine. The goal of the research was to control ignition timing in response to demand torque by a control algorithm based on this model.

3. Ignition Timing Simulation

Figure 1 shows the structure of the HCCI ignition timing simulation model used in this research. The model

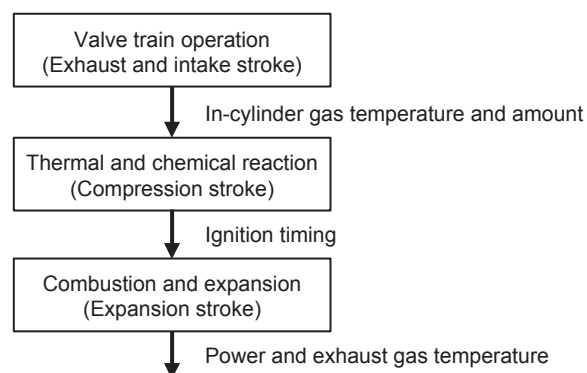


Fig. 1 Ignition timing simulation model

* Automobile R&D Center

Table 1 Specifications of single cylinder research engine

Bore × stroke (mm)	80 × 80
Displacement (cm ³)	402
Compression ratio	15
Valve diameter (mm) × number of valves	Intake 27.5 × 2 Exhaust 24.5 × 2
Fuel system	Port injection

consists of three parts: the intake and exhaust strokes, the compression stroke, and the expansion stroke. In the intake and exhaust stroke section of the model, the intake and exhaust valve train system controls the amount of internal EGR and intake air, and the in-cylinder gas temperature and inducted charge amount in the cylinders are determined. In the compression stroke section, the rise of temperature (ignition) due to the compression of the cylinder mixture (including the injection quantity) and the chemical reaction is modeled, and the ignition timing representing the initiation of the flame is determined. In the expansion stroke section, the work produced by the flame initiated by the ignition timing, as well as the exhaust temperature, are calculated. The models were determined as follows. The compression stroke model for determining the ignition timing was formulated on the basis of the results of HCCI tests conducted with intake air heating. The intake and exhaust stroke model, which determines the in-cylinder gas temperature and the charged amount of gas in the cylinders, was formulated on the basis of HCCI tests using the internal EGR and previously formulated compression stroke model.

Table 1 shows the specifications of the single cylinder engine used to confirm the characteristics of the HCCI combustion for the intake-air temperature and injection quantity. A 3-kW intake heater was positioned upstream of the intake port to enable HCCI tests with intake air heating.

3.1. HCCI with Intake Air Heating

When HCCI combustion is conducted with intake air heating, the charged gas is not heated by exhaust energy. Therefore, taking the intake conditions only into consideration, a model of the compression stroke in the HCCI combustion can be identified.

3.1.1. Results of tests with intake air heating

Figure 2 shows the heat release rate for crankshaft angles when the indicated mean effective pressure (IMEP) of the test engine was varied between 76 kPa and 327 kPa at an engine speed of 1000 rpm and an intake-air temperature of 200 °C. The heat release rate increases as the injection quantity increases. At the same time, the ignition timing advances, and the torque produced

increases. Figure 3 shows the indicated specific fuel consumption (ISFC) and NOx emission characteristics of the engine in relation to intake-air temperature and injection quantity. There is an intake-air temperature at which the ISFC for the IMEP is at its minimum, and the point at which the ISFC is at its minimum matches the point at which NOx emissions are at their minimum. From the perspective of ignition timing control, this indicates that the optimal control of in-cylinder gas temperature in response to demand torque can maximize thermal efficiency and can minimize NOx emissions at the same time.

3.1.2. HCCI combustion model

Assuming HCCI combustion modeled as a chemical reaction in a homogeneous mixture, the reaction speed can be expressed by Eq. (1) in accordance with the Arrhenius law.

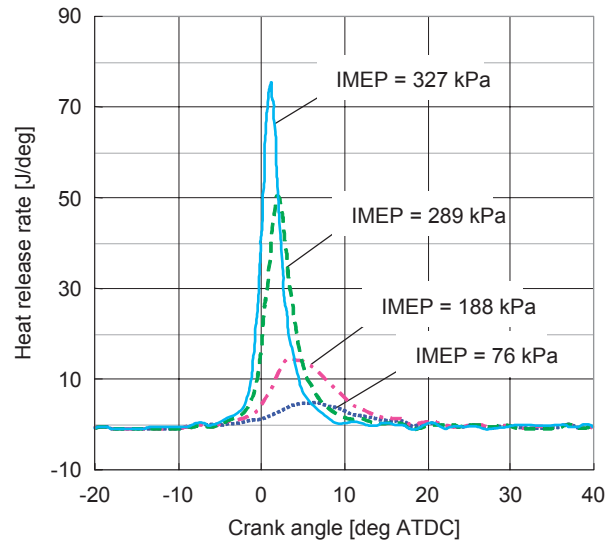


Fig. 2 Tendency of heat release pattern according to IMEP

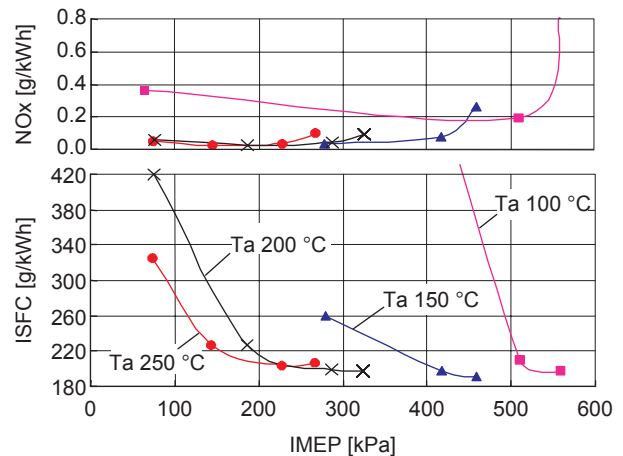


Fig. 3 ISFC and NOx characteristic of air intake heating HCCI

$$\frac{dN}{dt} = A[F][O]e^{-E^*/RT} \quad (1)$$

E^* : activation energy, A : frequency factor, $[F]$: mol number of carbon and hydrogen, $[O]$: mol number of oxygen, R : gas constant, T : mixture temperature

$[F]$ and $[O]$ are reduced in every reaction. With regard to the mixture temperature, if the reaction heat per reaction is q_0 and the cooling loss is dQ_c , the temperature change, dT , can be expressed by Eq. (2).

$$dT = \frac{q_0 dN - dQ_c}{C_v N_t} - (\kappa - 1) \frac{T}{V} dV \quad (2)$$

C_v : specific heat at constant volume, N_t : amount of gas in the cylinder, V : volume of cylinder, κ : ratio of specific heat

From Eqs. (1) and (2), the parameters controlling the HCCI combustion in an engine are the injection quantity, intake air amount, and in-cylinder gas temperature. For the initial conditions where the mol number of hydrocarbons which contributes to combustion from the injection quantity, the mol number of oxygen molecules which contributes to combustion from the intake-air amount, and the mol number of nitrogen molecules which does not contribute to combustion but only increases the temperature, the combustion model simulates the in-cylinder gas temperature by the compression and the chemical reaction. The reaction speed and reaction temperature parameters were identified to match the ignition timing output by this model and the results of engine tests. The ignition timing was defined as crankshaft angles of 2J/deg CA or above when the heat release rate clearly begins to increase.

3.1.3. Comparison with engine tests

Figure 4 shows the results for heat release rates obtained from the ignition timing simulation. The calculation conditions were identical to those used in the

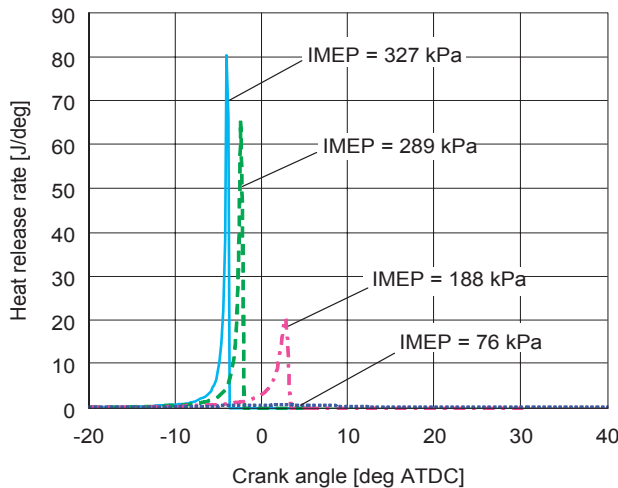


Fig. 4 Simulation result of heat release pattern

engine tests discussed previously (engine speed: 1000 rpm; intake-air temperature: 200 °C). The simulation reproduced the tendency displayed in the engine tests: as the injection quantity increased, ignition timing advanced and the amount of produced torque increased. Figure 5 shows a comparison of simulation results and engine test results when the intake-air temperature was varied. The ignition timing output by the simulation was within ± 5 deg CA of the ignition timing of the HCCI combustion in the actual engine.

3.2. Formulation of Ignition Timing Model using Internal EGR

Figure 6 shows the relation between engine control parameters and ignition timing model parameters for HCCI combustion using internal EGR. G_{fuel} represents the injection quantity, G_{air} represents the intake air amount, and T_{gas} represents the in-cylinder gas temperature. Because combustion gas remains in the cylinders when internal EGR is used, the mol number of carbon dioxide

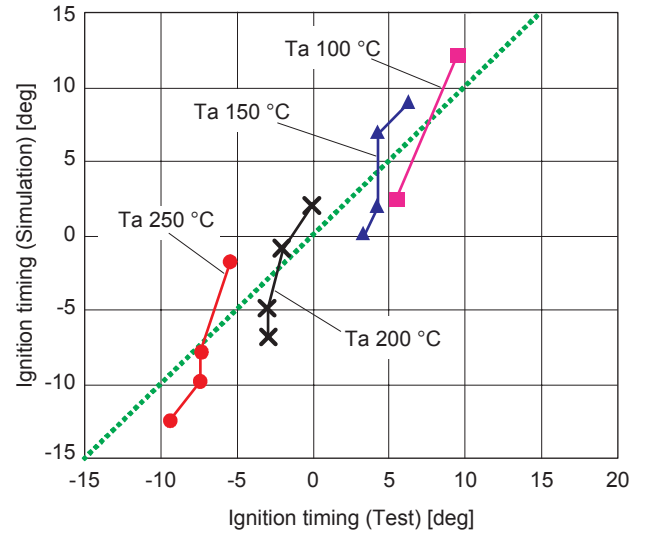


Fig. 5 Comparison of ignition timing simulation and test result

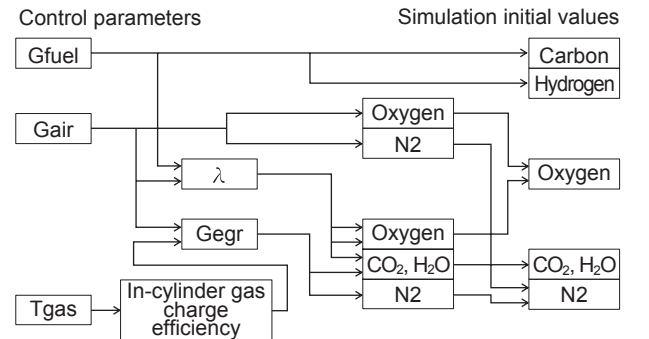


Fig. 6 Relationship between control parameters and ignition timing simulation parameters

molecules, oxygen molecules, and water vapor in the combustion gas should be considered. Because the mol number of oxygen molecules in the exhaust gas also contributes to ignition timing, an exhaust gas analyzer was used for measurement. Carbon dioxide and water vapor were treated as inert gases.

Figure 7 shows an outline of the method of determining the in-cylinder gas temperature, T_{gas} , in relation to the intake air and internal EGR. The intake air temperature, T_a , and the exhaust gas temperature, T_{ex} , are internally divided by the intake air amount, G_{air} , and the amount of internal EGR, G_{egr} , to give the in-cylinder gas temperature, which is used as the initial value of the combustion model. The in-cylinder gas temperature is determined by using Eq. (3).

$$T_{gas} = \frac{(T_{ex}G_{egr} + T_aG_{air})}{G_{egr} + G_{air}} \quad (3)$$

3.3. Comparison of Simulation and Test Results using Internal EGR

Valve timing and lift were modified for the single-

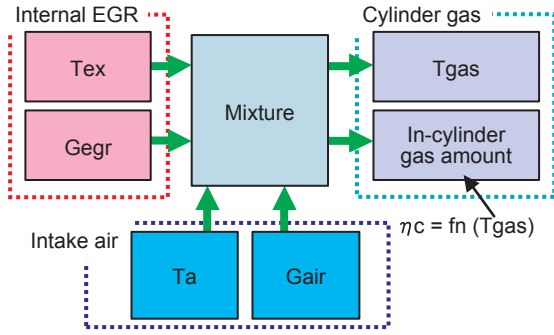


Fig. 7 Conceptual diagram of cylinder gas temperature and amount calculation

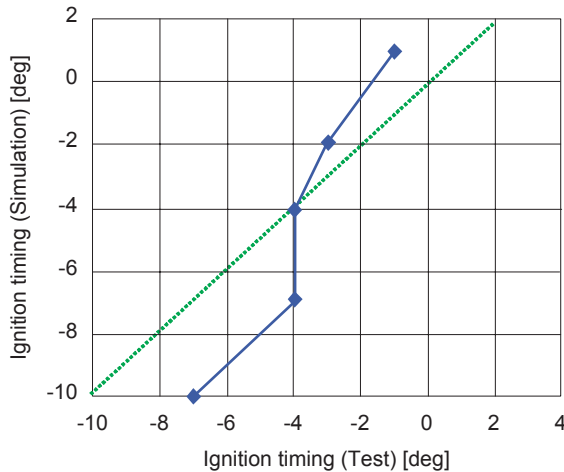


Fig. 8 Comparison of internal EGR HCCI ignition timing simulation and test results

cylinder engine mentioned above, and tests were conducted at 40% internal EGR. Various intake air temperatures at an engine speed of 1000 rpm and an IMEP of 300 kPa were studied.

As Fig. 8 shows, simulation results for ignition timing matched the test results within ± 3 deg CA or less even if internal EGR was employed.

4. Ignition Timing Control Algorithm

The ignition timing simulation model mentioned above was used to form an HCCI combustion ignition timing control algorithm. Figure 9 shows a block diagram of the control algorithm. The fundamentals of the HCCI ignition timing control system are calculating the injection quantity and the intake air amount from the demand torque, and concurrently calculating the target in-cylinder gas temperature as well as determining the amount of the internal EGR necessary to achieve the target temperature. The system also controls the valve close timing of the intake and exhaust valves to achieve the necessary intake air and internal EGR amounts.

4.1. Calculation of Target In-cylinder Gas Temperature

As indicated in Fig. 3, there are specific in-cylinder gas temperatures that produce optimum ignition timings for specific levels of demand torque. According to the Arrhenius law, when the engine is operating at high speed when a more rapid reaction speed is necessary, it will be necessary to increase the in-cylinder gas temperature, and when the engine is operating under a high load with high mol numbers of carbon and hydrogen, it will be necessary to lower the in-cylinder gas temperature to maintain the reaction speed at a fixed level. Figure 10 shows target in-cylinder gas temperatures for loads and engine speed parameters.

4.2. Method of Calculating Total Amount of Gas Charge in the Cylinders

The method of calculating the total amount of gas charge in the cylinders is as follows. The volume of the cylinder at the piston position when the intake valve is

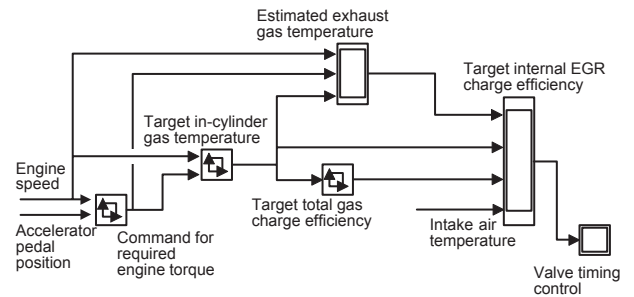


Fig. 9 Block diagram of ignition timing control for gasoline HCCI engine

closed can be measured. Assuming that the pressure in the cylinders is equivalent to the pressure in the intake manifold, and that the in-cylinder gas temperature matches the target temperature, the total amount of gas charge in the cylinders, including intake air and internal EGR, can be calculated from the gas equation of state.

Therefore, the target amount of gas charge in the cylinders, E_{tg} , can be calculated from the target in-cylinder gas temperature, T_{igas} , by using Eq. (4).

$$E_{tg} = \frac{\text{Constant}}{T_{igas}} \quad (4)$$

4.3. Exhaust Gas Temperature Estimation

As the calculation of the internal EGR temperature based on measurements of each exhaust gas temperature cycle would take considerable calculation time, this method could not be used in combustion control. Therefore, in this research, the exhaust gas temperature was estimated. Because the in-cylinder gas temperature in HCCI combustion is high at the commencement of the compression stroke, and the exhaust gas temperature is low in comparison to spark ignition combustion, the in-cylinder gas temperature has an effect on the exhaust gas temperature. Thus an exhaust gas temperature estimation logic, with consideration of the drop in temperature due to heat transfer from the walls, was used. The estimated exhaust gas temperature, T_{EX} , is calculated from the exhaust gas temperature at base engine speed T_{EX0} , cylinder wall temperature T_{EMPCYL} , and engine stroke cycle M_E , as shown in Eq. (5).

$$T_{EX} = T_{EX0} - AhM_E \frac{T_{EX0} - T_{EMPCYL}}{N_i C_p} \quad (5)$$

h : heat transfer coefficient, A : area of heat-transfer surface, N_i : amount of exhaust gas, C_p : specific heat at constant pressure

Figure 11 shows the results of tests conducted to verify the accuracy of the exhaust gas temperature estimation at different intake air temperatures. These

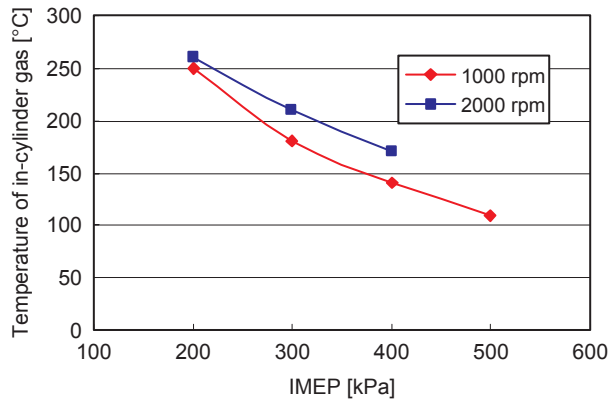


Fig. 10 Target temperature of in-cylinder gas for required torque

results show that the exhaust gas temperature could be estimated within ± 8 °C.

4.4. Method of Calculating the Necessary Amount of Internal EGR

The optimum amount of the internal EGR necessary for HCCI combustion in each engine operating state, N_{egr} , is calculated from the demanded in-cylinder gas temperature, T_{igas} , total amount of gas charge in the cylinders, E_{tg} , intake air temperature, T_a , and estimated exhaust gas temperature T_{EX} , by using Eq. (6).

$$N_{egr} = E_{tg} \frac{T_{igas} - T_a}{T_{EX} - T_a} \quad (6)$$

5. Test of Control Algorithm

Figure 12 shows the 4-cylinder HCCI engine with a variable valve timing system using an electromagnetic mechanism⁽⁶⁾, which also used the HCCI ignition timing control algorithm mentioned previously. Table 2 shows the specifications of this engine.

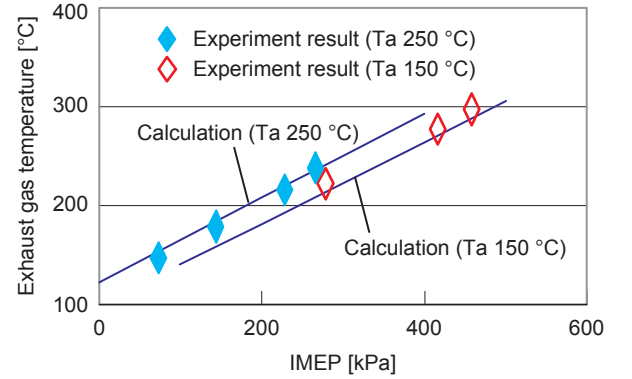


Fig. 11 Result of exhaust gas temperature estimation

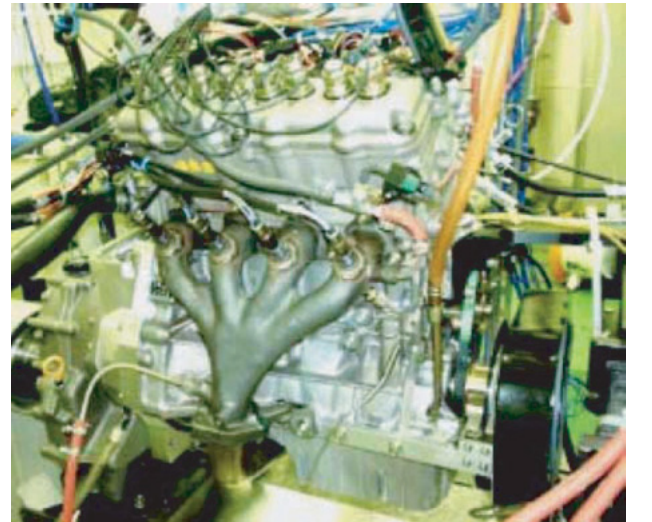


Fig. 12 Photograph of research 4-cylinder engine

Table 2 Specifications of research 4-cylinder engine

Bore × stroke (mm)	75 × 90
Displacement (cm ³)	1590
Compression ratio	15
Valve diameter (mm) × number of valves	Intake 30 × 2 (EMV) Exhaust 24 × 2 (EMV)
Fuel system	Port injection

5.1. Electromagnetic Valve Train Control

Figure 13 shows the outline of the electromagnetic valve train operation. As the internal EGR increases, valve control advances the close timing of the exhaust valve and delays the open timing of the intake valve at the same time.

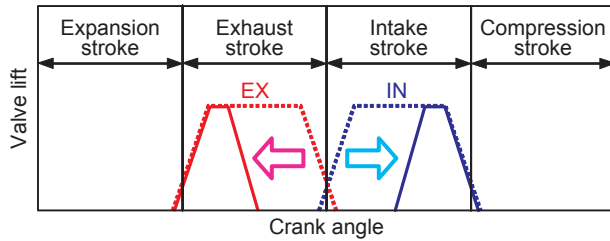


Fig. 13 Valve timing diagram of internal EGR HCCI condition

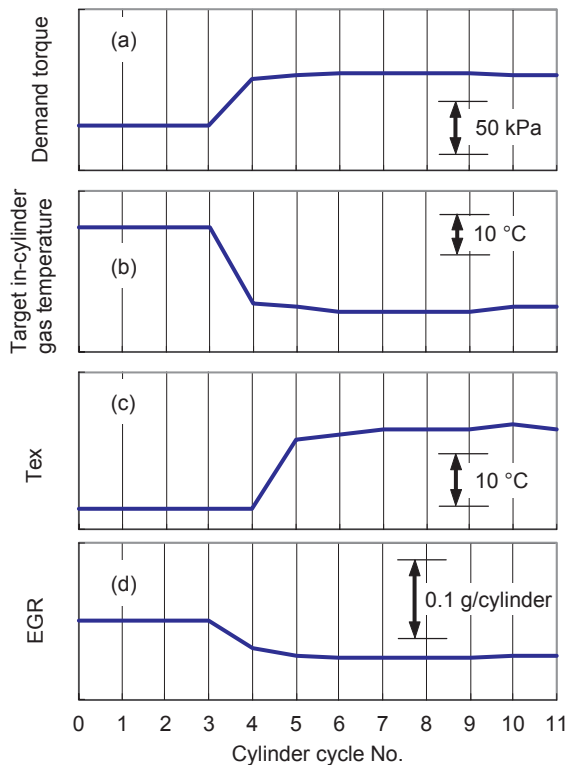


Fig. 14 Time chart of HCCI ignition timing control parameters of demand torque step

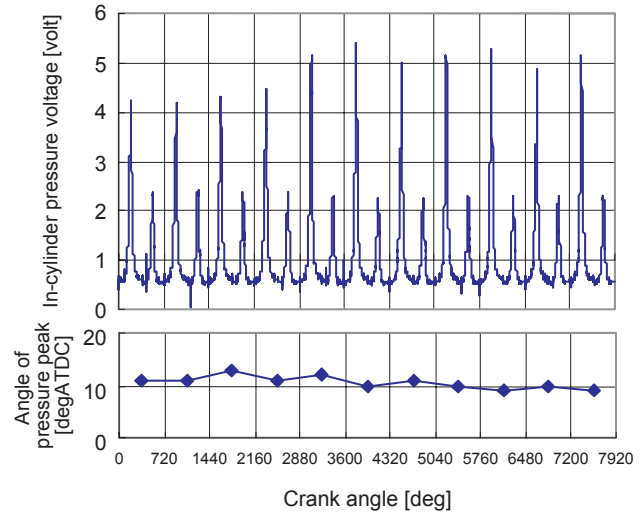


Fig. 15 Time chart of in-cylinder pressure and angle of pressure peak

5.2. Demand Torque Step Test

With the engine speed kept at 1500 rpm, torque responsiveness was tested by varying demand torque (BMEP) in steps between 300 kPa and 350 kPa.

Figure 14 shows changes in each control parameter. When demand torque is increased in Cycle No. 4 (a), the intake and exhaust valve timing functions to reduce the internal EGR (d) and to lower the in-cylinder gas temperature (b). Figure 15 shows the in-cylinder pressure waveform and the timing of pressure peaks for each cycle at this time. According to the in-cylinder pressure waveform of the fifth cycle, the peak value of the in-cylinder pressure changes in steps. The results also indicate that the timing of the in-cylinder pressure peaks can be constantly controlled to around an efficient ATDC 10 deg before and after torque changes.

6. Conclusion

This research was conducted on a gasoline HCCI ignition timing control algorithm focusing on the amount and temperature of in-cylinder gas when the intake valve is closed. The results are as follows.

- (1) An HCCI combustion ignition timing simulation model displayed an accuracy within ± 5 deg for both HCCI with intake air heating and HCCI using internal EGR, when compared to test results.
- (2) The ignition timing control algorithm of the HCCI combustion noted above was applied to a 4-cylinder engine with a variable valve timing system using electromagnetic mechanism, and a high level of responsiveness and high degree of accuracy in controlling ignition timing was confirmed for step changes in demand torque.

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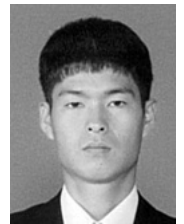
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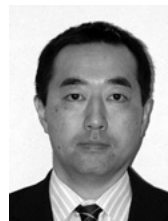
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Thermal Management of Air-cooled Motorcycle Engines Using Forced Oil-cooling System

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ABSTRACT

Engine cooling substantially affects power output, fuel economy, noise reduction and durability. Estimating each engine part's temperature in the early stages of design is important in the development of air-cooled motorcycle engines. In this study, engine temperature distributions were obtained through heat transfer coefficient calculations based on Computational Fluid Dynamics (CFD). From the analysis, it was found that by using air-cooling with a forced oil-cooling system, the cylinder head temperature sufficiently decreased, and the resultant temperature distribution was almost the same as that of water-cooled engines. It was also found that the temperature gradient among cylinders was manipulated by controlling the cooling-oil flow. Furthermore, heat loads were controlled as well, and the knocking limit was also enhanced when the engine was tested in a prototype motorcycle.

1. Introduction

Although customer needs with regard to motorcycles have diversified in recent years, motorcycles equipped with simple air-cooled engines are still popular. There is also high demand in the markets of developing countries for air-cooled engines, which have a comparatively lower number of parts and therefore lower costs.

An important issue with air-cooled engines is that the cooling depends on the air-flow generated by the moving motorcycle. Determining the total heat dissipation and correcting the temperature difference among cylinders in multi-cylinder engines are important factors that contribute to cooling performance. To pursue optimal engine cooling performance also means to pursue functional beauty in an air-cooled engine, which is also important from a design perspective. Conventionally, the development of air-cooled engines was based on the results of visualizing the cooling-air flow in wind tunnel tests and measuring the temperature in actual tests. Quantitative analysis of flow behavior and temperature forecasts at an early stage of engine design was considered too challenging. Controlling exhaust emissions and enhancing fuel economy have continued to become more important even for motorcycles, and controlling the engine temperature to

help achieve these objectives has become an important issue.

In this study, CFD analysis of the cooling-air behavior was performed to determine the cooling performance for an air-cooled engine. Some useful knowledge such as effective cooling-fin shapes and optimal positions for the oil cooler and air inlets was discovered⁽¹⁾. Secondly, the effectiveness of the oil-cooling system was studied to additionally enhance the cooling performance of air-cooled engines. To verify the effectiveness of the oil-cooling system, CFD analysis was used to obtain the local heat transfer coefficient for the cooling-air and the average heat transfer coefficient for the cooling-oil. These values were then used to calculate engine heat conduction and obtain a temperature distribution for each engine part. Note that the partial cell Cartesian Coordinate method^{(2), (3)} was used for the CFD analysis in this study. This method can be used in flow fields of various shapes and doesn't require many man-hours for mesh generation.

This paper describes the results of a theoretical study of an in-line four-cylinder, air/oil-cooled engine with a displacement of 900 cm³. These theoretical results were then utilized on a prototype motorcycle to test the effectiveness of using the forced oil-cooling system.

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2. Engine Outline

2.1. Concept of Air/Oil-cooled Engines

In the design concept stage of the new air-cooled engine, the development goal was to achieve the temperature distribution comparable to that of water-cooled engines, in order to meet the objectives for power output, fuel economy, noise reduction, exhaust emissions, durability and reliability.

The targets related to cooling and the technical methods (concepts) for achieving them are as follows:

- (1) To study cooling-fin shapes that increase the cooling-air flow rates and produce more effective cooling.
- (2) To determine an oil cooler position and internal structure that help achieve good cooling performance.
- (3) To reduce heat transmission and increase heat dissipation by shortening the length of the exhaust ports and modifying the thickness of the port walls, in order to reduce the temperature around the cylinder head combustion chamber which is the heat source.
- (4) To increase heat dissipation by providing a special oil-cooling system oil line to counteract temperature rises due to heat transmission around the combustion chamber. Also, to provide an orifice in the cooling system oil line and adjust the amount of oil supplied to each cylinder oil jacket in order to correct the temperature difference among the cylinders.
- (5) To add an oil temperature control function with oil temperature sensor information to the FI (Fuel Ignition) control, in order to adjust to changes in the riding conditions and environment.

Cooling-air flow analyses have been performed in the past for items (1) to (3) above, focusing on the cooling-fin shape of the air-cooled engine and the oil cooler position. Heat conduction was calculated for the entire engine to additionally develop these items and to study item (4), which enabled the temperature forecasts on each part theoretically. In particular, for multi-cylinder engines, it is important not only to reduce the maximum temperatures but to also correct the temperature difference among cylinders, in order to increase durability and reliability and stabilize quality. Consequently, fluid and heat analysis was used to quantitatively calculate the optimal amount of oil flow.

For item (5), the thermostat and temperature sensor layout positions were studied to determine an efficient temperature control method.

Note that although studies^{(4), (5)} on oil-cooling systems for engine cooling have already been reported, not many studies have been conducted using quantitative or visualized data.

In line with the study described above, it was decided to design and produce a new engine. It was also decided to combine both CFD analyses and actual tests to verify the accuracy and validity of the calculation results.

2.2. Forced Oil-cooling System

Figure 1 (a) shows the oil line systems within the structure of the forced oil-cooling system.

The green and red areas in the figure are the oil lines for lubrication and cooling respectively. It is a forced oil-cooling structure, where oil supplied from an oil pump with two rotors is divided into two oil lines. One oil line is for lubrication and the other is exclusively for cooling. A temperature sensor for the FI system was provided in the cooling system oil line to additionally enhance fuel economy and comply with emission regulations. A thermostat was positioned in front of the oil cooler to enhance the warming-up performance.

As a cooling medium, oil has a lower specific heat than cooling water, so the cooling performance of the oil jacket shape with the same amount of flow is lower than that of the water-cooling system. The oil viscosity coefficient changes according to the oil temperature. This characteristic is utilized in the system and shapes were

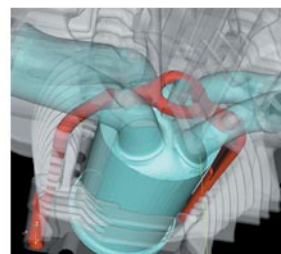
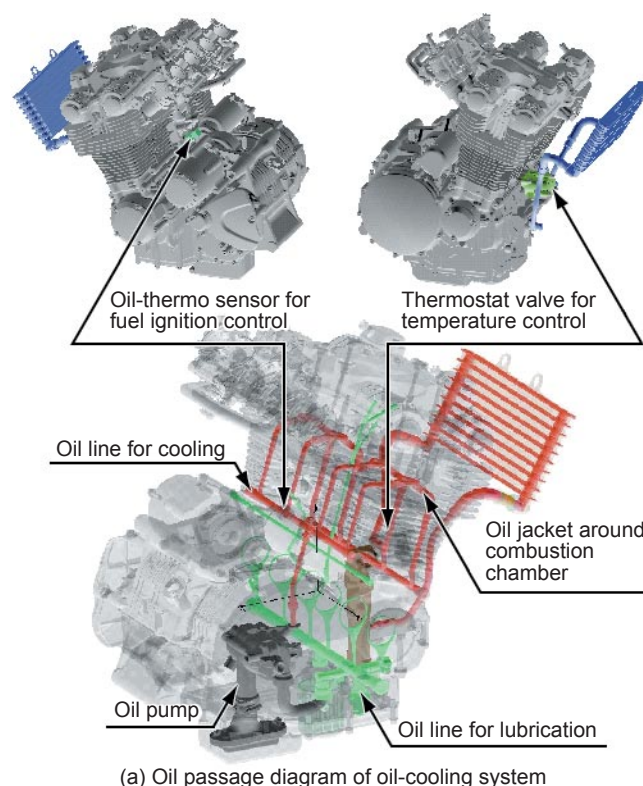


Fig. 1 Outline of engine oil-cooling system

studied to intensify the flow rate of the cooling-oil to increase the heat transfer coefficient. In order to achieve efficient cooling around the combustion chamber which is a heat source, the oil jacket was positioned near the cylinder head combustion chamber. In the case of the forced oil-cooling system, stagnations of oil-flow generate localized oil temperature spikes, resulting in the degradation of the oil. Another concern is when the surface area inside the oil line is too large, the heat receiving area will increase and the cooling-oil temperature will rise too much. Specifically, the shape that was used in the oil lines is different from the water jacket shape that covers the entire combustion chamber, which is similar to the design used in water-cooled engines. A line was provided between the respective intake and exhaust ports as shown in Fig. 1 (b) and items of concern in production (such as making the jacket core and casting of cylinder head) were also taken into account, to produce an oil jacket shape with less stagnation and the optimal heat receiving surface area.

3. Calculation Outline

3.1. CFD Analysis Properties

In the case of designing a new air-cooled engine, CFD analysis is effective as an examination item in an initial stage of design.

To effectively utilize the air flow generated by the moving motorcycle, the following properties were analyzed by the CFD: cooling-air flow, optimal shape for the cooling-fin, efficient position for the oil cooler, etc. In a wind-tunnel test, only visualization of air-flow is available. On the other hand, with the CFD analysis, it was possible to study the flow field under actual riding conditions before making the prototype, quantitatively study the cooling performance, and calculate the heat transfer coefficients for each engine area. By developing these results, it was also possible to calculate the temperature distribution of the engine and to link it to heat strain analysis.

In addition, when using an oil-cooling system, the study of the oil jacket shape, flow method of cooling-oil, optimization of the amount of cooling-oil recirculation, as well as the study of the exhaust port shape in order to minimize the heat source, was treated as analysis properties using CFD.

3.2. Calculation Procedure

Figure 2 shows the calculation procedure. With a numerical wind tunnel, shape data is created using 3D-CAD systems (Fig. 2 (a)) such as CATIA, and then written into a file in STL (Stereo Lithography) data format. Commercially-available software⁽⁶⁾ was then used to correct STL data and perform parts conversion. Figure 2 (b) shows some of the STL data after conversion.

To analyze the flow in a way equivalent to actual

riding conditions, a solid model of the completed motorcycle was placed in a numerical wind tunnel as shown in Fig. 2 (c), the Cartesian Coordinates (Fig. 2 (d)) were set, and the boundary conditions were created. There are approximately 2,370,000 cells of 133 x 128 x 139 in the calculation grid. After the calculation, the local heat transfer coefficient (Fig. 2 (e)) was obtained from the equation⁽⁷⁾ derived from Karman's analogy. The CFD analysis of the cooling-oil flow was also performed on the oil jacket.

The heat transfer coefficient obtained above and the heat flux in areas such as the combustion chamber and exhaust port were used to calculate engine heat conduction and obtain the engine temperature distribution.

Here, the heat conduction was calculated by developing the program of the Partial cell Cartesian Coordinate method in the CFD calculation. The same information is used for the Cartesian Coordinates as is used for the cooling-air flow CFD calculation, so interpolation calculations between coordinates for the heat transfer coefficient are not necessary.

3.3. Boundary Conditions

For the boundary conditions in the CFD calculation of cooling-air behavior analysis, a flow rate equivalent to that

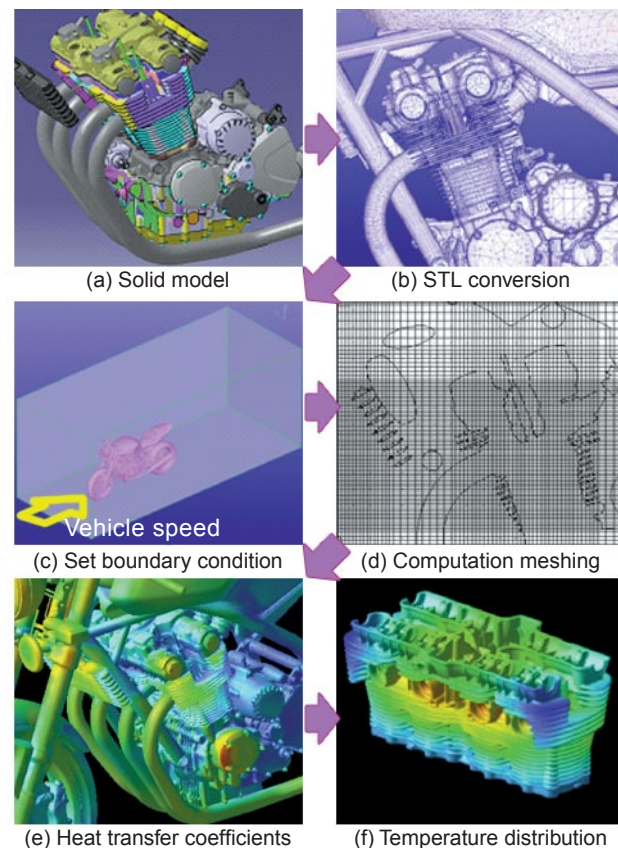


Fig. 2 Procedure of air-cooled engine analysis using CFD code

of motorcycle speed was applied to the entrance of the numerical wind tunnel as shown in Fig. 2 (c), with a turbulence intensity of 5% and length scale of 0.05m, and taking the ground and side walls as fixed walls.

For the boundary conditions in the calculation of the engine heat conduction, the local heat transfer coefficient for cooling-air obtained from the CFD calculation and the average heat transfer coefficient for the oil jacket were used. However, in addition to these, it was also necessary to take the following items into account even though these values were not very clear; cooling due to lubricating oil and the heat flux around the combustion chamber, in the cylinder walls and the intake and exhaust port inner walls⁽⁸⁾.

The objective of this study was to examine the cooling performance of air/oil-cooled engines, and in particular to study the effect of the oil jacket. Measured data from a test engine was substituted for the temperature at the bottom surface of the simulated cylinder barrel (upper crankcase deck surface), and this value was adjusted to approximately match the measurement points of the prototype engine.

The calculation grid in the heat conduction calculation was made uniform with CFD calculation of the cooling-air flow, so the grid of the heat conductive calculation is relatively rough. On the other hand, the minimum thickness of the cylinder head is thinner than the grid, so as Fig. 3 shows, in some areas it becomes necessary to determine two heat transfer coefficient settings for cooling-air and cooling-oil inside one cell. Furthermore, in some areas, boundary conditions are necessary for heat fluxes that make contact with the combustion chamber. To enable calculations even in cases such as those described above, the shape data for the calculation of heat conduction (STL data) is divided into groups by setting boundary conditions, and heat transfer coefficients, heat fluxes, and actual measured temperatures for each calculation cell to enable the setting of multiple boundary conditions.

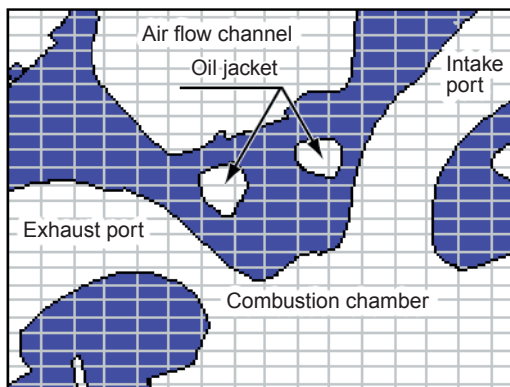


Fig. 3 Computation grids for heat conductive calculation

4. Calculation Results and Discussion

4.1. Heat Transfer Coefficient of Oil Jacket

The heat transfer coefficient of the cooling-oil (α : W/m^2K) was calculated in order to calculate the amount of heat received by the oil jacket. Figure 4 shows an example of these calculation results.

By decreasing the line cross section area between intake and exhaust ports, the flow rate rises and the heat transfer coefficient increases. On the other hand, the heat receiving area does not change much when the cross section shape is varied, so the cooling efficiency is enhanced. The wide area on the exhaust port side is a stand (base) that enables the oil jacket core to be stood up when casting the cylinder head. The optimal stand from a variety of shapes was selected using the CFD, to make sure it did not generate stagnation of cooling-oil.

In contrast to cooling-water, the viscosity of oil varies according to the oil temperature. Since the heat transfer coefficient varies according to the oil temperature, the relationship between the oil flow, oil temperature, and the heat transfer coefficient was obtained with CFD. However,

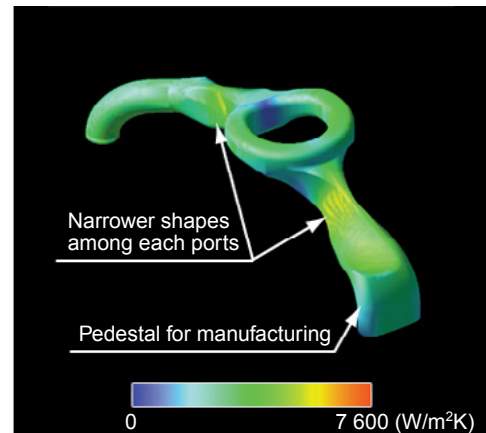


Fig. 4 Heat transfer coefficients of oil jacket

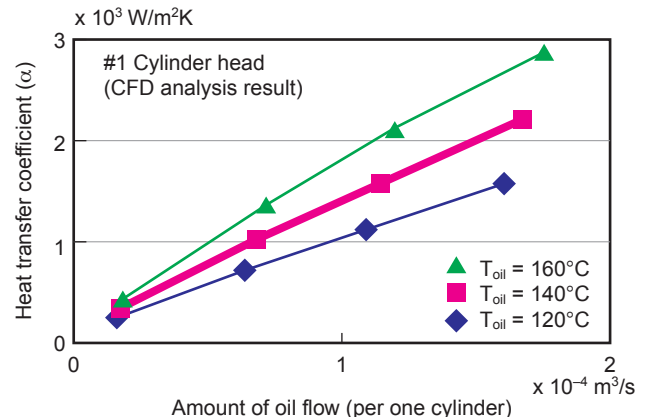


Fig. 5 Relationship between oil flow amount and heat transfer coefficients

the average heat transfer coefficient for the oil jacket was treated as the average values of only the oil jacket in the cylinder head area. Figure 5 shows these results. From these results it can be understood that the heat transfer coefficient is 70 to 80% higher when the oil temperature is 160 °C than when it is 120 °C.

With the water-cooling system, when the water temperature rises, the difference in temperature (ΔT : °C) between the coolant and the coolant line inner wall surface becomes smaller, and so the amount of heat dissipation drops.

However, with an oil-cooling system, when the oil temperature rises, α increase is larger than the reduction due to ΔT , so sometimes the amount of heat dissipation rises when the oil temperature is high. For example, if it is assumed that the oil jacket inner wall surface temperature is 250 °C, when the oil temperature rises from 120 °C to 160 °C, ΔT changes by 0.7 from 130 °C to 90 °C, while α changes by 1.7 and the heat transfer coefficient ($\alpha\Delta T$) changes by 1.19.

Consequently, while the water jacket in a water-cooling system covers the entire combustion chamber, for the oil jacket shape it is desirable to reduce the heat receiving area and design a simple shape that increases the oil flow rate.

4.2. Comparison with Surface Temperature Measurement

Figure 6 shows a comparison of the engine heat calculation results and the surface temperature measurement results using an infrared thermometer (thermoviewer) in a bench test. For the calculation and measurement conditions, the air-flow velocity and engine load were equivalent to 80 km/h.

In the calculation, α was obtained using the equation derived from Karman's analogy, and the heat was calculated based on the assumption that the cooling-air temperature is 25 °C. In the calculation results, the temperature of the cooling-fin on the exhaust port side is lower than the cooling-fin on the intake port side, but in the image shown by the thermoviewer, the cooling-fin temperatures on the exhaust port side and the intake port side are approximately the same. This is probably because the air temperature around the exhaust pipe is high in actual engines, and so the temperature difference is lower than in the calculation.

In summary, while there is still room to modify a number of items, it can be said that approximately appropriate calculation results were obtained for the engine temperature distribution.

4.3. Effect of the Oil-cooling System

In order to study the effectiveness of oil-cooling, heat conduction was calculated with and without oil-cooling. The calculation conditions were set as an engine load equivalent to 230 km/h with no oil-cooling, and so the

heat transfer coefficient for the oil jacket was set as zero. Figure 7 shows the temperature distribution of these calculation results.

From the figure it can be understood that the ends of the cooling –fins are cool but that the temperature is high on exhaust port inner walls in the intermediate cylinders (No.2, No. 3 cylinders) and that the cooling in the central area of the engine is insufficient. Part (a) in Fig. 8 shows the temperature (T_{plug} : °C) around the cylinder head ignition plug seat in the calculation. It is estimated that the difference of the T_{plug} temperature between cylinders is approximately 40 °C, and that the maximum temperature of T_{plug} for the intermediate cylinders is close to 300 °C.

Next, to verify the effectiveness of oil-cooling, the heat transfer coefficient of the oil jacket was recalculated with the equivalent cooling-oil amount for each cylinder of $1 \times 10^{-4} \text{ m}^3/\text{s}$ (6.0 l/min) and an oil temperature of 160 °C. Figure 9 shows these results. The temperature over the entire engine is lower, so it can be interpreted that oil-cooling is effective. As shown by (b) in Fig. 8, it is estimated that there is a temperature reduction of approximately 70 °C for T_{plug} with the intermediate cylinders having the highest temperature. Then, the calculation and measured values were compared for

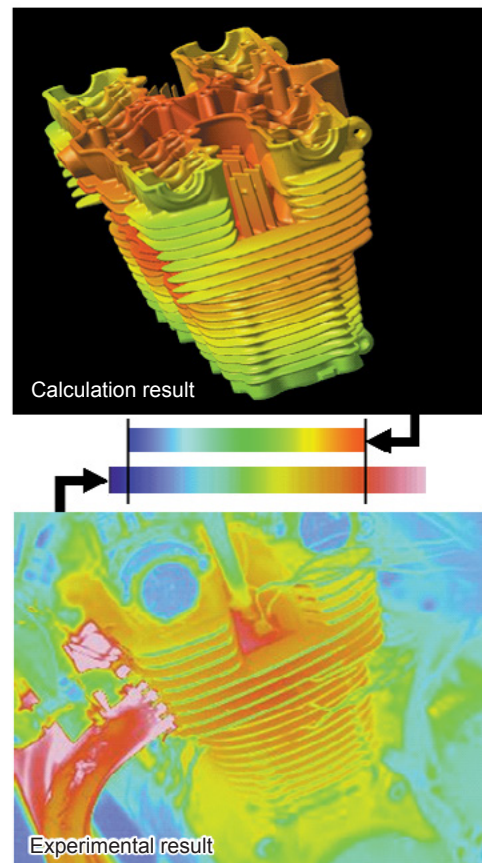


Fig. 6 Comparison of temperature (at 80 km/h)

verification. Part (c) in Fig. 8 shows the measured values.

Although the precision of the calculation values is insufficient, the temperature difference among the cylinders (difference in T_{plug} between each outer side (Nos. 1 and 4) and intermediate (Nos. 2 and 3) cylinders) is approximately the same for the calculated values and the measured values, and so experimentation using calculations is possible in order to correct the temperature difference among the cylinders.

When looking closely at Fig. 9 (a), first it can be seen that the corner areas of the cylinder head cooling-fin on the exhaust port side are well cooled. In contrast, the overall temperature around the combustion chambers in the central area of the engine is high. In particular, the temperature of the inner wall around the exhaust port is high. This is probably because the air flow generated by the moving motorcycle is blocked by the front tire and front suspension so that the cooling-air on the front center of the engine is insufficient.

Figure 9 (b) shows temperature distribution around the cylinder head combustion chamber. It can be understood that the temperature of the combustion chamber wall is higher on the exhaust port side than that

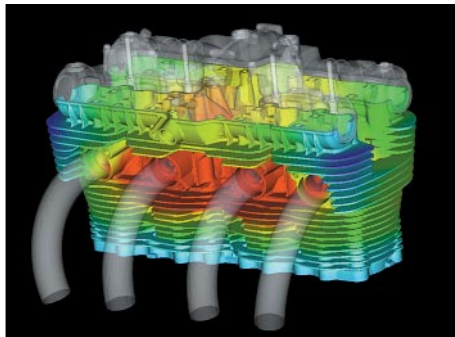


Fig. 7 Temperature distribution without oil-cooling

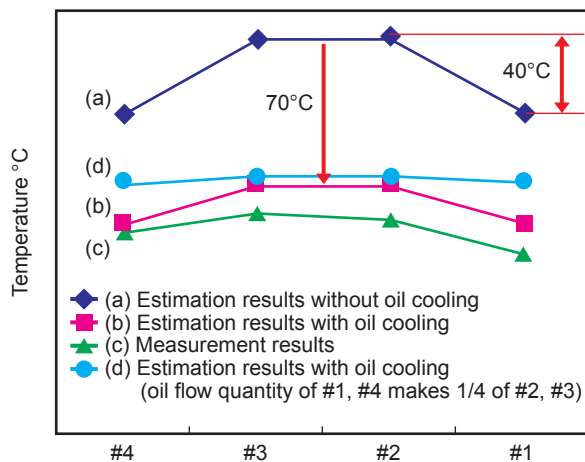
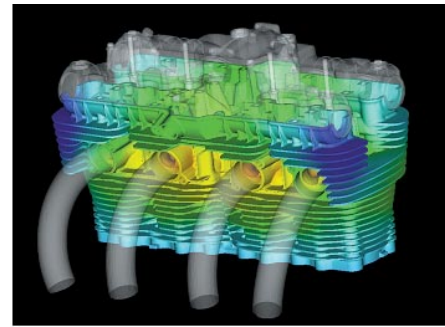
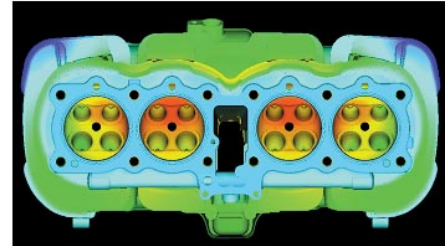


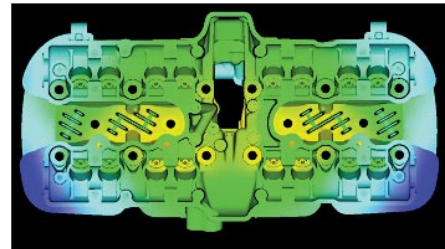
Fig. 8 Calculation of cylinder head temperature (at 230 km/h)



(a) Oblique perspective view

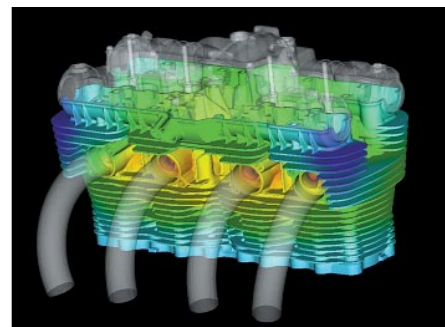


(b) Bottom view

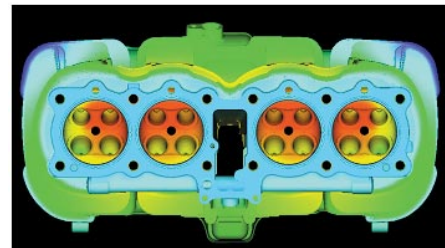


(b) Top view

Fig. 9 Temperature distribution with oil cooling ($T_{\text{oil}} = 160^{\circ}\text{C}$)



(a) Oblique perspective view



(b) Bottom view

Fig. 10 Temperature distribution of improving uneven temperature among cylinders ($T_{\text{oil}} = 160^{\circ}\text{C}$)

of the intake port, and that the combustion chamber temperature is particularly high between the two branching exhaust ports. When comparing the cylinders, it can be understood that the outer side cylinders are well cooled.

4.4. Correction of the Temperature Difference among Cylinders

In order to study the correction of the temperature difference among cylinders, calculations were made with many different amounts of cooling-oil for each cylinder. The calculation results shown in Figs. 10 and 8 (d) were obtained by leaving the amount of cooling-oil flow per cylinder to the intermediate cylinders (cylinder Nos. 2 and 3) unchanged and controlling the amount of cooling-oil flow to the outer side cylinders (cylinder Nos. 1 and 4) with an orifice to a flow rate of approximately 1/4 ($0.25 \times 10^{-4} \text{ m}^3/\text{s}$). As such, it was understood that the temperature difference among cylinders could be corrected.

5. Confirmation of Oil-cooling Effectiveness with an Actual Riding Test

5.1. Decrease of Cylinder Head Temperature around Ignition-Plug Seat

The cylinder head temperature around the ignition-plug seat (T_{plug}) was used as an indicator to evaluate the cooling performance of the engine.

Figure 11 shows a graph of the relationship between T_{plug} and the amount of cooling-oil per cylinder supplied to the oil jacket (for each cylinder), with the data obtained from the test. The results showed that when the engine speed was 7 000 rpm, the amount of cooling-oil that should be supplied to each cylinder is approximately $1 \times 10^{-4} \text{ m}^3/\text{s}$. As a result, the discharging volume of oil pump could be minimized from viewpoint of decreasing friction.

Figures 12 (a) and (b) show the temperature reduction effect in an actual riding test when using this oil-cooling system. Figure 12 (a) shows the relationship between T_{plug} and the indicated mean effective pressure (IMEP: MPa). The points indicated with round marks in the figure are the temperatures of representative mass-production models and the points indicated with square marks are the temperatures for this engine with and without oil-cooling. The overall temperature of the combustion chamber inner walls in water-cooled engines is lower than that in air-cooled engine, and consequently T_{plug} is lower. This is because the combustion chamber area in water-cooled engines is covered by cooling medium (coolant liquid) with good heat exchange properties. On the other hand, air-cooled engines do not have a cooling medium and have to depend on the heat mass, so there is a tendency for T_{plug} to sharply rise in proportion to the IMEP.

In the case of the air/oil-cooled engine that was

produced in this study, for the same IMEP, a temperature reduction effect of approximately 70°C is obtained as shown in Fig. 12 (a). The data without oil-cooling in the figure are the results with the same engine when only air-cooling was performed, without circulating the cooling-oil.

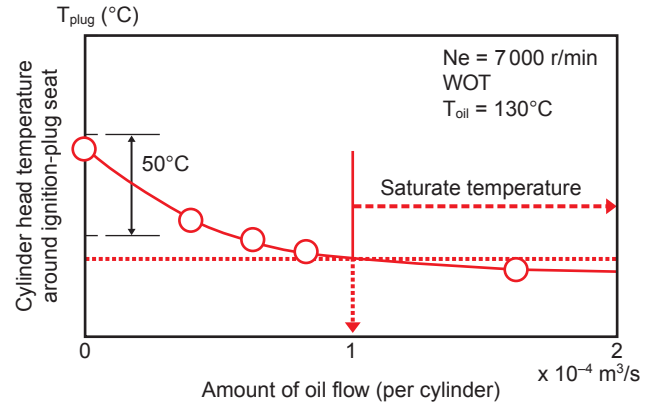
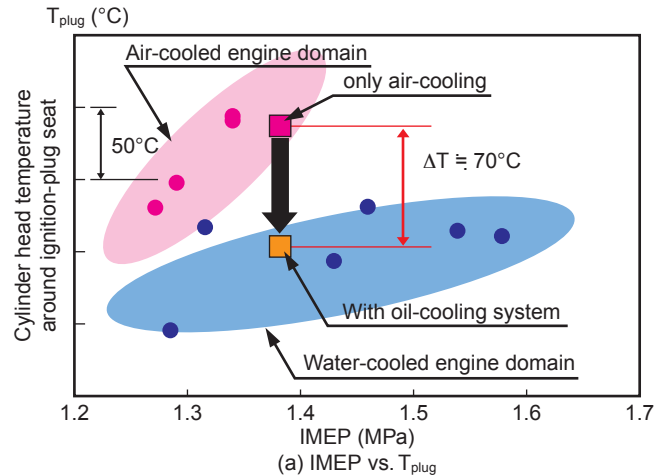
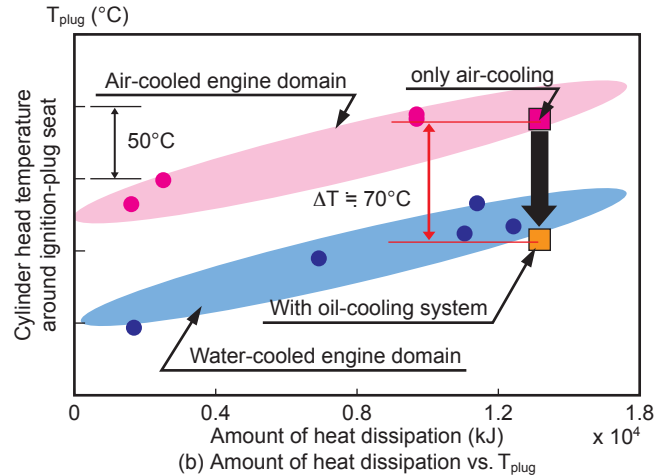


Fig. 11 Optimization of oil flow amount (per cylinder)



(a) IMEP vs. T_{plug}



(b) Amount of heat dissipation vs. T_{plug}

Fig. 12 Effect of oil-cooling

Figure 12 (b) shows the relationship between T_{plug} and the amount of heat dissipation (Q_t) (kJ) after correction of Fig. 12 (a). In the same way as Fig. 12 (a), it shows a temperature reduction effect of about 70 °C when compared with engines that have approximately the same amount of engine heat generation. Note, however, that in Fig. 12 (b) temperatures corresponding to representative mass-production models were used for each point. Q_t (kJ) is the amount of heat dissipation and was calculated from equations (1)⁽⁸⁾.

$$Q_t = Q_{t1} + Q_{t2} + Q_{t3} \quad (1)$$

Here, Q_{t1} (kJ) is the heat dissipation conducted from around the combustion chamber, Q_{t2} (kJ) is the heat dissipation radiated from around the combustion chamber, and Q_{t3} (kJ) is the heat dissipation from the friction heat in the reciprocating part of the piston.

Figure 13 shows the relationship between the IMEP and compression ratio (ϵ) by motorcycle category for representative Honda mass-production models. It can be understood from the figure that when comparing engines with the same output (same IMEP), it is possible to increase the compression ratio of an air-cooled engine to

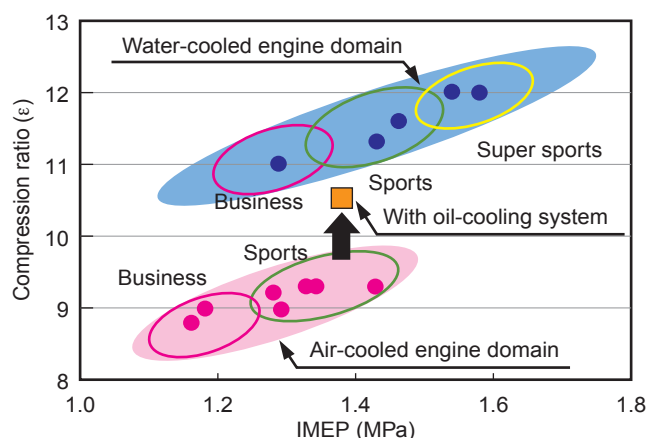


Fig. 13 Effect of oil-cooling for compression ratio

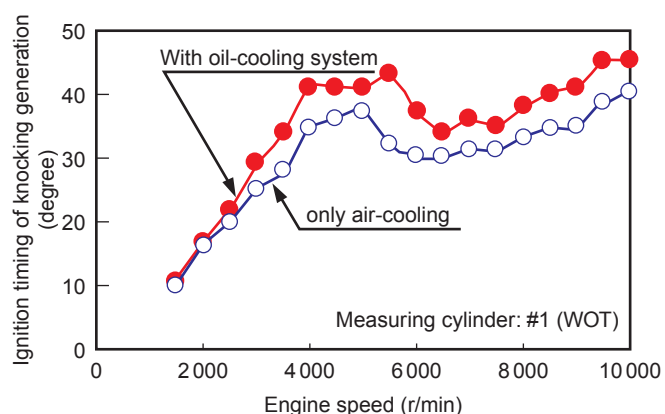


Fig. 14 Effect of oil-cooling for engine knocking limit

near that of a water-cooled engine for sports category of motorcycles produced in this study.

5.2. Increasing the Knocking Limits

Figure 14 shows the effect of increasing the knocking limits when using this forced oil-cooling system in an air-cooled engine.

In all engine speed ranges of 2 000 rpm or higher, an enhancement of about +5° or more was seen in the knocking generation ignition timing. It was assumed that the reason for this was because the temperature of the unburned mixed gas was reduced by lowering the wall surface temperature around the combustion chamber and making it more uniform.

In addition, from the results of the durability test, it was confirmed that the valve seat deformation and deterioration and blow-by gas discharge were corrected to levels comparable to those in a water-cooled engine.

6. Conclusion

- (1) The use of the forced oil-cooling system in an air-cooled engine has the potential to match the power output, environmental performance, durability and reliability of a water-cooled engine.
- (2) It was understood that the temperature difference among cylinders could be corrected by controlling the amount of cooling-oil.
- (3) It was confirmed with an actual test that, with the combined use of the forced oil-cooling system, temperature distribution comparable to that of a water-cooled engine was available even with an air-cooled engine.
- (4) It was also confirmed that the forced oil-cooling was effective in other areas, such as enhancing the knocking limits.

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Development of Lean NO_x Catalyst with High Thermal Durability

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ABSTRACT

A new metal substrate was developed to prevent the migration of alkali metals which work as NO_x adsorbents and maintain a higher NO_x conversion rate after thermal aging. A higher NO_x conversion rate was expected after thermal aging because the metal substrate contains almost no silicon (Si), however, the NO_x conversion rate after thermal aging fell below that of the cordierite substrate. This was not due to the migration of alkali metals to the metal substrate, but due to the chromium (Cr) in the metal substrate migrating to the washcoat and generating complex compounds formed with alkali metals. To help prevent this migration, the conventional metal substrate was heat-treated in air at 1100 °C to make an α -alumina barrier on the surface. This newly developed substrate showed no migration of alkali metals and almost no migration of Cr. When this substrate was used, the NO_x conversion rate after heat treatment at 830 °C for 280 hours was 31% higher than that of the conventional cordierite substrate.

1. Introduction

Calls for a reduction of the burden placed on the environment by automobiles are increasing. Carbon monoxide (CO), hydrocarbon (HC), and nitrogen oxide (NO_x) emissions have been subject to regulations for some time. Trends in the reduction of carbon dioxide (CO₂) emissions, which are not directly harmful to the human body but are a factor in global warming, have gathered at an increasing pace.

To reduce CO₂ emissions, it is necessary to increase fuel efficiency, of which lean combustion remains an effective means of achieving this goal. However, three-way catalysts cannot be used to convert NO_x in a lean burn engine, so it is necessary to use a lean catalyst that can adsorb and reduce NO_x when lean combustion is used.

Countries worldwide are establishing increasingly stringent regulations concerning CO, HC, and NO_x emissions. Because the performance of conventional lean

NO_x catalysts declines with thermal aging, the number of gasoline lean burn engines on the market has decreased as emissions regulations have become increasingly stringent.

The issue of thermal aging related performance decline in lean NO_x catalysts using cordierite substrates, which is caused by a chemical reaction between the silicon (Si) in the substrate and adsorbent alkali metals in the catalyst, was addressed by this work. The research discussed in this paper examined the causes of performance decline in a lean NO_x catalyst using a metal substrate containing almost no Si, in which there would be no migration of alkali metals to the substrate. The cause was identified as the migration of chromium (Cr) from the substrate to the washcoat, so a method of preventing the migration of Cr was developed. A lean NO_x catalyst using the developed substrate displayed a 31% higher NO_x conversion rate following thermal aging at 830 °C for 280 hours than a conventional catalyst using a cordierite substrate.

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2. NO_x Conversion Technology for Lean Burn Engine

Three-way catalysts were developed for use in gasoline engines in the 1970s. These catalysts provided a one-stage treatment of CO, HC, and NO_x at stoichiometric air/fuel (A/F) ratios. Used in combination with high oxygen storage capacity (OSC) materials which broaden the conversion window and the precision A/F ratio control, they are the main technology used to convert exhaust gases in gasoline engines.

The issue of global warming has increased in severity, generating demands for the reduction of CO₂ emissions, which does not directly harm the human body but contributes to global warming, to increase in intensity. The automotive industry is no exception to this trend. A variety of techniques, including the achievement of weight savings and the reduction of friction, have been used to increase fuel efficiency and thus reduce CO₂ emissions from automobiles. The lean burn technology is an effective method to increase fuel efficiency. The development of the direct injection engine technology was promoted in the latter half of the 1990s.

However, while three-way catalysts display excellent NO_x conversion performance at stoichiometric A/F ratios, they are virtually unable to convert NO_x in lean burn at an oxygen concentration of 0.5% and above. A replacement NO_x conversion technology able to function in lean burn is therefore necessary.

The following catalysts are able to convert NO_x in lean burn:

- (1) NO direct decomposition catalysts
- (2) HC-SCR (HydroCarbon-Selective Catalytic Reduction) catalysts
- (3) NO_x adsorption and reduction catalysts

Of these, neither (1) nor (2) possess sufficient NO_x conversion performance to meet the 2005 emission control standards in Japan.

Type (3), lean NO_x catalysts, use adsorbent materials to adsorb NO_x in lean burn and convert the adsorbed NO_x by means of intermittent rich spikes⁽¹⁾. Figure 1 shows the mechanism of NO_x conversion in lean NO_x catalysts. With a variety of modifications⁽²⁾, this type of

catalyst was used in the Stream with direct-injection system, and reduced NO_x emissions by 50% compared to the 2005 emission control standards in Japan. The majority of stoichiometric gasoline engines have reduced NO_x emissions to 75% of the level stipulated in the 2005 emission control standards. However, there are no catalysts that display a high NO_x conversion rate in lean burn gasoline engines, and the 75% reduction has not been achieved yet using this technology. The development of a catalyst able to reduce NO_x emissions by 75% compared to the 2005 emission control standards even in lean burn is therefore a desirable goal from the perspective of increased fuel efficiency.

3. Development of Metal Substrate for Lean NO_x Catalyst

3.1. Issues of Cordierite Substrate and Study of Replacement Substrate

As indicated in Section 2, there are three main types of NO_x-converting catalysts. The enhancement of the NO_x conversion performance of one of these, the lean NO_x catalyst, represents the most practical method of complying with future stringent emissions regulations.

The following represents methods to keep the performance of the conventionally used catalysts that support NO_x-adsorbing materials on a cordierite substrate:

- (1) Prevention of sintering of precious metals
- (2) Prevention of sulfur poisoning
- (3) Prevention of migration of alkali metals to substrate

Issue (1) is also an issue in conventional three-way catalysts, and is not easy to settle. In the case of (2), performance can be recovered by means of a sulfur purge, and this issue is expected to be settled by the use of lower-sulfur fuels.

Issue (3) is specific to lean NO_x catalysts. This phenomenon occurs because of the use of alkali metals as NO_x-adsorbing materials in these catalysts. At temperatures of 700 °C and above, the alkali metals migrate to the cordierite substrate. To provide an example, in the case of potassium, the following reaction occurs:

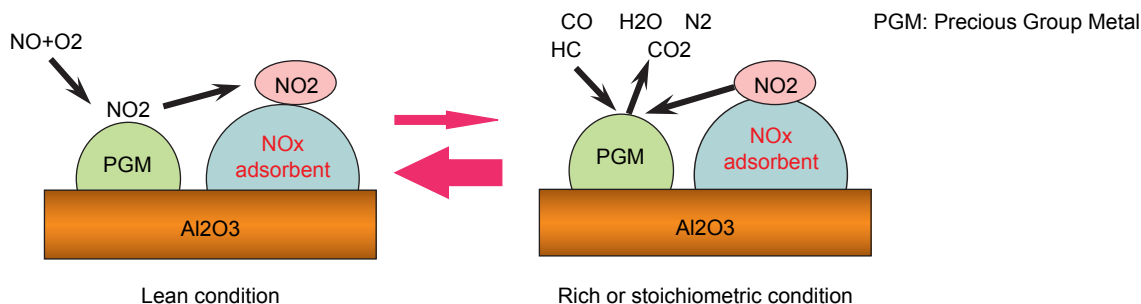


Fig. 1 NO_x Purification mechanism of lean NO_x catalyst

Figure 2 shows the results from an electron probe micro-analyzer (EPMA) image of potassium in a catalyst before and after thermal aging. These results show that the amount of alkali metals in the washcoat has been reduced, and that the alkali metals have migrated to the substrate due to thermal aging.

No method of reversing this reaction, as in the case of the use of sulfur purge when sulfur poisoning occurs, has been determined. The alkali metals that have undergone this reaction form complex compounds, which reduce its NO_x adsorption performance, causing the performance of the lean NO_x catalyst to decline. In addition, the compounds formed in the substrate have a different coefficient of thermal expansion from cordierite, and this can cause the substrate to crack. Figure 3 shows the surface of a catalyst following aging for 300 hours at 830 °C as an example. A crack due to thermal aging is visible on the external wall. To avoid this cracking of the substrate, it is necessary to restrict the supported amount of alkali metals.

The reason for the migration of alkali metals to the substrate is the fact that alkali metals react with Si, as Eq. (1) shows. Different methods have been proposed to

prevent this: specific elements can be added to stabilize the NO_x-adsorbing material⁽²⁾, or a K-capture layer can be formed between the substrate and the catalyst to prevent the alkali metals from reaching the substrate⁽³⁾.

In addition to cordierite substrates, metal substrates are also in mass production. These include substrates for three-way catalysts and the like. Metal substrates use 20Cr-5Al ferrite stainless steel alloy. Because the only Si found in these substrates occurs in the brazing material bonding the metal foil, it was considered that they might be effective in preventing the migration of alkali metals, and thus enhancing performance.

3.2. Results of Tests using Metal Substrates and Cause of Performance Decline

Based on the study mentioned above, lean NO_x catalyst was coated on a 20Cr-5Al substrate and subjected to thermal aging at 830 °C, after which its NO_x conversion rate was measured.

Figure 4 shows the results of these tests and the results of a lean NO_x catalyst supported on a conventional cordierite substrate as a comparison. Following thermal aging for 300 hours at 830 °C, the performance of the lean

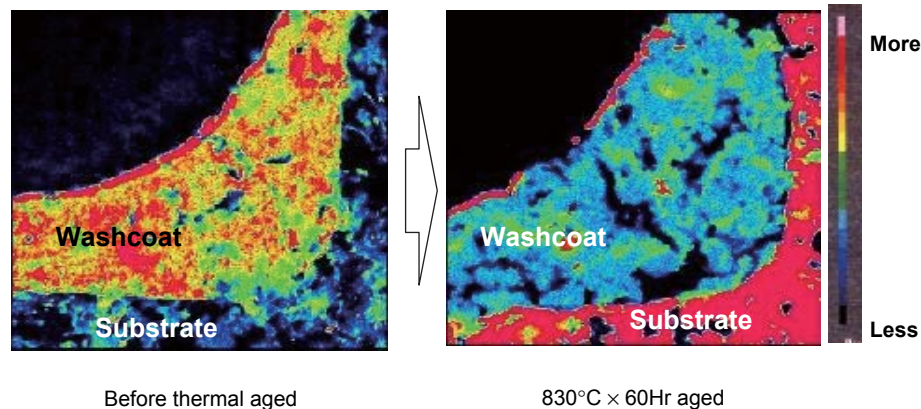


Fig. 2 EPMA map of potassium before and after thermal aging



Fig. 3 Surface crack after 830°C x 300Hr aging

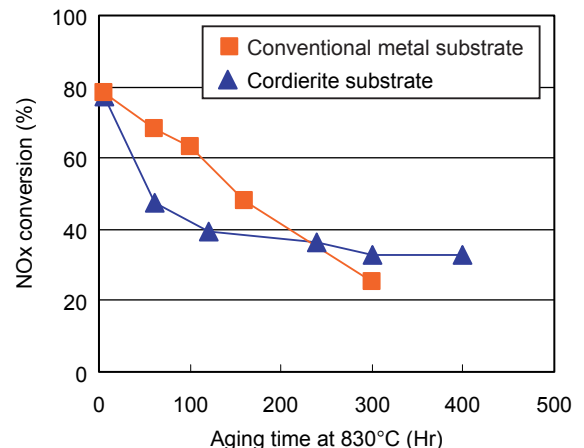


Fig. 4 NO_x Conversion with respect to aging time

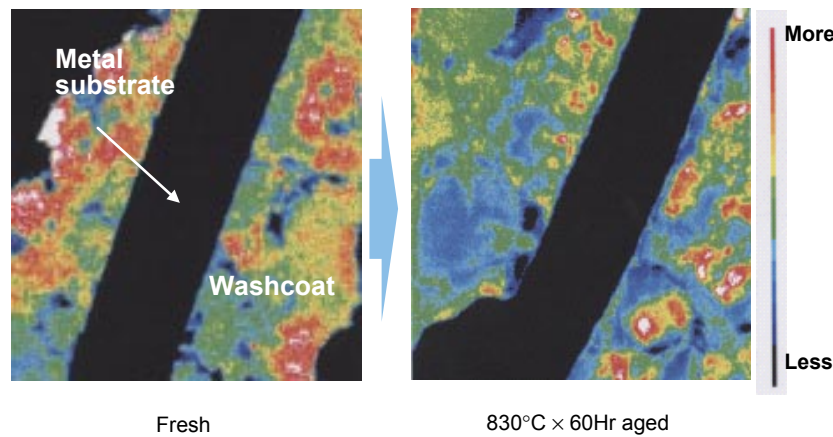


Fig. 5 EPMA map of potassium before and after aging

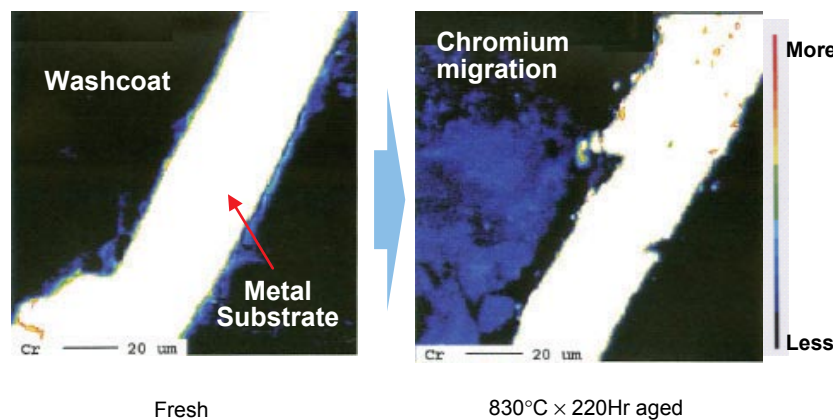


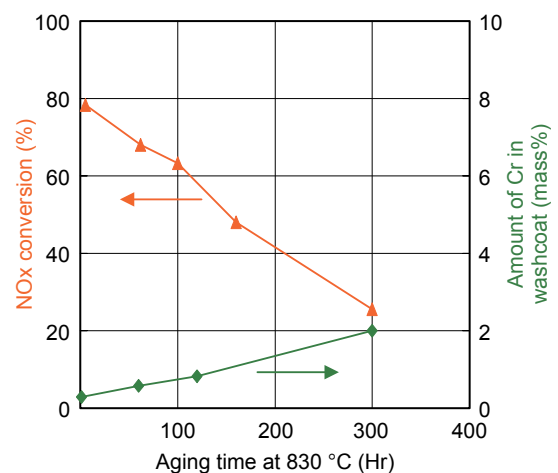
Fig. 6 EPMA map image of chromium before and after aging

NO_x catalyst supported by the 20Cr-5Al substrate declined to a level below that of the lean NO_x catalyst supported on the cordierite substrate. In addition, while the rate of performance decline with thermal aging time in the cordierite-supported catalyst was gradual, the metal-supported lean NO_x catalyst displayed an almost linear decline in performance with time.

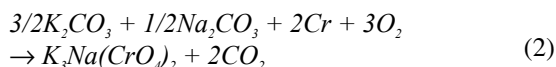
The catalyst and substrate were subjected to the EPMA analysis in order to determine the cause of this unexpected decline in performance. The results are shown in Figs. 5 and 6. As Fig. 5 shows, the analysis of the components of the substrate demonstrated that no alkali metals had migrated to the metal substrate. There was also no migration of other catalyst components to the substrate. However, as shown in Fig. 6, the results of analysis of the catalyst components indicated that Cr from the metal substrate had migrated to the washcoat. The relationship between the amount of Cr migration and the NO_x conversion rate is shown in Fig. 7. The decline in the rate of NO_x conversion is almost exactly proportional to the amount of Cr migration.

The washcoat was examined in isolation using powder X-ray diffractometry (XRD). The results showed that the

alkali metals and Cr had reacted to form chromium compounds. The XRD analysis results of the washcoat are shown in Fig. 8. These results indicate the following mechanism of performance decline in a lean NO_x catalyst

Fig. 7 NO_x conversion and amount of Cr in washcoat with respect to aging time

using a metal substrate: thermal aging causes the Cr in the metal substrate to migrate to the washcoat, generating a reaction of the type shown in Eq. (2), and complex oxides form in the alkali metals, making its adsorption performance to decline.



3.3. Development of a Metal Substrate with Limited Cr Migration

As discussed above, when a catalyst is supported by a conventional metal substrate, the rate of NO_x conversion declines as Cr migrates to the washcoat, where it forms complex compounds. A new method was therefore necessary to prevent Cr migration.

The metal substrate in a conventional metal substrate is formed from Fe-20Cr-5Al alloy. 5 mass% Al is used to increase the oxidation resistance of the metal substrate because the Al is preferentially oxidized and forms a protective alumina film when the metal substrate is exposed to high-temperature oxidizing air.

The present research project proceeded to examine the

possibility of using this protective alumina film as a barrier to prevent Cr diffusion after thermal aging in a lean NO_x catalyst. A conventional metal substrate was oxidized at 1100 °C in air, resulting in the formation of α alumina on the surface. Figure 9 shows a scanning electron microscope (SEM) image of this metal substrate (α-treated). The image shows that a uniform α alumina barrier has formed between the washcoat and the metal substrate.

To test the effectiveness of the α alumina barrier in controlling the migration of Cr, the catalyst was aged at 830 °C and its NO_x conversion performance was measured. Figure 10 shows EPMA maps of a conventional metal substrate and an α-treated metal substrate following thermal aging. Cr had migrated into the washcoat in the conventional metal substrate, but no Cr migration was observed in the α-treated metal substrate.

Figure 11 shows NO_x conversion rates for a cordierite substrate, a conventional metal substrate and an α-treated metal substrate following thermal aging. The rate of performance decline was slow in the case of the α-treated metal substrate, and it displayed a 31% higher rate of NO_x

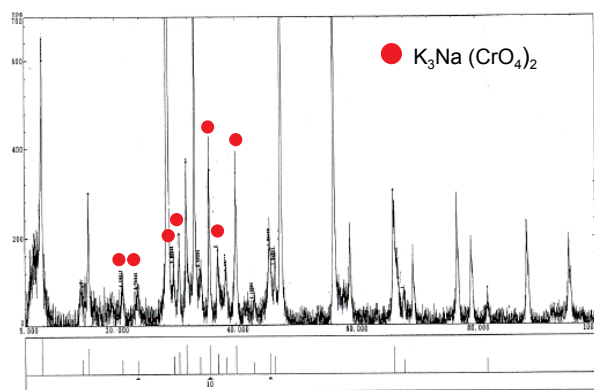


Fig. 8 XRD chart of washcoat after 830°C x 220Hr aging

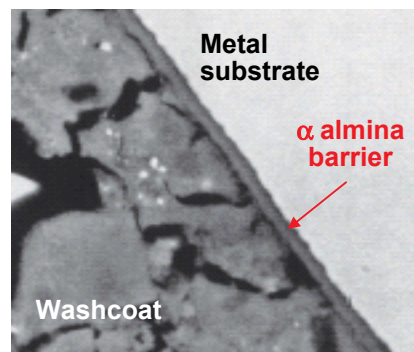


Fig. 9 SEM image of metal substrate with alpha alumina barrier

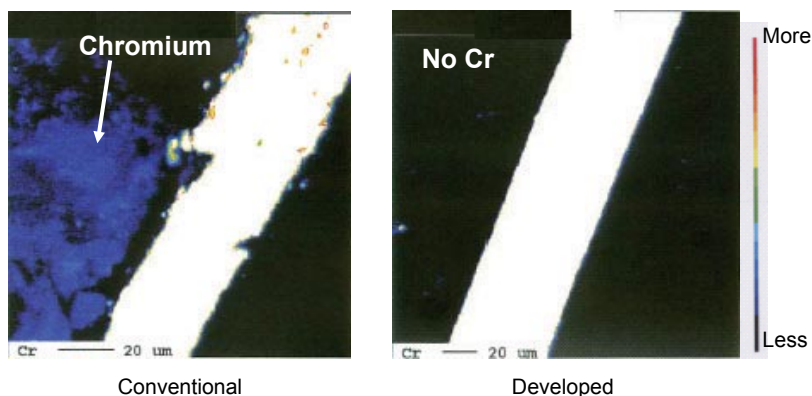


Fig. 10 EPMA map of chromium after 830°C x 220Hr aging

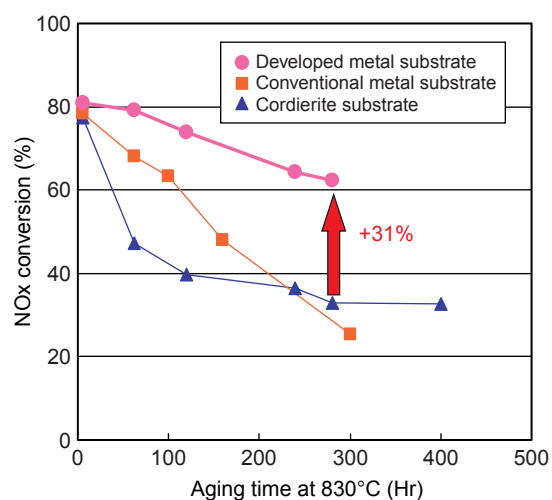


Fig. 11 Effect of metal substrate with α alumina barrier

conversion than the cordierite substrate following aging at 830 °C for 280 hours.

The developed α -treated alumina layer also displays reliable adhesion. Durability tests reproducing an automotive environment were conducted and no cracks were observed between the metal substrate and the α alumina film, indicating the developed substrate possesses sufficient durability.

4. Conclusion

A new catalyst substrate was developed by the thermal treatment of a mass-produced metal substrate at 1100 °C in air to create a fine α alumina film on the surface. The use of this substrate almost entirely controls the migration of Cr to the washcoat, which is the cause of performance decline in lean NO_x catalysts when a metal substrate is used. The developed substrate displays a 31% higher rate of NO_x conversion than a cordierite substrate following aging at 830 °C for 280 hours.

The development of the substrate led to the following findings:

- (1) Because metal substrates contain virtually no Si, alkali metals do not migrate to the substrate.
- (2) The NO_x conversion rate of a lean NO_x catalyst using a metal substrate declines because Cr in the metal substrate migrates to the washcoat and forms complex compounds with the alkali metals.
- (3) Thermal treatment of a metal substrate to precipitate a fine α alumina layer on the surface almost completely controls the migration of Cr, and the use of the substrate enhances the performance of lean NO_x catalysts.

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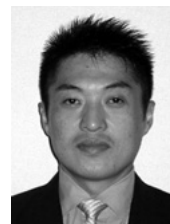
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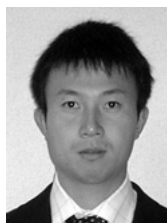
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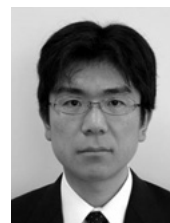
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Hydraulic Control Technologies with Robust Stability and Performance for CVT Start Clutch

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ABSTRACT

Honda Multi Matic S is a lightweight compact CVT with a hydraulically controlled wet multiple plate clutch (start clutch) used for starting. The start clutch is controlled with the torque converter as a reference, but the actual pressure is not sensed. This results in an issue in which the start characteristics are affected by temperature changes, age deterioration of the hydraulic circuit, and changes in hydraulic pressure characteristics due to process dispersion of components. Accordingly, in this research, a hydraulic servo system was configured in order to realize robust start clutch control with respect to temperature changes in particular. The controllers were composed of a two degree of freedom control system. In the design of the feedback controller, μ -synthesis was used, and variations in characteristics were actively taken into consideration.

As a result, the performance was enhanced, particularly at low temperatures, and a roughly equivalent rise time and overshoot rate were achieved at both ambient and low temperatures.

1. Introduction

The Honda Multimatic S⁽¹⁾ is a compact and lightweight continuously variable transmission (CVT) that has been developed independently by Honda. The components have been modified to produce a highly efficient and low cost system. Torque converters and electromagnetic clutches can be used as the CVT start equipment, but the Honda Multimatic S uses hydraulically controlled, wet multiple plate clutches (start clutches). In addition, a start clutch is positioned downstream from the pulley (Fig. 1), which achieves flexible start characteristics by controlling torque near the axle.

The start clutch system comprises the electronic control unit (ECU), linear solenoid (LSOL), hydraulic circuit, and wet multiple plate clutches. First, the ECU determines the target pressure from the start torque with reference to the torque converter, and outputs current based on a current versus pressure characteristic map. In accordance with this current, the pistons of the wet multiple plate clutches are operated with hydraulic pressure that was adjusted by the LSOL and hydraulic circuit, and start torque is generated [Fig. 2(a).]

However, since the actual hydraulic pressure is not sensed, the start characteristics are affected by changes in the temperature-dependent response characteristics of the clutch hydraulic pressure. In addition, the start

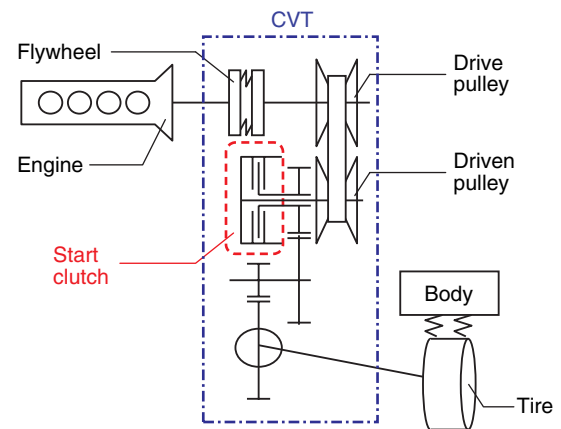


Fig. 1 Layout of start clutch

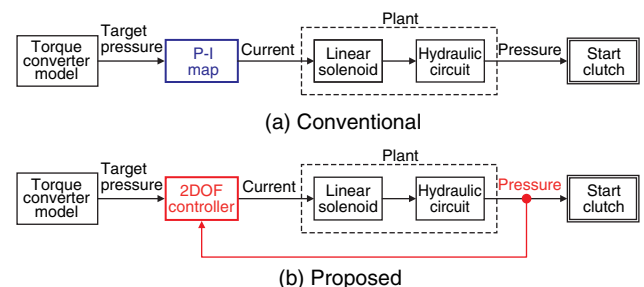


Fig. 2 Block diagram of pressure control

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characteristics may also be changed by variations in the adjustment pressure point, which may be caused by dispersion of the LSOL current, thrust, or the spring constant in the pressure control valve, and age deterioration of the clutch.

In response, this paper sets reducing the effects of temperature changes on the start characteristics as a special objective, and constructs a hydraulic servo system based on electronic control, which uses two degree of freedom (2DOF) control that enables the target tracking and disturbance rejection characteristics to be set independently [Fig. 2 (b)]. Many construction methods have been proposed for 2DOF control and their respective effectiveness reported^{(2), (3)}. Here, the method of positioning feedforward (F/F) control and feedback (F/B) control in parallel⁽⁴⁾ was used.

2. Plant

The start clutch operates when the piston is actuated by hydraulic pressure that has been adjusted by the LSOL (Fig. 3). When current is supplied to the LSOL, thrust is generated to operate the spool valve and open the oil feed port. Oil is supplied from the oil pump to generate clutch pressure. Conversely, the clutch pressure is reduced by lowering the LSOL thrust to open the drain. The pulsation of the line pressure also affects the clutch pressure, so an accumulator is provided to reduce these effects.

2.1. Modeling

In the LSOL, the characteristics of the input current I_{SC} and thrust F_{SC} generated by the LSOL are approximated by a first-order system.

$$F_{SC} = \frac{k_{IF}}{T_{IF} + 1} I_{SC} \quad (1)$$

The flow Q_{SC} at the LSOL opening depends on the LSOL spool displacement x_{SC} and clutch pressure P_{SC} . In

addition, the flow Q_f to the F/B area of the spool is proportionate to the square root of the pressure difference between P_{SC} and the spool pressure P_f .

$$Q_{SC}(x_{SC}, P_{SC}) = \begin{cases} -CA_{SC}(x_{SC})\sqrt{\frac{2P_{SC}}{\rho}} & (0 \leq x_{SC} \leq x_d) \\ CA_{SC}(x_{SC})\sqrt{\frac{2|P_{CR}-P_{SC}|}{\rho}} \text{sgn}(P_{CR}-P_{SC}) & (x_d < x_{SC} \leq x_{SCmax}) \end{cases} \quad (2)$$

$$Q_f(P_{SC}, P_f) = C \frac{\pi d_o^2}{4} \sqrt{\frac{2(P_{SC} - P_f)}{\rho}} \quad (3)$$

ρ : Oil density

x_d : Drain fully closed position

P_{CR} : Line pressure

x_{SCmax} : Stroke length

d_o : Orifice diameter

Here, C is the flow coefficient, which changes according to the oil temperature. The opening area A_{SC} depends on displacement x_{SC} and is expressed by the following equation.

$$A_{SC}(x_{SC}) = \begin{cases} \pi d_m(x_d - x_{SC}) & (0 \leq x_{SC} \leq x_d) \\ 0 & (x_d < x_{SC} < x_o) \\ 2L_N(x_{SC} - x_o) & (x_o \leq x_{SC} < x_N) \\ 2L_N(x_N - x_o) & \\ + \pi d_m(x_{SC} - x_N) & (x_N \leq x_{SC} \leq x_{SCmax}) \end{cases} \quad (4)$$

d_m : Spool diameter of the main area

x_o : Start position of the notch opening

L_N : Notch width

x_N : Fully open notch position

The following equations pertain to P_{SC} and P_f .

$$\begin{aligned} \frac{V_{CL}}{K} \frac{dP_{SC}}{dt} &= Q_{SC}(x_{SC}, P_{SC}) - Q_f(P_{SC}, P_f) - A_{AC} \frac{dx_{AC}}{dt} \\ \frac{V_f}{K} \frac{dP_f}{dt} &= Q_f(P_{SC}, P_f) + A_f \frac{dx_{SC}}{dt} \end{aligned} \quad (5)$$

V_{CL0} : Initial capacity of the accumulator area

V_{f0} : Initial capacity of the F/B area

A_{AC} : Pressure receiving area of the accumulator area

A_f : Pressure receiving area of the LSOL area

However, note that: $V_{CL} = V_{CL0} + A_{AC} x_{AC}$ and $V_f = V_{f0} - A_f x_{SC}$. K is the bulk modulus, which changes according to the oil temperature and air mix rate inside the oil.

The following relationship exists between x_{SC} , from the balance of forces on the spool, and spool displacement of the accumulator area, x_{AC} .

$$\begin{aligned} m_{SC} \frac{d^2 x_{SC}}{dt^2} + D_{SC} \frac{dx_{SC}}{dt} + k_{SC} x_{SC} &= F_{SC} - P_f A_f - F_{fld} - f_{SC0} \\ m_{AC} \frac{d^2 x_{AC}}{dt^2} + D_{AC} \frac{dx_{AC}}{dt} + k_{AC} x_{AC} &= P_{SC} A_{AC} - f_{AC0} \end{aligned} \quad (6)$$

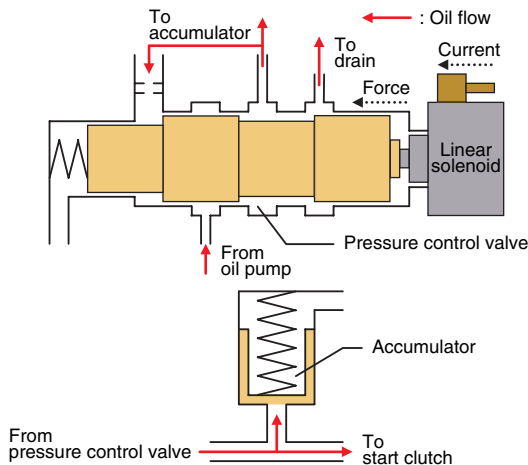


Fig. 3 Hydraulic circuit

m_{SC} : Spool mass of the LSOL area
 m_{AC} : Spool mass of the accumulator area
 k_{SC} : Spring constant of the LSOL area
 k_{AC} : Spring constant of the accumulator area
 f_{SC0} : Spring initial load of the LSOL area
 f_{AC0} : Spring initial load of the accumulator area
 F_{fld} : Fluid force

Here, the ranges of motion for the spools are $0 \leq x_{SC} \leq x_{SC \max}$ and $0 \leq x_{AC} \leq x_{AC \max}$ respectively. D_{SC} and D_{AC} are the coefficients of viscosity, which change according to the oil temperature.

2.2. Linear Model Derivation

The following state space is obtained by expressing the input as current δI_{SC} , the output as clutch pressure δP_{SC} , and linearizing the basic relationship equation obtained in the previous section.

$$\begin{cases} \delta \dot{x} = A\delta x + B\delta u \\ \delta y = C\delta x + D\delta u \end{cases}$$

$$\delta x = (\delta x_{SC}, \delta x_{AC}, \delta \dot{x}_{SC}, \delta \dot{x}_{AC}, \delta P_{SC}, \delta P_f, \delta F_{SC})^T$$

$$\delta u = \delta I_{SC}, \delta y = \delta P_{SC}$$

$$A = \begin{bmatrix} 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 & 0 \\ a_{31} & 0 & a_{33} & 0 & a_{35} & a_{36} & a_{37} \\ 0 & a_{42} & 0 & a_{44} & a_{45} & 0 & 0 \\ a_{51} & 0 & 0 & a_{54} & a_{55} & a_{56} & 0 \\ 0 & 0 & a_{63} & 0 & a_{65} & a_{66} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & a_{77} \end{bmatrix}, B = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ b_7 \end{bmatrix}$$

$$C = [0 \ 0 \ 0 \ 0 \ c_5 \ 0 \ 0], D = 0$$

$$a_{31} = -\frac{k_{SC} + K_{fldx}}{m_{SC}}, a_{33} = -\frac{D_{SC}}{m_{SC}}, a_{35} = -\frac{K_{fldP}}{m_{SC}}, a_{36} = -\frac{A_f}{m_{SC}} \quad (7)$$

$$a_{37} = \frac{1}{m_{SC}}, a_{42} = -\frac{k_{AC}}{m_{AC}}, a_{44} = -\frac{D_{AC}}{m_{AC}}, a_{45} = \frac{A_{AC}}{m_{AC}}$$

$$a_{51} = \frac{K_{KSC}}{V_{CL}}, a_{54} = -\frac{K_{AAC}}{V_{CL}}, a_{55} = \frac{K(K_{SCP} - K_f)}{V_{CL}}, a_{56} = \frac{KK_f}{V_{CL}}$$

$$a_{63} = \frac{KA_f}{V_f}, a_{65} = \frac{KK_f}{V_f}, a_{66} = -\frac{KK_f}{V_f}, a_{77} = -\frac{1}{T_{IF}}$$

$$b_7 = \frac{k_{IF}}{T_{IF}}, c_5 = \frac{1}{10^4 g}$$

Figures 4 and 5 show the step response when the oil temperature is 80 °C and -10 °C, and compare the actual response with a simulation using the obtained linear model. There is a good match in both cases. Figure 6 shows a large variation in performance, with a break frequency of approximately 100 (rad/s) when the temperature is 80 °C and approximately 40 (rad/s) when the temperature is -10 °C.

3. Controller Design

3.1. Controller Structure

The controller comprises a 2DOF control system to

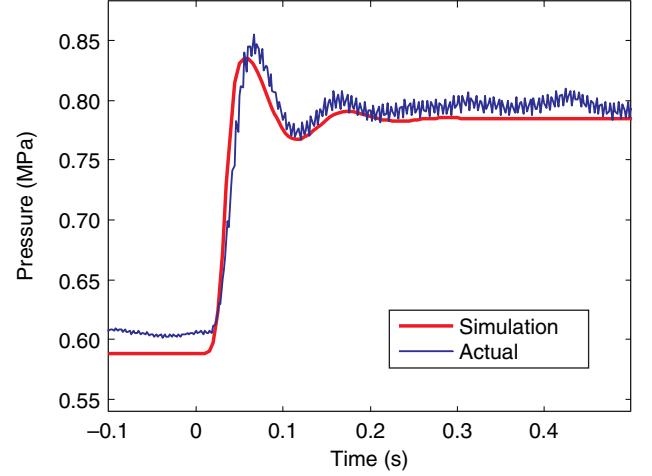


Fig. 4 Step response (80 °C)

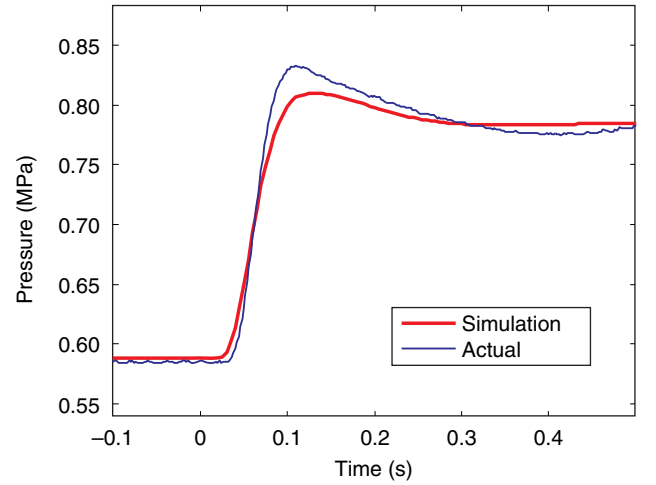


Fig. 5 Step response (-10 °C)

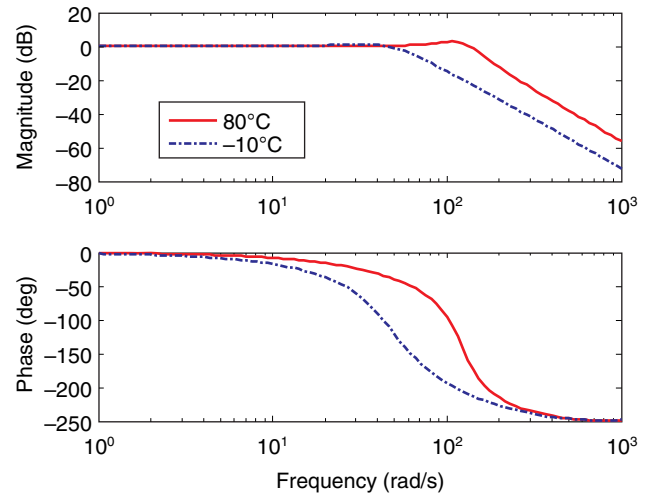


Fig. 6 Bode diagram

help achieve both a quick response and robustness. Although many construction methods have been proposed for 2DOF control, as shown in Fig. 7, the method of positioning an F/F controller and F/B controller in parallel was used.

First, to facilitate the setting of a target response, F/F control is expressed by the following equation, with $F(s)$ as the design parameter.

$$u_{ff}(s) = F(s)P(s)^{-1}r(s) \quad (8)$$

At this time, if there are neither disturbances nor modeling errors, then the following is established.

$$y(s) = P(s)u_{ff}(s) = F(s)r(s) \quad (9)$$

Furthermore, to prepare for cases where issues such as modeling errors invalidating the above equation, the following equation expresses the F/B for the difference between the desired output $F(s)r(s)$ and actual output $y(s)$.

$$u_{fb}(s) = K(s)(F(s)r(s) - y(s)) \quad (10)$$

The control input is established by the following.

$$u(s) = u_{ff}(s) + u_{fb}(s) \quad (11)$$

This structure shapes the target response with $F(s)$ and reduces the effects of disturbances and plant characteristic perturbations with $K(s)$, and so has the advantage that these characteristics can be set independently.

As shown in the previous section, there is a large variation in performance between 80 °C and -10 °C oil temperature. For a single F/B controller to maintain robustness under the influence of these kinds of large perturbations, there is no option except to sacrifice control performance and set a conservative F/B. Consequently, two ranges were set: 20 °C or below, and above 20 °C. Separate controllers were designed for these ranges, and the system switches between them. In this way, both control performance and robustness could be achieved.

3.2. F/F Controller Design

First, reference model $F(s)$ was determined to satisfy the specifications of the rise time. Next, the plant $P(s)$ inverse system was used to obtain $H(s)$. In other words: $H(s) = F(s)P(s)^{-1}$.

3.3. F/B Controller Design

H_∞ control is a typical method for robust control

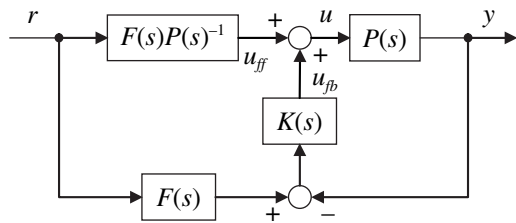


Fig. 7 Two degree of freedom control system

systems. Although design of a controller to stabilize plant set with uncertainties such as time delays and characteristic perturbations in the hydraulic circuit due to temperature changes can be achieved, as a rule, each uncertainty has to be gathered. However, in reality, uncertainties rarely gather in one location only. In general, they are found in multiple locations such as the actuator and sensor (Fig. 8). If these uncertainties are all dealt with together (Fig. 9), areas where uncertainties do not actually exist will also be included. As a result, unnecessary and excessive constraints will be placed on the controller, so that the disturbance rejection characteristics and quick response will not be sufficient.

In contrast, with the μ synthesis⁽⁵⁾, uncertainties can be dealt with as they are. In Fig. 10, the off-diagonal elements are 0 and only the uncertainties that actually exist are dealt with. Using this method, a controller with a better performance can be designed.

Here, a μ controller is designed for structured uncertainties, while considering the generalized plant shown in Fig. 11. The weighting function $W_2(s)$ represents the range of multiplicative perturbations. The broken line in Fig. 12 shows the result of plotting the difference between the nominal model obtained from the average value of anticipated perturbations, such as parameter changes dependent on the temperature and time delays,

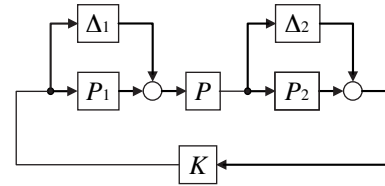


Fig. 8 Original uncertainties

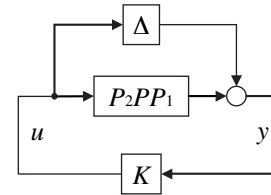


Fig. 9 Unstructured uncertainties

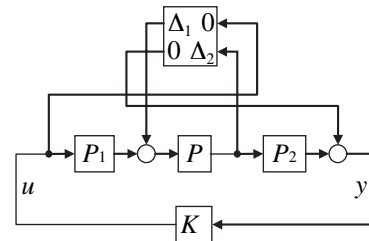


Fig. 10 Structured uncertainties

and the perturbation model that was obtained from the maximum and minimum values. By selecting $W_2(s)$ which covers these plots, robust stabilization was achieved for the anticipated perturbations. $W_1(s)$ is the dynamic characteristic of disturbance, $W_3(s)$ is the weight for the input, and $W_4(s)$ is the fitting parameter. These are selected as shown in Fig. 13 after trial and error.

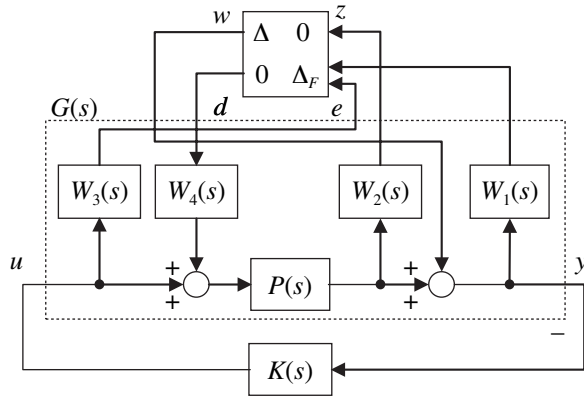


Fig. 11 Generalized plant

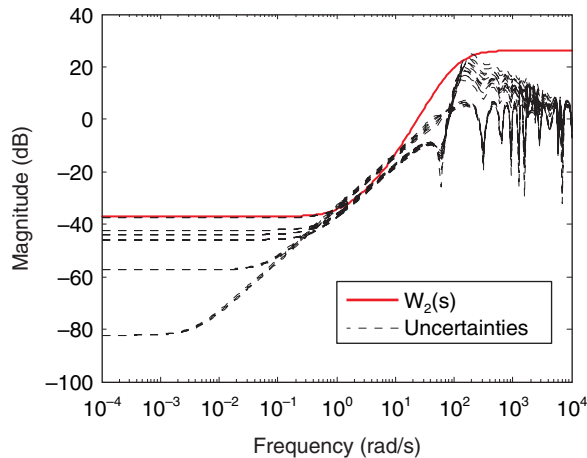


Fig. 12 Bode diagram of uncertainty and weighting function

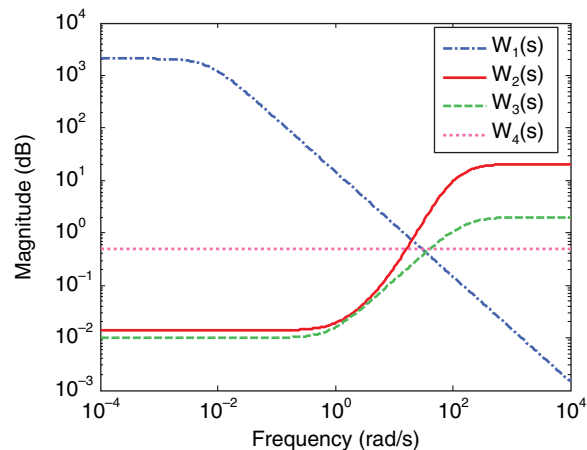


Fig. 13 Bode diagram of weighting function

Table 1 Result of DK iteration

Number of times	1	2	3	3'
Order of K	7	11	13	13
Order of D	0	4	6	6
γ	1.563	0.941	0.922	0.940
μ	1.451	0.941	0.922	0.939

However, DK iteration, the most typical method, was used to obtain the μ controller.

Table 1 shows the trends for μ and γ with DK iteration. 3' for "Number of times" in the table is a recalculation with γ set as the suboptimal. Figure 14 shows the μ plot for a closed-loop system constructed with a μ controller obtained with the first and third DK iteration. Figure 14 shows that for the first iteration, the μ value exceeded 1 in the vicinity of 10-100 (rad/s), but it was below 1 over the entire range for the third iteration. As shown in Fig. 15, this is because the weighting function is adjusted automatically in the DK iteration process, and in particular, the intermediate frequency band is optimized. The Bode diagram of the controller in Fig. 16 shows that the phase proceeds rapidly in the vicinity

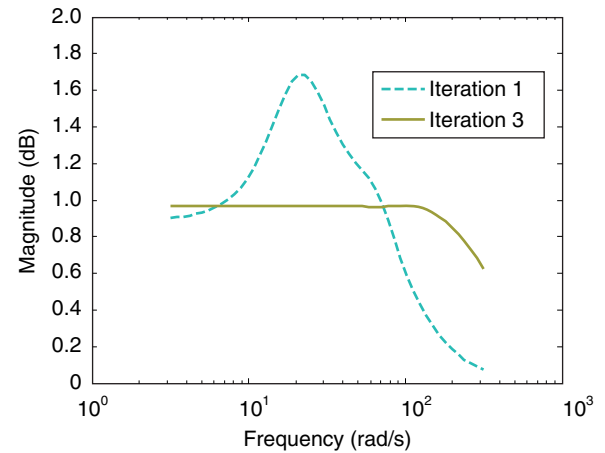
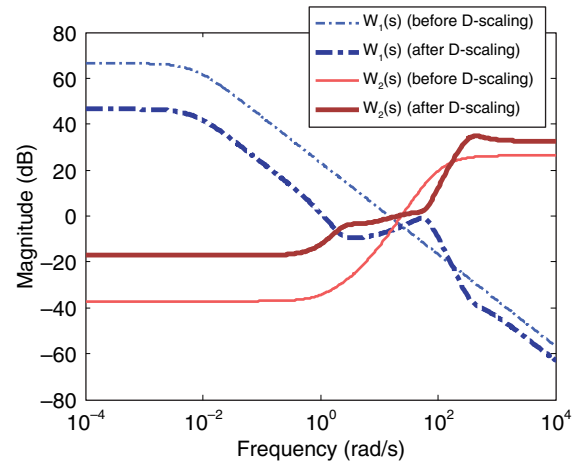
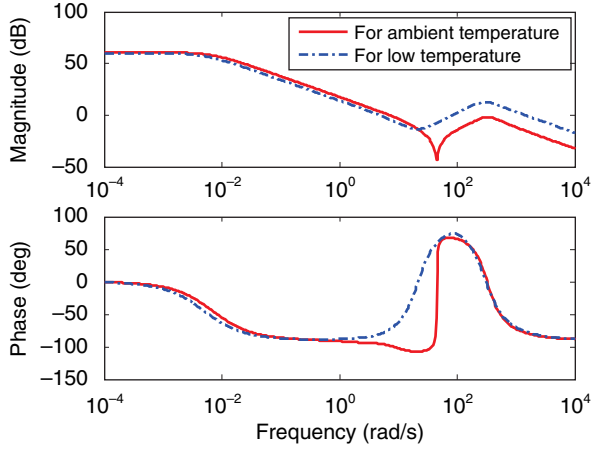
Fig. 14 μ plot

Fig. 15 Result of D-scaling


 Fig. 16 Bode diagram of μ controller

of 10-100 (rad/s), and it can be understood that strong stabilization has been provided to widen the control band.

The ideal frequency characteristics for the controller are typically to increase the gain in order to maintain steady-state characteristics in the low range, while having cut-off characteristics and a gradient of 40-60 (dB/dec) in the high range to reduce the influence of uncertainties and noise. However, other methods can also achieve these objectives with no difference. The intermediate frequency band, however, has a large influence on performance, for which adjustment is difficult. As already described, in μ synthesis, the intermediate frequency band is automatically optimized to a level that is hardly attained manually, so it is a useful tool.

4. Actual System Test Results

4.1. Step Response

Figures 17 and 18 show the step response performed with an actual system, with oil temperatures of 80 °C and -10 °C. At 80 °C, the rise time was approximately the same compared to the open-loop control and the overshoot rate was reduced by approximately 65%. At -10 °C, the

rise time was approximately 20% shorter and the overshoot rate was approximately 0%. These resulted in almost the same rise time and overshoot rate at both 80 °C and -10 °C.

From the controller outputs in Fig. 17 and 18, it can be understood that sufficient rising characteristics are maintained by the F/F controller output at first, then deviations from the target response, particularly the

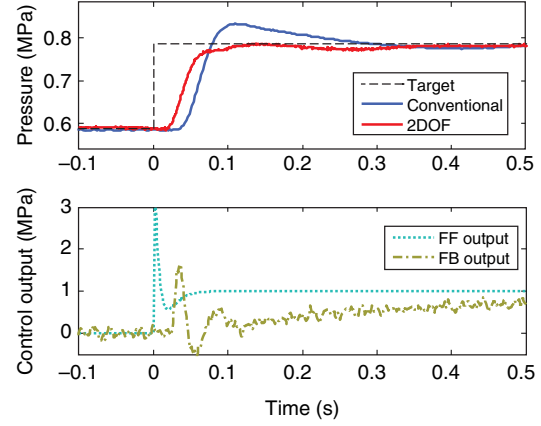


Fig. 18 Step response (-10 °C)

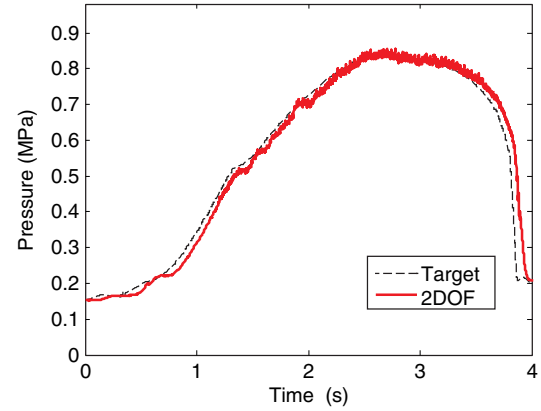


Fig. 19 Operating results (80 °C)

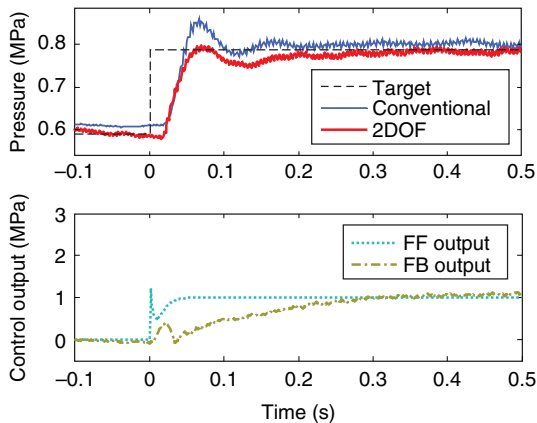


Fig. 17 Step response (80 °C)

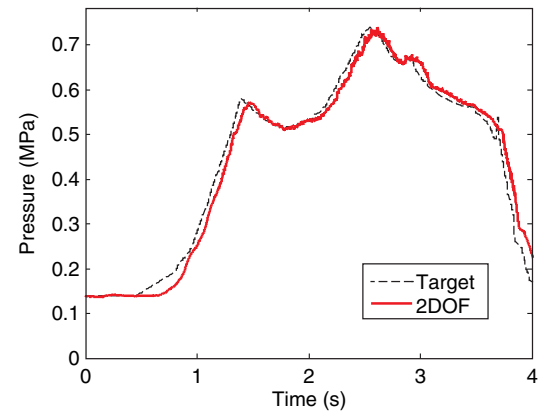


Fig. 20 Operating results (5 °C)

overshoot and steady-state errors, are reduced by the F/B controller, and the 2DOF control system is functioning effectively.

4.2. Operating Results

Figures 19 and 20 show the actual operating results. It can be understood that tracking is good at both 80 °C and 5 °C.

5. Conclusion

A hydraulic servo system was constructed using a 2DOF control system to achieve the robust start clutch control of the CVT start clutch for temperature changes. As a result, the following were achieved.

- (1) The overshoot rate was reduced by approximately 65% at ambient temperature (approximately 80 °C).
- (2) The rise time was shortened by approximately 20% and an overshoot rate of approximately 0% was achieved at low temperature (approximately -10 °C).
- (3) Almost the same values were achieved for the rise time and overshoot rate for both ambient temperature and low temperatures.

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Development of Non-normalized Nitrocarburized High Strength Crankshaft

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ABSTRACT

A non heat-treated nitrocarburized crankshaft was developed by designing the material composition so as to maximize the effectiveness of the strength increase due to nitrocarburization. This crankshaft has a fatigue strength that is approximately 16% higher than that of a conventional crankshaft of the same material hardness, while possessing the same workability and straightening ability. Strengthening ferrite by adding a relatively small quantity of molybdenum (Mo) gave the developed material both high fatigue strength and excellent straightenability. The mechanism of strengthening by the addition of Mo is thought not to be the secondary hardening due to the precipitation of molybdenum carbonitride, but the formation of a Mo cluster consisting of several atoms during the nitrocarburizing process. This results in a greater strengthening effect compared to solid solution strengthening, thus contributing to higher yield strength. It was found that compared to the conventional high strength crankshaft, the elimination of the normalization process and so on reduced CO₂ emission during and after the hot forging process by approximately 22%, and costs by approximately 8%.

1. Introduction

There are diverse demands on modern automotive engines. In addition to environmental performances such as high fuel efficiency and low exhaust emissions, demands for more weight reduction, more power, and lower costs also exist. To respond to these demands, it is necessary to increase the strength, reduce the cost, and reduce the production energy of the crankshaft that is one of the structural components of the engine. Carbon steel is the base material used for the majority of crankshafts that are manufactured by hot forging. The characteristics of the component are enhanced by additional processing and heat treatment such as secondary heat treatment, surface rolling, induction hardening and nitrocarburization^{(1) - (5)}. When a material is strengthened by nitrocarburization, steels with relatively high carbon contents (0.45 mass% or more) are used, and quenching and tempering or normalizing are often used to increase the strength of the base material. The nitriding elements such as aluminum (Al), chromium (Cr), vanadium (V), which promote the formation of nitrides, are added to

increase the strength around the surface of the product after nitrocarburization⁽⁶⁾. However, these methods all involve increases in production energy, CO₂ emissions, and costs due to the secondary processing. In addition, the high hardness of the materials reduces machinability, and the high hardness of the surface layers following nitrocarburization reduces straightenability.

The research discussed in this paper attempted to develop a nitrocarburized crankshaft that would display no reduction in machinability or straightenability while strengthened without any requirement for heat treatment.

2. Development Goals

Honda engines have used two different types of crankshafts, produced from the normal-strength and high-strength materials. Both types of material are nitrocarburized to enhance their mechanical properties. While the high-carbon steel is used in the high-strength crankshafts with normalization after hot forging to increase its strength higher than that of the normal-strength material, the normal-strength material does not

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undergo normalization. Since the normal-strength material is based on lowering the hardness of the raw material and that of the surface layers after nitrocarburization compare to high-strength material, it displays better machinability and straightenability. The production cost of normal-strength crankshafts is therefore lower than that of high-strength crankshafts. The goal of this research was to develop a crankshaft material that would display an equivalent level of strength to the current high-strength material while maintaining production costs equivalent to those of the current normal-strength material, and a material which would satisfy the following criteria was studied.

- (1) To be produced without normalization (Equivalent to the current normal-strength material process)
- (2) To have high fatigue strength (Equivalent to the current high-strength material)
- (3) To have low raw material hardness (Equivalent to the current normal-strength material)

3. Concept of Material Alloy Design

3.1. Basic Composition Setting

As shown in Fig. 1 (a), when the conventional high-strength material is not normalized, its structure becomes coarse with a low ferrite area ratio, and straightenability declines. It is therefore necessary to increase the ferrite area ratio and the nucleation sites of the proeutectoid ferrite so that the material does not need to be normalized. For this matter, the same method used for the normal-strength material, which is not normalized either, should be used⁽⁷⁾. This involves refining the structure by increasing the ferrite area ratio through optimization of the amount of carbon (C) and manganese (Mn), and by means of promoting the nucleation of proeutectoid ferrite through the addition of nitrogen (N). And as shown in Fig. 2, reducing the amount of C content and increasing the amount of Mn is effective in enhancing the fatigue strength and straightenability of this type of non-normalized nitrocarburized steel. Straightenability was evaluated by the value of bending strain when cracking occurred (crack strain)⁽⁸⁾. If a material displays a high degree of plastic deformation, i.e., if the level of bending that can be straightened is high, the material is said to

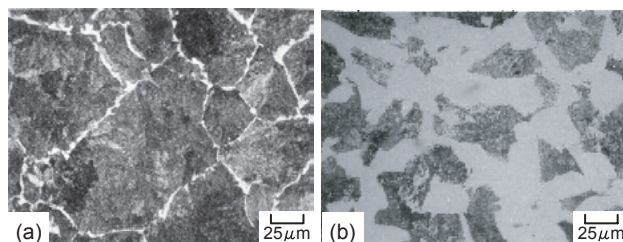


Fig. 1 Optical micrographs of as-forged state:
(a) Conventional high-strength steel
(b) Developed steel

possess a high degree of straightenability.

Crankshafts should also possess wear resistance at the points of contact with the oil seals, and the characteristics depend on the amount of C content as shown in Fig. 3.

Based on these considerations, the amount of C and Mn in the steel was adjusted according to the relationship shown in Fig. 4 as no less than 0.35 mass% for C and the ferrite ratio equivalent to that of the normal-strength

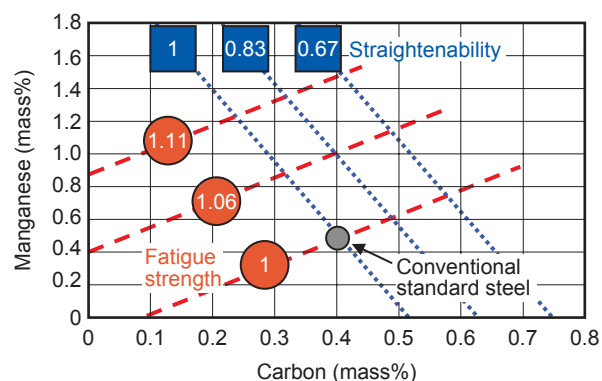


Fig. 2 Correlation diagram between content of alloying additions and properties of nitrocarburized specimen

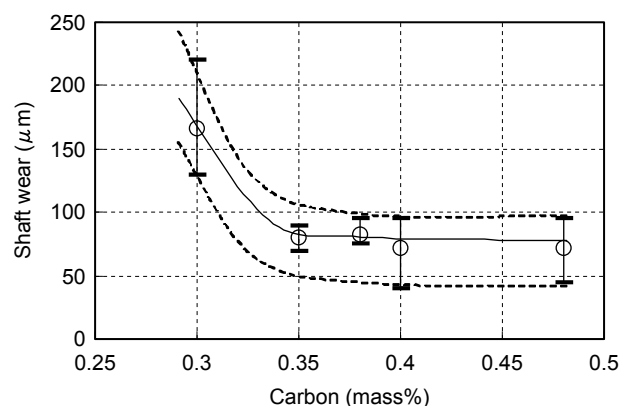


Fig. 3 Relationship between carbon content in steel and wear loss at oil seal section of crankshaft

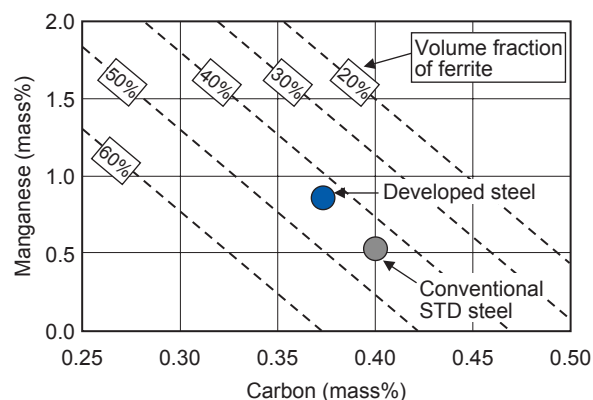


Fig. 4 Relationship between content of alloying additions and volume fraction of ferrite

material. This resulted in the achievement of a fine structure with a high ferrite area ratio after hot forging as shown in Fig. 1 (b).

3.2. Achievement of High-strength by Addition of Molybdenum (Mo)

Though a high ferrite area ratio was achieved, the achievement of the target fatigue strength would require strengthening of the ferrite structure and hardening of the surface layer after nitrocarburization. However, if the surface layers become excessively hard, ductility declines. The method used to strengthen the material, i.e., the selection of the additive elements, is therefore an important issue. Si, Ni, Mo, V, and Al are considered to be appropriate ferrite-strengthening agents in common structural steels^{(9), (10)}. However, Si and Ni make almost no contribution to increased strength after nitriding⁽¹¹⁾, and they are not expected to increase fatigue strength. On the other hand, Al, Cr, and V form hard nitrogen compounds,

hardening the surface layer^{(12), (13)}, and result in a significant decline in straightenability. For these reasons, none of these elements should be used.

Molybdenum (Mo), however, is effective in solid solution strengthening of ferrite and it also hardens the surface appropriately after nitriding. Test pieces were fabricated for the planar bending fatigue tests as shown in Fig. 5 to study the effect of Mo on fatigue strength. As Fig. 6 shows, even a small amount of the addition of Mo results in increased fatigue strength. According to this result, the amount of Mo to be added to increase strength was determined to 0.16 mass%. The next section will discuss the mechanism for the achievement of high strength by means of the addition of Mo.

4. Strengthening Mechanism by Addition of Mo

4.1. Increase in Yield Ratio during Nitrocarburization

Nitrocarburization is thought to have an effect on the mechanism of strength increase by the addition of Mo. The changes in mechanical properties after a thermal history corresponding to the nitrocarburization process (600 °C x 2h) were examined. To provide a comparison, tensile tests of test pieces that did not contain Mo were also conducted. Figure 7 shows the preparation procedure of the test pieces. Table 1 shows the chemical composition of the materials used for the tests, and Table 2 shows the test results. The material that did not contain Mo displayed no change in yield ratio (yield point/tensile strength) before and after the heat treatment, while the material containing Mo displayed an increase of approximately 6% in yield ratio. In general, the higher the

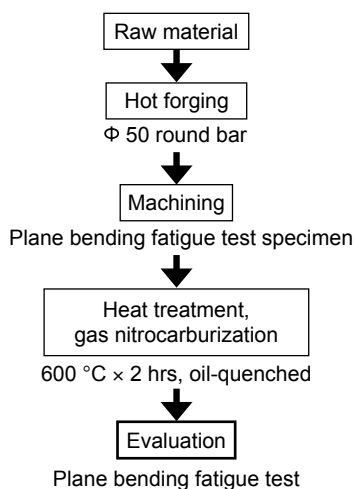


Fig. 5 Method of specimen preparation and property evaluation

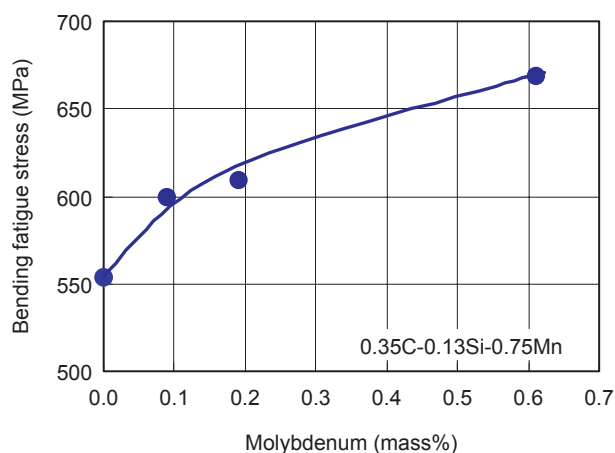


Fig. 6 Effect of Mo on plane bending fatigue strength in developed steel

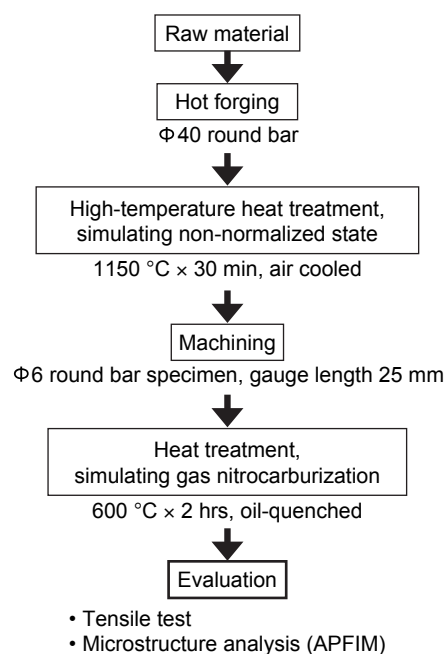


Fig. 7 Method of specimen preparation and property evaluation

Table 1 Chemical composition of Mo-added and Mo-free steel, in mass %

	C	Si	Mn	P	Cu	Ni	Cr	Mo	Ti	N
Mo-added steel	0.35	0.21	0.76	0.016	0.01	0.02	0.08	0.16	0.008	0.0180
Mo-free steel	0.41	0.19	0.58	0.014	0.01	0.02	0.06	0.01	0.009	0.0190

Table 2 Variation of mechanical properties of Mo-added and Mo-free steel, before and after heat treatment at 600°C for 2 hours

Material	Before 600 °C × 2h				After 600 °C × 2h			
	Tensile strength	Yield point	Elongation	Yield ratio (YP/TS)	Tensile strength	Yield point	Elongation	Yield ratio (YP/TS)
Mo-added steel	668 MPa	387 MPa	23.2%	0.579	658 MPa	402 MPa	23.2%	0.611
Mo-free steel	676 MPa	352 MPa	22.4%	0.521	653 MPa	340 MPa	22.8%	0.521

yield ratio of a material, the higher its fatigue strength⁽¹⁴⁾. This result therefore indicates that the fatigue strength of steel containing Mo increases merely by the temperature rise during the nitrocarburization treatment.

4.2. State Analysis of Mo

In order to clarify the strengthening mechanism by addition of Mo including the reason for the increase in the yield ratio from the perspective of the microstructure of the material, an atom probe field ion microscope (APFIM)⁽¹⁵⁾ was used to analyze the distribution of Mo at the atomistic level.

4.2.1. APFIM

The APFIM is a combination of a field ion microscope (FIM) and a time-of-flight mass spectrometer. By using the FIM, the arrangement of atoms on the surface of a material test piece can be observed as a collection of luminescent spots respectively corresponding to each atom. Figure 8 shows a schematic diagram of the APFIM, and the principle is as follows.

- (1) A strong electrical field, several V/Å, is generated at the tip of a needle-shaped test piece when high voltage is applied.

- (2) Atoms are successively stripped off the surface of the test piece, and are emitted into a vacuum as positive ions (field evaporation).
- (3) The mass of the ions, i.e., their atomic type, is identified by their time-of-flight to a detector and the applied voltage.

Ions reaching the detector are sorted out via a tiny hole called the “probe hole.” As a result, atoms are sampled from within a cylinder of a height (depth) of approximately 100 nm, the area of the bottom equivalent to the area of projection of the probe hole onto the surface of the test piece (diameter: approximately 2 nm). This volume is indicated by the dotted line in the figure. Two types of information regarding the individual atoms are obtained: the atomic type and the order in which the atom was detected (corresponding to its position in the direction of the depth). This enables the APFIM to detect fine precipitates at the early stage of precipitation and clusters of solute atoms, which are difficult to be detected with a transmission electron microscope (TEM)^{(16), (17)}. The APFIM analysis is usually conducted in an ultra-high vacuum. When extremely small amounts of neon or helium is introduced into the vacuum chamber, these inert gas atoms become ionized on the surface of the test piece

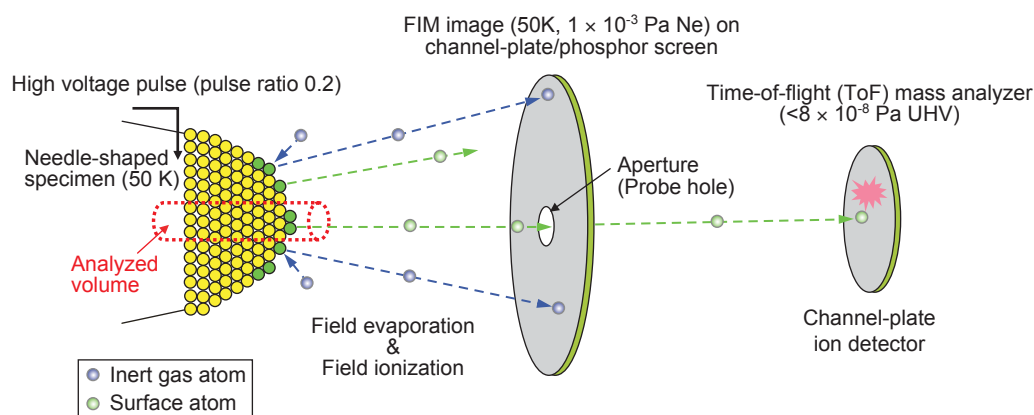


Fig. 8 Schematic diagram showing principles of atom-probe field ion microscopy (APFIM) and analyzing condition of present study

and produce light when they strike a facing fluorescent screen, and the atomic arrangement on the surface of the test piece can be observed as a collection of luminescent spots. These are called FIM images. As this ability to image individual atoms, the FIM provides several millions magnifications as a microscope.

4.2.2. Results of APFIM analysis

Figure 9 shows a FIM image of steel containing Mo that has been heated for 2 hours at 600 °C. The number of brighter spots that display a higher degree of luminescence than neighboring are looks increasing after heat treatment (marked by circles in the images). These spots are thought to be Mo clusters.

If atoms of alloying element with a higher evaporation

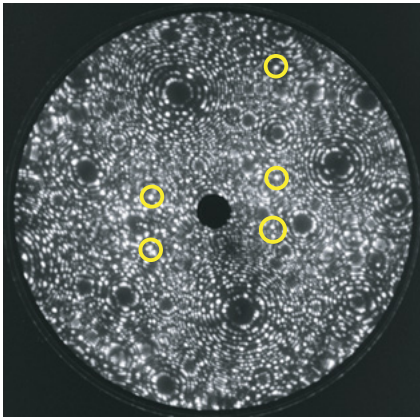
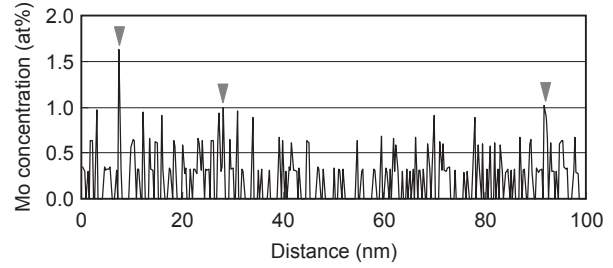
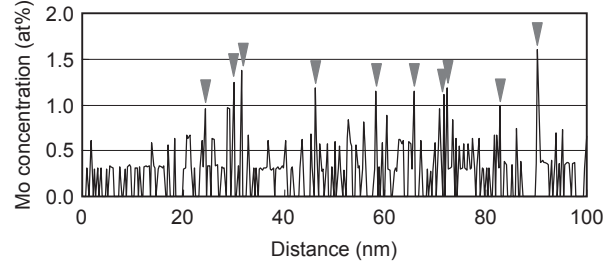


Fig. 9 FIM image of developed steel containing 0.2mass% Mo, after aged at 600 °C for 2 hours

field than the matrix aggregate and form precipitates in the material undergoing the FIM observation, the atoms in the surrounding matrix will evaporate preferentially first when the electrical field is applied. Precipitates therefore tend to remain on the surface. Due to the edge effect of these precipitates, atoms of the noble gases are more frequently field-ionized at the precipitates. Accordingly, these sections appear brighter than the surroundings. This also happens to the clusters of alloy elements that are much smaller than ordinary precipitates.



(a) As-forged (non heat-treated) specimen



(b) Aged specimen (aged at 600 °C for 2 hours)

Fig. 10 Concentration profiles of developed steel containing 0.2mass% Mo

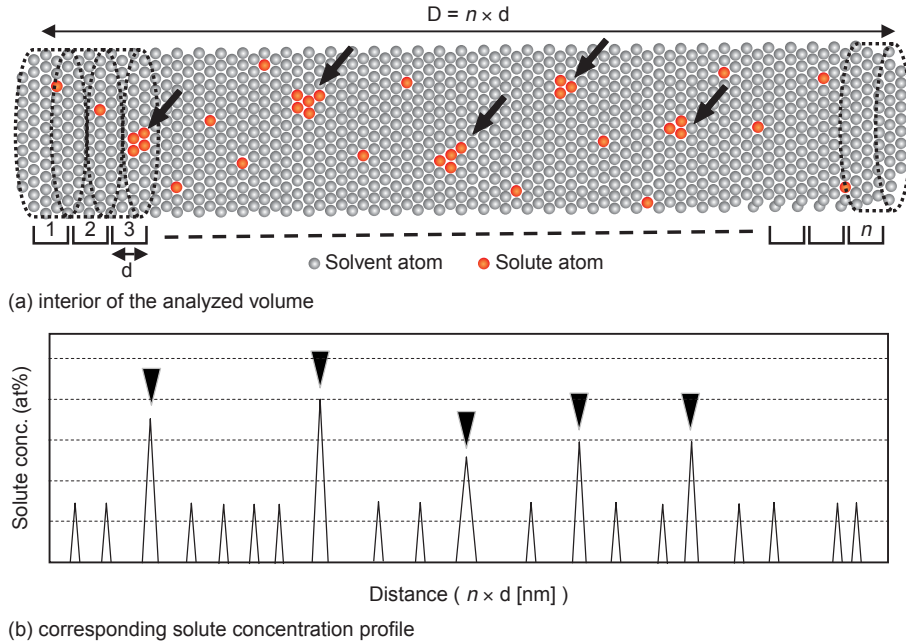


Fig. 11 Schematic diagram

Table 3 Chemical composition of developed and conventional steel, in mass %

	C	Si	Mn	P	S	Cr	Mo	N	Ti	Pb
Developed high-strength steel	0.37	0.20	0.82	0.015	0.07	-	0.16	0.015	0.01	-
Conventional standard steel	0.40	0.25	0.52	0.015	0.05	-	-	0.015	0.01	-
Conventional high-strength steel	0.48	0.30	1.42	0.015	0.05	-	-	-	-	-

To provide further verification of the existence of these Mo clusters, the APFIM was used to conduct a detailed examination of the distribution of Mo atoms. The results are shown in Fig. 10. The detected ions were sequentially divided into blocks of a depth of approximately 3 Å, basically corresponding to the lattice constant of iron, and the concentration of Mo in each block was determined. The Mo concentration profile resulting from this procedure is shown in Fig. 10. Each block contains approximately 300 ions.

Figure 11 is a schematic for interpreting the data shown in Fig. 10. When the analyzed volume where atoms are counted (the area indicated by the dotted line in Fig. 8) is enlarged, it should be as shown in Fig. 11 (a). The arrows in the figure are supposed to indicate the presence of clusters of solute atoms. By dividing this column into discs (unit blocks) of the appropriate thickness, d , the concentration of solute atoms in the disc can be calculated from the number of solvent and solute atoms. Plotting the concentration values sequentially in the direction of depth gives the solute atom concentration profile as shown in Fig. 11(b). If clusters of solute atoms exist in the unit blocks, the concentration of solute will be higher than in the case of random or homogeneous solid solution. The existence of these clusters is demonstrated by spikes in the concentration profile as shown by the ▼ mark in the figure.

Comparison of the as-forged material and the material subjected to heat treatment (600 °C x 2h) after hot forging demonstrates three to four times the frequency of aggregates, which are thought to be Mo clusters, in the heat-treated material. The Mo concentration in the case of uniform solid solution was approximately 0.1 at%. Concentrations of 1.0 at% and higher were regarded as representing Mo clusters. These results indicate that the formation of Mo clusters is promoted by exposure to a thermal history corresponding to the thermal conditions of nitrocarburization. In addition to the Mo clusters that form at 600 °C for 2 hours, the diffusion layers close to the surface of the material that underwent nitrocarburization are also thought to contain Mo and N clusters as a result of the absorption of N into the Mo clusters. However, because the APFIM cannot distinguish between bivalent Si^{2+} ions and monovalent N^+ ions, which both possess a mass-valence ratio of 14, this was not verified in the developed steel, which contains approximately 0.2% Si.

Next, a test piece taken from the nitrogen diffusion

layer (approximately 50 μm below the surface) of the steel containing Mo that had undergone nitrocarburization was subjected to TEM analysis. Mo nitrides were not observed in this analysis. As Fig. 12 shows, the only nitrides observed in the analysis were rod-shaped γ' - Fe_4N and particulate α'' - Fe_{16}N_2 , both iron nitride precipitates. In addition, as Fig. 13 shows, the concentration of N in the diffusion layer of this Mo cluster-forming steel following nitrocarburization is higher than that of the normal-strength steel, which does not contain Mo.

4.3. Considerations Regarding Strengthening Mechanism

The results described above led to the following

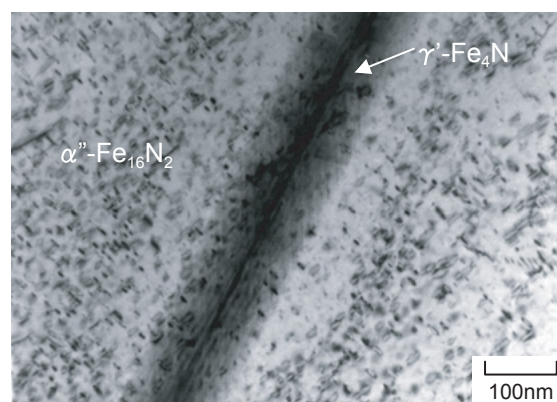


Fig. 12 Transmission electron micrograph taken from diffusion layer of developed steel containing 0.2mass% Mo, after nitrocarburization

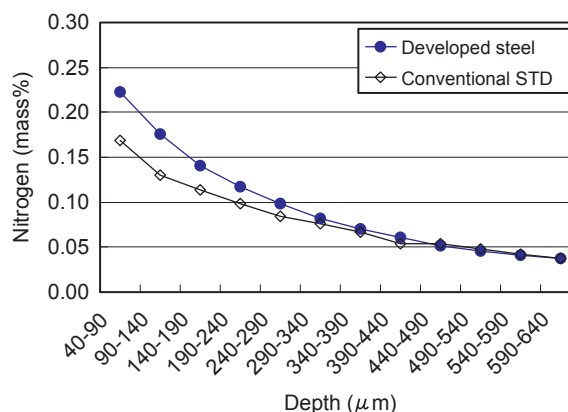


Fig. 13 Nitrogen content of steel after nitrocarburization

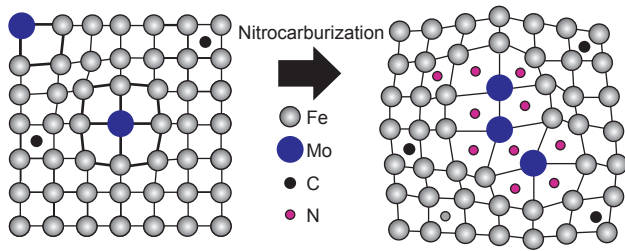


Fig. 14 Schematic illustration of Mo clustering process during nitrocarburization in developed steel

considerations regarding the mechanism that generates increased fatigue strength after nitrocarburization in steel containing 0.16 mass% Mo. As Fig. 14 shows, as the temperature during nitrocarburization rises, the Mo in solid solution in the ferrite forms clusters of several atoms, generating a higher level of lattice strain than single Mo atoms. During nitrocarburization, a higher amount of N than usual infiltrates the spaces between these distorted lattices and become trapped. These areas of high strain prevent dislocation motion, thus increasing yield stress and resulting in increased fatigue strength.

5. Crankshaft Characteristics

On the basis of the results discussed above, the chemical composition of the developed material was established as shown in Table 3. Table 3 also shows the composition of the current standard- and high-strength crankshaft materials. L4 crankshafts for 2.4L engines were manufactured using the developed materials and its performances were tested.

5.1. Mechanical Properties of Crankshaft

As shown in Fig. 15, the internal hardness of the newly developed crankshaft is lower than that of the current high-strength crankshaft, equivalent to that of the normal-strength crankshaft. However, the hardness around the surface layers after nitrocarburization is equivalent to

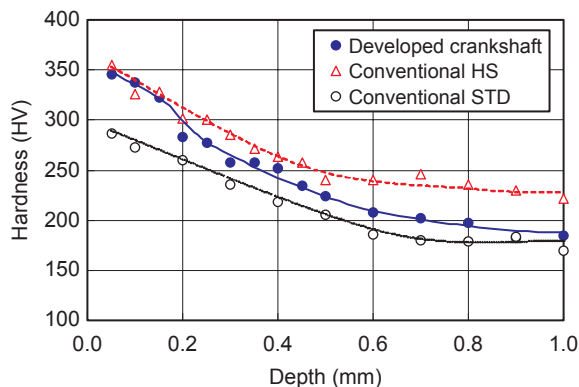


Fig. 15 Hardness distribution of crankshafts

Table 4 Bending fatigue strength and strain for cracking of crankshafts
(Conventional standard crankshaft = 100)

	Bending fatigue strength	Strain for cracking
Developed HS	116	90
Conventional STD	100	100
Conventional HS	113	73

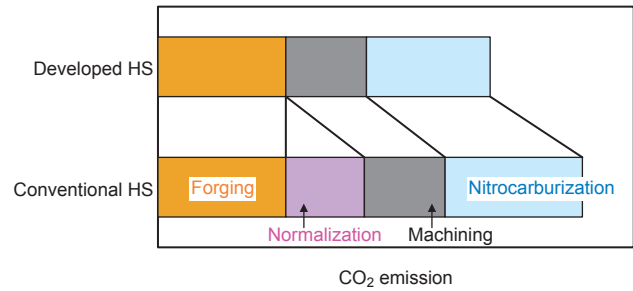


Fig. 16 Comparison of the total CO₂ emission and its constituents during production between developed and conventional high-strength crankshafts

that of the current high-strength crankshaft. As Table 4 shows, the bending fatigue strength of the developed crankshaft is equivalent to or higher than that of the current high-strength crankshaft, and 16% higher than that of the current normal-strength crankshaft. The crack strain of the new crankshaft is higher than that of the current high-strength crankshaft, but compares favorably with that in the current normal-strength crankshaft.

5.2. Merit by Application of Developed Material

As Fig. 16 shows, use of the new material has reduced CO₂ emissions in the procedures after hot forging by a maximum of approximately 22%, and has reduced costs by approximately 8%, compared to the current high-strength material.

6. Conclusion

A non-normalized high-strength crankshaft material was developed. The followings are the salient points of this development project:

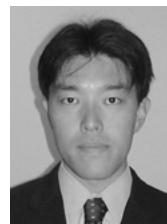
- (1) Addition of 0.16% mass% Mo to a crankshaft material increases fatigue strength by approximately 16% after nitrocarburization at 600 °C for 2 hours.
- (2) The yield ratio of steel containing Mo increases when the steel is only subjected to a thermal history corresponding to that of nitrocarburization (600 °C x 2h).
- (3) APFIM analysis showed that the formation of Mo clusters was promoted in the steel containing Mo that was subjected to a thermal history corresponding to that of nitrocarburization.

- (4) The N concentration in the diffusion layers of steel containing Mo after nitrocarburization was higher than that in steel that did not contain Mo.
- (5) The mechanism of the increase in fatigue strength after nitrocarburization in steel containing Mo is thought to be the high level of strain produced in the lattices due to the formation of Mo clusters and the consequent increase in the concentration of N.

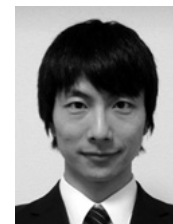
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Development of Piston Surface Treatment for Internal Combustion Engine

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ABSTRACT

Low-friction surface treatment technology that provided high wear resistance was developed for the piston skirt of an internal combustion engine. Conventional low-friction coating materials consist of a solid lubricant, such as molybdenum, and the effectiveness of performance was not expected to be maintained for a long time because lower friction was realized by self-lubrication, wear, and abrasion. The newly developed technology uses a high-strength film, which has an oil adsorption effect, enabling friction to be reduced by about 40% while increasing wear resistance about 5-fold as compared to conventional coating materials.

Concerning productivity, because this technology employs piston spray painting technology, the new film can be applied by simply replacing the material without any need for special production equipment.

1. Introduction

Efficient use of energy resources has been demanded due to social and economic background, such as the necessity of protecting the environment, emissions regulations, and depletion of fossil fuel reserves. Approximately 40% of the total energy generated by gasoline internal combustion engines is lost as mechanical loss⁽¹⁾. The reduction of mechanical loss therefore has a role to play in increasing the fuel efficiency and power of gasoline engines.

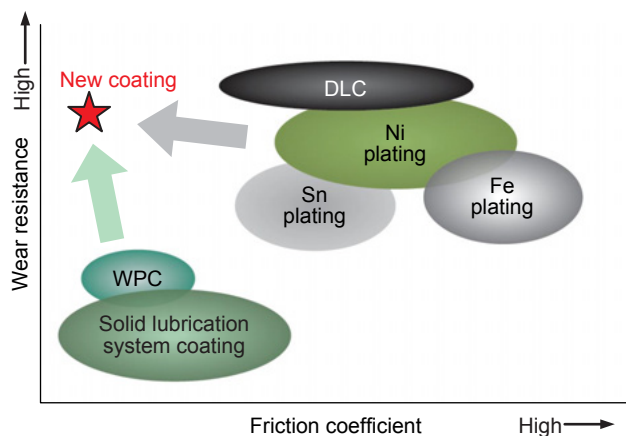


Fig. 1 Piston surface treatment

The sliding resistance of the pistons accounts for approximately 40% of mechanical loss⁽²⁾, and the development of technologies such as lighter components, low-tension piston rings, and piston surface treatments has resulted in increased mechanical efficiency.

The main objectives in the application of surface treatments to pistons are to enhance initial performance, to simplify the break-in run including adaptation to contact faces⁽³⁾, and to increase seizing resistance. The representative piston surface treatments are shown in Fig. 1.

The use of dry film lubricants is one typical piston surface treatment, which employs solid lubricants such as molybdenum disulfide, graphite, and PTFE. Solid lubricants reduce friction by means of cleavage (slipping on the faces of the crystal structure where the van der Waals force is weak) and other forms of self-lubrication, and abrasion is therefore unavoidable. Because of this, the retention of the film over an extended period remains an issue to be addressed.

2. Lubrication State of Piston Skirt

The important factor in piston tribology is that the cylinder bore and piston are able to maintain hydrodynamic lubrication. Figure 2 shows a Stribeck curve⁽⁴⁾, illustrating the state of lubrication and the friction characteristics of the sliding surfaces.

When the sliding speed is low, the surfaces are in a

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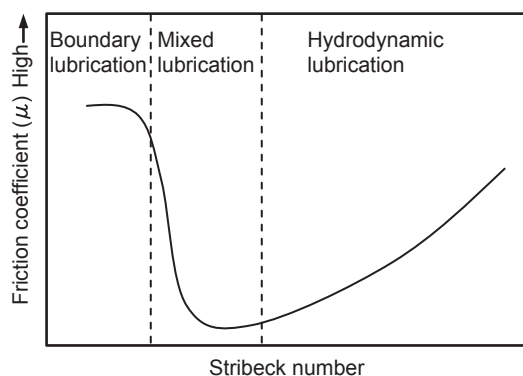


Fig. 2 Stribeck curve

state of boundary lubrication, in which they make contact and slide against each other. Therefore, the coefficient of friction is high. At high speeds the surfaces are in a state of hydrodynamic lubrication with almost no contact between them, and the coefficient of friction is low. At medium speeds, in the transfer zone between boundary and hydrodynamic lubrication, the surfaces are said to exhibit a mixed lubrication state.

When the conditions of sliding and state of lubrication of a piston skirt is represented as a Stribeck curve, at close to top dead center in the cylinder bore, the piston skirt surface is in a state of boundary or mixed lubrication. From the center to bottom dead center, it is in a state of hydrodynamic lubrication⁽⁵⁾.

Slight differences also occur in the state of lubrication of the piston in response to engine speed. In particular, boundary lubrication predominates at low engine speeds, and this surface state has an effect on sliding loss in the piston skirt⁽⁶⁾.

While the situation differs depending on conditions, the fact that the pistons display a state of boundary lubrication makes it challenging to reduce friction.

3. Material Design

The technology discussed here reduces sliding loss by forming a film on the piston skirt that ensures adsorption and retention of oil, thus extending the range of hydrodynamic lubrication to low-speed sliding, which had previously been characterized by boundary lubrication. In addition, the strength of the film was enhanced to increase abrasion resistance, which has conventionally been a challenging issue.

The following issues of manufacturability and practicality were considered in the design of the materials:

- (1) The material should not need special production facilities.
- (2) The material should not use environmentally harmful substances.
- (3) The material should be produced at low cost.

Dry film lubricants were focused on as materials satisfying the above requirements. The adhesive force of

the bonding resins used in dry film lubricants enables solid lubricants such as molybdenum disulfide, graphite, and PTFE to adhere to the sliding surfaces⁽⁷⁾. The coating is applied at the adequate thickness, after which it is dried or heat strengthened, which produces a lubricant film.

In general, when the strength of the film is increased its toughness declines, and the degree of adhesion is reduced. In other words, there is a trade-off between the strength of the film and its adhesive strength. In order to maximize each of these contradictory properties, the following three parameters were optimized in the development of the material:

- (1) Selection of added particles
- (2) Amount of added particles
- (3) Selection of resin and baking temperature (degree of crosslinking)

4. Tests of Materials

The dynamic friction abrasion testing machine shown in Fig. 3 was employed to simulate the reciprocal motion of the pistons in tests.

Surface treatments were applied to the disks used as a test piece. A pin was used to apply a vertical load to the disk, and the disk was subjected to right-left reciprocating motion under specific conditions of lubrication.

Performance measurements and durability tests in an actual engine were also conducted to evaluate the adhesion of the surface treatment and its performance in an actual engine.

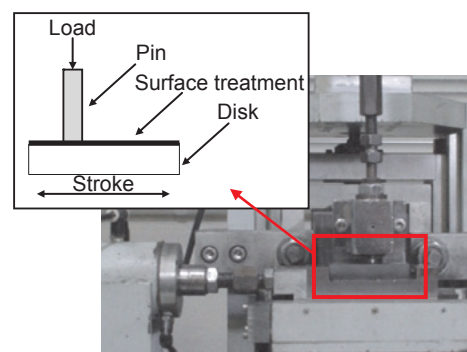


Fig. 3 Dynamic friction abrasion testing machine

5. Test Results

5.1. Oil Adsorption and Abrasion Resistance

Resin materials generally display low wettability and lipophilia⁽⁸⁾, and this was also true of the binding resin employed in this project. Because this research was attempting to use the oil-retaining properties of the film to extend the range of the state of hydrodynamic

lubrication of the piston skirt, it was necessary to increase lipophilia.

In order to increase lipophilia, the properties of the surface were modified by mixing microscopic hard particles in the binding resin that was used as the base of the surface treatment agent. The size of the used particles would affect the ratio of the surface area of the piston occupied by the particles and the binding resin, and would thus change the properties of the piston surface with respect to abrasion. Figure 4 (a) shows the degree of abrasion when the size of the grains is changed for a fixed amount of additive particles to the binding resin.

The graph shows that the degree of abrasion on the treated section of the surface decreases as the size of the grains diminishes. It can be considered that the increase in the specific surface area of the particles in the resin surface had an effect in modifying the abrasion properties of the piston surface, and thereby increasing oil lubrication performance. Larger additive particles were found to reduce the specific surface area of the piston surface. The particles were found to not have an effect in modifying the properties of the surface, but would instead increase the degree of abrasion by themselves causing abrasive wear.

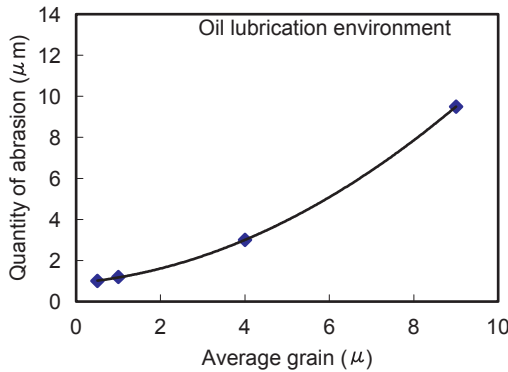
Figure 4 (b) shows the change in the degree of abrasion with respect to the quantity of added particles when the specific surface area of added particles in the resin surface is increased by dispersion of particles of 1 μm and below in size. Figure 4 (c) shows the oil adsorption and oil maintenance effects under the same conditions.

$$\text{Effect} = 1 - \frac{\delta A_{\text{oil}}}{\delta A_{\text{dry}}} \times 100(\%)$$

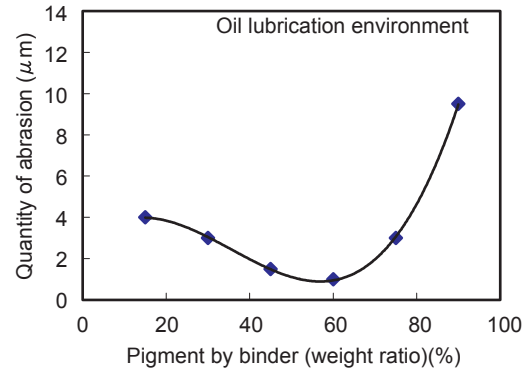
Here, the ratio between the degree of abrasion when oil is employed, δA_{oil} , and the degree of abrasion when no lubrication is employed, δA_{dry} , is used as an index of the oil adsorption and retention effect.

The graph indicates that the degree of abrasion can be controlled by increasing the amount of added particles to the material when oil is used. However, abrasion increases when the ratio of the particles exceeds 60%. This is because when the amount of additive particles exceeds the allowable limit, the binding resin cannot retain the particles and becomes brittle.

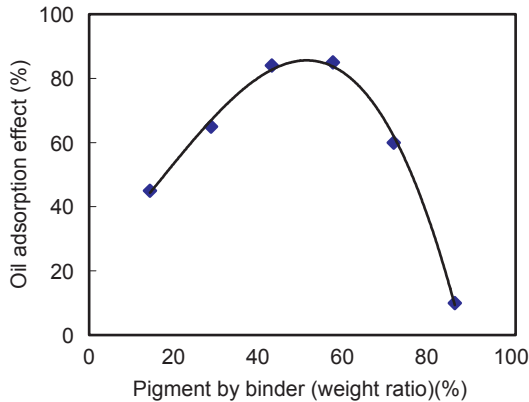
Figure 4 (d) shows a comparison on the oil adsorption and retention of the new coating with current mainstream coatings such as dry film lubrication, electro-less nickel



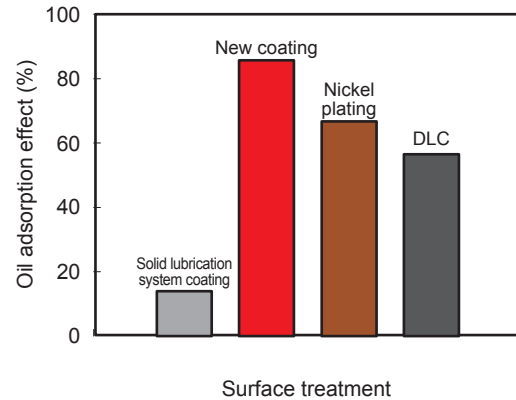
(a) Quantity of abrasion according to mixture particle size



(b) Quantity of abrasion according to particle mixture



(c) Oil maintenance effect according to mixture particle size



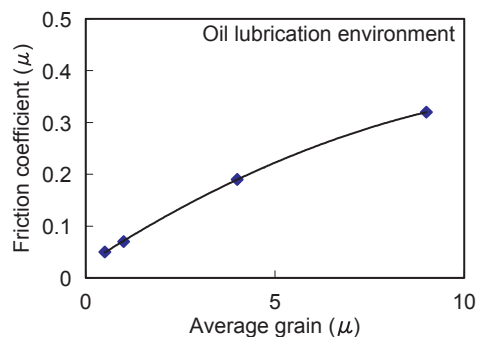
(d) Oil maintenance effect of each surface treatment

Fig. 4 Oil adsorption effect and anti-abrasivity

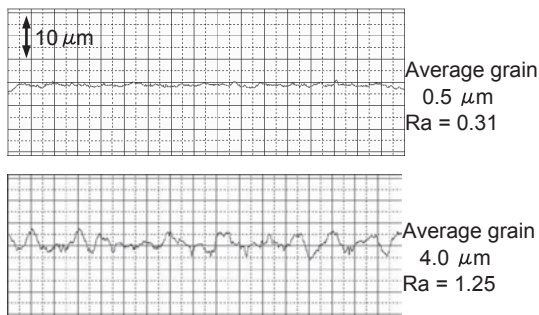
plating of anti-abrasion piston surface, and diamond-like carbon (DLC) which is a high strength film. The graph shows that the film developed in this project has the higher oil adsorption effect.

5.2. Oil Adsorption Effect and Low Sliding Friction Performance

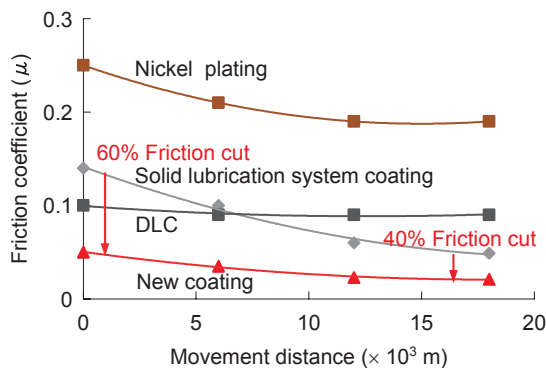
Modifying surface properties by means of the addition of microscopic hard particles not only affects abrasion resistance characteristics, but also affects sliding friction performance. The relationship between the average size of the added grains and the coefficient of kinetic friction when the amount of added particles to the binding resin is held constant is shown in Fig. 5 (a). The graph shows that sliding friction becomes lower as the average size of



(a) Relationship between friction and average grain



(b) Surface coarseness according to grain size



(c) Friction of each surface treatment according to movement distance

Fig. 5 Oil adsorption effect and friction

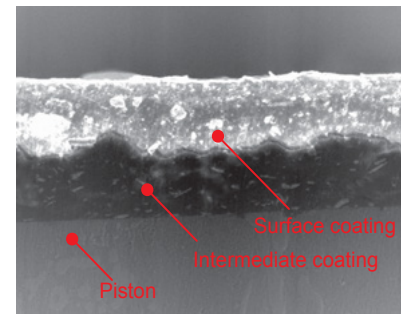
the grains decreases. Figure 5 (b) shows surface coarseness when two types of films with different average grain sizes are compared. The reduction of surface coarseness may be considered to reduce surface pressure by increasing the surface area of the sliding parts making actual contact, thus reducing the coefficient of kinetic friction.

Figure 5 (c) compares the new coating and other surface treatments in terms of the change in the sliding friction characteristic over the entire sliding distance. Due to the effect of adaptation to the contact faces, each treatment is saturated over a total sliding distance of approximately 10 000 m. The surface treatment developed in this project reduces sliding friction by 60% in the initial stage and 40% in a saturated state compared to a conventional dry film lubricant.

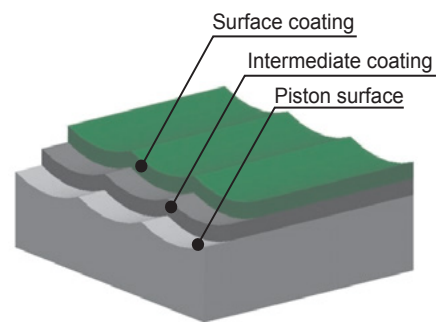
5.3. Surface Treatment Adhesion and Anti-abrasion Performance

Binding resins are chain polymers, and are baked to increase their strength by forming a bridged structure⁽⁹⁾.

In order to balance the contradictory characteristics of film strength and adhesiveness which are normally in a trade-off relationship, the project discussed here used an intermediate coating that is able to act as a buffer because of its toughness. The high-strength film adheres to the intermediate coating, and the intermediate coating adheres to the piston skirt. Figures 6 (a) shows an electron micrograph of a cross-section and Fig. 6 (b) shows an



(a) SEM image of piston surface



(b) Cut image of piston surface

Fig. 6 Image of piston surface

Table 1-(a) Engine specifications

Type	Air-cooled 4-stroke OHC single cylinder
Cylinder	Aluminum cylinder
Bore x stroke	64 x 50 mm
Displacement	160 cm ³
Compression ratio	8.5:1
Lubrication system	Forced splash

Table 1-(b) Test condition

Engine speed	3600 rpm
Throttle	Wide open throttle
Fuel	Regular (Density 0.7304 g/cm ³)
Oil	API SJ SAE 10W-30
Oil maintenance	Upper to lower

Table 2 Engine durability test result

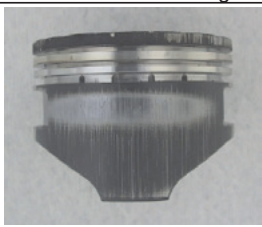
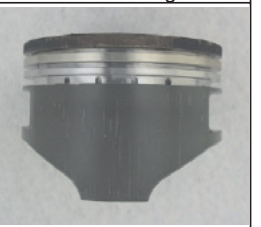
	Conventional coating	New coating	New coating
Piston surface (Thrust side)			
Operation time	20 h	20 h	150 h
Quantity of abrasion	Worn out (Coat 15 μm)	4 μm	6 μm

image of the piston surface.

Tests were conducted in an actual engine to evaluate the degree of adhesion. Table 1 (a) shows the specifications of the test engine and Table 1 (b) shows the test conditions.

This project assumed severe conditions for piston sliding motion, and used a general purpose engine with an aluminum sleeve in tests.

Table 2 shows closeup photographs of the pistons and indicates the degree of abrasion following the tests. In contrast to the pistons employing conventional dry film lubricants, the metal substrate of the piston skirts employing the film developed in this project was not visible following the tests for 20 hours, indicating enhanced anti-abrasion performance. This is considered to be a clear indication of the fact that the method developed in this project achieves lower friction by means of oil retention rather than self-lubrication as conventional dry film lubricants do. Tests were conducted for 150 hrs, and

results showed that the abrasion of the film was steady wear of 2 μm , indicating that the film maintained its performance over a long period.

In addition, the use of the intermediate coating enhanced the adhesion of the high-strength film, and no peeling of the surface coating was observed.

5.4. Results of Engine Performance Tests

Using the same test engine and replacing only the pistons, the power of the engine for each surface treatment was measured. Figure 7 shows the measurement results for power curves with a wide-open throttle.

The measurement data indicates the following effects at 3600 rpm under full load:

- (1) The use of the new surface treatment generates a 4.9% increase in power compared to that with no surface treatment.
- (2) The use of a conventional dry film lubricant generates a 2.4% increase in power compared to that with no surface treatment.
- (3) The use of the new surface treatment generates a 2.4% increase in power compared to that of a conventional dry film lubricant.

6. Conclusion

- (1) A high-strength film was developed by changing the characteristics of the piston surface with enhanced adsorption and retention of oil, enabling the achievement of a balance between low friction and anti-abrasion performance.
- (2) The results of abrasion and friction tests indicated the new surface treatment increased anti-abrasion performance by approximately five times and reduced friction by approximately 40% compared to those of a

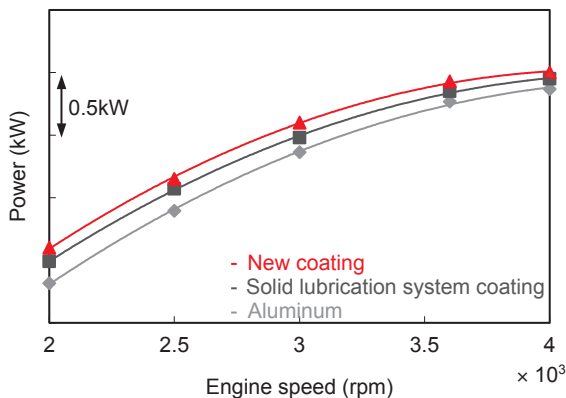


Fig. 7 Power curve

conventional dry film lubricant.

- (3) Power measurements conducted using an actual engine at wide-open throttle indicated that the use of the new surface treatment increases engine power by approximately 4.9%.

7. Afterword

New developments and modifications in relationship to the internal combustion engine are being conducted to respond to social and environmental requirements. The pistons are a key component of the internal combustion engine. Ensuring their continuing high performance and long life will have socioeconomic benefits and play an important role in responding to the depletion of fossil fuels and environmental issues. It is the hope of the author that the research discussed in this paper shall contribute to the achievement of these developmental goals.

Acknowledgments

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Kentaro SUGIMURA

Research on Optimization of Viscoelastic Properties of Tire Tread Rubber to Reduce Rolling Resistance and Decrease Braking Distance

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ABSTRACT

The achievement of reduced rolling resistance and lower braking distances is important in terms of automotive fuel efficiency and safety. These goals are strongly dependent on the performance of the tread rubber used in the tire, but are contradictory in terms of viscoelastic properties, an issue which needs to be addressed if these goals are to be satisfied simultaneously.

The research discussed in this paper involved analyses of the behavior of the contact patch of a vehicle's tires when the vehicle is cruising and braking. The results show that the two edge sections of a tire in the direction of width make the greatest contribution during braking, and the center section of a tire makes the greatest contribution during cruising. This discovery led to the proposal of a segmented tread configuration. Tires were developed on this basis, using different rubber compounds with optimized viscoelastic properties in the different segments. The distribution of contact pressure on the contact patch was also equalized. The newly developed tires displayed a 5-m reduction in wet braking distance when compared to other tires with identical rolling resistance characteristics.

1. Introduction

Ongoing efforts to increase fuel efficiency and enhance safety performance have highlighted the goals for tires to display reduced rolling resistance and decrease braking distance.

Figure 1 shows the relationship between the rolling resistance coefficient (RRC) of tires and their wet braking distance, as determined in tests based on the Japan New Car Assessment Program (JNCAP) of the National Agency for Automotive Safety and Victims' Aid (NASVA). As Figure 1 indicates, RRC and wet braking distance are contradictory parameters, and it is therefore difficult for both parameters to simultaneously enhance performance.

The research discussed in this paper focused on the behavior of the contact patch of a vehicle's tires while cruising and braking. The placement of the tread rubber

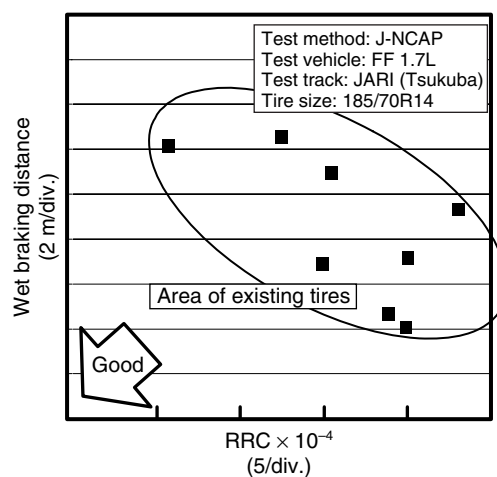


Fig. 1 Relationship between wet braking distance and RRC

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was optimized and the contact pressure on the contact patch was equalized to develop a tire that displayed a low RRC and reduced braking distance.

2. Viscoelastic Properties of Tire Tread Rubber

2.1. Contradictory Performance Goals and Relationship with Viscoelasticity

The RRC of tires has an effect on fuel efficiency: a 10% reduction in the RRC is known to result in a 1-2% increase in fuel efficiency⁽¹⁾. The RRC is generated by the loss of energy caused by repeated deformation of the rubber and high-molecular fibers that make up the greater part of a tire when a tire is turning. Rubber is a viscoelastic material, having both elastic and viscous properties. Because it has viscous properties, it loses some energy as heat when it is deformed, and cannot recover all the energy it had prior to deformation. Hysteresis loss therefore occurs. Hysteresis loss is proportional to the $\tan\delta$ of the material (loss tangent = loss elastic modulus/storage elastic modulus), and is therefore generally expressed using $\tan\delta$ ⁽²⁾. Figure 2 shows a simple schematic of hysteresis loss.

The contribution of the tire tread rubber to the RRC is approximately 50%⁽³⁾. The RRC can be reduced by reducing the $\tan\delta$ of the tread rubber.

However, the tread rubber is the only part of a tire that makes contact with the road surface, and therefore wet braking performance must be considered in addition to the RRC. Braking force is determined by the friction force of the tread rubber, which is expressed as the sum of the friction force due to hysteresis loss and the adhesion friction force. It has been demonstrated that the temperature-dependence of the braking distance of an actual vehicle can be accurately predicted from the viscoelastic properties of the tire tread rubber⁽⁴⁾. On a wet road, the water film reduces the area of contact between the road surface and the tread rubber, which reduces

adhesion friction force and thus increases the contribution of the friction force that is generated due to hysteresis loss. This means that increasing the $\tan\delta$ of the tread rubber would be an effective means of enhancing wet braking performance.

The preceding discussion indicates that different $\tan\delta$ characteristics are required for reduction of the RRC and enhancement of the braking performance. Conventionally, however, $\tan\delta$ control method has focused on the difference in the input frequency to the rubber when the vehicle is being driven and when it is braking, based on the relationship between tire behavior and the deformation frequency-dependence of $\tan\delta$ shown in Fig. 3.

The RRC of a tire is generated by the deformation of the tire by the road surface that occurs once every revolution of the tire. By contrast, braking force is generated by minute deformation of the tread rubber against the tiny (0.1-mm pitch) concavities and convexities of the road surface aggregates. If these deformations are considered in terms of frequency, the RRC corresponds to a deformation input frequency of 10^1 - 10^2 Hz, and braking force corresponds to a deformation input frequency of 10^4 - 10^6 Hz. Frequencies of higher than 1000 Hz are difficult to reproduce in the laboratory. Given this, time-temperature conversion using the WLF equation, which is often used in studies of viscoelastic substances, can be applied to convert these frequencies into temperatures. When this is done for measurement of $\tan\delta$ at around 10 Hz, the RRC corresponds to 50-60 °C, and the braking force during wet braking corresponds to a figure of close to 0 °C⁽⁵⁾⁻⁽⁷⁾. Therefore, a rubber with a low $\tan\delta$ at around 60 °C and a high $\tan\delta$ at around 0 °C would enable simultaneous reduction of the RRC and braking distance. Table 1 summarizes the requirements for a tire tread rubber to fulfill these two performance goals, as viewed from the perspective of hysteresis loss.

However, it would be difficult to achieve a simultaneous increase in $\tan\delta$ at 60 °C and 0 °C using conventional methods of controlling $\tan\delta$, which have aimed to decrease $\tan\delta$ at 60 °C and increase it at 0 °C.

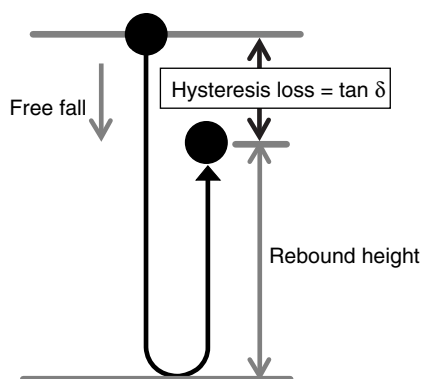


Fig. 2 Image of hysteresis loss of viscoelastic compound

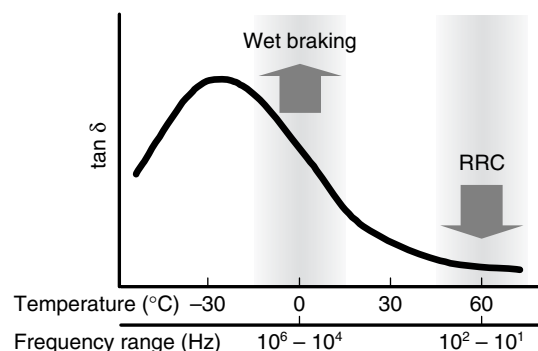
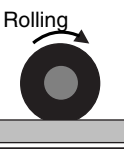
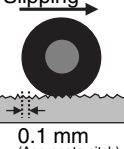


Fig. 3 Temperature dependence and frequency dependence of $\tan\delta$

Table 1 Comparison of direction of enhanced tread compound

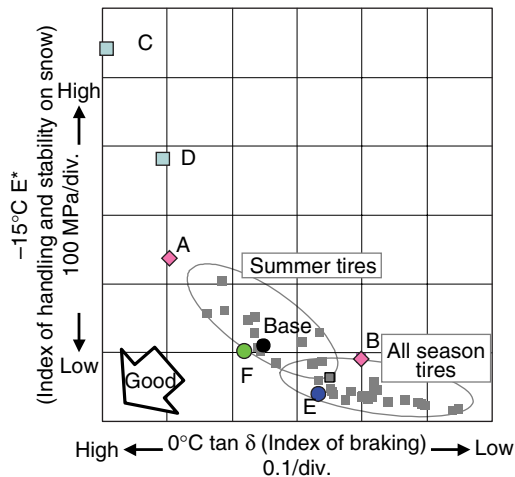
	Phenomenon	Frequency	Index Direction
RRC	Rolling 	$10^1 - 10^2 \text{ Hz}$	$60^\circ\text{C } \tan \delta$ Small
Wet braking	Slipping  0.1 mm (Aggregate pitch)	$10^4 - 10^6 \text{ Hz}$	$0^\circ\text{C } \tan \delta$ Large

Moreover, in addition to the RRC and the braking performance of the tires, their performance on snow, which is affected by the properties of the tread rubber, must also be considered. Figure 4 shows the relationship between the low-temperature (-15°C) elastic modulus, E^* , and $\tan \delta$ at 0°C for various rubbers. As Figure 4 shows, the range of distribution of $\tan \delta$ at 0°C differs for all season tires (M+S tires), which are required to perform well at low temperatures, and standard summer tires. In other words, the viscoelastic properties of all season tires and standard summer tires have been controlled to fall within different ranges of low-temperature E^* .

2.2. Study of the Relationship between Changes in Viscoelastic Properties and Tire Performance

Using the tread rubber of mass-production tires as a base, a variety of different test rubbers were formulated to clarify the relationship between the viscoelastic properties of the tread rubber and tire performance, as follows:

- A: $\tan \delta$ at 0°C was increased
- B: $\tan \delta$ at 60°C was reduced

Fig. 4 Relationship between $0^\circ\text{C } \tan \delta$ and $-15^\circ\text{C } E^*$

- C: $\tan \delta$ at 0°C was increased (an increase in E^* at -15°C was allowed)
- D: $\tan \delta$ at 60°C was reduced (an increase in E^* at -15°C was allowed)
- E: $\tan \delta$ at -15°C was maintained within the allowable range for all season tires, $\tan \delta$ at 0°C was increased, and $\tan \delta$ at 60°C was decreased
- F: E^* at -15°C was maintained within the allowable range for summer tires, $\tan \delta$ at 0°C was increased and $\tan \delta$ at 60°C was reduced

The following reasoning was used with regard to the chemical composition required to realize these properties. The type of polymer and the type and quantity of inorganic filler (carbon black or Silica) used has an effect on the viscoelastic properties of rubber. To reduce $\tan \delta$ at 60°C , it would be necessary to increase the dispersability and reactivity of the filler. A peripherally functionalized solution polymerized styrene butadiene rubber was therefore selected as the main polymer, and its reinforcement efficiency was increased. $\tan \delta$ at 0°C was increased by using a high filler content.

E^* at -15°C was adjusted by optimizing the amount of butadiene rubber and the type of added oil.

Figures 4 and 5 show the relative positioning of the tread rubber of mass-production tires and rubbers A-F in terms of viscoelastic properties.

Figure 6 shows the results of evaluation of the

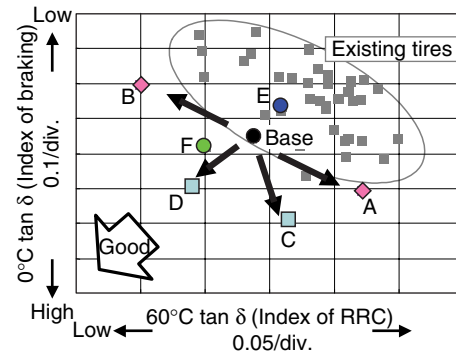
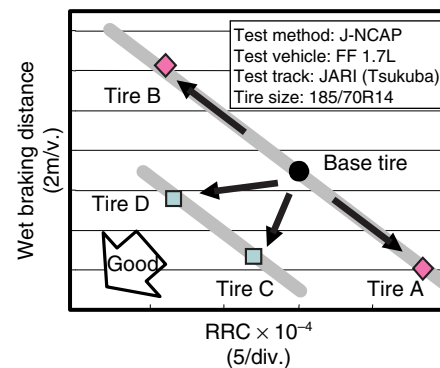
Fig. 5 Relationship between $60^\circ\text{C } \tan \delta$ and $0^\circ\text{C } \tan \delta$ 

Fig. 6 Relationship between wet braking distance and RRC

performance of Tires A-D, formed using rubbers A-D, against the performance of a base mass-production tire. The performance of tires A and B is situated on a line determined by the incompatibility between achieving enhancements in both the RRC and braking performance. By contrast, the performance of tires C and D, in which snow performance was not considered and E^* at -15°C was allowed to increase, shows enhancements in both the RRC and braking performance. This demonstrated that there was a limit to the enhancement in the RRC and braking performance that could be obtained by controlling viscoelastic properties if performance on snow was maintained. A new approach that could be used in combination with control of viscoelastic properties was therefore required.

The project therefore began to focus on the exchange of forces between the tire contact patch and the road surface, and conducted analyses of the contact patch behavior of tires during braking, cruising, and driving on snow.

3. Analysis of Contact Patch Behavior for Each Performance Parameter

The contact patch is the part of the tire that makes contact with the road surface. The form and pressure distribution of the contact patch of any individual tire will vary depending on the air pressure and the load on the tire. In addition, the state of the contact patch varies depending on whether the vehicle is being driven, braking, or being driven on snow. The team conducting the research conjectured that different areas of the contact patch contributed to different aspects of performance. Conventionally, the tread section has been formed from a single type of rubber, but optimizing the placement of different types of rubber in different parts of the tread section might generate performance enhancements. Tire behavior was analyzed on this basis.

In this paper, the length of the contact patch in the direction of the circumference is termed the contact length, and the length of the contact patch in the direction of width is termed the contact width. The central section in the direction of width is termed the center, and the two edge sections are termed the shoulders. Figure 7 shows these sections.

3.1. Analysis of Contact Patch Behavior during Braking

The JNCAP assessment includes full braking at 100 km/h. Figure 8 shows the changes in load on the front tire of a 1.7L FF vehicle in a wet braking test based on the JNCAP test. The load on the front wheel goes from a steady state to maximum load in 0.2 seconds, i.e. the change in the load on the tire occurs in an extremely short time period.

The test set-up shown in Fig. 9 was used to analyze the behavior of the contact patch under these conditions.

An electromagnetic shaker was used to impart a 5 Hz dynamic vibration to the tire, and a sensor (75 mm x 75 mm; resolution: 2 mm) with a responsiveness of 120 Hz was used to measure the pressure distribution on the contact patch. Measurements were also taken when a static

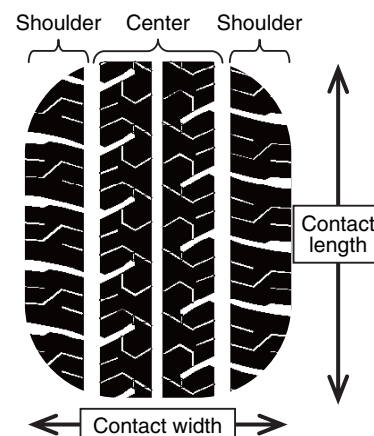


Fig. 7 Definition of footprint (Contact patch)

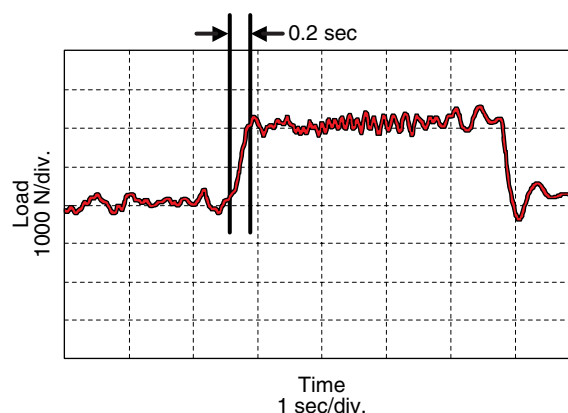


Fig. 8 Front wheel load variation in wet braking

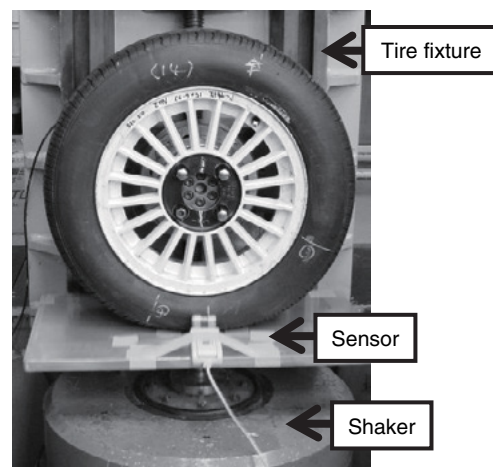


Fig. 9 Dynamic contact patch measuring equipment

braking load was applied (i.e. when the braking load causes saturation of the contact pressure), for comparison. The area in which measurements were taken is shown in Fig. 10, and the results of the measurements are shown in Fig. 11.

A comparison of the results of measurements of contact patch pressure under a normal load and maximum braking load in the dynamic braking tests (Fig. 11) shows no change in the contact pressure in the center, but a significant change in the shoulders. This indicates that the shoulders are the sections of the contact patch that are subject to the highest pressure load during braking. A

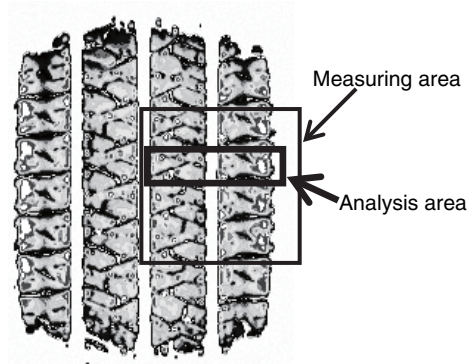


Fig. 10 Measuring and analysis areas of contact patch

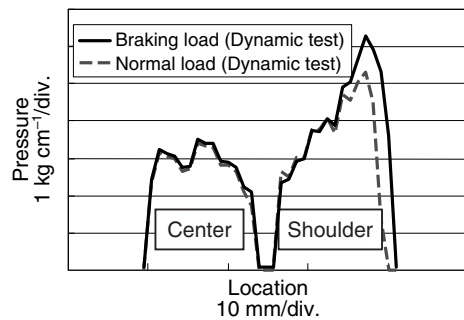


Fig. 11 Comparison of contact patch pressure distribution in normal load and braking load

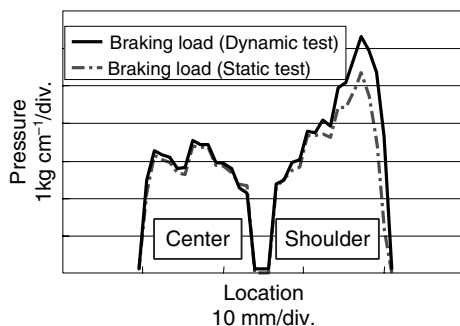


Fig. 12 Comparison of contact patch pressure distribution in dynamic and static tests

comparison of the contact pressure results from static and dynamic tests (Fig. 12) indicates that the contact pressure on the shoulders is higher when the load is applied dynamically.

In general, the lower the contact pressure of rubber the higher its friction coefficient (μ), therefore, it was conjectured that preventing the phenomenon of imbalance in the contact pressure of a tire could result in a reduced braking distance. Using the same rubber, two prototypes of tires were produced in which the high contact pressure on the shoulder section and the uniform contact pressure, as shown in Fig. 13. According to the comparison of these tires in the actual vehicle test regarding a wet braking distance, the braking distance of the tire with the uniform contact pressure was shorter than the tire with the high contact pressure on the shoulder section by 3 m or more. It was confirmed that equalization of the contact pressure enhanced the reduction of the braking distance.

3.2. Analysis of Contact Patch Behavior when Vehicle is Cruising

An FEM simulation input, with the properties of the various sections of the tire and conducted under conditions of usual driving, was done to analyze the behavior of the contact patch when the vehicle is cruising. The results, as shown in Fig. 14, indicated that energy loss was greatest in the rubber at the center of the tire, and that the rubber at the center therefore makes the greatest contribution in terms of the RRC.

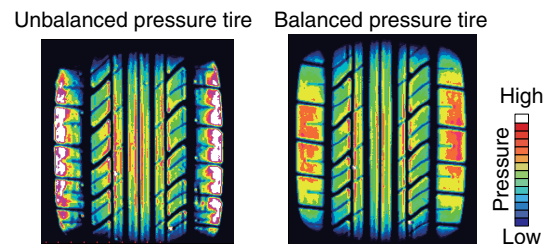


Fig. 13 Comparison of contact patch pressure under braking load

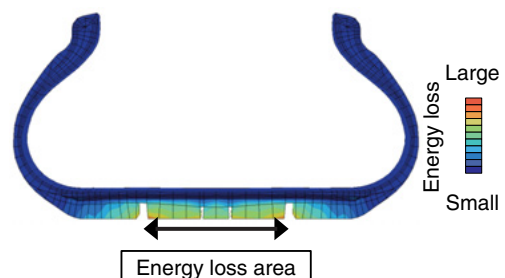


Fig. 14 Hysteresis loss distribution by finite element method analysis

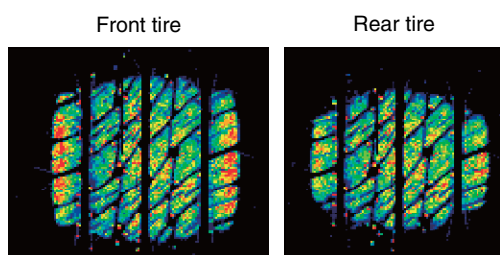


Fig. 15 Comparison of contact pressure

3.3. Analysis of Contact Patch Behavior when the Vehicle is Driven on Snow

On snow, compression resistance, shearing force of snow in void, adhesion friction force, and shearing friction force (edge effect) are the main mechanisms by which friction force is generated⁽⁸⁾. The design of the tread pattern and the elastic modulus of the rubber at low temperatures therefore determine the performance of the tire on snow. The contact state of the rear tires, i.e. the contact patch behavior under low loads, is important in ensuring the stability of a FF vehicle on snow. Figure 15 compares the contact patches of front and rear tires on snow for a FF 2.0L vehicle. The ratio of the surface area of the tire occupied by the rubber at the center is greater in the rear tire, and it is therefore important for handling on snow to help ensure that the rubber at the center is flexible at low temperatures.

4. Technology to Enhance Tire Performance and Results Achieved

Table 2 shows performance goals, the section of the tire that contributes to each performance goal, and the rubber property required for each section, as determined in the analyses of contact behavior. To realize these properties in a tire, a segmented tread structure was developed, as shown in Fig. 16. In this configuration, a rubber compound that is effective in reducing the RRC and helping ensure effective handling in snow is used in the center, and a rubber compound that is effective in terms of wet braking performance is used for the shoulders. The pressure distribution over the entire contact patch was also equalized to ensure that these properties of the rubber compounds could be displayed effectively.

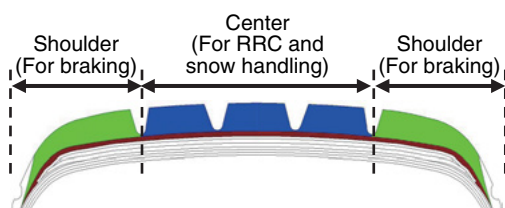


Fig. 16 Segmentation of tread construction

Table 2 Target tire characteristics

Performance goal	Part of contact patch	Index of tread compound property	Direction
Braking	Shoulder	0°C tan δ	Up
RRC	Center	60°C tan δ	Down
Snow handling	Center	-15 °C E*	Down

4.1. Rubber Compounds Used

In order to satisfy the requirements shown in Table 2, rubber E, as described in Section 2.2., was used for the center, and rubber F was used for the shoulders.

4.2. Equalization of Distribution of Contact Pressure

The following method was used to equalize the distribution of contact pressure on the tire. Figure 17 shows a cross-section of a tire. The distribution of contact pressure can be equalized by optimizing the crown shape⁽⁹⁾. The target function was therefore established as a minimum deviation in contact pressure and the design parameter was set as the crown R shape, and FEM analysis was used to determine the optimum cross-sectional shape. The results are reflected in the shape of the vulcanizing die.

4.3. Verification of Performance Enhancements

Tires were produced with optimized crown R shapes and optimized positioning of the developed rubbers in the three-segment structure, and were subjected to performance tests. The new tires displayed a uniform contact patch pressure at maximum braking load, as shown in Fig. 18. The relationship between the RRC and wet braking distance (JNCAP test) for the new tires is shown in Fig. 19, together with the relationship for conventional tires using a single rubber compound. The newly developed tires displayed a 5 m reduction in braking distance as compared to tires with an identical RRC, and an 18 reduction in the RRC as compared to tires with identical braking distances.

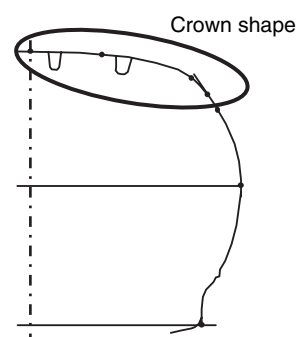


Fig. 17 Cross section of tire forming die

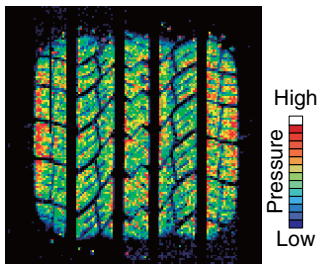


Fig. 18 Contact patch pressure of developed tire at braking load

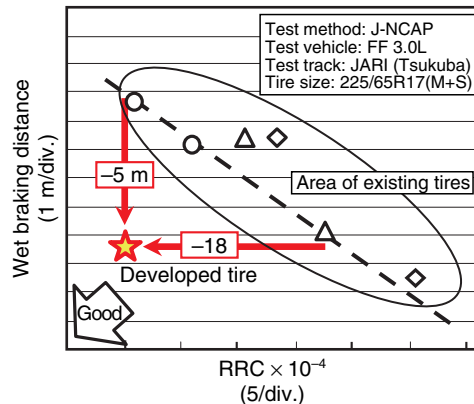


Fig. 19 Relationship between wet braking distance and RRC

These results indicate that this research has been successful in achieving a simultaneous enhancement in two antagonistic parameters, the RRC and braking distance.

The handling of the tires on snow (straight line vehicle stability, turning tracking performance, braking performance, and traction performance) was also verified. Further tests showed that the new tire technology had no adverse effect on the basic performance of the tires (durability, wear resistance, etc.) and their commercial appeal (handling, comfort of ride, road noise, etc.).

5. Conclusion

The following are the salient points of the research discussed in this paper.

- (1) The viscoelastic properties of tread rubber compounds place restrictions when trying to ensure adequate handling on snow, and there are difficulties to simultaneously achieve enhancement of the RRC and the braking distance of a tire by controlling the viscoelasticity of the rubber.
- (2) Analyses of the behavior of tires during braking indicated that the shoulder sections of the contact patch made the greatest contribution to braking performance. In addition, it was shown that equalizing the distribution of contact

pressure on the contact patch would be effective in helping ensure adequate friction.

- (3) An FEM simulation conducted under conditions of usual driving showed that the center of the contact patch made the greatest contribution to rolling resistance.
- (4) The center section of the rear tire represents the greater part of the contact patch. Therefore, the low-temperature elastic modulus of the rubber used in the center section makes a significant contribution to helping ensure the snow handling of FF vehicles.
- (5) Tires were constructed by placing rubber displaying properties that were effective in reducing rolling resistance and braking distance and maintaining snow handling in the optimal positions in a segmented structure. Also, using a cross-sectional shape that helped provide uniform distribution of contact pressure on the contact patch. These tires displayed a reduction of 5 m in braking distance on wet roads when compared to tires with identical rolling resistance characteristics, and an 18 reduction in the RRC when compared to tires with identical braking distances.

6. Afterword

Tires were developed for a 2.0 L SUV using the methods discussed in this paper. In line with the development goals set for the vehicle, reduction of the RRC was a main focus of the development. The developed vehicle displayed an enhancement of approximately 1.5% in 10-15 mode fuel economy with no decline in braking performance.

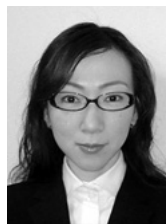
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Development of Microscopic Damage Model for Impact Failure Simulation of Polymeric Components

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ABSTRACT

For simulating the final failure of the component made by polymeric materials based on the microscopic material properties, the craze density-based softening law was proposed phenomenologically. The microscopic experiment was conducted while the craze nucleation, growth and fibril failure were simulated with the coarse grained molecular dynamics model. The impact failure test was also conducted on the actual automotive part. The deceleration time history and damage zone development were compared with those obtained by the finite element (FE) analysis. As a result, the usefulness of the constitutive equations with microscopic material properties was validated.

1. Introduction

Continuing efforts by automotive manufacturers to increase environmental and safety performance have generated increasingly diverse and sophisticated performance requirements for automotive components. To satisfy performance requirements that are expected to diversify even more in the future, further increases in product performance, crash performance and cost performance are being sought. This makes the study of the chemical composition of materials a greater part of the development process for automotive components.

Important components in terms of crash with pedestrians and the collision of passengers with the interior of the cabin are the instrument panel, the door lining, and the bumper face. These components are manufactured from plastics such as polypropylene, the crash performance of which is known to be determined to a large degree by the chemical composition of the material⁽¹⁾.

To analyze the large deformation impact characteristics of these plastic components, it is essential to determine material constitutive laws dealing with the microscopic damage (craze) resulting from micro-voids and fibril failure that can be observed in glassy plastics. A great deal of research has already been conducted on the craze using micro-scale simulations⁽²⁾⁻⁽⁶⁾. These studies have clarified the fact that the initiation of the

craze growth is affected by the length of high molecular chains⁽²⁾⁻⁽³⁾, and the fact that fracture energy on the macro-level in glassy plastics is determined by the elastic property and the fracture stress of the craze, which represent physical quantities on the micro-level⁽⁴⁾. The mechanism of the craze is also being clarified using micro-scale simulations, for example simulations that express the mechanism of nucleation of the craze by means of molecular chain entanglements⁽⁵⁾⁻⁽⁶⁾.

Considerable research has also been conducted using macro-scale simulations in attempt to predict the behavior of glassy plastics after yielding⁽⁷⁾⁻⁽⁹⁾. However, while the constitutive laws developed by this research enabled the prediction of the nucleation of the craze, they could not predict the growth of the craze and the formation of shear bands. Research in this area was conducted by Shizawa, et al., resulting in the proposal of material constitutive laws that enabled prediction of both phenomena⁽¹⁰⁾⁻⁽¹³⁾. These constitutive laws (the Shizawa, et al. model) comprise a law of strain rate hardening, a law of the craze nucleation, and a law of the craze growth. Their material constitutive model demonstrated excellent correlation with the results of static tensile tests on polypropylene and polycarbonate in micro-scale simulations, but the validation for predicting the deformation and failure at the medium and high strain rates, which are generated in automotive crash has not been conducted. In addition, to conduct analyses from the chemical composition level to

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the component level, it is necessary to formulate material constitutive models and analytical method that link the micro- and macro-scales.

The present paper therefore proposes a craze density-based softening law as a phenomenological model to enable simulations of large impact deformation processes at the component level that include consideration of micro-scale material parameters. To this end, the softening law is proposed phenomenologically based on the modeling of final failure due to the nucleation and growth of the craze and fibril breaks, using the results of tests at the micro-scales and coarse grained molecular dynamics. The above-mentioned Shizawa, et al. model is modified by means of the incorporation of this softening law, and material constitutive laws that can now be used in analyses at the component level are proposed. Finally, the results of tests and analyses at the component level are compared and then the analytical method is validated.

2. Test Method

2.1. Test Specimen and Test Material

The material used for this research was thermoplastic semicrystalline polypropylene often used for the interior and exterior parts of automobiles. When this material is to be used for parts which are to be loaded by impact, elastomer is added to toughen the thermoplastic. Therefore, the materials are ductile up to the high strain rate^{(14), (15)}. Polypropylene blended with elastomer was chosen as the test specimen for this research to evaluate the failure behavior of impact parts. Figure 1 shows a schematic of the test specimen. This was a compact tension test specimen, using the specifications stipulated by ASTM D5045-93. The specimen was cut from a bumper face. A 0.2-mm pre-crack was made at the crack tip using a sharp cutter knife prior to the tests.

2.2. Test Equipment and Methods

2.2.1. Tensile test inside environmental SEM

Tensile tests were conducted inside an environmental SEM to enable measurement of the large deformation and

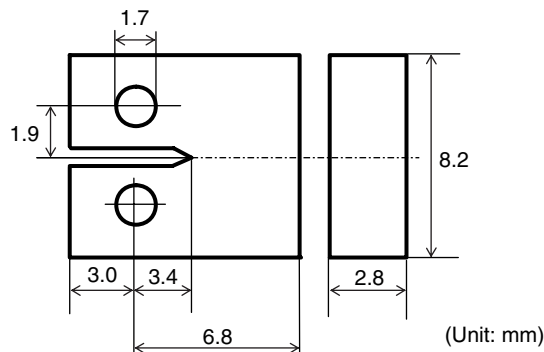


Fig. 1 Schematic illustration of compact tension test specimen

failure behavior of the craze at the micro level. A Deben M5000 micro tensile stage was placed in an environmental SEM (Quanta 600F manufactured by FEI Co.) and the compact tension test specimen was applied by tensile loading. Figure 2 shows a photograph of the specimen and the tensile stage in the ESEM. Tensile speeds were 0.5 and 4.0 mm/min, and the ambient temperature was 23 °C. As tensile force was applied, the deformed areas in which the craze occurred were recorded on video, at the same time as the load applied by the clamping devices was measured. After the tests, the fibril length was measured in the areas in which the craze had occurred using the video images until the macroscopic crack started to propagate. (Fig. 3).

2.2.2. Bumper face impact tests

Bumper face impact tests were conducted using the pendulum test equipment as tests at the automobile component level. Figure 4 shows the test set-up. A

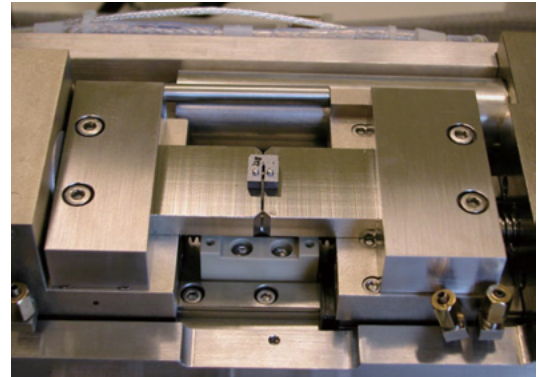


Fig. 2 Photograph of tensile stage and specimen in ESEM

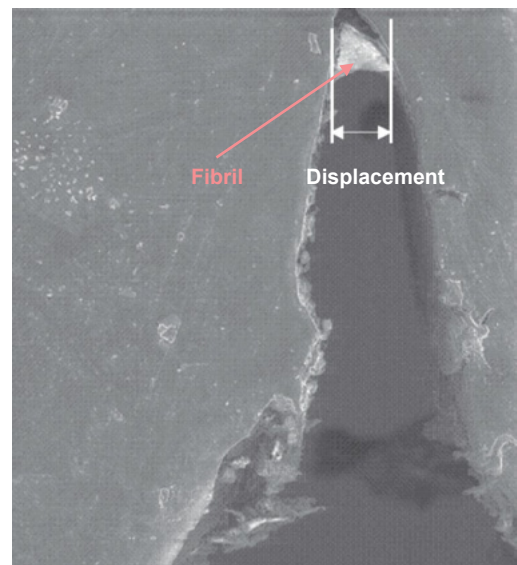


Fig. 3 Photograph of crack tip during tensile test

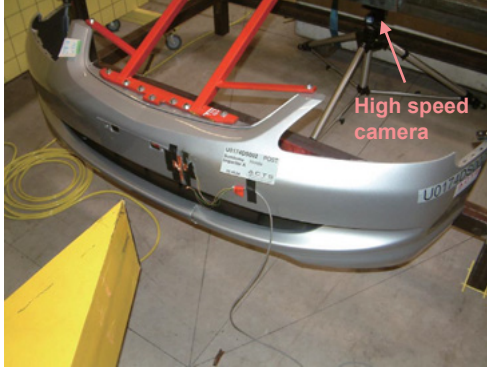


Fig. 4 Photograph of bumper face impact test

bumper face was fixed in an iron jig, and impacted by a wedge-shaped impactor at a velocity of 2.8 m/sec at an offset of approximately 40% from the center of the bumper face. The deceleration of the impactor was measured, and a high-speed camera positioned on the backside of the bumper face was used to record the progress of the craze damage development and fracture.

3. Coarse Grained Molecular Dynamics Model

This section will discuss the coarse molecular dynamics model used in the micro-scale modeling of the craze.

3.1. Potential Function

A coarse grained molecular dynamics model was used for micro-scale modeling of the nucleation and growth of the craze. OCTA⁽¹⁷⁾, an integrated simulator available in the public domain, was used as the analytic solver. A bead/spring model was used in this analysis. The Lennard-Jones (LJ) potential expressed by Eq. (1) was used as the van der Waals force between the beads.

$$U_{nonbond}(r_{ij}) = 4\epsilon_{ij} \left[\left(\frac{\sigma_0}{r_{ij}} \right)^{12} - \left(\frac{\sigma_0}{r_{ij}} \right)^6 \right] \quad (1)$$

Here, r_{ij} is the diameter of the LJ beads and ϵ_{ij} is the strength of the mutually acting force. The simple harmonic potential function expressed in Eq. (2) is used as the bonding force between the beads.

$$U_{bond}(r) = \frac{1}{2} k (r - r_0)^2 \quad (2)$$

Here, k is the spring constant, and r_0 is the equilibrium bond length of the spring in the initial balance state.

3.2. Polypropylene Bulk Model

Figure 5 shows the coarse molecular dynamics simulation model. The model has the shape of an oblong box with dimensions of 14.1 μm x 14.1 μm x 50.0 μm .

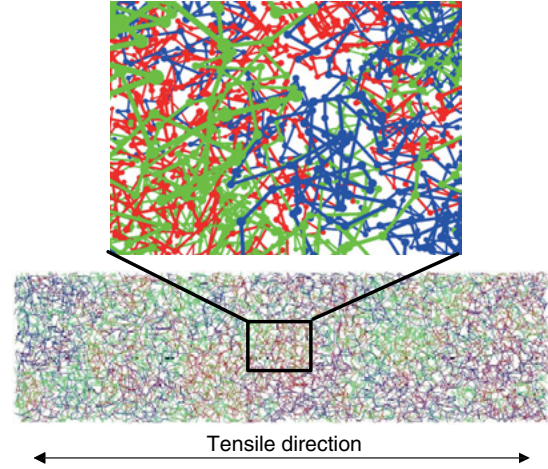


Fig. 5 Coarse molecular dynamics model

There are 10 000 beads in total and each bead contains a lamella structure of approximately 77 layers. Spring colors indicate different molecular chains respectively. After relaxation of the structure, the bond length is 1 μm , and a linear chain structure is assumed as a molecular chain shape. In this model, one chain therefore corresponds to a bundle of fibrils. Staggered reflection was used as the boundary condition in the tensile direction, and cyclic reflection was used in other directions. Potential was automatically scaled⁽¹⁶⁾ in the initial structural relaxation calculations; there were 50000 calculation steps. The mutually acting strength of the non-bond potential function, ϵ_{ij} , and the bond potential spring constant, k , were established to ensure that the relationship between tensile load and displacement in the environmental SEM tests correlated well with the results of the simulation.

4. Results and Consideration

4.1. Result of Tensile Test inside Environmental SEM

Figure 6 shows the relationship between load and fibril displacement as determined by the compact tension tests. This graph indicates that the maximum load

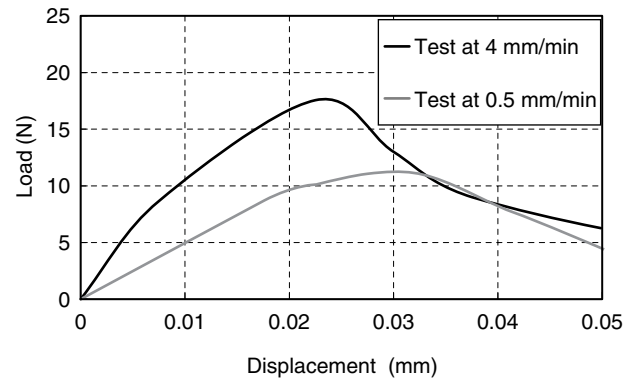


Fig. 6 Load displacement histories of crack tip

increases as the tensile speed increases. Figure 7 shows a photograph of nucleation of the craze at the crack tip during tensile loading. The fibril orients along the direction of tensile force, and then ruptures.

4.2. Results of Coarse Molecular Dynamics Simulation

By analyzing parameters, it was previously demonstrated that the spring constant of the bond potential function shows a high sensitivity toward the gradient of the load-displacement relationship, and both the mutually acting force in the non-bond potential function and the bond potential function showed high sensitivity towards the maximum value of the load-displacement relationship⁽¹⁸⁾. These results were used to establish the parameters of the potential functions in the present simulation. The simulation results are shown in Fig. 8, and the simulated deformation at final failure is shown in Fig. 9. Both test results and simulation results demonstrated that after reaching the maximum load, the

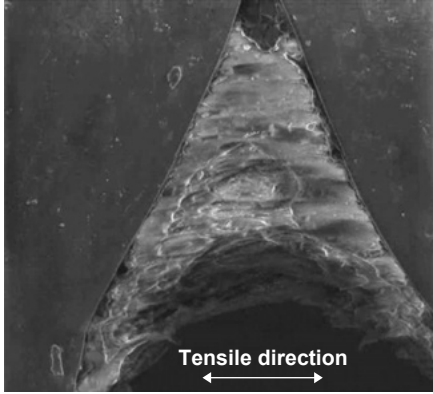


Fig. 7 Photograph of crack tip during tensile loading

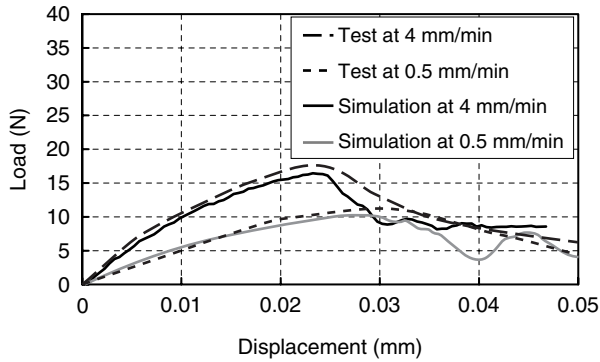


Fig. 8 Load displacement histories of bulk model



Fig. 9 Simulated deformation of crack tip in coarse molecular dynamics model

fibrils became oriented and the molecular chains loosened and fractured, making the load decrease. It is suggested that the incorporation of this softening behavior in FE models as a softening law would enable modeling the micro-parameters based modeling.

4.3. Craze Density-based Softening Law and Constitutive Law of Elasto-viscoplastic Materials

The coarse molecular dynamics simulations and tensile test inside environmental SEM discussed above demonstrated that the fracture mechanism in the micro scale is determined by continuous failure of the fibrils in the craze. A softening law dependent on the craze density is next proposed to enable FE simulation of this phenomenon at the macro level. Figure 10 shows a schematic diagram of this softening law. The craze nucleation and growth are divided into Zone A and Zone B. Zone A is an area of microscopic damage, and Zone B is an area of macroscopic damage. In Zone A, the softening law is not operative, and the Shizawa, et al. model is used. When the craze grows and reaches Zone B, the softening law becomes operative, and simulations are conducted using a modified flow stress. The strong softening or weak softening can be simulated by changing the degree of softening effect.

To present the concept noted above, the following discussion offers the constitutive law proposed by Shizawa, et al. which considers the nucleation and growth of the craze, in addition to the modified flow stress equation and softening law developed in this research.

$$\overset{\nabla}{\mathbf{T}} = \mathbf{C}^v : \mathbf{D} - \dot{\bar{\epsilon}}^p (\cos \delta) \mathbf{P}' - \dot{\omega} \mathbf{T} / (1 - \omega) \quad (3)$$

Here, $\mathbf{T}$ is Cauchy stress, $\overset{\nabla}{(\)}$ is Jaumann velocity, $\mathbf{D}$ is the deformation rate, $\dot{\bar{\epsilon}}^p$ is the corresponding plastic strain rate, ω is the craze density, and $\dot{\omega}$ is the craze growth rate. $\mathbf{C}^v$, $\mathbf{P}'$ and $\cos \delta$ are defined in the reference literature⁽¹⁰⁾⁻⁽¹³⁾, and therefore not noted here. The craze

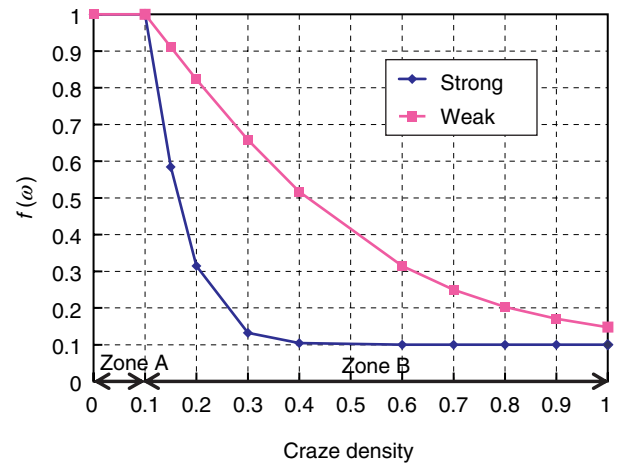


Fig. 10 Schematic explanation of softening factor and craze density

growth rate is expressed by the following equation:

$$\dot{\omega} = A(1-\omega)\langle \dot{\varepsilon}_m^p \rangle + B \cdot D_1 \cdot \dot{\varepsilon}_m^p \cdot 0.5[1 + \tanh\{-D_2(\bar{\varepsilon}^p - \varepsilon_c)\}] \cdot \dot{\varepsilon}^p \quad (4)$$

Here, A , B , D_1 , and D_2 are material parameters. $\dot{\varepsilon}_m^p$ is the mean plastic perpendicular strain rate, and is expressed by the following equation:

$$\dot{\varepsilon}_m^p = d\{(q_1\omega)\cosh((q_2\omega + q_3)\sigma_m/\sigma_y)\}/dt \quad (5)$$

where σ_m is hydrostatic pressure, σ_y is yield stress, and q_1 – q_3 are material parameters. The first term in Eq. (4) represents the creation of craze, and the second term represents nucleation and growth. ε_c is the strain at which the growth of the craze ceases. The craze is judged to have occurred on the basis of the following equation, which uses hydrostatic pressure criteria:

$$\sigma_b \geq A_1 + (B_1/3\sigma_m) \quad \sigma_b = \sigma_1 - \nu\sigma_2 - \nu\sigma_3 \quad (6)$$

Here, σ_b is the stress necessary to orient the fibrils, A_1 and B_1 are material parameters, and σ_i is the principal stresses. As shown in Eqs. (7) and (8), a strain hardening law is the general power law with the strain rate dependence coefficient m , and the flow stress, g , can express the increase of stress following secondary yielding, as observed in tensile load tests.

$$\dot{\varepsilon}^p = \dot{\varepsilon}_r \left| \frac{\bar{\sigma}}{g(\bar{\varepsilon}^p)} \right|^{1/m} \quad (7)$$

$$g(\bar{\varepsilon}^p) = \sigma_r \{ \tanh(k_1 \bar{\varepsilon}^p) + k_2 + H_e(\bar{\varepsilon}^p - \varepsilon_r)k_3(\exp \bar{\varepsilon}^p - \exp \varepsilon_r) \} \quad (8)$$

where $\dot{\varepsilon}_r$ is the reference strain rate, σ_r is the reference stress, and ε_r is the reference strain. k_1 – k_3 are material parameters. $H_e(x)$ is the step function expressed by Eq. (9):

$$H_e(x) = \begin{cases} 1 & (x \geq 0) \\ 0 & (x < 0) \end{cases} \quad (9)$$

Equations (10) and (11) were used as flow stress equations in this research. Fracture caused by the craze occurs due to continuous fracture of several hundred to several thousand molecular chains. Use of the craze density-based softening law and flow stress to model this process phenomenologically enables simulation using micro-parameters of final failure originating in the growth of the craze.

$$g(\bar{\varepsilon}^p) = \sigma_r \{ \tanh(k_1 \bar{\varepsilon}^p) + k_2 + H_e(\bar{\varepsilon}^p - \varepsilon_r)k_3(\exp \bar{\varepsilon}^p - \exp \varepsilon_r) \} \times f(\omega) \quad (10)$$

$$f(\omega) = 1 - \tanh((\omega - w_2)w_3) \quad (11)$$

Here, w_2 is the craze density at which the softening equation is first applied, and w_3 is a parameter that expresses the magnitude of softening. This softening law

Table 1 Material parameters

A	0.383	σ_r	100.55 (MPa)
B	1.226	σ_y	17.432 (MPa)
A_1	19.82 (MPa)	k_1	44.795
B_1	41766.5 (MPa ²)	k_2	0.863
D_1	0.999	k_3	1.243
D_2	100.541	q_1	0.0001177
ε_c	1.087	q_2	12.44
m	0.069	q_3	2.908
$\dot{\varepsilon}_r$	1	w_2	0.22425
ε_r	0.0001	w_3	4.0968

multiplied by the right side of Eq. (11) gives the flow stress, $g(\bar{\varepsilon}^p)$. FE analysis was conducted by incorporating the material laws as a user subroutine of RADIOSS Ver. 4.4, a general explicit solver. The FE model of the bumper face for impact tests is shown in Fig. 11. The bumper face was modeled using shell elements.

4.4. Results of Bumper Face Impact Tests and Simulation

A simple identification procedure of material parameters for the constitutive laws was already proposed, and the material parameters were identified for this concept⁽¹⁹⁾. Table 1 shows the materials parameters used in this FE analysis.

Figure 12 shows the results for deceleration obtained from the simulation and from actual bumper face impact

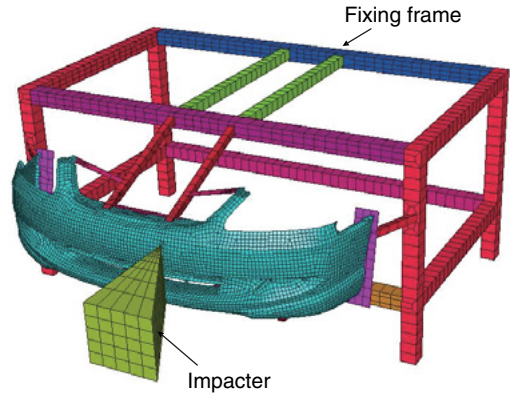


Fig. 11 Finite element model of bumper face

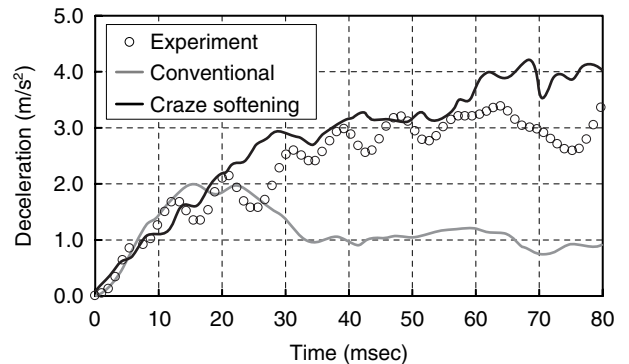


Fig. 12 Deceleration time histories of bumper face test

tests. The high-speed camera indicated that the craze occurred at 20 msec, and that cracks appeared at 40 msec. The results of the conventional simulation method that predicts fracture based on the strain showed fracture occurring earlier and at a lower level of deceleration than the test results. By contrast, the method using the constitutive material laws developed in this research showed good correlation with test results for the occurrence of the craze and the development of cracks.

Figure 13 shows the photographs of deformed bumper face at 40, 60, and 80 msec. Those photographs were taken by the high speed video camera located backside of the bumper face as shown in Fig. 4. The crack started propagating at 40 msec and continued propagating as the impactor penetrated to the bumper face. It was observed that the whitening at the crack tip was generated by the craze. It was found that the failure model observed in this test was formed due to the predominance of craze. Figure 14 shows the deformation with the simulated craze density distribution of bumper face at 40, 60, and 80 msec. In the simulated phenomena at 40 msec, the crack initiated and propagated show the same trend as the experiment. And the trends of the test and simulation results were the same in the craze density distributions at the crack tip and

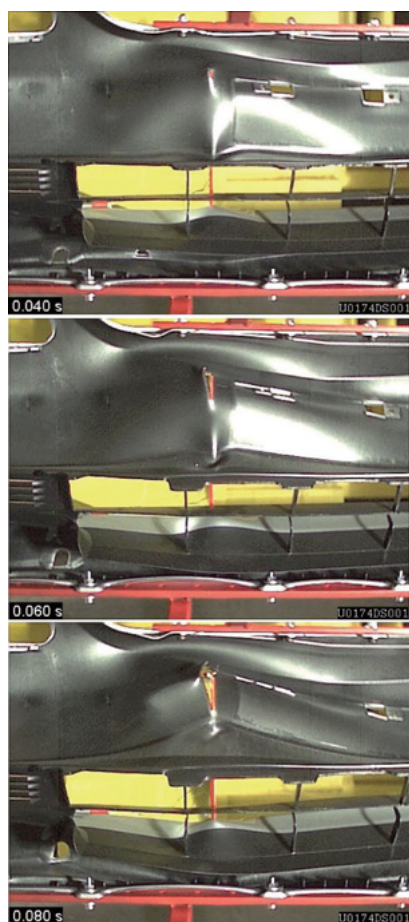


Fig. 13 Deformation of bumper face

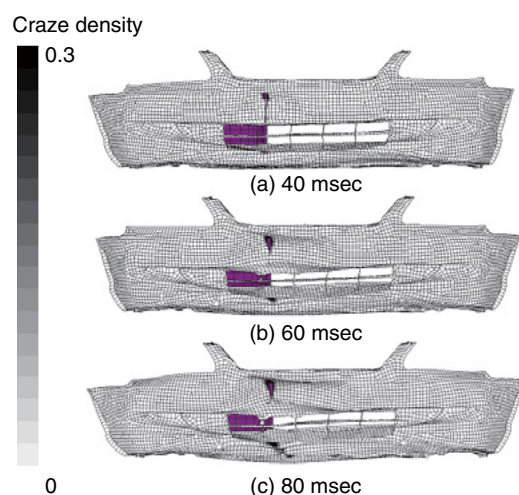


Fig. 14 Simulated deformation and craze density distribution of bumper face

around the impactor impact area. These results confirm the validity of the new method from the perspective of both deceleration and deformation. This confirms that the method proposed by this study can analyze macro-scale large deformation and impact behavior of polymeric materials using micro- material parameters.

5. Conclusion

The craze nucleation and growth and fibril fracture were modeled using the results of micro-scale tests and coarse grained molecular dynamics, and a phenomenological softening law that considered final failure process was proposed to analyze impact large deformation at the component level with micro-scale material parameters. This softening law was used to modify the model proposed by Shizawa, et al., and the modified model was used in the analysis of impact large deformation at the component level. Results obtained using the new method showed good correlation with the test results at the component level, confirming the validity of the method.

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Development of Orthotropic Damage Model for Crush Simulation of Extruded Aluminum Member

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ABSTRACT

It is well known that an extruded aluminum member has the different mechanical characteristics between the axial direction and the normal direction. It is important to consider the extruded direction dependence of mechanical characteristics in evaluating the energy absorption performance of the extruded aluminum member at non-uniform loading directions. For simulating such an orthotropic mechanical characteristic, the simple orthotropic damage model was proposed with the simulation method. The test specimens were cut out of the axial aluminum member in the axial and normal directions. The tensile, three-point bending and mixed-mode fracture tests were conducted in order to obtain the orthotropic characteristics. The simple orthotropic damage model was proposed and applied to the crush behavior of the extruded aluminum member. The load displacement curve and damage zone development were compared with those obtained by the experiments. As a result, the usefulness of the orthotropic damage model and simulation methods were validated.

1. Introduction

The mechanical properties of extruded aluminum members are known to differ in the axial and normal directions. It has been considered useful to develop a simulation method that takes this fact into consideration, for use in the evaluation of the energy absorption performance of extruded aluminum members under loads applied from various directions. A considerable amount of research has been conducted to characterize the impact energy absorption of extruded aluminum members and develop the simulation methods^{(1)-(6), (8), (9)}. The energy absorption characteristics of extruded aluminum members under various load conditions has been evaluated⁽¹⁾⁻⁽⁶⁾, and simulation methods that consider the orthotropic mechanical properties of the extruded aluminum members have been developed⁽⁷⁾⁻⁽⁹⁾, but the focus of research has been limited to microscopic deformation.

The research discussed in this paper therefore set out to develop a simple damage model that considered orthotropic mechanical properties in extruded aluminum members and to use this model in developing a simulation method. First, test specimens were cut of an extruded aluminum member in the axial and normal directions and subjected to tensile and three-point bending tests to

characterize the orthotropic mechanical properties of the material. In addition, mixed-mode fracture tests were conducted to measure the stress intensity factor in the mixed mode, in consideration of the complexity of the stress state at the points at which cracks initiate during crush deformation of an extruded aluminum member. The orthotropic mechanical properties measured by tensile and three-point bending tests are analyzed so that the material model would be proposed. Finally, numerical simulations of crush tests on extruded aluminum members were conducted using the simple orthotropic damage model that had been developed. The validity of the new method was then verified by comparison of simulation results with test results.

2. Material Constitutive Law

To simplify the material model employed in this research, the linear elastoplastic model expressed by Eq. (1) was used. Figure 1 shows an illustration of the model.

$$\sigma = a + b\bar{\epsilon}_p \quad (1)$$

Here, a is the yield stress, b is the strain hardening parameter, and $\bar{\epsilon}_p$ is the equivalent plastic strain. The simulation used in this study is conducted on the plane

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stress conditions, and the equivalent plastic strain is therefore given by Eq. (2).

$$\bar{\varepsilon}_p^2 = 4/3(\varepsilon_{xx}^2 + \varepsilon_{xx}\varepsilon_{yy} + \varepsilon_{yy}^2) + 1/3(\varepsilon_{xy}^2 + \varepsilon_{yz}^2 + \varepsilon_{zx}^2) \quad (2)$$

Here, ε_{ii} are the tensile plastic strain components in each direction, and ε_{ij} are the shear plastic strain components.

The relationship between stress and strain in the material is given by the following equations, based on the total strain theory:

$$\varepsilon_{all} = \varepsilon_e + \varepsilon_p \quad (3)$$

$$\sigma = E(1 - D)\varepsilon_e \quad (4)$$

$$\varepsilon_p = (\sigma - a)/b \quad (5)$$

Here, ε_{all} is the strain vector, σ is the stress tensor, and E is the elastic matrix.

The isotropic linear damage model proposed by Johnson and Cook⁽¹⁰⁾ was used as a reference in formulating the orthotropic damage model used in the simulation, which is expressed by the following equations:

$$D = \int_{\bar{\varepsilon}_1}^{\bar{\varepsilon}_2} 1/\bar{\varepsilon}_p d\bar{\varepsilon}_p \quad (6)$$

$$\bar{\varepsilon}_1 = 2/\sqrt{3} \{c_{1xx}\varepsilon_x^2 + c_{1xy}\varepsilon_x\varepsilon_y + c_{1yy}\varepsilon_y^2 + 1/4(c_{1xy}\varepsilon_{xy}^2 + c_{1yz}\varepsilon_{yz}^2 + c_{1zx}\varepsilon_{zx}^2)\}^{1/2} \quad (7)$$

$$\bar{\varepsilon}_2 = 2/\sqrt{3} \{c_{2xx}\varepsilon_x^2 + c_{2xy}\varepsilon_x\varepsilon_y + c_{2yy}\varepsilon_y^2 + 1/4(c_{2xy}\varepsilon_{xy}^2 + c_{2yz}\varepsilon_{yz}^2 + c_{2zx}\varepsilon_{zx}^2)\}^{1/2} \quad (8)$$

; where D is the orthotropic damage parameter, $\bar{\varepsilon}_1$ is the equivalent plastic strain at the initiation of damage, and $\bar{\varepsilon}_2$ is the equivalent plastic strain when damage ceases to occur. c_{ijkl} and c_{2ijkl} are orthotropic parameters. The use of different weight functions for the strain components in each direction enables their effect on the overall strain to be adjusted. Orthotropy can be treated phenomenologically by controlling the degree of contribution of these equivalent strains. In the present simulation, the material fractures when D reaches 1; when D reaches 1 at the integration points of the shell elements in the direction of thickness, the stress tensor at those integration points is calculated as 0. In addition, a multilayered integration

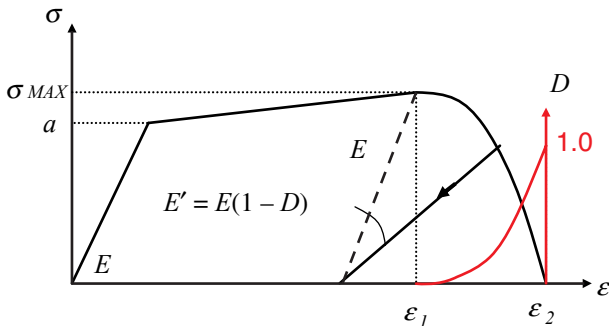


Fig. 1 Illustration of material model with damage

point determination method is used in which the shell elements are eliminated when D reaches 1 at all the shell element integration points. This enables simple expression of the initiation and propagation of cracks.

The material law was incorporated as a user subroutine of the general explicit solver RADIOSS Ver4.4 in the finite element models. Figures 2 and 3 show finite element models of a three-point bending test and an extruded aluminum crush test respectively. The models employed Belytschko shell elements, and five integration points were used in the direction of thickness.

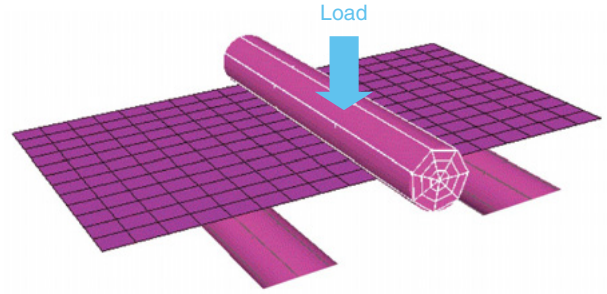


Fig. 2 Finite element model of three-point bending test

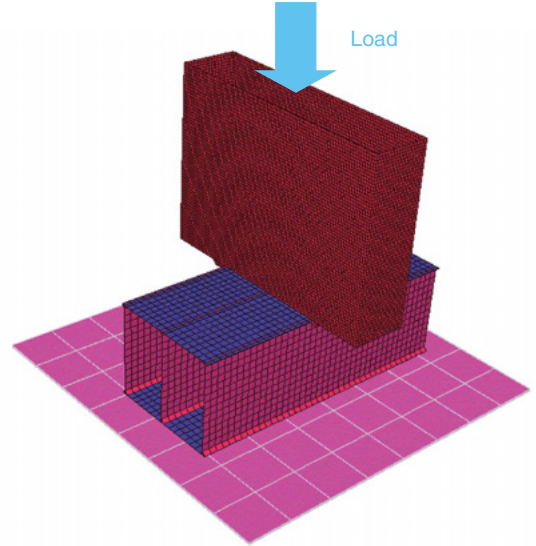


Fig. 3 Finite element model of aluminum extrusion crush test

3. Test Method

3.1. Test Material and Test Specimens

Extruded A6061-T6 aluminum alloys were used in the tests. Tensile, three-point bending, and mixed-mode fracture test specimens were cut in the axial and normal directions from thin rectangular pipes with closed cross-sections, formed from sheets of 2 and 3 mm in thickness. Table 1 shows the chemical compositions of the test

Table 1 Chemical composition of A6061 (wt. %)

Si	Fe	Cu	Mn
0.40-0.80	0.7	0.05-0.20	0.15
Mg	Cr	Zn	Ti
0.08-0.12	0.04-0.35	0.25	0.15

material. Two chemical compositions, using different ratios of Mg and Si, were used in the tests to understand the ductile material and brittle material behaviors. Composition A used small amounts of Mg and Si, while composition B used large amounts. The materials were aged for 3 hours at 190 °C. Figure 4 shows the geometry of the tensile test specimens, cut out of thin sheets (2 mm) in accordance with the specifications of JIS Z2201 14B. Thin oblong sheets ($L = 40$, $w = 20$, $t = 2$ mm) were used in the bending test. Figure 5 shows the geometry of the mixed-mode fracture test specimen proposed by Richard⁽¹¹⁾. The pre-crack was introduced by the fatigue test based on the specifications of JSME S001-1992. The extruded aluminum crush test specimens were 200 mm in length, and cut from a thin rectangular pipe with a closed cross-section.

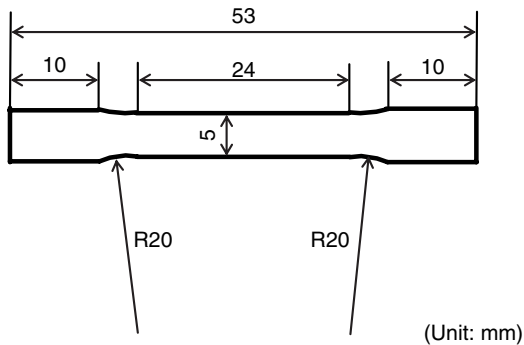


Fig. 4 Schematic illustration of tensile test specimen

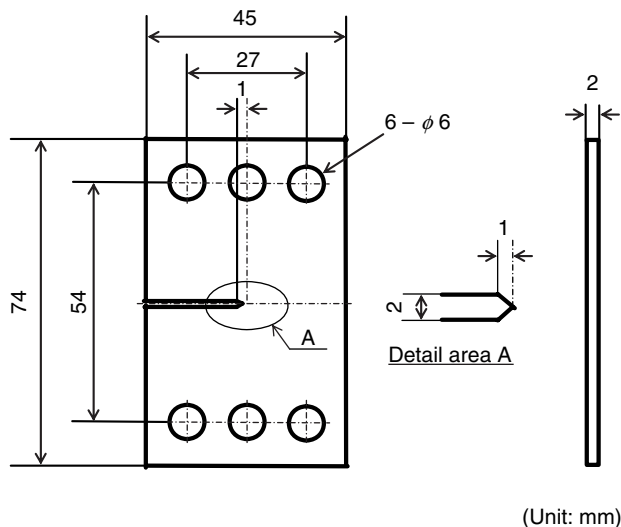


Fig. 5 Schematic illustration of mixed-mode fracture tests specimen

3.2. Test Equipment and Method

3.2.1. Tensile and three-point bending tests

A general screw driven test device manufactured by Shimadzu Corporation (rated capacity: 5 kN) was used in this research to measure quasi-static stress-strain behavior. A tensile speed of 0.5 mm/min was applied. A wedge speed of 50 mm/min was applied in the three-point bending test, and the load applied by the wedge and the displacement were measured. Figure 6 shows a photograph of the three-point bending test device.

3.2.2. Mixed-mode fracture test

Fracture tests were conducted using a mixed-mode fracture test jig (Fig. 7) in the same general screw driven test device used for the tensile and bending tests. Equation 9 defines the stress intensity factor, K , which is used for evaluating the stress state at the crack tip when the crack begins to initiate. This is the finite element analytic solution formulated by Richard and Benitz⁽¹²⁾. Suffix I and II indicate mode I, the fracture state under tensile predominance, and mode II, the fracture state under shear predominance.

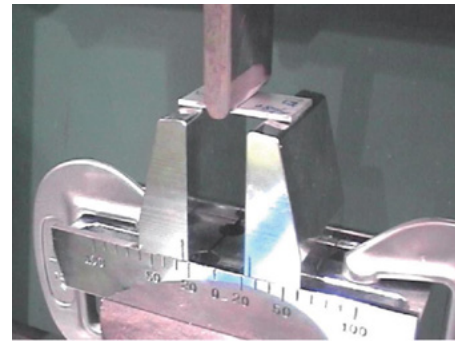


Fig. 6 Photograph of bending test device

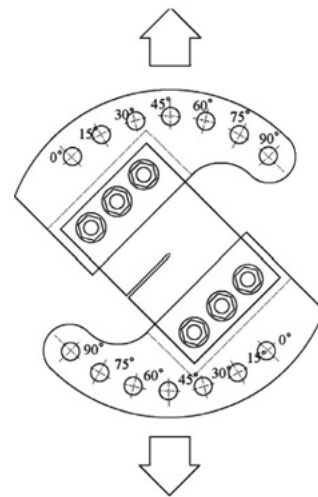


Fig. 7 Schematic illustration of mixed-mode fracture test device

$$K_{I,II} = F_{I,II} P / WB \sqrt{\pi a} \quad (9)$$

W is the width of the test specimen, B is the thickness of the test specimen, a is the length of the pre-crack, and P is the load. F_I and F_{II} are correction factors determined by the axial angle of the load and the length of the pre-crack.

During the tests, the pre-crack tip was monitored using a video microscope (Scala VMS-3000), and the initiation and propagation of cracks were measured. Figure 8 shows a photograph of the test device.

3.2.3. Crush test of extruded aluminum member

Crush tests of extruded aluminum members were conducted as component-level tests. Figure 9 shows a photograph of the test device. Extruded aluminum members were fixed in an iron jig and crushed by a square, columnar compression jig moving at 100 mm/min. A home video camera was used to record the deformation of the member and the development of cracks.

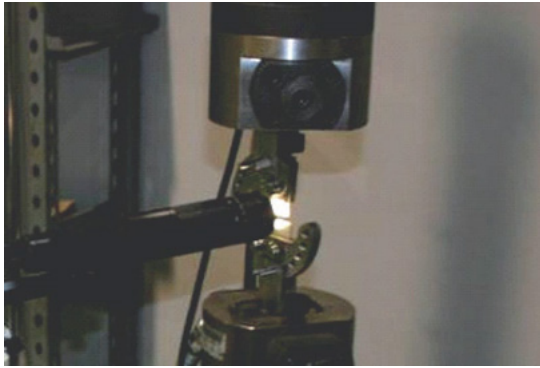


Fig. 8 Photograph of mixed-mode fracture test device

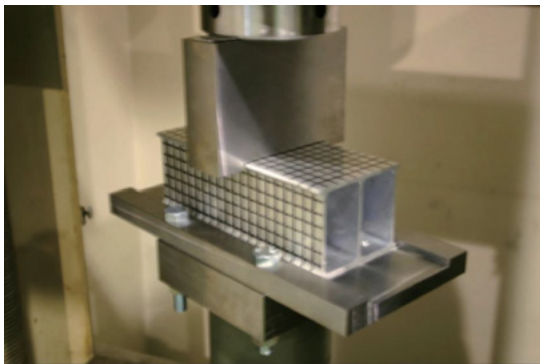


Fig. 9 Photograph of crush test device

4. Test Results and Consideration

4.1. Tensile Test Results

Table 2 shows the mechanical properties of compositions A and B, as determined from tensile test

Table 2 Mechanical properties

	Composition A		Composition B	
	Axial	Normal	Axial	Normal
Elastic modulus (MPa)	70.5	70.8	71.4	70.1
Yield stress (MPa)	278	270	303	291
Tensile strength (MPa)	316	311	336	331
Rupture stress (MPa)	270	264	296	308

results. Figures 10 and 11 show the stress strain curves of compositions A and B, respectively. There are slight differences in the stress values for the axial and normal directions, but no major differences are observed. As shown in Fig. 10 also indicates that orthotropy can be ignored in terms of yield stress and tensile strength. Composition A, however, did display a higher fracture strain than composition B, and displayed more ductile fracture behavior than composition B. Composition B, with a higher ratio of Mg and Si than composition A, recorded higher stress values than composition A.

4.2. Three-point Bending Test Results

Figures 12 and 13 show the relationship between load and displacement of the bending test for compositions A and B respectively. For both compositions, fracture displacement was greater in the axial than in the normal direction. The effect of orthotropy, which was negligible in the tensile tests, was prominent in the bending tests. In addition, composition A displayed greater ductility than

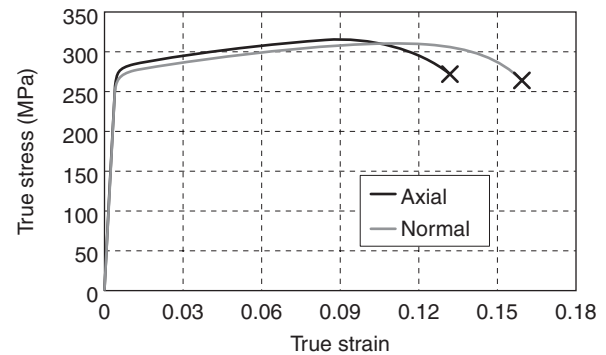


Fig. 10 Stress-strain curves of composition A

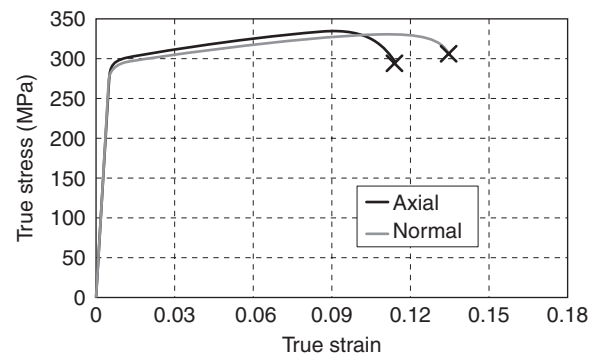


Fig. 11 Stress-strain curves of composition B

composition B, which contained a higher ratio of Mg and Si. Previous research⁽¹³⁾⁻⁽¹⁵⁾ has indicated that the particles of additives such as Si in Mg-Si aluminum alloys become the point of initiation of cracks, and this research produced identical results. Figure 14 shows the photograph of the backside of the bending specimen of composition A after three-point bending test. The area indicated in Fig. 14 was observed by a scanning electron microscopy (JEOL JSM-5300T). Figure 15 shows a scanning electron micrograph of the backside of the composition A test specimen cut in the axial direction after the bending test. Figure 16 shows the same for the composition A test specimen cut in the normal direction after the bending test. As Fig. 15 shows, the fractures in the test specimen are caused by tensile loading from the outer most layer, and numerous microscopic cracks can

be observed on the surface. By contrast, in the test specimen cut in the normal direction, fracture surfaces are formed tilted in the direction of thickness along shear bands, and localization of strain has occurred. This difference in fracture pattern is considered to cause the orthotropy observed in the three-point bending test results.

4.3. Mixed-mode Fracture Test Results

Stress intensity factors for the pre-crack tip were determined from the load-displacement curve and crack initiation points observed in mixed-mode fracture tests. Figures 17 and 18 show the stress intensity factors for compositions A and B respectively. Overall, stress intensity factors are higher for composition A than composition B, a result which supports the results of the three-point bending tests, in which composition A displayed more ductile fracture behavior than composition B. Comparing the stress intensity factors for each composition in the axial and normal directions, the stress intensity factor was higher for the axial than the normal direction for the stress state in which mode I was dominant, but the difference between the stress intensity factors in the axial and normal directions decreases as the mode II components increased. The stress state at the point of crack initiation in the three-point bending tests was close to mode I, but the dominant stress state at the

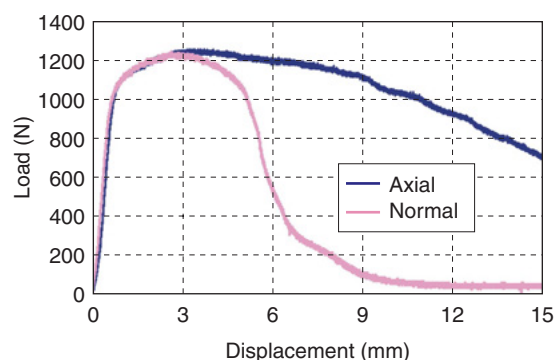


Fig. 12 Load displacement histories of composition A

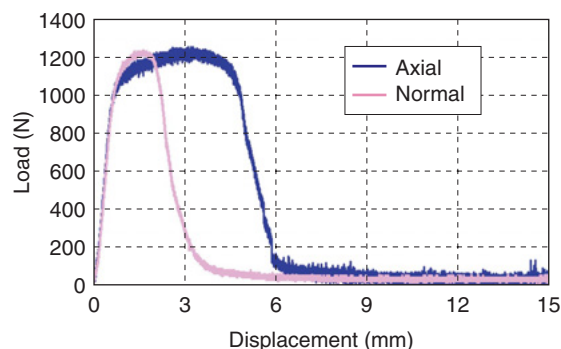


Fig. 13 Load displacement histories of composition B

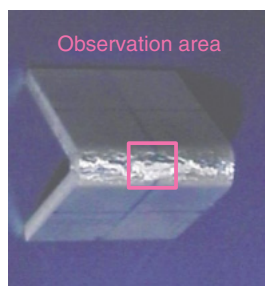


Fig. 14 Specimen of composition A after bending test

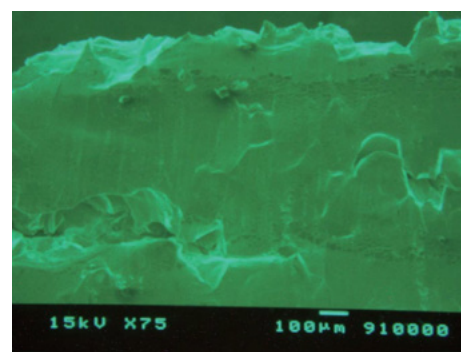


Fig. 15 SEM photograph of composition A cut in the axial direction after bending test

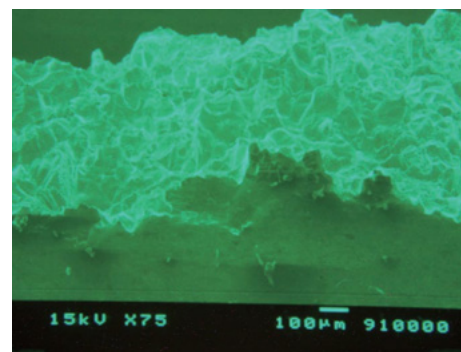


Fig. 16 SEM photograph of composition A cut in the normal direction after bending test

point of crack initiation in the tensile tests was considered to be the mode II state, when shearing predominates, rather than pure mode I. This is believed to be the cause of the mixed-mode fractures. These test results indicated that it is most reasonable to establish material parameters for an orthotropic damage model on the basis of the results of three-point bending tests, in which the effect of orthotropy is dominant.

4.4. Extruded Aluminum Member Crush Test Results

Figure 19 shows the compression load-displacement curve obtained from crush tests. As in the three-point bending tests, the maximum load for composition A was lower than for composition B. Figures 20 and 21 show extruded aluminum members following the tests. The crack of composition A did not reach the end area of the extruded aluminum members, however, the crack of composition B reached the end area. Therefore, composition A showed more ductile fracture behavior, with brittle fracture behavior observed in composition B.

4.5. Comparison of Three-point Bending Test and Simulation

All of the material parameters in the material constitutive law used in the simulations, with the

exception of the orthotropy parameters, were identified on the basis of the results of tensile tests. A set of orthotropic parameters was determined so as to ensure that simulation results for load-displacement curves corresponded well with the results of the three-point bending tests. Table 3 shows the material parameters identified for compositions A and B. Figures 22 and 23 compare the results of simulations of three-point bending tests using these parameters with the results of actual tests for compositions A and B respectively. The results show that the orthotropic damage model proposed in this research has simulated the orthotropy observed in actual test results at

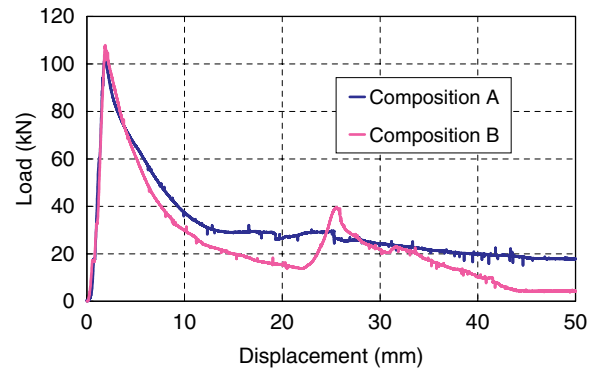


Fig. 19 Load displacement histories of crush tests

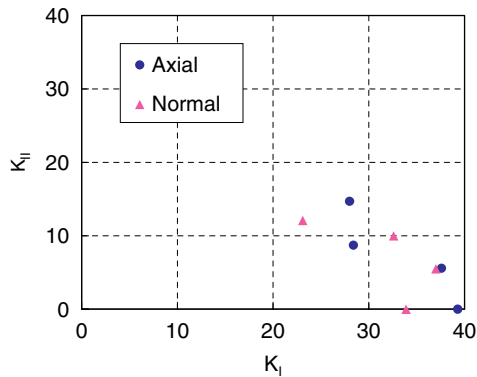


Fig. 17 Mixed-mode stress intensity factors of composition A

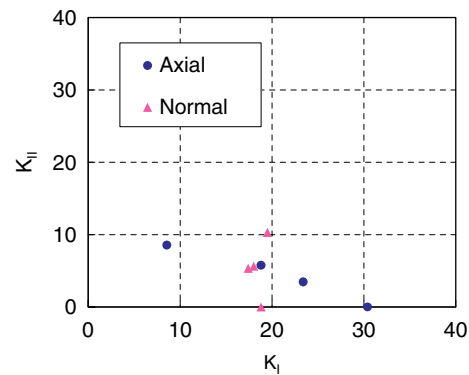


Fig. 18 Mixed-mode stress intensity factors of composition B

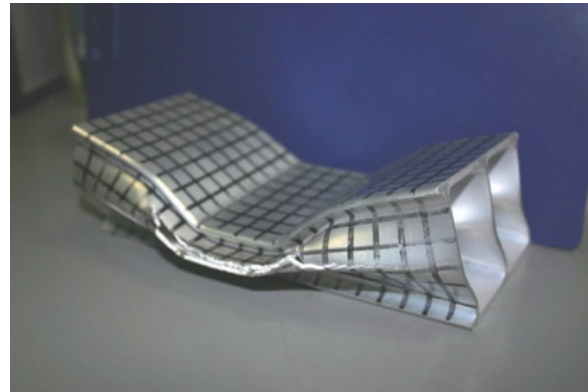


Fig. 20 Composition A after crush test

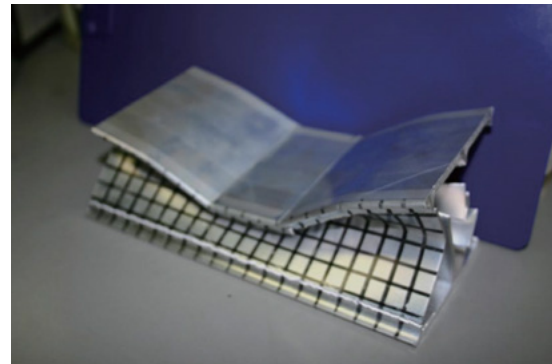


Fig. 21 Composition B after crush test

Table 3 Material parameters

	Composition A	Composition B
Elastic modulus (GPa)	71	71
Yield stress (MPa)	290	310
Strain hardening parameter	420	420
ϵ_1	0.19	0.09
ϵ_2	0.27	0.095
C_{1xx}	0.401	0.184
C_{1xxy}	0.633	0.429
C_{1yy}	1	1
C_{1xy}	1	1
C_{1yz}	1	1
C_{1zx}	1	1
C_{2xx}	0.071	1
C_{2xxy}	0.267	1
C_{2yy}	1	1
C_{2xy}	1	1
C_{2yz}	1	1
C_{2zx}	1	1

the high accuracy. In addition, the simulation accuracy was low when material parameters based only on the results of tensile tests, without consideration of orthotropy.

4.6. Comparison of Extruded Aluminum Member Crush Test and Simulation

Figure 24 compares the results of extruded aluminum crush tests and numerical simulations conducted using the

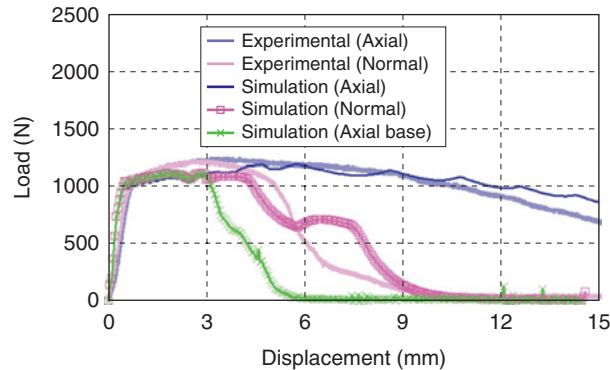


Fig. 22 Bending load displacement curves of composition A

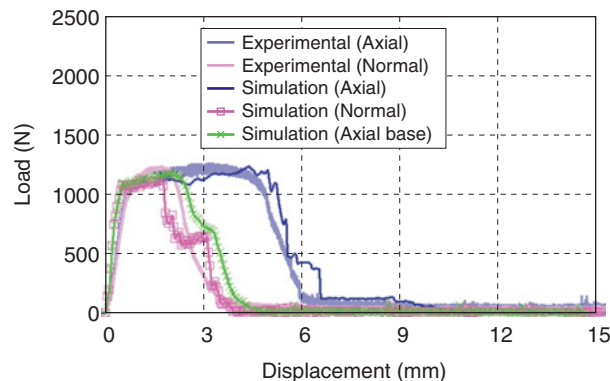


Fig. 23 Bending load displacement curves of composition B

set of material parameters established in this research. For both compositions A and B, the load-displacement histories simulated by using the new parameters show better agreement with the experimental result compared with those of a conventional simulation that does not consider orthotropy. Figure 26 shows the distribution of equivalent plastic strain simulated for composition A. Figure 27 shows a photograph of an extruded aluminum member after a crush test. As Figures 26 and 27 indicate, the simulation was able to reproduce test results in terms of deformation mode and areas in which cracks were initiated with a high degree of accuracy. This suggests that the orthotropic damage model proposed in this research enables simple simulation of crack initiation and development following the point of maximum load.

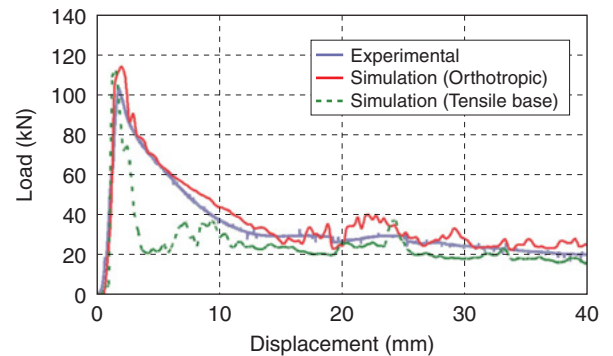


Fig. 24 Load displacement histories of composition A

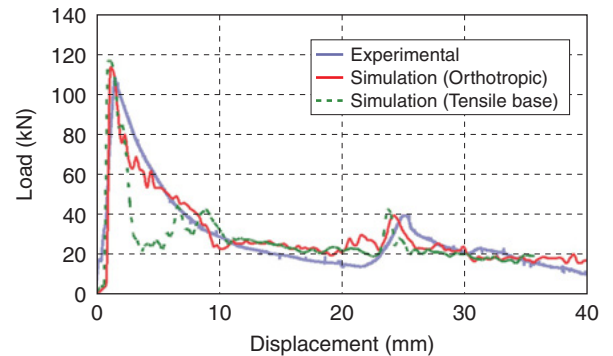


Fig. 25 Load displacement histories of composition B

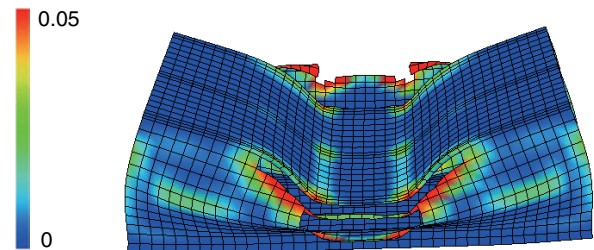


Fig. 26 Equivalent plastic strain distribution of composition A

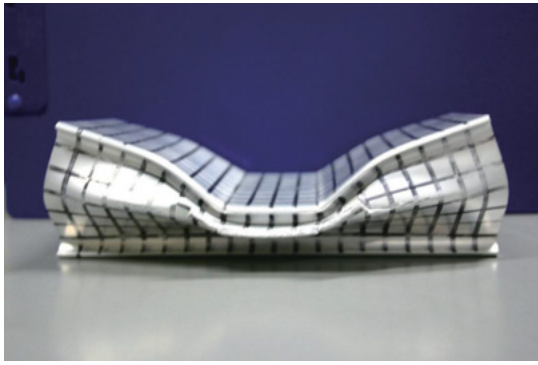


Fig. 27 Photographs of composition A member after crush test

5. Conclusion

A simple orthotropic damage model and simulation method were proposed to predict the crush behavior of extruded aluminum members. This method enables material parameters for the elastoplastic constitutive model to be determined on the basis of the results of tensile tests, and the material parameters for the orthotropic damage model to be determined on the basis of the results of three-point bending tests. Finite element simulations of extruded aluminum member crush tests using these parameters showed correlation with the results of actual tests in terms of load-displacement characteristics and areas in which damage occurred. This suggests that an orthotropic damage model is useful in simulating deformation and load when extruded aluminum members are crushed. The simulation method proposed in this research was shown to be effective in simulating the crush behavior of extruded aluminum members.

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Hiroyuki MAE



Kikuo KISHIMOTO

Study of Method to Reduce Low-frequency Noise by Controlling Panel Vibration Phase

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ABSTRACT

A method of reducing the sound pressure of low frequency noises such as booming and drumming noise was devised by focusing on the phase relation between multiple panel vibrations. Low frequency noise is affected by the acoustic modes, which are determined by the shape and the size of the cabin. Accordingly, an advantageous panel phase relation was clarified for the sound pressure inside the cabin in the low frequency region, and a method of realizing this phase relation by means of the distribution of the panel resonance frequencies was invented.

Also, technology for performing BEM analysis in the low frequency region was established. The sound pressure in the cabin was predicted according to the boundary conditions obtained from the measurement results, and the contribution of each panel was clarified. From these results, the multiple panel resonant frequencies of a completed vehicle were adjusted, and the necessary phase relation for canceling the sound pressure was realized, thus the sound pressure was reduced.

1. Introduction

Due to heightened demands for environmental performance, elements that increase engine and suspension vibration have proliferated in recent years, such as the use of direct injection engines and diesel engines, and expansion of the AT lock-up range to enhance fuel economy. In addition, demands for weight reduction are also increasing, and in many cases, conflict with vibration and noise performance. As a result, development to reduce low-frequency vibration and noise is becoming more challenging.

The acoustic 1st mode frequency inside a station wagon or minivan is determined by the shape and size of the cabin. However, this frequency is close to the resonance frequency of the front suspension, so low-frequency noise, such as booming and drumming (low-frequency road noise) easily become an issue, and many man-hours are spent developing countermeasures for these phenomena. Recent research^{(1), (2)} has investigated the structural-acoustic coupling system in this area, but most countermeasures consist of reinforcing the body framework or adding mass dampers or dynamic dampers, etc. There are some cases where countermeasures focus on panels, but most of these implement countermeasures

for only a single panel⁽³⁾, and few cases take into account the relation between multiple panels.

This paper focuses on station wagons and minivans, and describes a method for reducing the sound pressure inside the cabin by appropriately distributing the resonance frequencies of the windshield, roof, tailgate and other multiple panels that comprise the cabin. This method is applied for the frequency range in which the acoustic 1st mode is dominant. To validate this method, case studies using the boundary element method (BEM)⁽⁴⁾ for low-frequency booming noise and drumming are presented.

The objective of this method is to quickly determine specifications in the initial development stages by setting an appropriate resonance frequency for each panel.

2. Method for Reducing Noise by Controlling Panel Phase

2.1. Mechanism of Noise Generation

The mechanism of low-frequency noise generation is as follows: the vibration sources are the engine and the suspension, and that vibration is conveyed to the body through the engine mounts and suspension bushings, etc. This vibration then induces vibration in the windshield,

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roof, tailgate and other panels that comprise the cabin, which results in noise produced inside the cabin. Accordingly, the acoustic mode present in the cabin affects the sound pressure. However, the acoustic mode frequency is determined by the shape and size of the cabin, and the acoustic 1st mode frequency in station wagons and minivans is approximately 40 to 50 Hz. This mode produces a sound distribution with anti-nodes (high pressure fluctuation) in the front and rear of the cabin and a node in the center as shown in Fig. 1.

2.2. Issues with Conventional Methods

Most conventional methods for reducing low-frequency noise have focused on the panel with the highest contribution, and structural changes that modify the characteristics of panel response to reduce the vibration level. In such cases, there is often a limit to how much the vibration level can be reduced by changing the structure, so efforts then have also shifted to reducing the panel vibration with the next highest contribution. However, this method of development takes a lot of time to determine countermeasure specifications. Therefore, to quickly determine the specifications within the limited development period, a new method was necessary that focuses on the characteristics of multiple panels and does not rely simply on reducing the vibration level.

2.3. Method for Reducing Noise Focusing on the Phase of Multiple Panels

This study took into account the characteristics of the acoustic 1st mode, focused on the phases of two or more panels, and proposed a method for realizing a phase relation that reduces sound pressure by distributing the resonance frequencies.

For example, consider the case where sound pressure is produced from only two panels: the front windshield and the tailgate. At low frequencies that are relatively unaffected by the acoustic 1st mode, such as the idle noise range (20 to 25 Hz) for second-order vibration of a 4-cylinder engine, panel phase relations that increase the volumetric variation in the cabin space also increase the sound pressure. On the other hand, in the range where the acoustic 1st mode is dominant, phase relations that increase the volumetric variation exhibit a sound distribution with opposite phase at the front and back of the cabin due to the acoustic mode characteristics as

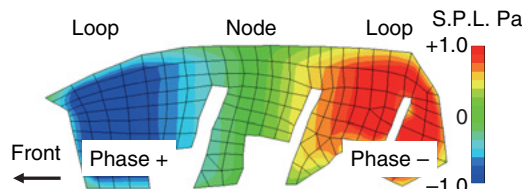


Fig. 1 Sound pressure distribution of 1st acoustic mode

shown in Fig. 2. This means that the sound produced from the windshield is cancelled by the sound produced from the tailgate, so the sound pressure is reduced. In other words, it is beneficial to have a structure where the front and rear panels vibrate out of phase with each other in the frequency range of the 1st acoustic mode of the cabin interior.

Investigations were performed for validation using this method of sound pressure reduction by controlling the panel phases.

2.4. Controlling the Panel Phase according to the Resonance Frequency

The fundamental characteristic of a basic single degree of freedom model is that the phase inverts at the resonance frequency. Therefore, it was thought that this feature could be used to realize a panel phase relation that reduces sound pressure by optimally distributing the resonance frequencies of the panels with high contributions to the sound pressure inside the cabin.

Vibration tests were conducted using completed vehicles, and the vibration phase characteristics were compared for the center of the windshield and front pillar section. The results confirmed that the phase inverts at the 1st mode resonance frequency of the windshield as shown in Fig. 3. These results indicate that the phase characteristics of each panel in a completed vehicle can be aligned by changing the respective resonance frequencies.

2.5. Panel Phase Combinations

A method of achieving a phase relation that cancels the acoustic 1st mode was investigated from the viewpoint of the resonance frequencies of two panels. By setting the resonance frequency of the windshield lower and the resonance frequency of the tailgate higher than the acoustic mode, the phases of the two panels in the acoustic mode range become opposite to each other and provide destructive interference with the cabin interior mode, and the sound pressure can be reduced. Figure 4

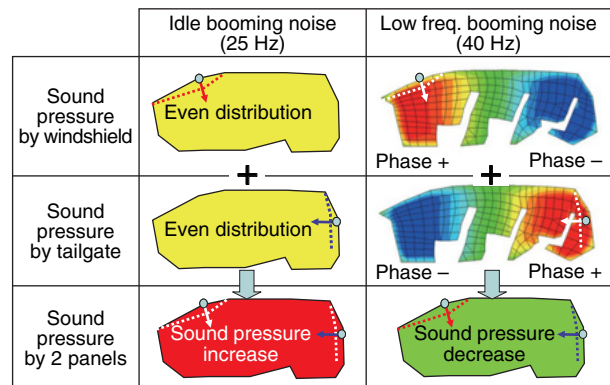


Fig. 2 Mechanism of noise generation

shows these resonance frequencies and phases.

It was found that noise is increased or cancelled depending on the phase relation of the panels, which is determined by the location of their respective resonance frequencies. Completed vehicles have high sound pressure levels at frequencies near the acoustic 1st mode, so this study set out to control the panel phase and reduce the sound pressure inside the cabin around the frequency of the 1st acoustic mode of the completed vehicles.

The panel phase relation is determined by whether the resonance frequency of each panel is set higher or lower than the acoustic 1st mode frequency of 43 Hz. Four panels had a high degree of contribution to the sound pressure inside the cabin for the vehicle used in these tests. In descending order of highest contribution, these panels were the windshield, the roof, the rear floor, and

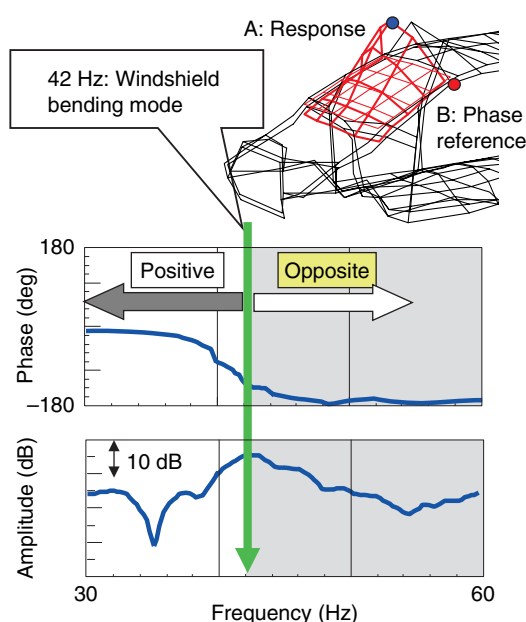


Fig. 3 Transfer function from front pillar to center of windshield

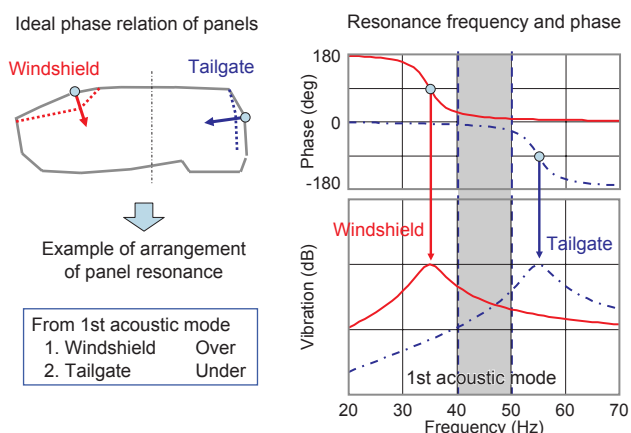


Fig. 4 Ideal arrangement of panel resonance

Table 1 Combination of panel resonance frequency

	Windshield	Roof	Rear floor	Tailgate	Phase condition
A	↑	↑	↑	↓	Even
B	↑	↑	↓	↓	Poor
C	↑	↓	↑	↓	Excellent
D	↑	↓	↓	↓	Good
E	↓	↑	↑	↓	Good
F	↓	↑	↓	↓	Excellent
G	↓	↓	↑	↓	Poor
H	↓	↓	↓	↓	Even

■ Unsuitable phase condition

↑ Resonance frequency : over 1st acoustic mode

↓ Resonance frequency : under 1st acoustic mode

the tailgate. Therefore, various combinations were investigated to determine whether the resonance frequency of each panel should be set higher or lower than 43 Hz. For structural reasons it is a challenge to set the resonance frequency of the tailgate higher than 43 Hz, so it should be set lower. As a result, the possible resonance frequency combinations are the eight combinations A to H shown in Table 1. In addition, whether a certain phase increases or cancels the noise can be determined from the test results by using the windshield as the reference. This is shown in the table, with shaded distributions indicating phase relations that increase the noise produced from the windshield. Combinations with few or no shaded areas are best, so this shows that combinations C and F are reliable.

Investigations were performed for validation using this approach of reducing sound pressure by controlling the panel phases.

3. Analysis Method Using BEM

3.1. Prediction of the Sound Pressure inside the Cabin Using BEM

The sound pressure inside the cabin is a complex summation of the noise produced from the vibration of each panel inside the cabin. To know which panel is important with respect to the generated the sound pressure, a method for clarifying the panel contribution is necessary. Therefore, an analysis method was introduced that models the cabin space using the boundary element method (BEM), and then calculates the sound pressure from the panel vibration.

First, a closed cabin space, including seats, was modeled as shown in Fig. 5.

When the boundary surfaces are modeled as rigid walls, the calculated acoustic mode frequency is higher and the damping is lower than the measurement results using a completed vehicle. This can cause a sharp peak to appear, which increases the difference between the calculated and measured the sound pressure inside the

cabin and lowers the accuracy. This is thought to be because the acoustic impedance of the boundary wall surfaces in the completed vehicle state is lower than that of the calculation model, due to the effects of the interior materials. Therefore, discrete characteristics tests were conducted on a number of interior material types, and except for the glass sections that have high acoustic impedance, the average results value was set for the boundary wall surfaces. This method reduced the difference between the calculated and measured values to within ± 3 dB as shown in Fig. 6.

3.2. Analysis of Panel Contribution

This method can be used to calculate the sound component produced by a panel from the vibration velocity of that panel. The panel contribution can then be analyzed by comparing the magnitude and phase of the sound component of each panel. A method for reducing sound pressure is examined below using the results of panel contribution analysis for two panels as an example.

When the windshield component and the roof

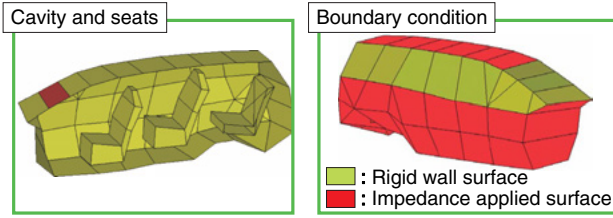


Fig. 5 BEM cavity model

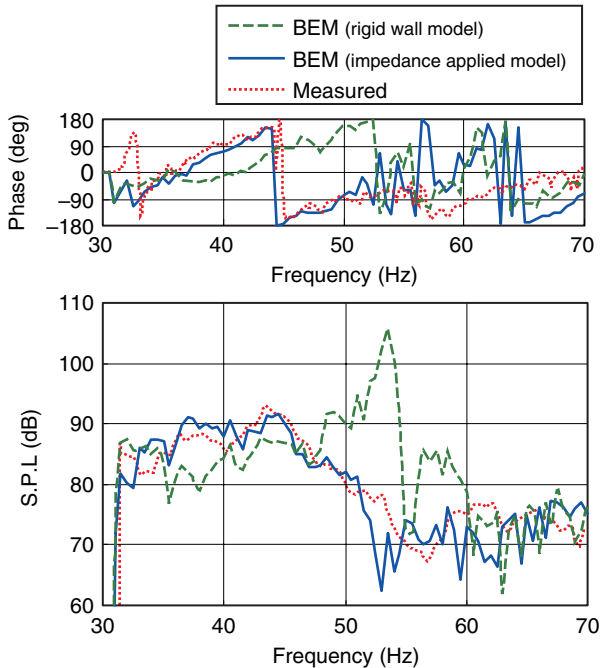


Fig. 6 Calculated S.P.L from actual car vibration

component make large contributions to the sound pressure inside the cabin, the sound pressure can be reduced by establishing a phase relation where these two sound components cancel each other. Figure 7 shows an example with the phase of the windshield component delayed by 90° . This makes the phase difference with the roof component approximately 180° , which reduces the total vector. In other words, it is thought that if this condition can be realized, the sound pressure inside the cabin can be reduced.

3.3. Application to Drumming (Low-frequency Road Noise)

3.3.1. Issues related to understanding the panel phase

The engine excitation force is the input source for the phenomenon of low-frequency booming noise, so the phase relation of multiple panels can be easily understood by using the engine pulse as the measurement reference. On the other hand, when applying this method to drumming (low-frequency road noise), the vibration input from the road surface is conveyed to the body through four tires. This means there are multiple uncorrelated inputs, so the signal processing should be enhanced in some way to understand the phase relations.

Therefore, the coherent component method was used for phenomena with multiple inputs. This method extracts a reference with a high degree of contribution to understand the phase relations as described below.

3.3.2. Coherent component method

The drumming input path is through the vibration of the suspension, so the phase relation of each panel was understood by breaking down the operational sound pressure and panel vibration into components that have a high correlation with the vibration of each wheel suspension.

Consider the case where k number of independent vibration inputs are conveyed from the suspension. Here, the measured the sound pressure inside the cabin P is expressed as the total of k components as shown in Eq. (1).

$$P = P_1 + P_2 + P_3 + \dots + P_k \quad (1)$$

In this manner, the phase relation of each panel is

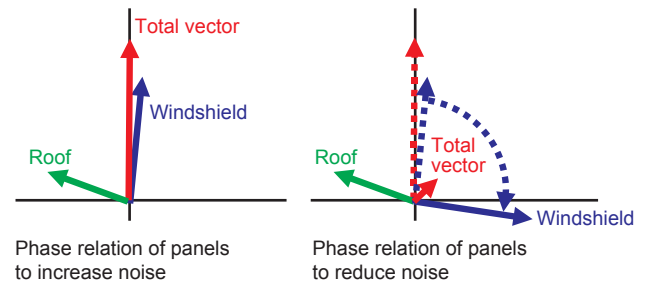


Fig. 7 Panel contribution analysis

investigated by breaking down the sound pressure into the components produced by each input. In this case, the sound component P_j attributable to the input j is obtained by multiplying the input j by the square root of the cabin sound coherence γ_j as shown in Eq. (2).

$$|P_j| = |P| \cdot \sqrt{\gamma_j} \quad (2)$$

The phase characteristics of P_j are obtained by the cross power spectrum of the input signal and the sound pressure inside the cabin.

Assuming there are m number of panels and focusing on the vibration of panel n , the component V_{nj} that correlates to input j is obtained from the coherence γ_j between the panel n vibration and input j as shown in Eq. (3).

$$|V_{nj}| = |V_n| \cdot \sqrt{\gamma_{nj}} \quad (3)$$

The phase characteristics of V_{nj} are obtained by the cross power spectrum of the panel n vibration. P_j is calculated as shown in Eq. (4) by applying the obtained vibration input V_{nj} that correlates to j to the BEM model, and at the same time the contribution of each panel can also be calculated.

$$P_j = \sum_m \{ [H_j] \cdot V_{nj} \} \quad (4)$$

H_j : Acoustic vibration transfer function of panel j

4. Results of Verification in Completed Vehicles

4.1. Low-frequency Booming Noise Reduction Effects

4.1.1. Sound pressure reduction effects of controlling the panel phase

Structural changes to the vehicle body were investigated using CAE with the goal of realizing combinations C and F in Table 1. The results showed that a significant increase in weight would be needed to raise the resonance frequency of the windshield for combination C. Weight restrictions are a major factor in the development of the subject vehicles, so combination F was determined to be more realistic, and this combination was verified using a completed vehicle. In combination F, the roof and rear floor sound components have phase relations that cancel the windshield sound component, and the tailgate sound component phase increases it. Therefore, overall noise reduction effects can be anticipated by achieving this phase relation.

Here, the test vehicle used for verification of low-frequency booming sound is labeled “a.” The resonance frequencies of each panel in vehicle “a” are 40 Hz for the windshield and 43 Hz for the roof, both of which are close to the acoustic 1st mode frequency. Structural investigations were performed using CAE to set the resonance frequencies of the windshield and the floor lower, and the resonance frequency of the roof higher than the acoustic 1st mode frequency, and these structural changes were then made on a test vehicle to realize

combination F. The acoustic 1st mode is dominant over a range centered on the mode frequency of 43 Hz, so the resonance frequencies were set at 33 Hz for the windshield, 60 Hz for the roof, and 36 Hz for the floor. This distributed the panel resonance frequencies sufficiently away from the acoustic mode frequency, and realized combination F in Table 1 as shown in Fig. 8.

After making these structural changes in a completed vehicle, the vibration phases of the windshield and roof were checked. The results confirmed that the phase relations cancel each other in the range where the acoustic 1st mode is dominant as shown in Fig. 9 (upper level). This resulted in significant low-frequency booming noise reduction effects of -10 dB or more around 43 Hz (engine speed: 1300 rpm), compared to before the structural changes.

The results of panel contribution analysis confirmed that the relation between the windshield sound component and roof sound component changed from characteristics that increased noise to those that cancel noise after the structural changes. (Fig. 10)

4.1.2. Comparison of results with the conventional method

The results were compared with the conventional method to compare the effects of phase control. The conventional method sets out to reduce the panel vibration level. Therefore, structural changes were made to the same vehicle “a” to realize combination A in Table 1, which has phase relations that increase the acoustic mode. Therefore, noise reduction effects of countermeasures that only reduce the panel vibration level were confirmed.

In this case, the resonance characteristics of the roof were kept the same as those of combination F, but the windshield resonance frequency was changed to 60 Hz, and the rear floor resonance frequency to 45 Hz (Fig. 11). The characteristic values were increased by increasing the rigidity of the panel support structures, and the structure was changed for the vibration level to be equal to that of combination F. Figure 9 (lower level) compares the low-frequency booming noise reduction effects with the windshield and roof phases. Unlike combination F, the

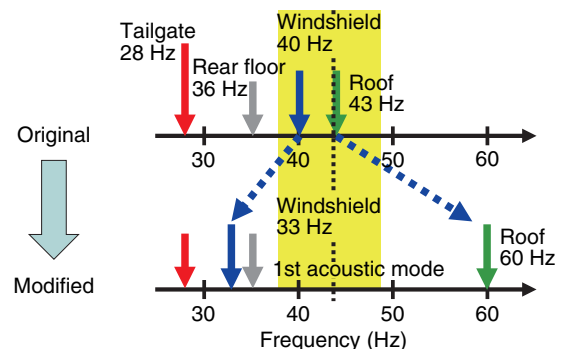


Fig. 8 Modification F of panel resonance

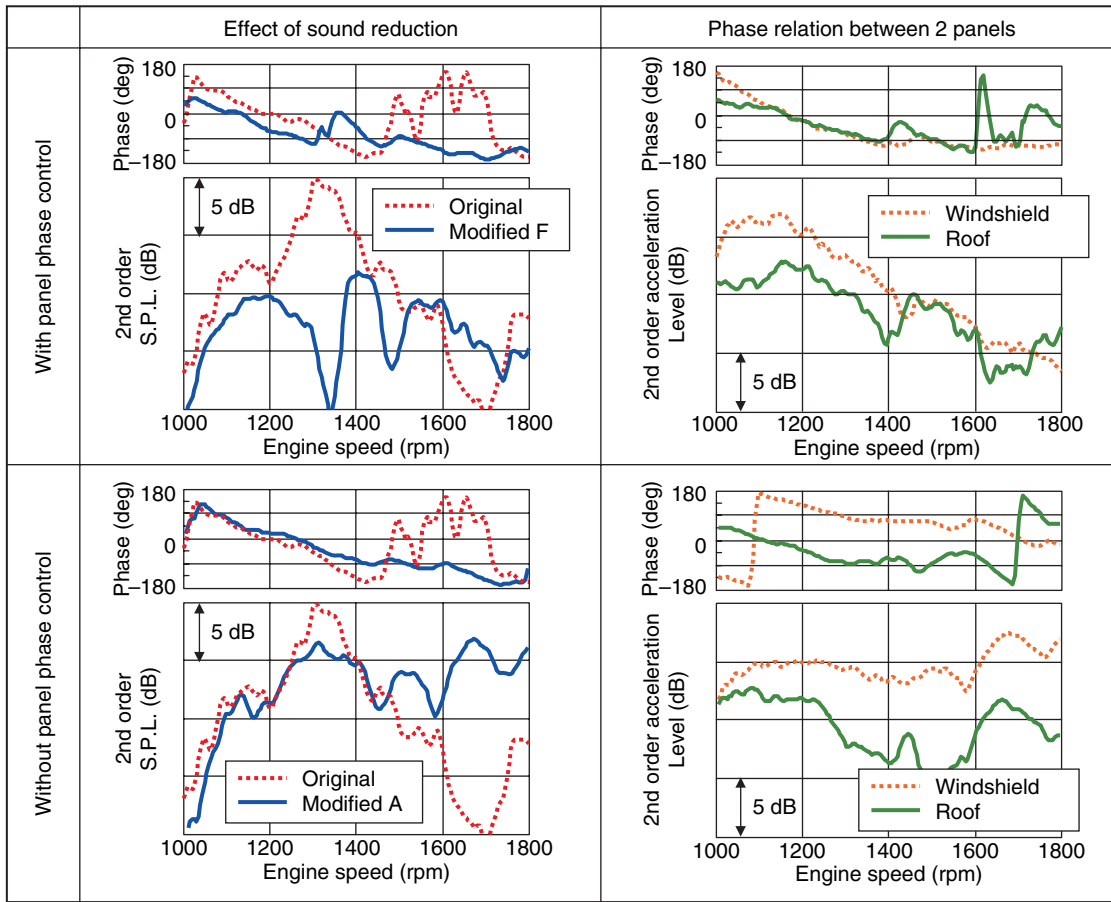


Fig. 9 Result of low frequency booming noise and panel phase relation

windshield and roof are out of phase. This is a phase relation that increases the acoustic response at the 1st mode, so while noise reduction effects of -4 dB were obtained, the effects were less than those for combination F. Both the windshield and roof vibration were reduced to approximately the same vibration level as those for combination F, therefore to obtain additional noise reduction, the vibration level should be reduced even more. However, it is a challenge to realize such a structure. This verified that combination F, which uses phase control, provides superior noise reduction effects as compared to the conventional method of combination A.

Thus, low-frequency noise issues can be quickly addressed by investigating this type of resonance distribution from the initial development stages.

4.2. Drumming Reduction Effects

4.2.1. Coherent component extraction results

Here, the test vehicle used to verify drumming is labeled "b." The input contributions were identified for vehicle "b." First, as the input components for the suspension vibration, the coherent components corresponding to the vibration of the four wheels in three directions were extracted using Eq. (1). In the subject

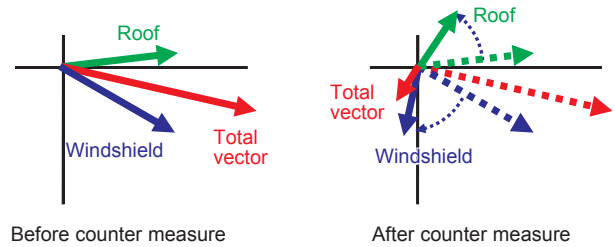


Fig. 10 Panel contribution vector at 43Hz for booming noise

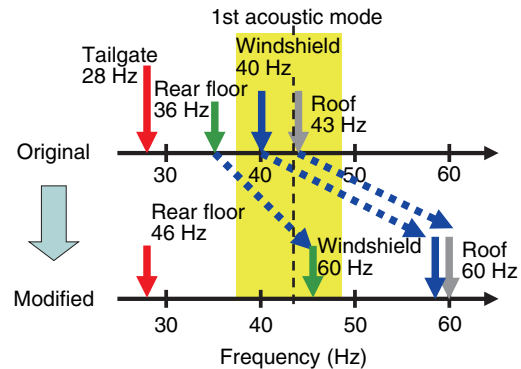


Fig. 11 Modification A of panel resonance

frequency range, the suspension vibration level is higher in the longitudinal direction than in the other directions, so the coherent components were compared for the longitudinal direction. Figure 12 shows the results.

These results show that the input component from the rear wheels has a high contribution in the 40 Hz range. Therefore, the panel vibration velocity components were calculated in the same manner from the measurement results using the longitudinal vibration of the rear wheel suspension as the reference, and applied as the boundary conditions in the BEM model.

Figure 13 compares the sound pressure component of the rear wheel suspension with the BEM model calculation results. Like the analysis results for low-frequency booming noise, an accuracy of within ± 3 dB was obtained. These results show that sufficient accuracy can also be obtained when applying the coherent component method to drumming.

The panel contributions at 43 Hz were confirmed from the BEM analysis results, and the tailgate component was found to be dominant and to have a phase relation that increases the acoustic mode and with the windshield component. Therefore, possible countermeasures would be to first reduce the amplitude of the tailgate vibration, and then to change the phases of these two panels to cancel the acoustic mode.

The panel resonance frequencies for this test vehicle "b" were confirmed to have a distribution as shown in Fig. 14. The distinguishing features are that the roof curvature is small so that the resonance frequency is high, and the tailgate resonance frequency is also high. The

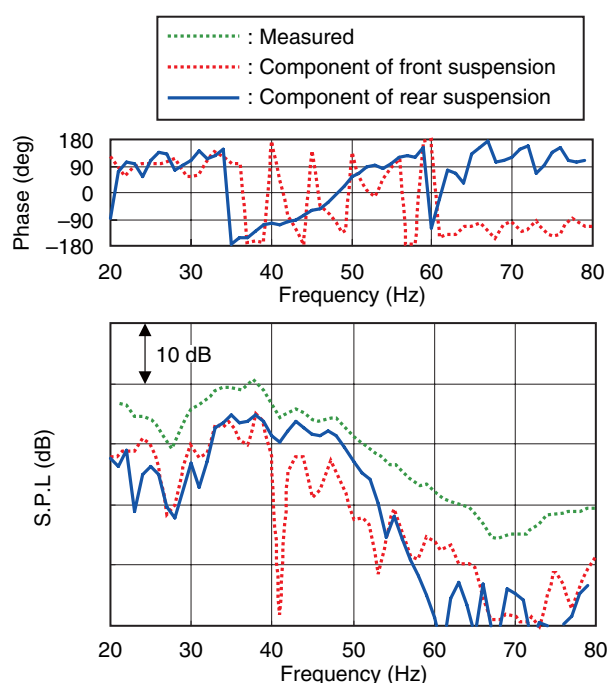


Fig. 12 Result of calculated coherent component

characteristic value of the tailgate is closer to the acoustic 1st mode frequency than that of the roof panel, so the tailgate is predicted to have a greater effect. Therefore, a combination corresponding to A in Table 1 was judged to provide significant noise reduction effects. That is to say, setting the windshield resonance frequency higher and the tailgate resonance frequency lower than those of the acoustic mode was predicted to cancel the acoustic mode.

In test vehicle "b", the windshield resonance frequency was raised by 5 Hz by countermeasures to increase the rigidity of the front roof rail, and the resonance frequency of the tailgate was lowered by 5 Hz by reducing the spring force of the tailgate stopper. As a result, the panel contributions changed as shown in Fig. 15, and the two panels changed to phases that cancelled the acoustic mode as expected. This phase relation provided reduction effects of -6 dB in the 40 Hz range as shown in Fig. 16.

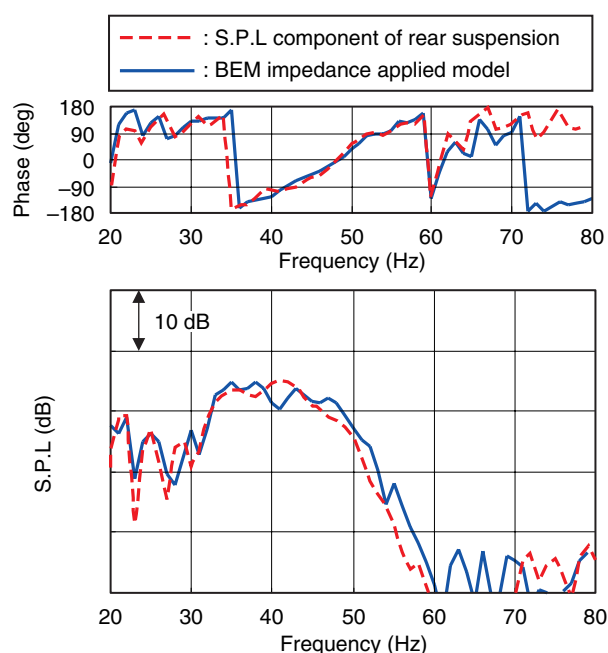


Fig. 13 Compared with S.P.L. measured and calculated

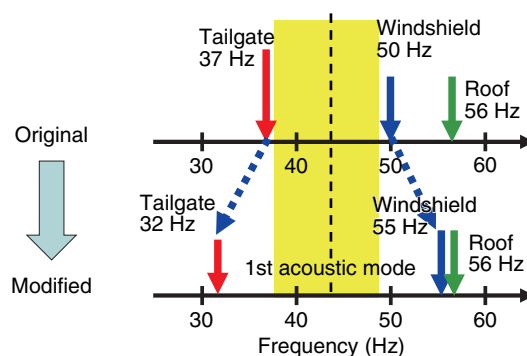


Fig. 14 Modification of panel resonance for drumming

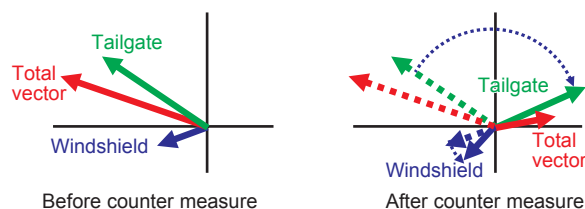


Fig. 15 Panel contribution vector at 43Hz for drumming

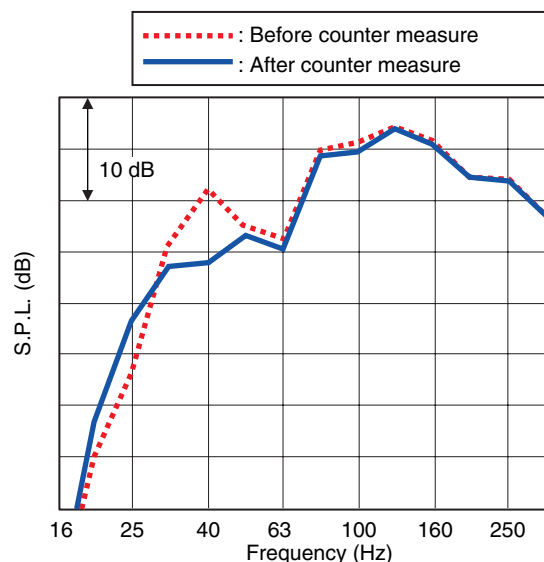


Fig. 16 Effect of panel phase control for drumming

5. Conclusion

The following points were clarified with respect to low-frequency noise, such as low-frequency booming noise and drumming, in station wagons and minivans.

- (1) It was shown that the panel phase characteristics can be controlled by the panel resonance frequency, and resulted in sound pressure reduction inside the cabin through cancellation by appropriately distributing the resonance frequencies of multiple panels.
- (2) The ability to reduce the sound pressure inside the cabin by controlling the panel phase was verified by making actual structural changes to completed vehicles; the effectiveness of these countermeasures was then confirmed.
- (3) Sound pressure was accurately calculated from the panel vibration measurement results using the boundary element method (BEM); the panel phase relations and contributions to sound pressure were then clarified.
- (4) The coherent component method was applied not only to low-frequency booming noise with a single input source, but also to the drumming phenomenon, which has multiple uncorrelated input sources. In both cases, analysis accuracy sufficient for practical application was obtained.

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Study of Door Closing Sound Mechanism Using Operational Modal Analysis and Acoustic Holography

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ABSTRACT

In order to keep the boundary conditions of door closings, operational modal analysis (OMA) was used to obtain the correlation between the operational deflection shape (ODS) and mode of vibration. As a result, it was possible to understand the time domain deflection shape. In addition, to verify the relationship between the deflection shape and phenomenon of noise radiation, the acoustical radiation phenomenon was determined using time domain acoustical holography. With this conventional method, there were limits to the measurement of low frequencies, so in this research a new method called statistical optimal near-field acoustic holography was used. As a result, it was possible to determine a correlation between the vibration phenomenon and phenomenon of noise radiation in transient conditions. A method of changing the door closing sound was developed by controlling the vibration flow.

1. Introduction

In recent years, the door closing sound (hereafter “closing sound”) has become a popular vehicle feature. However, the demand for lighter vehicles with reduced fuel consumption has also increased, so using materials to enhance the closing sound is a challenge.

In this study, a mechanism for generating a transient closing sound in the time domain (hereafter “time domain closing sound”) was researched to create a structure that can enhance the closing sound without adding any materials to current mass produced doors.

2. Closing Sound Elements

As shown in Fig. 1, the closing sound elements of low, medium and high frequencies closely correlate to the door outside panel, door seal and door lock respectively.

The sounds of the door seal (medium frequencies) and door lock (high frequencies) are generated by objects contacting against each other, and measures can be taken for the seal cross section and internal door lock structure. On the other hand, certain impact with the outside panel can cause the door outside panel to vibrate which generates door outside panel sounds. To analyze these phenomena, modal analysis which examines the eigenvalues of the door outside panel can be performed.

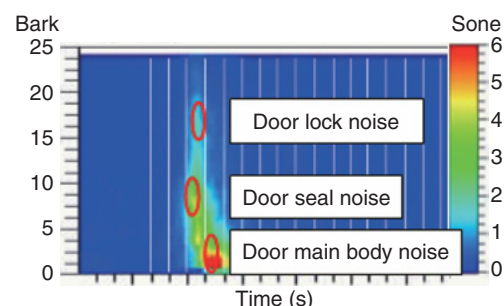


Fig. 1 Door closing sound element

In order to research the door closing sound mechanism, which is a transient phenomenon, the input and response processes should also be examined. For this reason, as described in this paper, OMA, time domain ODS and time domain acoustic holography were used for the analysis.

3. Analysis of Vibration Behavior in the Transient Condition

Generally, modal analysis is effective in performing vibration analysis of the door outside panel. Modal analysis is a technology that is used to estimate the mode model by extracting the mode parameters from the Frequency Response Function (FRF), which is obtained

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from the target structure. In FRF measurement, the target object is freely supported, and a tool such as a vibration exciter (shaker) is used to measure the input and output signals for the target object.

Such a method presents a challenge because the boundary conditions of the analysis differ from those of the actually used doors and the vibration behavior in a transient condition cannot be ascertained.

As a method to address these issues, OMA and time domain ODS were used to analyze vibration behavior in the transient condition.

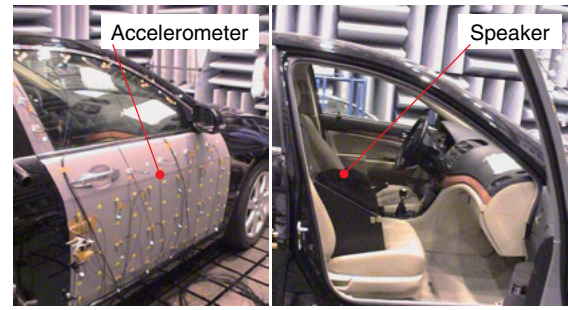


Fig. 2 Operational modal analysis

3.1. OMA

OMA (Brüel & Kjær, Co. Ltd.)⁽¹⁾ is a technology where the modal parameters are extracted only from the vibration response signal of the target object, and the vibration excitation does not need to be measured. However, since the target needs to be excited, acoustic excitation was performed for the door by radiating random signals from speakers positioned inside the vehicle, and the vibration was measured by using accelerometers at 36 points, which were installed on the door outside panel and sash (Fig. 2).

3.2. Results of OMA

As a result of the closing sound element analysis, maximum frequency in modal analysis was set as 200 Hz. The modal analysis was performed for the door outside panel, and it was able to estimate the modal parameters to a higher degree of accuracy, as shown by the mode shape MAC (SSI method of OMA)⁽¹⁾ in Fig. 3.

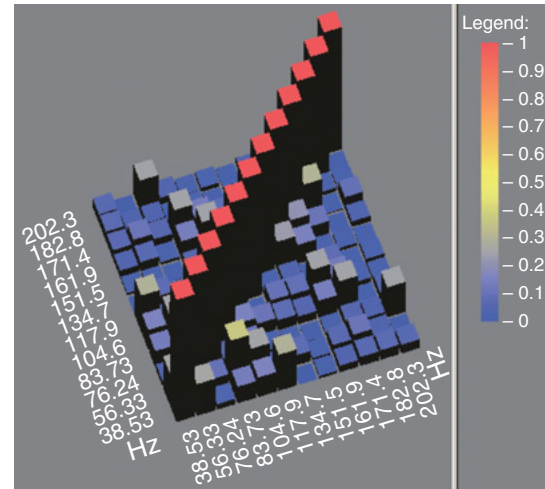


Fig. 3 MAC of door mode shape

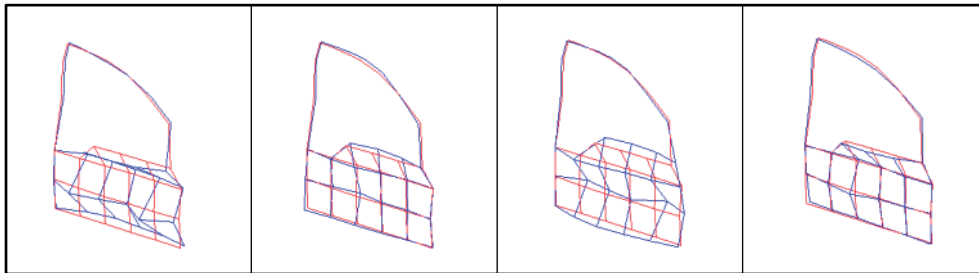


Fig. 4 Time ODS around 135 Hz

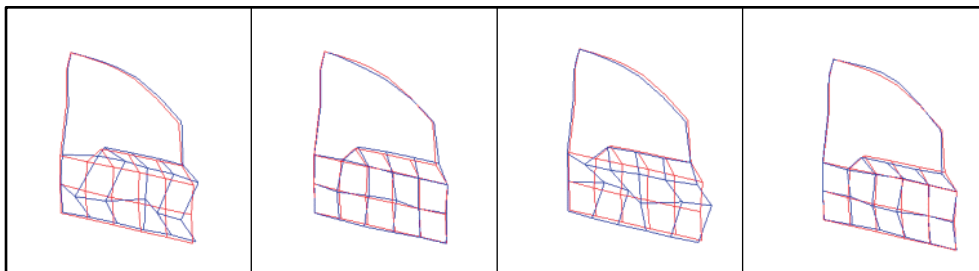


Fig. 5 Modal shape of 7th mode, 135 Hz

3.3. Time Domain ODS

ODS (Brüel & Kjær Co., Ltd.) is a technology that uses multiple sensors and simple signal processing (cross spectral measurement that obtains the amplitude ratio and phase difference or simultaneous time history recording) to express the vibration behavior of the target structure in the operational condition with moving images (of specified frequencies or time domain signals)⁽²⁾.

3.4. Results of ODS Transient Vibration Behavior Validation

To examine the relationship between the vibration mode estimated with OMA and the vibration behavior of the time domain, the mode shapes were compared with the ODS generated from signals filtered around the mode frequency. Figures 4 and 5 show examples.

Figure 4 shows one cycle of time domain ODS, based on a band path filtered signal with a width of 10 Hz around 135 Hz. Figure 5 shows mode shapes of the seventh mode (135 Hz), as estimated by OMA. The loops and nodes of vibration in both examples correlated well, so it was possible to perform modal analysis in the door closing boundary conditions.

In addition, as shown in Fig. 6, it was understood that the transient vibration behavior (which cannot be understood from OMA), is as follows: the impact generated on the top of the door sash at the moment the

door is closed is the vibration source, which is propagated to the door outside panel and excites the outside panel mode.

4. Validation of the Sound Radiation Phenomena

The transient vibration behavior could be ascertained from the ODS analysis results, but to examine the relationship between the vibration and sound, the sound radiation phenomena was also needed to be checked.

Therefore, time domain acoustic holography (Brüel & Kjær Co., Ltd., statistical optimal near-field acoustic holography) was used to determine the condition of the sound radiation for the door outside panel.

4.1. Issues Regarding Time Domain Acoustic Holography

Conventional near-field acoustic holography (NAH) is based, as a rule, on 2D discrete Fourier transformation. When using holography measurement for low frequency acoustic phenomena, all of the acoustic phenomena may not fit into the array size (spatial window) for the 2D discrete Fourier transformation, which is determined with a finite length. If this happens, by repeating the discrete Fourier transformation and reverse transformation for the spatial frequency domain, any discontinuities that appear

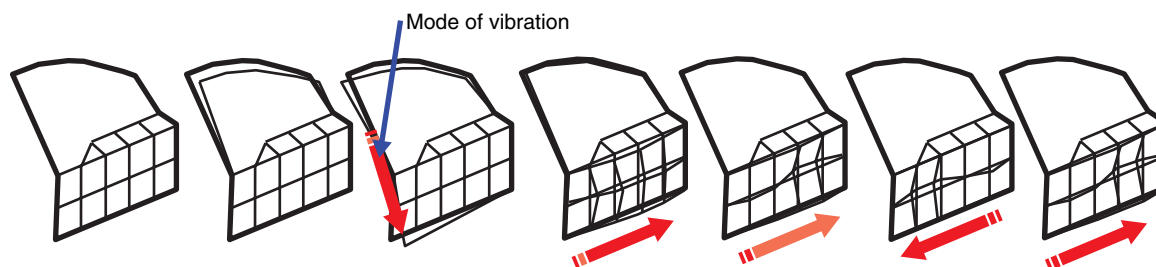


Fig. 6 Operational deflection shape

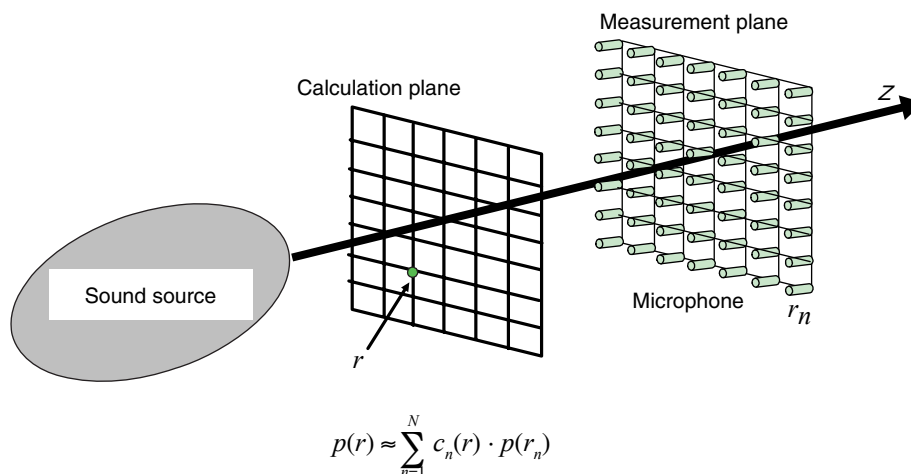


Fig. 7 Basics of statistical optimal near-field acoustic holography

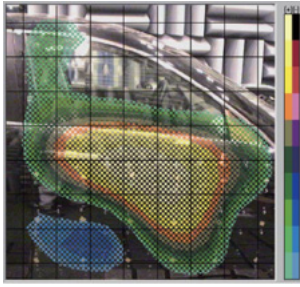


Fig. 8 Near-field acoustic holography (135 Hz)

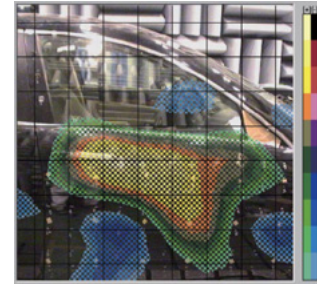


Fig. 9 Statistical optimal near-field acoustic holography (135 Hz)

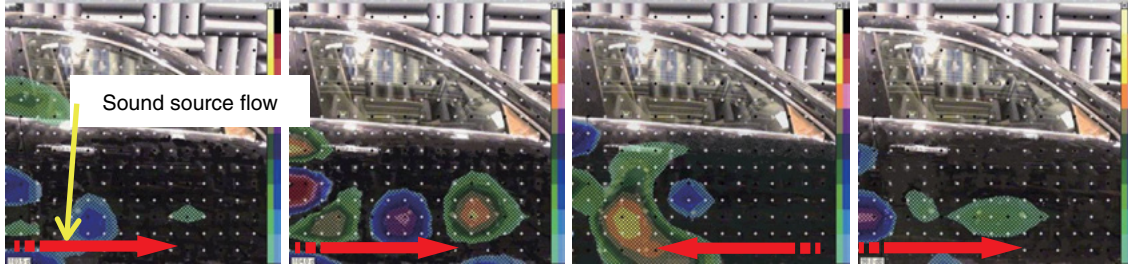


Fig. 10 Near-field acoustic holography of door

at the end of the spatial window will cause a ghost phenomenon in the analysis results⁽³⁾. Similar phenomena are seen with the window function when using the discrete Fourier transformation for time signals. For this reason, a large size of microphone array is necessary for the low-order mode frequency of the door outside panel that was previously estimated. Using the low-order mode frequency is not practical.

4.2. Statistical Optimal Near-field Acoustic Holography (SONAH)

SONAH is a modified form of NAH. In this method, a direct calculation is performed from sound pressure on an arbitrarily positioned measurement plane to an arbitrary calculation plane, without using 2D discrete Fourier transformation (Fig. 7). The time-harmonic sound pressure $p(r)$ at the arbitrary calculation point r can be expressed as a linear combination of complex, measured sound pressure $p(r_n)$, and the complex coefficient $C_n(r)$ can be statistically estimated with the least squares method⁽⁴⁾. In this case, the complex coefficient $C_n(r)$ is the specific information used to express the sound wave plane, and here is used for moving the Z axis.

The advantage of SONAH is the few restrictions (array size, microphone position) for the holography sensor and it can be used for small-sized arrays.

4.3. Validation of Low Frequency Limit

The results of active intensity with the conventional NAH (Fig. 8) method and SONAH (Fig. 9) for the door outside panel seventh mode frequency (135 Hz) showed that the large amplitude sections and positive/negative boundaries of the active intensity in the same time domain

match with the positions of the loops and nodes of the corresponding vibration modes (Fig. 5). In addition, it can be understood that the relationship between the positions of the loops and nodes of the vibration mode are more clearly shown (resolution quality) in the SONAH results (Fig. 9) than with the conventional method (Fig. 8).

4.4. Results of Sound Radiation Phenomena Validation

By using a SONAH holography calculation, it was confirmed in the same way as with the ODS animation that a low frequency sound source (30 Hz to 200 Hz) was generated at the rear end of the door outside panel, which moved on the outside panel as time elapsed (Fig. 10).

In addition, as shown by Fig. 11, it was understood from the SONAH results that the positional relationship and direction for the vibration phenomena matched.

From these results, it was discovered that transient vibration phenomena of the door outside panel are radiated as sound.

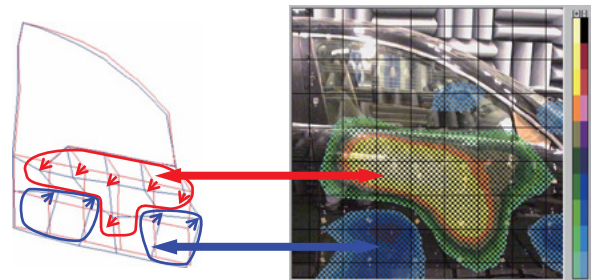


Fig. 11 Time ODS and near-field acoustic holography (135 Hz)

5. Enhancement Method for Low Frequency Elements

It was shown in the previous test that the vibration behavior of the door outside panel is radiated as sound. Consequently, methods for controlling the vibration flow were examined to control the tone of the low frequency elements.

When the door outside panel mode, which has many low frequency elements, was analyzed as sound pressure, it was understood that the first vibration mode of the door unit was 40 Hz or less. It was thought that lowering the first vibration mode and controlling the propagation of vibration would be a good way to enhance the low frequency elements.

Therefore, the doors of mass produced vehicles (hereafter “base doors”) were used to study control methods for vibration propagation.

5.1. Structure Study

Computer-aided engineering (CAE) was used to study a structure that could lower the first vibration mode.

This time, items such as materials with a vibration suppression effect for the door outside panel were removed from the base door study, and whether the first vibration mode could be changed by changing the supporting structure of the door outside panel was examined.

As a result, a supporting structure for the outside panel was created that lowered the first vibration mode.

5.2. Validation of the Effectiveness of the Door Outside Panel Supporting Structure Change

A door with a changed outside panel supporting structure was constructed, and modal analysis was performed. The first vibration mode was lowered as shown in Fig. 12.

The ODS results also confirmed that front-to-rear vibration flow could be controlled in the door by changing the outside panel supporting structure, as

Table 1 Sound pressure level (–200 Hz)

	SPL (dB)
Original door	96.0 dB
Modified door	99.2 dB

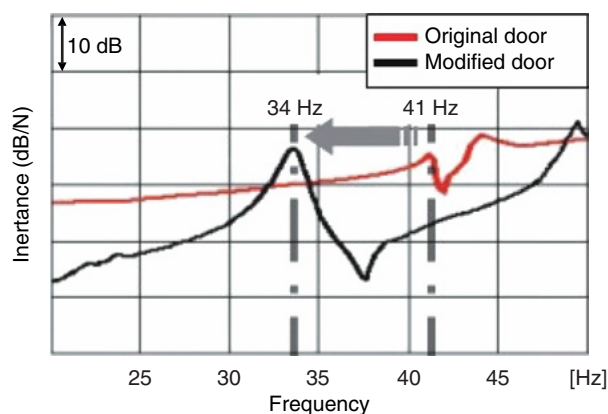


Fig. 12 Modal analysis of door

shown in Fig. 13, by comparing the door before and after the change (Fig. 6).

5.3. Validation of Sound of Door Outside Panel Supporting Structure

When the effectiveness of the closing sound was checked in an actual vehicle, it was confirmed, through the subjective evaluation of experts and general users, that the door with the changed outside panel supporting structure was effective in enhancing the low frequency sound elements⁽⁵⁾.

In addition, as shown by Table 1, it was understood that the low frequency (–200 Hz) sound pressure increased by more than 3 dB.

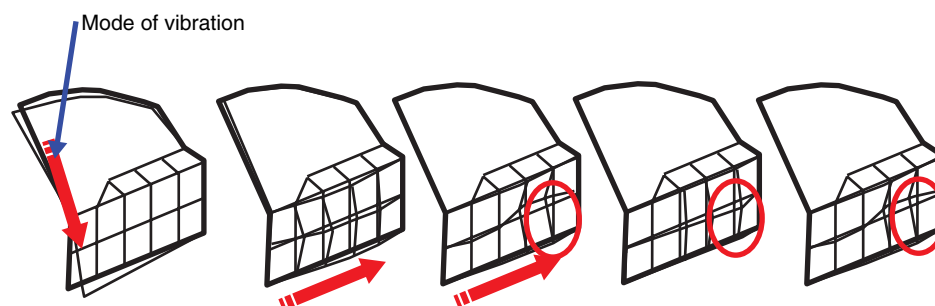


Fig. 13 Operational deflection shape

6. Conclusion

By examining door closing sound mechanisms, this study came to the following conclusions.

- (1) It was discovered by using OMA, time domain ODS and time domain acoustic holography that for low frequency elements, the vibration generated from the door closing impact is propagated to the door outside panel, which excites the vibration mode and radiates sound.
- (2) In the vibration modal analysis for the door outside panel, acoustic excitation OMA was effectively performed while maintaining the boundary conditions when closing the door.
- (3) As a method for ascertaining the acoustic phenomena of the door closing sound, SONAH time domain acoustic holography was used to clearly ascertain door acoustic phenomena in the low-order mode frequencies.
- (4) The low frequency elements of doors in mass produced vehicles were enhanced by changing the outside panel supporting structure.

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Junichi KIDACHI

Establishment of Method for Optimizing Multi Specifications for Sound-proof Packages

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ABSTRACT

This paper describes an optimization method for designing a vehicle sound-proof package. This method enables the achievement of an optimum balance between lower weight and higher absorption performance. In order to propose a sound-proof package in the early stages of vehicle development, it is necessary to assess the number of sound-proof elements required and to select the optimum design specifications for weight and absorption performance. However, it is difficult to achieve conflicting objectives simultaneously, and acoustic engineers therefore require special technical knowledge. The research discussed here developed a new automated optimization method that resolves these issues. The new method enables detailed sound-proof package specifications to be obtained, including the thickness distribution for each part. These specifications can be easily transferred to drawings. The method was successfully used to reduce the sound pressure level of a vehicle cabin, providing verification of its accuracy.

1. Introduction

Increasing the quietness of automotive cabins is an important element in enhancing commercial performance. The achievement of weight savings is also an indispensable element in ensuring fuel efficiency. However, in many cases increased quietness (enhanced acoustic performance) and the attainment of weight savings are contradictory objectives.

It is necessary to determine specifications from the initial stages of a development project in order to develop efficient sound-proof packages^{(1), (2)}. The flow of the process of determining specifications is as follows:

- (1) Determine the targets for in-cabin sound pressure level.
- (2) Determine the targets of individual vehicle components for achieving the overall in-cabin sound pressure target using SEA parameters of damping loss factor (DLF) and coupling loss factor (CLF).
- (3) Determine the targets of each sound-proof component based on absorption coefficient and transmission loss for achieving the targets of individual vehicle components.
- (4) Establish the material, thickness, and area coverage for each layer of each component in the sound-proof package for achieving targets.

Techniques are already available using the SEA method for items (1)-(3)^{(1), (2)}. However, a number of parameters, including acoustic material choice, thickness and application area, should be considered in order to decide on the specifications in item (4), and it is therefore challenging to determine specifications in a short period of time.

The purpose of the study discussed in this paper was to propose a method for optimizing multi-objective specifications for sound-proof packages that enables optimum specifications satisfying in-cabin sound pressure level targets. In addition, for verification, a comparison is made between the near-optimal and optimal results proposed by this method. In the result, the essence of this optimization technique can be verified by acoustic performance.

2. Analysis Method

Sound-proof package used in automotive applications is composed of layers of several different materials. Figure 1 shows an example of this structure, which will be referred to as a “multi-layered” structure in this paper.

The acoustic performances (absorption coefficient,

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transmission loss) are determined by the material specifications of each layer of the multi-layered structure, the thickness of these materials, and their surface area. These items are referred to as the sound-proof package specifications.

Figure 2 shows the flow of the process of designing sound-proof package specifications to satisfy in-cabin sound pressure level targets. This process is discussed in detail below.

(1) Specifying material and equivalent thickness in element

Limitations are placed on the potential thickness of the acoustic materials by other aspects of the layout, including the shape of the body panels, the air-conditioning ducts and the harnesses. Based on these restrictions, the acoustic materials are divided into a number of elements. Each layer of the multi-layered structure of each element is treated as being formed from a uniform material, and the layers are treated as being of equivalent thickness. Specifications for materials and the thickness of materials that satisfy the in-cabin sound pressure level targets are determined for each element. At the same time, the

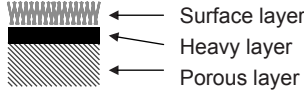


Fig. 1 Multi-layer sound-proof package

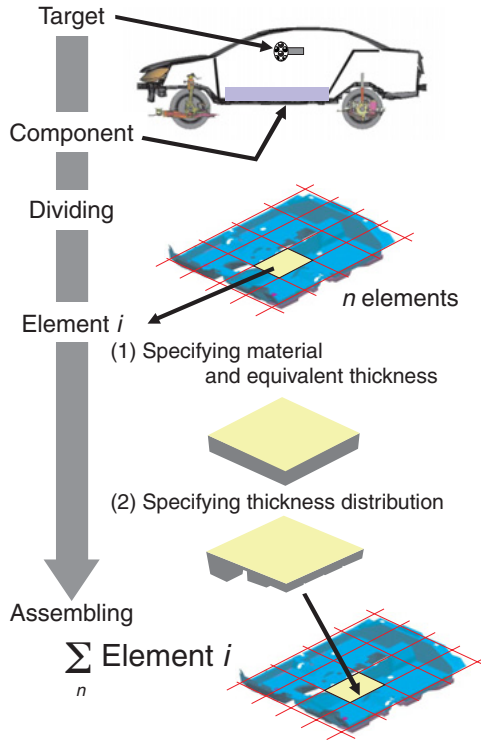


Fig. 2 Procedure for design of optimized sound-proof package

acoustic characteristics (absorption coefficient, $\bar{\alpha}_{if}$, and transmission loss, $\overline{TL}_{if}$) are calculated. Here, f is the center frequency of the 1/3 octave band.

(2) Specifying thickness distribution in element

As Fig. 3 shows, there is a range of variation in the thickness of the acoustic materials that are used in mass-production vehicles from thick to thin. This is termed the thickness distribution.

Because of the restrictions imposed by the shape of the panels and other aspects of the layout, even the elements into which the materials have been divided for optimization of specifications are unable to be assigned a constant thickness. Therefore, specifications for material thickness that reflect the design layout are determined, while maintaining the acoustic characteristics of the elements with materials of equivalent thickness divided in the study conducted in step (1). The relationship between the acoustic characteristics of the elements with materials of equivalent thickness and the elements with a thickness distribution reflecting the design layout is given by Eqs. (1) and (2).

$$\bar{\alpha}_{if} = \frac{\sum_j^n A_{ij} \cdot \alpha_{jf}}{A_i} \quad (1)$$

$$\overline{TL}_{if} = 10 \times \log \left(\frac{1}{\bar{\tau}_{if}} \right)$$

$$\bar{\tau}_{if} = \frac{\sum_i^n A_{ij} \cdot \tau_{jf}}{A_i}, \quad TL_{if} = 10 \times \log \left(\frac{1}{\tau_{if}} \right) \quad (2)$$

Here, A_i is the surface area of the element i , A_{ij} is the surface area of the material of constant thickness j in element i , α_j is the absorption coefficient, and τ_j is the transmission loss.

The method of optimizing the materials and the thickness distribution in each element is discussed below.

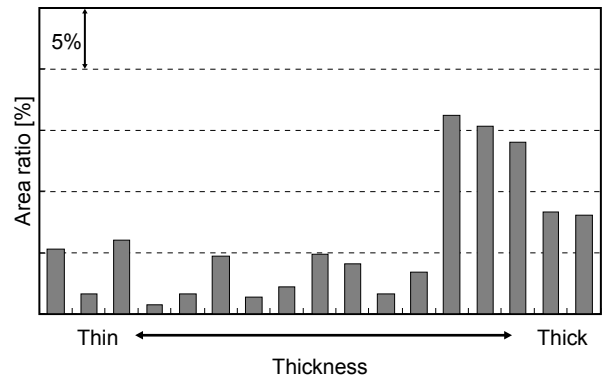


Fig. 3 Thickness distribution of mass produced floor carpet

2.1. Element Material and Equivalent Thickness Optimization

This section will discuss the flow of the procedure employed to optimize the specifications of the elements of the sound-proof package and the algorithm used in this procedure.

2.1.1. The optimization process

Figure 4 shows the process flow for calculating the optimum specifications for the elements of the sound-proof package. The details of each procedure can be summarized as follows:

- (1) The sound-proof package is divided into n elements, and an analytic SEA model is constructed treating the material used in each layer of the multi-layered structure of each element and equivalent thickness of each layer as design variables.
- (2) For each element, the thickness of the overall structure, the equivalent thickness of each layer, the analytic frequencies, and constraints for in-cabin sound pressure level are set.
- (3) The total weight of the sound-proof package and in-cabin sound pressure level are set as objective functions.
- (4) Optimum specifications are searched under the conditions described above.
- (5) Optimum acoustic performance (absorption coefficient, $\bar{\alpha}_{ij}$, and transmission loss, $\bar{T}L_{ij}$) are calculated⁽³⁾ for each satisfactory element determined in step (4).

2.1.2. Algorithm for optimization

A non-dominated sorting genetic algorithm (NSGAI)⁽⁵⁾ is used to search optimum design variables that satisfy the multiple objective functions (total weight of sound-proof package and in-cabin sound pressure level) set in step (3) above. However, because there are numerous design variables and an extensive search time is required, the variables are restricted in two stages. The methods

employed are discussed below⁽⁶⁾.

(1) Grouping

Elements possessing an identical multi-layered structure are grouped, and the design variables are restricted.

(2) Examination of objective function sensitivity to design variables

Using the orthogonal experimental method, the equivalent thickness of the layers in the multi-layered structures of the elements (which represent design variables) are converted to three levels; the design variable with the lowest sensitivity to in-cabin sound pressure level is used as an initial value, and design variables are restricted.

2.2. Element Thickness Distribution Optimization

This section will discuss the optimization process for thickness distribution of the elements of the sound-proof package and the algorithm employed in the process.

2.2.1. The optimization process

Figure 5 shows the flow of the process of searching optimum results for the thickness distribution of the elements. The following procedures are used:

- (1) The optimum acoustic characteristics of element i (absorption coefficient, $\bar{\alpha}_{ij}$, and transmission loss, $\bar{T}L_{ij}$) are set as objective functions.
- (2) The surface area of element i is termed A_i , and the surface area of the material of fixed thickness j in element i is termed A_{ij} . The ratio of each of these fixed thicknesses is expressed as A_{ij}/A_i , and is set as a design variable. Variations in this ratio of surface areas are treated as the thickness distribution.
- (3) Under these conditions, the optimum thickness distribution that achieves the optimum acoustic performance at the lowest weight is searched.

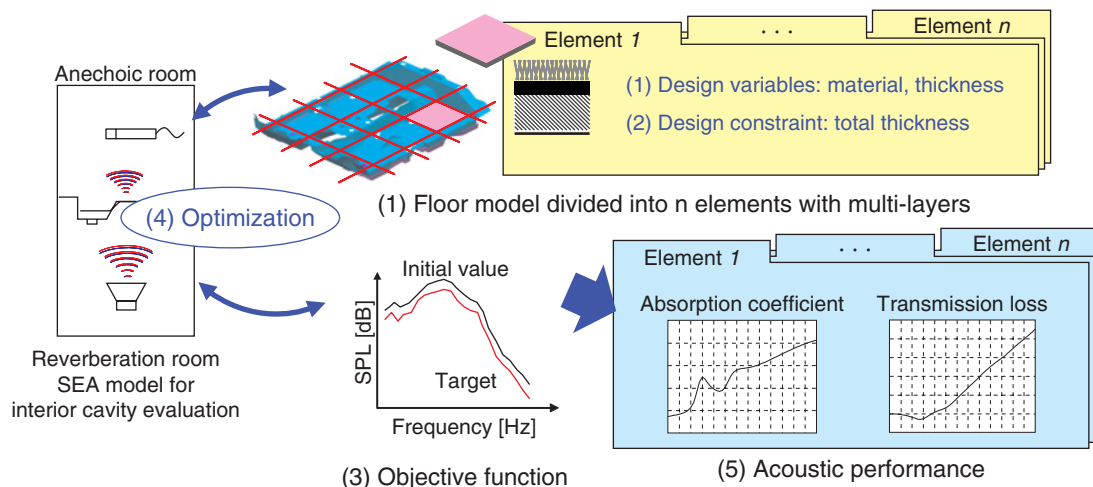


Fig. 4 Optimization process for design variables in elements

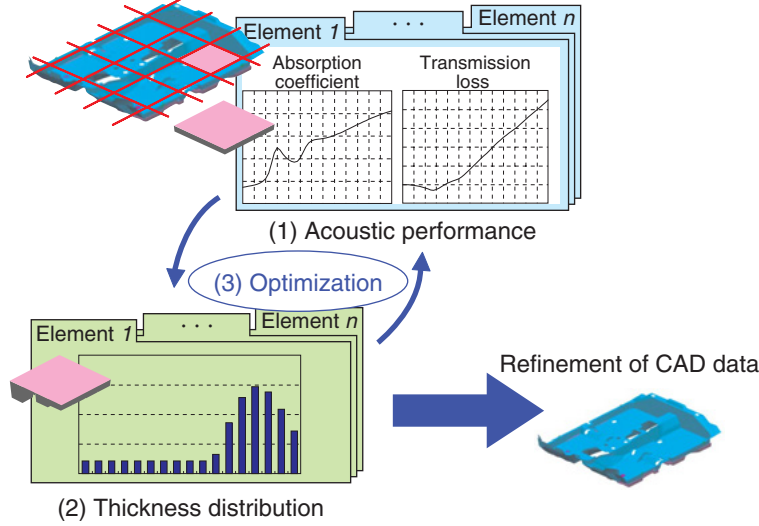


Fig. 5 Optimization process for thickness distribution

2.2.2. Algorithm for optimization

It is desirable that an optimum thickness distribution is realized in an actual vehicle, and that the search time is short. To satisfy these requirements, the three items discussed below are incorporated into the search algorithm.

(1) Expression by function of thickness distribution

As Fig. 3 shows, the thickness distribution of divided elements in mass-production vehicles is affected by the shape of the panels and other layout constraints. The thickness distribution tends to describe a smooth curve centering on a specific thickness. In the proposed method, the thickness distribution A_{ij}/A_i , which enables this tendency to be reproduced, is expressed as a function like Eq. (3).

$$A_{ij}/A_i = K_1 \cos(k_1 \times j) + L_1 \sin(l_1 \times j) + K_2 \cos(k_2 \times j) + L_2 \sin(l_2 \times j) + M \quad (3)$$

Here, the factors K_1 , k_1 , L_1 , l_1 , K_2 , k_2 , L_2 , l_2 and M in Eq. (3) are variables enabling determination of A_{ij}/A_i .

(2) Restriction in thickness distribution

Certain thicknesses, j , can generate surface area ratios that vary considerably from others in the thickness distribution A_{ij}/A_i expressed as a function for element i . These surface area ratios are issues to realize in the actual sound-proof package. To ensure that surface area ratios do not vary widely, constraints are placed on the surface area ratio of thickness j and the surface area ratios of the contiguous thicknesses $j-1$ and $j+1$. As an example, Fig. 6 shows constraints of 5% and 10% applied to $j-1$ and $j+1$ respectively. This ensures that the search for a thickness distribution is able to generate a result that can be realized in practice.

(3) Two-step optimization process

The absorption coefficient and transmission loss,

treated as objective functions, are independent indexes with no correlation. Because of this, there is the possibility that the algorithm employed will search for a local optimal result close to the initial value and ignore global optimal results.

In this project, the use of an algorithm combining NSGAII for global optimization, and sequential quadratic

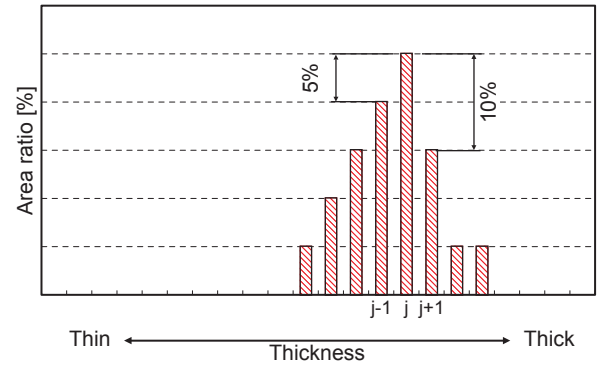


Fig. 6 Constraint for thickness distribution

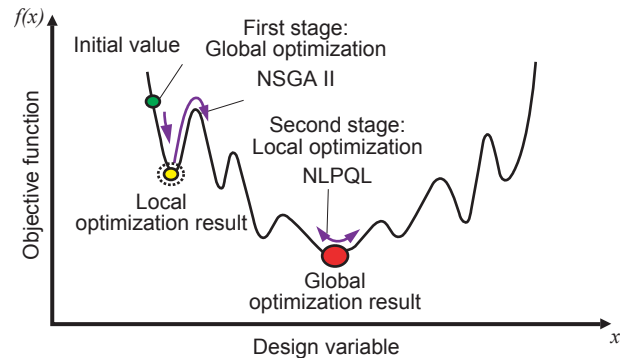


Fig. 7 Results of local and global optimization

Table 1 Optimization variables for elements

Objective component	Floor carpet divided along with floor panel shape into 50 elements
Design variables	Each layer thickness in multi-layer
Constraints	Total thickness of multi-layer in each part
Objective functions	Interior cavity sound pressure level in O.A. and high frequency range 1.25-5 kHz

Table 2 Target for floor optimization in interior cavity sound pressure level and floor carpet total weight

	High-frequency range 1.25-5 kHz	O.A.	Total weight
Target	1.0 dB reduction	1.0 dB reduction	Lighter than initial weight

programming (NLPQL)⁽⁴⁾ for local optimization, accomplished the method to search for global optimal results as shown in Fig. 7.

3. Analysis Results

The floor section was adopted as an example, and optimum specifications were searched for the sound-proof package using the proposed methods of optimizing materials specifications and thickness distribution. The details of each process are discussed below.

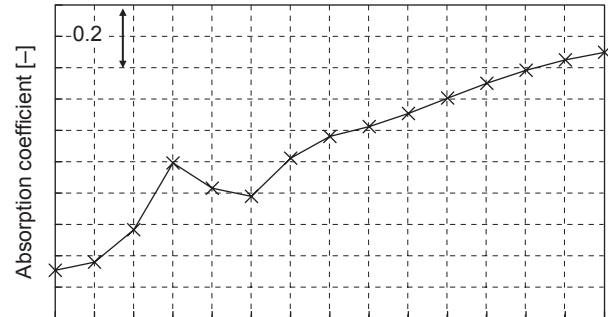
3.1. Element Material and Equivalent Thickness Optimization

A target was set for the effect on in-cabin sound pressure level to be achieved as a result of measures applied on the floor section, and materials specifications were searched for the floor elements, enabling this target to be satisfied. Table 1 shows the variables established for optimization of materials specifications.

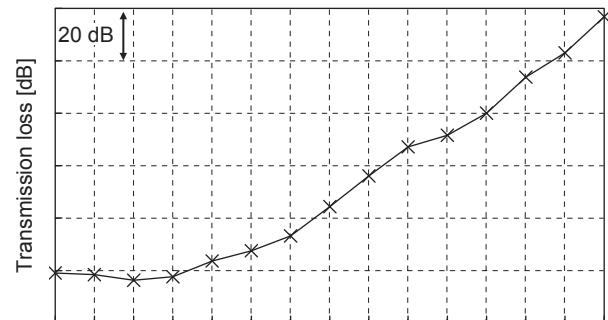
Table 2 shows the targets for the optimum specifications.

As Fig. 8 shows, using grouping and examination of objective function sensitivity, the approximately 100 design variables established in Table 1 are reduced to approximately 20.

Next, using the reduced design variables, the specifications of the elements that satisfied the conditions were optimized. An example of the optimum specifications determined by the method is shown in Fig. 8. The absorption coefficient, $\bar{\alpha}_f$, and transmission loss, $\overline{TL}_f$, of the optimum package are shown in Fig. 9.



(a)



(b)

Fig. 9 Sound-proof performance of optimal sound package: (a) absorption coefficient and (b) transmission loss

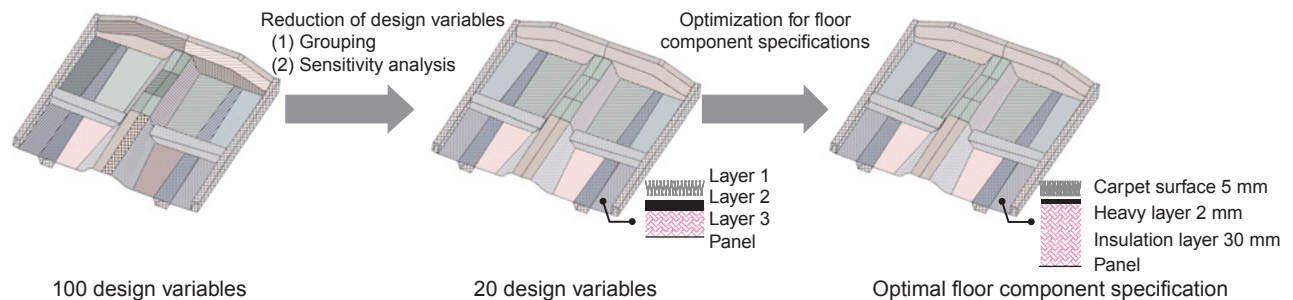


Fig. 8 Reduction of design variables and optimal result

Table 3 Optimization variables for thickness distribution

Design variables	Thickened in 2 mm pitch from 7 mm to 41 mm in an element
Constraints	Coefficients in Eq.(3) restrained as follows; $-10000.0 \leq K_1, L_1, K_2, L_2 \leq 10000.0$ $-10000 \leq k_1, l_1, k_2, l_2 \leq 10000$ $0 \leq M \leq 10000$
Objective functions	Absorption coefficient and transmission loss of the element

3.2. Element Thickness Distribution Optimization

An additional examination was conducted to determine a more attainable thickness distribution while maintaining the optimal acoustic performance (absorption coefficient, $\bar{\alpha}_{if}$, and transmission loss, $\overline{TL}_{if}$) of the elements as shown in Fig. 9.

Table 3 shows the variables established for the search for optimal results for the thickness distribution of the elements.

Under these conditions, a search was conducted for an optimum thickness distribution for the elements. A thickness distribution was determined that both maintained optimum acoustic performances and could be practically implemented on the vehicle. These results for optimum thickness distribution are shown in Fig. 10.

4. Verification of the Technique

The appropriateness of the proposed optimization process is verified with the optimized acoustic package. Moreover, applying the optimum specifications determined for the floor carpet to an actual vehicle, the majority of the proposed optimization process is verified.

4.1. Element Material and Equivalent Thickness Optimization

Optimum specifications were searched for using design variables that had been restricted in two steps. The $\times$ marks in Fig. 11(a) show the relationship between in-cabin sound pressure level in high frequency range and

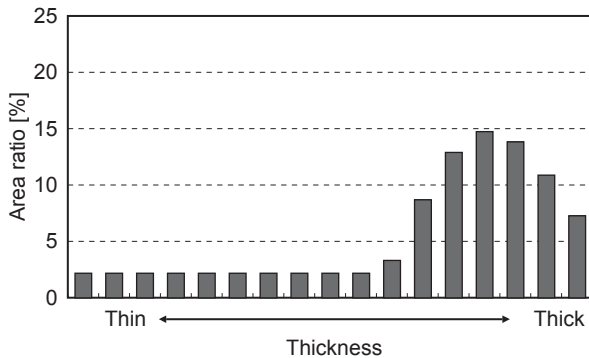


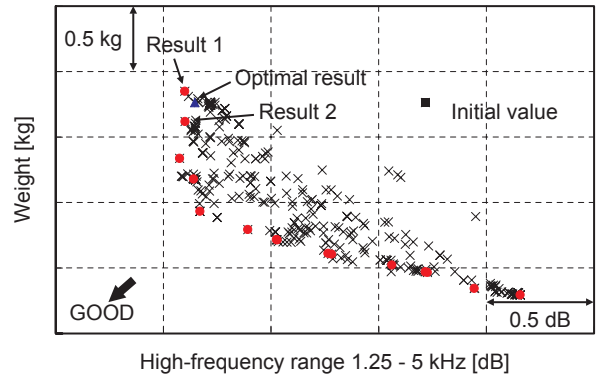
Fig. 10 Optimal thickness distribution for floor element

Table 4 Comparison between optimization results

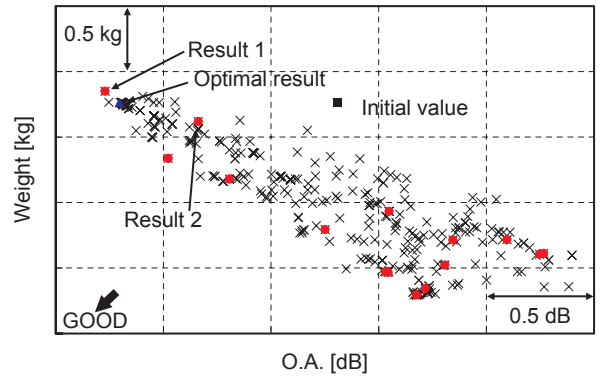
	High-frequency range 1.25-5 kHz	O.A.	Total weight
Target	1.0 dB reduction	1.0 dB reduction	Lighter than initial weight
Optimal result	Achieved	Achieved	Achieved
Result 1	Achieved	Achieved	Not achieved
Result 2	Achieved	Not achieved	Achieved

the weight of the floor carpet for the distributions determined in this process. The results for distribution (pareto solution) of in-cabin sound pressure level in high-frequency range by the lightest carpet weight are shown with circle marks, and the optimum specification by this method is shown with triangle mark. The relationship between the O.A. in-cabin sound pressure level and the weight of the floor carpet at this time is shown with triangle marks in Fig. 11(b).

Table 4 shows a comparison of the relationship



(a)



(b)

Fig. 11 Distribution of samples during optimization: (a) high-frequency range versus weight (b) O.A. versus weight

between the optimal result and target, and the relationship between near-optimal results and the target. As the table shows, while the optimal result satisfies all the pre-established targets, the near-optimal results did not satisfy certain targets.

These results indicate that the proposed method was able to select the optimal result for specifications satisfying in-cabin sound pressure level targets from among multiple specification patterns, thus confirming its validity.

4.2. Element Thickness Distribution Optimization

The distribution of results for acoustic performance during optimization of the thickness distribution of the elements is shown with $\times$ marks in Fig. 12. Acoustic performance at the initial thickness distribution is shown with square marks, and at the optimum thickness distribution with triangle marks.

As the method proceeds for the optimum result, the results become clustered in the lower left-hand corner of the graph, the area in which results match target

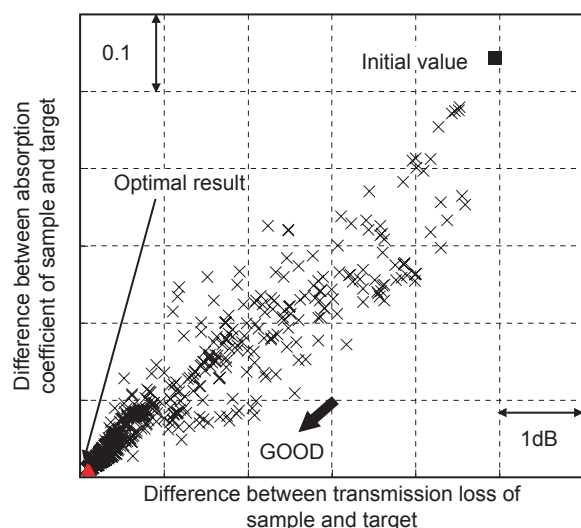


Fig. 12 Distribution of samples during optimization of thickness distribution

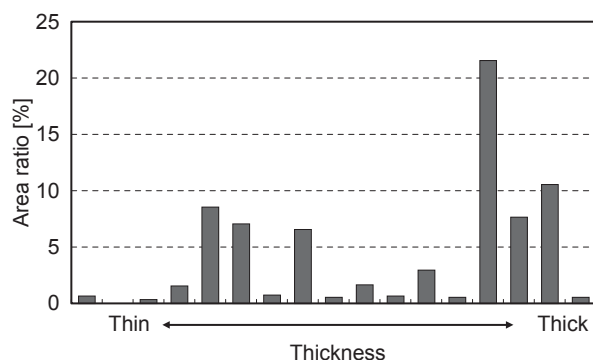


Fig. 13 Optimal random thickness distribution

performance. These results confirm that by using the proposed method, it can be achieved to select the optimum thickness distribution that enables targets for absorption coefficient and transmission loss to be satisfied from among a variety of thickness distributions rendered as functions.

In order to confirm the validity of the thickness distribution function expressed by Eq. (3), thickness distributions were expressed as random numbers without using Eq. (3) and an optimum thickness distribution was searched. The result is shown in Fig. 13. This thickness distribution differs from the distribution shown in Fig. 10 in that it does not describe a smooth curve centering on a specific thickness. The vehicle body panel is, in generally, formed in continuous shapes. The changes in thickness distribution shown in these results indicate the existence of unevenness on the surface of the floor carpet. These results demonstrate that the use of the function given by Eq. (3) to express thickness distributions enables determination of distributions that satisfy the targets for acoustic performance and are practically.

The results discussed above indicate that the new method enables a selection of an optimum result that is practically realizable from among multiple thickness distributions, thus confirming its validity.

4.3. Confirmation of Validity of Method in Actual Vehicle

The optimum specifications for acoustic materials for the floor carpet determined using the proposed method were applied to an actual vehicle. The effect of the use of these specifications on in-cabin sound pressure level satisfies the targets set in Table 2, additionally confirming the validity of the method.

5. Conclusion

A method of determining optimum specifications for sound-proof packages satisfying performance targets has been developed. The validity of this method was studied, and tests confirmed its viability. As a result, the following are stated:

- (1) The optimization procedure requires two steps. The first step is the element optimization of the acoustic package, and the second is the optimization of thickness distribution in elements.
- (2) The validity of the optimal results determined by the method was verified on the basis of the sample distribution maps.
- (3) The accuracy of the optimized specification is verified by comparing the reduction of the interior cavity sound pressure level of a vehicle with the proposed package versus the target.

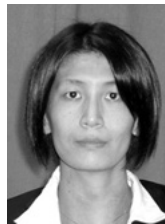
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Motorcycle Dynamic Simulation Model Incorporating Actual Rider Behavior Data

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ABSTRACT

For motorcycle dynamic simulation, the rider had previously been defined as a mass that is fixed on the motorcycle; therefore, examining and applying rider behavior to dynamic simulation models have been long-standing issues. This research established a dynamic model that simulates running on a winding road course while incorporating rider behavior. Rider behavior in actual running tests was digitalized using an image-processing-type motion capture system. By applying the digitalized behavior data to the motorcycle simulation model, the actual rider behavior was accurately modeled. Actual running paths for the simulation were measured using a Global Positioning System. Calculation accuracy of the rider model weight shift during acceleration and deceleration on the winding road course was enhanced by applying the acquired rider behavior data and the motorcycle dynamic simulation model which incorporates the multi-degree-of-freedom rider model, into the multi-body dynamics analysis software.

1. Introduction

Motorcycle running simulation using multi-body dynamics analysis software, such as straight forward stability and steady state circular simulations, have a higher correlation with actual running tests, and are widely used to develop products and to perform academic research⁽¹⁾⁻⁽³⁾. However, the rider model in these simulations is defined as a fixed mass or a simple rider model with one or two degrees of freedom. It was a challenge to apply this simple rider model to research on steering characteristics accompanying rider behavior. This research focused targets on examining rider behavior, including active weight shifting and steering maneuverability while accelerating and decelerating on a winding course, and also on realizing closed course running simulation for the development of super-sport type motorcycles. The first step was to construct measurement technology and simulation technology accurately reproducing rider behavior and motorcycle trajectory.

In order to reproduce actual rider behavior, it was necessary to establish a method of measuring the behavior of the various parts of the rider's body, such as the hips and shoulders, and to incorporate a body simulation model

that could reproduce this behavior more realistically into the motorcycle simulation model. In the past, the target running line was defined by a simple straight line or circle, but now it has become necessary to prepare one system for measuring actual motorcycle trajectory to perform calculations for winding courses. This paper describes a method of measuring rider behavior and motorcycle trajectory, and reports results when applying the rider's behavior data and actual trajectory data to a simulation, together in an actual case study.

2. Measurement Technology and Simulation Method

Figure 1 shows the flow from measurement to calculation. First, rider behavior, motorcycle trajectory, and motorcycle behavior when running on a winding course are measured, and the obtained data is incorporated into a simulation model in order to perform calculations for winding course running tests, which include acceleration and deceleration. Details are set out below.

2.1. Method of Measuring Rider Behavior

In the research carried out by Katayama and others, a rider behavior measurement system was devised that uses

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a link mechanism to couple the rider to the motorcycle⁽⁴⁾. Also, in recent years, progress has been made in body behavior measurements using motion capture technology, which is utilized in the fields of sports medicine and computer animation. A system that could accurately measure rider behavior during motorcycle riding was examined based on the prerequisites that the steering behavior of the rider be unrestricted, and that the system be small enough not to affect the motorcycle behavior. As a result, for this research, the decision was to use a motion capture method that obtains 3-dimensional coordinate data of the rider's body from video using one CCD camera⁽⁵⁾.⁽⁶⁾ To acquire video images that have little distortion and a shooting range that can cover the rider's range of motion, the camera lens and its position were studied (Fig. 2).

The basic theory of video processing is as follows. Figure 3 shows an example of the measured results obtained when a rider wearing a vest with a checkerboard pattern leans forward and moves laterally while straddling a stationary motorcycle. The yellow lines in the figure indicate the contours of the lattice pattern (hereafter called "pattern") on the back of the rider in the normal position. When the rider leans forward from the normal position,

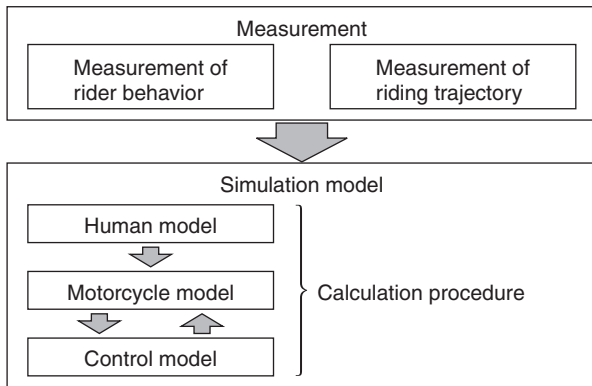


Fig. 1 Process flow from measurement to simulation



Fig. 2 CCD camera position

the pattern on the back of the rider in the moving images changes gradually from that in Fig. 3 (a) to Fig. 3 (b). Looking at the change in the pattern on the rider's back shown in Fig. 3 (b), it can be seen that the rider leans forward progressively, while moving his back forward. Figure 3 (c) shows the rider's position resulting from the normal position to the left by lateral movement. Looking at the change in the pattern in this case, it can be seen that as the rider moves laterally, his left shoulder moves forward. Motion capture software (hereafter called "capture software") traces and analyzes changes in the pattern on the moving images, enabling the positional change (behavior) in the human body to be extracted. It was confirmed that the accuracy of this software was within 10 mm, with respect to both longitudinal and lateral motion of the rider.

The method of acquiring rider positional behavior from the video information together with an example of implementation (Fig. 4) is described below. It is assumed that the images are processed in video units cut to an arbitrary length. First, the basic dimensions, such as the height, shoulder width, arm length, and the like (hereafter called "each part of the body") of a simplified body model on the capture software expressed using a yellow wire model, are matched to the actual dimensions of the rider. Next, the position and posture are adjusted (hereafter called "associated") so that each part of the body of the rider in the video coincides with each part of the body of the simplified body model in the first frame of the video unit. The capture software calculates the positional change of the simplified body model so that it matches the change in the video pattern, and continuously traces and extracts the position and posture of each part of the body. To minimize the cumulative error generated inside the capture

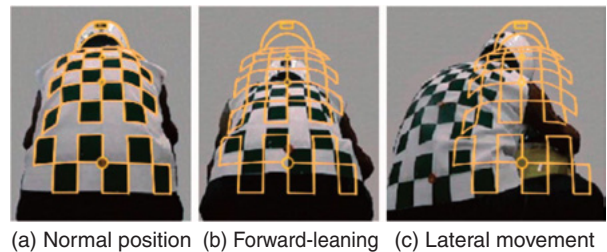


Fig. 3 Example of rider position while riding

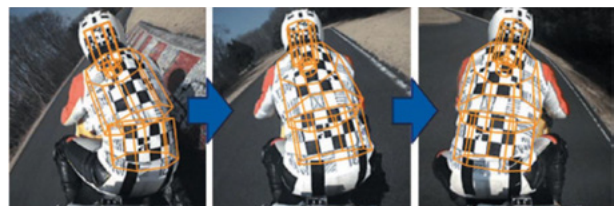


Fig. 4 Consecutive movement of rider

software, the rider in the video and the simplified body model are associated with each other even at the final frame of the video unit, and the positional change is calculated going in the reverse direction from the final frame to the first frame. The acquired data is used as behavior data for the human model that has been newly incorporated into the motorcycle simulation model.

2.2. Motorcycle Trajectory Measurement Method

For measurement of the winding course motorcycle trajectory, sufficient accuracy to obtain the difference between each rider was necessary. The inertial navigation system, remainder mark device, and the like, have been known for many years as methods of obtaining motorcycle trajectory. However, these days, a Global Positioning System (hereafter called "GPS") is generally used for vehicle speed and position, and has come into increasing use on motorcycles due to its compactness and higher performance. GPS was focused on for this research and the measurement of the motorcycle trajectory using GPS is described below. The large banking angle and the rolling and pitching behavior, which are characteristics of a motorcycle, often produce an angular change in the GPS antenna, which degrades reception, thus interfering with stable measurement⁽⁷⁾. For this reason, the position of the GPS antenna was determined not to shut off the radio waves from the GPS satellite, and one GPS that was suitable for mounting on the motorcycle was selected from the result of comparative testing of the motorcycle trajectories with several types of GPS systems. The information concerning the latitude, longitude, and the like, of the measured trajectory was coordinate-converted into ground position information. Also, because a discrepancy had occurred between the antenna mounting position and tire ground contact position, calculation using the measured data from motorcycle-mounted roll angle speed sensor was performed to locate the tire to ground contact point.

2.3. Simulation Model

The body model and body control model used in this research are described below.

2.3.1. Human body model

In this research, a body model that could accurately reproduce rider behavior on a computer was acquired. The body model was divided into 19 parts, and the mass and moment of inertia were defined for each of these parts. Here, the data was acquired by referring to the body database GeBOD⁽⁸⁾. In addition, because the load reaction and the like can be calculated, at each joint, it is considered that in the future this body model can be applied to technology for predicting rider fatigue, riding comfort, and the like. By inputting basic information, such as the rider's height and weight, a body model is made from a database that includes the actual mass, moment of inertia, and the like, in each part of the body. Next, by inputting the angle and coordinate position information of each part of the body, the body position on the motorcycle simulation model can be determined. The body model is connected to the motorcycle model at a total of five points, right and left handles, hips, and right and left steps. The rider behavior is treated as the shift at each part of the body along with the passage of time.

2.3.2. Motorcycle model

The motorcycle model is shown in Fig. 5. It consists of a total of nine elements, namely the sprung steering components, the unsprung steering components, the front tire, the rear body which includes the engine, the rider,

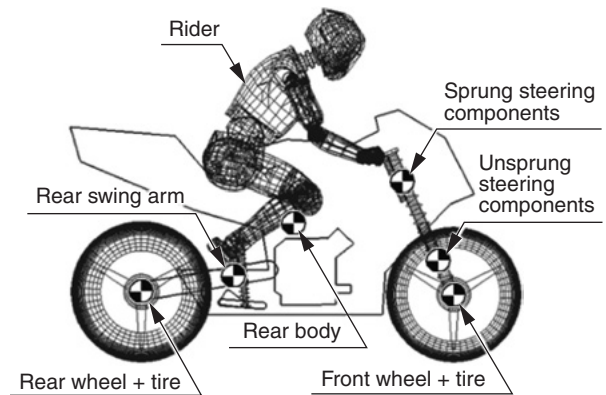


Fig. 5 Motorcycle simulation model

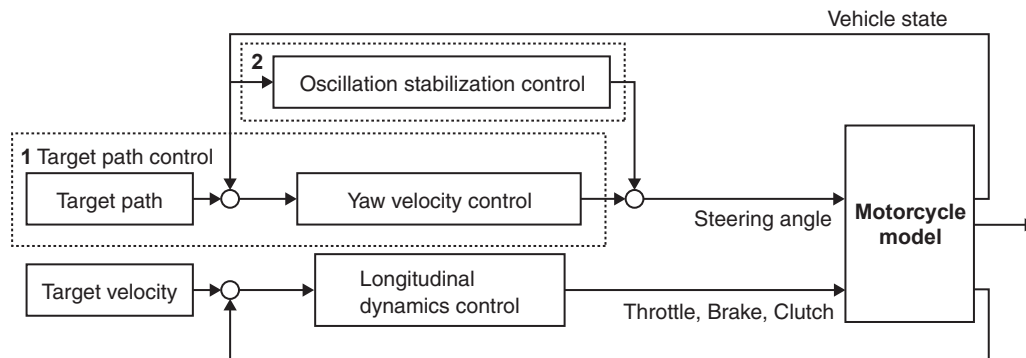


Fig. 6 Control model for motorcycle simulation model

the rear suspension and the link mechanism, the swing arm, and the rear tire. The frame body, front fork, and swing arm are treated as rigid bodies. The Magic Formula tire model is used as the tire model⁽⁹⁾.

2.3.3. Control model

The control model is divided into control in the lateral direction and control in the longitudinal direction. The control model used in this simulation model is shown in Fig. 6. The lateral direction control model, which controls the steering angle of the motorcycle model, consists of two controls: one control which follows the target running line that satisfies the balance in the rolling direction (1 of Fig. 6), and the other control for reducing vibration of weave and wobble (2 of Fig. 6). The algorithm of the control that follows the target running line uses optimization calculations to obtain the target yaw angular velocity that results in as smooth running as possible, from the deviation and curvature between the current motorcycle condition amount and target running line that is seen from the first prediction position after Δt seconds running. A feedback control model, which controls the steering angle according to the difference between the target yaw angular velocity and current yaw angular velocity, was achieved⁽¹⁰⁾.

The longitudinal direction control model was designed to control the throttle opening according to the engine speed with respect to the target motorcycle speed, and to select the appropriate gear and operate the clutch. The model was also designed to change the front and rear brake force distribution.

2.3.4. Calculation procedure

- (1) The rider behavior data is converted into the shift from the reference point on the motorcycle simulation model and created as a dedicated format.
- (2) The actual motorcycle trajectory data is created as a dedicated format converted into the tire ground contact

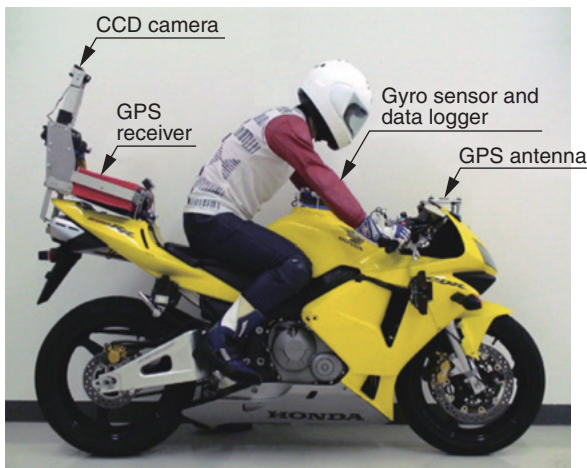


Fig. 7 Test motorcycle and location of measuring instruments

- point, and represents the target course.
- (3) The motorcycle speed data and motion distance data are created as a dedicated format, and represent the target speed.

These three files are read into the multi-body dynamics analysis software, and calculation is performed.

3. Running Measurement, Including Rider Behavior, and Measurement Results

3.1. Measurement Motorcycle

Figure 7 shows the measurement motorcycle. The motorcycle has a displacement of 599 cm³, mass of 209.3 kg, and a wheel base of 1.39 m. The measuring instruments were a motorcycle angular velocity sensor and data logger mounted on the fuel tank, a GPS mounted on the rear seat, and a GPS antenna located above the front meter. The CCD camera for recording rider behavior was fixed to the rear. The longitudinal and lateral acceleration, yaw and roll angular velocity, steering torque, steering angle, right and left step load, and front and rear suspension stroke were measured as the motorcycle behavior data. Also, to synchronize the rider behavior video and motorcycle behavior data, a photoelectric switch was mounted on the motorcycle and which reacts with a reflector plate, located along the side of the running course. One of the signals output from the switch was recorded on the video-side using an LED light emitting circuit, and the other signal was recorded to the logger for measuring data. The GPS was used to specify the position of the reflector plate, thus facilitating synchronization of the rider behavior, motorcycle behavior, and trajectory, as well as their application to simulation.

3.2. Results of Measuring Rider Behavior

Seven in-house test riders were asked to ride freely along the winding course, and measurements were

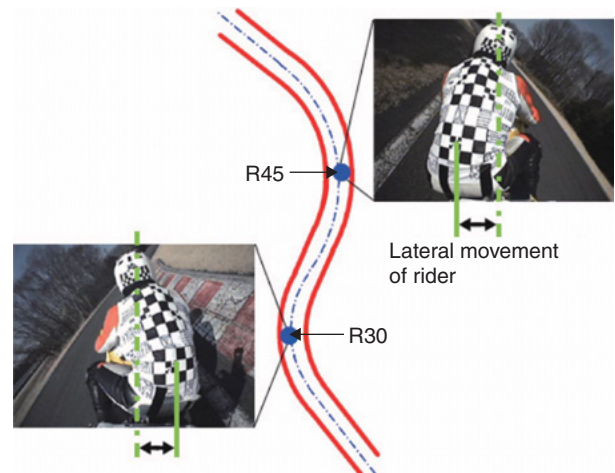


Fig. 8 Rider behavior on right-and-left turn

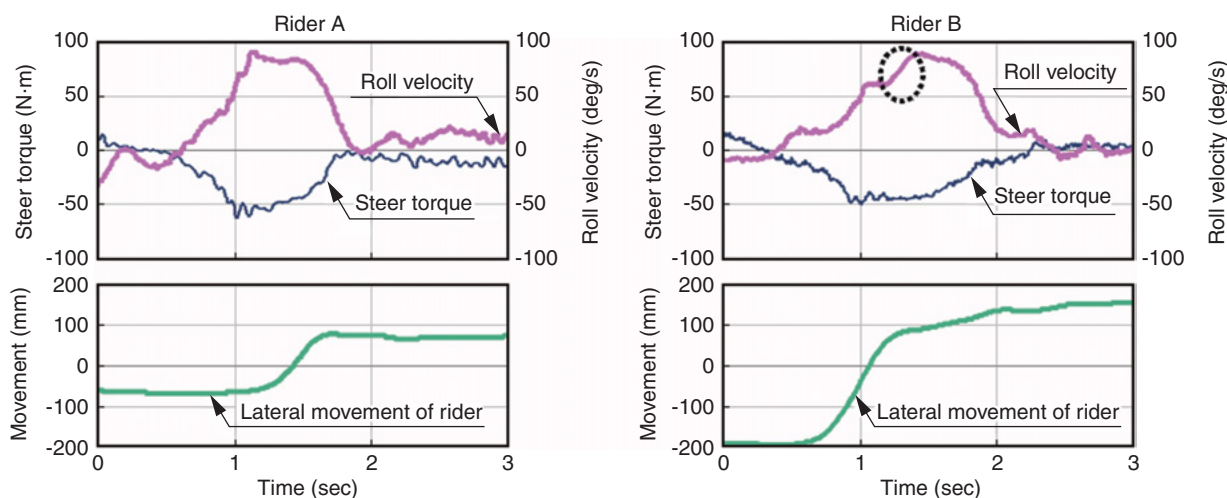


Fig. 9 Results of rider behavior

performed for each rider. As a result, the differences of the steering maneuverability and motion of the rider's body, the motorcycle behavior and trajectory, and the like, among different riders, were obtained. As an example, Fig. 9 shows the steering torque, roll angular velocity, and lateral motion of the hips, for two riders concerning which differences were found when the riders were making right and left turns (Fig. 8). For example, the lateral motion of the hips of rider A was small, and correlation was found between the steering torque and time series waveform of the roll angular velocity. In other words, in the case of rider A, the rider imparts roller behavior to the motorcycle by steering torque. The timing of the motion of rider B's hips is earlier than that of rider A, and the motion is also larger. Also, looking at the relationship between the motion of the rider's hips and the roll angular velocity, the roller angular velocity increases after the motion of the rider's hips is completed (indicated by the dotted line in Fig. 9). Because there is no notable change in the steering torque at this time, it is thought that this increase in the roller angular velocity is the roll behavior due to the shift of the rider's weight. Consequently, rider B features the use of a steering torque and weight shift to impart roll behavior. The measurement technology used in this research enabled the rider behavior

accompanying the weight shift to be converted into data, thus enabling the features of each rider to be clarified. In this measurement example, the measurement items were classified into rider behavior, handle operation, and motorcycle trajectory. From the seven riders, rider B was identified as a rider who performs standard operations. The rider behavior data obtained by measurement is applied to the motorcycle simulation model, and is used in calculations incorporating the features of the rider.

3.3. Results of Measuring the Motorcycle Trajectory

Figure 10 shows measurement results for the trajectory obtained from the GPS used in this research. A 30 m radius circle and straight lines indicating the approach and escape directions were drawn on the course. The riders ran a number of times trying to follow these lines. As a result, the dispersion of the trajectory was approximately ± 1 m in the width direction of the photoelectric switch reflector plate, as indicated in the figure. In the case of the winding course, on the other hand, generally satisfactory results were obtained, but on some corners an error of several meters occurred. It is thought that this was due to the GPS not being able to exhibit its normal performance because the GPS antenna is tilted. In the future, it will be necessary to focus attention on technology used to correct the measurement data and enhance accuracy of the GPS system.

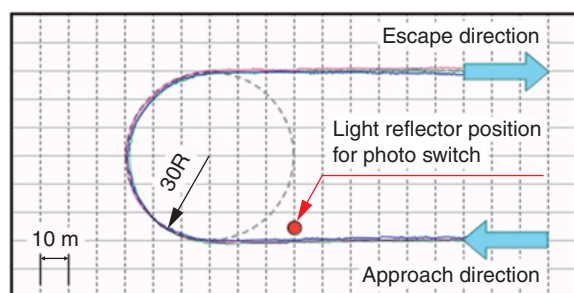


Fig. 10 Measurement of riding trajectory

4. Applied Case

As an applied case, Fig. 11 shows a simulation of running on the Twin Ring Motegi North Short Course of Mobilityland Corporation. The course has a total length of 982 m⁽¹¹⁾. Figure 12 shows the data obtained from an actual running test and the calculated results. The red line represents the test values and the blue line represents the calculated values. From top to bottom, the figure shows the motorcycle speed, roll angular velocity, and yaw

angular velocity. Apart from some exceptions, the calculated values basically agree with the test data, and it was confirmed that the rider behavior acquisition and trajectory at the tire ground contact point, which were constructed in this research, contributed to the increased degree of repeatability of the actual running test. It is considered that the difference between the test data and calculated values obtained at the low-speed corner indicated in the dotted line border in Fig. 12 is due to the fact that GPS motorcycle trajectory data was more inside than the actual trajectory.

Next, Fig. 13 shows an example of a comparative calculation between the cases with rider behavior and without it. There is a difference between the scale of the steering torque and the roll angle at a continuous corner which results in large rider behavior (Fig. 14). When

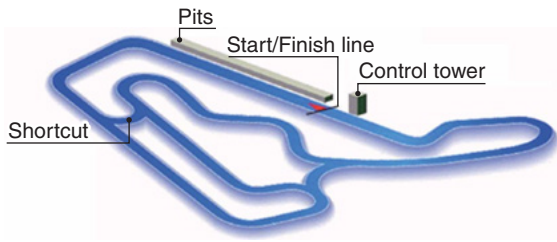


Fig. 11 Twin Ring MOTEGI (North short course)

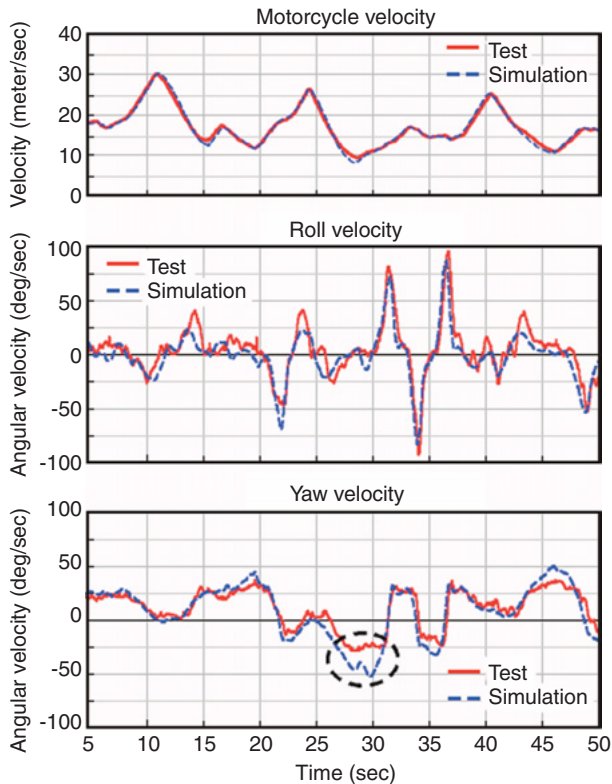


Fig. 12 Comparison between simulation results and test data

there is no rider behavior, the roll angle at the corner becomes larger, and there is a tendency to overshoot. By applying the rider behavior to the motorcycle simulation model, it was confirmed that the roll angle of the motorcycle had been reduced.

From the foregoing, it was found that by introducing a rider model with many degrees of freedom to the conventional motorcycle running simulation model, and by assigning the rider behavior and motorcycle trajectory obtained from actual running as data, composite course running, which included acceleration and deceleration, was realized.

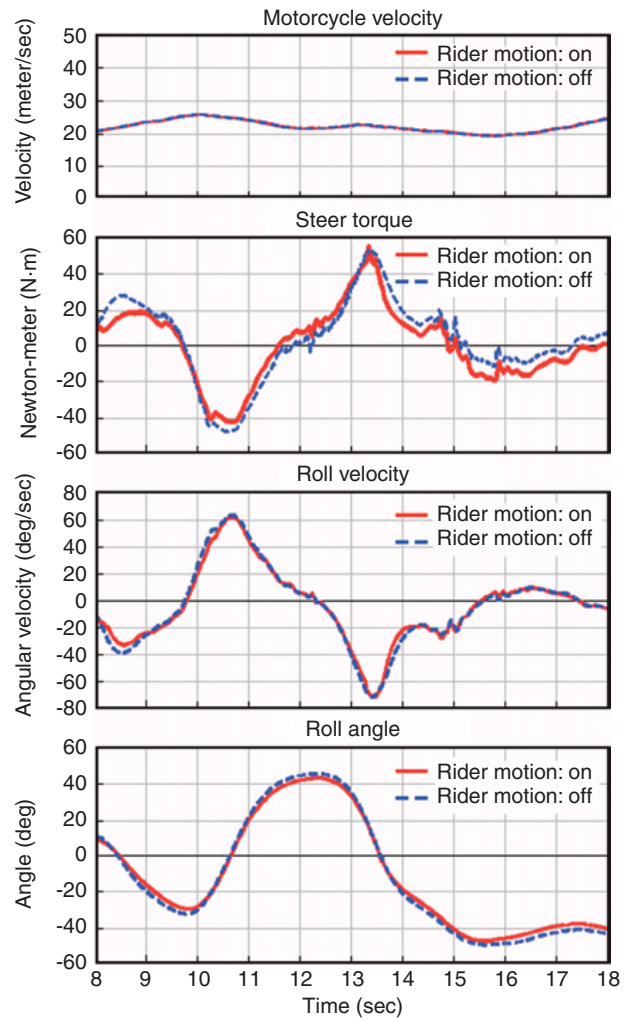


Fig. 13 Simulation results with or without rider movements

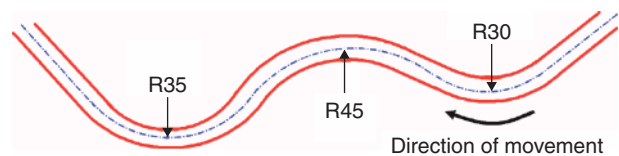


Fig. 14 Test course

5. Conclusion

- (1) By utilizing a motion capture system, a measurement method that acquired rider behavior during actual running as 3-dimensional data was established.
- (2) It was found that the actual motorcycle trajectory for a motorcycle was calculated by converting GPS observation data into the tire ground contact point using roll angular velocity information.
- (3) It was found that the motorcycle running simulation of a composite course, which included acceleration and deceleration (circuit, for example), was realized by accurately reproducing the rider behavior during actual running and assigning the actual trajectory as the target course.

Acknowledgments

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Measurement of Piston Skirt Oil-film Pressure when Piston Slap Noise Occurs

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ABSTRACT

A small thin-film pressure sensor that does not affect the piston skirt surface geometry, stiffness, and mass was directly deposited onto the piston skirt surface by physical vapor deposition, and the oil-film pressure on the skirt surface which is closely related to the cylinder liner excitation force when piston slap noise occurs was measured. The oil-film pressure corresponding to the cylinder liner acceleration, which is an indicator of piston slap noise, was measured in an engine under running condition, and the accuracy of this measurement was confirmed. It was also possible to measure the change in the oil-film pressure distribution on the piston skirt surface corresponding to changes in engine speed and the piston pin offset. The difference of the oil-film pressure on the piston skirt surface when the piston collides with the cylinder liner, which was not possible to explain from cylinder liner acceleration alone, allowed the mechanism where the piston secondary motion differs to be identified. As a result, the accuracy of the piston secondary motion simulation was enhanced, enabling effective measures against piston slap noise to be studied.

1. Introduction

Engines have been required to be compact, lightweight, high-output, highly fuel efficient and capable of producing lower levels of exhaust emissions and pollution in recent years. In addition, quietness is another strong commercial requirement for contemporary engines. To meet these performance requirements the reduction of piston slap noise, brought about by adjusting the piston offset and modifying the shape of the piston skirt, as well as the achievement of weight savings, durability, and friction reductions are design targets during piston development.

Along with the progress of continuing simulation technology, it is becoming possible to predict the piston secondary motion and the distribution of oil-film pressure on the piston skirt surface when piston slap noise occurs. However, the predictive accuracy of the simulation is insufficient and the development time required for piston development cannot yet be reduced, so there are demands to enhance the predictive accuracy.

Accordingly, it is necessary to measure the time career transition of the oil-film pressure distribution on the piston skirt surface, which is particularly closely related to the piston excitation force when piston slap noise occurs, and thereby clarify the mechanism of piston slap noise.

In this paper, the influential factors that would enhance the accuracy of the piston secondary motion were determined, which will contribute to the development efficiency. In the measurement, a compact thin-film pressure sensor was newly used in an attempt to directly measure oil-film pressure on the piston skirt surface when piston slap noise occurs.

2. Method of Measuring Oil-film Pressure on the Piston Skirt Surface

2.1. Piston Slap Noise and Oil-film Pressure Measurement

Piston slap noise occurring at the anti-thrust side was studied by measuring the oil-film pressure on the piston

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skirt surface. Mainly because of the increase and decrease of the inertial force of the reciprocating parts and in-cylinder pressure, the piston moves abruptly from the thrust side to the anti-thrust side in the top dead center vicinity of the compression and combustion strokes. At the same time, the moment generated around the piston pin makes the piston rotate, and the anti-thrust side skirt surface collides with the cylinder liner. Piston slap noise is considered to occur in this way⁽¹⁾. However, in an actual running engine, this behavior occurs in a short time, so when the cylinder liner acceleration was measured, the relationship with the piston secondary motion was unclear, and it was difficult to determine the extent to which the lateral motion and rotational motion of the piston contributed to the cylinder liner acceleration.

In order to clarify the mechanism of this piston secondary motion, it is necessary to measure the distribution of the oil-film pressure on the piston skirt surface. However, it is not possible to use large sensors that affect the mass of the reciprocating parts, or to use a measurement method that causes the piston skirt stiffness or the skirt surface properties to change.

Therefore, a sensor that can directly measure the oil-film pressure distribution on the skirt surface without affecting the piston mass, skirt surface stiffness, or shape was used.

2.2. Outline of Measurement Method

To measure the oil-film pressure distribution on the piston skirt surface, a thin-film pressure sensor that can be deposited directly on the surface of a sliding part of the engine was used. This sensor was developed to measure the oil-film pressure distribution on the main bearings of an actual running engine⁽²⁾, and has already been used to measure the oil-film pressure distribution on the piston pin boss and connecting rod big-end bearing⁽³⁾ - ⁽⁵⁾. Also during the research on the engine plain bearings⁽⁶⁾, this sensor has been used to verify the oil-film pressure distribution using a component jig tester. However, this is the first time that a thin-film pressure sensor has been used on the piston skirt, which is a representative reciprocating part of an actual running engine.

2.3. Sensor Construction

The pressure sensing part of the thin-film pressure sensor consists of a thin-film resistor deposited directly on the sensing part by means of physical vapor deposition. This sensor does not need mounting holes or an oil-leading hole for measurement, and can be installed in a shape that corresponds to the curvature of the sliding part. It therefore has the merit of exerting relatively little effect that would cause a shape change, stiffness reduction, or mass increase of the measurement object.

The thin-film pressure sensor has the shape shown in Fig. 1, and can measure pressure in as small an area of

Table 1 Alloy composition of sensors [wt%]

	Cu	Mn	Ni
Cu-Mn-Ni alloy	87.8	8.75	3.45

about 1 mm or less. As shown in Fig. 2, the sensor film consists of an intermediate film to increase bonding strength, an isolation film, pressure sensing film, lead film, and protection film on the surface of the piston, which is an electrical conductor. The total film thickness is 12.6 μm . The depositing area on the piston skirt surface used in this measurement was polished by an amount equal to the total film thickness, prior to the depositing operation, since the piston diameter had to be maintained the same as that of an actual engine after the sensor has been deposited.

2.4. Sensor Material and Pressure Characteristics

For the pressure sensing material, a copper, manganese and nickel (Cu-Mn-Ni) alloy was used for measurement of the oil-film pressure distribution on the main bearings and the connecting rod big end in an actual running engine⁽²⁾ - ⁽⁴⁾. Table 1 shows the composition of the sensor alloy.

Figure 3 shows an example of the output characteristics of the sensor fabricated from this material with respect to the pressure. High linearity and low hysteresis were obtained between the 0 and 10 MPa measurement range.

2.5. Measurement Position

Figure 4 shows the calculation results of the oil-film pressure distribution on the piston skirt surface when piston slap noise occurred at different piston pin offsets

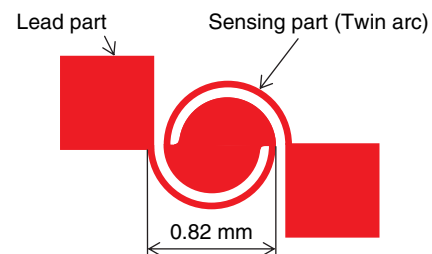


Fig. 1 Shape of thin-film sensor (Plain view)

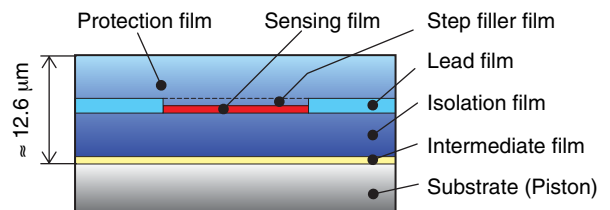


Fig. 2 Structure of thin-film sensor (Cross section)

obtained by using the existing piston secondary motion simulation. Based on these results and experience obtained during conventional development, a total of three thin-film sensors were deposited. One was deposited on the anti-thrust side piston skirt at the same position as the center of the piston pin hole, and the other two were deposited at progressive intervals of 4 mm going upward from the position of the first sensor as shown in Fig. 5.

2.6. Test Engine and Measurement Apparatus

Table 2 shows the basic specifications of the test engine.

Measurement was performed on the piston in cylinder

Table 2 Engine specifications

Engine type	Water-cooled, DOHC, 4-stroke 16 valves, In-line 4-cylinder, gasoline		
Bore × Stroke	[mm]	86.0 × 86.0	
Displacement	[cm ³]	1998	
Compression ratio		9.8 : 1	
Max. power	[kW / (r/min)]	118 / 6500	
Max. torque	[N·m / (r/min)]	191 / 4000	
Piston pin offset	[mm]	0.8 (Standard)	1.3
Cylinder offset	[mm]	0.0	

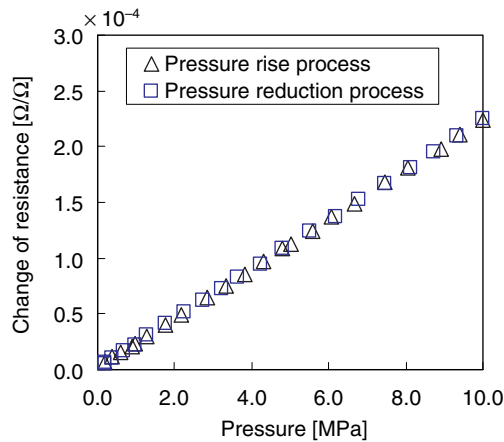


Fig. 3 Pressure-resistance characteristics

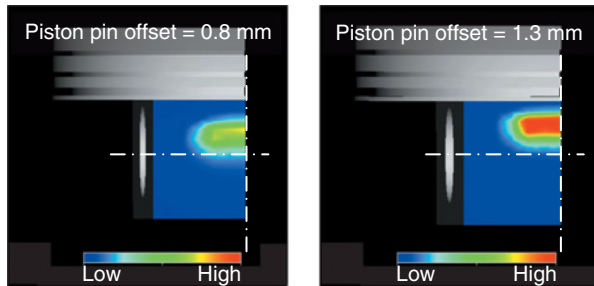


Fig. 4 Simulation results of oil-film pressure distribution on anti-thrust side skirt

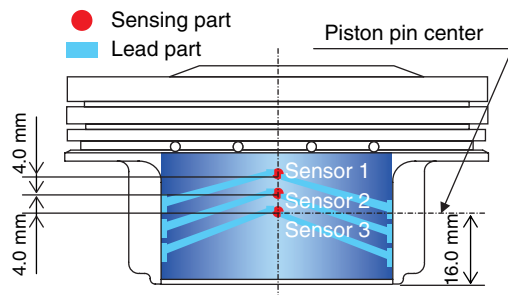


Fig. 5 Thin-film pressure sensors on anti-thrust side skirt

number 3. The linkage shown in Fig. 6 was used to run the wiring from the sensor signal terminal mounted on the piston to the outside of the engine. The linkage has three joints extending from the connecting rod big end. The wiring lead from the three thin-film pressure sensors is indicated by the arrows joining a point on the piston skirt surface to the connecting rod and linkage.

The thin-film pressure sensors are metallic resistance

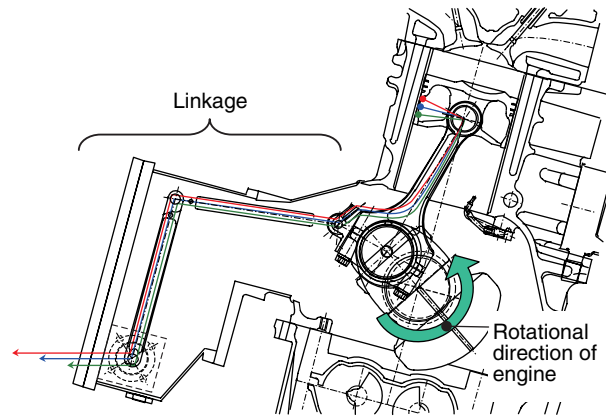


Fig. 6 Linkage and wiring from thin-film sensors

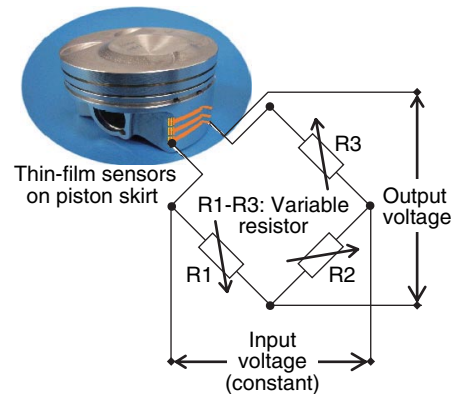


Fig. 7 Measurement circuit (Wheatstone bridge)

elements. The sensor resistance varies slightly according to the pressure applied to it. By incorporating each of these thin-film pressure sensors in one arm of the Wheatstone bridge circuit shown in Fig. 7, a minute voltage is generated between the terminals in the Wheatstone bridge circuit. This minute voltage output is amplified by a strain amplifier and measured.

3. Measurement Results and Considerations

3.1. Relation Between the Oil-film Pressure on the Piston Skirt Surface and Cylinder Liner Acceleration

The running conditions of the test engine were: engine coolant temperature 0 °C, firing, no load, mainly from 3000 r/min to 4000 r/min. When the piston pin offset is changed from 0.8 mm to 1.3 mm to the thrust side, piston slap noise is occurred at the anti-thrust side.

Accordingly, as shown in Fig. 8, a single axis acceleration sensor was installed in the normal line with respect to the circumference of the cylinder liner inside the water jacket on the outer wall of the cylinder. The measured value of the cylinder liner acceleration was

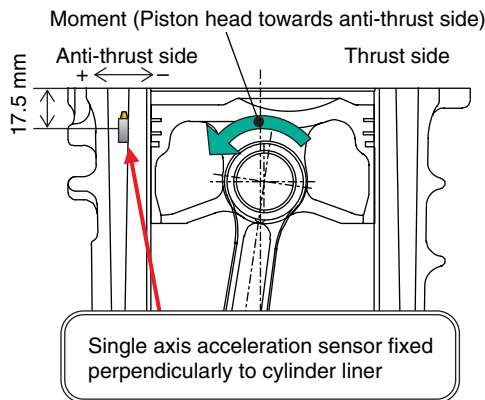


Fig. 8 Location of acceleration sensor

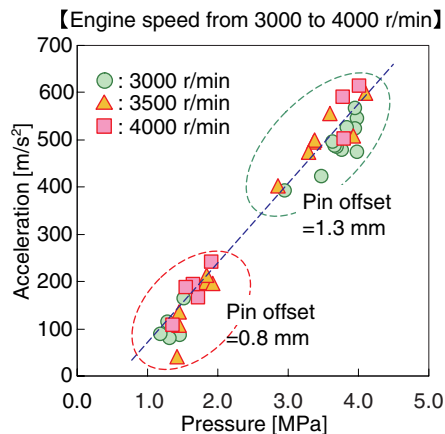


Fig. 9 Pressure measured by sensor 1 versus cylinder liner acceleration at two different piston pin offsets

treated as an objective value of the piston excitation force when slap noise is occurred, and correlation between this value and the measured value of the oil-film pressure on the piston skirt surface was investigated.

Figure 9 shows the measured values of the anti-thrust side cylinder liner acceleration sensor versus the measured values of the output from thin-film pressure sensor 1 on the anti-thrust side piston skirt surface when piston slap noise occurred under the above-mentioned running conditions.

In the figure, the area surrounded by the lower broken line represents the results for a piston offset of 0.8 mm, and the area surrounded by the upper broken line represents the results for a piston offset of 1.3 mm. The straight broken line is an approximation straight line based on the method of least squares of both results. The results obtained for the same engine speed are indicated by the same symbols.

Even taking random variations that occur due to fluctuations in combustion pressure and piston secondary motion from one cycle to another, and a slightly different contact point from the sensor position into consideration, a high proportional correlation coefficient, $R^2 = 0.980$, was obtained between the measured value of thin-film pressure sensor 1 on the anti-thrust side piston skirt surface, and the cylinder liner acceleration when slap noise occurred. Consequently, it is considered that the measured value obtained from the thin-film pressure sensor when slap noise occurs represents the piston excitation force accurately, indicating the effectiveness of the measurement method using the thin-film pressure sensor.

3.2. Comparison of Oil-film Pressure on the Piston Skirt Surface at Different Engine Speeds

Figure 10 (a) shows the measured combustion pressure values, Fig. 10 (b) shows the measured cylinder liner acceleration values on the anti-thrust side, and Fig. 10 (c) shows the measured oil-film pressure values on the anti-thrust side of the piston skirt surface. Figures 10 (d) and 10 (e) show the calculated piston side force values and the calculated piston rotational moment values around the piston pin center, obtained by using a simple mechanical model. The vertical broken line in each figure is the crank angle at which the cylinder liner acceleration occurs.

Figures 10 (b) and 10 (c) show that at 2000 r/min the cylinder liner acceleration is small, and the output from the thin-film pressure sensor on the piston skirt surface lags in phase with respect to the cylinder liner acceleration output. On the other hand, at 3500 r/min there is no phase difference between the thin-film pressure sensor output and crank angle at which cylinder liner acceleration occurs. However, the cylinder liner acceleration is large, and the three thin-film pressure sensors arranged in the vertical direction of the cylinder liner show a sequential reduction of output values from

sensors 1 to 2 to 3, as in the case of 2000 r/min.

In Fig. 10 (d), the calculated value of the side force at the crank angle where the cylinder liner acceleration occurs is roughly 0 N at 2000 r/min, but is around -700 N at 3500 r/min, and a force acts on the piston causing it

to move from the anti-thrust side to the thrust side.

In addition, in Fig. 10 (e), the calculated value of the rotational moment around the piston pin indicates a weak moment acts on the piston in the anti-thrust direction at 2000 r/min, but a strong moment acts in the anti-thrust

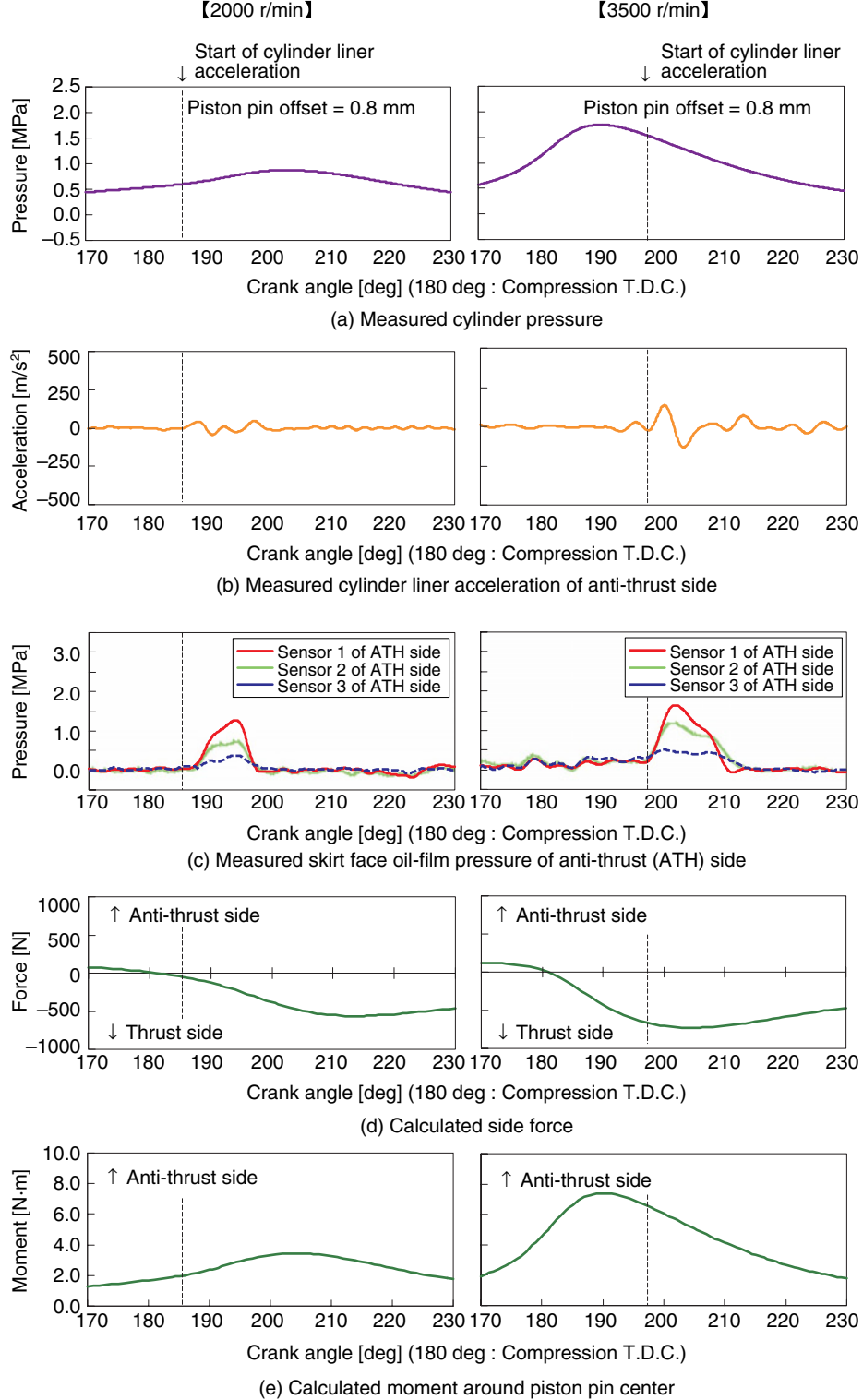


Fig. 10 Experimental and calculation results for different engine speeds

direction (direction causing the piston head to be pushed to the anti-thrust side) at 3500 r/min.

Based on the above results, Figs. 11 and 12 show the piston secondary motion at 2000 r/min and 3500 r/min, respectively.

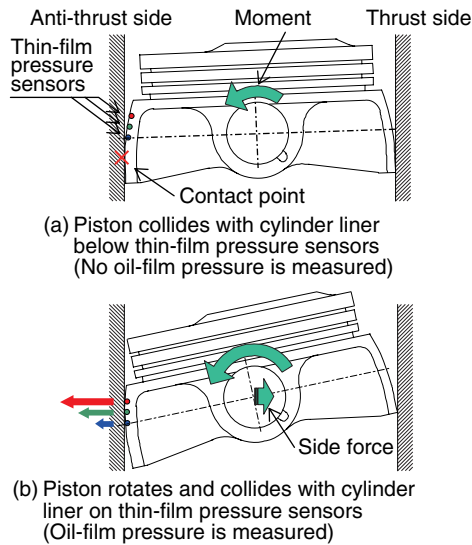


Fig. 11 Piston secondary motion at 2000 r/min

It is considered that the following occurred at 2000 r/min: Before the cylinder liner acceleration occurred, the piston was pushed to the anti-thrust side by a small side force. Subsequently, as soon as the effect of the side force disappeared, the piston, which had been subjected to the weak rotational moment around the piston pin, contacted the cylinder liner at a point below the thin-film pressure sensor, as shown in Fig. 11 (a), and only the cylinder liner acceleration occurred. After this, as shown in Fig. 11 (b), the moment around the anti-thrust continued to increase while the piston was separating from the cylinder liner, and the piston re-contacted the cylinder liner at the

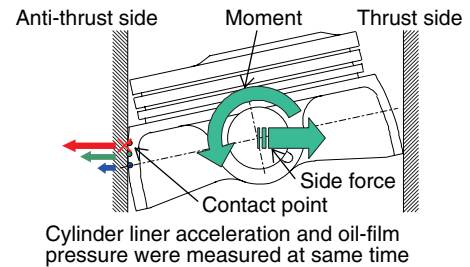


Fig. 12 Piston secondary motion at 3500 r/min

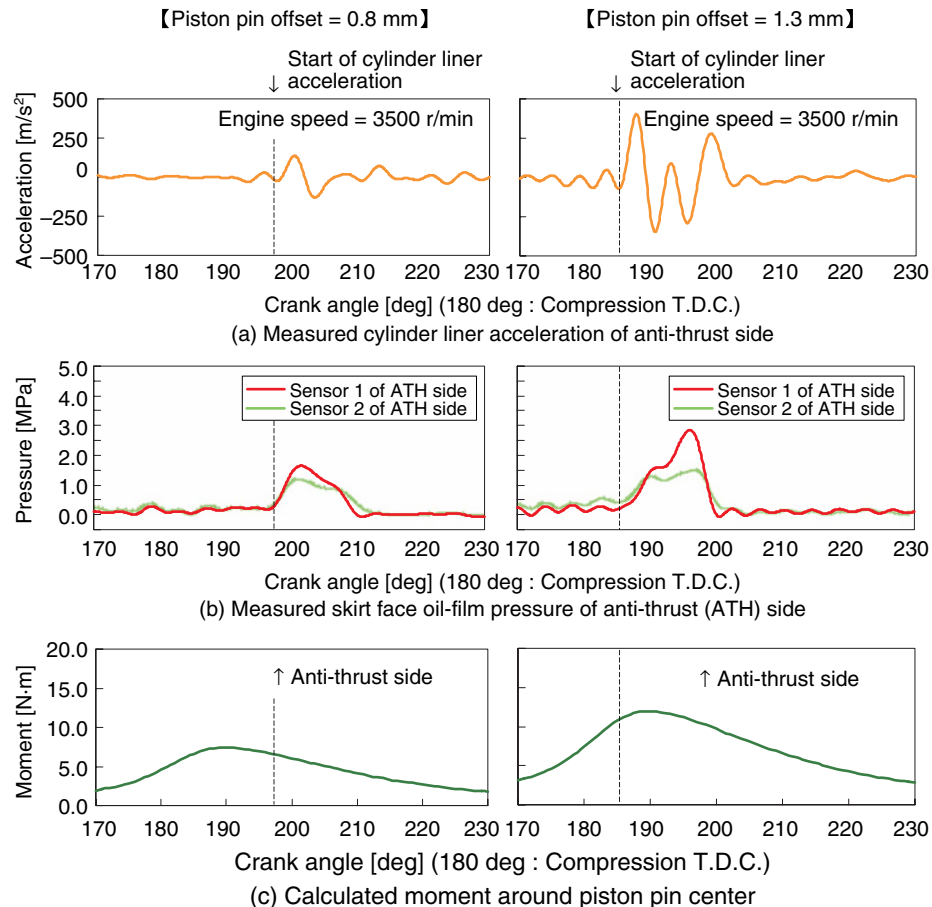


Fig. 13 Experimental and calculation results for different piston pin offsets at 3500 r/min

thin-film pressure sensor location. As a result, a phase of the oil-film pressure sensor output was delayed with respect to the crank angle at which the cylinder liner acceleration occurred.

Likewise, it is considered that the following occurred at 3500 r/min: The piston was pushed to the thrust side as shown in Fig. 12, and a clearance approximately equal to the piston clearance was formed at the anti-thrust side. In this condition, a strong moment around the piston pin acted on the piston in the anti-thrust direction. At this time, the piston rotated in the anti-thrust direction about the piston pin and contacted the cylinder liner in the vicinity of thin-film pressure sensor 1, located at the highest point. Consequently, the cylinder liner acceleration and oil-film pressure on the skirt surface were both large, and no phase difference occurred.

3.3. Comparison of the Oil-film Pressure on the Skirt Surface due to the Piston Pin Offset Difference at the Equivalent Combustion Pressure

Figure 13 (a) shows the measured values of the anti-thrust side cylinder liner acceleration, and Fig. 13 (b) shows the measured values of the thin-film pressure sensor on the piston skirt surface. Figure 13 (c) shows the calculated values of the rotational moment around the piston pin obtained by using a simple mechanical model. The vertical broken line in each figure is the crank angle at which the cylinder liner acceleration occurs.

In Figs. 13 (a) and (b), almost no difference could be seen in the maximum measured values that were output from thin-film pressure sensor 2 on the piston skirt surface due to the difference in the piston pin offset when the cylinder liner acceleration occurred, and a difference appeared only in the maximum measured values that were output from sensor 1. Consequently, it was found to be possible to evaluate the difference in the thin-film pressure on the piston skirt surface from the measured values output from sensor 1.

As a result, as shown in Fig. 13 (c), when the piston pin offset is increased, the moment around the piston pin increases, causing the top of the piston to be pushed hard against the anti-thrust side and the cylinder liner acceleration to increase.

From this fact, it was clarified that the main cause of piston slap noise is due to the rotational motion of the piston causing the contact of the top part of the piston skirt surface with the cylinder liner.

Consequently, it was possible to construct a model that considers the rotational motion of the piston in the simulation of the piston secondary motion that predicts piston slap noise, and the predictive accuracy was therefore enhanced⁽⁷⁾.

4. Conclusion

- (1) By directly installing a compact thin-film pressure sensor

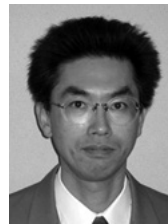
on the piston skirt surface using physical vapor deposition, it was possible to measure the oil-film pressure on the piston skirt surface when the engine was actually running.

- (2) When piston slap noise occurred, a proportional correlation was found between the cylinder liner acceleration, which is an indicator of excitation force, and the measured values obtained from the thin-film pressure sensor on the piston skirt surface, thus confirming the effectiveness of this measurement method.
- (3) The transition of the oil-film pressure on the piston skirt surface, utilizing different engine speeds and changes in the piston pin offset, was measured. The measured values of the oil-film pressure on the piston skirt surface clarified that the reason for the size of piston slap noise, which could not be explained previously from the cylinder liner acceleration alone, was the rotational motion of the piston.
- (4) Because it was possible to measure the oil-film pressure on the piston skirt surface directly under actual running conditions, the predictive accuracy of the piston secondary motion simulation was enhanced, enabling efficient countermeasures to be studied.

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Piston Slap-noise Prediction Using 3-dimensional Piston Secondary Motion Simulation

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ABSTRACT

A method of predicting piston slap noise was developed. Conventionally, piston slap noise was evaluated using moving energy or vibration of cylinder liner obtained by simulating the piston secondary motion. However, prediction accuracy and reliability were lower. This new method focuses on cylinder liner vibration, which has a high correlation with piston slap noise. Increasing the accuracy of predicting the cylinder liner thermal deformation, the piston and cylinder block dynamic stiffness model, and the 3-dimensional piston motion simulation program through oil film model modification enabled cylinder liner vibration to be predicted highly accurately. As a result, it was confirmed that piston slap noise could be predicted with sufficient accuracy for all practical purposes based on a comparison with the measured values for an actual engine under running conditions. By applying this method to a newly developed engine, the piston specifications could be determined at the initial stage of development.

1. Introduction

The importance of piston design in reciprocating engines is increasing due to requests in recent years for better environmental performance and comfort and overall drivability in automobiles. Amidst this matter, there have been many reported cases of analyses using piston secondary motion simulations to increase durability and reduce friction in the piston design^{(1) - (4)}. Various approaches using secondary motion simulations have also been applied to piston slap noise^{(5) - (7)}. In all of these studies, piston slap noise has been predicted based on the motion energy or cylinder liner vibration that is considered to correlate closely to slapping noise. However, the results have been limited to a qualitative range and the accuracy cannot be considered adequate from a practical viewpoint. A more accurate method for predicting piston slap noise is needed because the prediction of durability, friction, and piston slap noise during the initial stage of design can reduce the hours involved in testing and enhance the efficiency of design.

2. Outline of Prediction Method

2.1. Correlation between Piston Slap Noise and Cylinder Liner Vibration

Several kinds of piston slap noise are known to occur due to different kinds of secondary motion depending on the operating condition of the engine. In this research, these different forms of noise are stated collectively as slap noise. Particularly, two kinds of slap noise are covered here: slap noise that occurs at the anti-thrust side during no-load idling and slap noise that occurs at the thrust side during low-load at low-speed operation.

Figure 1 shows an example. It indicates the correlation between the relative change of cylinder liner vibration and subjective evaluation of piston slap noise under the conditions of different piston specifications and temperatures in a running engine. The cylinder liner vibration of each specification is defined as 0 dB at a borderline subjective rating. The subjective evaluation is a numerical value obtained by averaging the evaluation values assigned by several experts.

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From this graph, it is judged that there is a correlation between the change in subjective evaluation of piston slap noise and the relative change in the cylinder liner vibration and that the change in the subjective evaluation of piston slap noise is accurately predictable by the cylinder liner vibration.

2.2. Calculation Flow

Figure 2 shows the calculation flow. When predicting piston secondary motion, thermal deformation has a large effect on the clearance. Consequently, it is important to accurately predict the thermal deformation of the cylinder liner and piston when piston slap noise occurs.

A general purpose fluid analysis program was used to analyze the thermal distribution inside the water jacket,

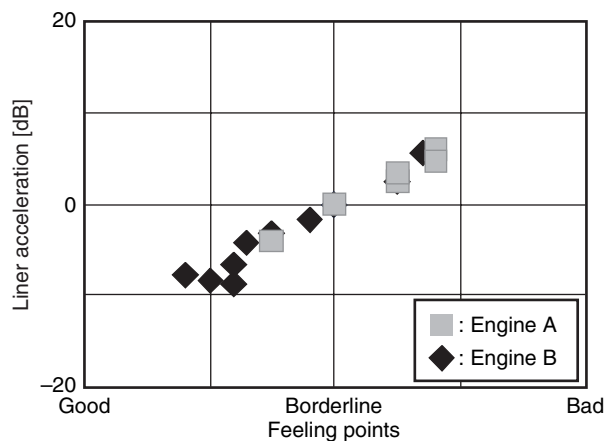


Fig. 1 Correlation of cylinder liner acceleration and feeling points

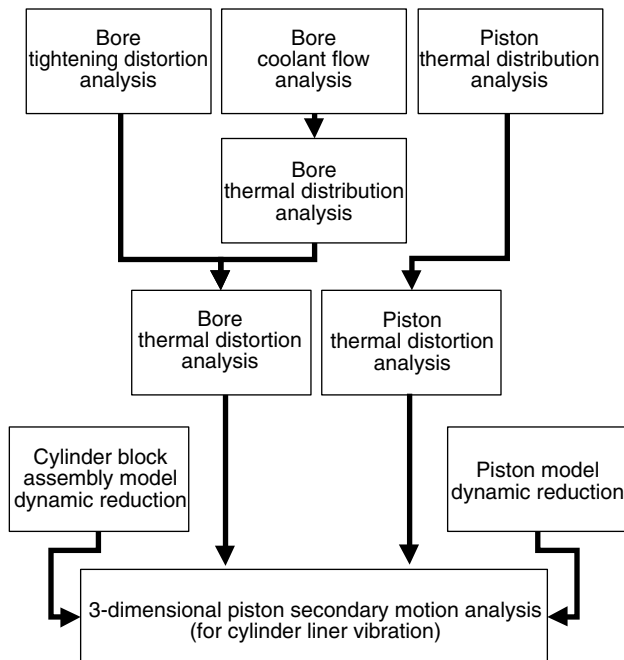


Fig. 2 Analysis flow

and a general purpose non-linear structural analysis program was used to analyze a tightening distortion of the cylinder block, enabling prediction of the thermal deformation of the cylinder liner during the low-temperature warm-up stage.

The thermal deformation of the piston is predicted by analyzing the thermal distribution taking the heat input from the combustion chamber into account.

Simultaneously, an FEM model of the cylinder block and piston, which were used in the previous thermal analysis for an accurate prediction of the cylinder liner vibration after the piston collision, is obtained by dynamic reduction in consideration of the vibration characteristics.

The cylinder liner vibration is accurately predicted by these models and a 3-dimensional piston secondary motion program.

3. Prediction of Thermal Deformation and Verification of Prediction

3.1. Cylinder Liner Thermal Deformation Analysis

Cylinder liner deformation in the engine running condition was predicted by combining the tightening distortion analysis and thermal deformation analysis based on the thermal distribution calculation using the heat transfer coefficient distribution with the coolant flow⁽⁸⁾. Figure 3 shows the FEM model of the cylinder block used in the analysis.

Figure 4 shows the results of a comparison between the temperature values obtained from the thermal distribution analysis performed with this model and measured cylinder liner temperature values obtained under

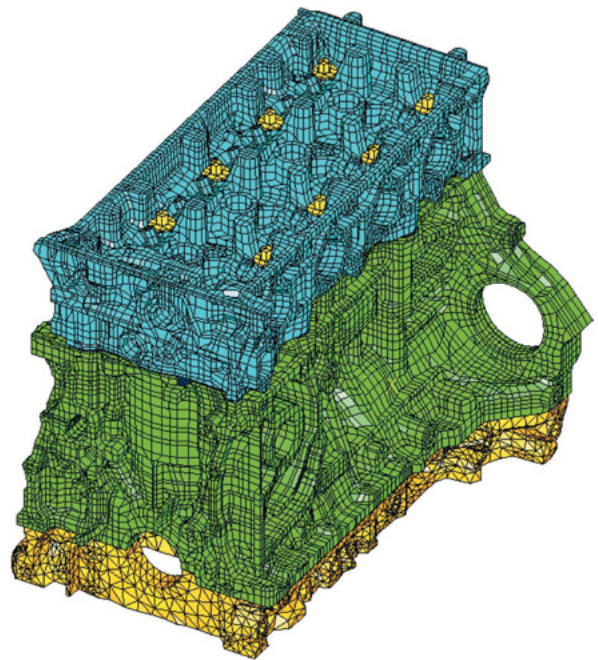


Fig. 3 Cylinder block FEM model

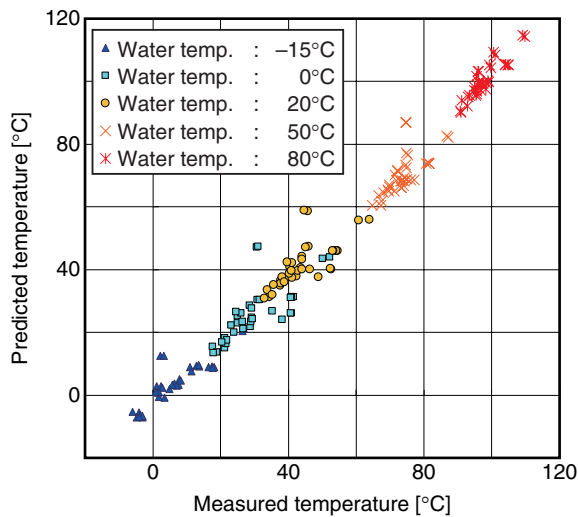


Fig. 4 Verification results for thermal distribution of cylinder block

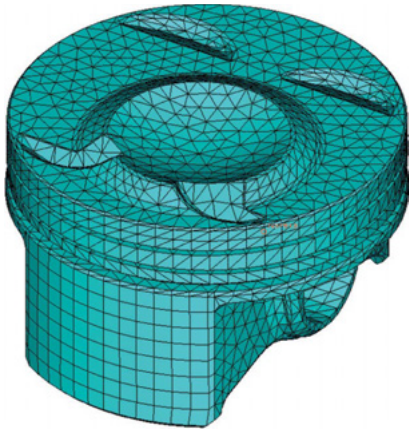


Fig. 5 Piston FEM model

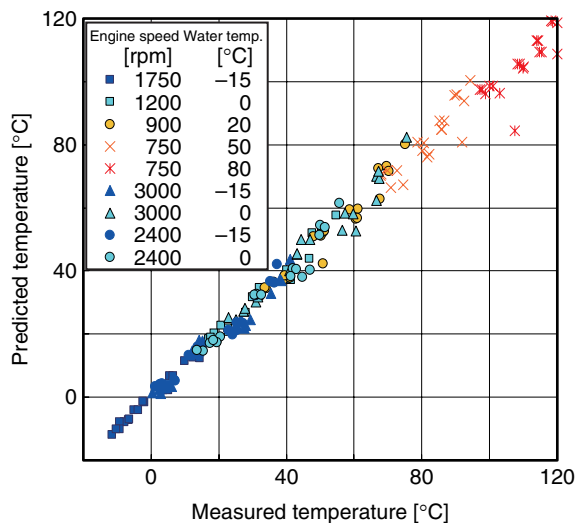


Fig. 6 Verification results for thermal distribution of piston

engine running conditions. Temperature deviations between the predicted and measured values obtained immediately after a cold start, during warm-up, and after warm-up were $\pm 17^\circ\text{C}$ in the analysis, while the standard temperature deviation was 5.8°C , and the obtained predicted temperatures were reliably accurate. Based on these analysis results, thermal deformation analysis is performed.

3.2. Piston Thermal Deformation Analysis

The piston deformation under engine running conditions was analyzed using a database of past analyses. The heat input value in the vicinity of the combustion chamber and boundary conditions were used as input conditions for the thermal distribution analysis. Figure 5 shows the FEM model of the piston used in the analysis.

Figure 6 shows a comparison between the predicted value of the thermal distribution analysis results and the piston temperature measurement value under engine running conditions. The temperature deviations at the measurement points were within 10°C , and the temperature prediction accuracy was judged adequate for the piston as well. Therefore the thermal deformation analysis was performed using the results of the thermal distribution analysis.

4. Three-dimensional Piston Secondary Motion Simulation

4.1. Reduction Model

In the 3-dimensional piston secondary motion program used in this research, the cylinder block model and piston model used in the thermal deformation analysis in Figs. 3 and 5 were reduced by the dynamic reduction method to enhance the accuracy of the dynamic stiffness model for a more accurate prediction of the cylinder liner vibration. Figure 7 shows the results of the verified accuracy of the cylinder block model.

The frequency response function was measured by taking the top part of the inside wall of the cylinder liner

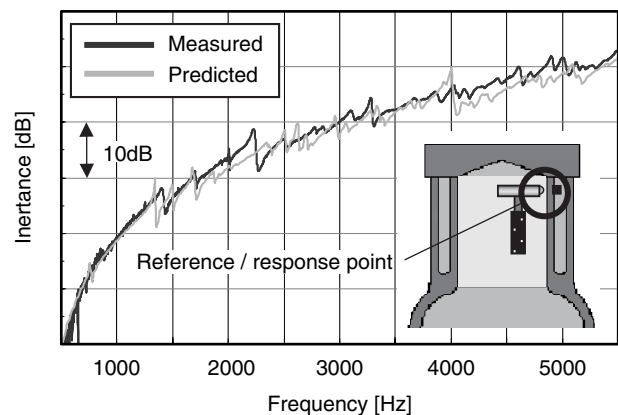


Fig. 7 FRF verification results of cylinder block

as an excitation point and the point for measuring the cylinder liner vibration in an actual engine as the response point. The same points were used as the excitation and response points for the frequency response analysis using the FEM model.

As a result, the predicted values were located nearly on the same line as the measured values, so it is judged that the model had sufficient accuracy as a cylinder liner vibration analysis model.

4.2. Oil Film Model

In the 3-dimensional piston secondary motion simulation program, the condition with the space between the piston skirt and cylinder liner completely filled with oil was represented by a model because the oil film characteristics incorporating elasto hydrodynamics (EHD) are applied to the sliding faces of the piston skirt and cylinder liner. However, in an actual engine, a mechanism that forces lubrication between the piston skirt and cylinder liner does not exist, and the actual lubricated condition is thought to result in air bubbles. For this reason, a model with air bubbles in the oil between the skirt and the cylinder liner was introduced as shown in Fig. 8⁽⁹⁾ to correct the oil film pressure due to the inclusion of air bubbles.

4.3. Piston Secondary Motion Analysis

After the analysis using the results in 3.1 and 3.2 and the model, the predicted values of the piston secondary motion and the measured values for a running engine were compared.

Figure 9 shows an outline of the device used for measuring the piston secondary motion. The secondary motion was measured by a method in which four pairs of gap sensors were mounted in the cylinder block and the lateral and rotational displacements of the piston calculated by using the outputs of two pairs of these sensors. Figure 10 shows the results of a comparison of the predicted and measured values of the lateral displacement of the piston secondary motion under the condition where the slap noise is pronounced (no-load idling at 2400 rpm). Figure 11 shows the results of a

comparison of the predicted and measured values of the rotational displacement of the piston secondary motion under the same conditions.

Comparing the predicted and measured values confirmed that the predicted values of the piston secondary motion accurately reproduced the measured values and taking air bubbles in the oil film into account enhanced the accuracy.

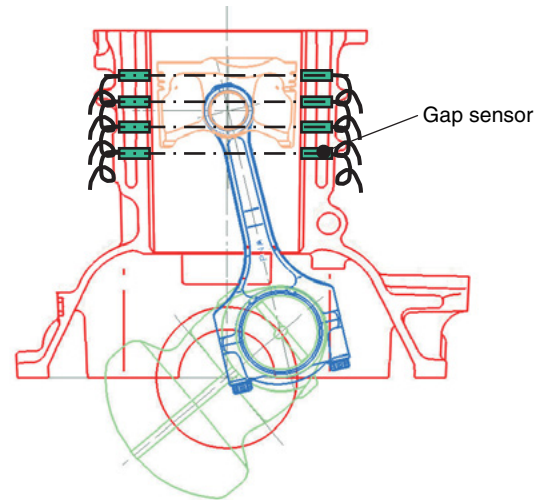


Fig. 9 Piston motion measurement method

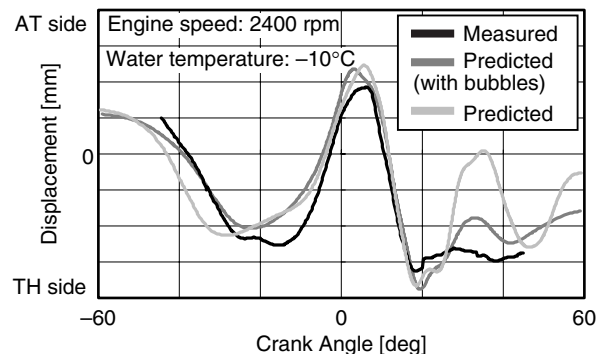


Fig. 10 Verification results for lateral displacement of piston

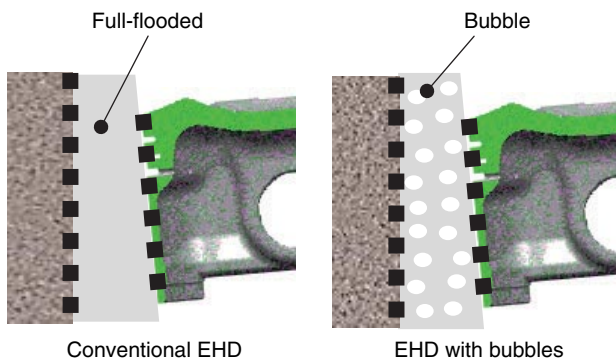


Fig. 8 Conception of bubble model

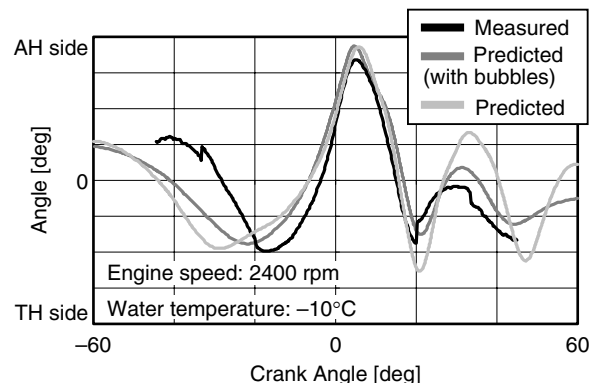


Fig. 11 Verification results for rocking angle of piston

Table 1 Specifications of engine and piston

Items	Specifications
Cylinder configuration	Inline 4-cylinder
Valve train	DOHC 4 valves
Crank offset length [mm]	0
Ignition system	Single point ignition
Fuel supply system	PGM-FI
Displacement [cm ³]	1998
Bore × stroke [mm]	86 × 86
Compression ratio	9.8
Maximum power [kW]	118
Maximum torque [Nm]	191
Compression height [mm]	26.2
Skirt width × height [mm]	34 × 29.5
Piston weight [g]	310

Table 2 Variations among pistons

	BASE	A	B	C	D
Pin offset [mm]	0.8	1.3	0.3	0.8	0.8
Skirt form	Base	Base	Base	Thin	Thick

4.4. Verification Results

As mentioned previously, it was confirmed that the prediction accuracy of the piston secondary motion was increased by enhancing the prediction accuracy of the cylinder block and piston thermal deformation, enhancing the accuracy of the dynamic stiffness model, and modifying the oil film model. This method is used to compare the cylinder liner vibration change with respect to a change in the piston specifications with the measured values of the cylinder liner vibration under engine running conditions.

Table 1 shows the specifications of the test engine and the pistons, and Table 2 shows the specifications changed for the verified piston of each specification.

The object of analysis was slap noise at the anti-thrust side under no-load idling around 2400 rpm where the difference among specifications is prominent in the measurement with the running engine.

The cylinder liner vibration was measured by installing a small acceleration sensor inside the water jacket of the engine, on the outside wall of the cylinder liner (15 mm below the gasket face in the thrust direction).

Figure 12 is a graph showing the relative change of the cylinder liner vibration obtained from the above measurement, taking the level of the base piston as 0. This

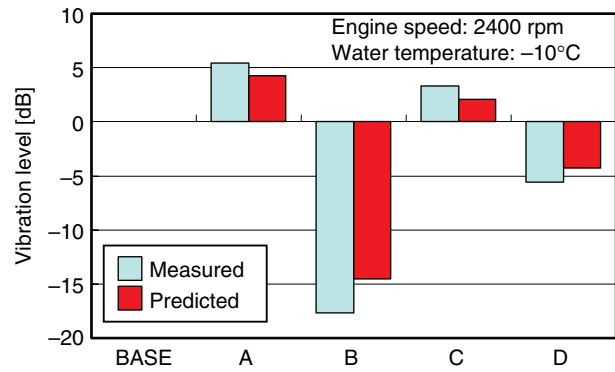


Fig. 12 Verification results of cylinder liner accelerations

graph shows a reliable prediction accuracy for all specifications, although the analysis results tended to be slightly lower.

5. Application to Actual Engine

Using this prediction method, a study of the skirt profile in the simulation was performed for a newly developed engine.

Table 3 shows the specifications of the newly developed engine and piston.

Piston slap noise is generated due to different secondary motions resulting from different engine operation conditions as mentioned previously. For this reason, when determining specifications, it is necessary to simultaneously reduce slap noise for these different regions.

Table 3 Specifications of newly developed engine and piston

Items	Specifications
Cylinder configuration	Inline 4-cylinder
Valve train	SOHC 2 valves
Crank offset length [mm]	14
Ignition system	Dual point sequential ignition
Fuel supply system	PGM-FI
Displacement [cm ³]	1339
Bore × stroke [mm]	73 × 80
Compression ratio	10.8
Maximum power [kW]	70
Maximum torque [Nm]	123
Compression height [mm]	26.2
Skirt width × height [mm]	38 × 29.5
Piston weight [g]	205

Therefore, it has to be confirmed that slap noise is not enhanced under a number of operation situations even when the skirt profile is changed.

The barrel form and ovality of the skirt profile were changed from the base piston shape.

Figure 13 shows the analysis results of the cylinder liner vibration at the anti-thrust side during 3000 rpm no-load idling and at the thrust side during 1500 rpm low-load operation for four kinds of pistons with modified skirt profiles. As mentioned previously, the graph

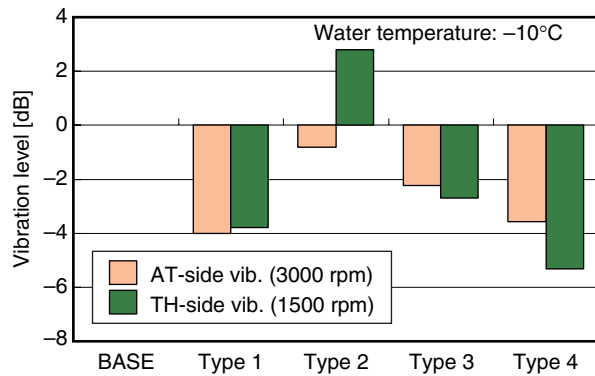


Fig. 13 Comparison of predicted cylinder liner acceleration (TH side and AT side)

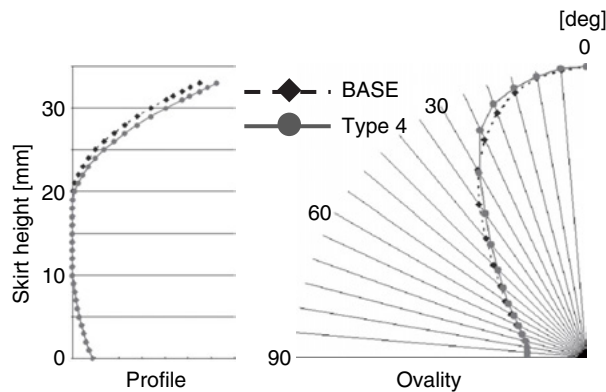


Fig. 14 Skirt profile

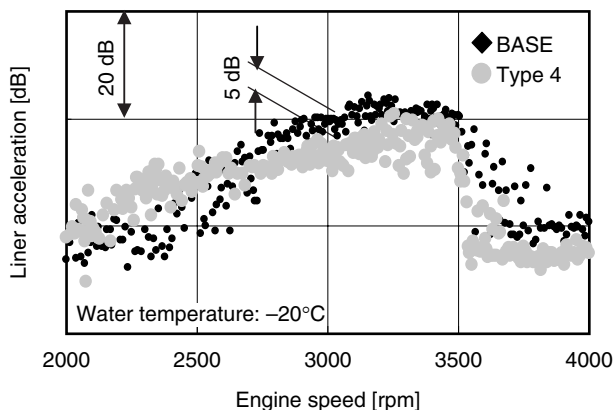


Fig. 15 Cylinder liner acceleration (BASE versus Type 4)

indicates the relative change when the cylinder liner vibration level of the base piston is defined as 0.

This graph shows the reduction of piston slap noise at both the thrust and anti-thrust sides would be predicted because the vibration levels of the Type 1 and Type 4 forms were reduced by about 4 to 6 dB.

At the same time, friction and scuffing was predicted⁽⁴⁾. Of the proposed specifications, an increase in scuffing and friction was not predicted with Type 4. Therefore, Type 4 was actually created and verified using a running engine.

Figure 14 shows an outline of the base and Type 4 skirt profiles.

By checking piston slap noise in a running engine, it was confirmed that as predicted, the cylinder liner vibration was about 5 dB lower than that of the base piston, and slap noise was also reduced in the case of the subjective evaluation.

Figure 15 shows the measured results of base and Type 4 cylinder liner acceleration on the anti-thrust side. This confirms cylinder liner vibration acceleration is reduced in the region between 2800 and 3200 rpm where piston slap noise was prominent.

6. Conclusions

- (1) In the region where piston slap noise occurs, the relative prediction of piston slap noise was performed by a reliable and accurate prediction of the cylinder liner vibration.
- (2) The accuracy of piston slap noise prediction using cylinder liner vibration obtained by analysis was verified and its effectiveness confirmed by measurement under actual engine running conditions.
- (3) This method was used for a newly developed engine to predict the reduction of piston slap noise due to the change in the piston form with the optimum form proposed. As a result, the specifications were determined in the early development stage, contributing to a reduction of hours involved in testing.

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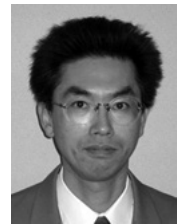
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Study of Modal Analysis from Order Tracking Data

Motoyasu SAKAGUCHI*

ABSTRACT

A Modal analysis technology using engine data taken under operational conditions is proposed. Taking the features of engine vibration into consideration, order tracking vibration data is used as an input to the subspace method and the identification of modal parameters is enhanced.

Paying attention to the mode shape used to validate FEM, and an accuracy validation by the proposed method was conducted. In the excitation test using pulses, pseudo tracking analysis was conducted. The results were compared with the modal analysis using the transfer function, and it was confirmed that a similar mode shape was obtained. By applying this method to actual engine vibration, the chain cover mode of diesel engines was identified, and it was confirmed that the orthogonality of the mode shape was enhanced compared to the mode shape obtained using conventional operational deflection shape and using average spectrum.

From the above, it is determined that the proposed method offers enhanced identification accuracy of the natural modes under actual engine operating conditions, as compared to the conventional method.

1. Introduction

Identifying the natural modes of actual structures is important for both identifying countermeasures to reduce vibration and increasing the accuracy of FEM models. Natural modes are usually identified through laboratory excitation tests. However, because of the effect of nonlinearity and differences in engine constraint conditions, the natural modes identified in excitation tests do not correspond to those in actual operation in some cases. For this reason, operational deflection shape (ODS) analysis, which simplifies the parameters by focusing on the amplitude of vibration at specific frequencies, has been used as a substitute procedure in the identification of natural modes in operation. This method is useful for studies of vibration countermeasures that visualize vibration⁽¹⁾. However, in analyses that focus on natural modes, such as comparisons between FEM and test results, and studies on the optimum measurement points for tests⁽²⁾, etc., the ODS is a fundamentally different phenomenon from natural modes, and the use of this method may cause reduced accuracy of the analysis.

On the other hand, an operational modal analysis

(OMA)^{(3),(4)} has been studied as a method of identifying natural modes from only the vibration response of actual machines in operation. The precondition for the OMA is that the excitation force acting on the structure should display no bias, such as white noise, in the frequency domain. However, a comb-shaped spectrum is generated due to rotational order harmonics dominant in the excitation force in an engine. Thus the application of the OMA is challenging, and accurate natural modes cannot be obtained.

Therefore, a method to identify natural modes in an operating engine accurately was proposed⁽⁵⁾. This method increased the accuracy of the modal identification by using order tracking data for the OMA input, and by taking the engine vibration characteristics into consideration. By focusing on the mode shape which is one of the natural mode characteristics and is frequently used in comparison with FEM results, the accuracy was verified again. From excitation tests simulating engine vibration, the mode shapes that were similar to the mode shapes obtained from conventional experimental modal analysis (EMA) using frequency response functions were obtained by the new method. Application of the new method to the actual

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engine vibration obtained better orthogonality of mode shapes than those obtained by the ODS. These results confirmed that the new method increased the identification accuracy of mode shapes.

2. Method Used to Increase the Identification Accuracy of Modal Parameters

2.1. Stochastic Subspace Identification Method

The method used in the OMA is known as the stochastic subspace identification method⁽³⁾ or the subspace algorithm⁽⁴⁾. The theoretical foundations of this method have been described in detail in the papers cited in the reference literature, and will therefore be dealt with here only in outline. The stochastic subspace identification method is a method of identifying natural modes by subjecting a Hankel matrix, which consists of the correlation functions of vibrational responses, to singular value decomposition, to determine the dominant matrix for the responses. Figure 1 shows an overview of the method. The correlation functions of the responses that are used for the input are obtained by inverse Fourier transformation of the cross spectrum. If the input excitation force satisfies the precondition of a low level of bias in the frequency domain, the stochastic subspace identification method can be used to obtain the natural frequency, mode shape, and modal damping ratio directly from the vibration data. However, because the method does not employ a frequency response function, scaling cannot be conducted using modal mass or other parameters. Because of this, the results obtained from this method can be used to verify FEM results, but cannot be used in substructure synthesis methods.

Since the stochastic subspace identification method does not need excitation tests as noted above, it has often been used in the analysis of structures for which such tests are inconvenient, such as large buildings. The method also enables natural modes to be determined under operating conditions in cases such as automotive suspensions⁽³⁾, in which modes are highly amplitude-dependent and it is difficult to identify modes in an excitation test that correspond to those occurring in actual operation.

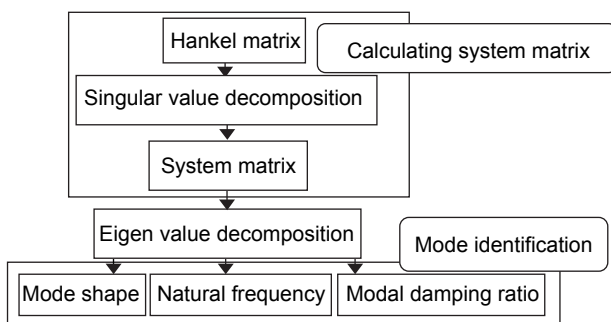


Fig. 1 Overview of OMA

However, when there is a specific bias in the excitation force, it is not possible to judge whether the result represents the true natural mode or the effect of the excitation force, decreasing the reliability of the analysis.

2.2. Issues Arising when the OMA is applied to Engines

As Fig. 2 shows, rotational order harmonics are dominant in engine vibration and generate a comb-shaped spectrum. The precondition that the excitation force should demonstrate a low level of bias in the frequency domain, as indicated in the preceding section, is therefore not satisfied, and the OMA cannot be used. To apply the OMA, it is necessary to reduce the bias of the excitation force by averaging the vibration spectrum over time or in relation to engine speed (Fig. 2). In many cases, commercially available OMA systems simply average the vibration spectrum, assuming white noise of the type generated by microtremors in buildings. When this method is applied to engine vibration, however, low response level vibration other than that of order harmonics is also included in the averaging. Therefore, the S/N ratio and other reliability of data decline and so does the accuracy of modal identification.

2.3. Proposal of a Method of Increasing Accuracy of Analysis in Engines

The order harmonic data obtained by tracking analysis should be used as the input data in the OMA to increase the identification accuracy of modal parameters when an engine is operating. The excitation generating the order harmonics originates from the inertial force of the interior of the engine, the excitation force produced by the operation of various mechanisms, including the meshing of the gears, polygon effect of the chain and the sprockets, and combustion pressure, among other sources, and therefore remains comparatively stable even when engine speed varies. By focusing on these matters, apply the order tracking analyses to vibration, and use the results as spectra. More accurate S/N ratios than those obtained by simple averaging of the vibration spectrum, as well as the response with a lower level of bias in the frequency of excitation force can be obtained by this method. It could therefore be assumed that the use of order tracking data would increase the accuracy of the OMA (Fig. 3).

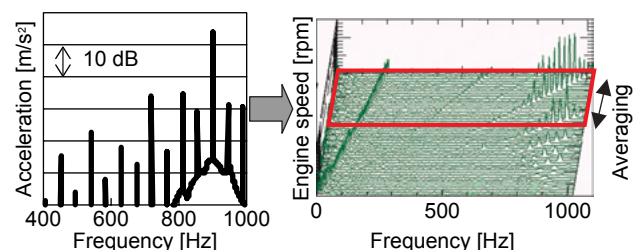


Fig. 2 Averaging engine vibration

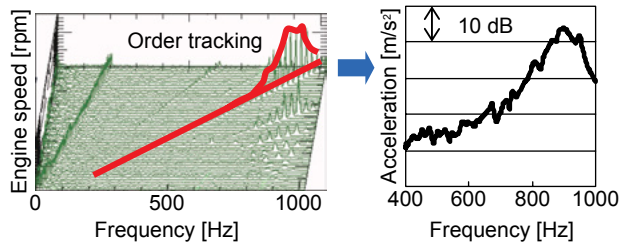


Fig. 3 Order tracking of engine vibration

3. Methods of Verifying Accuracy

3.1. Verification of Accuracy Using Excitation Tests

Prior to verifying the accuracy of the new method in operating engines, excitation tests were conducted to verify whether the OMA could be applied to order tracking data (Fig. 4). The targets for measurement were the cylinder head and cylinder block with the chain cover and head cover attached. An electromagnetic exciter was used to excite the oil pan deck of the cylinder block, and only the vibrational response of the head cover was measured. To enable a simulated order tracking analysis to be conducted in the excitation test, pulse waveforms such as saw-tooth waves were applied for excitation, and the interval between the pulses was continuously varied to simulate an increase in engine speed. Order tracking analysis was applied to this excitation, and the modes were identified by OMA. To provide a comparison, an excitation test was also conducted with random noise to compare the results of the OMA and EMA. The widely-used time domain poly-reference method was used for the EMA.

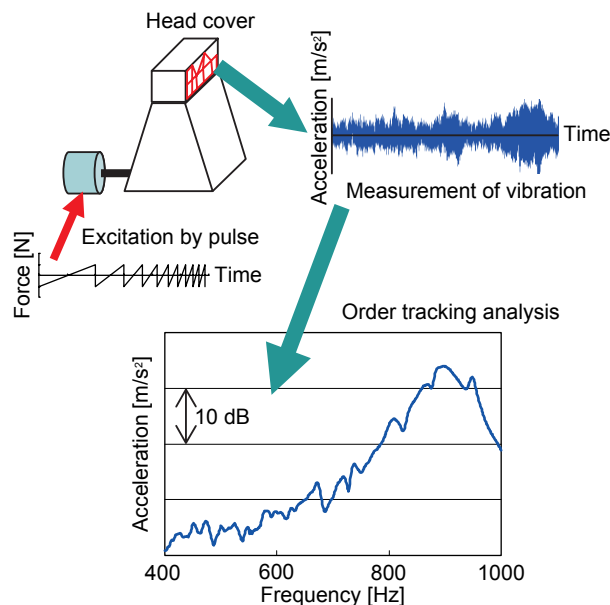


Fig. 4 Overview of pulse excitation test

3.2. Engine Operation Tests

Then the accuracy of the OMA was tested using the vibration generated by an engine operating on a test bench. Out-of-plane vibration on the surface of the chain cover in a 4-cylinder 2.2 L diesel engine was measured (Fig. 5). External drive was used to reduce noise generation by the combustion excitation force, and vibration was measured as engine speed was increased. An order tracking analysis focusing on the polygon order of the chain and sprockets was conducted as input to the OMA analysis. The conventional method of averaging the spectrum was also used, and the modes obtained from the OMA using the order tracking data were compared with the modes obtained using averaged data. The ODS, which has traditionally been used as a substitute to identify natural modes in operation, was also conducted by using the results of tracking analysis, and the results were compared to those by the OMA.

3.3. Comparisons of Mode Shape

A mode shape is an important parameter which is obtained from modal analysis and used in the visualization of vibration and comparison of results with FEM results. Therefore, the mode shape was focused on in testing the accuracy of the OMA. This section discusses a method of assessing the accuracy of mode shape identification using the Modal Assurance Criterion (MAC)⁽⁶⁾. When the mode vectors ϕ_r and ψ_s are given, the MAC can be found using Eq. (1), where the subscripts r and s express the order of the natural modes.

$$MAC(\phi_r, \psi_s) = \frac{|\phi_r^H \psi_s|^2}{\|\phi_r\|^2 \|\psi_s\|^2} \quad (1)$$

ϕ_r^H is the conjugate transpose of ϕ_r . The MAC expresses the degree of correlation between the two vectors ($0 \leq MAC \leq 1$). When $MAC = 1$, there is a complete correlation between the vectors; when $MAC = 0$, there is no correlation.

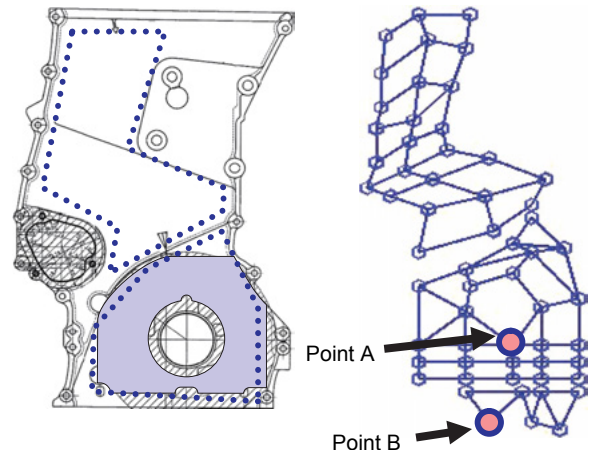


Fig. 5 Chain cover and measurement points

The MAC is usually calculated between multiple modes and expressed as a matrix. Correlation may be autocorrelation, i.e. correlation between the same modes, or cross correlation, i.e. correlation between different modes. In the case of autocorrelation, the lower the level of non-diagonal elements, the higher the orthogonality of the mode shapes, and the higher the degree of accuracy in identifying the mode. In the case of cross correlation, when the value of the diagonal elements is close to 1, there is a similarity between the mode shapes.

In order to objectively evaluate the accuracy of mode shape identification across multiple modes, in addition to formulating the matrix, the mean values of the MAC are evaluated. In the case of autocorrelation, by focusing on the orthogonality of the modes, the mean value of the non-diagonal elements expressed by Eq. (2) is evaluated. In the case of cross correlation, by focusing on the similarity of the modes, the mean value of the diagonal elements expressed by Eq. (3) is evaluated.

$$\text{Mean (Non - diag)} = \overline{\text{MAC}(\phi_r, \phi_s)} \quad (r \neq s) \quad (2)$$

$$\text{Mean (Diag)} = \overline{\text{MAC}(\phi_r, \phi_s)} \quad (r = s) \quad (3)$$

4. Results

4.1. Verification of Accuracy Using Excitation Tests

Figure 6 shows the results of a comparison of head cover mode shapes obtained by applying the OMA and EMA to an excitation test. Figure 6 (a) shows the MAC matrix for the OMA autocorrelation, 6 (b) shows the

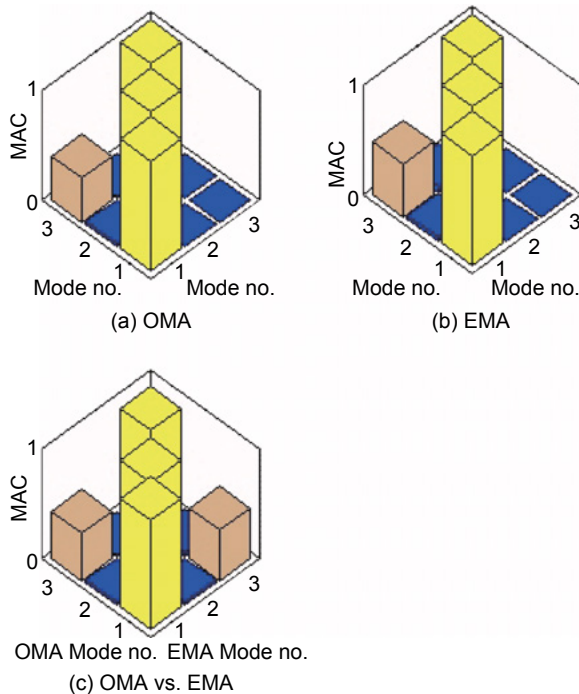


Fig. 6 MAC from excitation test

Table 1 Comparison of natural frequencies of head cover by OMA and EMA

Mode no.	Natural frequency [Hz]		Difference [%]
	OMA	EMA	
1	831	829	0.2
2	901	901	0.0
3	947	947	0.0

MAC matrix for EMA autocorrelation, and 6 (c) shows the MAC matrix for cross correlation between the OMA and EMA. In the case of the OMA and EMA autocorrelation, the MAC of 0.40 and above existed for the non-diagonal elements. As the OMA and EMA results display the same tendency, this can be assumed that the result of insufficient measurement points might be the reason rather than of the difference in accuracy between the OMA and EMA. The magnitude of the diagonal elements in the cross correlation shown in 6 (c) was 0.89 and above, and the mean value of the diagonal elements was high at 0.94. These results show that the mode shapes obtained using the OMA and EMA are similar, indicating that the OMA possesses the same level of accuracy in identifying mode shapes as the EMA.

Table 1 shows the natural frequencies determined by the OMA and EMA. The difference between the natural frequencies of the OMA and EMA was 0.2% or less. The results obtained from the both are almost the same, and thus the accuracy levels of the OMA and EMA in identifying natural frequencies are equivalent.

4.2. Engine Operation Tests

Figure 7 shows the results of a tracking analysis of 36th order vibration (the chain and the sprocket polygon order) of the chain cover. The measurement points were Point A and Point B shown in Fig. 5. Vibration peaks existed at 2200 rpm (1320 Hz), 2400 rpm (1440 Hz), and 2500 rpm (1500 Hz), and the analysis focused on these frequencies.

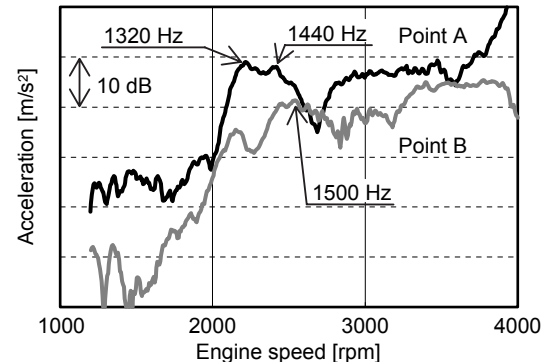


Fig. 7 36th order vibration of chain cover

Figure 8 shows the results of the ODS using tracking analysis results. Figure 9 shows the mode shapes obtained by applying the OMA to tracking data. The mode shapes shown in Fig. 9 are, respectively, the vibratory shapes of the upper section of the crankshaft through hole, the lower part of the crankshaft through hole, and the oil pan. All ODS results shown in Fig. 8 include the vibratory shapes of the lower part of the crankshaft through hole. This is assumed to be because modes cannot be separated in ODS, and are therefore displayed superimposed.

Figure 10 shows results for the autocorrelation MAC. 10 (a) shows the MAC matrix for the mode shapes obtained from the results of tracking analysis, 10 (b) shows the MAC matrix for the mode shapes obtained from averaging, and 10 (c) shows the MAC matrix for the ODS autocorrelation using the results of tracking analysis. Table 2 shows the mean value of the non-diagonal elements of MAC for autocorrelation. The maximum non-diagonal element for the OMA using tracking analysis was 0.11, the maximum for the OMA using averaging was 0.31, and the maximum for ODS was 0.66. The OMA using tracking analysis also displays the lower mean figure for non-diagonal elements in Table 2, with 0.06 compared to 0.20 for the OMA using averaging and 0.44 for the

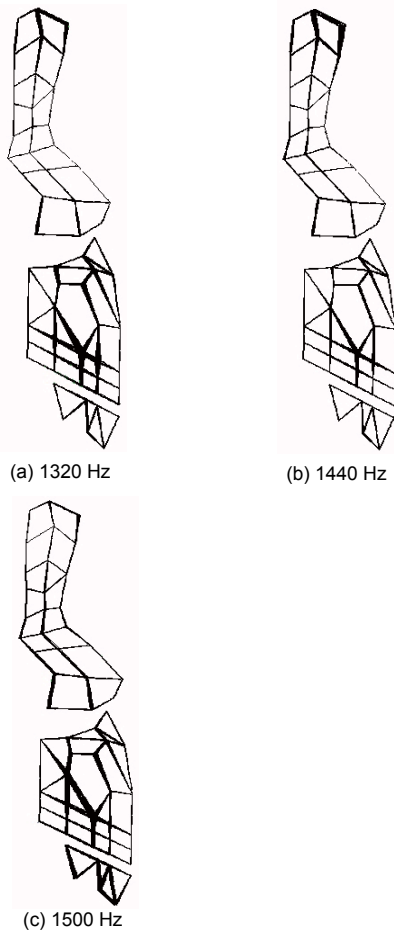


Fig. 8 ODS of chain cover

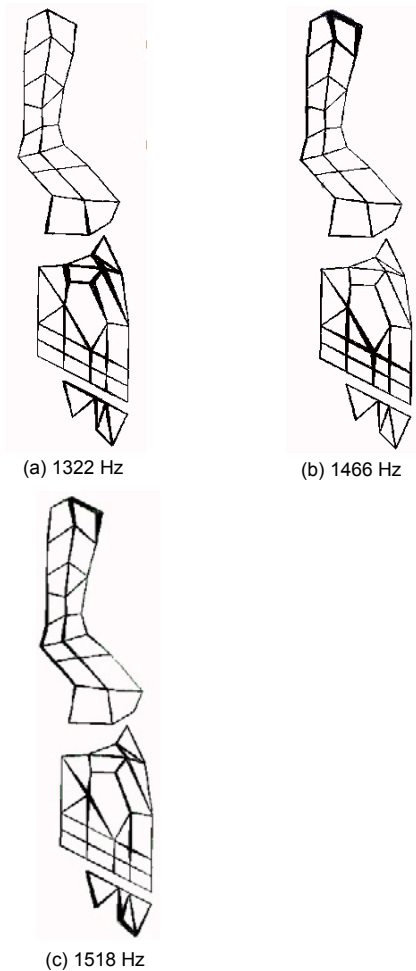


Fig. 9 Mode shape of chain cover

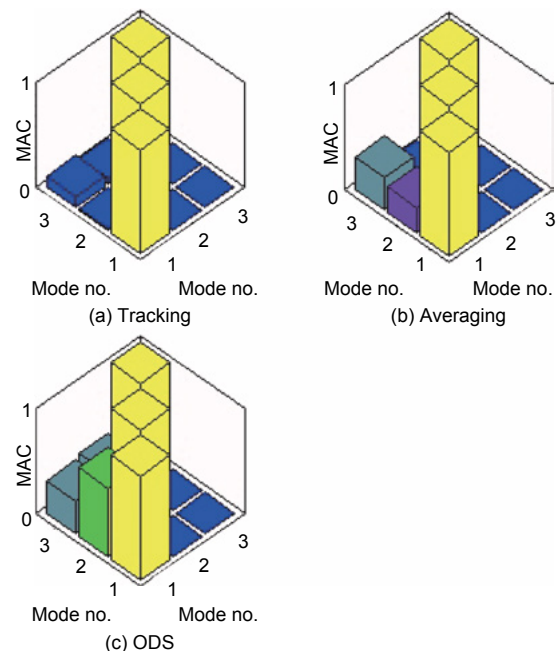


Fig. 10 MAC from operational condition

Table 2 Comparison of MAC mean from operational condition

Mean of non-diagonal MAC		
Tracking OMA	Averaging OMA	Tracking ODS
0.06	0.20	0.44

Table 3 Comparison of natural frequencies of chain cover by OMA from tracking and averaging

Mode no.	Natural frequency [Hz]		Difference [%]
	Tracking	Averaging	
1	1320	1322	0.2
2	1425	1466	2.9
3	1512	1518	0.4

ODS. The natural modes obtained by means of the OMA using tracking analysis demonstrate higher orthogonality and the mode shapes can be separated while the orthogonality of the mode shapes generated by the OMA using averaging and the ODS is lower and the shapes cannot be separated. The accuracy of the OMA using tracking analysis in identifying mode shapes is therefore higher than that of the conventionally used ODS or OMA using averaging.

Table 3 shows a comparison of natural frequencies obtained from the vibration data using tracking analysis and averaging. There is a 2.9% difference in the natural frequency identified in mode 2, but this would not represent an issue in practice. These results indicate that even the conventional OMA possesses a sufficient accuracy in identifying natural frequencies.

5. Conclusion

The order tracking vibration data has been used in the OMA to increase the accuracy of identifying natural modes in an operating engine. Studies focusing on mode shapes generated the following conclusions:

- (1) In a study using excitation tests with pulse waveforms, the OMA using the order tracking vibration data generated similar mode shapes to the EMA using frequency response functions, and demonstrates that the new method possesses identical accuracy with that of the EMA in identifying modal parameters.
- (2) In a study using the vibration of the chain cover in an operating engine, the orthogonality of natural modes obtained by means of the OMA using the order tracking data was higher than that of natural modes obtained by means of the conventional OMA that averages vibration and the ODS which is used as a substitute for the OMA, and demonstrated that the new method increases the accuracy of mode shape identification.

Only the objective natural modes under actual engine operating conditions can be identified by this method. In future, the application of this method to higher-frequency domains in which the mode density is higher can be expected to increase analysis efficiency.

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Motoyasu SAKAGUCHI

Development of Simulation Technology for Dynamic Behavior of Assembled Crankshaft Systems in Single-cylinder Engines

Tatsuya IWAMOTO* Junya SAITO*

ABSTRACT

When simulating an assembled crankshaft system of the type often used in single-cylinder engines, it is important, in terms of the accuracy of calculations, to adequately represent the influences of crankpin press fitting, the dynamic characteristics of the alternating current generator (ACG) rotor directly mounted to the crankshaft, and the characteristics of the rolling bearing. Actual measurements were performed to verify these characteristics, and a mechanical dynamic analysis model, which accurately represents the dynamic characteristics of the crankshaft system, was established. Also a rolling bearing model that represents the characteristics of three-dimensional elastic contact was developed. Using these in a mechanical dynamic analysis, the dynamic behavior of assembled crankshaft systems was simulated with enough practical accuracy for strength evaluations.

1. Introduction

Because assembled crankshafts are used in a wide range of models in single-cylinder engines powering motorcycles, all-terrain vehicles, and other vehicles, technology for predicting the durability of assembled crankshafts is important in development of single-cylinder engines. However, conventional simulation techniques for the assembled crankshafts offer inadequate accuracy due to the mounting of the crankshaft (in which one or both ends of the crankpin is press-fitted into the crank web) and due to the motion characteristics of the alternating-current generator (ACG) rotor, which is directly mounted to the crankshaft. Given the wide use of rolling bearings, simulation accuracy is also dependent on techniques for bearing modeling. In contrast, for the development of one-piece crankshafts used in multi-cylinder engines, durability prediction has been enabled early on by the dynamic simulation technologies^{(1), (2)} that employ mechanical analysis.

This paper discusses the enhanced prediction accuracy by simulation that is based on validation using unit measurements of the influences particular to assembled crankshafts, and based on mechanical dynamic analysis models that incorporate those properties.

2. Characteristics of Assembled Crankshafts

Figure 1 shows the structure of a typical assembled crankshaft. Assembled crankshafts used in single-cylinder engines exhibit the following five characteristics:

- (1) One or both ends of the shaft are press-fit between the crankpin and crank web.
- (2) Rolling bearings are mainly used for bearings.

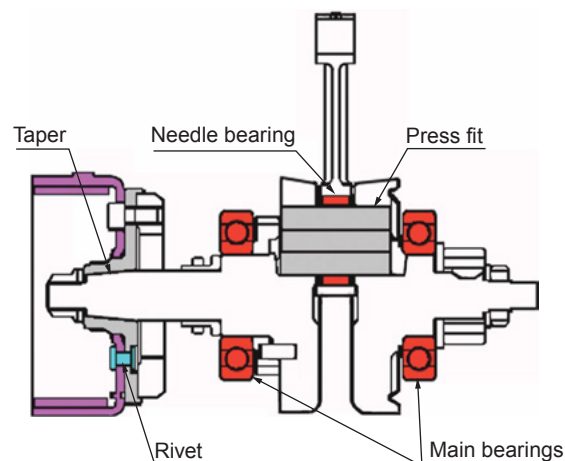


Fig. 1 Structure of assembled crankshaft

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- (3) With the ACG rotor having inertia mass almost equivalent to that of the crankshaft, it is thought that the dynamic properties of the ACG rotor affect crankshaft behavior.
- (4) Drive train units, such as a centrifugal clutch and a belt converter, are often directly connected to the crankshaft.
- (5) Needle roller bearings are used for the sliding portion between the connecting rod and crankpin.

Contact such as the press-fitted structure of the crankpin and rivet joints of the ACG rotor likely affect the natural frequency and damping more than in the case of one-piece construction. Also affecting crankshaft behavior is the dynamic characteristics of the drive train unit that is mounted to the shaft. In addition, because a rolling bearing is supported by a rolling element unlike a plain bearing, which is supported by an oil film, the bearing's quantity, size, and shape can affect bearing stiffness and damping characteristics.

After conducting validation bench tests on changes in dynamic characteristics with an actual crankshaft, a technique for using mechanical dynamic analysis models to represent the characteristics of press-fitting and rivet joints used in typical assembled crankshafts has been developed. To ensure applicability to the broad range of single-cylinder engines currently being mass-produced, mechanical dynamic analysis models for deep-groove ball bearings, cylindrical roller bearings, and needle roller bearings have been developed.

3. Results of Unit Measurements of Crankshaft Dynamic Characteristics

3.1. Characteristics Due to Press-Fitting of the Crankshaft

To determine how differences in crankshaft press-fit margins affect the crankshaft's dynamic properties, a test piece was fabricated and changes in dynamic properties in hammering tests were measured. Figure 2 shows the structure of the test piece. As for press-fitting of the pin, typical press-fit margins seen in mass-produced crankshafts were used to determine how changes in press-fit margins affected natural frequency and the damping ratio. Also, the original crank shaft that integrated the press-fit areas was fabricated, and the change between press-fit types and a one-piece type was verified.

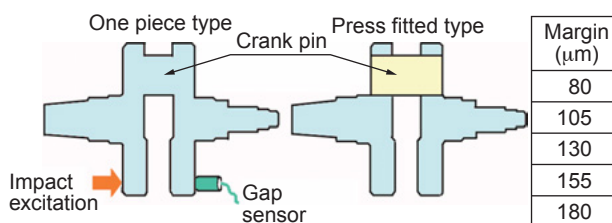


Fig. 2 Figure of press fit test piece

Figure 3 lists the rates of change in natural frequency relative to the one-piece crankshaft due to the differences in press-fit margins in the major vibration modes; Fig. 4 shows the changes in damping ratios. These results show that changes in press-fit margins produced almost no difference in natural frequency, regardless of the vibration mode, and that differences in natural frequency instead resulted from the presence of press-fit components. These differences were particularly noticeable in modes 1 and 4 (bending mode of pitching direction), with reductions of up to 2% compared to the one-piece crankshaft. Change in the damping ratio was greater in the press-fit crankshaft as compared to the one-piece crankshaft but small in absolute values, so that one can conclude that press-fitting had almost no influence.

3.2. Characteristics Attributable to Fastening Structure of the ACG Rotor

Structural features that greatly affect the ACG rotor's dynamic properties include the structure of taper that is bolted to the crankshaft and the rivet structure joining the boss and rotor. To verify the changes of dynamic properties by these connections, test pieces simulating an ACG rotor were fabricated and subjected to the same hammering test performed for press-fittings to measure frequency characteristics and the damping ratio. Figure 5 shows the shapes of the test pieces; Table 1 shows the specifications of test-piece combinations measured.

Figure 6 shows rates of change that each combination of connections produced in natural frequency relative to the one-piece equivalent in the major vibration modes. These results show that the tapered shaft generated the

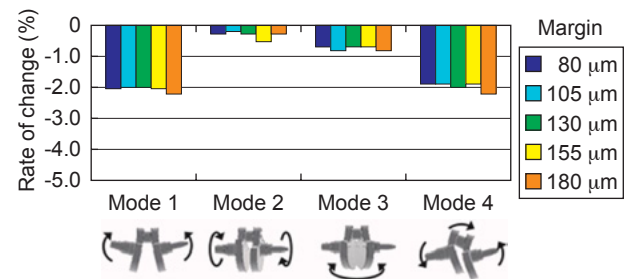


Fig. 3 Rate of eigen frequency with press fitting

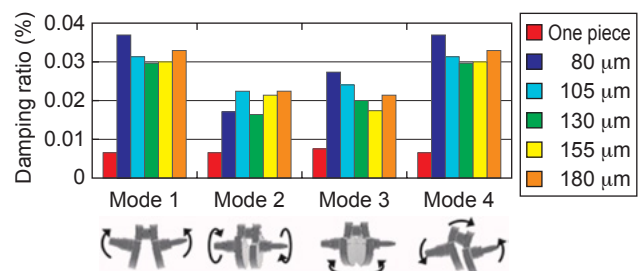


Fig. 4 Damping ratio with press fitting

Table 1 Specifications of ACG rotor test pieces

Type	Between rotor and boss	Between boss and shaft
I	One piece	One piece
II	One piece	Taper & Thread tightening
III	Riveting	Taper & Thread tightening

natural frequency to differ by only 2–3%, depending on the vibration mode. In contrast, the rivet joints generated a sharp reduction (relative to the one-piece equivalent) in the natural frequency. Varying according to mode, this decrease was the largest in Mode 3 (axial directional vibration), in which the rivet joints alone influenced a 14% reduction.

Figure 7 shows the results of unit measurements of damping ratio. Trends in the damping ratio were found to be similar to those of natural frequency, with the shaft's taper having almost no influence but the rivet connection

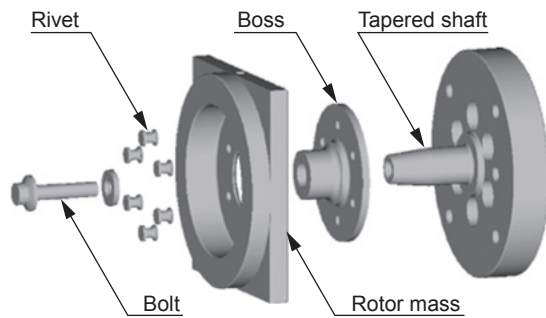


Fig. 5 Structure of ACG rotor test pieces

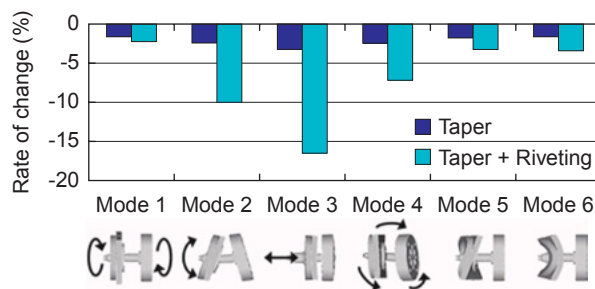


Fig. 6 Rate of change of eigen frequency with taper and rivet

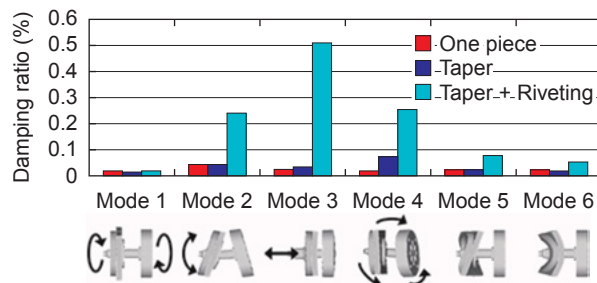


Fig. 7 Damping ratio with taper and rivet

having an influence. Change in the damping ratio was the greatest in mode 3 (in which change in natural frequency was also the greatest)—roughly 0.04% without the rivet versus an increase to 0.5% with the rivet, or an approximately tenfold increase.

4. Construction of the Mechanical Dynamic Analysis Model

The methodology with which an elastic model is constructed is an important issue when conducting a mechanical-dynamic-analysis-based simulation of the dynamic behavior of a crankshaft. Using the bench test described above to clarify the influences of press fitting and riveting on a crankshaft's dynamic characteristics, a finite element model (FE model) with elastic property was created to mimic the characteristics of press fitting and riveting to enhance accuracy of the simulation.

With regard to rolling bearings, a mechanical dynamic analysis model was developed to encompass not only standard ball bearings but also roller bearings and needle bearings (widely used at the big end of connecting rods, etc.) so as to assure applicability of this simulation to a diverse range of engine specifications.

4.1. Modeling of the Press-fitted Part of the Crankpin

Because actual testing showed that press-fitted components lower natural frequency, the elastic model was devised to account for the stiffness characteristics of press-fitted components. With the crankpin and right and left halves of the crankshaft represented with a separate FE model, the press-fitted surfaces were connected with spring elements to represent the change in stiffness due to press fitting. The spring elements were configured into a cylindrical coordinate system whose axes pass through the crankpin center, and stiffness characteristics were given in three axial directions. Regarding spring stiffness characteristics, actual test results were compared with the natural frequency determined by the FE model, and the optimum value was determined so as to minimize the error in each vibration mode.

Figure 8 gives the calculations made with this crankshaft FE model technique and the error in actual natural frequency. As the figure shows, this technique enabled an accuracy of within 1% in each major mode.

4.2. Modeling of ACG Rotors

Because of the considerable influence of riveting on the axial-direction vibration mode, the FE model developed to simulate the ACG rotor was made to represent the stiffness characteristics of the rivet–boss contact face. Three-axis spring connections were used for each rivet, and for the outermost edge of the boss–rotor contact face, a one-axis spring was added in the contact direction to make the connection. Single rivet stiffness characteristics were determined and used in defining the

rivet's three-axis spring characteristics. Optimum values for the uniaxial spring property at the boss outer diameter were determined according to correlation with the measurements. Figure 9 compares the calculated and measured results for the aforementioned ACG rotor measurement test piece. This shows that the technique is accurate to within 1.4%.

4.3. Treatment of Crank System Damping

Our measurements have shown that the ACG rotor and press fitting increase damping in the crankshaft system. However, in each actual test, measured damping values were small (Fig. 4) and not different from that of the one-piece crankshaft. As not only press-fitted components and the ACG rotor, but also such drive train units as the centrifugal clutch, belt converter, and torque converter are mounted directly to the crankshaft, the damping characteristics affecting the crankshaft can vary greatly, depending on the actual mechanism of each. Therefore, the damping value of the overall crankshaft system must be set according to each combination of drive train units. Currently, the only method for doing so is to estimate based on actual measurements. Measuring several types

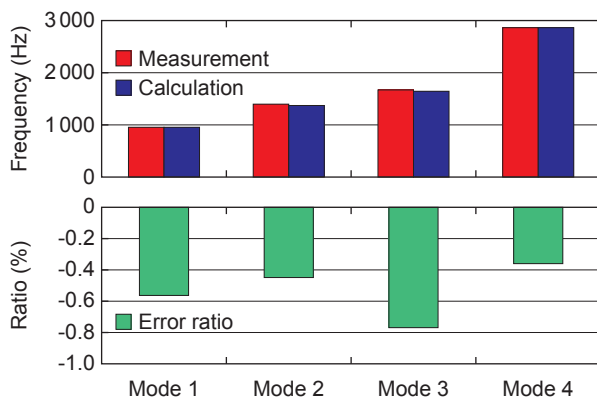


Fig. 8 Comparison of measured and calculated eigen frequencies with press fit

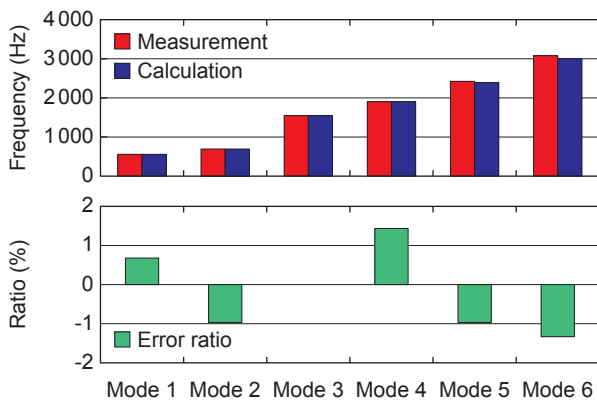


Fig. 9 Comparison of measured and calculated eigen frequencies with taper and riveting

of engines with different drive train units, it was found that single-cylinder motorcycle engine crankshafts had a damping ratio of approximately 1–5%. Based on these findings, simulations in which actual measured damping ratios for drive train units mounted to a crankshaft were assigned to the crankshaft system, were conducted.

4.4. Modeling of Rolling Bearings

Which method to use to model bearings is an important issue in simulating the dynamic behavior of crankshafts. For one-piece crankshafts, plain-bearing models based on elasto hydrodynamic lubrication theory are commonly used, but for assembled crankshafts, which contain rolling bearings, such three-dimensional elastic contact must be handled efficiently in the mechanical dynamic analysis^{(3), (4)}. The roller bearings used in single-cylinder engines are usually ball bearings, though roller bearings are also used, depending on the engine type. As the development of ball-bearing models has already been reported⁽⁵⁾⁻⁽⁷⁾, this paper discusses the development of a mechanical dynamic analysis model for roller and needle bearings.

4.4.1. Roller bearing model

For roller bearings, a crucial issue is how to represent large changes in contact points that occur with axis movement, as well as the corresponding change in stiffness. Because a roller bearing fundamentally has no component that constrains in the axis direction, for practical reasons it is treated as if no stiffness occurs until the rolling element contacts the side wall of the outer or inner raceway. As for the elastic characteristics of the contacted area, the finite-element method and general-purpose structural-analysis software were used to first calculate the relationship between load and reaction force for each contact surface in a single rolling element. Using these results, the variable stiffness function for each state as contact progresses was determined for different angles of contact β (Fig. 10). Saved as a bearing characteristic file, this data was referenced from the mechanical dynamic analysis software while determining load

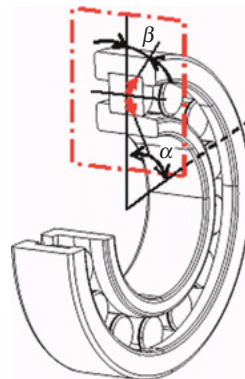


Fig. 10 Roller bearing model

corresponding to the displacement of each rolling element. Displacement of each rolling element was determined from the relative angle and distance between the inner and outer raceways according to the state of contact between them. Relative position along the circumference of each rolling element was determined geometrically according to the rotational difference between the outer and inner raceway⁽⁸⁾. In the mechanical dynamic analysis software, the resultant force of each rolling element's load was treated as a bearing reaction.

Next, an impact shaker tester (Fig. 11) was used to test the actual parts to verify that the bearing model was accurate. The crankpin was shaken by generating an impact load through breaking the bolt, and the resulting radius-direction displacement and axis-direction displacement on the right and left crank web were computationally compared with the results of measurement. The bearings used were a roller bearing in combination with ball bearings as the thrust bearings. Figure 12 lists the displacement data, which shows that the simulation accurately re-created actual behavior.

4.4.2. Modeling of needle bearings

The needle bearing used for the big end of the connecting rod can be treated as a case of extreme

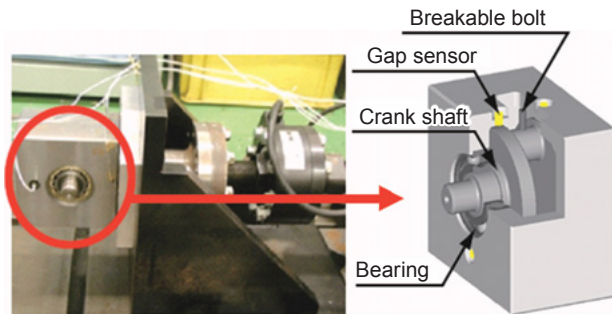


Fig. 11 Impact shaker tester

dimensions in the rolling element of a roller bearing. However, press fitting deforms the crankpin — which contains the bearing's rolling surface — into a barrel shape, with curvature in the cylindrical center axial direction. Consequently, for the inner raceway, the needle bearing model re-created this curvature in the axis direction.

5. Comparison with Actual Measurements

A single-cylinder engine typically contains two bearings, but because of differences in factors such as how crankshaft output is taken out, some single-cylinder engines contain three or more bearings. To verify the appropriateness of the mechanical dynamic analysis models discussed up to this point, engines with differing numbers of two types of bearings were used to test model accuracy.

5.1. Two-bearing Engine

Actual measurements and simulation results were compared for a single-cylinder engine with ball bearings on the right and left, the most common type of single-cylinder engine (Fig. 13). Because crankshaft output is applied directly through the gears, this type of engine is said to have the lowest damping attributable to drive train units.

Displacement in the right and left crank web in the normal direction of the center axis, as well as whirling displacement of the ACG, was measured with a gap sensor. Strain was measured by attaching strain gauges to the crank journal corner fillet on the ACG rotor side of the crankshaft.

For the simulation models, the aforementioned FE model designed to take crankpin press-fit characteristics and riveting characteristics into account was used, and a ball bearing model was used for the bearing.

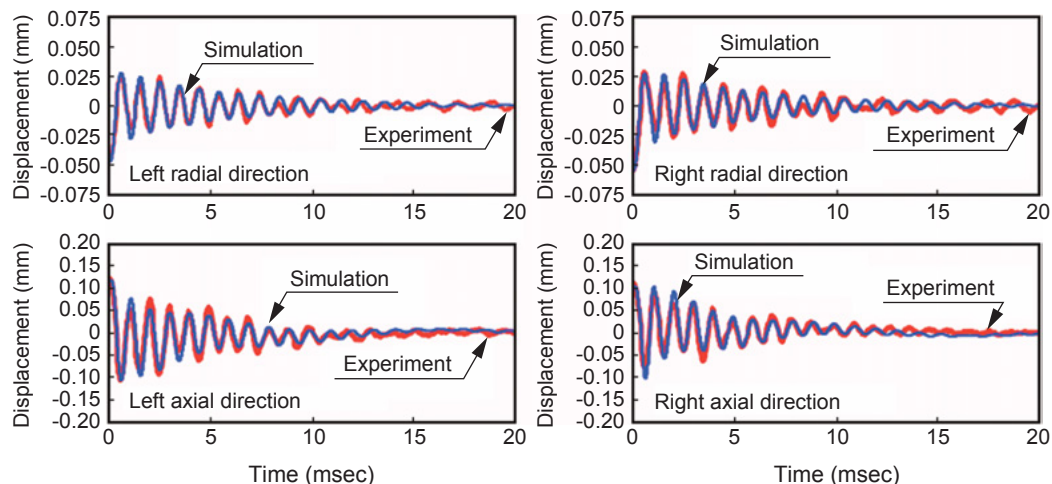


Fig. 12 Comparison of experiment and simulation of impact shaker test

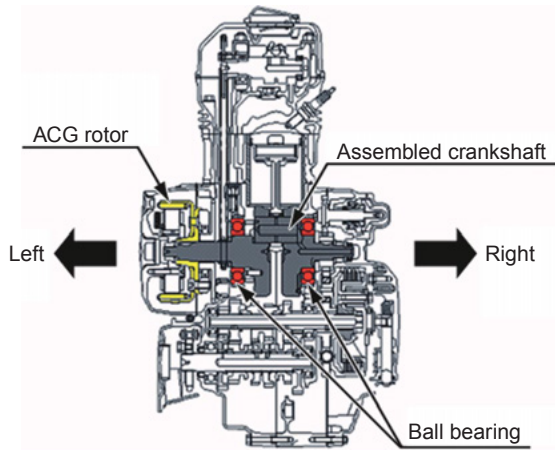


Fig. 13 Structure of 2-bearing type engine

Figure 14 shows the actual measurements of left web cylinder-direction displacement and strain in the crank journal corner fillet, along with the corresponding results for simulation. These results demonstrate the favorable correlation between simulation and actual measurements and, therefore, the appropriateness of this simulation technique.

In Fig. 15, actual strain measurements in the crank journal corner fillet are compared with simulations made with a conventional technique and those made with the developed method. Figures 15 (a) and (b) compare measurements with the results, respectively, of simulation using a model of conventional spring damper bearings and simulation that treated the press-fit part of the crankshaft as a single piece. These results show that the developed technique is superior in accuracy than that of the conventional technique.

5.2. Four-bearing Engine

This special type of single-cylinder engine, which is

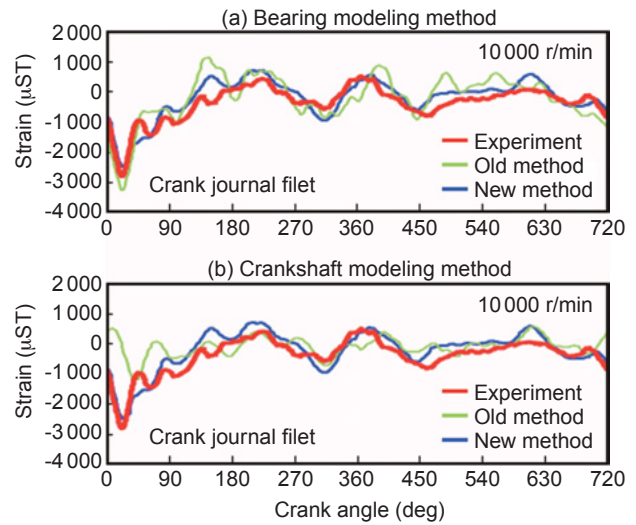


Fig. 15 Comparison of old and new methods of 2-bearing type engine

used in all-terrain vehicles, has two main bearings as well as two additional bearings that support the crankshaft at both ends. These engines sometimes have a torque converter as the drive train unit, and various additional mechanisms are installed on the crankshafts, resulting in a complex geometry. Consequently, actual measurements were conducted to compare the results for correlation with the results of simulation.

Figure 16 is a schematic of the engine in question. As with the two-bearing engine, displacement in the normal direction of the center axis at both crank webs, displacement in the ACG rotor in the normal direction of the center axis, and torsional strain in the front and rear axes were measured.

The simulation model is depicted in Fig. 17. This engine had a roller bearing as the front main bearing and ball bearing for the rear main bearing. At both ends, ball

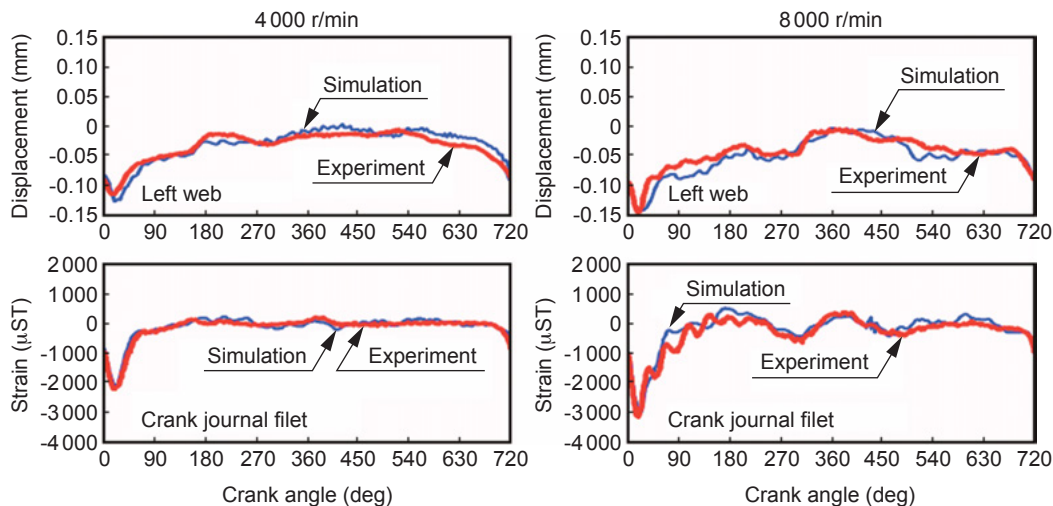


Fig. 14 Comparison of experiment and simulation results of 2-bearing type engine

bearings also supported the space between the ends and the engine case cover.

Measured and calculated displacements are shown in Fig. 18, which indicates the favorable correlation for front and rear crank web displacement and ACG rotor displacement. The larger measured displacement of the ACG rotor is due to the sensor passed through a balance-adjustment hole. Overall, these results demonstrated that this simulation technique accurately reproduces crankshaft bending behavior, regardless of the number of bearings.

Figure 19 shows measured and calculated torque converted from torsional strain. This data reveals a slight discrepancy between the measured and calculated values.

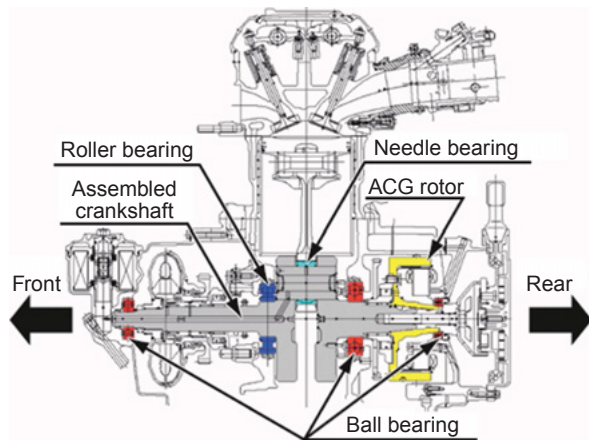


Fig. 16 Structure of 4-bearing type engine

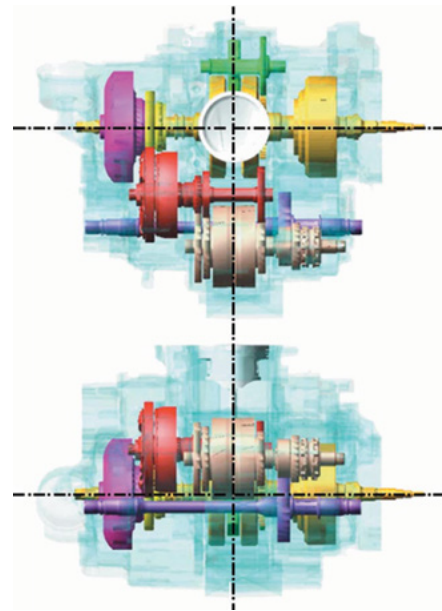


Fig. 17 Mechanical dynamic analysis model for 4-bearing type engine

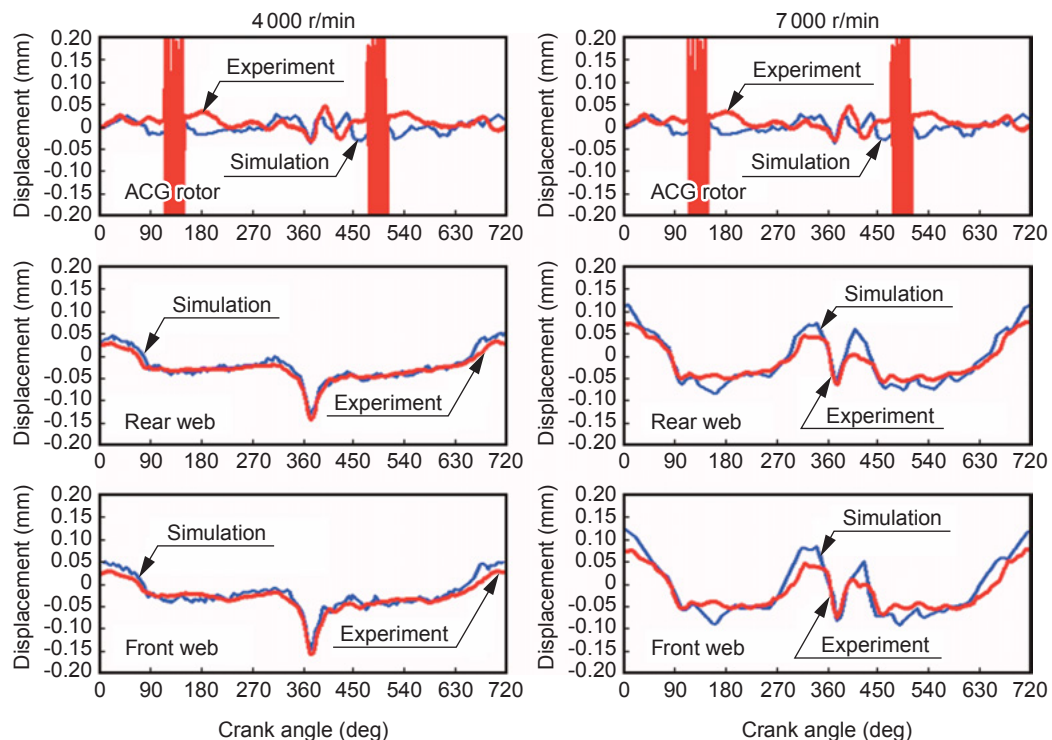


Fig. 18 Displacement of crank-web and ACG rotor of 4-bearing type engine

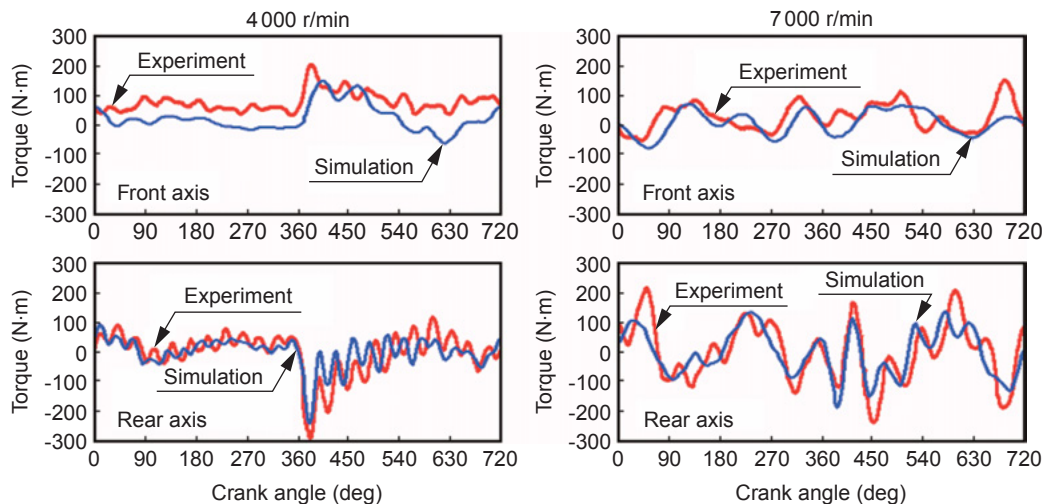


Fig. 19 Axial torque of 4-bearing type engine

6. Conclusion

- (1) The following actual characteristics necessary for simulating the dynamic behavior of assembled crankshafts were verified.
 - The natural frequency of assembled crankshafts was approximately 2% lower than that of one-piece crankshafts, regardless of the crankpin's press-fit margin. However, press fitting had almost no influence on the damping ratio.
 - The taper of the ACG rotor decreased the natural frequency approximately 3% compared to the one-piece crankshaft but had almost no influence on the damping ratio.
 - Rivet joints decreased natural frequency up to 14% and increased the damping ratio approximately tenfold.
- (2) A method was established to enable the construction of an elastic crank model for mechanical dynamic analysis that accounts for the dynamic characteristics of the crankshaft's press-fitting and rivet joints.
- (3) A rolling bearing mechanical dynamic analysis model considering the three-dimensional elastic contact characteristics of differing types of rolling elements was developed.
- (4) Using the mechanical dynamic analysis models referred to in (2) and (3) above, two types of single-cylinder engines were simulated, each with a different number of bearings, and compared the computed results with those of actual measurements, thereby demonstrating that the technique provides sufficient accuracy for practical applications in evaluating the strength of assembled crankshafts.

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