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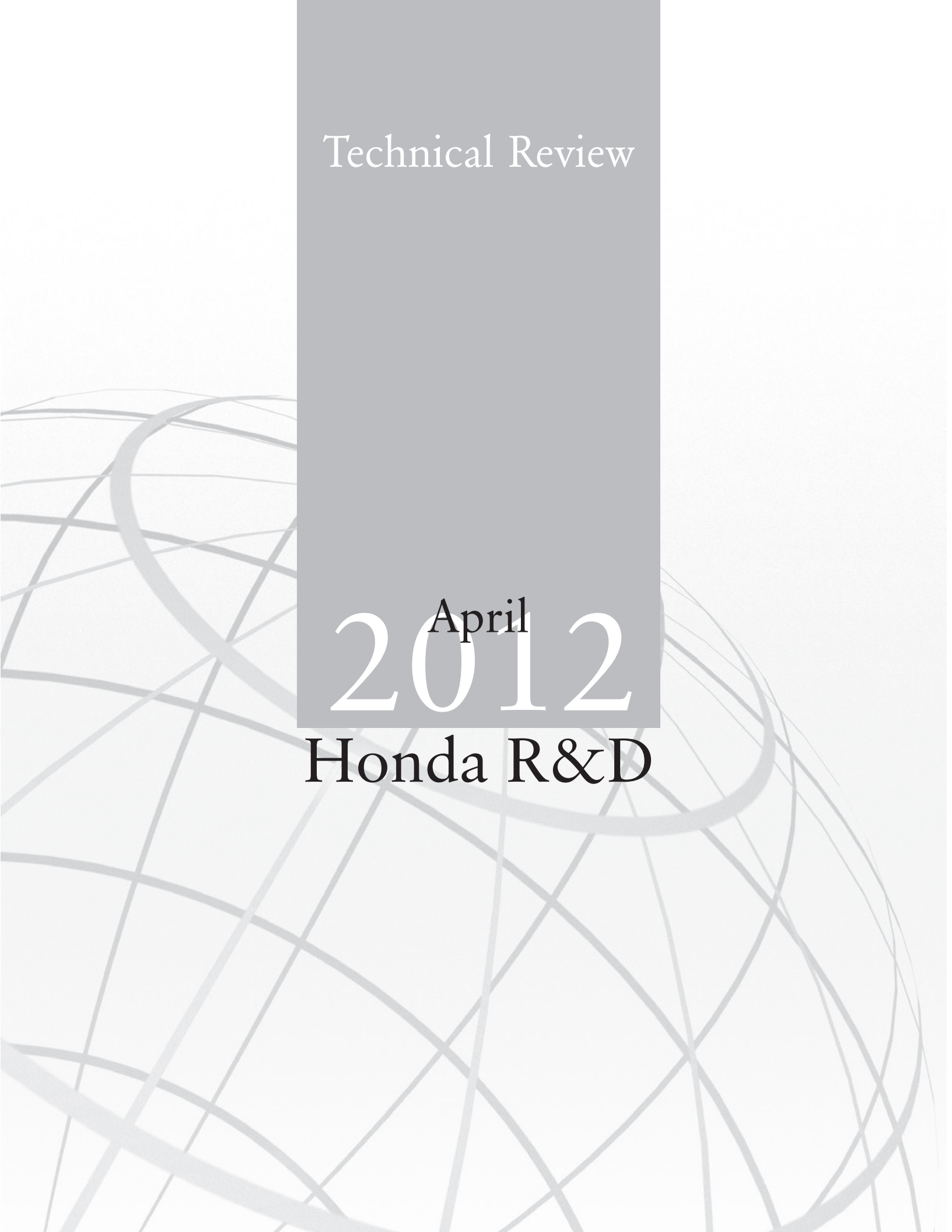


Comment on the Cover

An essential component of any Honda product is excitement. With their breathtaking appeal, Honda products traverse the world, delivering joy. The cover of this issue expresses Honda's desire to create similarly stimulating products.



Motorcycle R&D Center
Noriyuki Mori

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

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Styling Design of New CIVIC

Toshiyuki OKUMOTO*

Yoshiya KITAJIMA*

Akihiro HAYASHI*

Yoko TAJIMA*

ABSTRACT

The new CIVIC was developed under the grand concept of “Distinctive Smart CIVIC” following the keywords “Clean & Energetic” and “Smart Technology” (new values that anticipate the era) and “Fun to Drive/Fuel Efficiency” (the basic performance expected of a car).

The exterior design is based on the concept “Clean & Energetic” and expresses a unique dynamic monoform framework that looks powerful and clear. The interior design is based on the concept “Smart Futuristic Cockpit” and embodies a functional, spacious and advanced cockpit. The colors and fabric were also developed to bring out these characteristics.

1. Introduction

Since the arrival of the first-generation CIVIC in 1972, successive generations have offered innovative packages, leading-edge technologies, and new value for each era so that customers in 170 countries of the world, and particularly the US, have loved each model.

The previous generation of the CIVIC, the eighth overall, offered distinctive new value with a streamlined silhouette and a monoform framework creating an interior that evoked a sense of spaciousness and the near future. In the main market of the US, several generations including the baby boomers have endorsed such value, as has Generation Y, which particularly hopes to lead a smart lifestyle. The advanced technology and refined exterior design were highly praised.

Generation Y is the main target of the recent full model change. The aim of the designers was to make Generation Y glad to own the CIVIC by building new value with an evolved interface that makes their lifestyle rich and smart through the new CIVIC’s unique and dynamically evolved monoform framework. Figure 1 shows the grand concept of the new CIVIC.

2. Design Goals

The styling design of the new CIVIC was developed with the keywords “Clean,” “Smart” and “Refreshing” so that the main target Generation Y customers could lead rich and smart lifestyles.

The aim was a novel and advanced interface which the driver could instantly recognize and intuitively operate without having to think about it, and a roomy interior space beyond that of the car’s class to ensure that driving the car is refreshing. Another aim was to achieve a design with unique presence and new value by creating a dynamic monoform framework characterized by a large cabin that makes the roomy interior space obvious from the outside.

3. Package Layout

The package layout was based on the concept “Distinctive Package,” aiming to achieve a unique monoform framework with presence and a spacious and refreshing interior beyond that of the car’s class. To prevent a loss of fuel efficiency, the exterior size was kept the same as the previous CIVIC generation, but to achieve the concept within that size the designers endeavored to make the interior space as efficient as possible while pursuing



*DQR: Durability, Quality and Reliability

Fig. 1 Grand concept

an open and visually roomy feel with a wide front field of vision (Fig. 2).

To make the interior space efficient they went back to Honda's starting point of vehicle manufacturing - "Man maximum, mechanism minimum." In doing so they specified, in 1 mm units, the dimensions of those parts in which people are particularly sensitive to roominess.

A look at how the CIVIC is used in the US market indicates that it is primarily by the driver alone for work and school commuting, etc. To provide comfort in the front seat space and impart the feel of an automobile one rank higher, the space between passengers on the left and right was increased by 10 mm, and at the same time the interior width was increased by 75 mm by making the door lining thinner.

The rear seat space, moreover, is important to the global market. The distance between front and rear passenger is about the same as in the previous CIVIC generation, but a lower floor makes it possible to give back seat passengers more foot space, while improvements to the shape of the front seat-backs increase knee space for a roomier back seat space (Fig. 3).

With the aim of further increasing the roomy feel, the lower end of the front pillar was positioned forward similar to the previous CIVIC. The front pillar was made slender and improved in cross-section ensuring a wide and open-looking forward field of vision.

Meanwhile, the front quarter-window was positioned on the door side to widen the field of vision. This offers a refreshing and open space with a clear view above the instrument panel to the left and right (Fig. 4).

The lower end of the front pillar is positioned forward to also contribute to the visual roominess that one can feel from the back seat. Further, the front seat-back was shaped to appear smaller so that the interior space looks wide open even from the back seat.

The monoform framework developed not only presents such a refreshing space, but also embodies the CIVIC's unique proportions - namely the long cabin, short nose and short deck.



Fig. 3 Rear space



Fig. 4 Front space

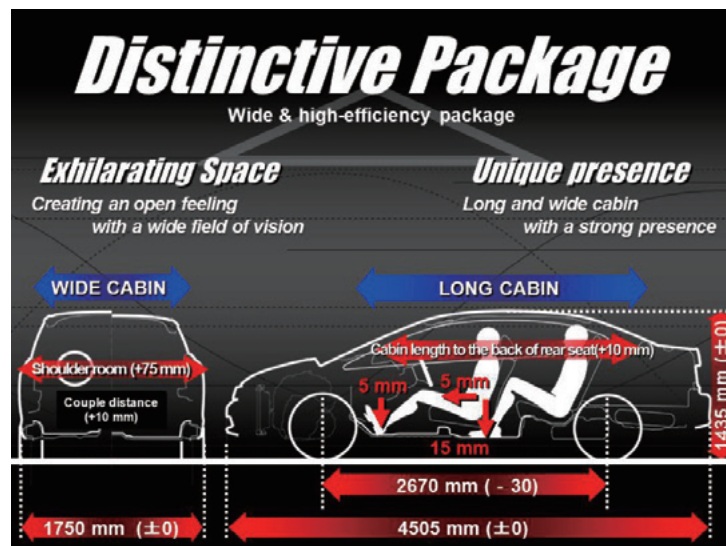


Fig. 2 Package layout

4. Exterior Design

The exterior was styled with the Clean & Energetic concept and expresses a unique dynamic monoform framework that looks powerful and clear. The styling emphasizes form continuity flowing from the engine hood to the cabin and trunk. The basic motif is a simple, clean monoform framework that looks honed and integrated (Fig. 5).

A surface has been built with an image of energy bubbling out from the inside of this clean body. An emphasis on the modulation and lively feel of the tension-filled body enabled a design evolved beyond that of the previous CIVIC generation. What particularly expresses this is the dynamic form modeling in which the surfaces twist and project outwards from the side shoulder line to the rear fender.

There is also a character line running down the lower part of the body side that gives a nimble and casual impression. Additionally, the lower end of the front pillar is positioned forward, making the side window graphics look long from front to back. This makes the interior space appear refreshing as well as gives the body a low, sporty bearing (Fig. 6).

The front view's design aimed to impart the feel of an automobile one rank higher by emphasizing the wide feel. The corner of the front bumper was modeled to project outwards, which achieves an automobile rank with modulation, a lively feel, and a stately presence. The front mask is designed as a horizontal line connecting the front

grill and headlights with a long-slit expression and intrepid-looking face. The front face succeeds to the identity of the previous CIVIC generation so that it is immediately recognizable as a CIVIC (Fig. 7).

The rear view features tension-filled modeling with continuity from the trunk center to the outside and emphasizes roundness through the rear combination lamp that wraps around to the body side. The outer lens's outside corner is designed to have an edge shape that optimizes air flow on the body side face and boosts aerodynamic performance.

The rear bumper, like the front, has corners that project outwards to emphasize the wide look and a design with lively feel that looks like an automobile one rank higher (Fig. 8).



Fig. 7 Front design



Fig. 5 Exterior design concept



Fig. 8 Rear design



Fig. 6 Final rendering

5. Interior Design

The interior was styled on the concept of "Smart Futuristic Cockpit," seeking to be an advanced driver's cockpit and a functional, roomy interior space (Fig. 9).

The cockpit panel was arranged so that the meters, where the driver's line of vision is most likely to settle, are

set back the farthest, and the surface modeling gives a sense of vigor from front to back in order to draw the occupant's vision. As it shifts toward the passenger side, the instrument panel is designed with a large curve as if it had been scooped out with a spoon. Both the upper and lower parts of the door lining use negative faces to achieve the greatest interior width in the class. The result is refreshing and clean form modeling that gives a feel of expansiveness in the horizontal direction (Fig. 10).

The design concept of the control system and display was to place driving interface including the speed meter in front of the driver, and information interface including the audio information display at the center of the instrument panel. Operability of the instrument panel was enhanced by positioning important displays on the upper section closer to the outside view, and control system near the driver's hands (Fig. 11).

The upper level of the multiplex meter, which offers excellent visibility, is used for the "Multi-Information Display" that provides a variety of information in an easy to understand way. The simple graphic design helps the screens providing fuel efficiency, and audio and driving support information to evolve into a functional, advanced interface that minimizes movement of the line of vision while cruising, and offers excellent instant recognition.

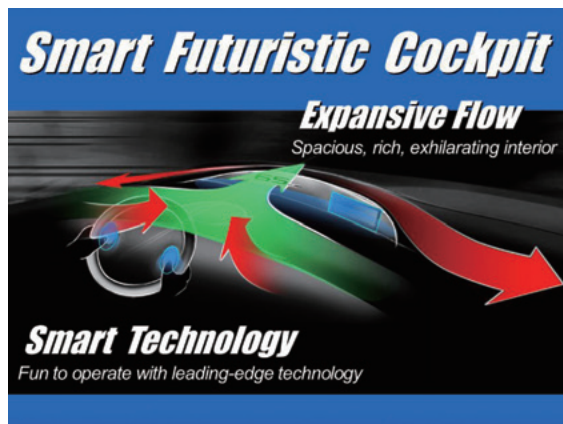


Fig. 9 Interior design concept

Additionally, a round cross-switch⁽¹⁾ has been used as the audio steering wheel switch for the first time in a Honda. By using a design that makes it easy to imagine the screen display content and switch operating direction, this smart operating system can be used intuitively without having to think about it or look at what the fingers are doing (Fig. 12).

The front seat has sufficiently thick parts where it supports the lower back from the side, yet it makes the space seem bigger with its smooth, round shape where it stretches from the lower back to the shoulders at the top. Furthermore, for rear seat passenger comfort, the back of the front seat has a pared-down shape providing more knee space.

The rear seat emphasizes spaciousness with horizontally oriented graphics, and features side supports of a three-dimensional shape, enhancing riding comfort for rear seat passengers (Fig. 13).

Storage space retains the basic specifications of the previous CIVIC generation, including the position and module, and for the first time includes a USB terminal in the console box which allows a music player, etc. to be put away while still plugged in. And because the music player, etc. can be finger-operated with the aforementioned steering wheel switch while cruising, it is possible to enjoy music in smart fashion (Fig. 14).

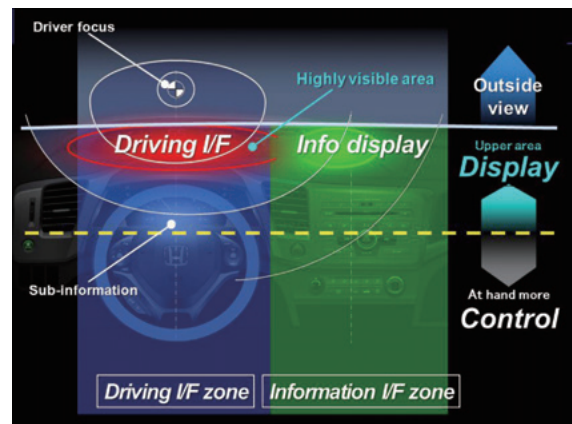


Fig. 11 Multi-information display concept



Fig. 10 Instrument panel



Fig. 12 Drivers cockpit



Fig. 13 Seat design



Fig. 14 Storage design

6. Color and Fabric Design

Colors are coordinated to fuse function and design based on the “Clean & Emotional” concept.

The keywords for the exterior colors are a clear feel like glass and strength of hard metal. There is a lineup of 11 colors, including five new.

The theme color COOL MIST M. is a white metallic color with fine polarized pearl pigment. Soft color-shifting and multilayer-looking highlights invoke a futuristic feel and make the styling presence all the more prominent. GREEN OPAL M. is a color for hybrid cars only and directly expresses an advanced eco-responsible image. Bright aluminum flakes emphasize the rich form, and the color shift from blue to green offers a refreshing look.

CRIMSON P., DYNO BLUE P., and SUNBURST ORANGE P. (which is for the Si sports grade only) were each selected for their lucid character to meet the preferences of various customers. Each color contains coarse, bright aluminum flakes to emphasize three-dimensional shadowing with deep shades and highly chromatic highlights (Fig. 15).

Interior coloring was based on the keywords of advanced and confidence-inspiring character. The lineup of four colors makes the roomy styling more prominent.

The interior theme color, PALE MOSS GRAY, is a new

light color whose essence is pale green. Coordinated to be bright and clean, it expresses a refreshing and advanced feel.

Other interior colors are a modern gray (with a bright gray base), beige (expressing a warm, high-quality feel), and black (expressing a powerful sporty feel and available only on the Si sports grade) (Fig. 16).

The new concept of “Hide and Seek Surface,” which harmonizes organic surfaces and form, was used on the instrument panel and upper part of the door.

Two different organic textures are blended to express a gentle texture. Shape and the reflection of light cause the look to change into various expressions in which the two textures are sometimes independent, sometimes overlapping. This visual effect imparts the look of even greater expansiveness and depth to the already roomy interior space (Fig. 17).

The seat fabric consists of tricot suede, pleasant to the touch, printed with a line-based design that looks as if it were drawn freehand and gives a light and lively feel. The line-based design running freely over the entire seat is repeated three times more often than normal, making the dynamic nature and roominess of the seat styling more prominent (Fig. 18).

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Fig. 15 Exterior color image and line-up

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Fig. 16 Interior color variation

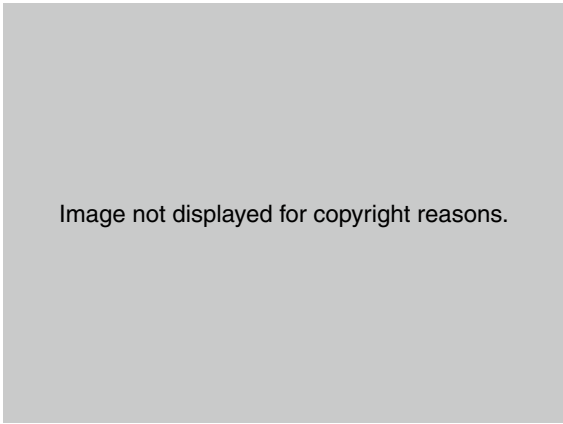


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Fig. 17 Surface design

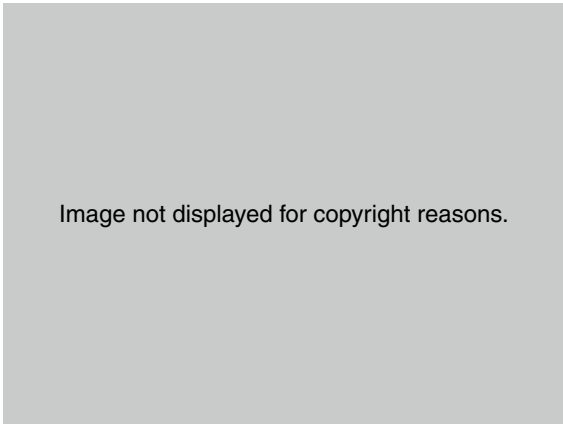


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Fig. 18 Fabric design

7. Conclusion

Forty years have passed since the arrival of the first-generation CIVIC, and customers around the world have consistently loved the model over all of this time. Honda has continued to provide original, innovative packages, leading-edge technologies, and new value to delight customers in 170 countries around the world.

For the recent full model change that resulted in the ninth generation of CIVIC, Honda endeavored to develop a design based on the grand concept of the “Distinctive Smart CIVIC.” With its powerful, clear and original dynamic monoform framework, and its functional, advanced interface offering excellent instant recognition, the new car expresses the unique presence handed down from the first generation plus new value.

It is Honda’s hope that over the coming decades customers around the world will continue to love not only the new CIVIC but the entire CIVIC line.

Reference

- (1) Abe, M.: Operability Enhancement of Steering Wheel Switch for Audio Remote Control, Honda R&D Technical Review, October, 2011, p. 66-69

■ Author ■



Toshiyuki OKUMOTO



Yoshiya KITAJIMA



Akihiro HAYASHI



Yoko TAJIMA

Styling Design of N BOX

Takeo HAMANO*

Akio HAYASHI*

Masayuki TOGAWA*

Yasuko SAITO*

ABSTRACT

Designed to the grand concept of a “Green Mini Minivan,” the N BOX utilizes an FF center tank layout (a first for a kei car) in a new-generation platform based on the concept of “man maximum, mechanism minimum,” making it a “kei minivan” for families with young children.

Starting with the design concept “Relax Flex Mini Minivan,” the N BOX has achieved the sense of presence, space and utility of a family minivan. The exterior has the styling theme of a “sense of reassurance in a box that evokes a friendly feeling,” while the styling theme of the interior is “pursuit of spaciousness.” The colors and seat fabric further accentuate those characteristics.

1. Introduction

Since entering the automobile market in 1963 with a kei car,* Honda has offered unique products unlike those of any other manufacturer.

Kei cars have come to be used as a highly accessible means of transportation. The market for them has undergone a number of standard changes that have brought safety and product marketability to mature levels, and thus their market has further diversified and continued to grow. The core of the market presently is taller models: these so-called height wagons emphasize the space value.

As part of product development, Honda researched kei car users nationwide and determined that space value is what is needed by families with young children, which make up the biggest share of users.

Some new models are appearing that are over 1700 mm high (so-called super-height wagons), and Honda has become strongly aware of the importance of space value in kei cars.

As the research has continued, the product image of an “authentic family minivan in the kei car size (a Mini Minivan)” has taken solid form, and Honda has found three themes that define this image: “comfortable space,” “reassuring with a commanding view,” and “utility.”

With these themes in mind, Honda aimed to build a “Green Mini Minivan” with optimal packaging, quality and fuel economy (Fig. 1).

To realize this objective, Honda developed a new-generation kei platform based on the concept of “man

maximum, mechanism minimum” (MM concept) and characterized by a high-efficiency package and FF center tank, thereby achieving an automobile with as much or more interior space, utility, handling, and sense of presence as an ordinary minivan while conforming to kei car standard (Fig. 2).

*Kei cars (or k-cars) are automobiles meeting the smallest specifications in Japan’s unique automobile classification system. The enforcement regulations of the Road Transport Vehicle Law stipulate that kei cars shall be no longer than 3400 mm, no wider than 1480 mm and no higher than 2000 mm, with engine displacement of no more than 660 cm³, rated passenger capacity of no more than four, and cargo capacity of no more than 350 kg.



Fig. 1 Grand concept

* Automobile R&D Center



Fig. 2 Overview of N BOX

2. Design Concept

The design concept selected for the N BOX is “Relax Flex Mini Minivan.” This concept expresses the N BOX’s characteristics of plenty of interior space for relaxing and utility with abundant flexibility (Fig. 3).

The N BOX, however, was designed and developed not simply to be a spacious kei height wagon, but to be a minimum minivan with exceptional quality offering a sense of reassurance so that customers could use it with pride as a family car.

Every effort was made to create a functional and



Fig. 3 Design concept

timeless expression that would not lose its appeal so that families could continue to love and use it for many years.

3. Package Layout

Through Honda’s pursuit of a high-efficiency package, the N BOX’s Mini Minivan Package was newly developed to achieve more space than an ordinary minivan while keeping the kei size (Fig. 4).

With the three key elements making up the Mini Minivan package — reassuring with a commanding view, comfortable space and utility — Honda repeatedly tested the package as it established the specifications.

Details of each of the elements are described below.

3.1. Reassuring with a Commanding View

One highly regarded aspect of minivans is the commanding view afforded by their high eye points. Honda drivers tried various models before setting the eye point height of the N BOX at 1360 mm off the ground. This eye point, which rivals that of the STEP WGN, aims to ensure a commanding view and a sense of reassurance that the N BOX will not be towered over by other automobiles on the road. Honda found a good balance of forward and downward vision by having drivers, especially women who are the main users, test-drive models from kei cars to minivans. The N BOX is the only FF kei car with an eye point of the same height as an ordinary minivan.

The design stretches the distance from the eye point to the top edge of the front window, forming a roomy forward space that even ordinary minivans do not offer. The angle of the front window was made nearly vertical to form this space. Moreover, the A pillar was narrowed and the shape of the front window perimeter was designed to create a sense of spaciousness laterally expanding the forward field of vision and creating a panoramic view (Fig. 5).

3.2. Comfortable Space Surpassing that of a Minivan

To achieve minivan space in a kei car size, the space

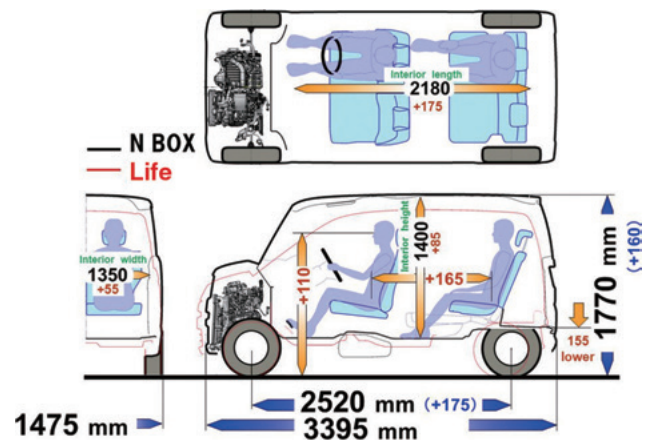


Fig. 4 Mini Minivan package

was constructed after researching the values of the first and second row seats of an ordinary family minivan of engine displacement class 2 L.

The new platform offers an interior length (catalog value) that is 175 mm longer than the Life thanks to such changes as shortening the engine room. The FIT's well-received center tank layout was used, which combined with the effect of extending the wheel base 100 mm, made it possible to expand the tandem distance (distance between front and rear seat passengers) by 165 mm over the Life.

Furthermore, the overall height was raised as high as possible (taking maneuvering stability into account), thus securing as much interior height as a high-end minivan. This achieved a roomy rear seat space that lets a child up to 140 cm stand up in the interior (Fig. 6).

3.3. Utility beyond Expectations

A study focused on family consumers in Japan found that many wanted to be able to carry a bicycle in their automobiles for those times when dropping off or picking up the child who bicycled to school. Honda's designers studied the cargo compartment and seat arrangement with the idea of making sure that a particular strength of this Mini Minivan would be that it is easier than ever before to load both people and bicycles. This study revealed that for family consumers, bicycles are the largest items that get loaded and unloaded on a day-to-day basis.



Fig. 5 Reassuring and commanding field of vision

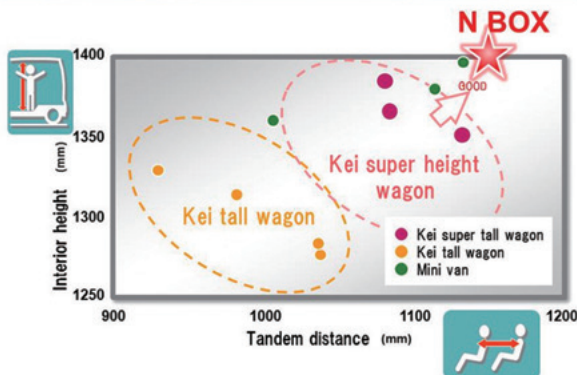


Fig. 6 Comfortable space

The interior length of the MM high-efficiency package and the low cargo compartment afforded by the FF center tank layout make it possible to easily load a bicycle.

And to take full advantage of these easy-to-use features, the rear side door on both sides is a sliding door that opens 640 mm, achieving class-leading ease of ingress/egress and accessibility (Fig. 7).

With one-motion dive-down, the rear seat folds up easily to ensure a large loading space for a bicycle. The height of the tailgate's lower edge is set low at just 480 mm above the ground, making it easy to load a bicycle. The cargo compartment is tall, at 1190 mm in the trunk area and 1155 mm over the folded seat back, so that a person can climb into the vehicle while loading the bicycle and then easily engage the stand. Loading and unloading a bicycle requires no effort at all, even on a daily basis.

The center tank platform also has a rear-seat tip-up mechanism so that with a single touch the user can open up a large space with the 1400 mm interior height from the floor beneath the rear seat to the roof. This allows many types of uses different from the trunk space such as a place to change clothes, or put hobby equipment like gardening tools and large musical instruments, or a dog carrier (Fig. 8).

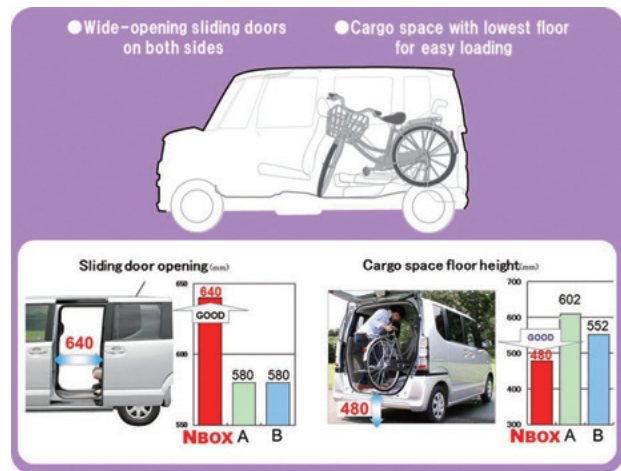


Fig. 7 Utility

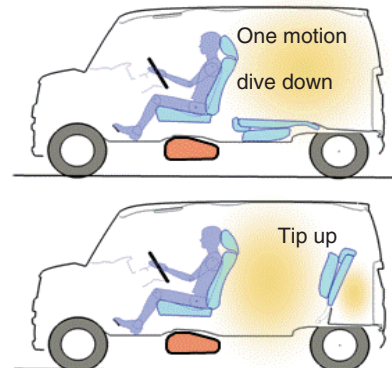


Fig. 8 Seat arrangement

4. Exterior Design

The exterior design is based on the theme “a strong and reassuring Mini Minivan form” in order to express the “Relax Flex Mini Minivan” design concept.

Though based on the box form that one associates with spaciousness, the model is more than just a box; it has a moderate amount of tension and roundness to make it seem friendly, and a cantilever window graphic that expresses the excellent panoramic field of vision. To provide reassurance as a minivan, the thick engine hood and the fenders that project outward express a sense of stability (Fig. 9).

The front view aims to express a sense of reassurance



Fig. 9 Exterior design concept



Fig. 10 Exterior rendering

and friendliness. To ensure this, the fenders project outward from the short but thick and sturdy front mask to express dynamism and stability. The panoramic window simultaneously expresses the good field of vision and the roominess of the interior.

Large-diameter cylinder reflector faces give the headlights a friendly expression. The front grill consists of a combination of a plated bar and a louver of the body color that conspicuously projects the same level of quality as an ordinary minivan.

In side view the long roof and wide side panel face express the spaciousness and sense of reassurance of a minivan. This was achieved by the A pillar that stands straight up, and a higher window position than in an ordinary kei car.

In rear view, the firm D pillar supports the long roof enabling both the excellent field of vision and the sturdiness of the box. With a wide rear gate, the hatch, which is the size of that of an ordinary minivan, emphasizes the excellent utility and ease of use, while the plated garnish creates an accent that projects quality. The blacked-out housing puts a graphic emphasis on the tail light giving it both a fresh and functional look (Fig. 10).

5. Interior Design

The interior was developed with the themes of “spacious feel, sense of enrichment and novelty” to provide the spaciousness and utility of a minivan but in the kei car class. Honda designed the instrument panel, door lining, seats and so on to appear larger, and studied the utility of the N BOX as a minivan (Fig. 11).

The middle tray extends to the right and left setting the base tone for the instrument panel that is modeled to impress as functional and wide. The meters, air conditioning panel, audio panel and other major information indicators and controls, and other functions are arranged to be intuitive at a glance and easy to use. The instrument panel’s upper surface has a wide visor to impart a sense of the automobile



Fig. 11 Interior design concept

rank of minivan, and a visual sense of reassurance while driving (Fig. 12).

A three-part meter design comes standard on all grades of the model.

The speedometer sits in the middle flanked by the tachometer on the left and information display on the right. Of the three parts, the speedometer has a fresh and clean image with a green scale on a white background. It has three ring-shaped lights on the periphery, with the middle one changing color from white to green when driving in eco-mode to inform the driver that he or she is saving fuel (Fig. 13).



Fig. 12 Instrument panel rendering



Fig. 13 Meters



Fig. 14 Interior rendering

The seat design aims for the sense of spaciousness and comfort of a minivan; both the front and rear rows are large, flat bench seats. While the flat appearance of the seat design creates a roomy feel, the subtle undulation of the seat backs and cushions provides sufficient hold so passengers feel comfortable and refreshed even on long drives. The right and left back seats also have independent center armrests making them as comfortable as the front seat (Fig. 14).

6. Color and Fabric Design

6.1. Exterior

Following the theme of “Relax Flex Mini Minivan,” the eight exterior colors available give the N BOX its friendly and fun look as well as a sense of fine quality reminiscent of a minivan.

The new color Cherry Shell Pink mimics the coloring of the shellfish *sakuragai*. The elegant, slightly pale pink color has the transparent texture of sunlight. The addition of this color gives the entire lineup a brilliant look and makes color selection more enjoyable.

The lineup also includes Premium Deep Rosso Pearl and Premium Bronze Pearl for a high-quality, grown-up look. Other choices are Cool Mist Metallic (the model’s appealing character color), HIDAMARI Ivory Pearl (with a friendly looking and gentle impression) and the basic colors Premium White Pearl, Alabaster Silver Metallic and Crystal Black Pearl (Fig. 15).

6.2. Interior

Honda designed the interior so that the person would be surprised by its roominess and comfort the moment he or she opened the door. With a simple and waste-free design, the aim is to project an image of a spacious room where one can relax.

Beige, with enough brightness to emphasize the spaciousness but also giving a modern impression of subdued chroma, is used throughout the interior. The seat fabric starts with a soft cloth base with a pearl print that projects elegant glossiness for a quality look. The result

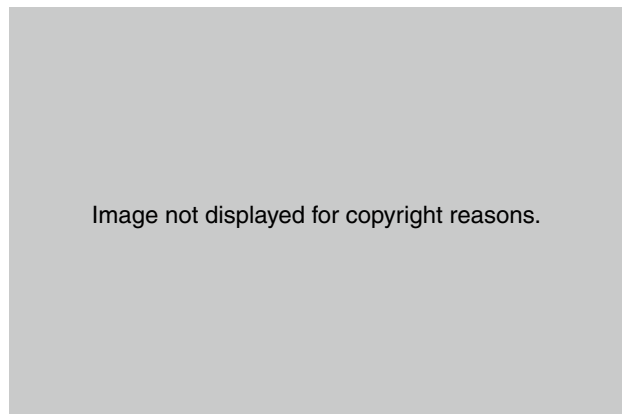


Fig. 15 Exterior colors

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Fig. 16 Interior color material

is a high-quality interior that is reminiscent of an ordinary minivan and unlike any kei car available (Fig. 16).

7. Logo Design

The “N” in the car name was chosen to imply the ideas “New, Next, Nippon (Japan) and Norimono (vehicle)” in a new-generation kei series. The N BOX is the first in this series.

The logo design layout features a large letter N to leave a lasting impression of the N brand as a series (Fig. 17).



Fig. 17 N BOX logo

8. Conclusion

The N BOX was developed under the concept “Green Mini Minivan.”

It is Honda’s hope that many drivers with young children will choose the N BOX and that the N Series will expand in the future.

■ Author ■



Takeo HAMANO



Akio HAYASHI



Masayuki TOGAWA



Yasuko SAITO

Development of DOHC VTC 0.66 L Gasoline Engine for New K-car

Ryutaro TAGISHI*

Shigemi KOBAYASHI*

Junya IINO*

ABSTRACT

A lightweight, compact DOHC VTC 0.66-L 3-cylinder gasoline engine was developed for use in K-cars, achieving a balance of environmental performance (low fuel consumption and low emissions) and high power output. The new model engine was developed on the assumption that it would be combined with a newly developed continuously variable transmission (CVT) for k-cars. It was accordingly given a long-stroke structure and variable timing control (VTC) on the intake side as well as technology for enhancement of combustion and friction reduction. These measures achieved high power with maximum torque of 65 Nm and maximum power of 43 kW in the naturally aspirated engine together with 10% enhancement in specific fuel consumption over the previous engine model. In the turbocharged engine, use of a higher compression ratio in addition to fuel consumption enhancement technology in common with that of the naturally aspirated engine achieved a 10% enhancement in specific fuel consumption over the previous engine model as well as maximum torque at the high level of 104 Nm.

1. Introduction

Emissions reduction technology for protection of the environment and low fuel consumption technology for reduction of CO₂, which is a cause of global warming, have become important in the development of motor vehicle engines in recent years. There has also been rising demand in Japan for k-cars because of sharply rising crude oil prices and other such factors, and raising the level of environmental performance is an urgent task. As the technologies adopted for these purposes, the engine used in the 2004 model of the LIFE⁽¹⁾, a k-car, incorporates the short-stroke structure for friction reduction and the dual and sequential ignition (i-DSI)⁽²⁾, also adopted in the 2002 model Fit, to provide more rapid combustion and a higher compression ratio by having two spark plugs for each cylinder. Although the short-stroke type of i-DSI engine has an enlarged cylinder bore diameter that is not advantageous in terms of heat loss and knocking, it is advantageous in terms of fuel consumption due to the lower friction when combined with a four-speed automatic transmission that makes intensive use of relatively high speed domains. Achieving further enhancement of fuel consumption, however, will require operation at lower speeds and enhancement of brake specific fuel consumption (BSFC). To that end, the new model engine was aimed for enhancement of low-speed torque and enhancement of brake specific fuel consumption under high load. Development was therefore carried out so as to effectively achieve enhanced fuel consumption when combined with a

CVT. The following will report on the technology adopted in the new model engine.

2. Development Goals

In considering engine structure, the ways by which the environmental performance of the new model engine could be enhanced were studied. Figure 1 shows the engine's

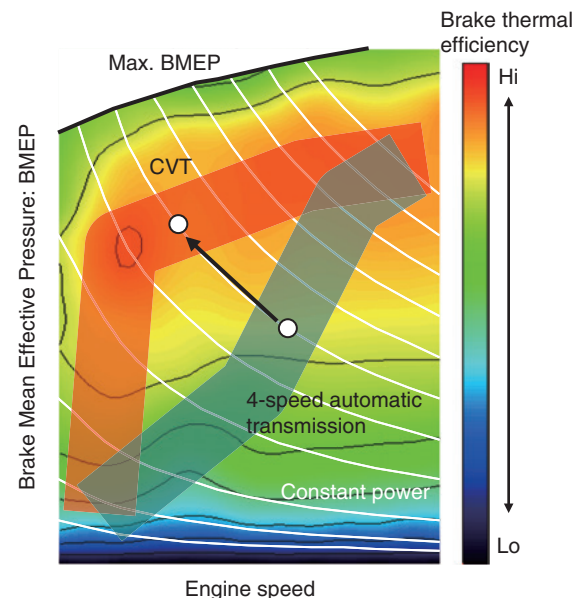


Fig. 1 Brake thermal efficiency map and typical-use domain under 10-15 mode operation of 4-speed automatic transmission and CVT

* Automobile R&D Center

Table 1 Engine specifications

		Developed model		Previous model	
Air intake system		Normal aspiration	Turbo charged	Normal aspiration	Turbo charged
Cylinder configuration		In-line 3-cylinder	In-line 3-cylinder	In-line 3-cylinder	In-line 3-cylinder
Bore × stroke (mm)		64 × 68.2	64 × 68.2	71 × 55.4	71 × 55.4
Displacement (cm ³)		658	658	658	658
Compression ratio		11.2	9.2	11.2	8.5
Valve train		DOHC intake VTC	DOHC intake VTC	SOHC i-DSI	SOHC i-DSI
Number of valves		4 per cylinder	4 per cylinder	2 per cylinder	2 per cylinder
Valve diameter (mm)	in.	24.5	24.5	35.5	35.5
	ex.	20.5	20.5	30	30
Ignition system		Single-point ignition	Single-point ignition	Dual-point ignition	Dual-point ignition
Max. power (kW/rpm)		43/7300	47/6000	38/7000	47/6000
Max. torque (Nm)		65/3500	104/2600	60/3600	93/4000

brake thermal efficiency map overlaid with typical-use engine speed and load domains for the four-speed automatic transmission and CVT operating in 10-15 mode. Adoption of the CVT enables lower-speed, higher-load operation with the same driving force and increases the frequency of use of domains with greater brake thermal efficiency. Consequently, adoption of the CVT alone enables enhancement of fuel consumption even with the same engine, but the new model engine was aimed to enhance performance in the low-speed domain so it could produce still greater fuel consumption enhancement effect. Figure 2 shows the brake specific fuel consumption (BSFC) of the engine measured at the same brake horsepower and speed in the previous model engine together with the pumping mean effective pressure (PMEP) and friction mean effective pressure (FMEP). As the engine speed drops, the PMEP and FMEP also drop, enhancing the BSFC. Below a certain speed, however, no further enhancement is obtained. That is due to the retarded ignition timing used for knocking avoidance, and knocking enhancement has the potential for

further enhancement of fuel consumption. Furthermore, even in cases when there is no necessity to avoid knocking, the use of lower engine speeds is constrained by the limitations on charging efficiency. It is apparent, therefore, that the maximum fuel consumption impact from low-speed operation can be obtained effectively by knocking enhancement and charging efficiency enhancement in low-speed domains, which is to say, by enhancement of low-speed torque.

Given the above, the goals for the new engine aimed at enhancement of fuel consumption performance were defined by the following items:

- (1) Achieve a balance of enhanced low-speed torque and maximum power; and
- (2) Further enhancement of brake thermal efficiency.

3. Engine Overview and Main Specifications

In order to achieve the performance goals for the new model engine, the following kinds of engine structure specifications and main technologies were applied.

- (1) Adopt continuous variable valve timing control (VTC) mechanism on the intake side
 - Enhance charging efficiency in the low to medium speed domains through the scavenging effect
 - Enhance thermal efficiency by internal exhaust gas recirculation (EGR)
- (2) Long-stroke structure
 - Enhance knocking by strengthening the in-cylinder flow and thermal efficiency by increasing the combustion speed
 - Enhance thermal efficiency by decreasing the surface/volume ratio (S/V ratio)
- (3) DOHC four-valve cylinder head
 - Achieve balance of small bore and maximum power
 - Adopt Tumble port
- (4) Reduction of friction in every part

Table 1 shows the main specifications compared with the previous engine model, and Fig. 3 shows an

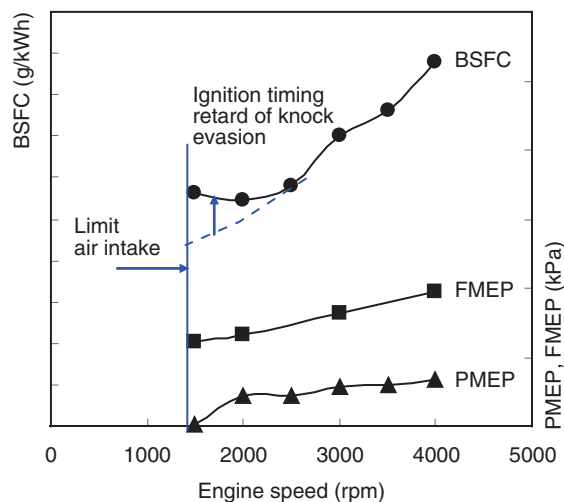


Fig. 2 Effect of engine speed on BSFC, PMEP and FMEP

external view of the engine. The new model engine was given a long-stroke structure for the purpose of enhancing combustion, but steps were also taken to make it compact, without increasing its height relative to the previous model.

Figure 4 shows a sectional view of the cylinder head. The new model engine was given roller rocker arms for reduced friction as well as hydraulic lash adjusters to make it maintenance-free, and the size of the cylinder head was reduced.

Figure 5 shows a comparison of the combustion

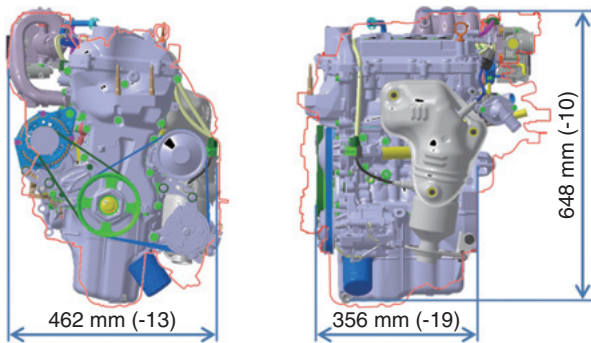


Fig. 3 View of new engine and comparison of engine size

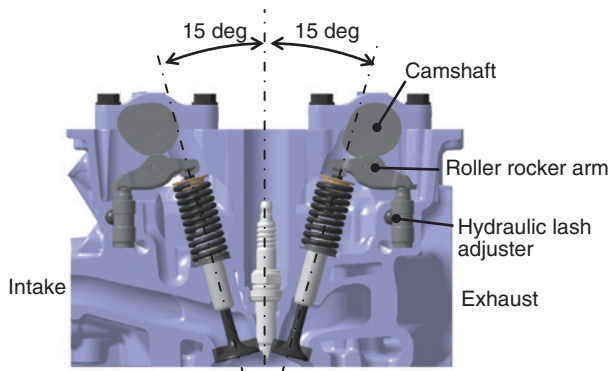


Fig. 4 Sectional view of cylinder head

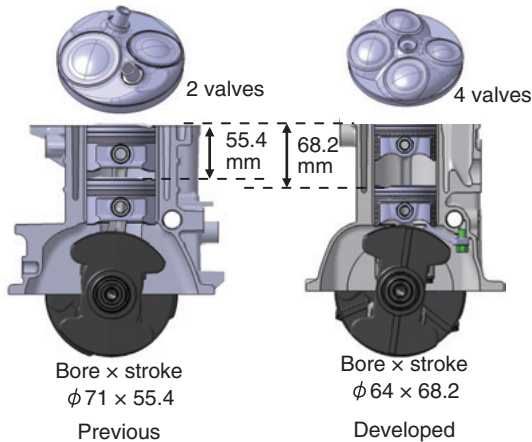


Fig. 5 Comparison of combustion chamber configuration of previous and developed engine

chamber configuration with the previous model. Where the previous model had a short stroke with a stroke-bore ratio (S/B ratio) of 0.78, the new model engine was given a longer stroke, with an S/B ratio of 1.07, to reduce the flame propagation distance and strengthen the in-cylinder flow. The limitation on valve diameter due to the smaller bore was addressed by changing from the previous two valves to four valves. Tumble ports were used for the intake ports as a way of strengthening the in-cylinder flow. Taking the conceptual approach of F1 technology for the shape of the combustion chamber, a conical shape was formed concentrically around each valve to control the flow, providing a more advantageous shape than the pent-roof of the previous model in terms of S/V ratio and flow coefficient.

4. Power Output Characteristics

4.1. Naturally Aspirated Engine

Figure 6 shows the power output characteristics at wide-open throttle in comparison with the previous model. The new model engine has VTC and optimized intake and exhaust system specifications that yield higher torque than the previous model across all speed domains. It achieves 8% torque enhancement at maximum torque and 18% at a speed of 2000 rpm, obtaining the stronger low-speed torque that was the aim. The maximum power is enhanced by 13% over the previous model, achieving a balance of low-speed torque and maximum power.

4.1.1. Charging efficiency enhanced by VTC

Figure 7 shows measurements of intake and exhaust

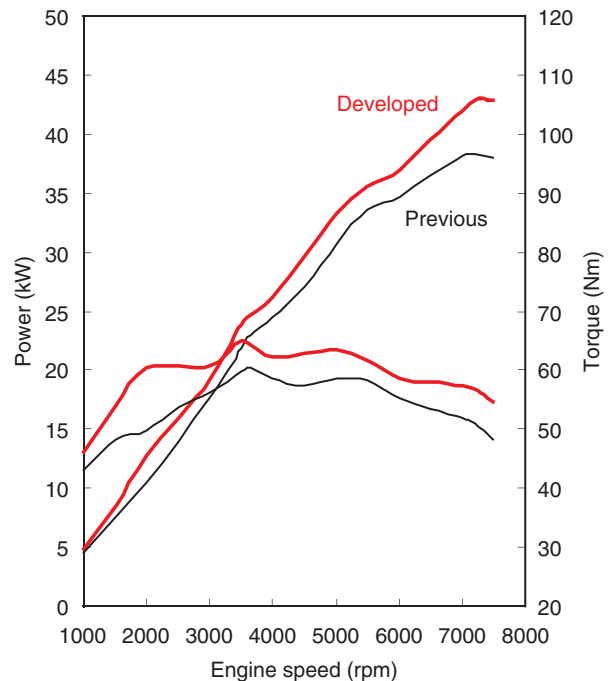


Fig. 6 Engine performance of normal aspirated engine

pulsation at the torque peak of 3500 rpm with the throttle wide open. The timing for opening the intake valves is controlled by the intake VTC timing advance action (Fig. 7A), and the timing for closing the exhaust valves (Fig. 7B) is retarded to the limit for providing stable combustion under low load, as found when idling. This extends the period when the exhaust port pressure is exceeded by the intake port pressure during overlap (Fig. 7C), enhancing the scavenging effect. Figure 8 shows charging efficiency characteristics when operating with the throttle wide open. Due in part to the effect that changing to a four-valve configuration has in high-speed domains, this engine achieves enhanced charging efficiency relative to the previous model across all speed domains.

4.1.2. Knocking enhancement

In addition to strengthened in-cylinder flow due to the long-stroke structure, the adoption of piston oil jets and the change of coolant circulation method yielded knocking enhancement. Figure 9 shows the ignition timing characteristics as measured with varying charging efficiency at an engine speed of 2000 rpm in comparison with the previous model. The new model engine has been able to shift charging efficiency some 10 points more toward the high-load side than the previous model at the point where the knocking ignition timing begins to be retarded beyond

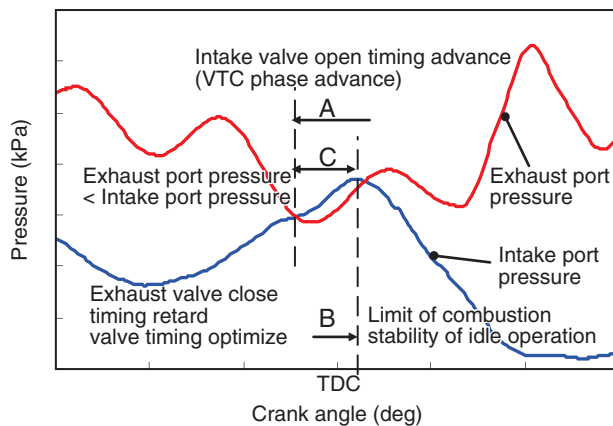


Fig. 7 Pulsation of intake port and exhaust port, and valve timing

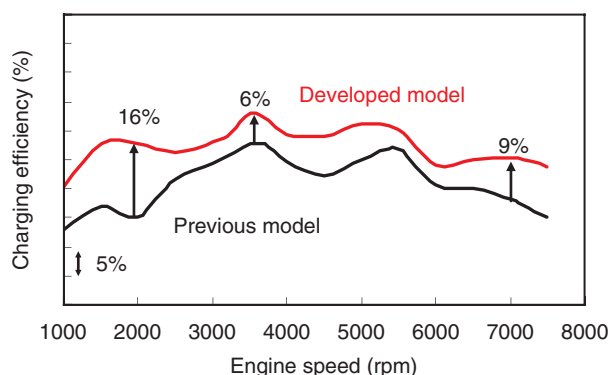


Fig. 8 Comparison of charging efficiency

the minimum advance for best torque (MBT).

Figure 10 shows the piston oil jet shape. The oil from the oil jet has been directed so as to actively cool the exhaust side behind the piston. Unlike the conrod jet in the previous engine that sprays oil intermittently, here the constant spray of oil increases the buildup of oil behind the piston, enhancing the cooling effect on the piston and enabling the knocking ignition timing to be advanced by one degree.

Figure 11 shows the coolant circulation system by comparison with the previous model. In the previous model, the total amount of coolant discharged by the water pump entered the cylinder heads, after which a portion of the coolant was distributed to the cylinder block. The coolant on the sides of the cylinder block flowed in from the cylinder-head side, which is to say from above, so that there would be a stronger flow at the bottom of the water jacket than in the vicinity of the head gasket. In the new model engine, the coolant discharged from the water pump

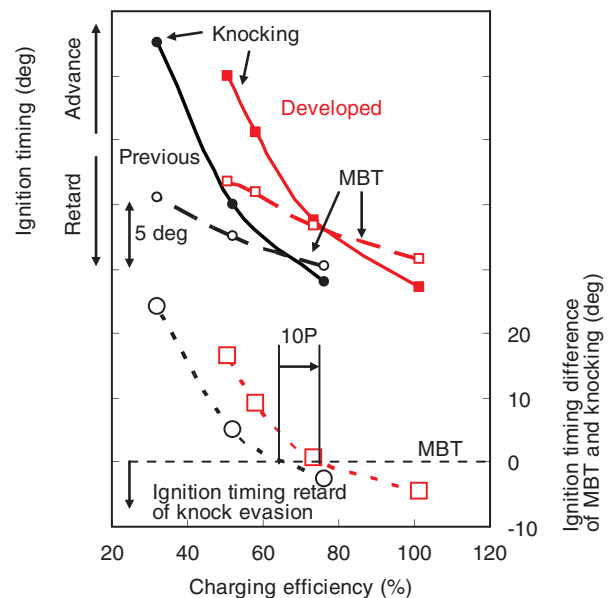


Fig. 9 Characteristics of ignition timing versus charging efficiency

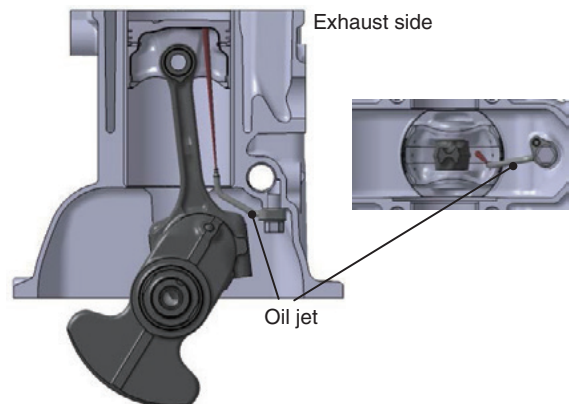


Fig. 10 Piston oil jet

is distributed simultaneously to the cylinder block and the cylinder head. The coolant flowing in the cylinder block enters the cylinder block directly from the water pump, and the coolant is thrown up to the upper areas by the ramp-shaped structure built into the water jacket.

This allows an active flow of coolant to be directed to the vicinity of the gasket surface, heightening the cooling effect on the combustion chamber and yielding knocking enhancement. Also, as shown in Fig. 12, the aluminum portion around the sleeve was made thinner, only on the upper part of the water jacket in the cylinder block, to provide active cooling of the combustion chamber.

4.2. Turbocharged Engine

Figure 13 shows the power output characteristics at wide-open throttle in comparison with the previous model. The new model engine was equipped with intake VTC, which increased the compression ratio from the previous 8.5 to 9.2 while achieving a 12% enhancement at maximum torque. This further enabled enhancement of the torque start-up characteristics.

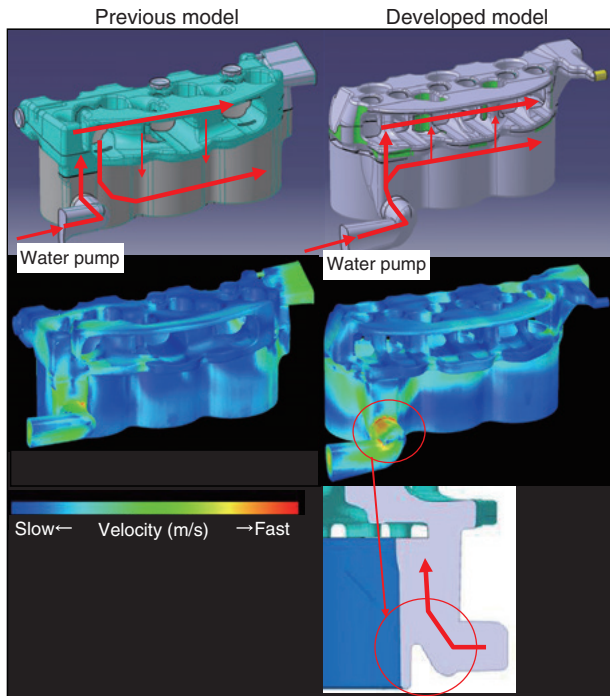


Fig. 11 Cooling circulation system

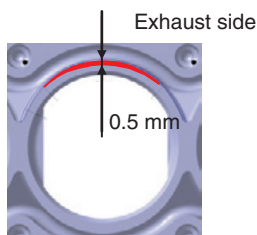


Fig. 12 Enhanced shape of water jacket

4.2.1. VTC knocking enhancement effect

Figure 14 shows the relationship of boost pressure and ignition timing to charging efficiency at an engine speed of 3000 rpm with and without VTC. As with a naturally aspirated engine, the adoption of VTC increases the scavenging effect, enabling a reduction of about 20 kPa in boost pressure at the same charging efficiency. Reducing the boost pressure results in lower intake temperature and yields knocking enhancement together with reduction in the amount of residual gas. This enabled an increase in the compression ratio from 8.5 in the previous model to 9.2, enabling achievement of a balance between high torque and fuel consumption performance.

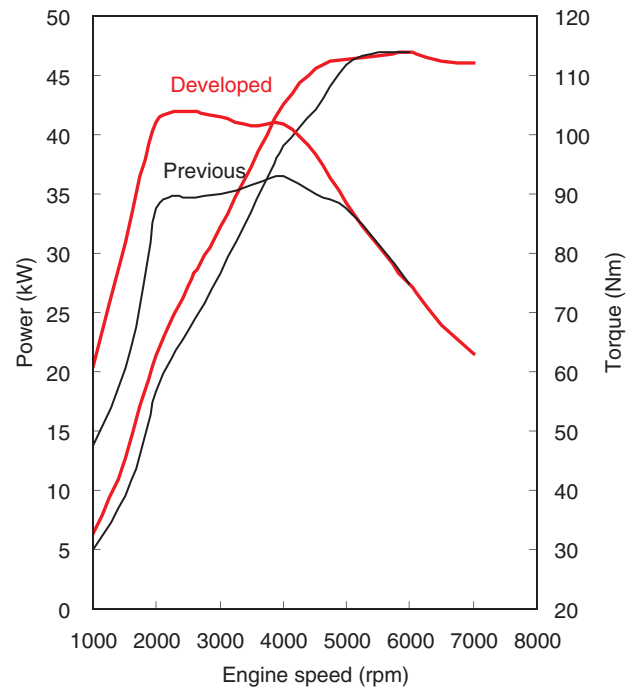


Fig. 13 Turbocharged engine performance

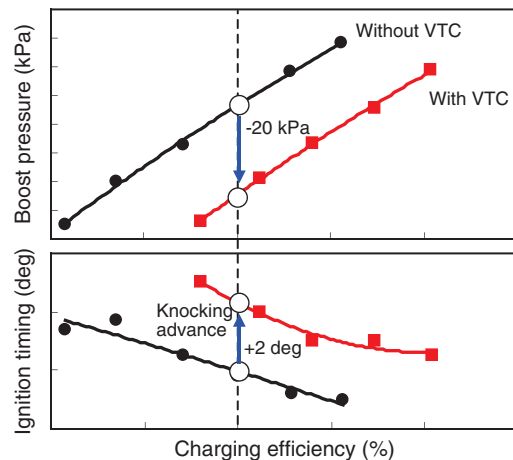


Fig. 14 Comparison of boost pressure and ignition timing between engines with and without VTC

4.2.2. Boost pressure start characteristics enhanced by VTC

Enhancement of the scavenging effect in low-speed domains due to the adoption of VTC also contributes to enhancement of transient boost pressure start characteristics. Figure 15 shows transient characteristics when acceleration of a completely built vehicle from a standing start with wide-open throttle is simulated on the bench. Due to VTC, the time to reach target boost pressure was shortened about 18% and the engine speed rises more quickly, contributing to enhancement of standing start performance in completely built vehicles.

4.2.3. Measures addressing exhaust soot

The greatest issue for adoption of VTC in a turbocharged engine was the increase in exhaust soot caused because the advanced intake valve timing increased the overlap. Figure 16 shows smoke and other particulate matter density when measures to be described below are taken to address soot and other particulate matter and when a previous model engine without VTC is operating at wide-open throttle power, both with VTC angle advanced to increase charging efficiency. When the VTC angle is advanced in order to increase the scavenging efficiency, there is an increase in the amount of blow-by from fuel injected into the intake port and the amount of soot and other particulate matter increases. The increase in soot and other particulate matter can be avoided by limiting the VTC advance angle, but this will also lower the charging efficiency, so it is desirable to keep the limitation on the advance angle to a minimum. This was addressed by

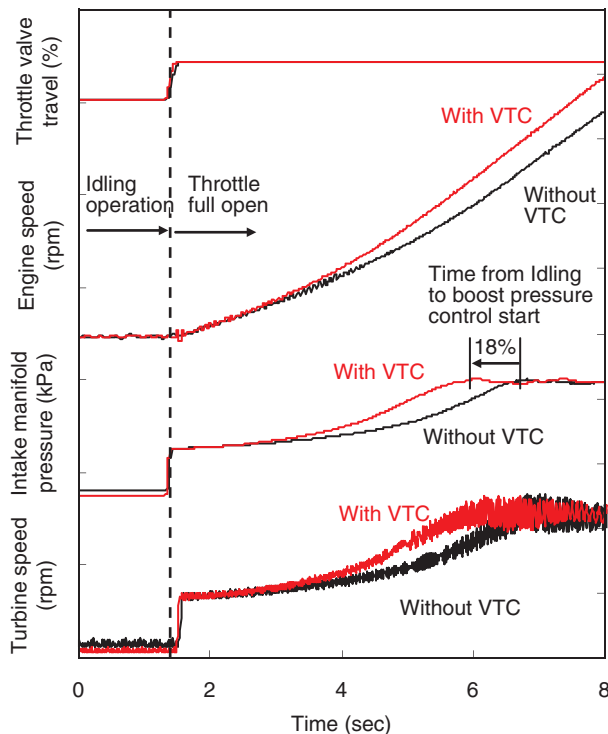


Fig. 15 Transient performance comparison with and without VTC

increasing the pressure of fuel injection from 294 kPa to 343 kPa and shortening the injection time while also optimizing the injection timing. This reduced the amount of fuel injected before overlap, which reduced the amount of blow-by without placing any limit on the VTC advance angle, and achieved a balance between reduction in the amount of exhaust soot and increase in charging efficiency.

5. Brake Specific Fuel Consumption Performance

Due to VTC, both the naturally aspirated version and the turbocharged version of the new model engine achieved enhancements over the previous model engines of 4% in internal EGR and 6% in combustion enhancement and friction reduction, for a total enhancement of 10% in specific fuel consumption. Figure 17 shows BSFC characteristics at 2000 rpm in the previous model and new model of naturally aspirated engine.

5.1. Enhancement of Indicated Thermal Efficiency

Figure 18 shows the cylinder pressure diagram for the previous model engine and for the new model engine with the VTC angle not advanced, both at 2000 rpm and an indicated mean effective pressure (IMEP) of 710 kPa. The

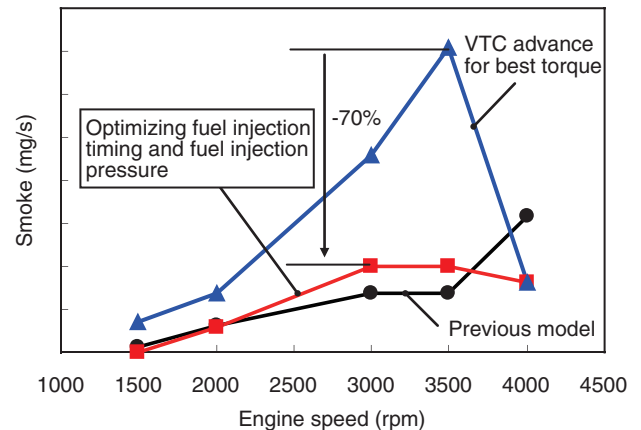


Fig. 16 Comparison of amount of smoke

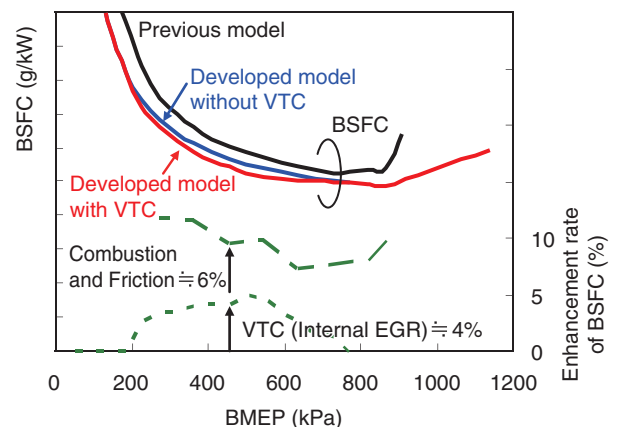


Fig. 17 Comparison of characteristics of BSFC

increased stroke length relative to the previous model yielded a lower S/V ratio, which together with other factors enabled a 14% reduction in heat loss; the strengthened in-cylinder flow together with other factors enabled a 5% reduction in time loss; and the optimized exhaust valve timing enabled a 70% decrease in exhaust loss. The VTC also has an internal EGR effect that yielded enhancement of brake specific fuel consumption relative to the previous model engine.

5.2. Friction Reduction Technology

Relative to the previous model engine with short stroke and SOHC, the new model engine involves numerous factors that increase friction. These include the adoption of a long stroke, which increases piston sliding resistance; of DOHC, which increases valve assembly friction; and of hydraulic VTC and the piston oil jet, which increases the lubricating oil requirement and the oil pressure. The adoption of the following technologies, however, enabled a 13% reduction in mechanical friction and a 3.5% enhancement of brake specific fuel consumption relative to the previous model. Figure 19 shows an analysis of

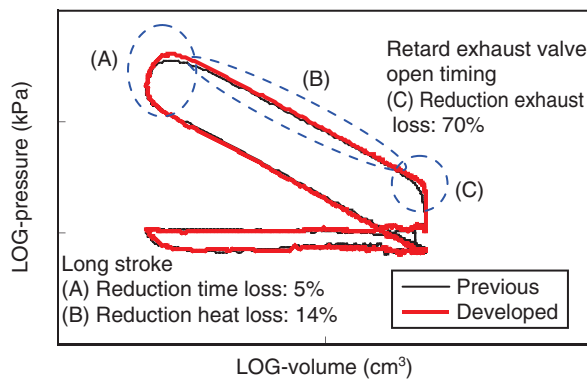


Fig. 18 Comparison of cylinder pressure

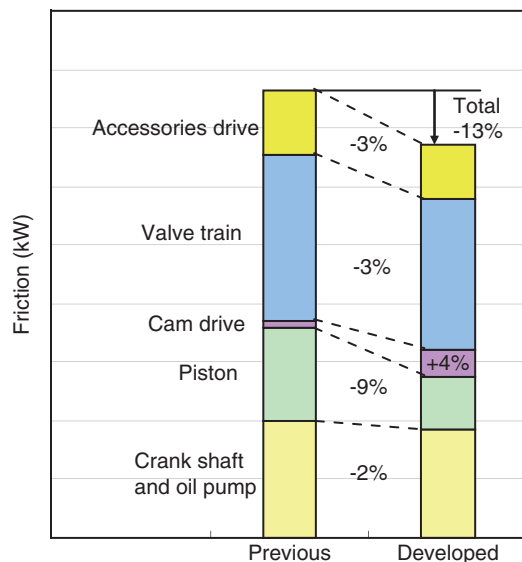


Fig. 19 Mechanical friction of previous and developed model

mechanical friction at 2000 rpm in comparison with the previous model.

(1) Crank shaft, oil pump drive

- High-efficiency, rotor-type, two-stage relief oil pump
- Crankshaft journal bearings with eccentric grooves on the bottom side and with molybdenum disulfide coating
- Adopt oil seal with low binding force and Teflon coating
- Reduce crank journal surface roughness
- Reduce viscosity of engine oil
- Minimize oil leakage in each part

(2) Piston sliding resistance

- Piston skirt with alternating-pattern molybdenum disulfide coating
- Optimize piston clearance
- Reduce piston ring tension

(3) Cam drive

- Reduce timing chain width
- Adopt cam journal holder (thinner cam journal axis)
- Reduce cam journal surface roughness

(4) Valve train

- Reduce load on valve springs

(5) Auxiliary drive

- Optimize auxiliary belt layout

The high-efficiency two-stage relief oil pump will be described as an example of mechanical friction reduction technology. In the new model engine, the shape of the rotor teeth in the oil pump was enhanced to increase the rate of volumetric change. This was done to raise the oil discharge rate per rotor turn and reduce the rotor diameter by 2%. Adoption of the two-stage relief structure lowered the oil pump driving force. Figure 20 shows the two-stage relief structure of the oil pump. Figure 21 shows the engine oil pressure and friction with two-stage and one-stage relief oil pumps. The pressure relief has two stages, and the first stage is configured to assure the minimum necessary oil pressure at low speeds. This enables a 36% lowering of the oil pressure, yielding a 9% reduction in friction relative to the one-stage relief oil pump.

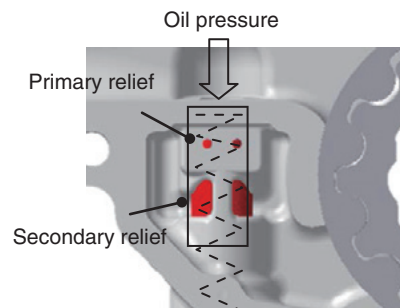


Fig. 20 2-stage relief oil pump

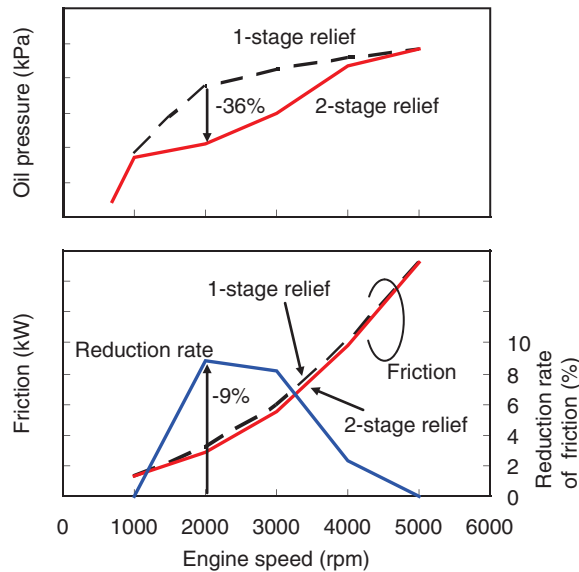


Fig. 21 Effect of 2-stage relief oil pump for oil pressure and friction

6. Ignition Timing Control

6.1. Calculation of Ignition Timing

As discussed in the preceding, VTC is being used to increase charging efficiency and enhance specific fuel consumption. The combination of VTC with a three-cylinder engine readily yields a scavenging effect and internal EGR effect from the VTC. On the other hand, however, there is little influence from exhaust interference, and internal EGR under the same intake pressure is subject to considerable influence from atmospheric pressure and varies accordingly. Where the previous control system calculated the ignition timing from the intake pressure measured by a pressure sensor and from the engine speed, therefore, it could not track the variations in internal EGR and could not calculate a correct control value. The present engine adopted a control system that calculates the proper ignition timing by using the output from an atmospheric pressure sensor built into the electronic control unit (ECU) to correct for the intake pressure, thereby cancelling out the variations in internal EGR. Therefore, the internal EGR remains always constant with respect to the intake pressure corrected for atmospheric pressure, even if the atmospheric pressure varies, and the fresh air volume is also subject to a similar relationship. Consequently, even if the atmospheric pressure changes when the VTC angle is being advanced, the system will be able to calculate the correct ignition timing. Figure 22 shows how the internal EGR rate with respect to the intake pressure at varying atmospheric pressures and the internal EGR rate with respect to the intake pressure corrected for atmospheric pressure are different.

6.2. Optimizing the Retardation of Ignition Timing by means of Knocking Detection

The knocking control system applied from the 2011 model CIVIC uses information on multiple frequencies by employing a non-resonant knocking sensor and short-time Fourier transform (STFT) processing. By comparison with the previous method, which only determined whether or not knocking had occurred by the magnitude of the knocking sensor output, the new method uses frequency and time component pattern matching to enable identification of knocking intensity⁽³⁾. Therefore, as shown in Fig. 23, the previous method could not identify the intensity of knocking, so retard was always at a constant value. The new method, however, can provide retard amounts appropriate to the intensity. This new control method has also been applied in the present engine, and it has produced enhancement of specific fuel consumption by reducing the retard amount during trace knocking, which occurs frequently in completely built vehicles.

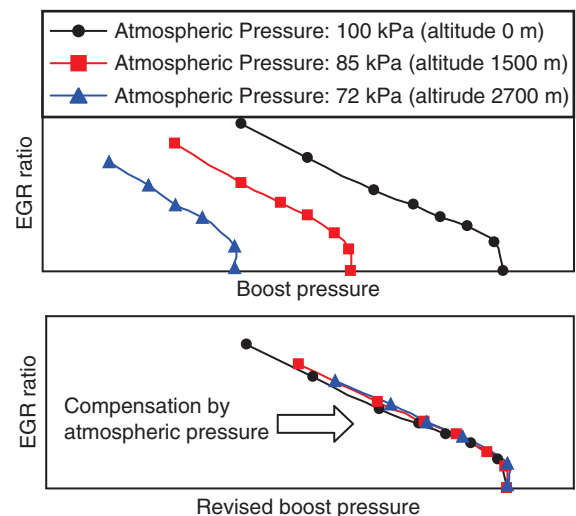


Fig. 22 Compensation by atmospheric pressure of internal EGR rate

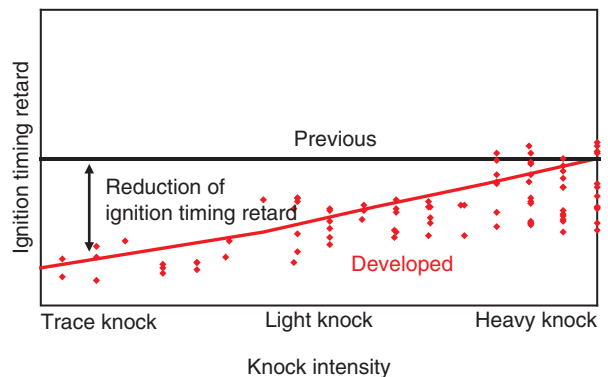


Fig. 23 Ignition timing retard depending on knock intensity

7. Conclusion

An engine was developed that achieves a balance of low fuel consumption and high power through the adoption of intake VTC technology, combustion enhancement technology, and friction reduction technology as well as the revision of ignition timing control. The effects of this engine are as follows:

- (1) In comparison with the previous model engine, it achieved a 13% enhancement of maximum power and 8% enhancement of maximum torque in the naturally aspirated engine and obtained a 12% enhancement of maximum torque in the turbocharged engine; and
- (2) Brake specific fuel consumption was reduced 10% in both the naturally aspirated and turbocharged engines.

References

- (1) Wada, Y., Ohtsu, K., Narita, K., Shinohara, T.: Development of i-DSI 2Plug Engine for 2004 Model Year Honda LIFE, Honda R&D Technical Review, Vol. 16, No. 1, p. 93-102
- (2) Nakayama, Y., Suzuki, M., Iwata, Y., Yamano, J.: Development of a 1.3L 2-Plug Engine for the 2002 Model "Fit," Honda R&D Technical Review, Vol. 13, No. 2, p. 43-52
- (3) Akimoto, K., Komatsu, H., Kurata, A.: Construction of Knock Detection Logic by Pattern Recognition Using Short-time Fourier Transform, Honda R&D Technical Review, April, 2012, p. 128-135

■ Author ■



Ryutaro TAGISHI



Shigemi KOBAYASHI



Junya IINO

Development of New CVT for K-car

Takashi NAKAMURA*

Keiichi HANYU*

Koichi KONISHI*

ABSTRACT

A continuously variable transmission (CVT) adopting a parallel-shaft, helical gear primary reduction mechanism for greater transmission efficiency was developed for K-car use. The distance between the input shaft and output shaft in the front-to-back direction was reduced and the hydraulic control system was positioned all together below the transmission. This supported the reduction of engine compartment size and, by extension, contributed to the extended length of the K-car cabin interior.

Two types of 0.66-L engine, naturally aspirated (NA) and turbocharged (TC), are supported, as well as two-wheel drive (2WD) and four-wheel drive (4WD) systems, with a minimum configuration of special parts. Idle stop was also supported with just the addition of an electric oil pump.

1. Introduction

K-car fuel-saving performance has become a major focus of attention in recent years. An enhanced level of fuel consumption is also being demanded of Honda K-cars, and a new model CVT (hereafter, the present CVT) was developed for the K-car.

This CVT is to replace the lightweight, reasonably priced four-speed automatic transmission (AT)⁽¹⁾ being used in Honda K-cars, and so a lightweight, compact structure was adopted. The previous CVT had the drive pulley positioned coaxially with the input shaft, so the separation between the input shaft and the output shaft where the differential assembly is located could not be reduced. It was reasoned that the adoption of a parallel-shaft, helical gear primary reduction mechanism would allow the drive pulley to be located above the input shaft, in which case the distance between the input shaft and output shaft could be reduced.

The adoption of a primary reduction mechanism lowers the rotational speed of the CVT's metal V-belt and increases the transmitting torque. It was reasoned that setting an optimal primary reduction ratio would enhance the transmission efficiency of the CVT.

The previous CVT had a high gear-meshing frequency, and therefore required high machining accuracy for the gears, careful attention to the shaft supports, and other such considerations. However, the present CVT has adopted a primary reduction mechanism with parallel shafts and helical gears. This allows a lower gear-meshing frequency than other mechanisms, and this was therefore considered an advantageous structure for the quietness of the cabin.

The NA and TC engine types could be supported by

differentiating specifications for torque converter and primary reduction gear, the 2WD and 4WD drive system types by differentiating specifications for the torque converter case (T/C case) and the differential case, and the idle stop specification just by the addition of an electric oil pump, with a minimal configuration of spatial parts.

2. Development Goals

The development of a new CVT for use with K-cars was carried out with the following items as the development goals:

- (1) Light weight and compactness at the top of its class;
- (2) Transmission efficiency at the top of its class;
- (3) Lower gear-meshing frequency in order to enhance quietness of the cabin; and
- (4) Commonization of parts
 - Support NA and TC with the same construction
 - Support 4WD with minimum parts change
 - Support idle stop with just the addition of an electric oil pump.

3. Basic Layout and Main Specifications

A main section view and a perspective external view of the present CVT are shown in Fig. 1 and Fig. 2. The major specifications are shown in Table 1.

The basic layout of the present CVT has the following features:

- (1) A primary reduction mechanism and forward-reverse switching mechanism using helical gears between parallel shafts were adopted and set upstream of the main transmission mechanism (continuously variable

* Automobile R&D Center

transmission mechanism using a belt and pulley). These together eliminated any secondary reduction mechanism. Torque flows from the input shaft, to the drive pulley shaft, then to the driven pulley shaft, and finally to the differential shaft, in a four-shaft structure.

- (2) Parts for the NA and TC engine types downstream from the main transmission mechanism were commonized by adjusting the fluid characteristics of the torque converter and the primary reduction ratio.
- (3) A shaft arrangement that supports reduction in the front-to-back length of the engine compartment was adopted.

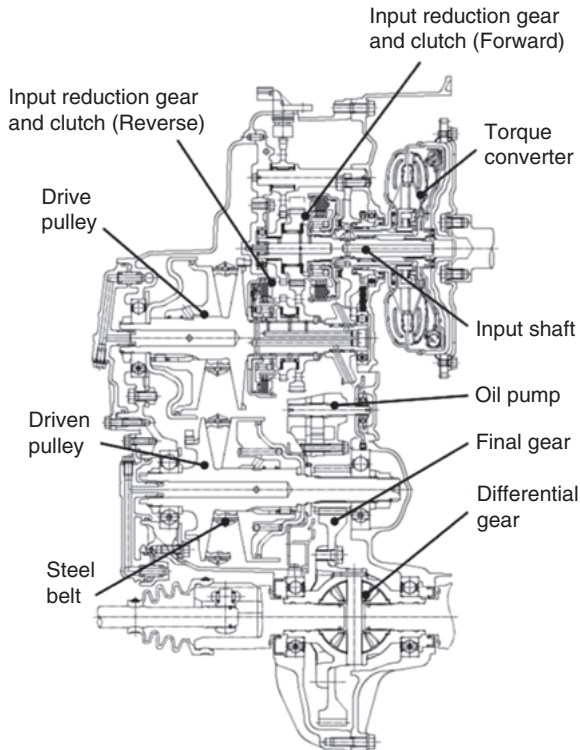


Fig. 1 Main section of developed CVT

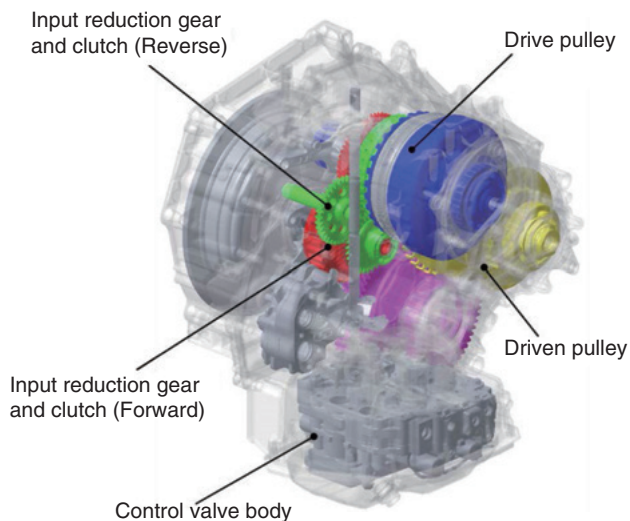


Fig. 2 Perspective view of developed CVT

Table 1 Specifications of developed CVT

Engine specifications	Type		0.66 L NA	0.66 L TC
	Max. power [kW/rpm]		43 /7300	47 /6000
	Max. torque [Nm/rpm]		65 /3500	104 /2600
Ratio	Primary reduction	Forward	1.605	1.375
		Reverse	1.187	
	Pulley (low-overdrive)		2.293-0.420	
	Final		5.176	
Dimensions	Overall length [mm]		348	
	Shaft axis distance [mm]	Input /Drive	87	
		Drive /Driven	147	
		Driven /Final	123.5	
Weight (wet) {Idle stop}	[kg]		58.5 {59.6}	58.6

- (4) The case was made of a simple two-piece construction.
- (5) 4WD was supported by adopting different T/C case and differential specifications for 4WD.
- (6) The hydraulic control system was placed all together under the transmission, and hydraulic circuits capable of supporting 4WD were adopted.
- (7) Idle stop capability was supported merely by adding an electric oil pump.

4. Main Technologies

4.1. Parallel-shaft, Helical Gear Primary Reduction Mechanism

For the present development of a new CVT for K-car use, the development of a new metal V-belt could also be considered. This would have involved elimination of primary reduction and development of a metal V-belt matched to the maximum power of the K-car with the width reduced and the circumferential length adjusted to the necessary ratio range. On the other hand, common use of the previous metal V-belt with a 24-mm width would provide an extra margin of allowable input torque that would enable use with primary reduction and with increased input torque to the metal V-belt. The choice was made to take the previous 24-mm wide metal V-belt for use in the present CVT.

A primary reduction mechanism using helical gears for location between parallel shafts was placed upstream of the continuously variable transmission mechanism operated using a metal V-belt. The forward gear set and clutch were placed inversely with the reverse gear set and clutch for a compact construction. The reverse clutch was positioned on the driven side of the parallel shafts so as to halve the differential rotation within the clutch and reduce friction. That structure is shown in Fig. 3.

In general, the metal V-belt has the greatest transmission efficiency when operated close to the belt's allowable

input torque. As the input speed increases, the transmission efficiency tends to diminish.

The rate of fuel consumption enhancement in the NA engine due to the difference in primary reduction ratio was simulated by taking fuel consumption in 10-15 mode and without primary reduction (primary reduction ratio of 1.0) as the base. Figure 4 shows the results. In the NA engine, a 1% or better enhancement in fuel consumption was obtained with primary reduction ratios from 1.45 to 2.0.

When the primary reduction ratio is higher, the torque passing through the belt increases. When the torque passing through the belt is greater, an axial thrust matched to the torque becomes necessary, and a larger hydraulic pressure is required by the system. The relationship between the primary reduction ratio and system-required pressure is shown by the red lines in Fig. 5 and Fig. 6. Increasing the pressure generated by the oil pump also allows the primary reduction ratio to be higher. However, this also means the various parts need greater capability to resist pressure. Considering such factors and the capacity of the oil pump, it was decided that the maximum pressure generated in the present CVT should be 5 MPa.

The elimination of a secondary reduction mechanism also means that the overall gear ratio is determined by the product of the stepless gear ratio of the metal V-belt and

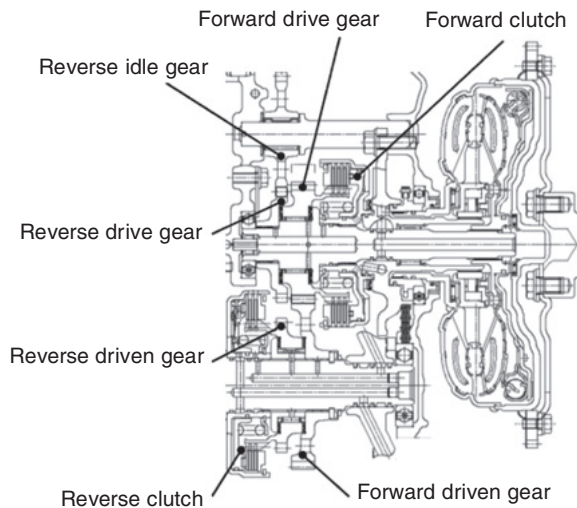


Fig. 3 Structure of helical gear input reduction

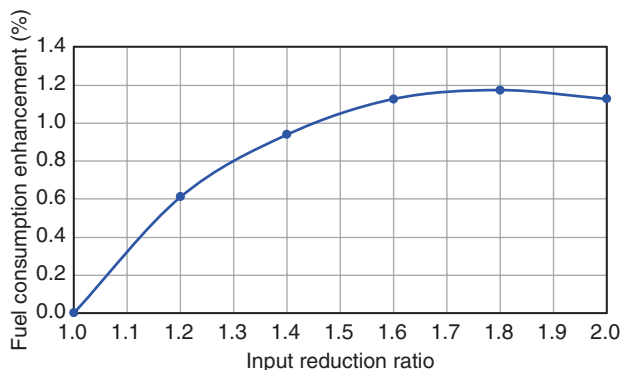


Fig. 4 Input reduction effect for fuel consumption of NA

the primary reduction ratio and the final reduction ratio. When optimal overall gear ratios are respectively set for the NA and TC, the final reduction ratio is determined by the primary reduction ratio. The relationship between the primary reduction ratio and final reduction ratio is shown by the blue lines in Fig. 5 and Fig. 6. Limitations on the front-to-back length of the transmission place constraints on the maximum value that can be obtained for the final reduction ratio (the maximum diameter of the final gear), and as a result there are constraints on the primary reduction ratio that can be obtained. In the present CVT, the upper limit on the final reduction ratio is 5.5.

Consequently, the primary reduction ratios within the shaded regions of Fig. 5 and Fig. 6 can be set. Since the gear ratio for a parallel-shaft, helical gear primary reduction mechanism can be readily set, separate primary reduction figures of 1.605 for the NA and 1.375 for the TC were adopted. The product of this primary reduction ratio, the torque generated during a stall in the NA and TC, and the torque ratio of the respective torque converters is the maximum torque that is input to the belt transmission mechanism. Setting the maximum input torque to the belt transmission mechanism more or less equal for the NA and TC specifications enabled their use of common parts downstream from the belt transmission mechanism.

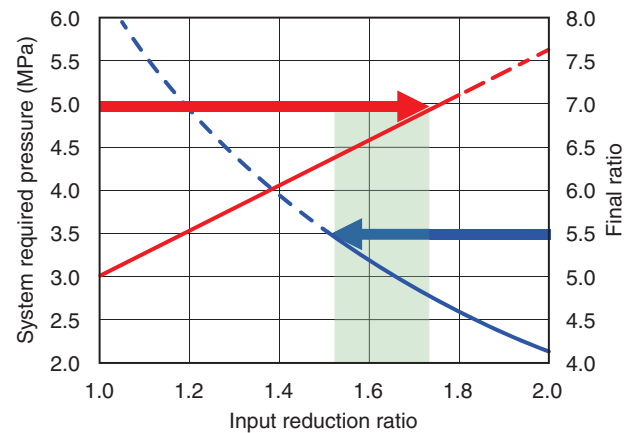


Fig. 5 Input reduction ratio for NA

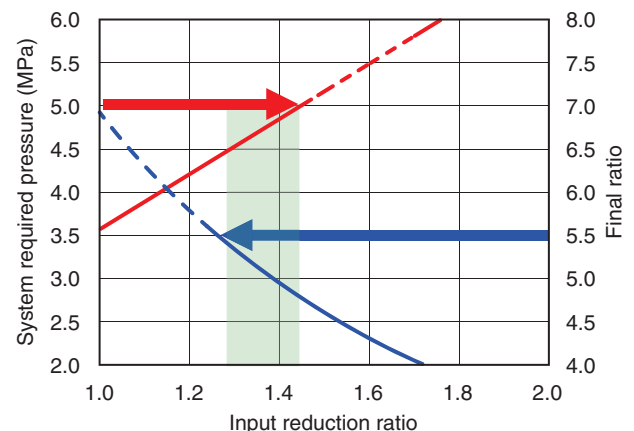


Fig. 6 Input reduction ratio for TC

4.2. Gear-Meshing Frequency Reduction

In general, the noise level of the cabin of a vehicle has greater sound pressure on the low-frequency side, and lower pressure on the high-frequency side of the sound. Moreover, if gear sounds are at or below the sound pressure of the noise level of the cabin, then they will not be audible. When the meshing frequency of the gears is high, then there is a need to lower the level of the gear sound pressure, and it will be necessary to give consideration to the machining accuracy of the gears and to gear support stiffness. In the previous CVT, torque flows from the belt-pulley transmission mechanism, to secondary reduction, and to final reduction. When in overdrive, the belt-pulley transmission mechanism speed rises and then slows down so that the gear-meshing frequency of the secondary reduction gear rises higher, and this means that gears with higher-precision tooth profiles will be required. The present CVT adopts a primary reduction mechanism and eliminates the secondary reduction mechanism, thereby holding the gear-meshing frequency down to a low level. This structure is advantageous for the quietness of the cabin. The previous CVT also adopted a planetary gear mechanism for the forward-reverse switching mechanism, so that the planetary gear mechanism would operate when the vehicle was driven in reverse. At that time, the planetary gear would have a high gear-meshing frequency, so it was also necessary to consider such measures as enhancing the precision of the gear tooth profile in order to lower the gear sound pressure. The present CVT also has helical gears between parallel shafts configured for use in reverse. This structure provides lower gear-meshing frequency and also allows the gear tooth profile to be adjusted readily.

Gear-meshing frequencies were calculated with the ratios for use with a K-car and on the assumption of a CVT (in the conventional arrangement) equipped with a secondary reduction mechanism and of a CVT (planetary gear-type primary reduction) that would perform primary reduction using a planetary gear mechanism. The results are shown in Fig. 7. When cruising in forward gear at 60 km/h, the present CVT can hold the gear-meshing frequency to a level 1000 Hz or more lower than the conventional arrangement CVT and the planetary gear primary reduction CVT.

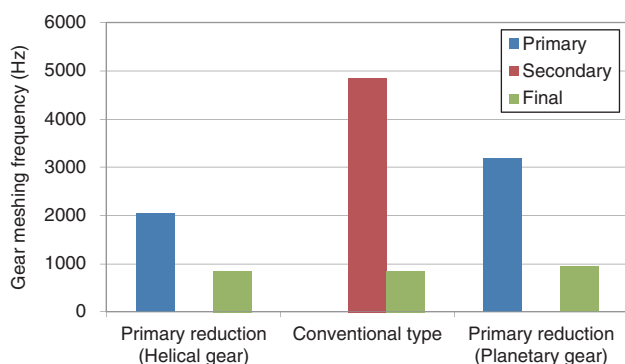


Fig. 7 Gear meshing frequency at 60 km/h cruise

Figure 8 similarly shows the gear-meshing frequency when moving in reverse at 20 km/h. The primary reduction mechanism of the present CVT has a gear-meshing frequency at a level one-half or lower than the CVT that activates a planetary gear mechanism.

4.3. Case Construction

4.3.1. Shaft arrangement

In order to obtain as much cabin space as feasible within the limited space of a K-car, the dimensions of the input shaft and output shaft in the front-to-back direction were brought as close together as could be managed. The front-to-back length of the transmission was made 4 mm shorter than that of the previous four-speed AT⁽¹⁾ for K-car use. The hydraulic pressure control solenoid was also moved from in front of the transmission to a position below it, and the profile along the outside diameter was matched to the dashboard profile. This pushed back the amount of engine travel (stroke) during a collision and contributed to a 70-mm increase in the interior length of the cabin. A side view of the present CVT is shown in Fig. 9.

4.3.2. Two-piece case construction

A structure that supports the drive and driven pulley shafts at both ends was adopted, and a two-piece basic case structure was adopted, with a T/C case and a transmission

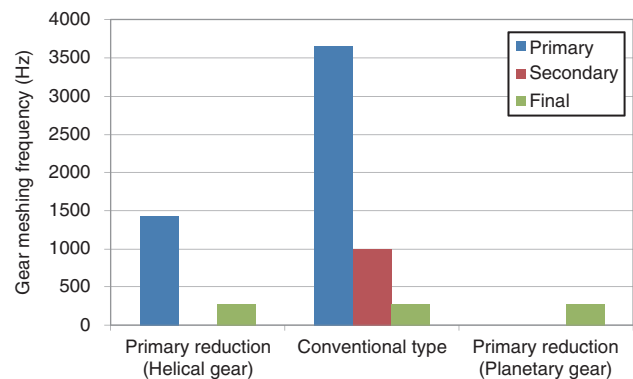


Fig. 8 Gear meshing frequency at reverse 20 km/h

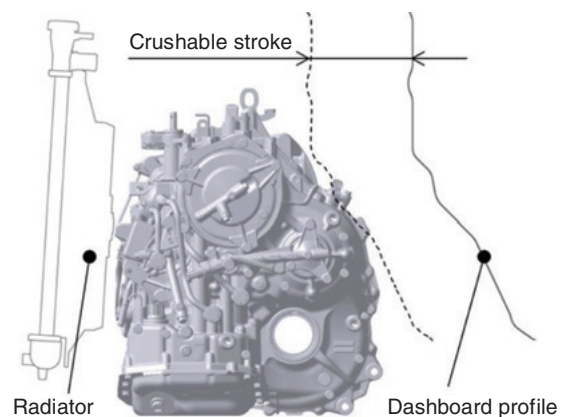


Fig. 9 Side view of developed CVT

case (T/M case). The partition between the cases and the flange mating surfaces could be reduced, realizing a reduction in weight. The case construction is shown in Fig. 10.

4.3.3. 4WD support

Here, a transfer was developed that is coaxial with the differential and that connects the differential case and the transfer drive gear with a spline. The only special 4WD parts in the CVT are the T/C case and the differential case. The differences between the specifications for the 2WD T/C case and the 4WD T/C case were minimized, and the machining and casting plans for the manufacturing process were commonized, making the structure advantageous in terms of manufacturing cost as well as quality. That configuration is shown in Fig. 11.

4.3.4. Formed-in-place gasket

The paper gasket used before was replaced with a

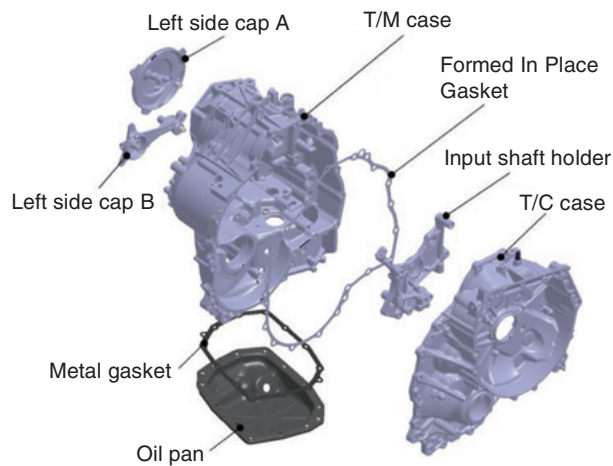


Fig. 10 Case construction

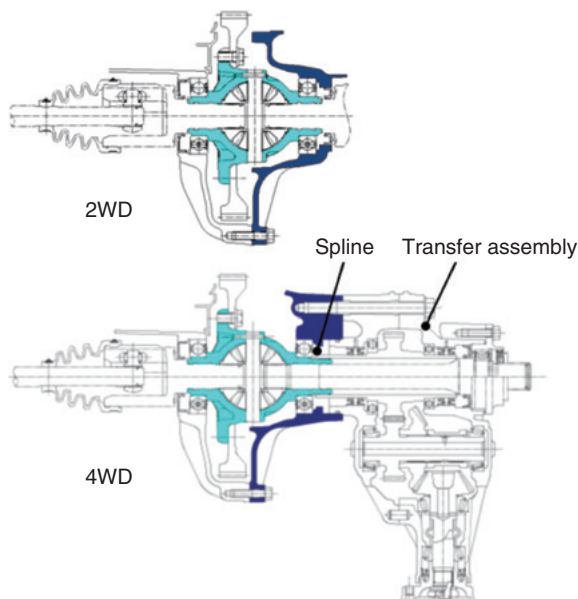


Fig. 11 Configuration of transfer coaxial with differential

formed-in-place gasket (FIPG) for the CVT in order to reduce the case weight and lower its cost. The elution of silicone from the FIPG into the CVT fluid has raised issues of reduced metal friction coefficient μ and diminished defoaming property of the CVT fluid. The FIPG components were therefore reexamined and a new gasket material was developed that would not affect the CVT fluid. Adoption of the FIPG enabled the gasket to follow tensile relative movement and lateral relative movement at the case mating surfaces. This allowed the design to shift from the previous paper gasket, which depended on bolt axial force, to instead optimize tightening stiffness. This enabled the use of shorter bolts as well as reduction in the number of bolts used. This achieved a reduction of 1100 g in weight compared to when the paper gasket was used, and also yielded a reduction in cost.

4.3.5. Metal gasket

In the interest of maintainability and compactness, the hydraulic control system consisting of the valve body and other such elements was positioned together under the transmission. A metal gasket was adopted for use as a seal between the steel-sheet oil pan and the aluminum case. This enabled use of a steel-sheet oil pan, which is lower in cost than an aluminum oil pan. This configuration is shown in Fig. 12.

4.4. Hydraulic Control System

A reasonably priced hydraulic circuit structure with a smaller number of solenoids was adopted for the hydraulic control system. A new control that uses the forward gear clutch to cut off sudden excessive input torque was also adopted. An electric oil pump was additionally configured to support idle stop.

4.4.1. Hydraulic circuit

The previous hydraulic circuit was configured from three linear solenoids and two on-off solenoids. One of those linear solenoids was switched back and forth between use for forward-and-reverse clutch control and use for lock-

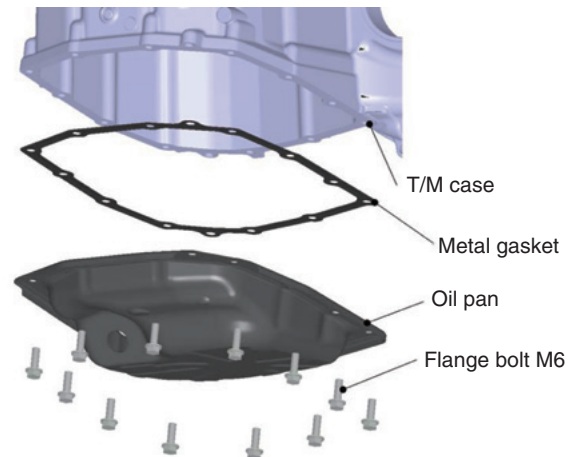


Fig. 12 Metal gasket of oil pan

up clutch control. The circuit was therefore configured so that, while it was being used to control lock-up clutch capacity, it could not control forward clutch capacity. The previous hydraulic circuit is shown in Fig. 13.

The hydraulic circuit for the present CVT controls forward clutch capacity and lock-up clutch capacity independently. A special linear solenoid was therefore set up for each. The linear solenoid for the forward clutch constantly controls forward clutch capacity so it will be at or below belt transmission capacity. The lock-up clutch control is configured for the capability to switch the lock-up clutch on-off circuit and to control the lock-up clutch capacity within the pressure control range of the linear solenoid. The switching solenoids have been eliminated.

A direct control system has been adopted for the pulley pressure control circuit, which follows the configuration of the previous control circuit. It uses two linear solenoid valves to perform separate control of the drive and driven pulley pressure while constantly keeping line pressure at the minimum necessary level. The above configuration achieved a balance between enhancing its functionality and making it more compact and lighter in weight. Figure 14 shows the hydraulic circuit of the present CVT.

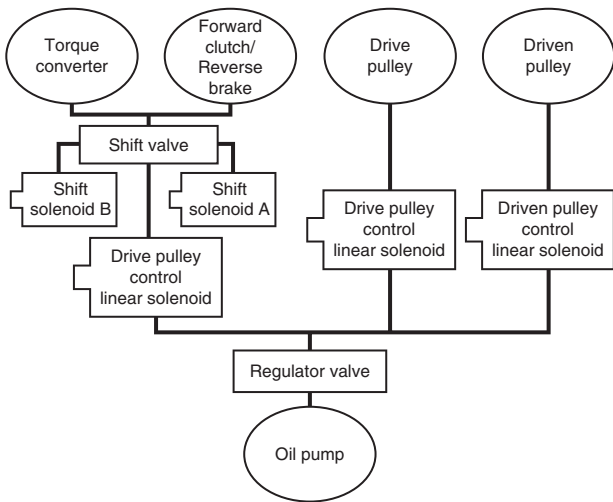


Fig. 13 Hydraulic circuit of previous CVT

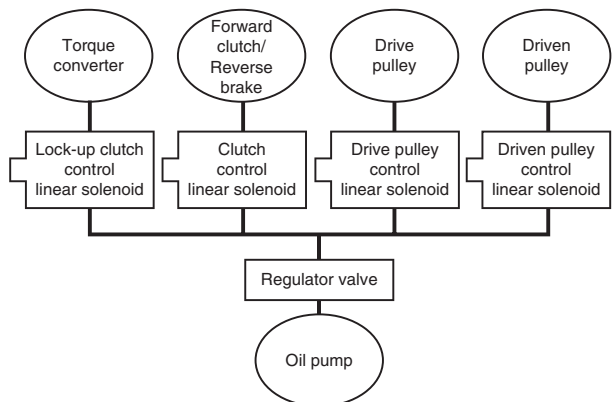


Fig. 14 Hydraulic circuit of developed CVT

4.4.2. Control for 4WD support

The adoption of constant clutch capacity control eliminated the difference between 2WD and 4WD belt input torque and allowed the use of common component parts. A new torque-on-demand control at launch was also adopted to comply with the rear differential input torque characteristics, achieving smooth launch characteristics.

4.4.3. Control for idle stop support

With the adoption of an electric oil pump, the engagement of the forward clutch is not completely prevented even during idle stop. This achieved a highly responsive and smooth launch when restarting. Figure 15 shows launch responsiveness from idle stop.

4.5. Increasing Efficiency

In addition to optimizing the primary reduction ratio, each part of the CVT was reexamined with a view to further enhancement of efficiency.

4.5.1. Reduction of oil agitation resistance

The adoption of the primary reduction mechanism resulted in the drive pulley being located above the oil level, and reduced the agitation resistance from rotation of the pulley. This was combined with adoption of a baffle plate to reduce the agitation resistance of the large-diameter final driven gear.

4.5.2. Oil pressure sensor

An oil pressure sensor was placed in the oil pressure control system to constantly sense the CVT working oil pressure, and learning control that provides feedback was adopted. This enabled reduction of the pressure margin and also lowered the oil pump drive torque and belt friction.

4.5.3. Cruising efficiency results

Figure 16 shows a comparison of the cruising efficiency of the present CVT and the CVT of other vehicles. An advantage of about 2% was obtained at vehicle speeds from 40 km/h to 100 km/h.

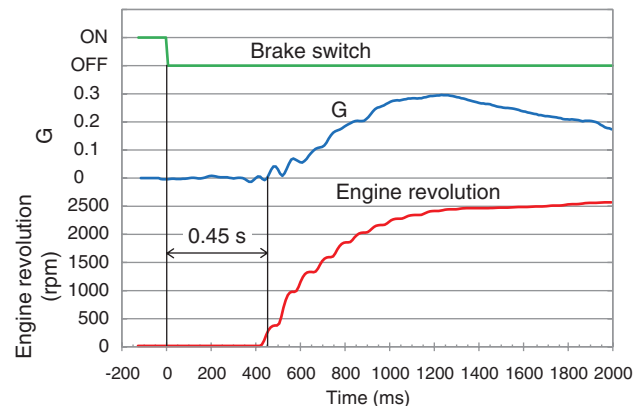


Fig. 15 Response time from idle stop

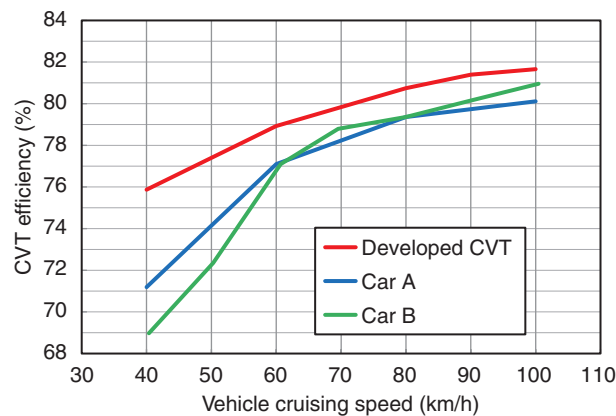


Fig. 16 CVT efficiency at cruise

5. Conclusion

A new model CVT was developed for K-car use and yielded the following results:

- (1) The adoption of a parallel-shaft, helical gear primary reduction mechanism achieved a CVT weight in the top class for CVTs in the 110-Nm allowable input torque class.
- (2) Optimal primary reduction ratio settings achieved cruising transmission efficiency in the top class for a K-car CVT.
- (3) Compared to the previous CVT, the gear-meshing frequency was lowered by 1000 Hz or more when cruising at 60 km/h, enhancing the quietness of the cabin.
- (4) The specifications for the torque converter and forward gear set were differentiated to support NA and TC engine versions. Support for 4WD was provided with the T/C case, the differential case for 4WD use, and a special transfer. Idle stop was supported with just the addition of an electric oil pump.

References

- (1) Uchida, Y., Miyamoto, O., Ootani, T., Maruyama, T., Sakai, K., Tanimoto, S.: Development of New Compact 4-speed Automatic transmission for Honda LIFE, Honda R&D Technical Review, Vol. 16, No. 2, p. 79-86
- (2) Yamaguchi, M., Ootaki, M., Ito, K., Nirasawa, H., Totsuka, H.: Development of CVT for 2008 Model Year FIT, Honda R&D Technical Review, October, 2008, p. 12-17
- (3) Sato, Y., Urano, J., Ohashi, A., Sakuma, S., Yamaguchi, M.: Development of CVT for 2004 Model Odyssey, Honda R&D Technical Review, Vol. 16, No. 2, p. 87-94

■ Author ■



Takashi NAKAMURA



Keiichi HANYU



Koichi KONISHI

Development of Electric Motor for “EV-neo”, Electric Motorcycle for Business Use

Naoki KOBAYASHI*

ABSTRACT

A new DC brushless motor, which has an equivalent driving performance to a 50 cm³ displacement scooter engine, has been developed to be equipped in a new electric motorcycle for business applications. The requirement for the maximum motor torque was mitigated by applying reduction gears and the drive unite that can be mounted on the wheel side was realized by reducing the size of the motor, together with the united structure of PDU (Power Drive Unit). In this motor, IPM (Interior Permanent Magnet) structure was implemented to cope with the high rotation speed of the motor, and the concentratedly wound stator coil was implemented for downsizing. The motor performances with the maximum power of 2.8 kW, the maximum torque of 11 Nm, the practical efficiency of 75% or more, and the maximum motor speed of 9000 r/min or more, were achieved, satisfying the specifications required for the drive train unit of business application motorcycles. In production processes, the split core stator structure and the bus rings for coil wiring were adopted to make it possible to manufacture them in the existing mass production facilities for the electric carts production, taking the cost reduction into good account as well.

1. Introduction

The developments of electric vehicles are under way as a means to reduce CO₂ emissions due to the growing concern on environmental issues in recent years. Ever since the start of mass-production of the electric scooter the CUV-ES (Fig. 1) in 1994, we have been continuing the researches for the technology of electric motorcycles up to now. Following that development, further enhancing the conventional techniques and know-how applied to the previous mass production model, we have newly developed the electric motorcycle EV-neo suitable for business applications in 2010 (Fig. 2). The specifications of the new and previous models are shown in Table 1. The new electric motorcycle employs the high-power lithium-ion battery to achieve a quick charge operation which

Table 1 Specifications of previous and developed vehicle models

	Previous model	Developed model
Dimensions L X W X H (m)	1.785 x 0.615 x 1.035	1.830 x 0.695 x 1.065
Vehicle mass (kg)	130	106
Rated output (kW)	0.58	0.58
Maximum power (kW)	3.3	2.8
Maximum torque (Nm)	9	11
Battery type (V-Ah)	Ni-Cd 86.4 V-20 Ah	Lithium-ion 72 V-12.6 Ah



Fig. 1 Previous mass production electric motorcycle (CUV-ES)



Fig. 2 Newly developed electric motorcycle (EV-neo)

* Motorcycle R&D Center

requires only 20 minutes from 0% up to 80% SOC (State of Charge), while the battery package also provides a satisfactory in driving power (discharging performance) and quick charging (charging performance) even in temperatures as low as -10°C . It achieved hill-climbing departure capability of 12-degree with a rider and a 30 kg load, which is comparable to a conventional scooter equipped with a 50 cm³ displacement gasoline-fueled engine. The motor introduced here was developed uniquely targeting the above-mentioned driving performances. In this paper, the technologies applied for this motor development and its performances are discussed.

2. Development Aims

Normally, attempts to realize an adequate driving distance in electric vehicles (EV) result in increased battery load, making the vehicles heavy and costly. Given this, it was particularly emphasized within the development requirements of EV-neo that it should achieve low weight and low cost. Consequently, the requirement for the driving distance should be met while employing the minimum battery load. Based on the above mentioned circumstances, the following were set as the main objectives for the development:

- (1) Reduction of size of drive unit
- (2) Realization of increased power efficiency
- (3) Realization of low cost

In the case of an ordinary engine-powered scooter, the power from the engine is transmitted to the rear wheel via a belt-converter-type transmission system. The belt converter with reduction gears was also used in the CUV-ES, which is the former model earlier mentioned (Fig. 3). For that reason, the motor had to be mounted in the same location as where the engine was originally mounted; hence the entire drive train unit was as bulky as that of an engine-powered scooter. In our development, we aimed at downsizing of the whole drive train unit and enhancing the transmission efficiency by elimination of a belt converter and downsizing of the motor. We also aimed at realization of low cost with increased productivity by employing such a structure that permits motor production to be carried out in the existing mass-production facility for the electric carts manufacturing.

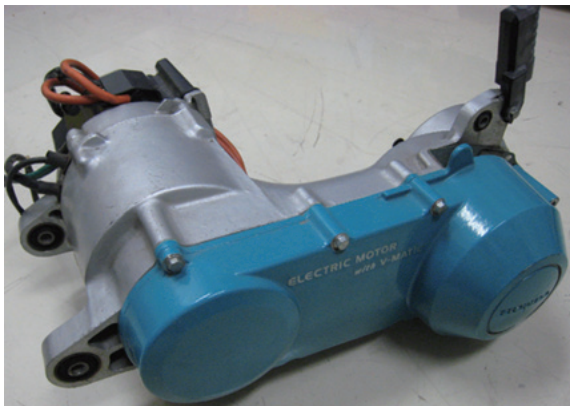


Fig. 3 External view of belt converter-type transmission

Table 2 Major specifications of new and previous motors

	Previous motor	Developed motor
Type	DC brushless	DC brushless
Magnets	Nd-Fe-B	Nd-Fe-B
Number of poles	4	8
Rotor type	SPM	IPM
Winding type	Distributed winding	Concentrated winding
Cooling system	Forced air cooling	Natural air cooling

3. Applied Technologies

Major specifications of the new and previous motors are shown in Table 2 and the entire drive train unit is shown in Fig. 4. Both motors are DC brushless type motors. Compared to the conventional SPM (Surface Permanent Magnet) structure, which was applied to the CUV-ES, the IPM (Interior Permanent Magnet) structure was chosen for the new motor as it excels in the efficiency and strength when spinning at high revolutions.

For cooling of the motor, we selected the natural air cooling assisted by the heat transfer to the drive train unit and by the headwind caused by riding forward. As a result, additional parts to cool the motor were unnecessary, and a weight increase was avoided.

3.1. Drive Unit Configuration

3.1.1. Use of coaxial clutch

The detail structure of the drive train unit is shown in Fig. 5. The automatic centrifugal clutch was employed to improve driving efficiency by avoiding operations in the low-revolution, high-torque zone where the motor efficiency lowers. Also, by avoiding a high stalling torque at 0 r/min, the currents in the motor and PDU are reduced, and temperature rises due to current are suppressed. As a result, the reliability in low-speed, high-load operations such as a start on an uphill slope was ensured without increasing heat resistance of the motor and PDU. Moreover, as the clutch separates the motor from the driving wheel after stopping, it is made easy to manually move the electric motorcycle when the power is turned off.

To downsize the drive train unit, the above-mentioned

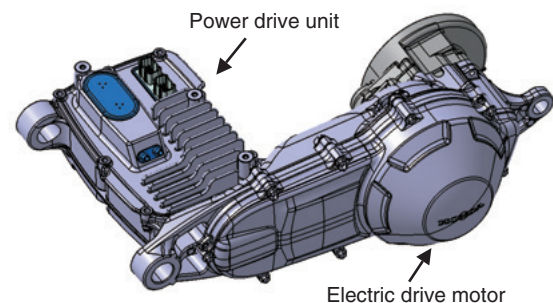


Fig. 4 External view of drive unit assembly (EV-neo)

clutch and motor were laid coaxially. The motor and the clutch were laid on the same axis by using a hollow shaft for motor rotor, inside of which the drive shaft passes through. The coaxial clutch and the downsizing of motor permit mounting of the motor on the wheel side while ensuring adequate lean angle of the vehicle, thus enabling high transmission efficiency and downsizing of the drive train unit.

3.1.2. Integration with PDU

The PDU and drive train were integrated into a single unit to downsize the drive train unit. As the coaxial clutch allows mounting of the motor on the wheel side, the PDU was installed in the space in front of the driving wheel and integrated into the drive train unit. This integration allowed the system to have the swingarm as an adequate heat mass for the natural cooling of the PDU, while, in the CUV-ES, air-cooling system was installed. Also, as the motor and the PDU were connected inside of the unit, the power lines were shortened over 50% from 500 mm to 240 mm. The transmission loss from the PDU to the motor was reduced by more than 50% compared to the system having a PDU separately on the vehicle body. Moreover, the integrated design allowed the elimination of wiring parts protection from excessive bending and the elimination of waterproof structure, which are necessary when wires are connected from outside of the unit. This structure also contributed to the reduction of cost by decreasing the number of parts necessary.

3.2. Increased Motor Speed

By applying the reduction gears to the motor while designing the motor to adapt to the high-revolutions,

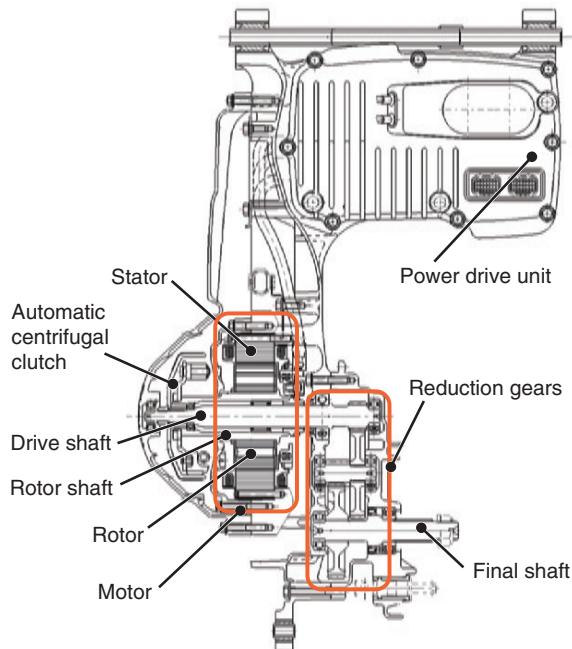


Fig. 5 Sectional view of drive train unit

downsizing and weight reduction of the drive train unit had been achieved.

Incidentally, when the main concern in the system design is focused on the increase of frictions in the drive train caused by the use of reduction gears, the in-wheel motor, which directly drives the wheel, may be one of the solution options. However, compared to a system using reduction gears, the in-wheel motor system requires a larger motor torque due to the limitations of reduction ratio.

In the case of a motor using the magnetic torque, in order to attain higher motor torque, it is necessary to increase the inter-linkage magnetic flux by the magnet and/or the number of poles, which causes an increase of motor outer diameter and/or the stator thickness. As more magnets, magnetic steel plate, and coils have to be used, the motor becomes bulky and its weight increases.

Consequently, we chose to provide the drive train unit with reduction gears and it was set as a target specification to have hill climbing capability of 12-degree carrying a rider and a 30 kg load. With that condition, the necessary torque on the drive wheel axle was 150 Nm. To attain this necessary axle torque considering the motor shape limitations demanded from the manufacturing equipments and the increased friction by the use of high revolution range, we chose the reduction ratio of 13.9, and set goals of maximum torque at 11 Nm and maximum motor speed at 9000 r/min.

3.3. IPM Motor

To eliminate the belt converter, it was necessary to increase motor revolution by 50% compared to the motor of the CUV-ES. Moreover, as efficiency enhancement is necessary across a broader revolution zone, we have chosen the IPM type motor which uses the reluctance torque in addition to the magnetic torque. To ensure rotor strength in high revolutions, the magnet for each pole was split into two pieces and was placed inside of the rotor yoke. Figure 6 shows comparison of SPM and IPM shapes in the rotor yoke and stator core. As for the rotor dimensions, the external diameter was increased by 10% compared to the CUV-ES rotor, and the magnetic flux needed to satisfy the maximum torque requirement was ensured by increasing the number of poles from four to eight. In addition, the shapes of the magnets were made thinner by using magnets

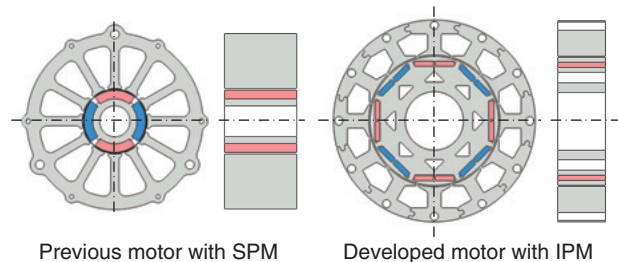


Fig. 6 Comparison of shapes between previous and developed motors

which had greater magnetic power by 7% than the magnets used in the CUV-ES, making it possible to keep the same volume of magnets as used in the CUV-ES, even though the external rotor diameter was increased.

3.3.1. Combined Use of Reluctance Torque

When the motor torque with combined use of reluctance torque is denoted as T_q , the torque constant of magnetic torque as K_t , the torque constant of reluctance torque as K_{rt} , the electric currents for d axis and q axis as i_d and i_q respectively, the inductances for d axis and q axis as L_d and L_q respectively, the number of pairs of magnetic poles as P_n and the electromotive force constant as K_e , the motor torque T_q can be expressed by the formulae (1), (2) and (3). The first term in the right side of the formula (1) indicates the magnetic torque, and the second term indicates the reluctance torque.

$$T_q = K_t \cdot i_q + K_{rt} \cdot i_d \cdot i_q \quad (1)$$

$$K_t = P_n \cdot K_e \quad (2)$$

$$K_{rt} = P_n (L_d - L_q) \quad (3)$$

In the motor with SPM structure, which is also applied to the CUV-ES, L_d and L_q are equivalent, and no reluctance torque is produced, i.e. only the magnetic torque is produced.

When the induced voltage in the motor exceeds the voltage of the power source during high-speed operation of the motor, it is necessary to apply field-weakening control in order to restrict the magnetic field generated by the magnets. However, as the application of field-weakening current for magnetic field restriction does not contribute to the torque generation, the efficiency of motor generally declines.

For motors with SPM configuration, in which magnetic torque is solely generated, it is necessary to suppress the increase of induced electromotive force when enhancing efficiency at high revolutions by reducing the field weakening current. Though an increase of induced electromotive force can be controlled by reducing K_e by changing the number of turns of coils, K_t also decreases at the same time as shown in the equation (2). This means that the efficiency in low-revolution with high-torque operations goes down because the decrease of K_t causes an increase of necessary current to produce torque, though the efficiency at high revolutions is enhanced. In the SPM-type motor using only the magnetic torque, it is therefore difficult to enhance efficiency across a wide revolution range.

Meanwhile in the case of IPM-type motor, which uses the reluctance torque in addition to the magnetic torque, the magnetic torque under the same current condition decreases because K_t decreases along with a decrease of K_e just like in the SPM-type motor. The decrease of magnetic torque, however, can be compensated by the reluctance torque indicated by K_{rt} . It is therefore possible to maintain efficiency at the time of low revolutions while preventing

lowering of efficiency in high revolutions resulting from the field weakening control. It is therefore possible to enhance the efficiency for wider revolution speed range compared to the motors with SPM structure because high revolution operation is feasible without depending on the field weakening control not sacrificing the efficiency at low revolution⁽²⁾.

From the equation (3), the reluctance torque occurs from the variance between L_d and L_q , and L_d becomes larger than L_q in the case of the IPM type. Accordingly, the reluctance torque can be added to the magnetic torque when the current phase angle is 90-180 degrees⁽³⁾. In the case of the new motor for EV-neo, by the modification of rotor shape, we have attained such a rate of reluctance torque that is more than 20% of the entire torque when producing the maximum torque. Figure 7 shows the results of the developed motor on magnetic field analysis of the current phase angle and the produced torque.

3.3.2. Rotor Strength

To make a motor to bear high rotation speed operation, rotor strength becomes an issue. In order to relieve the stress on the rotor yoke due to centrifugal force, the magnet for each pole was divided into two pieces, and those were positioned inside the yoke (Fig. 8). The number of positions designed for bearing the stress was increased from two to three by adding the center rib to the both ends of the

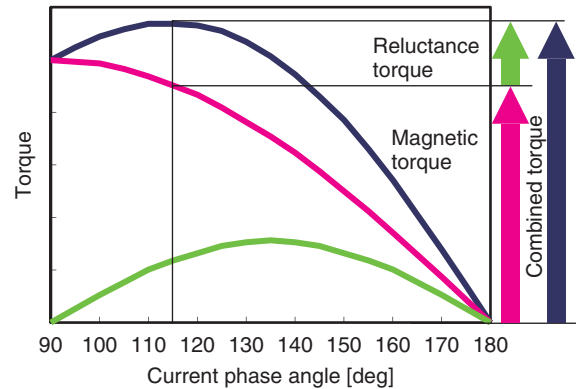


Fig. 7 Torque characteristics in relation to current phase angles

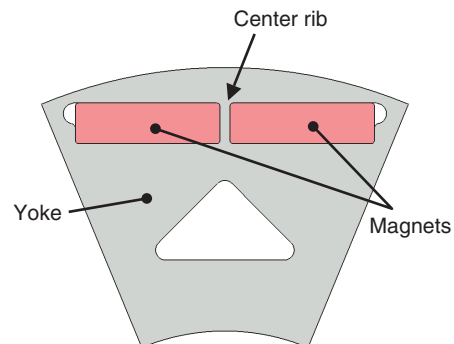


Fig. 8 Rotor structure

magnet. This made it possible to reduce the concentration of stress caused by centrifugal forces and to fulfill the requirement for the rotor strength at high revolution speed.

3.3.3. Elimination of Magnet Pasting

The efficiency of production had been enhanced by eliminating the process of pasting magnets to the rotor yoke and the cost was reduced by that. The use of adhesive was making the production efficiency enhancement difficult because it required the process controls for adhesive application and heat hardening. With IPM structure, which is adopted this time, magnets are maintained in the positions where they are originally inserted into the yoke because they are enclosed within the motor yoke. However, a clearance of around 0.1 mm is necessary between the rotor yoke and the magnets for insertion of the magnets, and the magnets can move within this clearance space. With this clearance, magnets can move and hit the rotor yoke repeatedly and that may result in cracks and wear on rust-prevention coatings applied to the magnets. In order to respond to this issue, a highly flexible nickel sulfamate coating was employed. To tackle this issue, the application of nickel sulfamate coating was intended and the conditions of application such as thickness of the coating were examined to meet the durability requirements by conducting vibration tests assuming accelerations and decelerations of the vehicle. By these measures, the feasibility was verified for elimination of the magnet pasting.

3.4. Concentrated Winding in Stator

Whereas the distributed winding had been adopted in the motor for the CUV-ES, the concentrated winding, which permits lowering of coil end height, was employed for the stator winding. Figure 9 shows the stator shape. With the employment of concentrated winding, the coil end height was reduced by 52% compared to that for the CUV-ES, contributing to the downsizing of the motor. In addition to the employment of concentrated winding, in order to make the motor shape suitable for mounting on the wheel side, the stator outer diameter was increased by 17% and the stator thickness was reduced by 31% from the motor of the CUV-ES (Fig. 10).

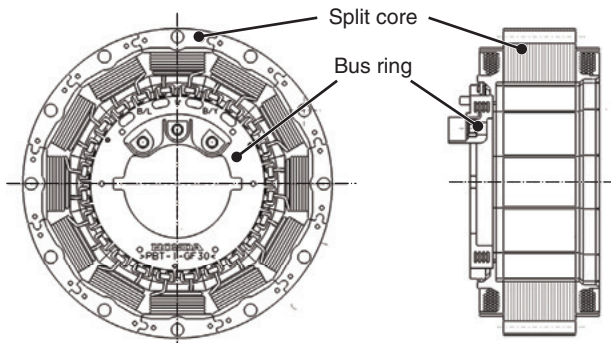


Fig. 9 Structure of stator

3.4.1. Split Core

To allow manufacturing using the existing motor mass-production equipment, the split core stator, which is the same structure as formerly developed motors, was employed⁽⁴⁾. The use of split core stator allowed application of the existing bobbin-type winding machine suitable for the concentrated winding. Also, when compared to the manufacturing of one-piece stator core, the split structure allowed the yield rate enhancement of material, and this contributed to cost reduction. The shape of the split core structure is shown in Fig. 11.

3.4.2. Self-bonding Wires

Self-bonding wires, which can be fixed to each other by adhesion after they are heated, were employed for the coils. Figure 12 shows a cross-section of a self-bonding wire. An

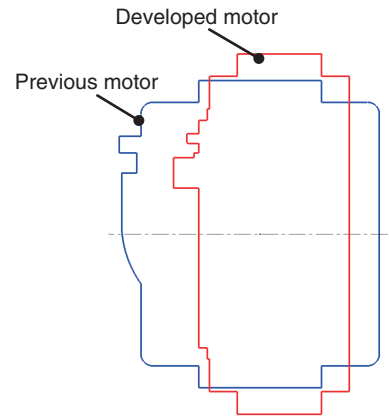


Fig. 10 Comparison of contours of new and previous motors

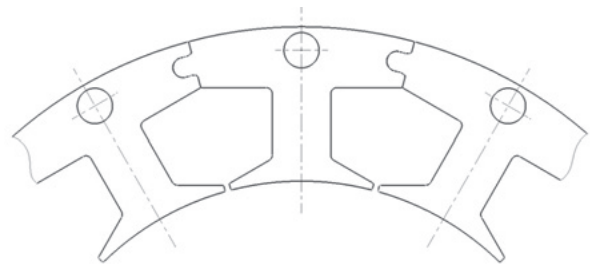


Fig. 11 Split core structure

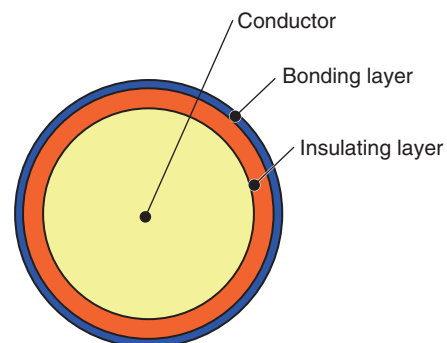


Fig. 12 Cross-section of self-bonding magnet wire

insulating layer is placed between a base layer formed by a conductor and an upper bonding layer. Wires of the coils can be fixed together by applying heat on the coils melting the uppermost bonding layer after the winding process. As the use of self-bonding wires had made it possible to dispense with the wire adhesion process using impregnated varnish, it was realized to increase the productivity by reducing the number of processes.

3.4.3. Bus Ring

To connect the concentrated-winding coils, the bus-ring was employed. Figure 13 shows structure of the bus-ring. The equivalent circuit, after completion of connection between coils and bus rings, is shown in Fig. 14, which shows the composition of four coils connected in parallel for each phase of UVW. By locating the bus-ring in the inner side of the coil ends, the thickness of the motor was reduced as an increase of motor thickness caused by connecting wires was reduced, enabling the miniaturization of electrodes in the bus-ring as well.

Fusion was used as the method to connect the coils and the bus ring. As Fig. 15 shows, firstly, the coil end is mechanically bound to the U-shaped section of the bus ring temporarily. Next, the heat generated by the prior current, which goes through in vertical direction in the figure between the poles, removes the films of the insulating layer. Following this, electrical bonding is completed

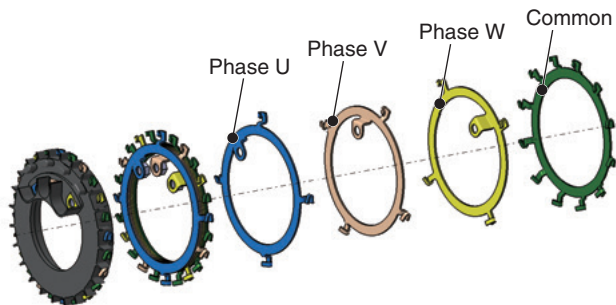


Fig. 13 Structure of bus-ring

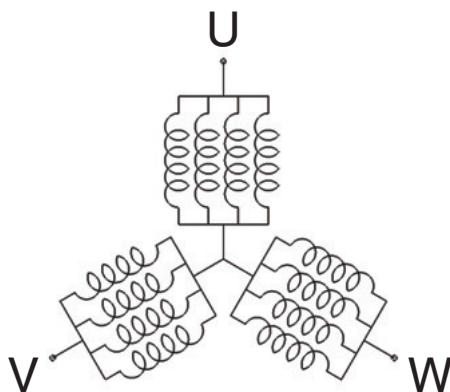


Fig. 14 Equivalent circuit for the motor

by fusing the surface of conductor by means of primary current. The application of fusing technique contributed to the enhancement of productivities through making the time required for wire bonding shorter by eliminating the process of removing insulating layers needed in other wire bonding methods.

As for the bus ring configuration, the exclusive design was implemented for the inner structure in which the creeping distances between electrodes of each phases are enlarged. This enabled the shapes of bonding portions to be the same structure commonly handled with bus rings used in exiting motors while enhancing the isolation performances. This made it possible to share the use of existing fusion equipment, resulting in cost reductions.

3.5. Rotation Angle Sensor

The employment of a Hall IC-type rotation angle sensor to the rotor and the application of PDU, with an error correction function to the detection angles, made it possible to eliminate the mechanical adjustments of sensor positions, increasing the productivity while reducing the working hours needed when sensor components are to be replaced. Figure 16 shows the detailed layout of the sensor. Hall IC-type sensors have the characteristic to produce errors when operated at high temperatures, and the sensor was therefore positioned on the opposite side of the clutch in order to

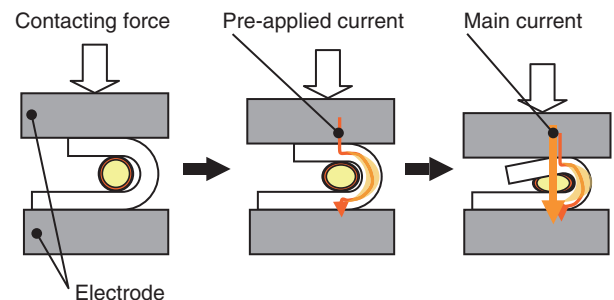


Fig. 15 Process of welding by fusing

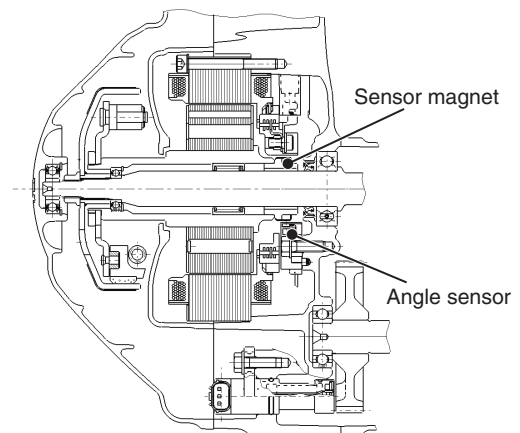


Fig. 16 Layout of angle sensor

ensure the reliability. By locating the ring magnets for the sensors on the inner side of the bus ring, any increase of the sensor dimension in the axial direction had been mitigated, contributing to the size reduction of the motor.

In brushless motors, the variations in output power and efficiency are derived from the angular sensing errors caused by the deviations in positions of installed sensors and/or positions of the magnetic phase of the sensor magnet, because the current control to the motor is performed referring to the index angular signals generated by the angular position sensor. In order to cope with this issue, the installation position of the sensor was mechanically adjusted for the motor used in the CUV-ES, and it was challenging to increase the productivity. In the EV-neo, the circuit, which compares the motor induced voltage waveforms with the sensor signals, was included within the PDU in order to detect angular errors, and it was made possible to correct these errors electronically in the process of the inspections for complete vehicles while increasing the productivity as well. This correction function contributes to the reduction of working hours when parts are replaced for maintenances as it is possible to adjust angle errors without dedicated tools.

4. Motor Performance

4.1. Output

With the above-mentioned techniques applied, we attained the target motor output performance; maximum power 2.8 kW, maximum torque 11 Nm, maximum motor speed 9000 r/min. Compared to the motor for the CUV-ES, the torque density was increased by 37%, and the motor weight was reduced by 11%. Figure 17 shows results of output measurement. With the increase of maximum torque by 22% and maximum motor speed by 52% compared to the motor for the CUV-ES, we achieved the following: the performance of the departure on a 12-degree uphill slope with a rider and a 30 kg load on the vehicle, and a maximum speed performance using the fixed reduction ratio without using a belt converter.

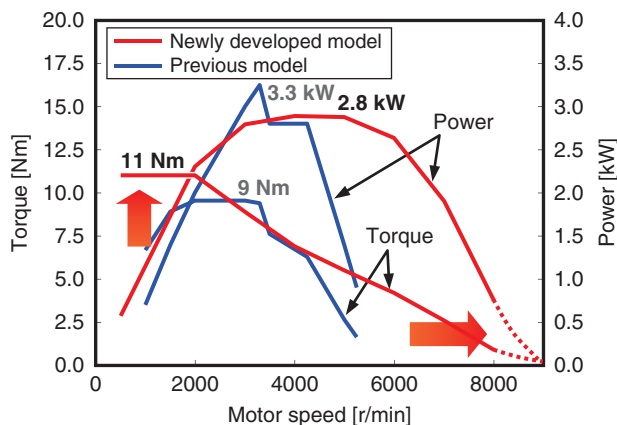


Fig. 17 Comparison of output performance for developed and previous motors

4.2. Efficiency

We attained the maximum motor efficiency of 92.7%, and 75% or higher efficiency in the actual vehicle operation zone. The result of confirmed efficiency is shown in Fig. 18. As the performance as a whole vehicle, the range of 34 km at a constant speed of 30 km/h was achieved, by reducing friction in the drive system through the elimination of a belt converter and the elimination of forced air cooling system together with the weight reduction by downsizing.

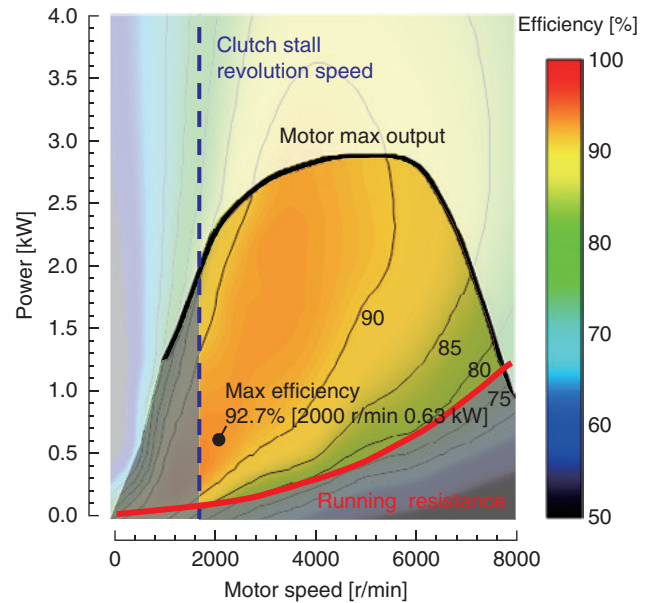


Fig. 18 Measured motor efficiency characteristics

5. Conclusion

The following has been achieved through the development of the motor for the electric motorcycle of business applications.

- (1) The drive train unit was downsized through the coaxial layout of automatic centrifugal clutch and motor, and the size reduction of the motor by adoption of concentrated winding to the stator coils.
- (2) A high-revolution motor was realized through the development of IPM motor, exclusively designed for electric motorcycles.
- (3) The cost reduction was achieved through the production efficiency enhancement by making the motor to be manufactured in the existing production facilities by adopting the same structure as the formerly developed motors, such as split core stators and bus rings for coil wire bonding.

References

- (1) Nakazawa, Y., Kumagai, C.: Development of Electric Scooter for Practical Use, HONDA R&D Technical Review, Vol. 6, p. 17-23

- (2) Takizawa, K., Itoh, T., Satoh, H.: Development of High Power Density and High Efficiency Traction Motor for FCV, EV, HONDA R&D Technical Review, Vol. 12, No. 2, p. 105-112
- (3) Fukushima, T., Iwai, A., Yamaguchi, N., Shinoki, H.: Electric Drive Motor for Fuel Cell Vehicle FCX Clarity, Honda R&D Technical Review, April, 2009, p. 63-70
- (4) Ichikawa, K., Hirose, Y., Yamamoto, T., Yamagishi, Y., Ishikawa, T.: Development of Monpal ML200 4-Wheel Scooter, Honda R&D Technical Review, Vol. 18, No. 2, p. 28-35

■ Author ■



Naoki KOBAYASHI

Development of New ASIMO

– Realization of Autonomous Machine –

Satoshi SHIGEMI*

Koji KAWABE*

Takahiro NAKAMURA*

ABSTRACT

The new ASIMO was developed as a step toward the goal of achieving “a robot that coexists and cooperates with people and performs useful functions in society.” The purpose was to give it the capability to sense changes in the circumstances of real-life environments where people are present, to take autonomous action, and to continue acting. New multi-modal sensing technology, situation estimation and prediction technology, and autonomous behavior generation technology were developed, greatly advancing the robot’s ability to adapt to situations. The development of physical capabilities included a many-axis hand capable of independent control of five fingers and lightweight, highly responsive hardware with heightened agility to enable continuing operation in a mixed environment with people. In terms of human response functions, therefore, the robot became able to adapt to situations to work appropriately with people. In terms of walking functions, it became able to predict the movements of multiple people and cross their paths without stopping, as well as to maintain its balance even on irregularities or other changes in the road surface. In terms of work functions, it became able to hold an object with a stable grip and to manipulate things with its fingers.

1. Introduction

Honda has for a long time engaged in research on humanoid robots in its efforts to create further innovative forms of mobility. Such robots function usefully in society by coexisting and cooperating with people, which adds new value. Findings from analysis of how humans walk obtained in the course of research, designs for actuator mechanisms for bipedal walking, methods for controlling bipedal walking, and other such technologies were put to use in the world’s first autonomous bipedal walking humanoid robot, the P2, the prototype of which was publicly announced in 1996⁽¹⁾. In November 2000, Honda introduced ASIMO, which was made a size adaptable to the human living environment with the aim of making it a genuinely useful robot that could function in close proximity to people⁽²⁾. ASIMO was made available for rent on the Japanese market in 2001, and it was exhibited in 30 or more countries around the world in 2002. Honda has explored the interaction between people and robots in search of ways to make robots more useful in real-life situations, and has further equipped robots with human interactive functions, including the capability for recognizing people’s posture and actions as well as distinguishing among individual people’s faces so that robots could interact more smoothly with humans⁽³⁾. In December 2005, Honda presented an ASIMO with body functions significantly enhanced by highly accurate, highly responsive hardware and new posture control technology that makes active use

of bending and twisting of the upper body⁽⁴⁾. It was also equipped with technologies for autonomous, continuous movement as well as for obstacle avoidance and track generation⁽⁴⁾. In December 2007, ASIMO was given new functionalities for cooperative work, autonomous recharging, crossing paths without colliding, and obstacle avoidance, and operational testing in a real-life environment was begun. Through activities like these, Honda has sought to further its dream of a humanoid robot that functions together with human beings, to illuminate new frontiers for exploration, and to develop a wellspring of technological know-how.

This article presents an overview of the new ASIMO (Fig. 1) that was presented in November 2011.



Fig. 1 Humanoid robot “New ASIMO”

* Fundamental Technology Research Center

2. Development Goals

Figure 2 shows a development roadmap that leads toward the ultimate goal for the ASIMO, which is to achieve “a robot that coexists and cooperates with people and performs useful functions in society.” The new ASIMO is positioned at the second step, aiming for the capability to engage in continuing work in a real-life environment where it coexists with people, though in a limited manner.

The previous model of ASIMO has been employed to date in operation tests of use as a reception guide and in a delivery service in office lobbies and other real-life environments. It was found through this testing that there are many situations in a real-life environment where a robot is unable to recognize the presence of people, and that the people a robot interacts with often do not act as anticipated. When the previous model of ASIMO, which acted automatically in accordance with scenarios defined in advance, encountered these situations, it was unable to act appropriately, making it necessary for an operator to intervene.

The aim in developing the new ASIMO, therefore, was to achieve an autonomous machine with the capability for autonomous decision-making and behavior in changing circumstances. The elements that are necessary to make a robot an autonomous machine were defined as:

1. **External recognition:** This capability detects people's changing movements and other such actions around the robot using multiple sensors and integrates that data to infer the situation.
2. **Autonomous behavior generation:** This capability predicts the subsequent situation from collected data and autonomously makes decisions about its next behavior without the intervention of human operation.
3. **High-level postural balancing:** This is the robot's capability to maintain a stable posture by quickly extending a foot forward.

The development effort was then directed toward technology that would achieve these three elements.

This paper introduces the main specifications of the

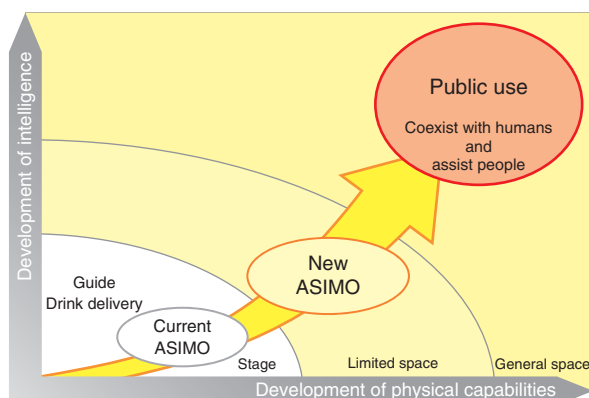


Fig. 2 ASIMO development roadmap

new ASIMO in Chapter 3. Chapter 4 then describes the system architecture that provides the foundation technology for enhancement of robot intelligence, which achieves the external recognition and autonomous behavior generation noted above. This architecture integrates information obtained from external observation of people and things by multiple sensors, determines what action to take with regard to whom, and takes the action. Next, Chapter 5 describes the technology for adapting to exterior disturbances such as differences in level, unevenness, and other such changes in road surface conditions, in terms of the high-level postural balancing noted above. There will also be some description of local path planning technology used to move without stopping in a mixed environment with people. Chapter 6 will introduce the many-axis hand that was developed in order to support the increase in kinds of work objects and the diversification of positions and postures.

3. Main Specifications

3.1. Basic Specifications

Table 1 shows the main specifications of the new ASIMO. Its total height is 1300 mm and its weight is 48 kg. A weight reduction of 6 kg was achieved without altering the robot's total height, which is the same as the previous ASIMO announced in December 2005. It was given a walking speed of 4.0 km/h, which is the same as normal human walking speed, in order not to hinder people's walking. Its running speed was raised significantly, to 9.0 km/h, and the robot was also given the ability to move in reverse. The exterior design continues to express a harmony of innovativeness and familiarity while embodying evolution in terms of gentleness, comprehensibility, and distinguishability for the purpose of engaging with people.

3.2. Joint Configuration

The new ASIMO has been given a total of 57 degrees of freedom (Fig. 3) with the addition of one axis in the waist and 11 axes in each hand. The pitch axis in the waist was added to expand the robot's range of work by including the ability to bend at the waist. The degrees of freedom in the hands were increased significantly by adding to the previous count of one axis in the thumbs and one axis in the four other fingers. The new configuration enables an optimal grasp of objects in accordance with their position and orientation, enabling the robot to grip objects of various shapes and pinch them with its fingertips, and enabling the manipulation of objects using a twisting motion.

The new ASIMO's range of movement in each axis has also been expanded compared to the previous model. A wide-angle six-link neck construction with a changing instantaneous center was adopted to avoid interference with the chest from up-and-down movement of the head. This significantly (by a factor of 1.7) expanded the angles of movement from the previous 22 degrees up and 19 degrees down to 26 degrees up and 32 degrees down. The ability to

Table 1 Specifications for ASIMO

Height	1300 mm	(self-standing)
Weight	48 kg	
Speed	0 – 4.0 km/h	(Walk)
	9.0 km/h	(Run straight)
	3.0 km/h	(Run backwards straight)
Sensors	Dual resolution stereo camera	(in head)
	8-ch microphone array	(in head)
	G/Gyro	(in body)
	Floor mark sensing camera	(in waist)
	Force sensors	(in legs, wrists, hands)
	Tactile sensors	(in hands)

rotate the foot in the lateral direction at the ankle (tilting the foot so the sole will remain flat on a floor that slants upward or downward from left to right) is important to the cross-step leg movement, which expands the range of landing points for the feet. By reducing the link-leaning angle, the range of this lateral rotation increased from the previous ± 24.5 degrees to ± 32.5 degrees.

3.3. Configuration of the Electrical System

High-response systems that support all of the robot's recognition, movement, and work functions were achieved for the new ASIMO.

One of the pair of stereo cameras previously mounted in the head was made into a multiple resolution camera that uses a prism. This achieved a balance between recognition of objects across a wide range and face detection at a distance. Moreover, where the previous model had microphones mounted at three locations on the front of the head, the new ASIMO has them at eight locations around the head to enable detection of sound from the total periphery. Ring-shaped vibrating gyroscopes, which are less influenced by external shock and vibration, were used for acceleration and angular velocity sensors. This achieved a 95% reduction in volume compared to the previous model, contributing to the increased compactness of the system.

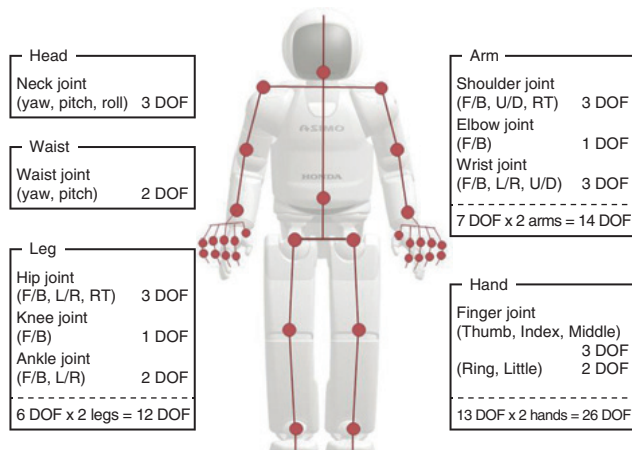


Fig. 3 Joint configuration and DOF (Degree of freedom)

In order to support the increased processing volume resulting from enhancement of the robot's functions, the central processing unit (CPU) parallelism was increased and a dedicated leg control CPU was added. A hierarchical memory structure using serial input and output was also adopted for the main CPU, achieving conflict-free memory access. The speed of the internal communications system linking the main CPU, sensors, and motor drivers was increased (by a factor of four relative to the previous model), as well, enabling synchronization of all 57 axes on a cycle of 250 μ s.

4. Autonomous Behavior Control

A new system architecture for autonomous behavior control was built with the aim of giving the new ASIMO a heightened ability to adapt to situations. Working on the assumption that the robot would be assigned to office reception, a system applying this architecture to human response functions (interaction at encounter, interaction by trigger, guiding), interactive explanation functions (presentation question and answer (Q&A)), and work functions (drink delivery) was first created (Fig. 4).

4.1. Sensing

In order for the robot to act in a manner appropriate to changing circumstances, it is necessary for the robot to have an accurate grasp of the circumstances around it. In addition to the sensors mounted on the previous robot body, sensors were placed in spaces for the new ASIMO in order to detect conditions in the total space where the robot operates, not just in its immediate vicinity.

4.1.1. Spatial sensing

Two types of sensor were used: Lateral laser range finders (LRF) that conduct one-dimensional scans were placed mainly on wall surfaces, and vertical LRF that conduct two-dimensional scans were placed on ceilings. These measure distances from the sensors to an object. The differences between those measured distances and the distances to the background measured in advance when there were no moving bodies present are obtained. The vertical LRF further help eliminate data that does not match with people's heights. The remaining data is subjected to clustering by the lateral LRF on the basis of adjacency of data and by the vertical LRF on the basis of distance between data. Those clusters that are determined not to be people due to the distances between data on either end of the clusters are eliminated from the clusters created by the lateral LRF. The centers of the clusters that remain after this processing are output as the locations of people.

4.1.2. Image recognition

Office reception areas are unlike an area under controlled illumination, such as a stage, because their lighting environment generally varies significantly with the

weather and the time. Such areas therefore call for image sensing that remains robust against varying illumination. A technology for automatically adjusting shutter speed in coordination with face recognition at the time of image acquisition was developed for the new ASIMO.

In cases when the robot is interacting with multiple people, it is not necessarily the case that the face of the person the robot should be interacting with will always be directly in front of the robot. A facial image recognition technology was therefore developed that would enable face detection, face direction estimation, and face identification regardless of the person's location in the camera's field of view. Instead of the previous method, which took the center of the image as the center for application of distortion correction to the entire image field, this technology uses a method of distortion correction known as virtual pan and tilt, which virtually directs the camera to the face location and places the face at the center (Fig. 5).

4.1.3. Audio recognition

Taking proxemics as a reference, the distance between the robot and people it converses with was set at 1-2 m for the new ASIMO. The aim was to recognize only utterances by people who are at this distance from the robot. Given that there is some breadth to the distance, and that, unlike demonstrations on stage, the people interacting with the robot will not all speak at the same volume level, the audio intensity input to the robot will vary considerably. In the conventional audio recognition method that is generally used, sounds that have multiple envisioned signal-to-noise (S/N) ratios are given as individual models to be learned, and the model that is considered optimal in terms of

distance and loudness of the voice is selected. The distance and loudness of voice can vary dynamically, however, which means that the optimal model cannot necessarily be chosen in every case, and there was therefore a risk that the recognition rate would decline.

In order to deal with these varying circumstances, a technology was developed for the new ASIMO that uses a single model to learn sounds over a wide range of S/N ratios. Compared to audio recognition that uses individual models for each distance, this method has achieved a favorable recognition rate using a single model for sounds at every distance.

In addition, acoustic feature, sound volume, and fundamental frequency are used to detect word lengthening within audio. This has achieved elimination of misrecognition caused by the filled pauses that occur frequently in speech.

4.2. Situation Estimation

Situation estimation uses spatial sensing information

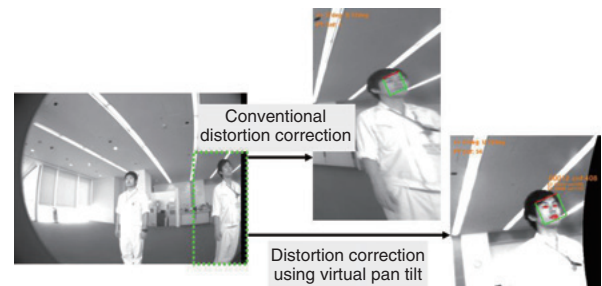


Fig. 5 Correction of image distortion

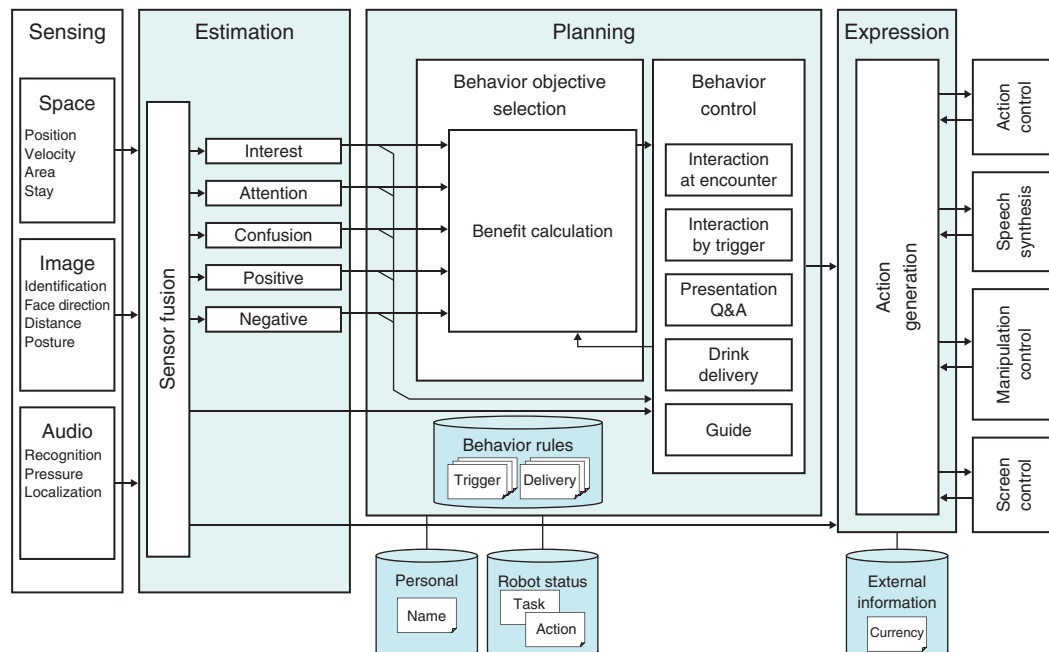


Fig. 4 Interaction system architecture

as a basis for estimating people's attributes that are needed in the robot's interactions with people. Here, people's attributes refers to the states of people, including stopped, coming closer to the robot, going farther away from the robot, facing the robot, arriving at the workplace, leaving, and waiting. This is information needed to determine the robot's course of action. The previous ASIMO could interact only with people who came close to it. By using this information, however, the new ASIMO has become able to approach people on its own.

The system uses a Bayesian network that takes the audio, image, and spatial sensing information as input and provides the situations of the people as output in the form of probabilities. Then it extrapolates where the people in the vicinity of the ASIMO are directing their attention and how much interest they are showing in the ASIMO, and it outputs this to the behavior generation unit, which is described below.

4.3. Behavior Generation

An example of the operation of behavior generation will be explained using Fig. 6. This scenario has five people designated A-E as candidates for interaction with the robot. The behavior objectives are of five types: interaction at encounter, interaction by trigger, acting as a guide, presentation Q&A, and drink delivery.

In behavior objective selection, the effectiveness of behavior is first calculated for each behavior objective according to the situations of people output from situation estimation and the robot's action history. There are combinations formed from multiple people and multiple behavior objectives. From among those, the robot selects the target person and behavior objective that yields the greatest benefit of behavior. In Fig. 6, the robot chooses to engage in interaction at encounter with person C. The robot's choice of an action with greater benefit of behavior enables the robot not to continue speaking to a person who

does not respond no matter how many times the robot speaks to him/her, and instead to change its action to work with a different person.

Next, in behavior control, the decision is made on specifically which action to take among those behavior objectives. The choice of action is made here on the basis of predefined action rules and under constraint conditions, which are that the same thing is not said twice and another action that uses the same part of the robot's body is not performed simultaneously.

The system allows parallel processing and interrupt processing to occur in multiple behavior controls, and when the robot is taking a certain action, it can, according to the circumstances of use of the robot's resources, continue that action or temporarily suspend it and undertake a separate action. Specifically, this means that when someone says "thank you" to the robot while the robot is placing a tray on a table during drink delivery, the robot now has the capability to respond, "you're welcome" at that time, and when a person asks the robot a question during the robot's explanation of a presentation, the robot now has the capability to answer the question at that time.

5. Mobility Functions

In a real-life environment, the positions of people and obstacles change dynamically. There may also be cases when people cannot readily predict the robot's movement, resulting in situations when the people's movements and the robot's intersect. The new ASIMO has been given significantly enhanced capability for detection of obstacles, including people, relative to the previous model, and new leg control technology has been developed to serve as the basis for smooth movement that will not confuse people.

5.1. Dynamic Route Guidance

The new ASIMO has been equipped with new local path

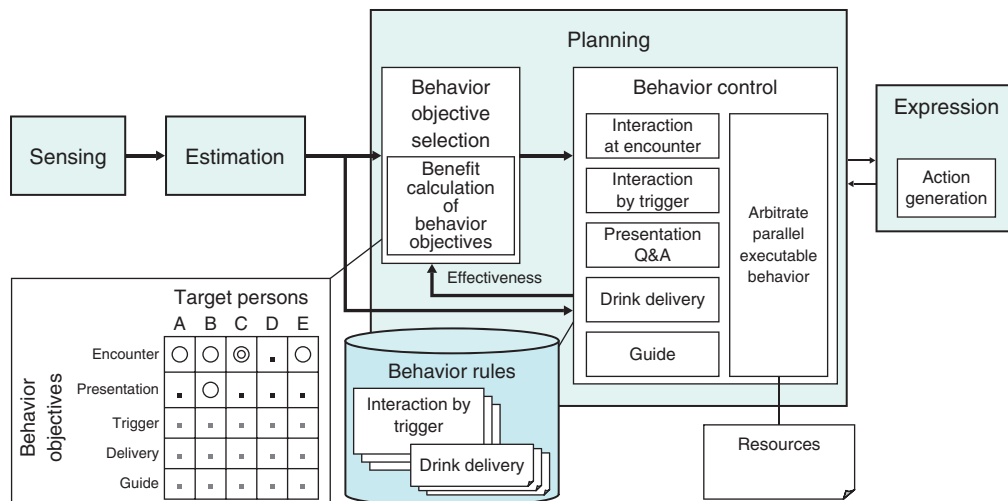


Fig. 6 Behavior control

planning technology and upgraded detection functionality for obstacles including persons. The purpose was to enable the robot to generate a path that allows it to accelerate, decelerate, change course, and so on without stopping, even in a location where multiple robots and multiple people are present.

5.1.1. Person and obstacle detection by spatial sensors

Detection was previously conducted by sensors on the body of the robot, so that only people and obstacles that were at close proximity and to the front could be detected. Since courses were generated on the basis of information from such an extremely narrow range, the robot sometimes moved jerkily or it would move toward a person behind it because it could not detect a person in that location. For the new ASIMO, therefore, new spatial sensing was introduced in order to detect the approach of people from the rear and the movement of people in blind spots, not just in the direction the robot is moving. Since the robot is able to detect people at an earlier stage, a smoother walking path can be generated for it.

5.1.2. Local path planning

The robot's path was previously decided as a matter of keeping the predicted locations of people and the path of the robot from overlapping on a two-dimensional plane. When a person was walking on a path that would cut across the robot's path, therefore, as in Fig. 7 (left), the robot would be unable to generate a course that avoided the person. Also, as explained in the previous section, the new ASIMO detected the movements of people in the entire space, so that when multiple people were present in the vicinity of the robot, the further increased likelihood that a path could not be generated would be anticipated. For the new ASIMO, therefore, a technology was developed for seeking paths whereby the person's predicted location and the robot's location would not overlap within a three-dimensional space that included time, as shown in Fig. 7 (right).

By means of this technology, the robot became able to generate a path of movement such that the robot could pass the person's path ahead of the person, or slacken its speed to allow the person to pass in front of it. The robot was thus enabled to continue walking without pointless stopping even if circumstances changed.

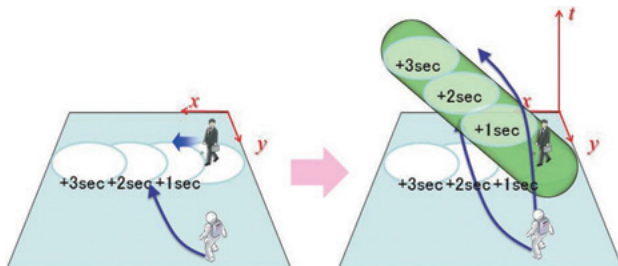


Fig. 7 Local path planning

5.2. Leg System

The previous model of ASIMO was limited in its ability to deal with changes in circumstances for various reasons. For instance, it could not generate more than one type of gait at a time, it could not adjust its landing point while it was moving, and it could not exert restoring force with regard to inclination errors due to external disturbance except within the area covered by its feet.

The new leg system configuration shown in Fig. 8 was adopted for the new ASIMO in order to overcome these issues. This system conducts gait generation in parallel and in real time so that the optimal gait pattern can be selected at any time from among walking, running, and hopscotch movements. This has also yielded significant enhancements in the robot's ability to exert restoring force when inclination error occurs not only by staying on its foot, but also by changing its landing point in real time by means of the horizontal acceleration of its upper body and by bending at the hip.

5.2.1. Expansion of landing points

It is important that the robot be able at any time to change the landing point where it places its foot and the timing of that action in order to reinforce its capability for dealing with external disturbances. Figure 9 shows the range within which the foot can be placed on the next step when the left leg is the supporting leg. The range of possible landing points for the previous ASIMO is the red area in the

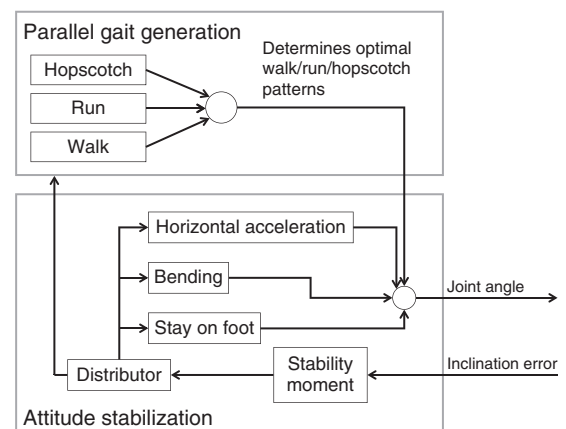


Fig. 8 Leg control system

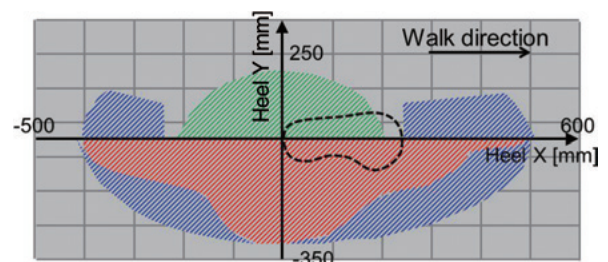


Fig. 9 Extended foot landing position

figure. The foot and leg cannot be extended in the direction of the supporting leg (positive direction on the Y axis), and there was therefore a risk that the robot would fall over if it were subjected to an external disturbance on the supporting-leg side. The new ASIMO has been given the capability to move laterally toward the supporting leg by means of the cross-step leg movement and hopscotch movement. The blue areas show the idling-leg landing area, which has been expanded by the new ASIMO's cross-step leg movement. The green area shows the landing area for the supporting leg from the hopscotch movement. The cross-step leg movement and hopscotch movement have added a remarkable degree of stability when the robot is subjected to external disturbance in the direction of the supporting leg.

5.2.2. Real-time gait generation

Even the previous ASIMO had the conceptual approach of overall stability, which seeks to stabilize the robot against inclination error by bending at the hip or accelerating the upper body. Since it did not have lateral movement or real-time gait generation, however, its capability for stabilization was low and the circumstances under which the capability could be used were limited.

Figure 10 shows the landing point determination method used when exerting restoring force against inclination error. When the zero moment point (ZMP), calculated from the moment of the restoring force required when trying to land the foot at the desired landing point in the diagram, goes beyond the foot, the sensitivity of the landing point and ZMP compensation is calculated and the ZMP landing point is decided analytically.

The new ASIMO has achieved overall stability that can be used on uneven road surfaces and other such real-life environments. It was achieved by this real-time gait generation technology as well as by the expansion of the range within which the foot can be placed by means of the cross-step movement and hopscotch movement, as described in the previous section.

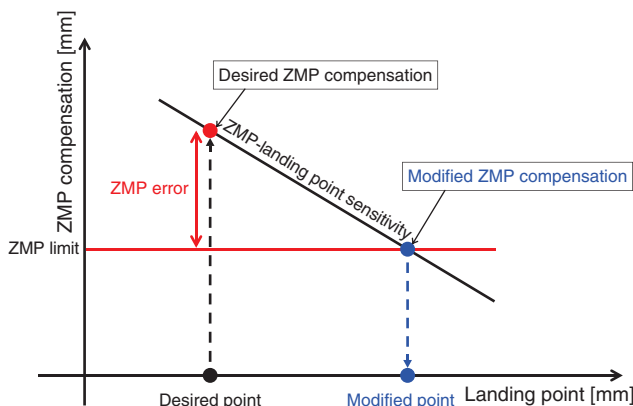


Fig. 10 ZMP landing point compensation

6. Compact, Hydraulic Many-Axis Hand

The previous ASIMO's hands were grippers with two degrees of freedom that did not even have sensors. Consequently, they were only capable of gripping a limited range of objects in a fixed manner. In working toward achievement of work that is in the interest of coexisting with people and performing useful functions, it becomes necessary to increase the types of object and to enable manipulation even when object positions and postures become more diverse. With the new ASIMO, therefore, the number of degrees of freedom in the hand was increased significantly without undercutting its design quality. Control for a stable grip using sensor information and degrees of freedom was also developed.

6.1. Hardware

The enhanced compactness of every functional component in the new ASIMO achieved multiple degrees of freedom in the ASIMO-sized hands. A mechanism was adopted that uses hydraulic pressure to transmit driving force from the motors, which support each axis on a one-to-one basis⁽⁵⁾. The hydraulic pressure master cylinders for all 26 axes were made more compact by means of a newly developed motor integrated with a ball screw, a 13-axis motor driver, and the integration of multiple axes in one. The master cylinders are mounted inside the robot torso, as shown in Fig. 11. Each finger uses low-expansion, small-diameter, flexible mesh-reinforced tubing to provide full responsiveness under the increase in pressure that occurs when the crank is made more compact.

Newly developed, compact six-axis force sensors were mounted at the fingertips of all five fingers on each hand. Tactile sensors were also placed under the skin on the palms of the hands. These together with the six-axis force sensors on the wrist provide 96 channels of signals. A compact electronic control unit that can capture these signals on a cycle of 250 μ s and be mounted in the hand was also developed.

Measures were also taken to enable diverse manipulations of work objects. The angular mobility of the wrists was increased 90% in the vertical direction and 60% in a twisting direction compared to the previous model, significantly increasing the degree of postural freedom of

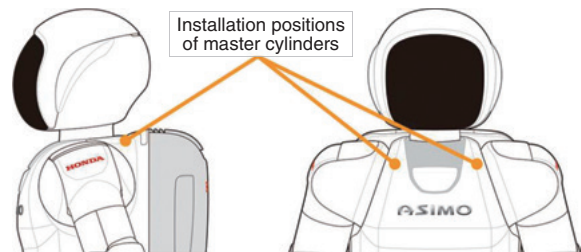


Fig. 11 Installation positions of hydraulic pressure master cylinders

the hands.

Application of these technologies achieved a compact hand weighing 0.38 kg and capable of gripping 1.8 kg, an increase of 360% over the previous capability.

6.2. Stable Grip and Shift of Grip

In order to secure a stable grip on an object, it is necessary to distribute finger forces properly to every finger so that the object will not slip. Determination of the internal forces for the new ASIMO required development of a method for formulating the grip conditions in terms of force distribution such that the contact point on the finger side and the contact point on the object side have a relative velocity of zero. This method remains effective even if the number of contact points is increased.

Figure 12 shows the relationships of forces when multiple fingers grasp an object. The blue in the figure signifies the designed finger forces, the red signifies the combined force of those forces, and the black signifies gravity. The internal forces are distributed so that the grip on the object will be stable even if the number of gripping fingers changes. In the horizontal direction, the forces of the fingers cancel each other out, and in the vertical direction, the combined forces balance out with the force of gravity. A divergence between the designed grip point and the actual grip point occurs when the object is actually gripped, so that the anticipated center of gravity and the actual center of gravity diverge. In order to extrapolate this divergence and compensate for it, the model of the object is corrected using a center of gravity error that is obtained as the difference between the designed finger force value and the actual value. Gripping force transition control that determines the gripping force by continuously changing the constraining conditions was also incorporated.

The hardware described in the previous section together with these items of control technology enabled the robot to perform the task of using one hand to pick up a drinking flask from a tabletop, unscrewing the cap, and pouring a drink from the flask into a cup that was picked up with the other hand.

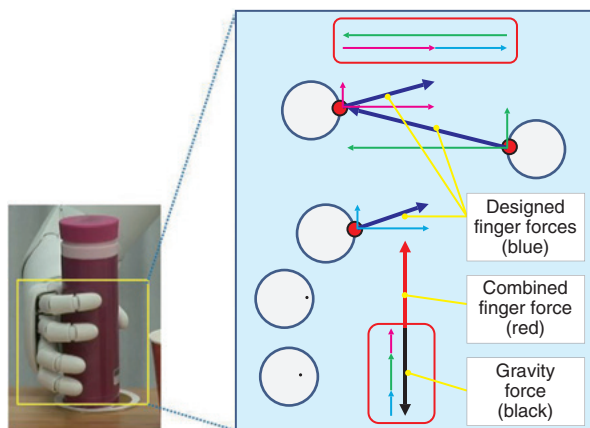


Fig. 12 Multi-finger grasping

6.3. Sign Language

The aim for the ASIMO is to achieve a high level of expressiveness that communicates readily to people using the robot's body and taking advantage of its humanoid characteristics. Sign language was selected as the means for expression by the new ASIMO, which was equipped with the actions to express greetings, self-introduction, and comments on the weather, which are frequently used in everyday life (Fig. 13). Sign language is a means of communication that functions through bodily expressiveness, and it includes large numbers of effective mechanisms, schemes, conventions, and other such resources that can be applied to bodily expression technology for robots.

The sign language words and messages that were created were then tested on a native user of sign language. This confirmed that fully adequate expression was taking place.

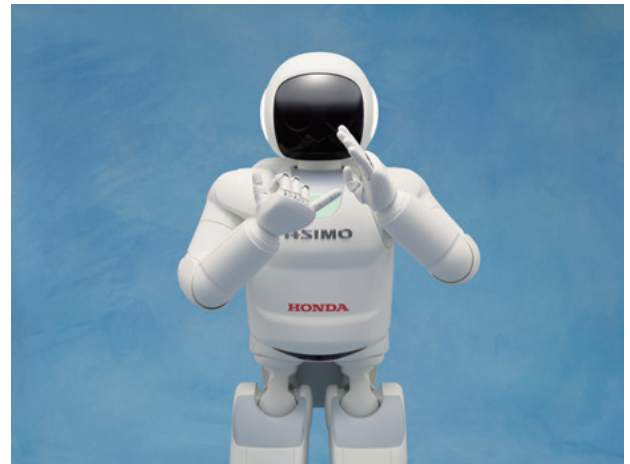


Fig. 13 Sign language "family"

7. Conclusion

The new ASIMO was developed with the capability for continued working in a real-life environment where it coexists with people. This was a step toward achieving "a robot that coexists and cooperates with people and performs useful functions in society." The new ASIMO represents an evolution from an automatic machine that performs fixed actions to an autonomous machine that matches its actions to changes in surrounding circumstances. This resulted in a system built with an architecture that constitutes the foundation technology for enhancement of robot intelligence, and it established the fundamental technology for engaging in appropriate action toward people in changing circumstances. In terms of bodily abilities, the development of lightweight mechanisms with a wide range of movement, high-response electrical systems, and new control technologies achieved an agility capable of dealing

with changing circumstances.

Our intention at Honda is to make use of operational trials in real-life environments to further refine the robot's capability for adapting to circumstances within environments where it coexists with humans, and to expand the locations in which the robot can operate.

References

- (1) Hirai, K., Hirose, M., Haikawa, Y., Takenaka, T.: The Development of Honda Humanoid Robot, IEEE International Conference on Robotics and Automation (1998)
- (2) Hirose, M., Takenaka, T.: Development of Humanoid Robot ASIMO, Honda R&D Technical Review, Vol. 13, No.1, p. 1-6
- (3) Higaki, N., Yokoyama, T., Fujimura, K.: Gesture Recognition Algorithm for Understanding of Man-Machine Interaction and Its Application for New ASIMO, Honda R&D Technical Review, Vol. 15, No. 1, p. 95-102
- (4) Shigemi, S., Kawaguchi, Y., Yoshiike, T., Kawabe, K., Ogawa, N.: Development of New ASIMO, Honda R&D Technical Review, Vol. 18, No. 1, p. 38-44
- (5) Takahashi, K., Waita, H., Kokusho, M., Hayakawa, M.: Development of Dexterous and Powerful Multi-fingered Hand for Humanoid Robots, Proceedings of the 27th Annual Conference of the Robotics Society of Japan, 1A1-01 (2009)

■ Author ■



Satoshi SHIGEMI



Koji KAWABE



Takahiro NAKAMURA

Fingertip-size Six-axis Force and Moment Sensor Using Semiconductor Piezoresistance Effect

Takeshi OHSATO*
Shigenori YASUIE*

Nobuhiro SAKURAI*
Yusuke HIRABAYASHI*

Yasuhiko JINBU*
Hiroshi YOKOBAYASHI*

ABSTRACT

A compact six-axis force and moment sensor that can be mounted in a fingertip is proposed to enhance the holding function of robots.

Specifically, the sensor system uses the semiconductor piezoresistance effect to detect the six-axis force and moment with a silicon chip (hereafter, “chip”). In addition, a resistive element for temperature compensation is fabricated into the chip to stabilize the temperature characteristics. The structure combines the chip with a metallic stress attenuation mechanism, enabling prevention of the load applied to the sensor from damaging the chip. Accurate sensor characteristics are obtained by integrating the components using anodic bonding and precision integration.

A fingertip-size prototype sensor based on the above concepts realized a rated load of 30 N (moment: 30 Ncm). The output accuracy at the rated load was confirmed to be 2.4 %FS (FS = rated equivalent output), and the temperature characteristics accuracy in the zero load state was confirmed to be 0.19 %FS/°C (0 to 60°C).

1. Introduction

As shown in Fig. 1, six-axis force and moment refers to the forces (F_x , F_y , F_z) and moments (M_x , M_y , M_z) in the three axial directions. It is mainly used for industrial machine applications, and various six-axis force and moment sensor products have been commercialized such as strain gauge, electrostatic capacity, and optical types. All types require precise location of the detecting elements to obtain accurate sensor characteristics, which requires a high level of manufacturing ability, and the manufacturing issues increase further as the size decreases. For example, in case of a strain gauge-type sensor, several to several ten metallic strain gauges need to be affixed to the structure

(flexure element) to which the load is transmitted, and when reducing the size to fingertip size, some physical layouts may not be feasible depending on the number and dimensions of the strain gauges. In consideration of these circumstances, a six-axis force and moment sensor using the semiconductor piezoresistance effect was proposed to help reduce the size to fingertip size.

2. Basic Concepts

Points (1) to (3) below list the basic concepts of the developed sensor.

- (1) The six-axis force and moment detection system using the semiconductor piezoresistance effect⁽¹⁾ enables detailed and accurate placement of compact detecting elements that are a challenge to manufacture using conventional technology. In addition, a resistive element for temperature compensation is fabricated into the chip to stabilize the temperature characteristics.
- (2) The structure combines the chip with a metallic stress attenuation mechanism, enabling prevention of the load applied to the sensor from damaging the chip.
- (3) Accurate sensor characteristics are obtained by integrating the components using anodic bonding⁽²⁾ and precision integration.

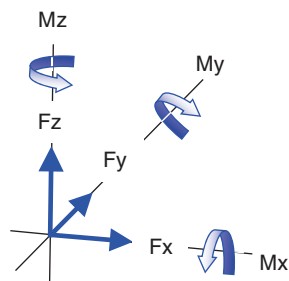


Fig. 1 Six-axis force and moment

* Fundamental Technology Research Center

3. Sensor Structure

The sensor structure is described below.

Figure 2(a) shows the external appearance of the sensor. The sensor has a cylindrical shape with a diameter of 13 mm, and grooves machined into the metallic body function as a stress attenuation mechanism. Figure 2(b) shows the internal structure. The chip and the glass are joined by anodic bonding, and the glass and the metallic body are also joined by anodic bonding. In addition, the dotted line in the figure shows an image of the chip deformation when force is applied in the F_z direction. This figure omits the metallic body that corresponds to the bottom half of the sensor and functions as the housing. The shape of this bottom body closely resembles the body for the top half of the sensor, and the structure is integrated by precision integration.

Figure 3 shows an image of chip deformation when a load (F_x in this example) is transmitted to the chip. T-shaped beams are formed on the chip using the MEMS process^{(3), (4)}, and these beams deform to transmit strain to the piezoresistive elements. An additional piezoresistive element is also fabricated in a location other than the T-shaped beams. Strain is not transmitted to this location, so this is used as a resistive element for temperature compensation.

The dimensions and shapes of the sensor components were determined according to knowledge of strength of materials and structural analysis using the finite element

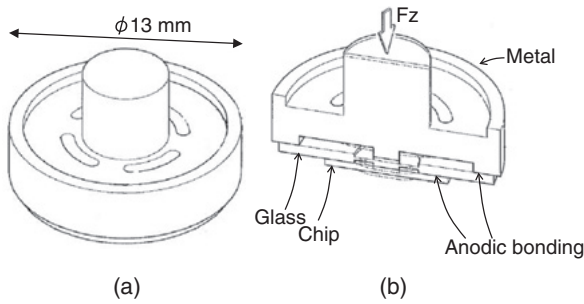


Fig. 2 Oblique perspective view of sensor

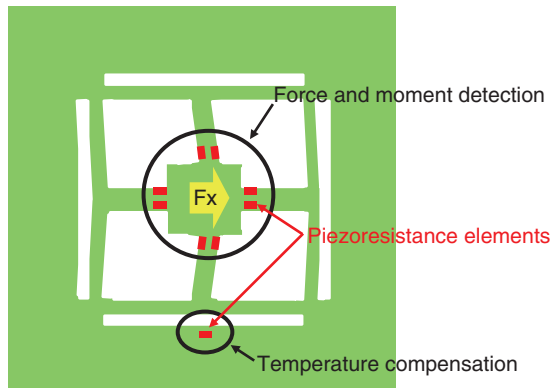


Fig. 3 Image of chip deformation

method, with the design goals of fingertip size ($\phi 13$ mm) and the rated load (force: 30 N, moment: 30 Ncm).

4. Six-axis Force and Moment Calculation Method

The chip of this sensor has eight piezoresistive elements fabricated into the T-shaped beams, and strain generated according to the load can be detected by these piezoresistive elements. The detected strain is converted to the six-axis force and moment as follows. First, a known load was applied to each axis separately, the change in resistance of the eight elements (R1 to R8) was measured, and the components of the matrix $[A]$ in Eq. (1) were obtained beforehand. Then, when an unknown load is input, calculations are performed using the inverse matrix $[A]^{-1}$ shown in Eq. (2).

$$\begin{bmatrix} R1 \\ R2 \\ R3 \\ R4 \\ R5 \\ R6 \\ R7 \\ R8 \end{bmatrix} = \begin{bmatrix} A_{11} & & & & & & & A_{81} \\ & & & & & & & \\ & & & & & & & \\ & & & & & & & \\ & & & & & & & \\ & & & & & & & \\ & & & & & & & \\ A_{81} & & & & & & & A_{86} \end{bmatrix} \begin{bmatrix} Fx \\ Fy \\ Fz \\ Mx \\ My \\ Mz \end{bmatrix} \quad (1)$$

$$[F, M] = [A]^{-1} [R] \quad (2)$$

In addition, temperature compensation can also be performed using the calculation method shown in Fig. 4. Specifically, the output when the temperature is varied between 0 and 60°C in the zero load state is measured beforehand to obtain the correlation with the temperature compensation resistor. Temperature compensation can then be performed in an environment of unknown temperature by subtracting the load detection resistor output obtained from the correlation with the temperature compensation resistor output.

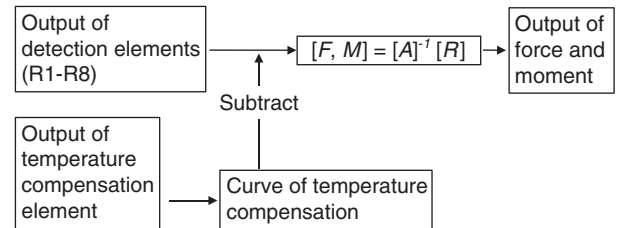


Fig. 4 Block diagram of force and moment detection

5. Evaluation of Characteristics Using a Prototype

5.1. Load Application Characteristics

A prototype sensor was made and the output accuracy was evaluated by applying loads in a constant room

temperature environment. This paper shows an excerpt of the results when applying F_x and M_x .

Figure 5 shows the output results when force equivalent to the rated load ($F_x: \pm 30$ N) is applied. As shown in this graph, it was confirmed that only F_x is detected, and other loads are almost not detected. Figure 6 shows the offset (error) from the line obtained by linear approximation using the least-squares method, for the output characteristics shown in Fig. 5. The M_z output exhibited the largest offset of 0.4 Ncm relative to the F_x load of 30 N, so the output accuracy for F_x application was confirmed to be 1.4 %FS (FS = rated equivalent output).

Figures 7 and 8 show the output data for the moment ($M_x: \pm 30$ Ncm). Here, the F_z output exhibited the largest offset of 0.7 N, so the output accuracy for M_x application was confirmed to be 2.4 %FS.

The output accuracy when the other loads (F_y , F_z , M_y , M_z) not shown in this paper are applied was confirmed

to be higher than the results for M_x application described above. Therefore, the output accuracy of this sensor at the rated load is thought to be 2.4 %FS.

5.2. Temperature Characteristics

The output accuracy when the temperature is varied between 0 and 60°C was evaluated with the sensor in the zero load state. Figure 9 shows the relationship between the load detection resistors (R1 to R8) and the temperature compensation resistor. Note that the change in resistance is measured as a voltage, so the graph shows the voltage output. This relationship was substituted into the calculation method shown in Fig. 4 to evaluate the accuracy after temperature compensation.

Figure 10 shows the results after temperature compensation of the sensor output when the temperature is varied between 0 and 60°C. An offset of 3.4 N occurred in the F_z output, which is equivalent to 11.3% of the rated load (30 N). Therefore, the temperature characteristics accuracy of the developed sensor within the measurement temperature range is thought to be 0.19 %FS/°C.

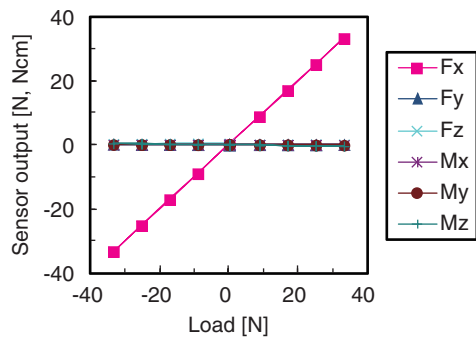


Fig. 5 Characteristic of load impression (F_x)

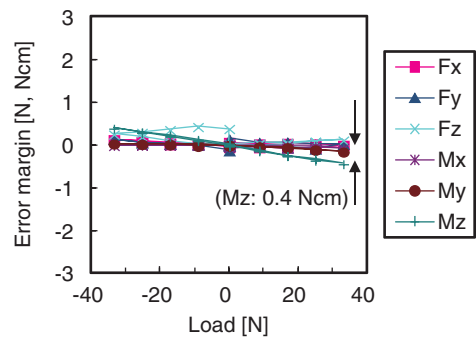


Fig. 6 Detection error margin of load impression

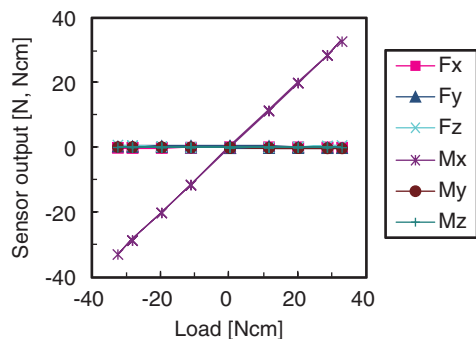


Fig. 7 Characteristic of load impression (M_x)

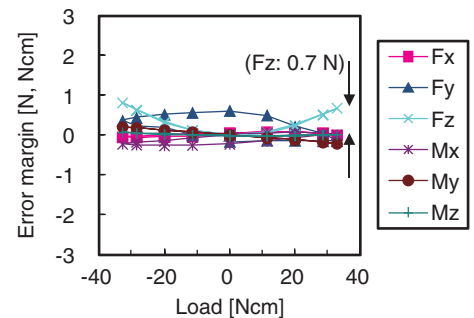


Fig. 8 Detection error margin of load impression

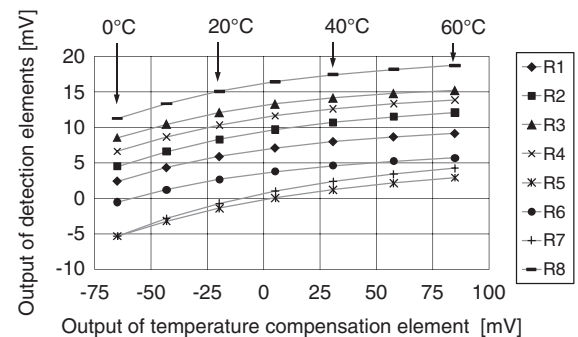


Fig. 9 Curve of temperature compensation

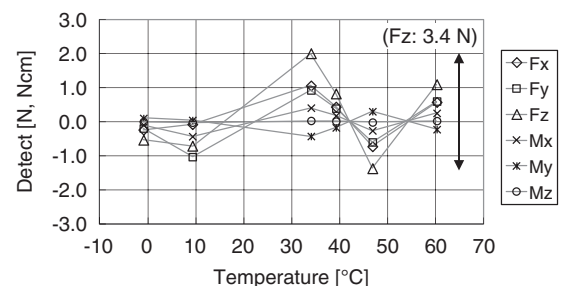


Fig. 10 Sensor output (after temperature compensation)

Table 1 Sensor characteristic results

Diameter		13 mm
Load range	Force	30 N
	Moment	30 Ncm
Accuracy	Load	2.4 %FS
	Unload 0 – 60 °C	0.19 %FS/°C

6. Conclusion

Table 1 shows the characteristics of the prototype sensor. This shows that a fingertip-size sensor capable of providing accurate output was realized.

A compact six-axis force and moment sensor is needed to enhance the holding function of a robot's fingertips, and the basic sensor concept proposed in this paper is thought to be an effective means of achieving this goal.

Furthermore, this sensor uses semiconductor technology, so in addition to a compact size and high accuracy, the power consumption can be reduced by increasing the resistance of the piezoresistive elements. It is expected that mass-productivity can also be increased by manufacture using a semiconductor process that does not require special skills.

References

- (1) Kanda, Y.: A Graphical Representation of the Piezo-resistance Coefficients in Silicon, IEEE Transactions on Electron Devices, Vol. ED-29, No. 1 p. 64-70 (1982)
- (2) Wallis, G., Pomerantz, D. I.: Field assisted Glass-Metal Sealing, Journal of Applied Physics, 49, p. 3946-3949 (1969)
- (3) Hynes, A. M., Ashraf, H., Bhardwaj, J. K., Hopkins, J., Johnston, I., Shepherd, J. N.: Recent advances in silicon etching for MEMS using the ASE™ process, Sensors and Actuators A: Physical Volume: 74, p. 13-17 (1999)
- (4) Laermer, F., Urban, A.: Milestones in Deep Reactive Ion Etching, Digests of Technical Papers, Transducers' 05, p. 1118-1121 (2005)

■ Author ■



Takeshi OHSATO



Nobuhiro SAKURAI



Yasuhiko JINBU



Shigenori YASUIE



Yusuke HIRABAYASHI



Hiroshi YOKOBAYASHI

Research on Silent Chain Elongation Mechanism and Prediction

Masao SEKI*

Motoki TANAKA**

ABSTRACT

The silent chain elongation mechanism due to soot in the engine oil was analyzed and the relationship between chain elongation factors and the chain elongation rate was obtained through experiments in order to construct technology for predicting silent chain elongation.

The results showed that almost no silent chain elongation occurred when there was no soot in the engine oil. However, when soot is present on the pin and plate sliding surfaces, three-body abrasive wear produces plate wear, which increases chain elongation. In addition, experimentally deriving a silent chain elongation predictive formula from the relationship between chain elongation factors and the chain elongation rate enabled estimation of silent chain elongation from the initial design stages.

1. Introduction

Chain-driven systems are more durable and compact than belt-driven systems, so chain-driven systems are often used in the cam-driven systems of engines. In addition, silent chains have come to be used as they are quieter than roller chains. When using a chain-driven system, chain elongation influences the valve timing, chain behavior and other factors. Therefore, design should take into account chain elongation in the actual operating environment to help assure engine reliability and product appeal. However, various factors are associated with chain elongation, which makes it a challenge to predict chain elongation in the actual operating environment from the initial design stages.

A number of studies have been reported on simulation of chain tension, noise, friction and other factors in chain-driven systems⁽¹⁾⁻⁽³⁾. However, the chain elongation mechanism has not yet been clarified, so prediction using simulations is a challenge, and there are few reports of technology for predicting chain elongation.

Silent chain elongation occurs when the plate holes and pins become worn due to sliding between the pins and plates when the chain meshes with the sprockets and bends. Therefore, the engine operating status, oil supply status and other factors when the chain bends must be taken into account to predict chain elongation.

On the other hand, soot in the engine oil is known to

influence the wear of engine sliding parts. Pin and plate sliding surface wear is also a potential cause of silent chain elongation, so soot in the engine oil is thought to influence silent chain elongation. This means that the chain elongation mechanism due to soot in the engine oil must be clarified to predict silent chain elongation.

Therefore, this study used experimental results to clarify the chain elongation mechanism due to soot in the engine oil. In addition, a chain elongation predictive formula was experimentally derived by analyzing the relationship between chain elongation factors and the chain elongation rate. This enabled prediction of silent chain elongation.

2. Silent Chain Elongation Mechanism

2.1. Elongation Due to Soot in the Engine Oil

Engine bench tests were performed to confirm the effect of the soot concentration in the engine oil on silent chain (hereafter, “chain”) elongation. In these tests, carbon black was mixed into the engine oil in place of soot, and the engine was operated at the constant speed of 5000 rpm with different carbon black concentrations. Table 1 shows the specifications of the engine used in the tests and the test conditions. In addition, Fig. 1 shows the chain composition, Fig. 2 shows the chain elongation measurement results, and Fig. 3 shows the pin and plate wear amounts. Figure 2 shows that almost no chain elongation occurred under

* Automobile R&D Center

** Daido Kogyo Co., Ltd.

Table 1 Engine specifications and test conditions

Cylinder configuration	Inline 4
Bore × Stroke (mm)	87.0 × 99.0
Displacement (cm ³)	2354
Compression ratio	10.5
Chain type	Silent chain
Chain pitch (mm)	6.35
Number of link plates	4/3
Number of links	176
Engine speed (rpm)	5000
Engine oil	ILSAC GF-4, 5W-20
Soot concentration (wt%)	0, 0.4, 1.0

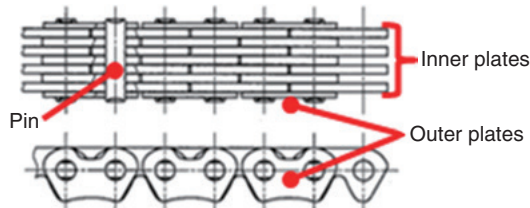


Fig. 1 Composition of silent chain

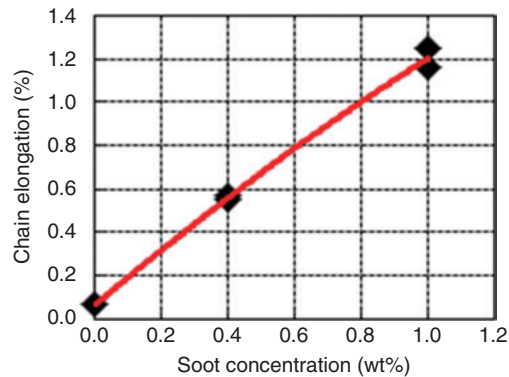


Fig. 2 Influence of soot concentration on chain elongation

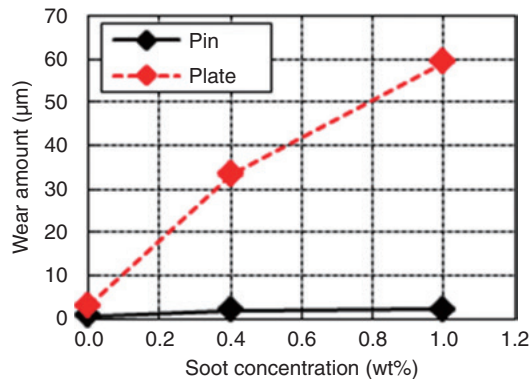


Fig. 3 Wear amount of pin and plate

Table 2 Chain specifications and test conditions

Chain type	Silent chain
Chain pitch (mm)	6.35
Number of link plates	4/3
Number of links	102
Load (N)	1500
Oil supply (l/min)	0.1, 1.0
Soot concentration (wt%)	0, 1.0

conditions of a carbon black concentration of 0 wt%. In addition, within the concentration range confirmed in these tests, chain elongation increased together with the concentration. These results indicate that chain elongation occurs when soot is present in the engine oil, and that the chain elongation rate is influenced by the soot concentration in the engine oil.

Next, Fig. 3 shows that the pin wear amount did not change much and there was also little wear as the carbon black concentration was increased. In contrast, the plate wear amount increased together with the carbon black concentration. The chain pins are surface-treated to a hardness equal to or greater than that of soot, but the plate hardness is lower than that of soot. Abrasive wear is known to generally be inversely proportional to hardness. These test results show that almost no chain elongation occurs when there is no soot in the engine oil, only the low-hardness plates become worn when soot is present in the engine oil, and only the plate wear amount increases together with the soot concentration. In consideration of these results, it is thought that chain elongation occurs due to the presence of soot between the pins and plates, and that the main factor of this plate wear is three-body abrasive wear.

2.2. Influence of Different Oil Supply States on Chain Elongation

As described in Section 2.1, plate wear due to three-body abrasive wear is thought to be the main factor increasing chain elongation. However, when soot is present in the engine oil, increasing the oil supply rate increases the supply of soot that is a potential cause of three-body abrasive wear, so supplying oil is thought to increase the wear amount. Therefore, wear tests were performed with different oil supply rates and carbon black concentrations in the engine oil to confirm the influence of the oil supply status to the chain pin and plate sliding surfaces on chain elongation. The tests used a discrete wear tester in order to stabilize the oil supply conditions. Table 2 shows the specifications of the chain used in the tests and the test conditions. In addition, Fig. 4 and Fig. 5 show the pin and plate wear amounts. Figure 4 shows that increasing the oil supply rate reduced the pin wear amount for both carbon black concentration conditions of 0 wt% and 1.0 wt%. This

is thought to be because adhesive wear between the pins and plates is the dominant mode for pin wear, so increasing the oil supply rate helped prevent the oil film between the pins and plates from breaking, which reduced wear. In addition, under conditions of a carbon black concentration of 1.0 wt%, increasing the oil supply rate reduced the pin wear but increased the plate wear. This is thought to be because increasing the oil supply rate increased the amount of carbon black present between the pins and plates, so plate wear was accelerated by three-body abrasive wear.

Next, Fig. 5 shows that increasing the carbon black concentration increased the pin and plate wear under both oil supply rate conditions of 0.1 l/min and 1.0 l/min. The increase in plate wear is as described previously, but the pin wear also increased together with the increase in the carbon black concentration. This is thought to be because carbon black accumulated near the pin and plate sliding surfaces, which obstructed oil supply to the sliding surfaces and accelerated abrasive wear.

These results confirmed that oil supply is necessary to suppress pin surface wear in order to reduce chain elongation, but oil supply may also increase plate wear under conditions where soot is present in the engine oil. Therefore, excessive oil supply may accelerate chain elongation.

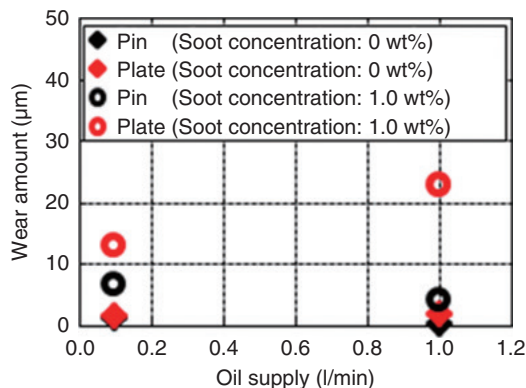


Fig. 4 Relation between wear amount and oil supply

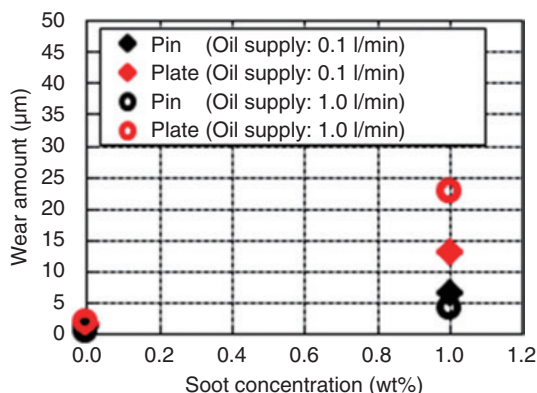


Fig. 5 Relation between wear amount and soot concentration

3. Chain Elongation Prediction Technology

3.1. Chain Elongation Predictive Formula

It was confirmed that the soot concentration in the engine oil is a factor increasing chain elongation. However, to predict the chain elongation rate, it is necessary to understand the influence of factors such as the contact surface pressure and velocity of the pin and plate sliding surfaces. Therefore, the chain elongation under different engine operating conditions was confirmed in engine bench tests to perform factor analysis. Figure 6 shows the test results arranged according to the condition of equal chain bending times. The bending and sliding distance results in Fig. 6 were obtained through tests with other factors held constant, so the trend can be easily understood. However, other factors mutually influence each other even when arranged according to the relationship between the respective chain elongation factor and the chain elongation rate. This makes it a challenge to understand the trends in the relationships between the chain elongation rate and factors such as contact surface pressure and velocity, and the influence of separate factors cannot be derived.

Therefore, correction was performed using the regression curves of other graphs in Fig. 6 so that factors other than the factor shown on the horizontal axis of each graph were constant conditions, and iterative calculations were performed until the slopes of all chain elongation factors converged. This enabled separate derivation of the influence of each factor on chain elongation. Figure 7 shows these calculation results. Whereas the results in Fig. 6 before the iterative calculations do not show a clear correlation between chain elongation factors and the chain elongation rate, the results in Fig. 7 after the iterative calculations show a sufficient correlation. Figure 7 also shows that chain elongation increases together with the pin and plate contact surface pressure and the sliding distance. In addition, chain elongation tends to increase as the sliding velocity decreases. Next, regarding the plate material characteristics, chain elongation decreases as the ratio of shear surface at the plate hole (the percentage of the plate hole machined surface accounted for by the shear surface) increases. In addition, chain elongation tends to decrease as the plate hole hardness increases, up to a certain hardness. This is thought to be because three-body abrasive wear due to soot is the dominant mode of plate wear, so the plates become resistant to wear as the hardness increases. However, these test results showed that chain elongation increases together with the hardness above a certain hardness. This is presumed to be due to the following reasons. A preload is applied to the chain to adjust the length and increase the fatigue strength. The chain plate holes deform when this preload is applied, and as the plate hardness decreases, the deformation amount increases, reducing the contact surface pressure between the pins and plates. The results in Fig. 7 also show that chain elongation tends to increase together with the contact surface pressure, and the phenomenon where chain

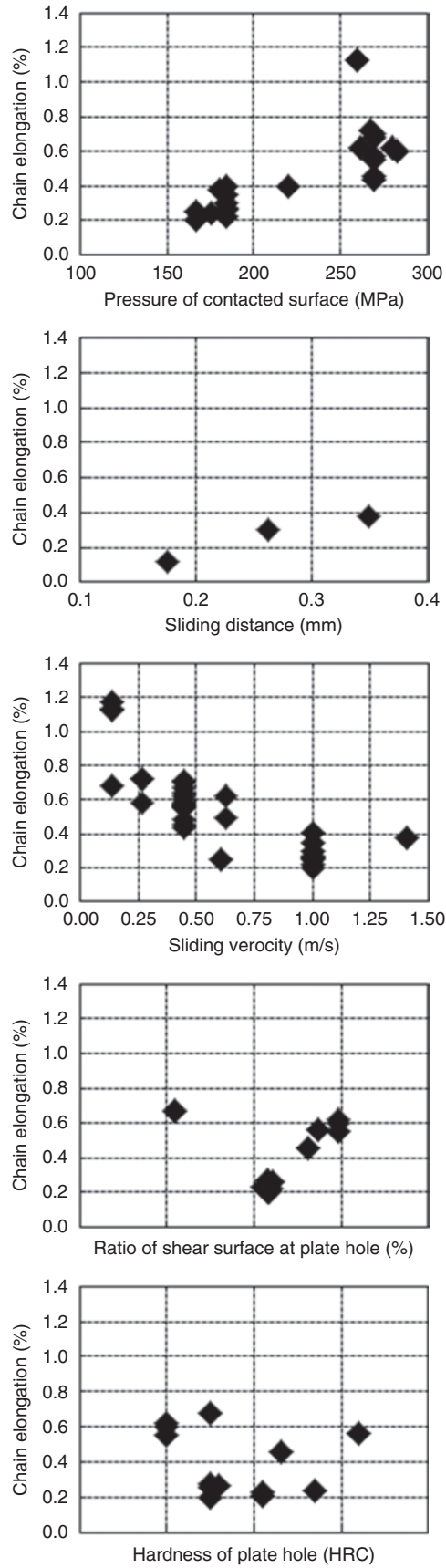


Fig. 6 Test results of relation between chain elongation and chain elongation factor

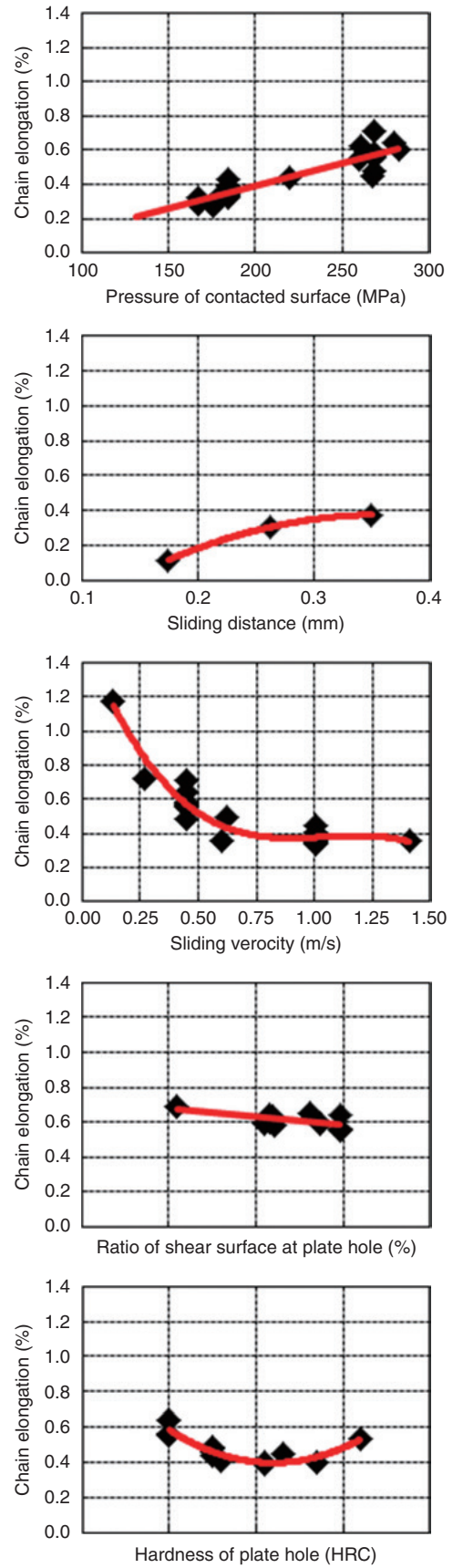


Fig. 7 Relation between chain elongation and chain elongation factor

elongation increases together with the hardness is thought to result from increased contact surface pressure due to a smaller deformation amount when the preload is applied. These results show that increasing the chain plate hardness effectively suppresses chain elongation, and reducing the hardness effectively lowers the contact surface pressure. Therefore, there is thought to be an optimum hardness for suppressing chain elongation.

Based on the results in Fig. 2 and Fig. 7, a formula was constructed that can predict the chain elongation rate from the specifications of the chain-driven system, the chain tension, engine speed $\times$ frequency (total revolutions), engine operating time, and the soot concentration in the engine oil. An outline of this predictive formula is shown below.

$$L = \{f(p) + g(v) + h(\alpha) + i(\beta)\} \times j(l) \times k(\rho) \times N$$

Here,

L : Chain elongation rate (%)

$f(p)$: First-order function of the contact surface pressure p

$g(v)$: Third-order function of the sliding velocity v

$h(\alpha)$: First-order function of the ratio of shear surface at the plate hole α

$i(\beta)$: Second-order function of the plate hole hardness β

$j(l)$: Second-order function of the sliding distance l

$k(\rho)$: Second-order function of the soot concentration in the engine oil ρ

N : Chain bending times

3.2. Verification of Chain Elongation Predictive formula

The chain elongation occurring in engine bench tests was compared with the calculation results to verify the chain elongation predictive formula. To perform verification under various operating conditions, tests were performed under various conditions other than the test conditions used to derive the predictive formula. These included constant speed tests at different engine speeds from low speed to high speed, variable speed tests in which the

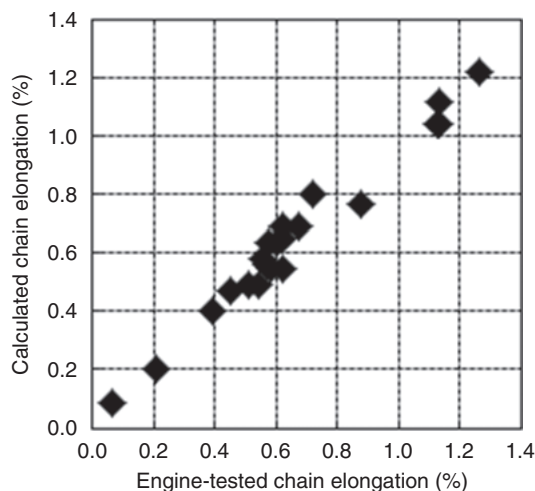


Fig. 8 Comparison of engine-tested values and calculated values

engine speed was varied during the test, and constant speed and variable speed tests using different load conditions. Figure 8 shows the verification results, and shows a high correlation between the test results and calculation result. This confirmed that the constructed predictive formula can be used to estimate chain elongation in an actual engine.

4. Conclusion

The chain elongation mechanism due to soot in the engine oil was analyzed and experiments were performed to obtain the relationship between chain elongation factors and the chain elongation rate with the aim of constructing chain elongation prediction technology. The following knowledge was gained as a result.

- (1) Chain elongation due to soot in the engine oil occurs as a result of soot present on the pin and plate sliding surfaces, and three-body abrasive wear is the dominant mode of plate wear.
- (2) Adhesive wear is the dominant mode of pin surface wear, so oil should be supplied to the pin and plate sliding surfaces. However, under conditions where soot is present in the engine oil, increasing the oil supply amount increases the plate wear, so excessive oil supply may accelerate chain elongation.
- (3) Use of an experimentally derived chain elongation predictive formula enabled estimation of chain elongation in the initial design stages.

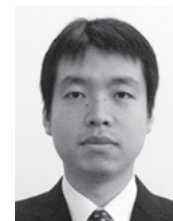
References

- (1) Takagishi, H., Sugawara, H.: Prediction of Chain Load in Timing Chain System, Honda R&D Technical Review, Vol. 14, No. 1, p. 223-230
- (2) Sakaguchi, M., Aguro, M., Yamada, S., Matsumochi, Y., Seki, M., Wakabayashi, T., Koiwa, Y.: Study of Chain Meshing Noise Using Multi-body Dynamics Simulation, 2010 JSAE Annual Congress Proceedings, No. 20105525 (2010)
- (3) Sakaguchi, M., Yamada, S., Seki, M., Koiwa, Y., Wakabayashi, T., Yamauchi, T.: Prediction of Friction in Timing Chain Using Multi-body Dynamics Simulation, 2011 JSAE Annual Congress Proceedings, No. 20115527 (2011)

Author



Masao SEKI



Motoki TANAKA

Development of Cylinder Pressure Sensor Integrated with Direct Injector

Yuji YAMAMOTO*

Gaku SATO**

Takayuki HAYASHI***

ABSTRACT

Polycrystalline piezo-electric element-type cylinder pressure sensors developed thus far require correction of the output characteristics due to pyroelectricity and other factors. This type of sensors also made it necessary to use plug clamp-type sensors, which had a large influence on engine layout.

This issue was addressed by integrating a single crystal piezo-electric element-type cylinder pressure sensor with a direct injector, and this paper introduces the effects and characteristics of this new sensor. Use of an integrated sensor and injector structure provided commonality with existing parts and enabled easy installation in turbocharged engines with direct injection that are expected to become widespread in the future. In addition, with regards to cylinder pressure detectability, the use of single crystal piezo-electric elements with good temperature characteristics achieved a detection accuracy of $\pm 3\%$ compared to the reference sensor (Kistler (6052B)). Furthermore, the knocking waveform detectability was also confirmed to be the same level as that of the reference sensor.

1. Introduction

Recently in Japan, technologies that enhance fuel economy are being mass produced such as small cars equipped with high-efficiency internal combustion engines as opposed to electric cars and hybrid cars, and compact cars equipped with idling stop technology. In addition, technologies that enhance fuel economy centering on increasing combustion efficiency are gaining attention in Europe and other regions, as seen by the increasing popularity of down-sized turbocharged engines and research on Cooled-EGR and Homogeneous Charge Compression Ignition (HCCI) combustion.

Gasoline engines are also evolving from port injection types to direct injection types that spray fuel directly into the combustion chamber, and combustion control that was formerly a challenge using only the combustion chamber shape and ignition timing control has also become possible. As a result, more parameters are being used, and there are demands for even more detailed control. Various companies are developing cylinder pressure sensors to meet these demands, but so far these sensors have been mass produced in only some gasoline and diesel engines⁽¹⁾⁻⁽⁷⁾.

To meet these requests for complex and detailed

control and apply combustion control technology to mass production, research has been conducted over many years to develop methods that combine real-time measurement of gasoline engine cylinder pressures with CPU capable of executing heat and efficiency calculations at high speed. These efforts include the research and development of a plug clamp-type cylinder pressure sensor^{(8), (9)}. This sensor featured a structure that enabled mounting without affecting the valve structure, but assembly limitations required changes to the plug hole and the head cover.

In addition, the sensor used polycrystalline piezo-electric elements, so there were the issues of varying output sensitivity due to the temperature characteristics and output drift due to pyroelectricity, which required special software⁽⁸⁾. These issues were addressed by developing a new sensor. The details are described below.

2. Development of Cylinder Pressure Sensor

2.1. Features of the New Cylinder Pressure Sensor

The use of direct injectors in gasoline engines has increased in recent years. As shown in Fig. 1, the new sensor locates ultra-compact single crystal elements around the fuel injection port at the tip of the direct injector.

* Automobile R&D Center

** Keihin Corporation

*** CITIZEN FINETECH MIYOTA CO., LTD.

The new cylinder pressure sensor is integrated with the injector, so there is no need for special changes to the hardware and head layout. In addition, the new sensor has the characteristic that the element temperature can be maintained at 150°C or less by the cooling effect of the fuel sprayed from the injector, so there are no thermal issues. Furthermore, there is also no change to the shaft seal structure, making it a highly versatile sensor. On the other hand, conventional plug clamp-type sensors detect the combustion pressure as head strain, so they are subject to the influences of intake valve and exhaust valve closing. However, the new cylinder pressure sensor directly detects the cylinder pressure, so it is not subject to these influences.

Figure 2 shows the sensor structure. The basic structure has piezo-electric elements assembled in a ring shape at the tip of the injector. One of the piezo-electric element electrodes is connected externally via electrode material coated onto the pressure transmission location.

The piezo-electric elements use single crystal piezo-electric elements with good temperature characteristics. These piezo-electric elements can accurately detect the cylinder pressure without the need to correct sensitivity due to varying temperature characteristics or to correct drift due to pyroelectricity.

2.2. Characteristics of the Piezo-electric Elements

The piezo-electric elements use langatate single crystals. As shown in Fig. 3, the crystallization structure of a langatate single crystal is that of a trigonal crystal belonging to the space group P321. It is a single crystal without pyroelectricity that exhibits only piezo-electricity,

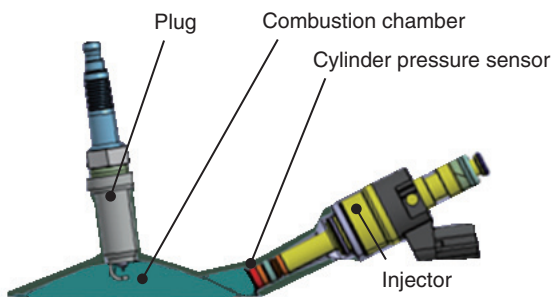


Fig. 1 Installation of injector with cylinder pressure sensor

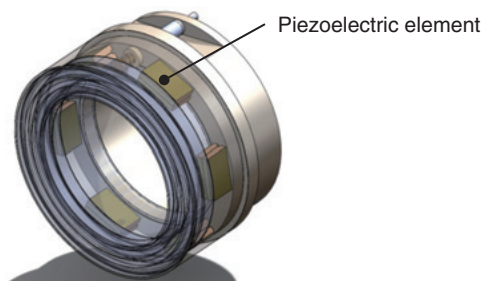


Fig. 2 Cylinder pressure sensor structure

and comprises the four different cation sites A to D. The compositional formula is expressed as $A_3BC_3D_2O_{14}$, and typical langatate is expressed by the chemical formula $La_3Ga_{5.5}Ta_{0.5}O_{14}$.

Polycrystalline piezo-electric elements are molded from powder, and polarization does not occur in the firing stage. A piezo-electric crystal with the entire polycrystalline structure polarized in a single direction is obtained by externally applying high voltage under high temperatures of the Curie point or more to induce unidirectional polarization and forcibly unify the self-canceling spontaneous polarization. Piezo-electric elements manufactured by this method return to the original polarization direction and lose piezo-electricity due to repeated pressure amplitude, and in particular when pressure amplitude is repeated under high temperatures or near the Curie point.

In contrast, langatate piezo-electric single crystals have spontaneous polarization in a single direction from the end of the growth stage, so polarization treatment is not required, and there is no Curie point up to the melting point, so the structure also does not change. In addition, langatate piezo-electric single crystals are disordered crystals that experience differences in melt composition and crystal growth composition during crystal growth, but they can be grown by melt solidification such as the Cz method, so larger crystals are easier to obtain compared to AlN or GaPO₄. Furthermore, the lack of a Curie point up to the melting point is also a reason for the easier growth and stability. For these reasons, langatate single crystals have good stability and the piezo-electric constant has a low temperature dependency over a wide temperature range from low to high temperatures, which gives them good characteristics as piezo-electric elements for cylinder pressure sensors (Table 1).

The sensor structure transmits pressure from the tip of the sensor facing the combustion chamber to the elements, and extracts the stress generated on the sensor as the output. Therefore, the target sensitivity was set at 50 pC/Mpa, based on an S/N ratio that does not affect the combustion parameter calculations, and the sensor layout and structure

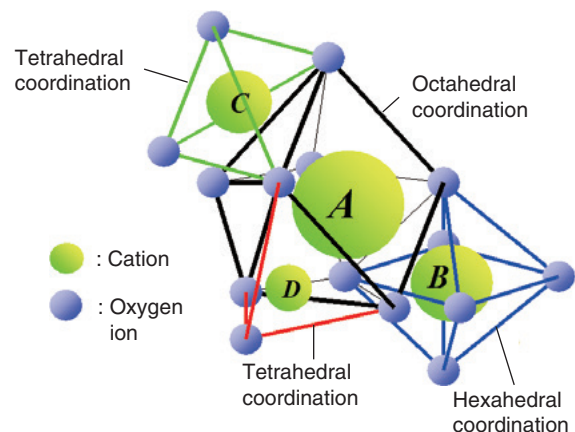


Fig. 3 Langatate crystallization structure

Table 1 Piezoelectric element comparison

		Single crystal		Polycrystalline	
		La _{0.5} Ga _{0.5} Ta _{0.5} O ₁₄ langatate	SiO ₂ quartz	PbTiO ₃ lead titanate	PbNb ₂ O ₆ lead niobate
Performance characteristic	Piezoelectric constant (pC/N)	7.5	2.5	60	80
	Pyroelectricity (pC/°C)	0	10	20000	15000
	Temperature characteristics (%/°C)	0.5<	0.1	0.4	0.3
Degradation characteristic	Curie temperature (°C)	—	570	490	570
	Piezoelectric property degradation temperature (°C)	—	350	300	350
	Breaking strength (MPa)	800	400	500	300
	Piezoelectric property degradation pressure (MPa)	800	300	400	200

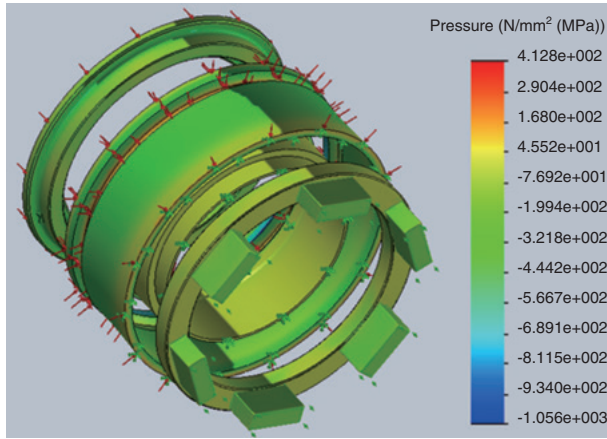


Fig. 4 Sensor stress simulation result

were optimized using simulations. Figure 4 shows the output simulation results for the cylinder pressure sensor. When 15 MPa was applied in the output simulation, the load applied to the piezo-electric element was 103 N. When piezo-electric elements are manufactured using the longitudinal piezo-electric effect Q_x of langatate, the discrete crystal output is approximately 7 pC/N.

The electric charge generated at this time is 721 pC, and the sensor output is estimated to be 48 pC/MPa.

An actual cylinder pressure sensor was manufactured and the output was confirmed to be 50 pC/MPa. This is a difference of 2 pC/MPa from the simulation, which achieved the target performance.

3. Development of Injector Integrated with Cylinder Pressure Sensor

3.1. Injector Structure

To enable mounting of the sensor in the tip of the injector without changing the outer diameter, the valve seat needs to be able to fit inside the ring-shaped sensor. This was achieved by reducing the valve seat and body wall thickness. Structural analysis simulations were used to optimize the valve seat and body to obtain the same strength

as that of the base injector.

Figure 5 shows the structure of the developed injector. The external dimensions are the same as those of the base injector, and the pressure signal cable is wired from the sensor mounted on the tip of the injector, through the inside of the seal ring, to the connector side. The pressure signal cable uses a thin coaxial cable in consideration of electrical noise.

The valve seat inside diameter greatly influences the injector flow rate characteristics. Therefore, simulations were performed and the valve seat inside diameter was set so that the valve seat deformation was the same as that of the base injector or less. Using the valve seat outside diameter D as the variable, the outside diameter was set to suppress the deformation Δd of the valve seat inside diameter d after press-fitting to the same as that of the base injector or less, as shown in Fig. 6.

Stress due to the fuel pressure concentrates on the body undergroove, which is used to lead the pressure signal cable. Figure 7 shows the relationship between the undergroove wall thickness t and the stress. This development set the undergroove wall thickness based on these results, achieving sufficient body strength relative to the stress generated by the fuel pressure.

Next, the methods used to ease assembly and help

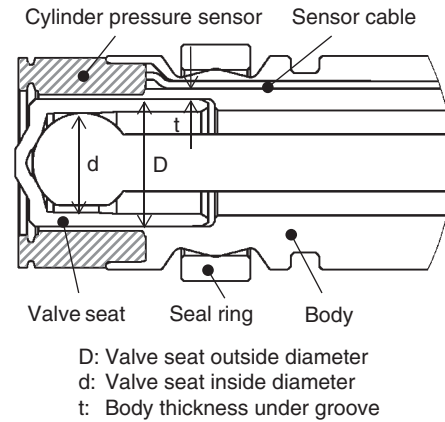


Fig. 5 Injector structure

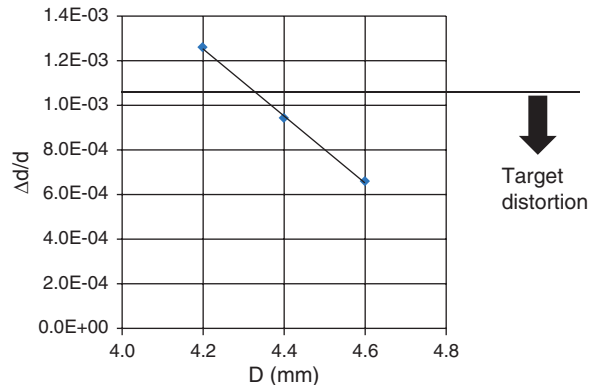


Fig. 6 Valve seat diameter simulation result

prevent broken cables are described. As shown in Fig. 8, the structure leads out the pressure signal cable via a coupler, so a 3-pin-type connector was used. The structure leads out the sensor signal from the center terminal.

The method of dealing with the injector drive noise is as follows. Magnetic field analysis simulation was performed, and the magnetic path structure was determined that minimizes the magnetic field strength. Figure 9 shows the magnetic field analysis simulation result. This figure shows that the housing outside diameter should be increased to reduce the effects of the magnetic field generated by injector operation. This development set the maximum housing outside diameter D_h in accordance with this simulation.

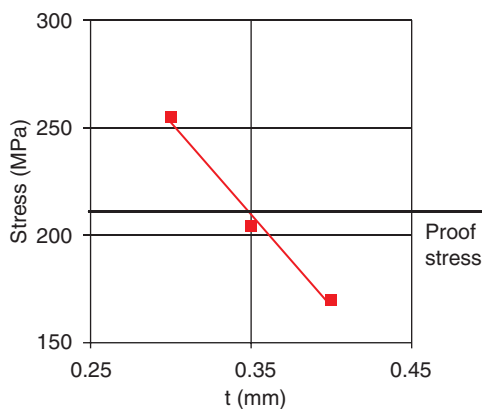


Fig. 7 Body stress simulation result

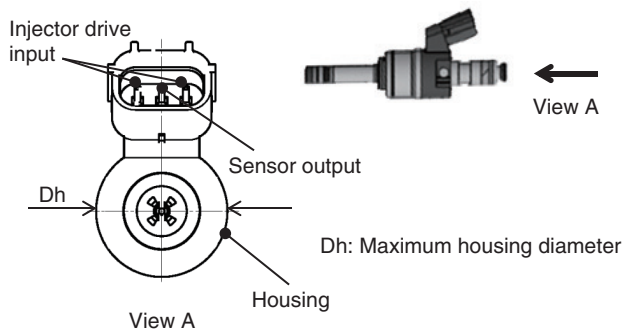


Fig. 8 Signal connector layout

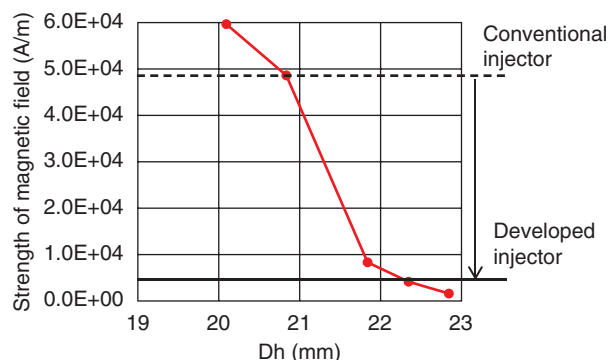


Fig. 9 Magnetic field simulation result

3.2. Evaluation of Injector Performance

The injector flow rate characteristics were verified to confirm the effects of sensor installation. As shown in Fig. 10, the flow rate characteristics proved to be the same as those of the conventional injector.

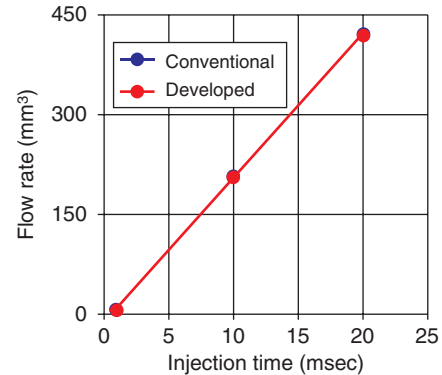


Fig. 10 Flow rate characteristics

4. Verification Results Using an Actual Engine

The cylinder pressure detecting performance was evaluated by comparison with the reference sensor (Kistler (6052B)) using a test bench. The engine used was an inline-four 2000 cm³ engine, and comparison was performed under the conditions of 2000 rpm and 180 Nm.

Figure 11 compares the cylinder pressure waveforms of the reference sensor and the new cylinder pressure sensor, and Fig. 12 shows the correlation with the reference sensor. The developed sensor obtained output results within $\pm 3\%$ of the reference sensor.

In addition, the ability to detect knocking waveforms near 10 kHz is an important factor as a cylinder pressure sensor. Therefore, the knocking waveform under conditions of 1500 rpm and a medium load was compared with that of the reference sensor. As shown in Fig. 13, the developed sensor obtained the same pressure as that of the reference sensor.

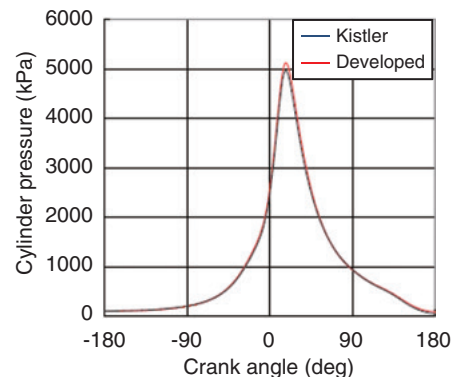


Fig. 11 Cylinder pressure measurement result

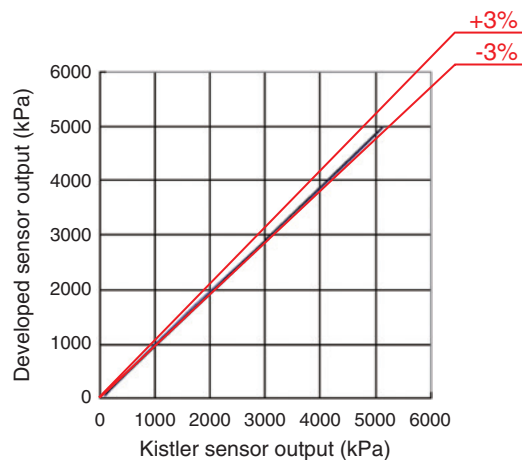


Fig. 12 Cylinder pressure correlation result

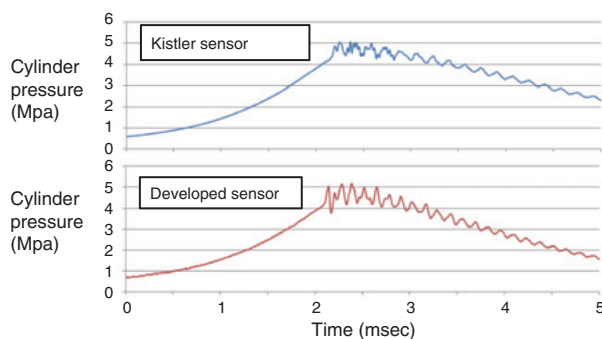


Fig. 13 Cylinder pressures with knocking

5. Conclusion

Ultra-compact single crystal piezo-electric elements were arranged around the fuel injection port of a direct injector. This enabled maintaining the element temperature at 150°C or less due to the active cooling effect of the fuel, thus realizing a lower temperature environment than the conventional technology. In addition, locating the sensor in close proximity to the combustion chamber also enabled detection of knocking. At the same time, the cylinder pressure sensor could be integrated with the direct injector without changing the injector hardware configuration.

- (1) The developed injector can be contained within the same dimensions as the conventional injector, and can be mounted without any change to the engine hardware.
- (2) The developed injector with integrated cylinder pressure sensor was confirmed to have no effect on the flow rate characteristics compared to the conventional direct injector.
- (3) The cylinder pressure detection accuracy was confirmed to be within $\pm 3\%$ of that of the reference sensor, and the knocking pressure could also be detected to the same degree as the reference sensor.

References

- (1) Kondo, K., Niimi, A., Nakamura, T.: Indiscope-A New Combustion Pressure Indicator with Washer Transducers, SAE Paper, 750883 (1975)
- (2) Sawamoto, K., Kawamura, Y., Kita, T.: Individual Cylinder Knock Control by Detecting Cylinder Pressure, SAE Paper, 871911 (1987)
- (3) Kusakabe, H., Okauchi, T., Takirawa, M.: A Cylinder Pressure Sensor for Internal Combustion Engine, SAE Paper, 920701 (1992)
- (4) Wlodarczyk, M. T.: High Accuracy Glow Plug-integrated Cylinder Pressure Sensor for Closed Loop Engine Control, SAE Paper, 2006-01-0184
- (5) Hasegawa, M., Shimasaki, Y., Yamaguchi, S., Kobayashi, M., Sakamoto, H., Kitayama, N., Kanda, T.: Study on ignition Timing Control for Diesel Engines Using In-Cylinder Pressure Sensor, SAE Paper, 2006-01-0180
- (6) Shimasaki, Y., Kobayashi, M., Sakamoto, H., Ueno, M., Hasegawa, M., Yamaguchi, S., Suzuki, T.: Study on Engine Management System Using In-cylinder Pressure Sensor Integrated with Spark Plug, SAE Paper, 2004-01-0519
- (7) Murai, H., Kameshima, A., Moriya, H., Miyamoto, Y.: Glow Plug with Combustion Pressure Sensor, Denso Technical Review, Vol. 8, No. 2, p. 74-78 (2003)
- (8) Yamamoto, Y., Akazaki, S., Ohara, M., Mizuno, H.: Development of Cylinder Pressure Measurement System Using Adaptive Calculation, SAE Paper, 2008-01-1009
- (9) Okuyama, T., Fukuzawa, T., Hashimoto, A.: Development of V10 500HP Engine -Technologies for High-performance and Low-emission Clean Powertrain-, Honda R&D Technical Review, October, 2010, p. 41-49

■ Author ■



Yuji YAMAMOTO



Gaku SATO



Takayuki HAYASHI

Evaluation Condition for Auxiliary Belt Tension Fluctuation in Engine Bench Test

Hiroki TAKAZAWA*

ABSTRACT

In order to evaluate the reliability of an auxiliary belt system consisting of a Poly-V belt and the auxiliary equipment driven by that belt, it is necessary to measure belt tension fluctuation that becomes an input load to the auxiliary component parts. This study proposes an example of a measurement method, clarifies the conditions that increase belt tension fluctuation in the actual use condition, and describes the engine running condition when substituting the actual vehicle running condition with an engine bench. The proposed measurement method enables measurement with the engine in the actual vehicle condition, and does not require changes to the auxiliary belt layout. Strain gauges are adhered to the auxiliary part adjacent to the span where the maximum belt tension fluctuation occurs, and the strain due to the belt tension fluctuation at that position is measured. The measured strain is then converted to the belt tension fluctuation using the calibration equation of belt tension and strain. In order to substitute actual vehicle tests with evaluation by engine bench tests, it is necessary to apply the running condition that results in the maximum belt tension fluctuation as the measurement condition. This requires measurement in the engine starting and compressor on/off conditions, which are transient conditions unique to an actual vehicle, in addition to the condition in which the engine speed is increased at the engine full load plus auxiliary maximum load.

1. Introduction

Many recent engine models use serpentine drive that drives multiple auxiliary equipment with a single belt. The driven AC generator (ACG) and air conditioner compressor have enhanced their efficiency over the years, but driving loads are increasing in order to further enhance automotive comfort. In addition, increased combustion excitation force due to higher combustion efficiency, together with weight reduction around the crankshaft (CS), is increasing the CS angular velocity fluctuation, which is a factor increasing belt tension fluctuation. As a result, auxiliary belt tension fluctuation is increasing, and the loads input to each auxiliary part via the belt are also increasing. To help ensure the reliability of auxiliary belt system component parts under harsh actual use conditions, it is necessary to accurately measure and verify the maximum value of the belt tension fluctuation that becomes these input loads.

Thus far, the reliability of auxiliary belt system component parts has ultimately been verified in the actual vehicle condition. However, demands to front-load engine development have made it necessary to create the specifications in the engine development stage before starting actual vehicle tests. Therefore, it is necessary to understand belt tension fluctuation occurring in an actual vehicle, and to evaluate this belt tension fluctuation by

engine bench tests.

This study proposed an example of a method for measuring belt tension fluctuation, clarified the conditions that increase belt tension fluctuation in the actual use condition, and investigated proper engine running conditions in order to evaluate auxiliary belt tension fluctuation by engine bench tests.

2. Auxiliary Belt System

2.1. Auxiliary Belt System Reliability

The auxiliary belt system refers to the series of functional part groups consisting of the Poly-V belt, the ACG, compressor and other auxiliary equipment driven by that belt, and the belt auto tensioner.

Auxiliary belt system reliability is verified by first measuring and checking the stress with S-N diagrams as the parts durability certification of each component part, and then durability is evaluated in the assembled engine state. In all cases, reliability verification should use the maximum value of the belt tension fluctuation that is the source of the load received by each component part.

2.2. Belt Tension Fluctuation

Friction drive-type auxiliary belts apply the set tension for transmission of the driving force by pressing the driving

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surface against the pulley surface as shown in Fig. 1. For this reason, the tension of each span is maintained uniformly at the set tension in the engine-stopped condition. However, when the engine is operated, the tension fluctuates due to the CS rotation fluctuation, the moment of inertia, and load of the auxiliary equipment (Fig. 2). This fluctuation of the tension in the engine-running condition is called belt tension fluctuation.

Belt tension fluctuation is amplified by the resonance of the auxiliary belt system, and in some cases can result in excessive tension that damages the auxiliary component parts.

The forced vibration model is generally expressed by the model shown in Fig. 3(a). When this model is applied to the auxiliary belt system, the external forced vibration force generated by the CS angular velocity fluctuation is $f_0 \sin \omega t$, and the inertia mass I of the ACG pulley can be expressed by a model consisting of an auxiliary belt comprising the spring constant k and the viscous damping constant c (Fig. 3(b)). In addition, when the natural frequency of the auxiliary belt system matches the forced vibration frequency ω , resonance occurs in the auxiliary belt system. The maximum amplitude occurs at this time, and the phase lag becomes $\pi/2$ ⁽¹⁾⁻⁽³⁾.

Figure 4 shows the CS and ACG pulley velocities when the belt tension fluctuation is increased by resonance. This shows that the ACG pulley velocity phase is delayed by approximately $\pi/2$ relative to the CS pulley velocity, and it can be confirmed that the auxiliary belt system is in the resonance zone and the belt tension fluctuation increases. This condition is referred to hereafter as auxiliary belt system resonance.

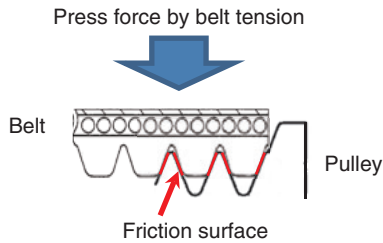


Fig. 1 Friction transmission of auxiliary belt

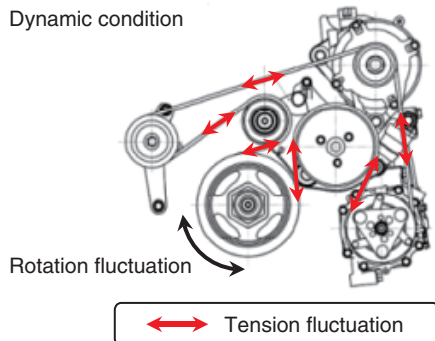
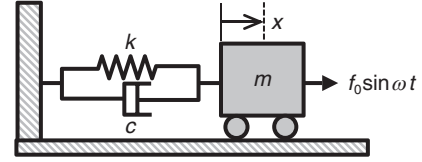


Fig. 2 Auxiliary belt tension fluctuation

Conversely, when a one-way clutch⁽⁴⁾ is mounted on the ACG pulley and the natural frequency of the auxiliary belt system is offset from the forced vibration frequency of the CS pulley to suppress resonance, the CS and ACG pulley velocity fluctuation become in-phase and belt tension fluctuation is reduced as shown in Fig. 5.

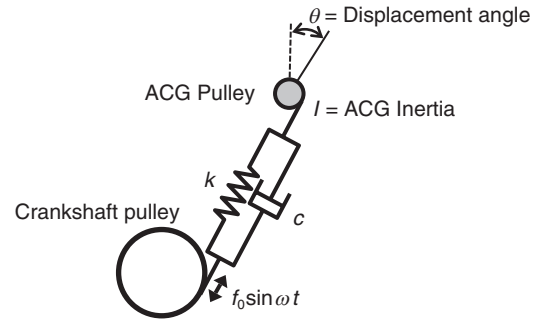


Equation of motion

$$m \frac{d^2x}{dt^2} = -c \frac{dx}{dt} - kx + f_0 \sin \omega t$$

$$\left[\begin{array}{l} c : \text{Viscous damping constant} \\ f_0 \sin \omega t : \text{Forced vibration} \\ k : \text{Spring rate} \\ m : \text{Weight} \\ x : \text{Displacement} \end{array} \right]$$

(a) General model of forced vibration



(b) Auxiliary belt system forced vibration model

Fig. 3 Forced vibration model

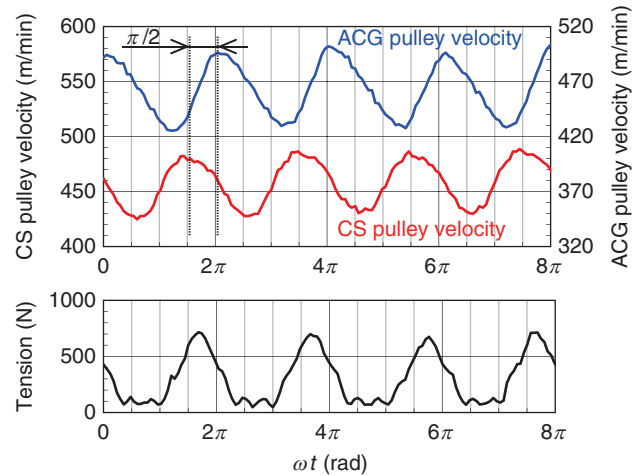


Fig. 4 Auxiliary belt tension fluctuation in belt system resonance zone

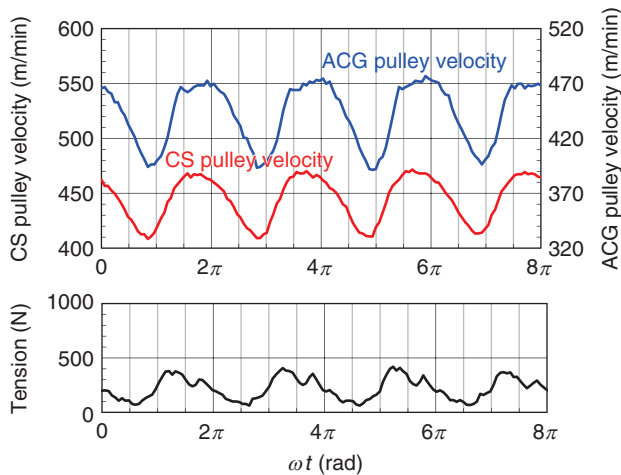


Fig. 5 Auxiliary belt tension fluctuation in belt system non-resonance zone

In addition, belt tension fluctuation also increases under transient operating conditions where the auxiliary load fluctuates.

3. Measurement Method

3.1. Belt Tension Fluctuation Measurement Position

Belt tension fluctuation occurs in all spans, but is the highest before the CS pulley (immediately upstream from the direction of rotation), where all of the auxiliary loads are added. When auxiliary belt system resonance occurs, the resonance is added to the tension in each span, but the position where the maximum tension is generated does not change. Therefore, the maximum tension in the belt layout shown in Fig. 6 is generated between the CS pulley and the water pump (WP), so the tension in that position (red arrows in Fig. 6) was measured.

3.2. Load Cell Creation Method

A known general method of measuring auxiliary belt tension fluctuation is to set a load cell pulley that outputs bending stress in the center of the span where the belt tension is to be obtained. In order to measure belt tension fluctuation in the actual use condition, it is necessary to set the load cell in the condition with the engine mounted in an actual vehicle. However, it is often a challenge to set a load cell given the space constraints in an actual vehicle. Therefore, this study employed a method that uses strain gauges to make an auxiliary component part that does not require special space into a load cell. Strain gauges were adhered to the WP housing to make the WP into a load cell in order to measure the tension between the CS pulley and the WP pulley, which is the position where the maximum tension is generated. The belt tension fluctuation was measured as the WP bending reaction shown in Fig. 6. No machining was performed to grind the ribs and reduce stiffness in order

to increase the sensitivity, and the original shape was maintained. Two strain gauges each were adhered to the base of the WP housing ribs at the tensile position and the compression position as shown in Fig. 7(a). In addition, a bridge circuit that connects the opposing strain gauges was configured as shown in Fig. 7(b) to simultaneously increase the sensitivity and cancel the apparent output due to temperature change.

3.3. Load Cell Calibration Method

Next, the method of converting the strain output of the WP load cell into the auxiliary belt tension fluctuation is described. The WP with strain gauges adhered was assembled into the engine used for measurement, and a jig capable of generating arbitrary tension that can be substituted for the auxiliary belt auto tensioner (AT) was set

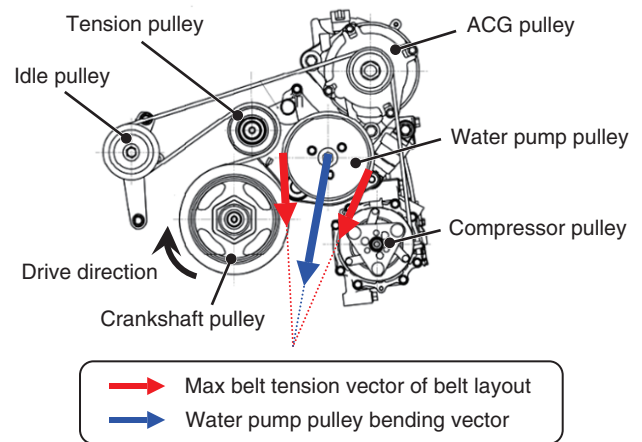


Fig. 6 Auxiliary belt layout of tension fluctuation measuring engine

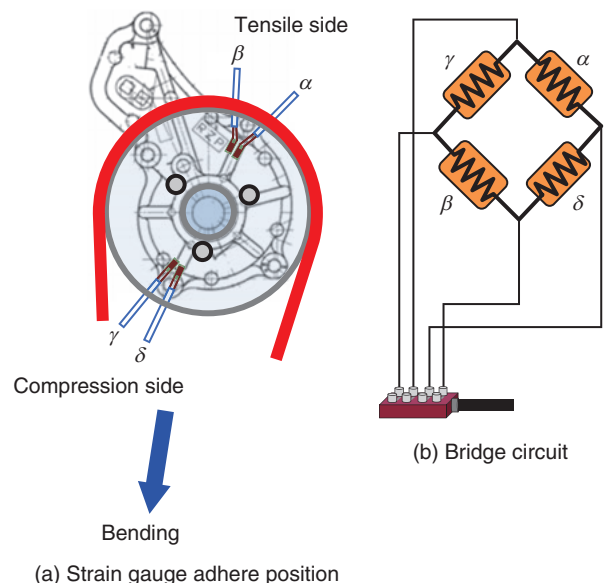


Fig. 7 Water pump load cell-making method

as shown in Fig. 8. The set tension of the belt was varied by rotating the hexagon nut in the center of the jig. The tension was obtained from the vibration frequency when the span between the ACG and the idler was flicked in the engine-stopped condition, and the calibration line was obtained from the relationship between the tension and the strain output at that time.

Figure 9 shows the relationship between the belt tension and the WP housing strain. The belt tension has a directly proportional relationship with the WP housing strain, and the belt tension fluctuation was calculated by multiplying the strain generated on the WP housing with the calibration line slope.

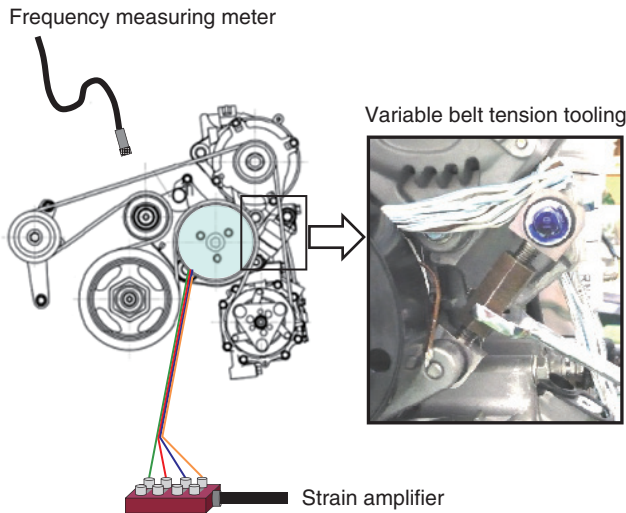


Fig. 8 Variable belt tension tooling and belt set tension measuring by span frequency at static condition

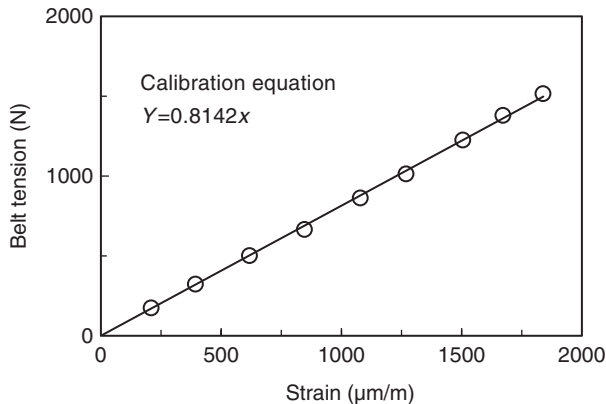


Fig. 9 Static calibration line between belt tension and water pump load cell strain

4. Test Conditions

4.1. Engine Specifications

The auxiliary equipment specifications may differ according to the vehicle, even when the engine uses the

Table 1 Measuring engine specifications

Part	Specification
Engine	Higher crankshaft angle acceleration engine of same belt layout engine series
Transmission	Lower crankshaft axis inertia of all transmission applications For firing engine bench, lockup clutch tight condition
Auxiliary belt	Belt length lower tolerance
Auto tension	Spring force higher tolerance Damping force higher tolerance
AC generator	Higher volume of same belt layout engine series
Compressor	Higher refrigerating capacity of same belt layout engine series

Table 2 Engine and auxiliary load conditions of firing bench

Equipment	Load type		Condition
Bench	Engine load		Continuous full load
	Auxiliary load	AC generator	Continuous full load
		Compressor	Continuous full load

same type of engine block as the frame. Therefore, the specifications that result in the highest belt tension should be selected when measuring belt tension fluctuation.

Table 1 lists the items related to engine specifications that increase belt tension fluctuation. Measurement used the auxiliary belt system shown in Fig. 6, with each item in Table 1 set so as to increase belt tension fluctuation. Engine and transmission specifications were selected that result in high CS rotational fluctuation, which is a source of auxiliary belt tension fluctuation. Auxiliary belt system specifications were selected that increase the belt set tension to the tolerance upper limit, and also increase the maximum tension. Auxiliary load specifications were selected that result in high driving torque and increase the driving transmission force of the belt.

4.2. Running Conditions

The maximum belt tension in the auxiliary belt system resonance condition was measured using engine bench tests. Measurement was performed in the running mode where the engine speed is increased from the lowest speed at which the throttle can be fully opened up to the maximum output engine speed in approximately 30 s. Next, an actual vehicle was used to investigate whether there are any conditions under which the maximum belt tension in the resonance condition is exceeded due to transient running conditions in the actual use condition.

Table 2 lists the load conditions used to obtain the maximum belt tension in the resonance condition on the

Table 3 Engine and auxiliary transient load conditions

Equipment	Load Type		Condition
Vehicle	Engine Load		Engine start
			Put in gear
			Racing
			Shift change
			Torque converter stall condition
	Auxiliary Load	AC generator	On-Off
		Compressor	On-Off

engine bench. The engine load was the maximum load that results in the maximum CS rotation fluctuation. The auxiliary load was the maximum load that results in the maximum driving torque. This mode in which the engine speed is increased at the maximum engine and auxiliary loads is referred to hereafter as the continuous load condition.

Table 3 lists the conditions hypothesized to increase belt tension during transient conditions in an actual vehicle; that is to say, in the actual use condition. First, the running conditions were extracted that involve a change in velocity that increases the belt tension on both the drive-side CS pulley and the driven-side auxiliary pulleys. Next, these running conditions were sorted into factors deriving from engine loads and auxiliary loads.

5. Test Results

5.1. Continuous Load Condition Tests Using an Engine Bench

Figure 10 shows the results of measuring the belt tension fluctuation on an engine bench using an engine with the specifications listed in Table 1, under the continuous load conditions listed in Table 2. This shows that large resonance tension originating in the torque fluctuation due to combustion occurred in the low-engine-speed range of 2000 rpm or less. Conversely, the range of 2000 rpm or

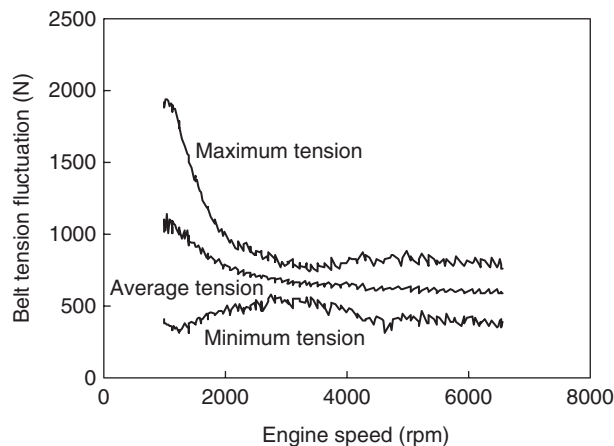


Fig. 10 Belt tension fluctuation of continuous load condition in firing engine bench

more was outside the resonance zone, so the belt tension fluctuation stabilized at a lower level.

5.2. Transient Running Condition Tests in the Actual Use Condition

The specifications listed in Table 1 were assembled into an actual engine, and the belt tension fluctuation was measured under each transient condition listed in Table 3. These results were then compared with the results obtained in the continuous load condition (Fig. 10) to verify the need to add each transient condition to the engine bench running conditions.

5.2.1. Engine starting

Figure 11 shows the belt tension fluctuation at engine start. Combustion starts after cranking, then the engine speed rises and the peak tension occurs partway to the first idle state, after which the belt tension fluctuation drops.

Figure 12 shows the relationship between the CS pulley

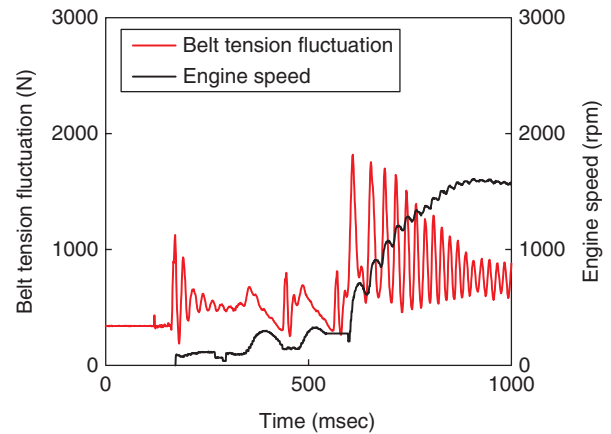


Fig. 11 Auxiliary belt tension fluctuation at engine start

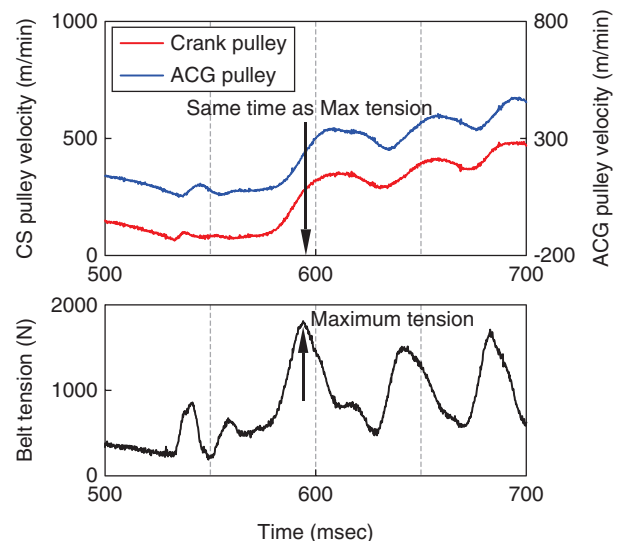


Fig. 12 Relation between belt tension fluctuation and pulley velocity

and ACG pulley velocities when the belt tension fluctuation rises immediately after engine start. The initial tension peak near 2000 N is synchronized with the rise in the CS pulley velocity, and it is thought that the maximum tension was generated as the force to increase the rotations of all driven auxiliary equipment. In addition, the rise in the CS pulley velocity matches with the ACG pulley velocity phase, indicating that auxiliary belt system resonance does not occur.

Figure 13 compares the tension peak at engine start with the engine bench test results (Fig. 10). The tension peak exhibits a high value at engine start, even though there is no auxiliary load. In addition, there is no auxiliary belt system resonance and belt tension is generated by only CS acceleration, so the generating factors differ from the maximum belt tension in the continuous load condition. Therefore, the engine starting condition should be added to the continuous load condition for evaluation.

5.2.2. Putting in gear

Figure 14 shows the belt tension fluctuation when the transmission select lever is changed to the D position

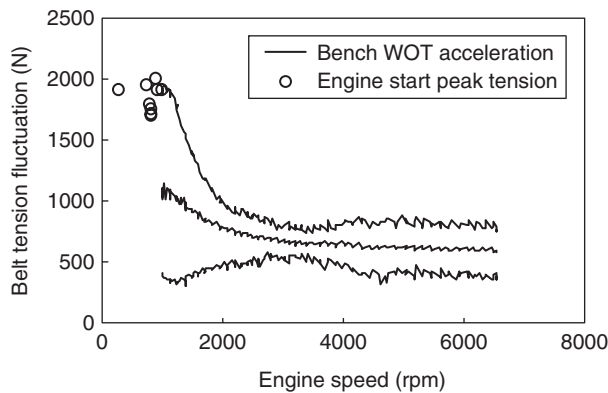


Fig. 13 Belt tension comparison between peak belt tension at engine start and bench continuous load condition

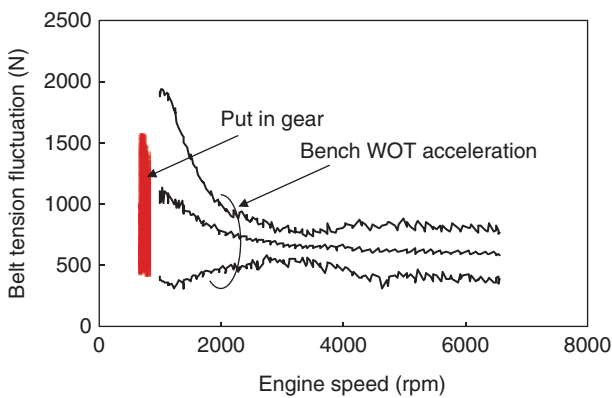


Fig. 14 Belt tension fluctuation comparison between putting in gear position and bench continuous load condition

from the condition with the full electrical load and air conditioning load applied during idle operation with the transmission in the neutral position. This operation to change the gear from neutral to the D position is hereafter referred to as putting in gear.

In order to maintain the idle engine speed with respect to the auxiliary load and the drive system loads such as the torque converter, the engine control system increases the intake manifold pressure, which increases the CS rotation fluctuation. In addition, the idle engine speed is within the auxiliary belt system resonance speed zone (Fig. 10), so the belt tension fluctuation increases. However, the continuous load condition on the engine bench performs engine full-load operation from near the idle engine speed, so the belt tension fluctuation is higher than when putting in gear. This means that the putting in gear condition is included in the continuous load condition, so there is no need to add it to the engine bench evaluation conditions.

5.2.3. Racing

Figure 15 shows the time series measurement results of the belt tension fluctuation and the change in engine speed when racing is performed.

Compared to the belt tension fluctuation due to auxiliary belt system resonance at the idle engine speed, both the engine load and the belt tension fluctuation increased the instant the throttle was opened, but the tension then dropped when the engine speed rose above the resonance zone.

Figure 16 arranges the belt tension fluctuation results in Fig. 15 according to the relationship with the engine speed, and compares it with the results in the continuous load condition on the engine bench. In the racing condition, auxiliary belt system resonance occurred in the same low-speed range as in the continuous load condition. However, the engine speed rises above the resonance zone before the engine full load is reached in the low-speed range where resonance occurs. As a result, the belt tension fluctuation in the racing condition is lower than that in the continuous load condition, so there is no need to add it to the engine bench evaluation conditions.

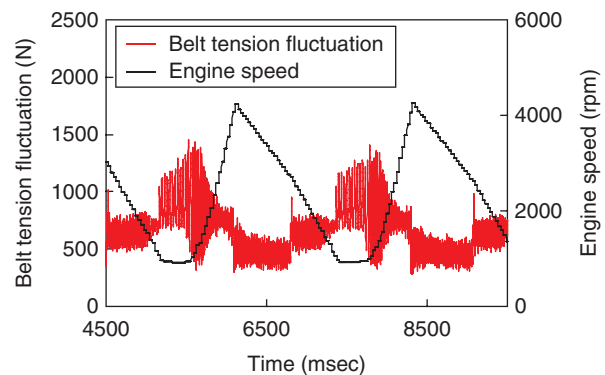


Fig. 15 Belt tension fluctuation and engine speed at racing condition

5.2.4. Shifting

The belt tension fluctuation was confirmed when accelerating and shifting in a CVT vehicle. Figure 17 shows these results. No influence whatsoever of CVT shifting was confirmed, and the tension fluctuation resembles that during the continuous load condition on the engine bench. However, in the actual use condition the torque converter absorbed the CS rotation fluctuation in the low-engine-speed range, so the belt tension fluctuation did not increase compared to the engine bench test condition. This means that belt tension fluctuation in the shifting transient condition is included in the continuous load condition, so there is no need to add it to the engine bench evaluation conditions.

5.2.5. Torque converter stall

Figure 18 shows the belt tension fluctuation when the transmission select lever is changed to the D position, the accelerator pedal is pressed while stepping on the brake until the throttle is fully open, and the output is absorbed by the torque converter. This running condition is referred to hereafter as torque converter stall.

The accelerator pedal was pressed until the throttle is fully open, so the engine load increased, but the engine

speed rose to the torque converter stall speed, so the low engine speed could not be maintained with the throttle fully open as in the continuous load condition, which resulted in a low maximum tension. This means that the belt tension fluctuation in the torque converter stall condition is included in the continuous load condition, so there is no need to add it to the engine bench evaluation conditions.

5.3. Auxiliary Load Transient Condition Tests using an Actual Vehicle

The auxiliary belt tension fluctuation when auxiliary loads are applied to an actual vehicle in a transient manner was measured to confirm whether there is a difference from that in the continuous load condition, and it was verified whether these conditions can be included in the continuous load condition.

5.3.1. ACG load

Figure 19 shows the belt tension fluctuation and the ACG load when electric loads such as the headlights and rear defogger are applied in succession. The ACG load rises as the electric equipment part switches are turned on, and at the same time the belt tension fluctuation also increases.

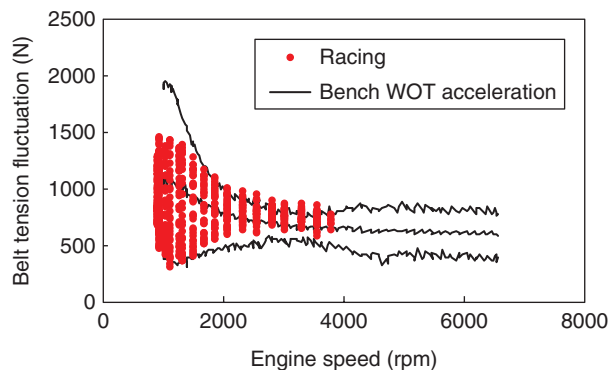


Fig. 16 Belt tension fluctuation comparison between racing and bench continuous load condition

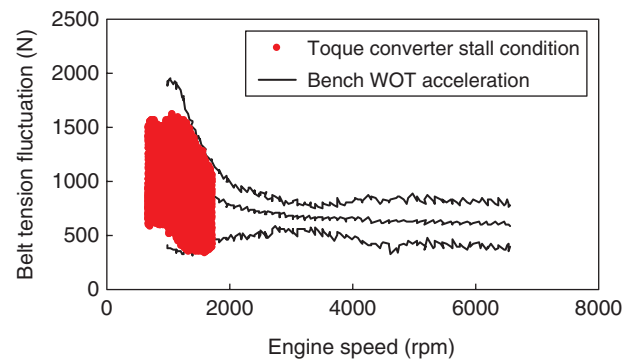


Fig. 18 Belt tension fluctuation comparison between torque converter stall condition and bench continuous load condition

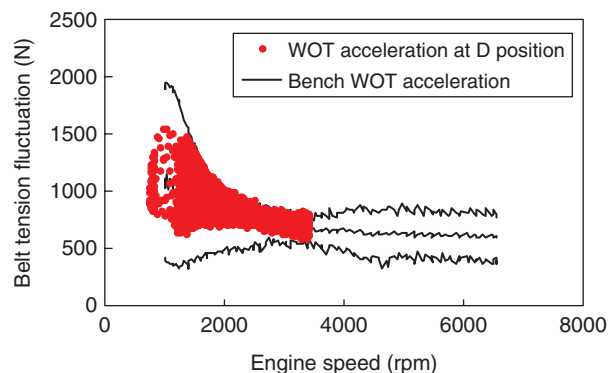


Fig. 17 Belt tension fluctuation comparison between WOT acceleration at select lever drive position and bench continuous load condition

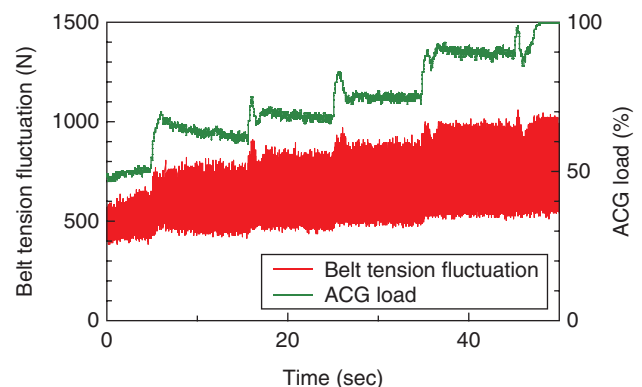


Fig. 19 Belt tension fluctuation and ACG load when ACG load rises in succession

Figure 20 shows the relationship between the fluctuation amplitude peak value on the high tension side of the belt tension fluctuation shown in Fig. 19 and the ACG load. This shows that the ACG load and maximum belt tension fluctuation value are correlated, and that the belt tension fluctuation did not have a peak value due to the transient manner that applied the ACG load.

The belt tension fluctuation changes according to the amount of the ACG load, but it does not change according to the manner in which the load is applied. Therefore, the condition with the maximum electric generation load results in the largest ACG load, and that condition is the same as the continuous load condition.

5.3.2. Compressor load

Figure 21 shows the belt tension fluctuation when the compressor switch is turned on and off. The belt tension rose and indicated a peak when a load was applied, but then dropped instantly thereafter. This is because the compressor is accelerated from a velocity of zero to the crankshaft velocity the instant the clutch is engaged, so the work to accelerate the compressor appears as belt tension.

Figure 22 shows the results of comparing the peak tension at compressor on/off with the continuous load condition. The peak tension in the high-engine-speed range

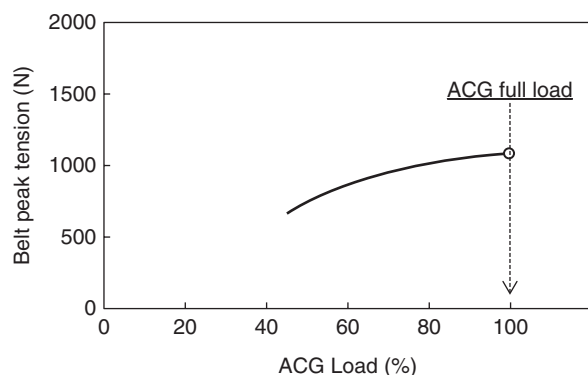


Fig. 20 Relation between belt peak tension and ACG load

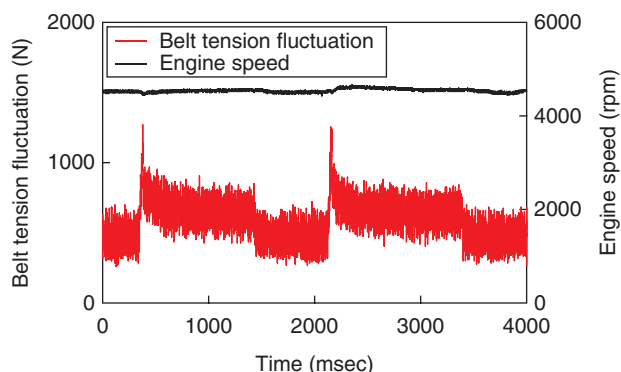


Fig. 21 Belt tension fluctuation at compressor clutch engage

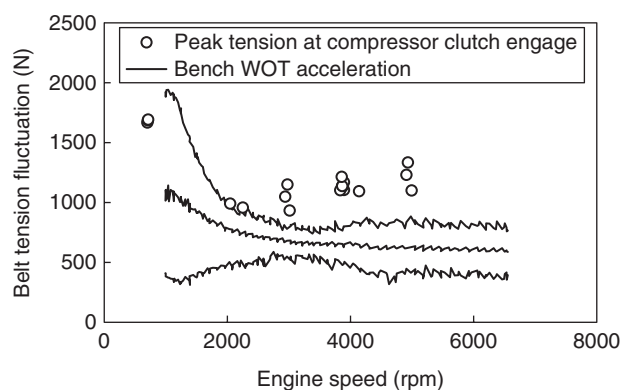


Fig. 22 Belt tension comparison between peak belt tension at compressor clutch engage and bench continuous load condition

exceeds that in the continuous load condition, and this difference became noticeable at engine speeds of 2000 rpm or more that are outside the auxiliary belt system resonance zone. This shows that the peak tension at compressor on is generated by a different factor than that for tension fluctuation due to auxiliary belt system resonance, and cannot be included in the continuous load condition. Therefore, the compressor on/off running condition should be added to the engine bench test.

6. Conclusion

In order to evaluate auxiliary belt system reliability, an example of a belt tension fluctuation measurement method was proposed, and the engine running conditions and auxiliary load conditions were clarified that enable evaluation of an auxiliary belt system in the actual use condition by engine bench tests.

(1) Belt tension fluctuation measurement method

A method was proposed to enable measurement of belt tension fluctuation using an engine in the actual vehicle condition, without changing the auxiliary belt layout. This method adheres strain gauges to the auxiliary part adjacent to the span where the maximum belt tension fluctuation is generated, to enable use as a substitute load cell.

(2) Running conditions used to measure belt tension fluctuation

The engine starting and compressor on/off conditions are transient conditions unique to actual vehicles. When evaluating belt tension fluctuation by engine bench tests, these transient conditions should be added to the running condition that increases the engine speed in the continuous load condition, which consists of the engine full load and maximum auxiliary load.

References

- (1) Watari, A.: Forced vibration, Kikaishindou, Maruzen publishing CO., LTD., p. 103-107 (1966) (in Japanese)

- (2) Sakata, M.: Vibration of degree-of-freedom, Forced vibration, Hadoutoshindounokougaku, Kyoritsu shuppan CO., LTD, p. 53-59 (1980) (in Japanese)
- (3) Meirovitch, L.: Forced vibration of single-degree-of-freedom system, Elements of vibration analysis, Burein tosyō CO., LTD., p. 39-46 (1984)
- (4) Sato, K., Mikuriya, I.: Compact Clutch Integrated Pulley for Alternator, NTN Technical Review, No. 75, p. 123-127 (2007) (in Japanese)

■ Author ■



Hiroki TAKAZAWA

Visualization of Air Current when Noise Occurs with Throttle Rapidly Opening

Jin YAMAGISHI*

Takeshi WAKAMATSU*

ABSTRACT

Pressure measurements were conducted and airflows were visualized inside the intake manifold to help enable an overall understanding to be obtained of the phenomenon of noise occurring under conditions of rapid throttle opening during engine operation. The use of plastic intake manifolds and drive-by-wire (DBW) has made this noise more conspicuous. The results of attempts to identify the sound source showed that the noise was radiated from the surface of the intake manifold. For measurements of the behavior of the airflows inside the intake manifold, airflows were visualized while the engine was in operation using a high-speed camera (taking advantage of recent performance enhancements) and a special lighting system in a method that can be applied during the development process. The results of these measurements demonstrated that noise resulting from the collision of high-speed airflows is the origin of the noise under study.

1. Introduction

Modularization and the achievement of weight savings have long been demanded for engine components, and the use of plastic as a material and the reduction of the thickness of parts have been pushed ahead in order to realize these goals. The use of plastic for intake manifolds came into use around 15 years ago, and today many engines employ plastic intake manifolds. At the same time, the expansion of the degree of freedom of throttle control with the application of DBW has increased the need for rapid opening of the throttle to control the resting position of the pistons, in order to respond to demands for enhanced driving performance and environmental performance.

This has resulted in noises, which were previously not heard when aluminum intake manifolds and mechanically controlled throttles were employed, becoming conspicuous when the throttle is rapidly opened.

Figure 1 shows the results of measurements of the frequency characteristics of transmitted sound and sound pressure for different intake manifold materials.

These results indicate that for intake manifolds of the same shape, the transition from aluminum to plastic as the material decreases sound transmission loss, thus increasing noise. Automakers have been conducting research on the reduction of noise under conditions of rapid opening of the throttle for more than 10 years⁽¹⁾. Technologies for noise reduction employing nets and other devices⁽²⁾ are widely applied in mass-production vehicles.

Previous research has conducted analyses using measurements of sound inside intake manifold pipes and computational fluid dynamics (CFD); there are few reported examples in which measurements of airflow behavior in actual pipes have been conducted⁽¹⁾.

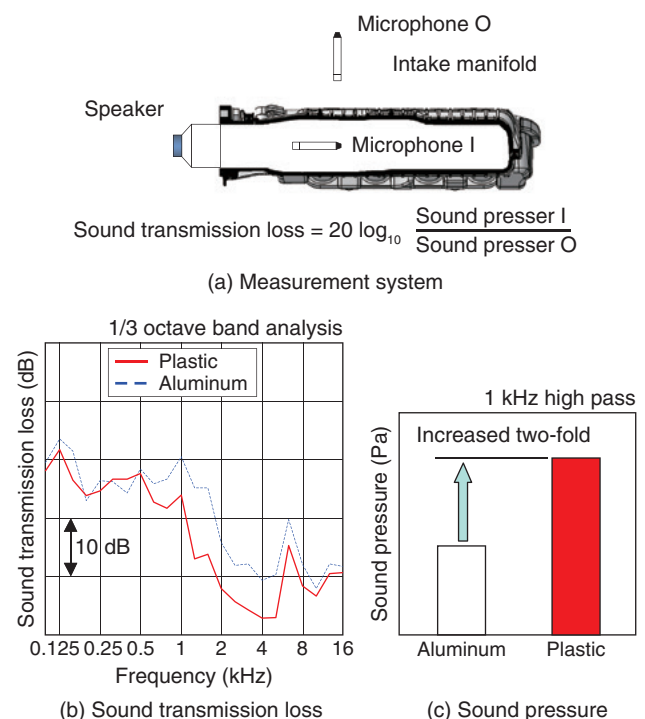


Fig. 1 Comparison of sound transmission loss and noise

* Automobile R&D Center

The research discussed in this paper sought to provide guideposts towards measures to respond to noises by visualizing and studying airflow behavior when noise is generated during engine operation.

2. Development Goals

The goals of the research were to:

- (1) Identify the location in which intake manifold noise is generated; and
- (2) Visualize airflow behavior in the intake manifold pipes. By these means, the relationship between noise and airflow behavior and fluctuations in the internal pressure of the intake manifold pipes is determined, and an understanding of the phenomenon of the noise is achieved.

3. Identification of Location of Noise Generation

A 2.4 L engine fitted with a plastic intake manifold was employed in tests. Table 1 shows the engine specifications.

The noise under consideration is most conspicuous when the throttle is rapidly opened while the driver races the engine and engine speed has dropped to 1000 rpm. These conditions were considered as one racing cycle, and tests were conducted in which this cycle was repeated. The setting for the degree of opening of the throttle was identical to the conditions under which the noise phenomenon becomes noticeable.

Airflow behavior when the throttle is rapidly opened is a transient phenomenon, and obtaining stable results therefore represents a challenge. Because of this, the maximum values for sound and pressure during a single racing cycle were measured several tens of times, and average values were obtained.

3.1. Tests using Sound Source Identification Device

A sound source investigation device employing multiple microphones, which is generally used for unsteady-state sound source investigation, was employed to examine the noise occurring during rapid opening of the throttle. Figure 2 shows the device employed and an example of the results obtained. The tests indicated that the noise is radiated from the surface of the intake manifold, as shown in Fig. 2.

3.2. Detailed Analysis of Intake Manifold

A detailed analysis of the intake manifold was conducted in order to further clarify the location in which the noise is generated.

3.2.1. Measurements of near-field sound, surface vibration, and internal pressure of intake manifold

Figure 3 shows the test intake manifold.

The intake manifold was divided into three sections (the section close to the throttle body, the chamber section, and

Table 1 Engine specifications

Item	Specification
Engine type	4-stroke gasoline engine
Cylinder configuration	In-line 4-cylinder
Bore x stroke (mm)	87 x 99
Displacement (cm ³)	2354
Compression ratio	10.5
Valve train	16-valve DOHC i-VTEC
Intake manifold	Plastic with glass fiber
Max. power (kW [PS] / rpm)	142 [193] / 7000
Max. torque (Nm [kgm] / rpm)	220 [22.4] / 4400

the port section), and the internal pressure of the manifold was measured in addition to near-field sound and surface vibration.

Because the internal pressure of the intake manifold instantaneously increases to atmospheric pressure from a state of vacuum when the throttle is rapidly opened during engine operation, high-frequency fluctuating components are generated in addition to a significant change in pressure. Given this, it is desirable to employ measurement equipment with a wide dynamic range, up to high frequencies. Conventionally, the fluctuating components of sound pressure have been measured using probe microphones, but the present project employed pressure sensors that have minimal effect on airflow behavior and are able to take measurements up to high frequencies.

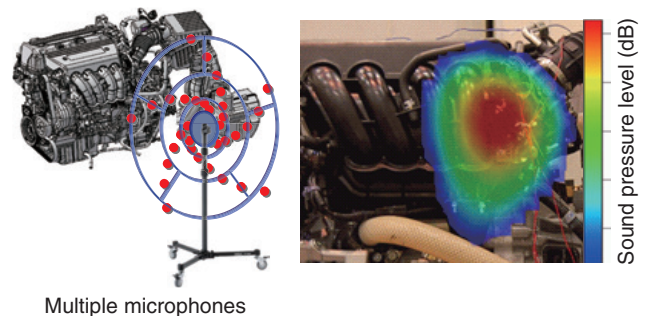


Fig. 2 Sound source investigation analysis

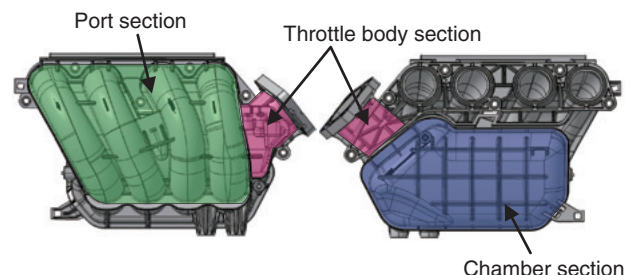


Fig. 3 Measurement intake manifold

The measurement results shown in Fig. 4 indicate the following:

- (1) (a) and (c) indicate that sound pressure is highest in the section close to the throttle body, where pressure fluctuation is high;
- (2) (a) indicates that the internal pressure fluctuations in the intake manifold are approximately five times greater in the section close to the throttle body than in the port section; and
- (3) (a), (b), and (c) indicate that an area in which the surface vibration of the intake manifold is high exists in the port section (in which internal pressure fluctuation is low), and the level of near-field sound is also high in this section.

3.2.2. Modal analysis of intake manifold

Given that an area in which levels of intake manifold vibration and near-field sound are high exists in the port section, in which internal pressure is low, it was possible to consider that a vibration transmission effect was occurring due to the structure. A modal analysis was therefore conducted. Measurement conditions were identical to firing operation, and measurements were conducted with the intake manifold coupled to the engine and the engine supported by engine mounts. A film surface mode exists in the area in which vibration is high in the high-frequency range at 1 kHz and above. Figure 5 shows an example of the mode, surrounded by a red circle.

3.3. Summary of Clarification of Location of Noise Generation

Based on the results above, it was conjectured that the noise is airflow noise, produced by the sound component generated when pressure fluctuations close to the throttle

body are transmitted through the intake manifold, and the component radiated from the area in which the mode is generated when the intake manifold is excited from the interior and vibration is transmitted.

Given this, understanding airflow behavior close to the throttle body could be considered important in identifying the source of the noise.

4. Visualization of Airflow Behavior in Pipes

4.1. Study of Visualization Methods

Based on the results discussed in Chapter 3, it could be conjectured that the airflow behavior in the intake manifold close to the throttle body was a factor in the occurrence of noise. The potential for visualization of airflow in the intake manifold was therefore studied.

The following methods were available for overall visualization of the airflow:

- (1) Use of a transparent intake manifold of the same shape as the manifold under study; or
- (2) Insertion of a transparent cylinder in the area in which the airflow behavior occurs.

With regard to method (1), it would be possible to consider the airflow behavior occurring in the intake manifold identical to the actual airflow behavior, but the method presented numerous issues, including the setting of the light source, the position for measurements, the distortion of the image due to the shape of the manifold, and the transmission coefficient of the light. This method was therefore not employed in the present study.

Method (2) could be considered to offer measurements that resolved the issues indicated above, but also presented the possibility that the behavior of the airflow might differ. A test was therefore conducted to determine whether this would be the case.

Internal pressure was measured and the effect of the addition of a transparent extension pipe was studied in order to determine whether it produced any difference in internal airflow behavior.

The results are shown in Fig. 6. Comparisons at the same distance from the throttle body show almost identical values for fluctuation in internal pressure, indicating that airflow behavior remains the same even when a transparent extension pipe is used.

Based on this result, measurements were taken using method (2).

4.2. Overview of Visualization and Measurement

Figure 7 shows a diagram of the visualization and measurement system.

An oil mist is introduced upstream from the throttle body for the purpose of visualization. High-speed camera images in the transparent section and signals for sound, pressure, and other parameters transmitted via an A/D board are synchronized and input to a PC. By this means, it is possible to determine the behavior of the airflow when

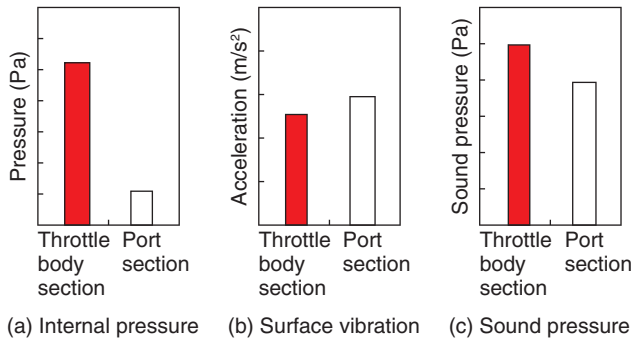
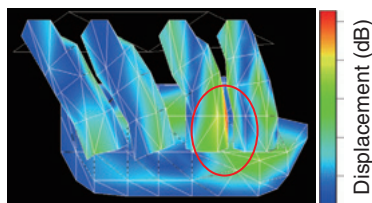


Fig. 4 Sectional analysis of sound source



Port section mode at 1280 Hz

Fig. 5 Modal analysis result

airflow noise occurs.

When imaging using a high-speed camera under normal light, light from the exterior continuously radiates onto the surface, and this means that the images become darker the higher the speed at which they are taken.

When measuring airflow behavior in a transparent pipe, noise generated by surface reflection and internal reflection, in addition to darkness, makes it challenging to isolate the airflow to be studied.

Particle image velocimetry (PIV) measurement using laser sheet light, a method in which strong light energy that can be synchronized with the operation of a high-speed camera is radiated onto the subject, is available to respond to these issues, but this method necessitates large equipment and advanced technology, and it is therefore not employed for measurements while engines are being operated in development laboratories.

It proved to be possible to visualize the airflow during engine operation using a special light (manufactured by nac

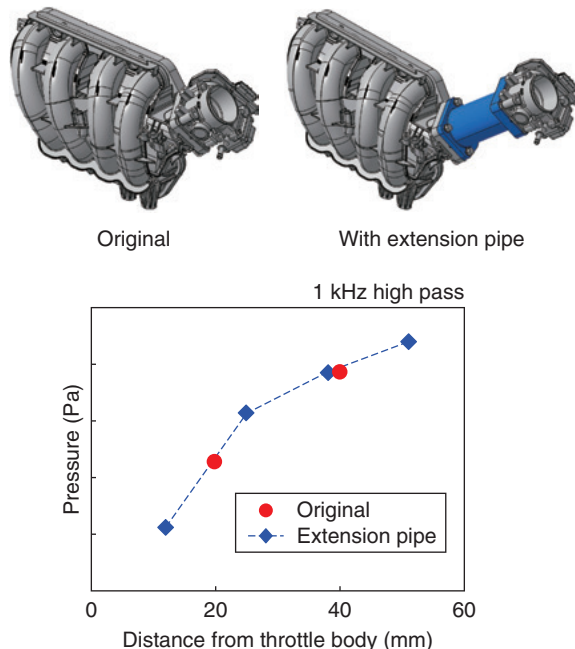


Fig. 6 Comparison of internal pressure

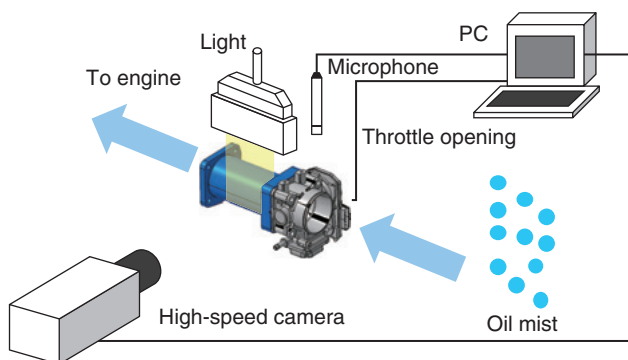


Fig. 7 Visualization system

Table 2 Specifications of light source

Light	250 W metal halide
Power consumption	290 W
Voltage	100 V 50/60 Hz
Filter	UV, IR cut
Shutter	Continuously mechanical type
Cooling	Fan

Image Technology) that employs linearly arrayed optical fibers with a metal halide light as a light source; the system concentrates scattered light by means of a cylindrical lens to form a light sheet.

Table 2 shows the specifications of the special light system, and Fig. 8 shows its configuration.

4.3. Measurement of Internal Pressure Distribution in Pipes

First, measurements of the internal pressure distribution in the pipes were conducted.

Figure 9 shows a cross-section of the extension pipe

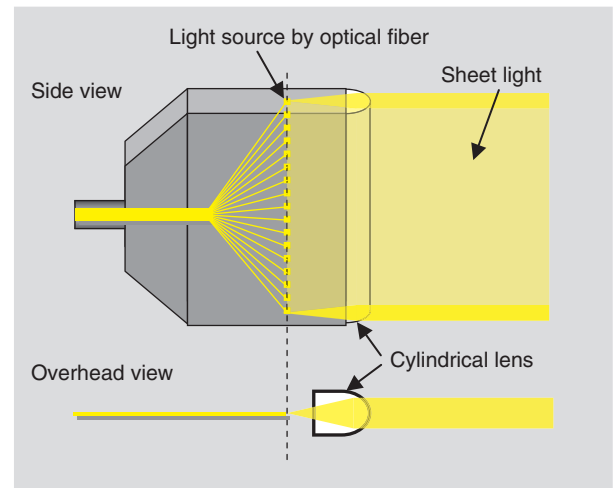


Fig. 8 Special light system

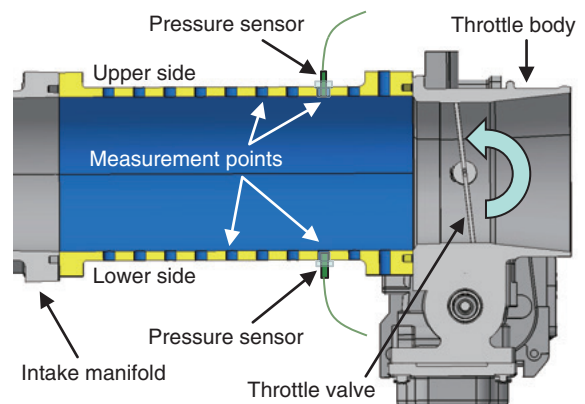


Fig. 9 Measurement extension pipe

used in the measurements. Ten pressure sensors were positioned at uniform intervals on the interior surfaces of the upper and lower sides of the pipe in relation to the throttle, for a total of 20 sensors. As described in Chapter 3, the maximum values during a single racing cycle were measured several tens of times, and expressed as an average.

The results shown in Fig. 10 demonstrate the following:

- (1) The amount of pressure fluctuation close to the throttle body is higher on the lower side of the pipe than on the upper side; and
- (2) Pressure fluctuation reaches its maximum at a distance of between 40 and 90 mm from the throttle body.

These results indicate that when airflow noise is produced, there is intense pressure fluctuation close to the throttle body.

4.4. Visualization of Airflow

The imaging conditions for the high-speed camera were set at a resolution of 448 x 368 pixels and a frame rate of 10000 fps, and moving images were taken of one racing cycle. Synchronized with the camera, sound and degree of throttle opening were measured, and visualization results with values close to average measurements for sound were employed.

Still images were captured from the sequence of moving images where the airflow noise was at its maximum.

Figure 11 shows an image of the measurement planes.

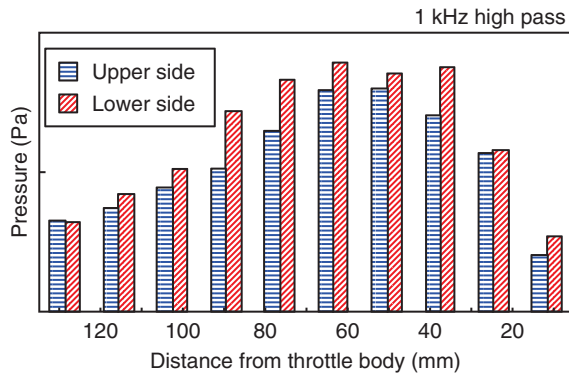


Fig. 10 Inner pressure distribution of extension pipe

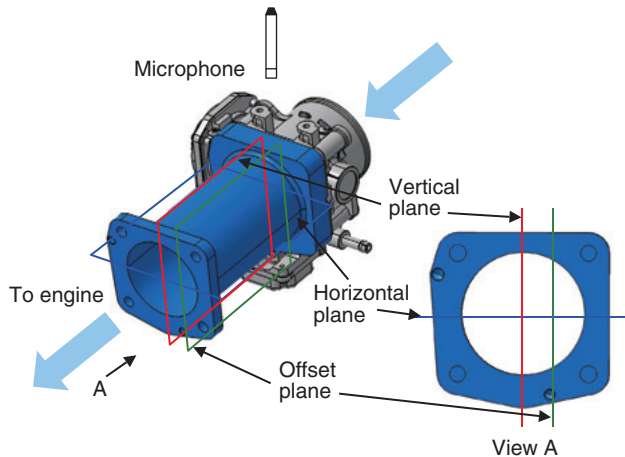


Fig. 11 Visualization planes of air flow

Measurements were conducted in a perpendicular direction to the throttle valve axis, where airflow fluctuation was considered to be most intense, and in a central cross-section in the direction of the axis of the extension pipe (vertical plane). The conditions for isolation of the still images are shown in Fig. 12, and the measurement results are shown in Fig. 13.

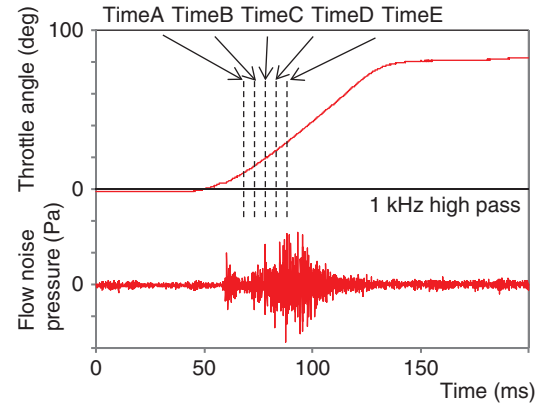


Fig. 12 Condition of visualization of air flow at vertical plane

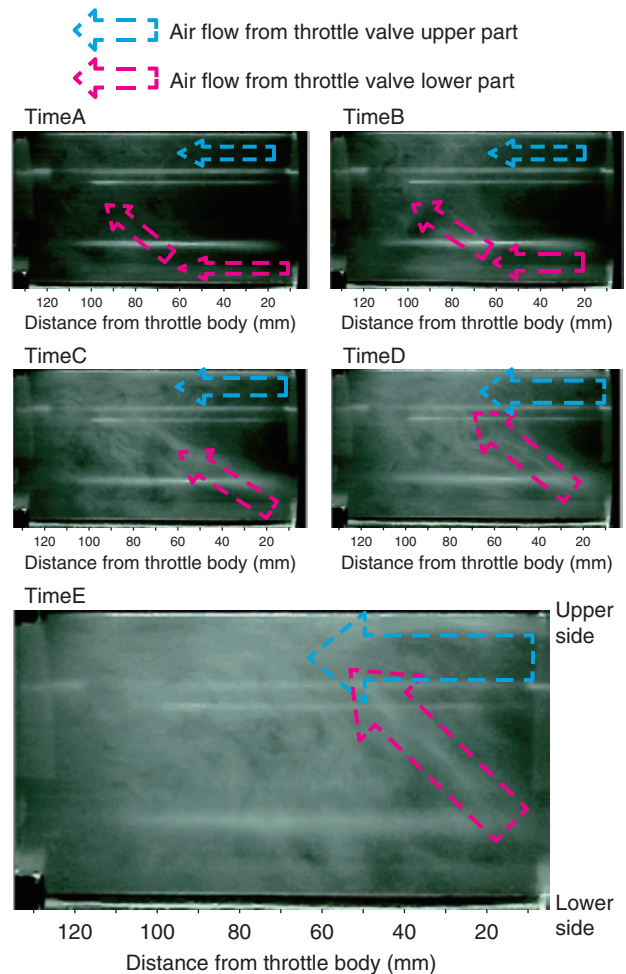


Fig. 13 Visualization of air flow at vertical plane

Figure 14 shows results for visualization with the position of the light offset 20 mm towards the camera in the direction of photography (offset plane). Figure 15 shows results for visualization in a horizontal cross-section through the center of the throttle valve axis (horizontal plane).

Conducting measurements in each of these planes makes it possible to easily obtain a three-dimensional grasp of airflow behavior in the pipes.

The following results were obtained using this method:

- (1) Results for Time A to E in Fig. 13 show that when the throttle valve is opened, the airflow flowing from the lower side of the valve rises diagonally within the pipe at a distance from the throttle body, and during the timing when noise is at its maximum, it collides, at the distance of 40 to 90 mm, with the airflow flowing directly along the surface of the inner wall of the upper side of the pipe and the upper side of the throttle valve;
- (2) For the airflow flowing along the upper side of the throttle valve, there is a high-speed flow flowing directly along the upper side of the interior of the pipe at Time E (Fig. 13), and a high-speed flow flowing diagonally downwards along the surface of the pipe wall (Fig. 14); and
- (3) The airflow flowing from the upper side of the throttle valve in a downwards direction along the inner wall of the pipe (Fig. 14) collides with the airflow from the lower side of the throttle body at a position from 40 mm to 90 mm from the throttle body (Fig. 15).

4.5. Changes in the Behavior of the Airflow when the Throttle Opening Speed is Reduced

Measurements were conducted after reducing the speed of throttle opening in order to measure behavior when the airflow was altered. The results of visualization are shown in Fig. 16, and results for airflow noise are shown in Fig. 17.

Comparison with the results shown in Fig. 13 indicates that the reduction of the speed of opening of the throttle eases the behavior of the inflowing airflows. This results in a reduction in sound pressure.

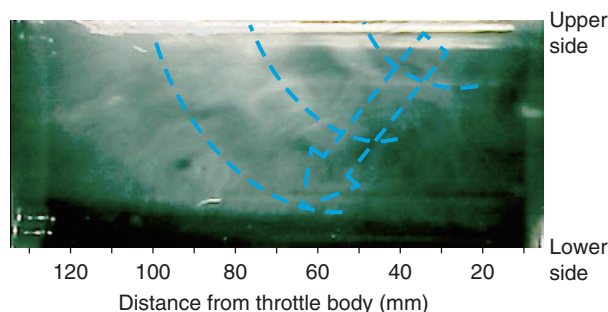


Fig. 14 Visualization of air flow at offset plane

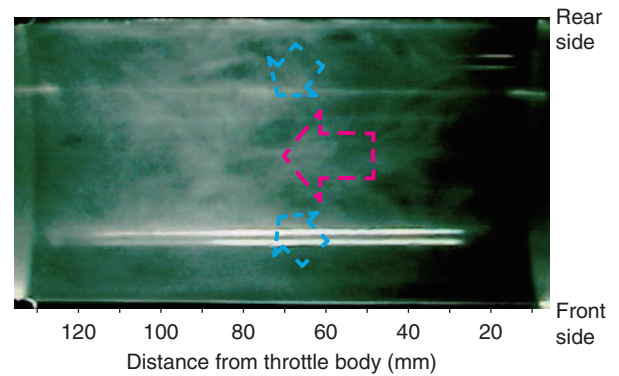


Fig. 15 Visualization of air flow at horizontal plane

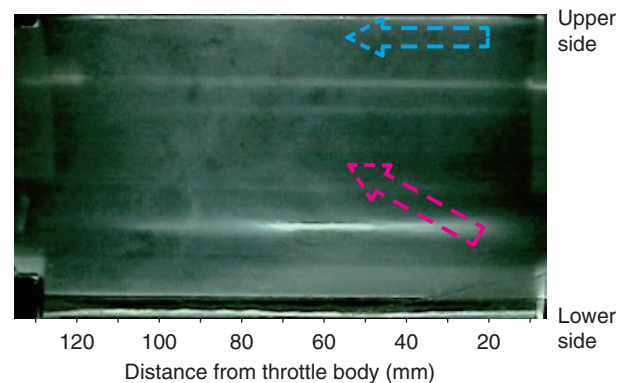


Fig. 16 Visualization of air flow at vertical plane when decreasing throttle speed

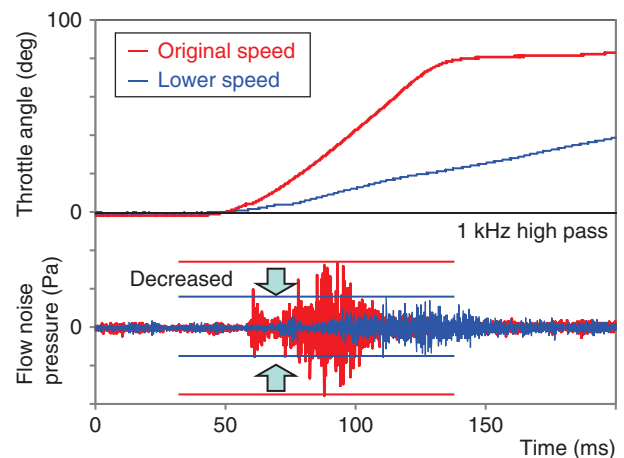


Fig. 17 Comparison of flow noise when decreasing throttle speed

5. Study of Reduction of Airflow Noise and Summary of Visualization

5.1. Study of Reduction of Airflow Noise

Measurements were conducted with a split board dividing the interior into upper and lower sections in order to alleviate the collisions between airflows. The results of

visualization are shown in Fig. 18, and results for airflow noise are shown in Fig. 19.

Comparison with the results shown in Fig. 13 indicates that the use of a split board helps to prevent the collision between the inflowing airflows, thus reducing sound pressure.

5.2. Summary of Visualization of In-pipe Airflow Behavior

Because measurements of in-pipe pressure and the results of visualization show that airflow noise occurs at the same timing as the maximum fluctuation in the pressure inside the pipe occurs in the region in which the airflow flowing diagonally upwards from the lower side of the throttle valve collides with the airflow flowing directly along the wall surface of the upper side of the pipe and the upper side of the throttle valve, it is believed that this collision between airflows is the major factor in the occurrence of airflow noise.

This conjecture is also supported by the fact that measurements conducted when the speed of throttle opening was reduced and measurements conducted when a split board was added showed a reduction in sound pressure.

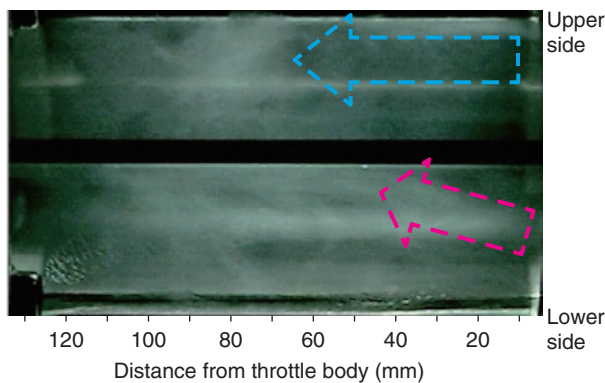


Fig. 18 Visualization of air flow at vertical plane when appending separator

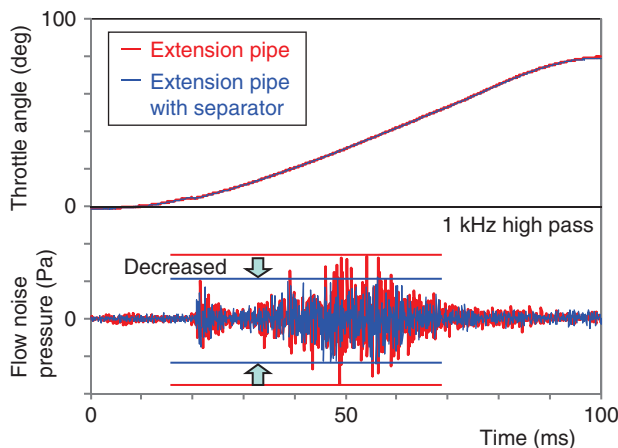


Fig. 19 Comparison of flow noise when appending separator

6. Conclusion

- (1) It was determined that noise occurring when the throttle is rapidly opened is airflow noise originating in airflows inside the intake manifold, radiated from the surface of the intake manifold.
- (2) The phenomenon of collision between high-speed airflows close to the throttle body inside the intake manifold when the engine is in operation was clarified, and was found to contribute to airflow noise.
- (3) The results of measurements of airflow behavior inside the pipe during engine operation conducted with the aid of a transparent pipe and special lighting clarified the factors involved in airflow noise. The method was shown to be an effective method of visualization able to be used in the vehicle development process.

The results of the study discussed in this paper have clarified the behavior of the in-pipe airflows when airflow noise occurs. They have also provided guideposts for ameliorating airflow noise.

References

- (1) Nakase, Y., Kanehara, K., Ohara, K., Yamamoto, K., Miyaji, Y.: The Flow Noise Reduction when Rapidly Opening The Throttle, JSAE Paper, No. 12-00, p. 13-16 (2000)
- (2) Nakase, Y., Kanehara, K., Ohara, K., Yamamoto, K., Miyaji, Y.: Flow Noise Reduction upon Quick Opening the Throttle, SAE Paper, 2001-01-1429 (2001)
- (3) JP, 3337630, B (1997)

Author



Jin YAMAGISHI



Takeshi WAKAMATSU

Battery Control Technology of Li-ion Battery System for HEV

Naoki MARUNO*

Satoru KOMODA*

Yoshikazu NISHIDA*

ABSTRACT

The use of high-output lithium-ion batteries (LIB) is expected to enhance the vehicle performance of hybrid electric vehicles (HEV). However, LIBs are required to withstand long-term use when mounted in HEV.

This paper reports on a battery control system developed to help achieve LIB system durability. The use of an overcharge prevention system based on reliable voltage detection and a deterioration control system based on efficient charge state control enabled enhancement of durability while obtaining the maximum battery performance.

1. Introduction

Recently, various proposals have been made from a wide range of industrial sectors for the realization of a global low-carbon society. One such proposal is the further enhancement of hybrid electric vehicle (HEV) efficiency and popularization of HEV as a means of achieving higher automobile fuel efficiency. One device expected to help enhance HEV performance is the lithium-ion battery (LIB). The reason is that LIBs feature higher output and are lighter in weight compared to nickel metal hydride (Ni-MH) batteries.

Honda developed a LIB system for the 2011 model year CIVIC Hybrid. This system consists of 40 rectangular battery cells and is used for compact and lightweight parallel hybrid systems^{(1), (2)}. The motor is located between the engine and the continuously variable transmission (CVT), and assists running and performs regeneration when decelerating. Figure 1 shows the structure of the parallel hybrid system. The use of LIBs increased the assisting and regenerating outputs by 33% from the previous 15 kW to 20 kW, and further enhanced fuel efficiency. The battery box is 30% lighter and has 20% less volume than the previous model, and is contained in the Intelligent Power Unit (IPU). This IPU has approximately the same volume as the previous model but with higher system output. Figure 2 shows the structure and dimensions of the developed IPU.

In order to mount LIBs in vehicles, a system that can reliably prevent overcharging and deterioration control that extends battery life are needed to withstand long-term use. This paper reports on the system development.

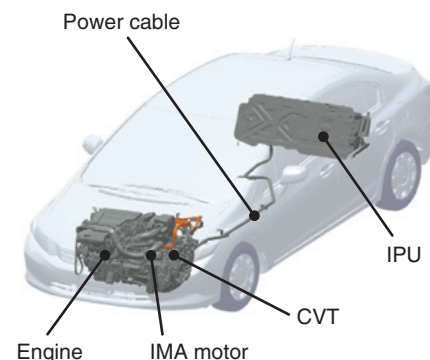


Fig. 1 Construction of Honda hybrid system

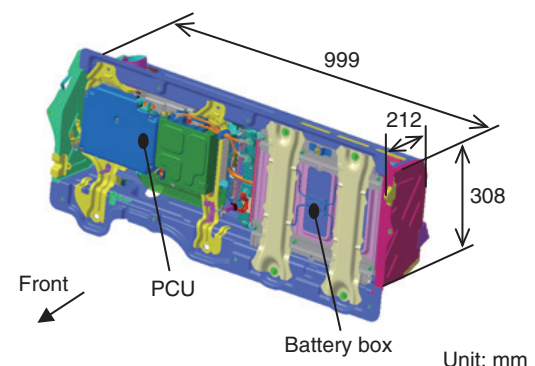


Fig. 2 Construction of IPU

* Automobile R&D Center

Table 1 Comparison of battery box specifications

	Developed	Previous
Voltage/Number of cells	144 (V)/40	158 (V)/132
Weight	21.7 (kg)	31.3 (kg)
Volume (Battery module)	14.4 (L)	15.4 (L)
Maximum output	20 (kW)	15 (kW)

2. System Configuration

2.1. Battery Box Specifications

Table 1 lists the specifications of the developed battery box. The use of LIBs enabled a 33% increase in output with weight reduction of approximately 10 kg. The battery box consists of four modules, each of which contains ten cells.

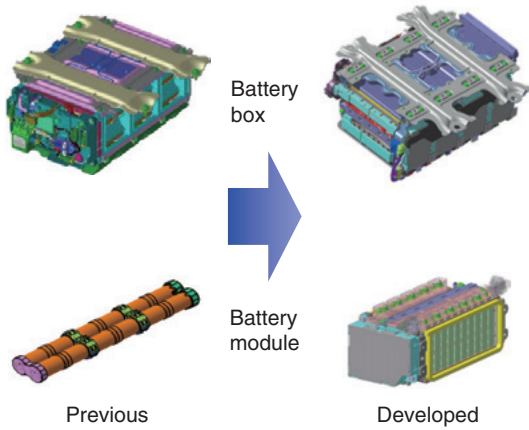


Fig. 3 Comparison of battery box and module construction

Figure 3 shows the battery box and module structures, and Fig. 4 shows the battery system diagram. The voltages of all cells are constantly monitored by voltage sensors and controlled by a battery ECU. In addition, the box has a mechanical switch and fuse at the electrically neutral point, and a contactor in the high-voltage output block. When the battery ECU detects a battery error, it operates the contactor to cut off the high voltage.

2.2. Battery Cell Specifications

Figure 5 shows the cell structure, and Table 2 lists the cell specifications. The criteria for selecting the battery materials were high thermal stability and high durability.

Figure 6 shows verification results obtained from the DSC (differential scanning calorimetry)⁽⁴⁾ tests for the thermal stability of different positive electrode materials. The developed battery uses ternary materials for the positive electrode, which gives it high thermal stability specifications.

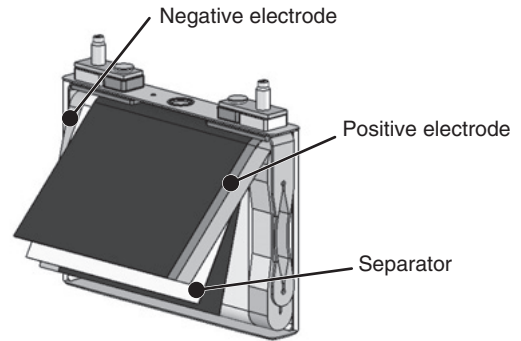


Fig. 5 Construction of battery cell

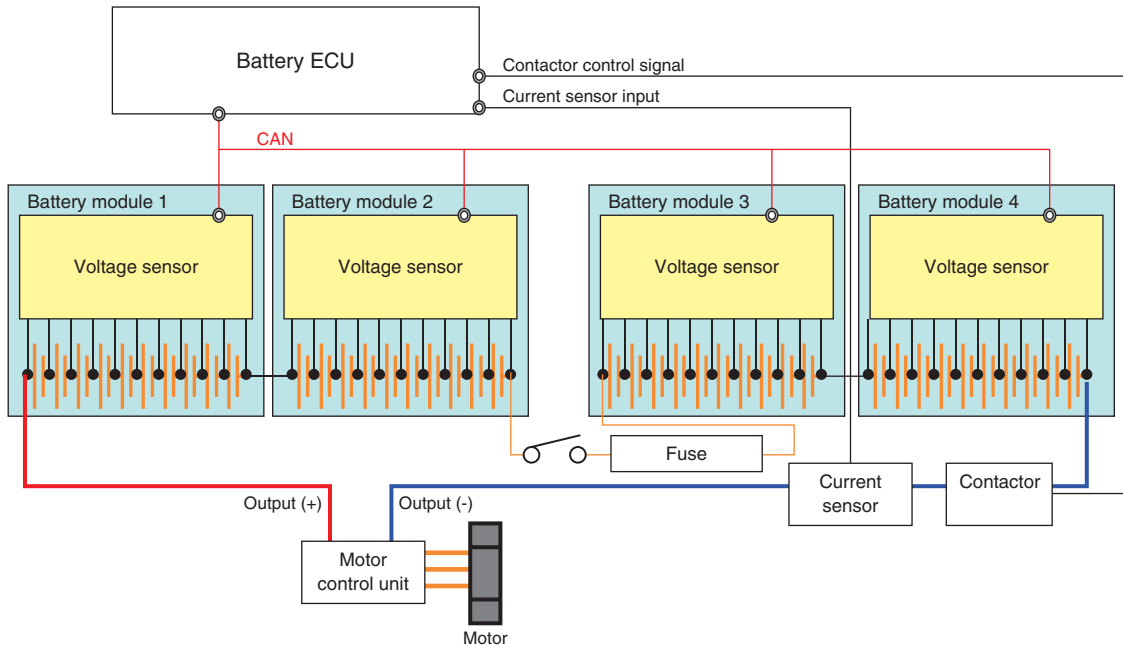


Fig. 4 System diagram of developed battery

Table 2 Battery cell specifications

Item	Description
Voltage	3.6 (V)
Volume	0.136 (L)
Weight	279 (g)
Positive electrode	LiNiCoMnO ₂
Negative electrode	Hard carbon
Case material	A3003 ⁽³⁾

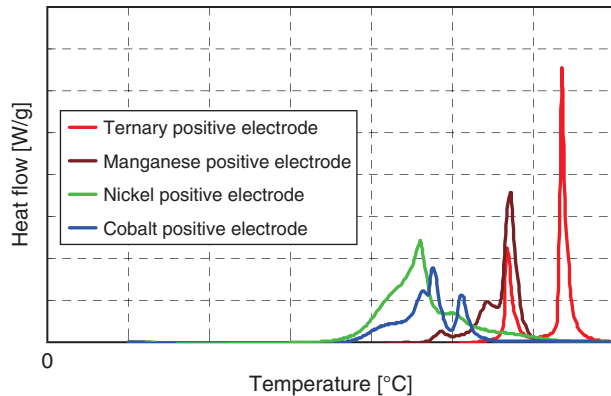


Fig. 6 Comparison of thermal stability among different positive electrode materials

3. Technology to Provide Durability

3.1. Overcharge Prevention System

Figure 7 shows the battery voltage behavior for differing states of charge (SOC). The Ni-MH battery voltage characteristics are shown on the left side, and the LIB voltage characteristics are shown on the right side. The Ni-MH battery charging efficiency drops in the high SOC range, and there is an SOC range in which the voltage does not rise even when charging is performed. On the other hand, LIBs have characteristics such that the voltage rises linearly with charging time even at high SOC. In addition, the cell voltage changes proportionately to the input charging current value, so SOC can be detected based on the cell voltage and current values. In order to reliably detect overcharging, the developed system detects the overcharge state using only the voltage value of the open circuit voltage (OCV) when SOC reaches its critical value.

To increase the voltage value detection accuracy, the Ni-MH battery performs voltage detection in 12-cell units as shown in Fig. 7. However, the LIB system has a cell voltage sensor for each cell to monitor each cell voltage value. The system also connects two voltage detection circuits with identical performance in parallel for each single cell as shown in Fig. 8, and constantly monitors and compares the values from these circuits to detect voltage detection circuit failures based on the voltage difference between the two circuits as shown in Fig. 9. In addition,

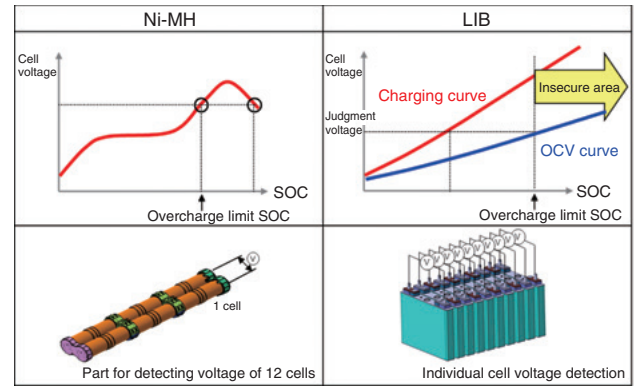


Fig. 7 SOC characteristics and voltage monitoring unit

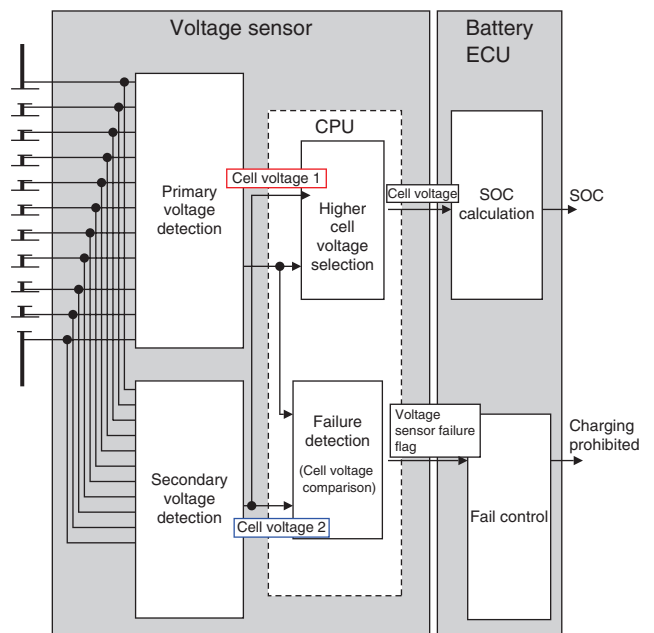


Fig. 8 Diagram of cell voltage sensor system

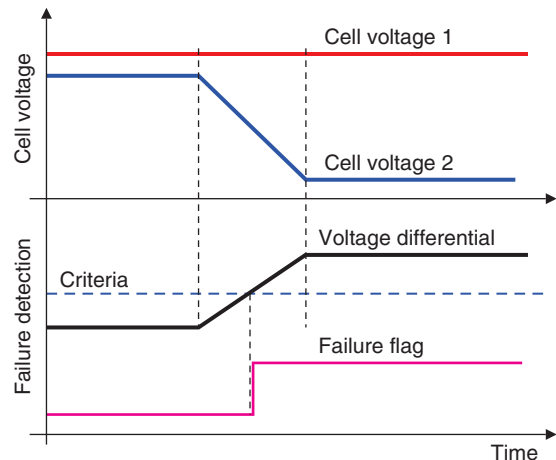


Fig. 9 Detection of cell voltage sensor failure

overcharge prevention reliability is further increased by comparing the voltage values measured by the two voltage detection circuits and selecting the higher value of the voltages.

3.2. Battery State Estimation and Deterioration Control

As batteries deteriorate, the usable capacity and output drop, so to provide the battery performance required by vehicles, it is necessary to accurately understand the SOC and cell capacity relative to battery deterioration. An adaptive battery state estimation system was constructed as a calculation method for this purpose. This system continuously estimates the battery internal resistance based on the voltage sensor, current sensor and temperature sensor information input to the battery ECU, and calculates the battery SOC and cell capacity.

The battery state estimation system consists of a real-time internal resistance estimator, an OCV observer, an SOC calculation block, and a battery capacity estimator (Fig. 10). The OCV observer continuously estimates the OCV based on the battery internal resistance value estimated by the internal resistance estimator. The SOC calculation block calculates the SOC from the estimated OCV. The battery capacity estimator continuously processes the change in the OCV and the change in the electric charge calculated from the current sensor information, estimates the battery capacity, and monitors the deterioration state (Fig. 11).

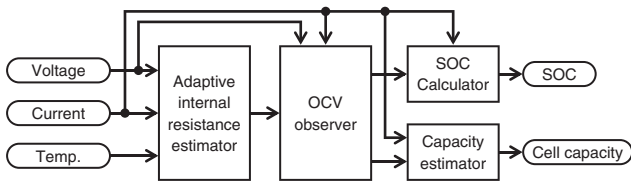


Fig. 10 Composition of adaptive battery-state estimation and control system

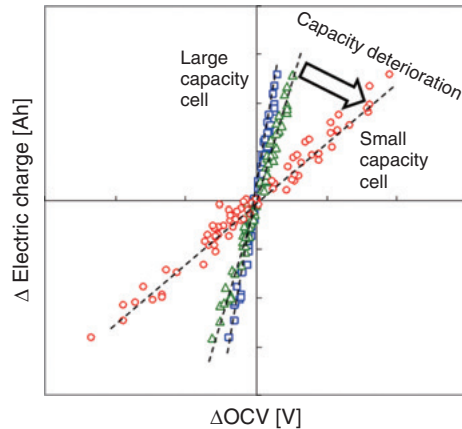


Fig. 11 XY plots showing characteristic change by battery capacity deterioration

Figure 12 shows comparative verification results between estimation results for SOC and battery capacity using the developed system and measurement results. This system enabled accurate estimation of the SOC and accurate calculation of cell capacity at the initial state as well as after battery deterioration.

Figure 13 shows the relationship between the battery SOC and durability. As shown in Fig. 13, battery deterioration progresses as the operating SOC increases. Based on this characteristic, it can be expected that battery life can be increased by performing control to achieve a lower SOC. However, there is the issue that, depending on the battery deterioration state, the energy and driving force required by vehicles cannot be secured at a low

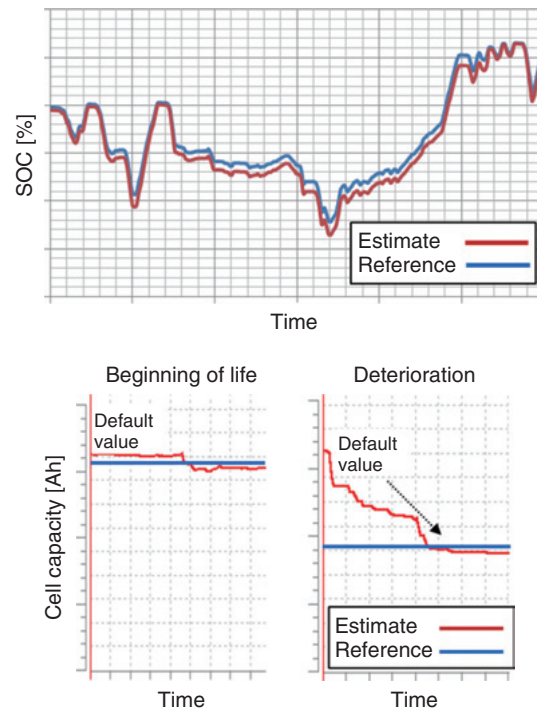


Fig. 12 Time charts of SOC calculation (upper) and battery capacity estimation (lower)

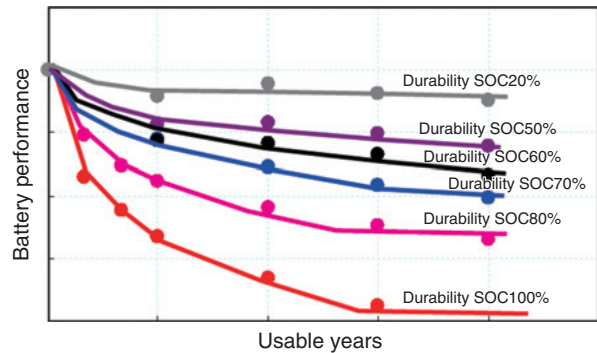


Fig. 13 Cell deterioration characteristics at different SOC

SOC. Therefore, the developed battery state estimation technology was used to construct the ideal battery control system.

In the initial state, when the battery capacity and output are sufficient, the SOC required for operation can be obtained without performing control to boost the SOC. When the battery has deteriorated after long-term use and a drop in the battery energy is detected, the battery performance required by vehicles is provided by shifting the SOC to the high SOC side as necessary. This method achieves the battery performance and battery durability required by vehicles (Fig. 14). This SOC control system suppressed the drop in performance as shown in Fig. 15, and extended the service life of the battery.

3.3. SOC Control between Cells

The developed SOC control system calculates the SOC separately for each cell. To provide durability, control must be performed so that the maximum SOC of each cell does not exceed the control SOC upper limit, and the minimum SOC does not fall below the control SOC lower limit.

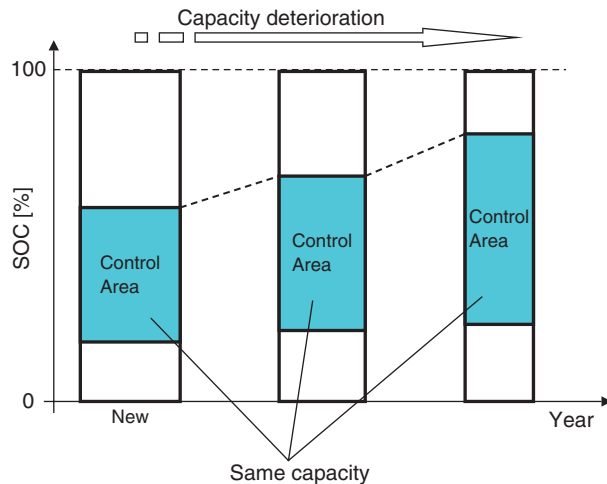


Fig. 14 Battery capacity deterioration

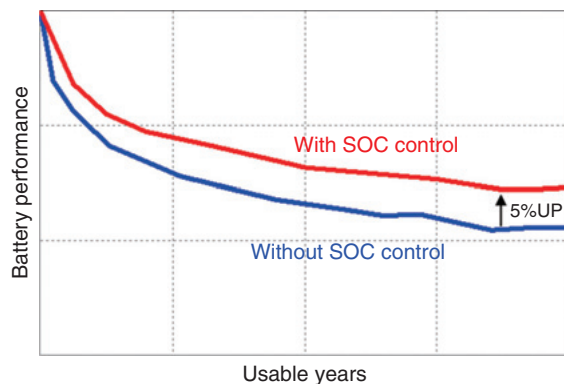


Fig. 15 Effect of deterioration control

When the SOC differs between cells due to the service conditions, as shown in the top part of Fig. 16, the upper and lower SOC limits of different cells restrict the available capacity width, even though the capacity of each cell has not dropped. This results in a deteriorated state with reduced effective charging capacity for a battery pack. To suppress this, the individual discharge circuits provided for each cell are operated to discharge cells with a high SOC in order to level the SOC and restore the chargeable capacity width for a battery pack (Fig. 16 bottom). This technology enables effective use of the battery pack performance.

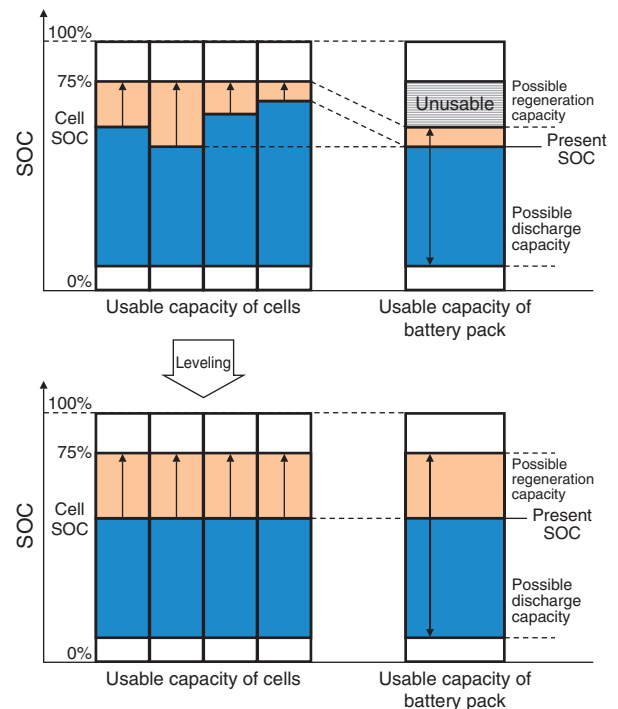


Fig. 16 Cell capacity leveling system

4. Conclusion

The following battery control technology was developed for LIB systems mounted in HEV.

- (1) To prevent overcharging, a system was constructed that can reliably detect overcharging. This system has a cell voltage sensor in each cell and also detects voltage sensor failure.
- (2) Technology was established that can simultaneously achieve the energy and driving force required by vehicles and durability. This technology constantly calculates the battery state (SOC and capacity of each cell) and performs control to maintain the ideal SOC.
- (3) A discharge circuit is provided for each cell, and when the SOC differs between cells, discharge circuits are operated to suppress this difference in SOC to enable effective use of the battery performance.

References

- (1) Komoda, S., Nishida, Y., Maruno, N.: Development of Lithium-ion Battery System for Hybrid System, JSAE Annual Congress Proceedings, No. 107-11, p. 1-4 (2011)
- (2) Takahashi, Y., Inoue, K., Matsuoka, T: Battery cooling technology for Hybrid Vehicles, JSAE Annual Congress Proceedings, No. 68-06, p. 11-14 (2006)
- (3) ISO 2107: 1983, Aluminum, magnesium and their alloys-Temper designations
- (4) JIS K0129: 1994, General Rule of Thermal Analysis

■ Author ■



Naoki MARUNO



Satoru KOMODA



Yoshikazu NISHIDA

Design of Experiments on Steering Shimmy in Hydraulic versus Electric Power Steering Systems

Michael HULLIHAN*

ABSTRACT

Research was conducted on automotive power steering systems at a component level to examine system sensitivity to an oscillatory loading condition. This series of experiments was aimed at quantifying system response of hydraulic power steering (HPS) and electric power steering (EPS) to objectively compare vibration attenuation capability of the two systems.

Steering shimmy is caused by an imbalance in a wheel/tire rotational assembly. The energy of this phenomenon is usually too great for a HPS system to completely attenuate while an EPS system is commonly capable of absorbing most of the vibration. With a HPS system, vibration can be partially attenuated through steering system parameter variation such as increasing friction or inertia. In an EPS system, the motor inertia, programming control, and gear reduction make it comparatively simple to attenuate the vibration before it reaches the steering wheel. This ability of an EPS system to partially or completely attenuate vibration caused by other chassis components has yet to be quantified. Furthermore, it has yet to be objectively compared with HPS and is thus the primary focus of this research.

The test results confirm that increased inertia and friction have a dramatic effect on steering vibration regardless of application. There is a friction threshold that can be used in partially mitigating some small vibrations such as tire imbalance. Furthermore, it shows that in an EPS system, the motor inertia and electric motor gear reduction are beneficial in stopping vibration from reaching the steering wheel. Through this research, we have shown that future developments are better equipped to attenuate steering shimmy per the application of an EPS system.

1. Introduction

In the manufacturing of tires and wheels, it is typical for an imbalance to exist about the rotational axis and/or rolling center line of the component. While the modern process of wheel/tire balancing is largely effective, it is neither permanent nor impervious to the inevitable effects of year-round vehicle operation. Dynamic wheel imbalance can result from uneven tire wear, dislodged wheel balance weights, and build up of mud and snow within the wheel. This imbalance has no effect on the continued safe operation of the wheel/tire assembly, but does transmit an oscillatory load into the chassis that in turn can create objectionable levels of vibration throughout the passenger compartment. When this wheel/tire imbalance causes vibration at the steering wheel, it is referred to as shimmy.

Vehicle manufacturers go to great lengths to attenuate such vibrations to improve the marketability of the vehicle. With the recent widespread application of EPS, this capability has grown because of the unique capabilities of that system. Past research on steering rotational

vibration⁽¹⁾⁻⁽³⁾ explored the response of an entire chassis system, including an HPS system, to an oscillatory input load. These studies looked into system correlation and prediction as well as component level factor analysis. The research described in this paper expands on that topic by exploring HPS versus EPS component level parameter variation and the benefits of EPS with regard to attenuation of steering shimmy.

2. Design of Experiments on Steering Shimmy

Steering shimmy is typical in the event of a wheel/tire imbalance because these assemblies are directly connected as shown in Fig. 1. However, because steering systems inherently possess characteristics that can attenuate vibration, it is possible to tune-out this reaction at least partially.

System characteristics such as friction do not exist solely for vibration attenuation, so a balance must be reached. In order to reach that balance, every system characteristic affected by friction must be considered.

* HONDA R&D AMERICAS, INC.

To that end, it is prudent to first look at the system at a fundamental level.

The free body diagram of the system (Fig. 2) shows why this phenomenon occurs. An oscillatory load parallel to the ground plane causes a rotational vibration of the wheel/tire assembly about the kingpin axis. This vibration is transmitted to the steering system by the now axial vibration into the gearbox (G/box). This forcing function then acts on a seemingly simple mass, spring, and damper system, although the “damper” is actually a by-product of the power steering fluid in the gearbox cylinder. In an EPS system, the damper is replaced by a gear reduction and motor inertia, although there is damping designed into the software. For the purposes of this fundamental analysis, this section focuses on HPS only.

To more clearly show the areas of focus for attenuation of this vibration, this system can now be characterized by its equations of motion, Eq. 1 and Eq. 2:

$$F_{input} \left(\frac{r_{input}}{r_{gbox}} \right) + c_{gbox} v_{rack} + (-F_f) + \frac{k_{col} \theta_{col}}{r_{col}} + \frac{I_{str} \alpha_{str}}{r_{str}} = F_{response} \quad (1)$$

Where F_{input} is described by the equation for centripetal force:

$$F_{input} = m_{imbalance} r_{imbalance} \left(\frac{v_{vehicle}}{r_{tire,dynamic}} \right)^2 \quad (2)$$

The term $r_{tire,dynamic}$ is the dynamic loaded radius of the tire at the vehicle speed, $v_{vehicle}$, of interest. Shimmy happens when there is a net imbalance described by the mass of that

imbalance, $m_{imbalance}$, and it's radius from the wheel center, $r_{imbalance}$. When there is shimmy it causes some oscillatory steering rack velocity, v_{rack} . The vibration is then felt in the steering wheel as an oscillatory angular acceleration, α_{str} at some radius, r_{str} . The fourth term in Eq. 1 is for the torsional spring rate of the steering column where r_{col} is the outer radius. Since the displacement terms are determined by the input loading and the gearbox damping (c_{gbox}) is fixed, the tuning variables are friction (F_f), torsional spring rate (k_{col}), and rotational inertia (I_{str}). Within the realm of tuning friction we can change normal force and rack friction material to change the coefficient, μ . These are the areas of focus in this design of experiments.

This research treated rack, housing, and mounting stiffness as rigid. This is a safe assumption because the relative stiffness to that of the components described in the above equations of motion are several orders of magnitude higher.

Up until now, shimmy has been attenuated to an acceptable level by trial and error, but the full range of this solution has not yet been quantified. Commonly this attenuation is achieved by increased friction. So in addition to quantifying friction changes, we will see the empirical effects of changes to the other components as well.

In this research, the steering system was isolated on bench and studied at a component level to quantify the steering system's response to varying inputs and component parameter variation. Through this study we will demonstrate the similarities and differences of HPS and EPS steering response, and finally highlight the advantages for future developments with the application of EPS systems.

3. Test Method

3.1. Test Setup

Figure 3 shows a cross section of a steering G/box rack-and-pinion interface.

In order to isolate the steering system on-bench, a test fixture had to be created to exactly mimic the setup on

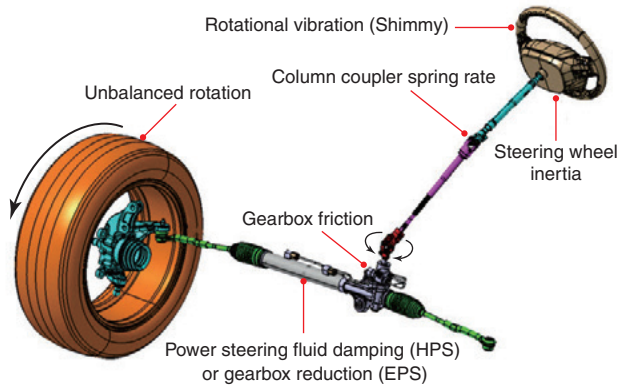


Fig. 1 Steering system model of shimmy condition

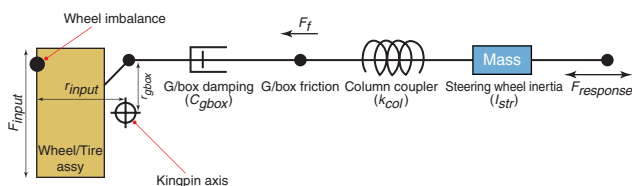


Fig. 2 Steering shimmy free body diagram

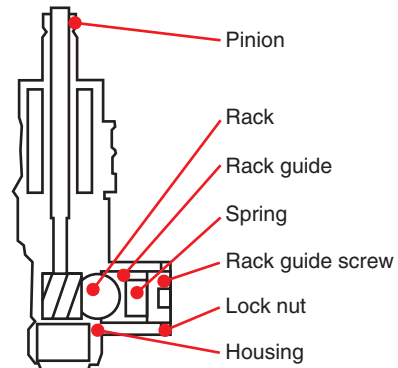


Fig. 3 Cross-section of a rack-and-pinion interface

a vehicle. Once built, the test setup would be able to be mounted alongside a power steering pump tester so a HPS system can function just as it would in vehicle at speeds that shimmy is most likely to occur. The required equipment to mimic vehicle speed with EPS is mobile and therefore not a concern in the fixture's design. Figure 4 shows the test fixture with a steering system mounted to it.

To capture the input measurements of force and acceleration, a force transducer and accelerometer are mounted at the interface of the shaker and G/box. The response at the steering wheel is captured via the accelerometer mounted at the nine o'clock position. The rotational vibration at the pinion was measured to look at potential damping provided by the steering column. As shown in Fig. 4, several other points were measured for vibration response to make sure that results were not being skewed by vibration energy being lost in shaking the test fixture itself.

3.2. Test Setup Confirmation

3.2.1. HPS correlation of on-vehicle to on-bench data

Before this testing began, measurements of tie rod load and steering wheel vibration were taken during another similar steering vibration phenomenon, brake judder. Though this mode has greater amplitude than shimmy, it provides a legitimate reference point for validating the test setup since resonant frequency is independent of amplitude. Figure 5 shows the comparison between this data and the test setup for an HPS system. A steering wheel vibration response of 1.9 m/s^2 was measured with a 750 N input force at a frequency of about 15 Hz on-bench and in-vehicle. This test setup proved to be a good model of the in-vehicle situation.

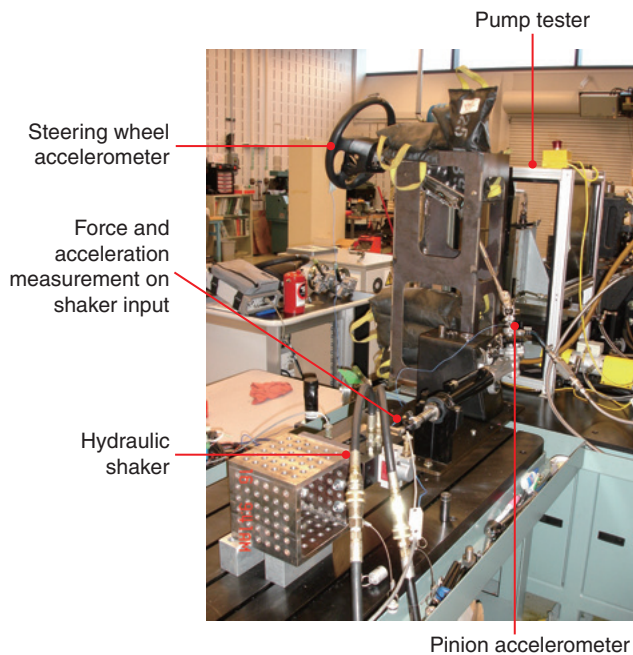


Fig. 4 Steering vibration sensitivity test bench setup

3.2.2. EPS correlation of on-vehicle to on-bench data

A similar trend was found with an EPS system. As shown in Fig. 6, a natural frequency peak exists at 10 Hz on-bench as well as on-vehicle. Though the HPS and EPS

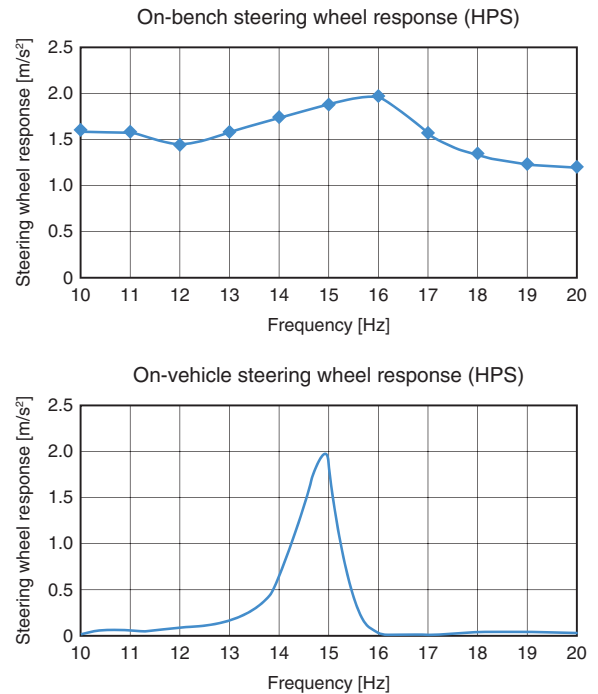


Fig. 5 HPS vehicle versus bench data

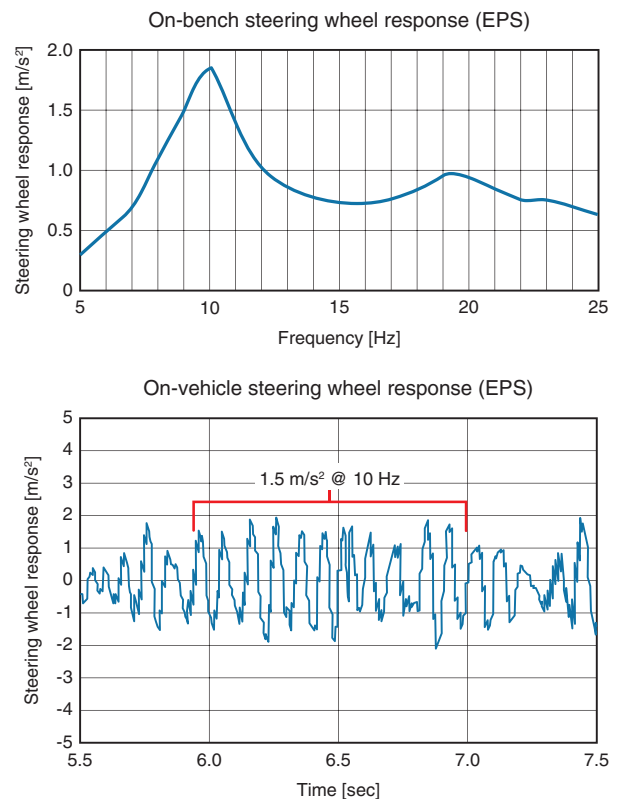


Fig. 6 EPS vehicle versus bench data

correlations were done at different load levels (750 N and 250 N respectively), once again the amplitude of a sinusoidal forcing function is not related to the response frequency, only the response amplitude. Therefore this has no effect on system resonant response and offers a sound method for test setup validation.

3.3. Test Procedure

In the HPS and EPS systems, it was initially important to find where the natural frequencies of the systems were because they would be the most likely to occur on-vehicle. After resonate frequencies are determined, testing can proceed to other frequencies if necessary. However, since the frequency peaks found on-bench were the same as those found on-vehicle, it was immediately apparent that this needed to be the area of focus. As shown in Figs. 5-6, the HPS and EPS systems being examined in this research have peak responses around 15 Hz and 10 Hz, respectively. After discovering the systems' behavior in the frequency domain, testing from then on was in the form of a force sweep. This was done for two reasons. First, it allowed for observing a particular setting's small and large input behavior all at once. Secondly, if there is a resonance frequency shift, it is important to see that effect at the natural frequency of the vehicle chassis, that is, 10 Hz for EPS or 15 Hz for HPS. Therefore, a force sweep was run while changing each variable component of the system in the following design of experiments:

- (1) G/box Rack Guide Screw Torque
- (2) G/box Rack Guide Material Friction

- (3) Steering Column Coupler Rate
- (4) Steering Wheel Rotational Inertia
- (5) P/S assist On and Off
- (6) Simulated Vehicle Speed

With regard to variable (6), in order to simulate different vehicle speeds, the P/S pump speed was varied to change flow rate and pressure. Later on during testing it was found that there was no change in response from this variation, so these results are omitted. This response is expected because the system typically does not provide power assist on-center, where this event occurs.

For EPS, software was used to send the ECU an ignition signal and CANalyzer software was used to send the ECU a speed signal. It is important to simulate vehicle speed because at any given speed, there are unique power assist characteristics that have been intentionally tuned into the system; therefore, system behavior will vary from a static to dynamic condition. This is discussed in more detail in section 4.

After the validity of the test setup was confirmed, another sampling of on-vehicle steering shimmy data was referenced to find the proper load range for testing that phenomenon. As shown in Fig. 7, shimmy occurs at around 250 N at the tie rod. Therefore this would be our primary area of focus.

4. HPS Parameter Variation

4.1. Friction

When the setup was verified and ready, testing began by varying the aforementioned HPS variables (1-5) to find the more sensitive areas in the system. Each setup was run through a force sweep to study response at all input levels. First, the rack guide friction variables (1) and (2) were investigated. Figure 8 shows that varying rack guide screw torque changes the steering response. This was tested by tightening or backing off the rack guide screw by incremental amounts, thereby changing the rack guide's normal force and therefore the friction. Changing the rack guide material (2) has the same effect on system response.

This result is because of the threshold of static and

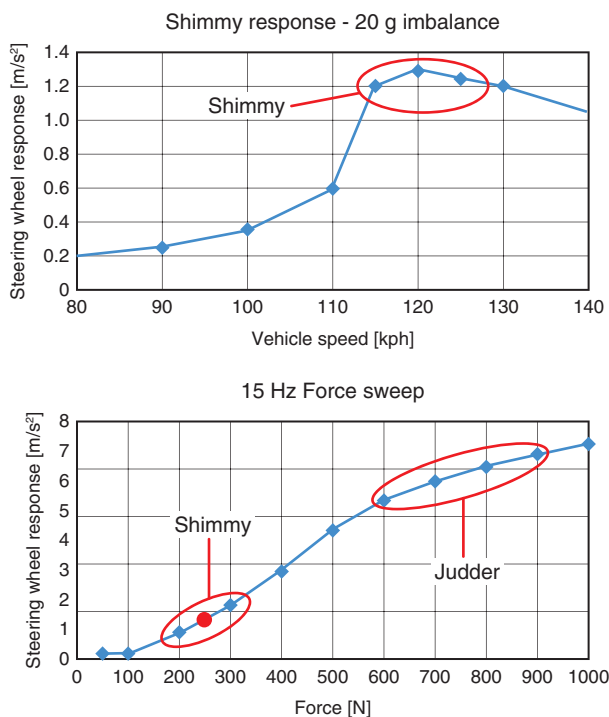


Fig. 7 Vehicle versus bench shimmy data

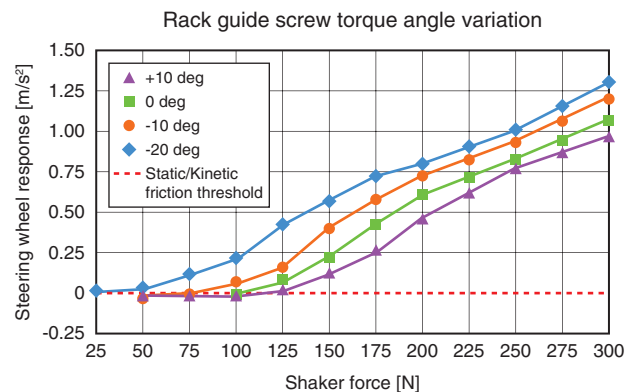


Fig. 8 G/box rack guide screw torque with HPS

kinetic friction between the steering rack and the rack guide. As that friction is increased via increased normal force, the threshold at which the rack transitions into dynamic oscillation increases. If this point is increased high enough, shimmy can be completely attenuated. However, even if that threshold cannot be moved above the shimmy load region for a particular vehicle, it still has an effect of reducing the vibration response due to the increased kinetic friction.

4.2. Spring Rate

Next, the angular spring rate of the steering column coupler was varied to determine its effect on vibration attenuation. This potential solution had a notably different effect than a change in rack guide friction. As shown in Fig. 9, it had a noticeable effect at higher vibration levels, but had no discernable trend in the oscillatory loading range where shimmy happens.

The reason for this response is due to the nature of changing spring rates in a mass-spring-damper system. Given the frequency sweep run early on in this testing, it is clear that there exists a system natural frequency for this HPS system around 15 Hz. It is a physical property of harmonic motion that if the spring rate of a singular component of a system changes, the natural frequency of the entire system also changes. However, given the relatively small spring rate variation possible with this

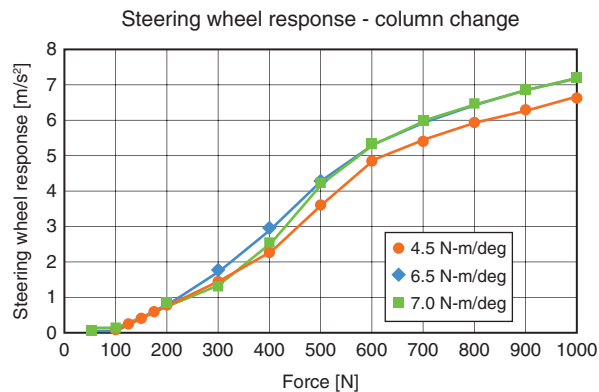


Fig. 9 Steering column coupler rate change

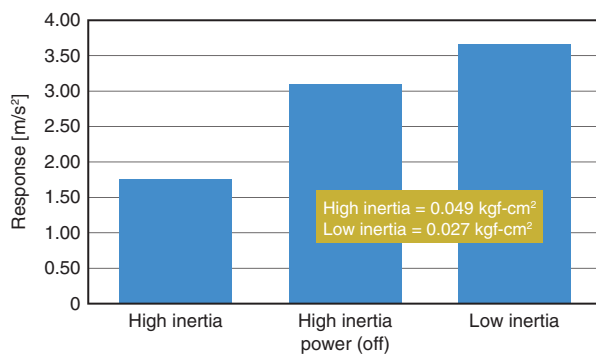


Fig. 10 Steering wheel inertia and power off

component, it is certain that any shift in resonant frequency will be too small to mitigate the rotational vibration at the steering wheel. This is shown at the much higher loads, where any frequency shift at all only resulted in a small change in response as opposed to meaningful attenuation.

4.3. Inertia

Moving further up the steering system, variation in steering wheel inertia was tested. Figure 10 shows the effects of this variation under a 250 N oscillatory load at 15 Hz. By cutting the steering wheel inertia in half, the vibration response is doubled.

This result is expected considering the basic equation $T = I\alpha$, which relates torque, T , to an object's moment of inertia, I , and angular acceleration, α . Since rotational vibration is a form of angular acceleration, at a given pinion torque, the response is directly dependent upon the inertia of the system. If the system has half the inertia, then it should have twice the angular acceleration for a given torque.

Another test involved checking the response with the P/S pump completely turned off. Although this is a scenario that is unlikely, it was tested for further understanding. As shown in Fig. 10, the damping of the system is enhanced with P/S fluid continuously circulating through the system.

5. EPS Parameter Variation

In a rack EPS system, a radial motor drives a ball screw mechanism that in turn provides power steering assist by pushing on the rack in the desired direction. Spatial limitations in an engine room require the motor to be relatively small and consequently, a sizeable gear ratio is needed to provide adequate assist. This gear ratio and rotational inertia of the motor have effects on system response to external inputs.

5.1. EPS versus HPS

5.1.1. Similarities

When varying the aforementioned mechanical variables, the EPS steering system response was predictably similar to HPS. For example, as shown in Fig. 11, changes in friction

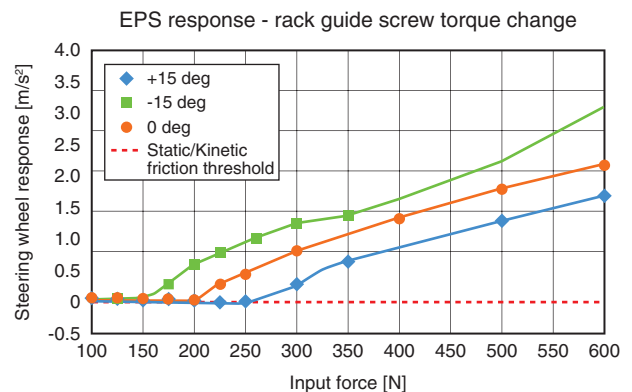


Fig. 11 G/Box rack guide screw torque with EPS

can cause the static/kinetic friction threshold to shift just as it did in the HPS setup. However, in comparison to Fig. 8, the EPS friction threshold and dynamic response is better attenuated than HPS. As mentioned before, the sizeable gear reduction provides a great deal of disturbance rejection to inputs such as vibration or kickback.

With changing variables (3) and (5), the EPS system response is virtually identical to HPS. This is an expected result. The principals that governed the response to a column coupler rate change on HPS still apply in an EPS system. Also, as mentioned, EPS inherently possesses some level of damping as a result of the motor programming; therefore, it would respond just as an HPS system with P/S fluid flowing through it.

5.1.2. Differences

The effects of changing steering wheel inertia did have a unique result. As shown in Fig. 12, varying steering wheel inertia had no effect on system response in the shimmy load range. This is due to the relatively large gear reduction and motor inertia in a rack assist EPS system. The changes in inertia offered by different steering wheels are small by comparison although they did begin to have an effect at higher loads. The reasoning behind this result is somewhat similar to that of changing column coupler rates. Removing

the steering wheel airbag had no discernable effect because the component's mass is essentially along the rotational axis.

5.2. Software Programming Variation

EPS system response was also tested by software tuning. In an EPS system, the steering control unit is constantly receiving a vehicle speed signal while the vehicle is in operation. This signal is useful because some steering assist software maps are speed-based, which allows an engineer to, for example, decrease power assist when the vehicle is being driven straight (on-center) as speed increases. Such adjustments can enhance on-center stability.

Using CANalyzer software, the vehicle speed signal was varied and system response to the resonance frequency during a force sweep was recorded. Figure 13 shows the results.

There is no change in system response in the load range of the shimmy phenomenon. There is a pronounced effect at the higher load range, and the dynamic threshold is shifted when the vehicle is in motion versus statically. This shows the need to simulate vehicle speed, but in-motion response is virtually the same at all speeds.

6. Conclusion

In this study, it is shown that the mechanical tools for attenuating steering shimmy include increases to:

- Rack guide friction by increasing
 - Normal force (rack guide screw torque)
 - Material coefficient of friction
- Steering wheel inertia

However, contrary to what one might expect given the properties of system response to forced harmonic motion, this research showed that varying the column coupler rate has little effect on mitigating vibration for a small input. Another interesting point is that a power assist "on" condition is beneficial toward attenuation of steering shimmy. As shown in this research, EPS system response holds similarities with that of HPS. For example, steering shimmy can be attenuated by increasing system friction. However, this increase may not be necessary because of the already higher static/kinetic threshold that exists. There are differences also such as the system response to steering wheel inertia variation. Largely in part to the significant inertia of the EPS motor, changes to steering wheel inertia had no effect on system response in the load range of steering shimmy.

Additionally with EPS, software programming had to be considered. This research showed that the vehicle speed simulation caused system response to change everywhere except in the shimmy region. However, a speed signal did cause the static/kinetic threshold to shift for the resonant frequency of the system.

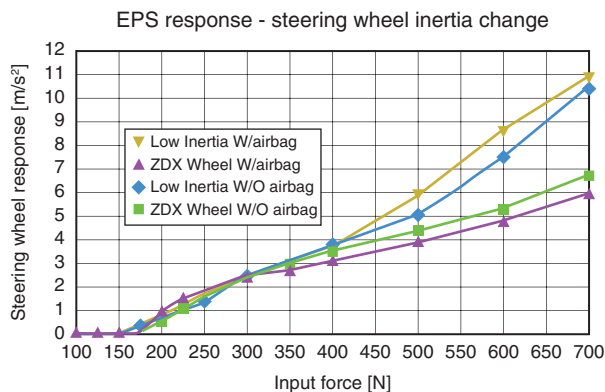


Fig. 12 Steering wheel inertia variation

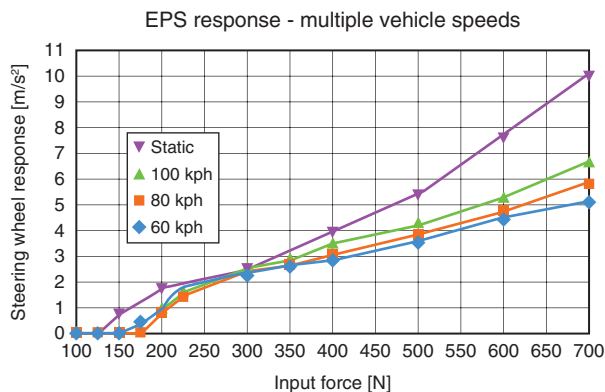


Fig. 13 EPS response under varied vehicle speeds

References

- (1) Yu, J., Nutwell, B., Brickner, B., Johnson, M.: Analysis of Vehicle Chassis Transmissibility of Steering Shimmy and Brake Judder: Mechanism Study and Design of Experiments, SAE Technical Paper, 2007-01-2342 (2007)
- (2) Yu, J.: An Analysis of Fluid-Mechanically-Coupled, Non-Linear Rotational Vibration Sensitivity of Automotive Hydraulic Power Steering Systems, International Mechanical Engineering Congress and Exposition, Anaheim, California, U.S.A. (2004)
- (3) Yu, J., Slattengren, J., Motoyama, K.: Modeling and Factor Analysis of Hydraulic Power Steering Systems for Rotational Steering Vibration by Using MSC. ADAMS, MSC Software Virtual Product Development Conference, Huntington Beach, California, U.S.A. (2004)

■ Author ■



Michael HULLIHAN

“Statistical Tire Model” Identified from Data Group of Multiple Tires

Tomokazu SUZUKI*

Kaoru KUSAKA*

ABSTRACT

A statistical tire model has been developed. This tire simulation model is for use in the feasibility study of performance design at the initial stage of vehicle development. Using this method, existing data sets for the performance and characteristics of multiple tires are statistically modeled. The model is available to estimate values for tire performance and characteristics based on the input values for specifications such as tire size. By contrast with conventional tire models, the statistical tire model is able to estimate the characteristics of multiple tires of differing sizes and specifications using a single model without the measurement and modeling of data for each single tire.

1. Introduction

1.1. Background

The process of setting performance targets at the initial stage of vehicle development is termed performance design. Today, with the greater proportion of CAE in use in vehicle development, it is becoming increasingly important to conduct performance design at the beginning of the development process systematically and with a high degree of accuracy.

The tires are an important element in performance design, having a significant impact on many aspects of the performance and commercial appeal of the completed vehicle, including the chassis layout, handling performance, vehicle stability, brake performance, ride comfort, quietness and fuel efficiency. The multiple characteristics demanded from tires exert a mutual effect on each other, and are sometimes antagonistic. Because of this, it is necessary to realize a more accurate balance between these characteristics.

In addition, major changes in the size of the tires during the development process can have a significant effect on the chassis layout and the overall performance of the vehicle, and it is therefore desirable to decide on the specifications of the tires based on sufficiently accurate and broad-ranging studies at the initial stage of the development process.

1.2. Conventional Tire Model Approach

A variety of tire models are employed in performance design, including experimental models, theoretical models, and finite element models (FEM), a type of theoretical model (Table 1).

Table 1 Tire models for vehicle dynamics simulation

Model type	Application	Tire size effect simulation	Source data	Arbitrary tire size characteristic estimation
Experimental model	Statistical model	Single model for any size	Multi-tire data group	Available
	Magic Formula, etc.	Separate model for each size	Single tire data	Not available
Theoretical model	Brush model SWIFT, FTire etc.	Separate model for each size	Single tire data	Not available
	FEM model	Separate model for each size	Single tire data	Not available

Experimental models are produced by curve-fitting test data using approximation equations. The Magic Formula tire model is a standard example of an experimental model, widely used in applications including mechanism analysis and driving performance simulations. Standard examples of theoretical models include SWIFT and FTire. In theoretical models, the physical phenomena associated with the tires are expressed using theoretical expressions. These models frequently employ empirical formulas and constants obtained by experiment in specific sections. In the case of FEM models, characteristics are determined using CAD models of detailed structures. In many cases, data obtained from tests on actual tires are used in the verification and correlation of the models^{(1), (2)}.

Irrespective of the type of model used, it is standard procedure to model the various characteristics based on test data obtained separately for various conditions, such as tire sizes, inflation pressures, and loads. When these models are used in the performance design of parameters such as

* Automobile R&D Center

handling performance and vehicle stability at the initial stage of vehicle design, and tire size and use conditions are changed in studies of performance, it is necessary to utilize models identified for each respective set of conditions. However, in actual practice it is challenging to prepare tire models for all sizes and conditions in advance, and in addition there are also cases in which tire sizes for which no previous data exists are used, and it is necessary to study vehicle performance on the basis of predictions.

To respond to these issues, for the performance parameters of the greatest importance, in addition to conducting test measurements and modeling the tires, designers substitute tire models with similar characteristics and correct similar models based on estimates. However, in order to increase the precision and efficiency of performance design and expand the scope of comparative study, tire models able to estimate the characteristics of tires of various sizes and specifications without the necessity for measurement and modeling of individual tires are necessary.

2. Overview of Development of New Tire Model

2.1. Concept

In order to resolve the issues discussed above, it would be necessary to be able to estimate the characteristics of tires of varying sizes under varying conditions using a single model. It was considered that a tire model capable of estimating the characteristics of tires of differing sizes and specifications could be obtained by expanding an experimental model identified from data for a single tire into a model identified from a data group encompassing tires of multiple sizes and types (Fig. 1).

2.2. Method of Statistical Modeling

In order to realize a model of this type, a system identification method was used to formulate a single multi-dimensional approximation equation model from a data group encompassing tires of multiple sizes and specifications. System identification is a form of

statistical identification that takes into consideration the conformity to physical laws of system characteristics between input (explanatory variables) and output (objective variables)⁽³⁾. The flow of the procedure is as follows (Fig. 2):

- (1) Projection of model equations based on prior information (existing knowledge);
- (2) Identification of approximation equation constants using test data;
- (3) Verification of accuracy of model by evaluation of error;
- (4) Repeated correction of model until adequate accuracy is obtained; and
- (5) Determination of model by introduction of constants to approximation equations.

This method is widely employed in all modeling. From the perspective of the fact that it models the relationship between input and output using approximation equations, the method is the same as that employed in creating a single-tire experimental model. However, it differs in the fact that it identifies a single model from a data group for multiple tires of differing sizes. This means that it is possible to estimate tire characteristics by employing variables related to sizes and specifications, which are not normally incorporated in a single-tire model, as input (Fig. 3).

The application of this method to existing, previously accumulated data groups makes it possible to obtain models

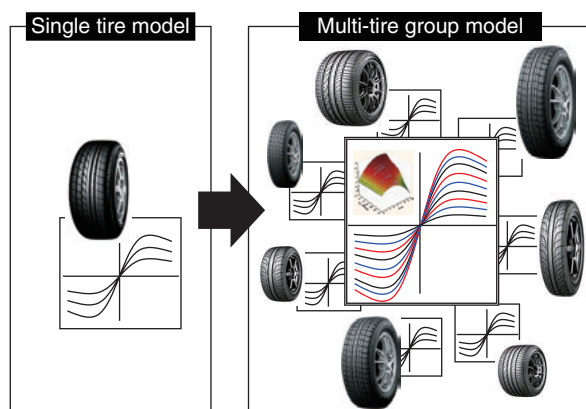


Fig. 1 New tire model from tire data group

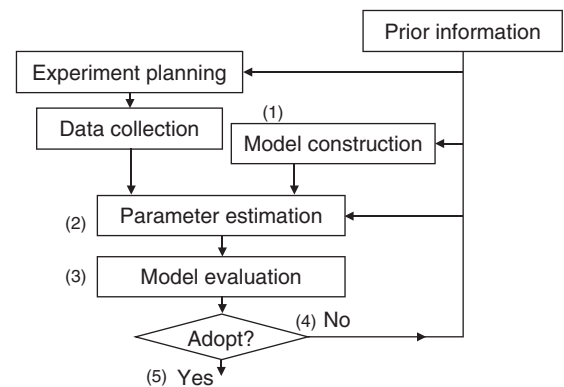


Fig. 2 System identification flow chart

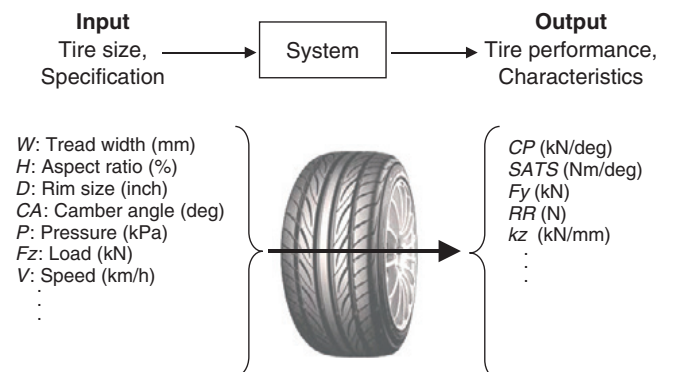


Fig. 3 System identification parameters

able to estimate the characteristics of tires of varying sizes and specifications.

Research has proceeded towards the application of tire models of this type, termed “statistical tire models.” To date, research has been conducted on the modeling of tire characteristics including cornering power, CP ; vertical spring stiffness, k_z ; self-aligning stiffness, $SATS$; side force characteristic, F_y ; and rolling resistance, RR , using this method, and the application of the method to the statistical generation of Magic Formula models. This paper will provide an overview of this research, with a focus on statistical CP models as a representative example of its application.

3. Statistical CP Models

3.1. Benefit of Statistical CP Models

Tire CP expresses the side force generated per unit angle of the tire’s side-slip angle, and is an important tire characteristic, affecting parameters such as handling performance and vehicle stability. The load-dependence of CP expresses the characteristic of change in CP in relation to vertical load, F_z . This is an important characteristic because it changes the ratio of front-rear tire force in response to the front-rear distribution of load, and has a significant effect on the steering characteristic of the vehicle.

Tire size has a strong effect on the tire’s CP characteristic, and is a factor that affects elements of performance including handling performance and vehicle stability. Because of this, it is important to examine it quantitatively at the initial stage of performance design. In addition, tire size also affects numerous other vehicle characteristics including brake performance, ride comfort, fuel efficiency, and quietness, and it is necessary to also take these characteristics into consideration when studies are conducted.

Given the above, in order to study vehicle dynamic performance, it is essential to estimate the CP characteristic for tires of diverse sizes and to conduct computational studies. Using a statistical CP model capable of estimating CP when tire size is given, it would be possible to efficiently study the tire size necessary for realizing target vehicle performance at the initial stage of performance design.

3.2. Formulation of Statistical CP Model

In order to clarify the relationship between tire CP and the basic tire characteristics width, W ; profile, H ; wheel diameter, D ; inflation pressure, P ; and vertical load, F_z , single-dimensional regression analyses were conducted for test data for approximately 130 types of tires. The results are shown in Figs. 4 and 5.

As is generally known, CP tends to increase when the width of tires is increased, their profile is reduced, and their diameter is enlarged. The results show that this tendency is

basically linear (Fig. 4).

In addition, it can be observed from Fig. 5 that CP behaves as a quadratic function in relation to F_z , and becomes saturated at high loads.

As this indicates, the general characteristics of tires appeared in a single-regression analysis, but the gap between regression lines or curves and test data was too great for the estimation of CP from tire size during initial performance studies, and the level of accuracy could not be considered sufficient. The change in CP in relation to F_z and W was therefore observed in order to investigate the multiple correlations between variables. Figure 6 shows an example of the results.

Figure 6 indicates that the increase in CP close to zero

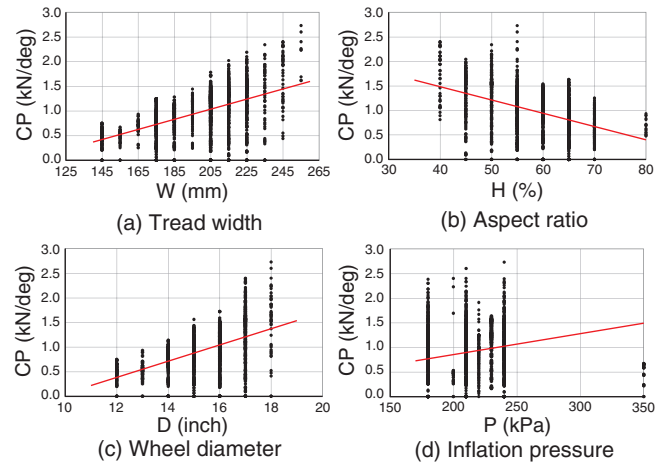


Fig. 4 CP relation to tire size parameters

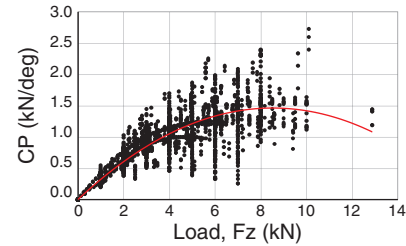


Fig. 5 CP relation to tire load, F_z

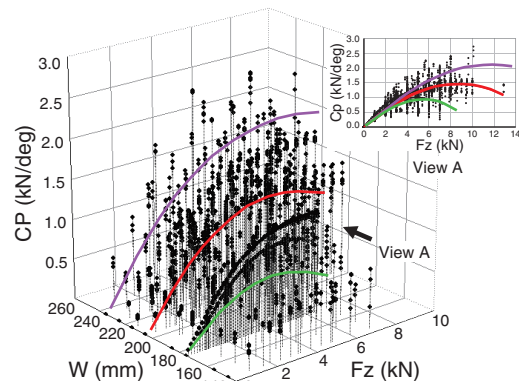


Fig. 6 CP relation to both F_z and W

load becomes steeper the wider the tire is made, and the load at which CP becomes saturated also shifts to the high-load side. This suggests that the second-degree behavior of CP in relation to Fz represents functions for parameters such as tire width, W .

Figure 7 shows the results of plotting test data for CP against Fz , W , H , D , and P , respectively, and approximating for smooth curved surface functions. The obtained approximated curved surfaces described quadratic function curves in relation to Fz , and their curvature and peak positions had all been transformed to basically linear functions in relation to W , H , D , and P .

Based on the above considerations, the function expression for the multi-dimensional regression of CP was hypothesized as shown below. If CP is approximated by a quadratic function of Fz , its curvature is termed A , and Fz that imparts the maximum value is termed B , CP is expressed by Eq. (1a). If it is assumed that $CP=0$ when $Fz=0$, the approximation function of CP is expressed by Eq. (1b).

$$CP = A(Fz - B)^2 + C \quad (1a)$$

$$CP = A(Fz^2 - 2BFz) \quad (1b)$$

The variables are substituted as follows.

$$A = \alpha$$

$$-2AB = \beta$$

The multi-dimensional regression model of CP in which curvature and peak position are transformed into linear functions in relation to W , H , D , and P can be expressed as Eq. (2). $a, b, c, \dots, j$ are multi-dimensional regression model coefficients.

$$\left. \begin{aligned} CP &= \alpha \cdot Fz^2 + \beta \cdot Fz \\ \alpha &= aW + bH + cD + dP + e \\ \beta &= fW + gH + hD + iP + j \end{aligned} \right\} \quad (2)$$

Here,

W : Tread width, mm

H : Aspect ratio, %

D : Rim diameter, inch

P : Inflation pressure, kPa

Fz : Normal load, kN

$a, b, \dots, j$: Coefficients of model

Next, the coefficients $a, b, c, \dots, j$ were identified by means of maximum likelihood estimation from CP_i, W_i, H_i, D_i, P_i , and Fz_i in test data. Equation (3) gives the error function of this model, J , for n units of test data.

$$\begin{aligned} J &= \sum_{i=1}^n \{CP_i - CP(a, b, c, d, e, f, g, h, i, j / W_i, H_i, D_i, P_i, Fz_i)\}^2 \\ &= \sum_{i=1}^n \{CP_i - (aW_i Fz_i^2 + bH_i Fz_i^2 + cD_i Fz_i^2 + dP_i Fz_i^2 + eFz_i^2 \\ &\quad + fW_i Fz_i + gH_i Fz_i + hD_i Fz_i + iP_i Fz_i + jFz_i)\}^2 \end{aligned} \quad (3)$$

In order to determine the multi-dimensional regression coefficients $a, b, \dots, j$ so as to minimize J , 10 linear partial differential equations, in which J is partially differentiated for each coefficient and set at 0 are established simultaneously and solved. Taking the equation for coefficient a as an example:

$$\frac{\partial J}{\partial a} = 0 \rightarrow a = \tilde{a} \left| \frac{\partial J}{\partial a} = 0 \right. \quad (4)$$

The partial differential equation that provides conditions of minimum error is therefore Eq. (5) below. Similarly, when partial differential equations are established simultaneously for coefficients b, c, d, e, f, g, h, i , and j , the linear matrix equation Eq. (6) is obtained. A and b are the numerical matrices obtained when test data is substituted, and it is therefore possible to directly derive the multi-dimensional regression model coefficient x from these matrices.

$$\frac{\partial J}{\partial a} = \frac{\partial}{\partial a} \sum_{i=1}^n \left[CP_i - (aW_i Fz_i^2 + bH_i Fz_i^2 + cD_i Fz_i^2 + dP_i Fz_i^2 + eFz_i^2 + fW_i Fz_i + gH_i Fz_i + hD_i Fz_i + iP_i Fz_i + jFz_i) \right]^2 = 0 \quad (5)$$

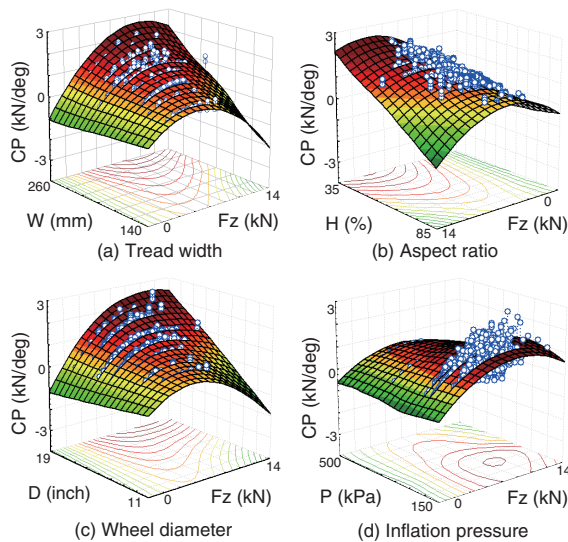


Fig. 7 CP contribution to Fz and tire size parameters

$$Ax = b, \therefore x = A^{-1}b \quad (6)$$

$$A = \begin{bmatrix} \sum W_i^2 Fz_i^4 & \sum W_i H_i Fz_i^4 & \sum W_i D_i Fz_i^4 & \sum W_i P_i Fz_i^4 & \sum W_i Fz_i^4 & \sum W_i^2 Fz_i^3 & \sum W_i H_i Fz_i^3 & \sum W_i D_i Fz_i^3 & \sum W_i P_i Fz_i^3 & \sum W_i Fz_i^3 \\ \sum W_i H_i Fz_i^4 & \sum H_i^2 Fz_i^4 & \sum H_i D_i Fz_i^4 & \sum H_i P_i Fz_i^4 & \sum H_i Fz_i^4 & \sum W_i H_i Fz_i^3 & \sum H_i^2 Fz_i^3 & \sum H_i D_i Fz_i^3 & \sum H_i P_i Fz_i^3 & \sum H_i Fz_i^3 \\ \sum W_i D_i Fz_i^4 & \sum H_i D_i Fz_i^4 & \sum D_i^2 Fz_i^4 & \sum D_i P_i Fz_i^4 & \sum D_i Fz_i^4 & \sum W_i D_i Fz_i^3 & \sum H_i D_i Fz_i^3 & \sum D_i^2 Fz_i^3 & \sum D_i P_i Fz_i^3 & \sum D_i Fz_i^3 \\ \sum W_i P_i Fz_i^4 & \sum H_i P_i Fz_i^4 & \sum D_i P_i Fz_i^4 & \sum P_i^2 Fz_i^4 & \sum P_i Fz_i^4 & \sum W_i P_i Fz_i^3 & \sum H_i P_i Fz_i^3 & \sum D_i P_i Fz_i^3 & \sum P_i^2 Fz_i^3 & \sum P_i Fz_i^3 \\ \sum W_i Fz_i^4 & \sum H_i Fz_i^4 & \sum D_i Fz_i^4 & \sum P_i Fz_i^4 & \sum Fz_i^4 & \sum W_i Fz_i^3 & \sum H_i Fz_i^3 & \sum D_i Fz_i^3 & \sum P_i Fz_i^3 & \sum Fz_i^3 \\ \sum W_i^2 Fz_i^3 & \sum W_i H_i Fz_i^3 & \sum W_i D_i Fz_i^3 & \sum W_i P_i Fz_i^3 & \sum W_i Fz_i^3 & \sum W_i^2 Fz_i^2 & \sum W_i H_i Fz_i^2 & \sum W_i D_i Fz_i^2 & \sum W_i P_i Fz_i^2 & \sum W_i Fz_i^2 \\ \sum W_i H_i Fz_i^3 & \sum H_i^2 Fz_i^3 & \sum H_i D_i Fz_i^3 & \sum H_i P_i Fz_i^3 & \sum H_i Fz_i^3 & \sum W_i H_i Fz_i^2 & \sum H_i^2 Fz_i^2 & \sum H_i D_i Fz_i^2 & \sum H_i P_i Fz_i^2 & \sum H_i Fz_i^2 \\ \sum W_i D_i Fz_i^3 & \sum H_i D_i Fz_i^3 & \sum D_i^2 Fz_i^3 & \sum D_i P_i Fz_i^3 & \sum D_i Fz_i^3 & \sum W_i D_i Fz_i^2 & \sum H_i D_i Fz_i^2 & \sum D_i^2 Fz_i^2 & \sum D_i P_i Fz_i^2 & \sum D_i Fz_i^2 \\ \sum W_i P_i Fz_i^3 & \sum H_i P_i Fz_i^3 & \sum D_i P_i Fz_i^3 & \sum P_i^2 Fz_i^3 & \sum P_i Fz_i^3 & \sum W_i P_i Fz_i^2 & \sum H_i P_i Fz_i^2 & \sum D_i P_i Fz_i^2 & \sum P_i^2 Fz_i^2 & \sum P_i Fz_i^2 \\ \sum W_i Fz_i^3 & \sum H_i Fz_i^3 & \sum D_i Fz_i^3 & \sum P_i Fz_i^3 & \sum Fz_i^3 & \sum W_i Fz_i^2 & \sum H_i Fz_i^2 & \sum D_i Fz_i^2 & \sum P_i Fz_i^2 & \sum Fz_i^2 \end{bmatrix}$$

$$x = \begin{bmatrix} a \\ b \\ c \\ d \\ e \\ f \\ g \\ h \\ i \\ j \end{bmatrix}, \quad b = \begin{bmatrix} -\sum W_i Fz_i^2 CP_i \\ -\sum H_i Fz_i^2 CP_i \\ -\sum D_i Fz_i^2 CP_i \\ -\sum P_i Fz_i^2 CP_i \\ -\sum Fz_i^2 CP_i \\ -\sum W_i Fz_i CP_i \\ -\sum H_i Fz_i CP_i \\ -\sum D_i Fz_i CP_i \\ -\sum P_i Fz_i CP_i \\ -\sum Fz_i CP_i \end{bmatrix}$$

3.3. Characteristics of Statistical CP Model

Figure 8 shows estimates of CP calculated by varying tire size variables and inflation pressure using the statistical CP model obtained above. With 195/60R15, an inflation pressure of $P=200$ kPa, and a load of $Fz=4$ kN as the benchmark, W , H , D , and P were separately varied with other variables held constant, and the respective contribution of each variable to CP was determined independently.

A comparatively relaxed positive correlation can be observed between H and CP . The single-dimensional regression analysis performed on the test data in Section

3.2 indicated a comparatively strong negative correlation between the two, but this is thought to reflect the fact that most low-profile tires on the market tend towards a broad W and large D , and CP was high as a result of the effect of this tendency. By contrast, in the case of the statistical CP model, which helps to enable multiple factors to be broken down, it is believed that because it is possible to isolate the separate contribution of the rate of flatness of the tires, a different correlation to that demonstrated by the single-dimensional regression analysis was observed.

The results obtained from the statistical CP model accord with the results of the single-dimensional regression analysis conducted in Section 3.2 in that no clear correlation can be observed between inflation pressure, P , and CP . It is believed that this tendency results from the action of multiple factors, including the canceling effect between the increase in CP due to the increased tire stiffness resulting from increased P and the reduction in CP due to the decreased surface area of the tire in direct contact with the road due to the increase in P .

CP and W display a comparatively strong positive correlation, and this correlation quantitatively expresses the increase in CP due to increased tire width. The contribution of D similarly quantifies the effect of increasing the diameter of the tires.

Figure 9 shows estimated values for the load-dependence of CP . As in Fig. 8, Fig. 9 shows the relationship between Fz and CP when (a) W , (b) D , (c) H , and (d) P are separately varied, this time with 205/55R16 and $P=210$ kPa as the benchmark.

Increasing W increases CP from the low-load range. Increasing D increases the external diameter of the tires,

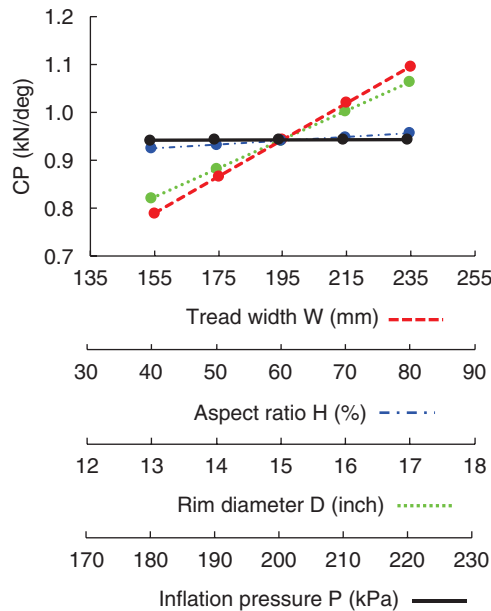


Fig. 8 CP variation by size factor

resulting in a slight reduction in low-load CP and an increase in high-load CP . A reduction in H (flattening the tires) significantly increases CP from the medium- to high-load ranges. An increase in P results in a slight decrease in CP in the low-load range and an increase in the high-load range.

The above characteristics all correspond with already known qualitative data, and it is therefore considered that this model generalizes and quantitatively expresses changes in tire characteristics with changes in tire size and inflation pressure.

3.4. Comparison of Statistical CP Model and Test Data

Test data and estimated values obtained using the statistical CP model were next compared. Figure 10 shows

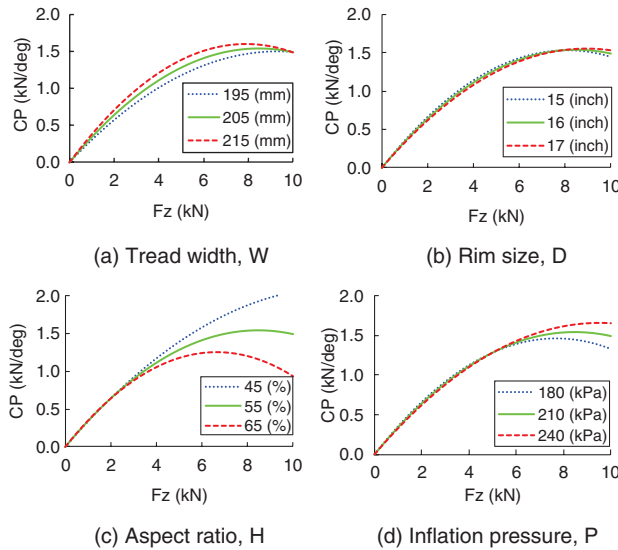


Fig. 9 Relationship between CP and Fz when size parameters varied

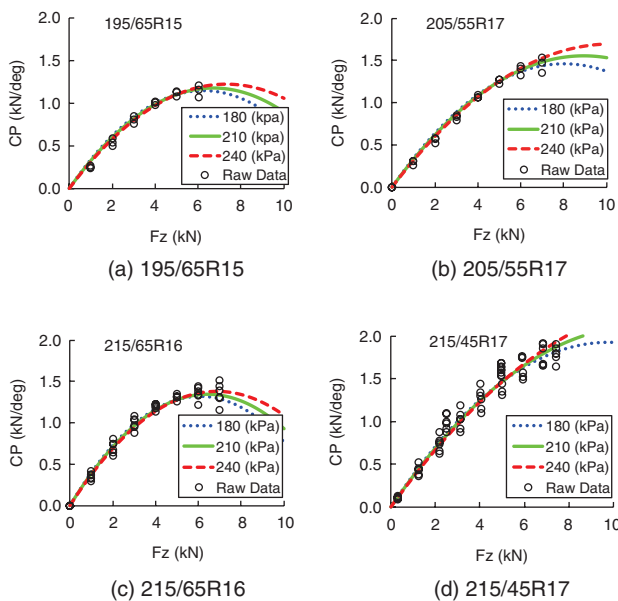


Fig. 10 Experimental data and curves obtained using statistical model

comparisons for four tire sizes. There is some variation in the characteristics displayed by the test data for each size because data for multiple tires of the same size but different type is included, but the statistical tire model reproduces the average characteristics of those tires.

4. Examples of Application of Statistical Tire Model

4.1. Study of Dynamic Performance

This section will look at the application of the statistical CP model to the stability factor, K , and the steering-stability capacity, C_s , as examples of its application to the study of vehicle dynamics. Equation (7), for K , gives an indicator of the vehicle's steer characteristic, while Eq. (9), for C_s , gives one of the indicators that determine Eq. (8), for $\zeta\omega_n$, which expresses steering responsiveness and convergence⁽⁴⁾. Here, an attempt will be made, when K and C_s are changed due to modification of certain vehicle specifications, to return these parameters to their values prior to the changes in specifications by modifying tire size.

$$K = \frac{C_r l_r - C_f l_f}{C_f C_r} \times \frac{m}{L^2} \quad (7)$$

$$\zeta\omega_n = \frac{1}{2V} \left(\frac{C_f + C_r}{m} + \frac{C_f l_f^2 + C_r l_r^2}{I_z} \right) = \frac{C_s}{2V} \quad (8)$$

$$C_s = \frac{C_f + C_r}{m} + \frac{C_f l_f^2 + C_r l_r^2}{I_z} \quad (9)$$

C_f/C_r : Front/Rear axle equivalent CP , N/rad

m : Car mass, kg

WD : Weight distribution to front axle, %

I_z : Yaw moment inertia, kgm^2

L : Wheelbase, m

V : Vehicle speed, m/s

l_f/l_r : Front/rear axle distance from COG, m

ζ : Yaw damping rate, -

ω_n : Yaw natural frequency, sec^{-1}

The case to be considered here will involve a change in specifications from Car A, the benchmark case, to Car B (Table 2).

Figure 11 shows the degree of contribution to K and C_s of the changes in each specification. The arrows show the direction in which each variable increases. When vehicle mass (m : red), yaw inertia moment (I_z : green), and the allocation of weight to the front axle (WD : blue) are increased simultaneously, differences in K and C_s appear between Cars A and B. An appropriate range exists for K and C_s , and the specifications for Car A produce values that basically fall within this range. Here, the potential to realize

Table 2 Car and tire specifications

	Car				Tire			
	L (m)	m (kg)	Iz (kgm ²)	WD (%)	W (mm)	H (%)	D (inch)	P (kPa)
Car A	2.4	950	1400	62.5	175	65	15	210
Car B	2.4	1000	1500	65	-	-	-	-

the same values for K and C_s in Car B as in Car A by means of changing the tire size will be examined.

Figure 12 shows an example of estimates of the degree of contribution of tire size variables in realizing the same level of K and C_s in Car B as in Car A, obtained using the statistical CP model. The arrows indicate the direction in which each variable increases, making it possible to quantitatively grasp the effect of W (blue), H (red), D (green), and P (orange) on K and C_s .

When tire size is changed from 175/65R15 to 185/60R15 under these conditions, the values of K and C_s for Car B, which were previously changed with the modification in the vehicle's specifications, become almost exactly the same as the values for Car A. In actual vehicle developments, other factors are also taken into consideration in determining tire size, but the ability to implement the initial performance design while obtaining a grasp of average tire characteristics

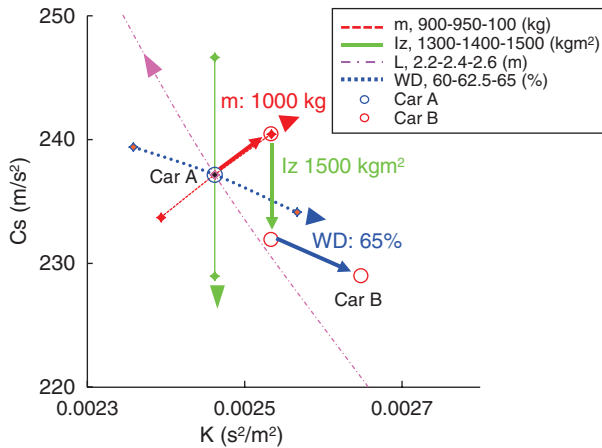


Fig. 11 Effect of changes in car specifications on K and C_s

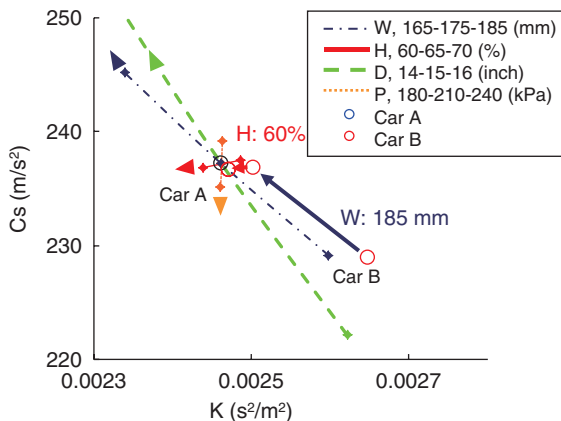


Fig. 12 Effect of tire size parameters on K and C_s

and their effects in this way will make it possible to balance the realization of greater commercial appeal in the finished product with increased efficiency in the development process.

4.2. Study of Dynamic Performance and Ride Comfort

The statistical tire model can be applied to the estimation of tire characteristics other than CP . Figure 13 shows an example in which k_z is modeled at the same time as CP and the effect of tire size on the parameter is studied. k_z is a fundamental tire characteristic that affects ride comfort and other parameters, and it is necessary to conduct studies in order to balance changes in size or specifications to increase CP with ride comfort.

The various color lines in Fig. 13 show the changes in CP and k_z against the benchmark tire specifications (185/65R15, $P=220$ kPa, $F_z=4.5$ kN; the central black point in the graph) when W , H , D , P , and F_z are individually increased and decreased with other characteristics held steady (the arrows show the direction of increase of each variable). The surrounding points show plots of the patterns generated by the combination of increases and decreases in each parameter.

Balancing increased CP with reduced k_z , moving towards the lower right-hand corner of the graph is the generally desirable direction. In actual practice, however, tire size and specifications are set in line with other constraining conditions. It is possible to determine the combinations of size, inflation pressure and load that help to enable the realization of target CP and k_z from the values of the input variables for points in the area that corresponds to the target characteristics. This makes it possible to obtain multiple candidate tire sizes. Because vehicle developers are also able to determine the degree of contribution of each size parameter at the same time, it is possible to proceed with quantitative studies towards the selection of the optimum tire size from among the candidates, while also taking other factors into consideration.

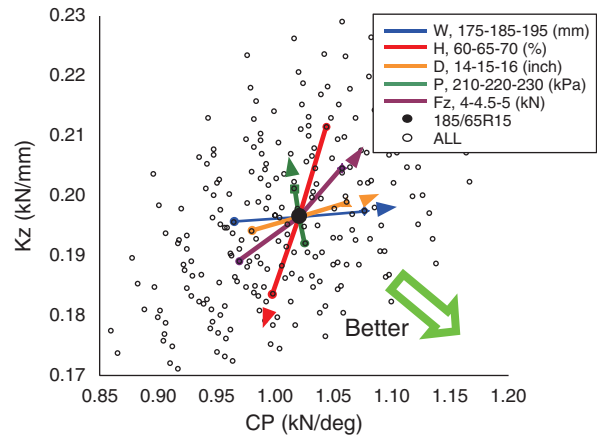


Fig. 13 Variation of CP and k_z when size parameters, inflation pressure and tire load changed

5. Conclusion

A statistical tire model making it possible to quantitatively estimate tire characteristics from tire size and specifications has been developed.

- (1) This tire model makes it possible to estimate tire characteristics for tires of various sizes without individually measuring and modeling the target tires.
- (2) For the modeling method, an experimental model for a single tire is expanded, and a system identification method is used to generate a multi-dimensional approximation equation from a data group encompassing multiple tires.
- (3) The use of this model, which is able to treat changes in tire size, will make it possible to conduct performance design at the initial stage of the development process more accurately and more efficiently.

In future, the authors intend to conduct further measurements of tire data, seeking, in addition to enhancing the physical theory basis of the statistical model, to realize further data accumulation in order to increase the accuracy of model identification, expand the range of applicable tire sizes, and incorporate qualitative variables such as tire material and structure into the model.

References

- (1) Pacejka, H. B.: Tyre and Vehicle Dynamics, Second Edition, Butterworth-Heinemann, p. 2-215 (2009)
- (2) Bridgestone Corporation: Pneumatic Tire Technology, Tokyo Denki University, p. 142-154 (2008) (in Japanese)
- (3) Katayama, T.: System Doutei Nyuumon, Asakura Shoten, p. 7-10 (1994) (in Japanese)
- (4) Miura N., Fukuda, M.: Jidoushasekkei to Kaiseki Simulation, Baifukan, p. 178-182 (1990) (in Japanese)

■ Author ■



Tomokazu SUZUKI



Kaoru KUSAKA

Study of Evaluation Method for Knee-grip Feeling of Motorcycles

Kobue IWATA* Koki KUDO*

ABSTRACT

Simulation technology is demanded that enables quantitative and unified evaluation of the knee-grip feeling of motorcycles in a short time. This study used a computer mannequin, which can simulate various rider physiques, and ergonomic techniques to derive the appropriate relative positions of the foot peg and knee contact area that enable riders to exert sufficient gripping force. In addition, the knee-grip status was reproduced using simulations, and the relationship with the subjective rating was investigated. This enabled establishing of evaluation parameters that express the feeling of discomfort during knee gripping and threshold values, and deriving of a predictive formula. As a result, it became computationally possible to estimate the feeling of thigh discomfort with an accuracy of 93%.

1. Introduction

When riding motorcycles, riders clasp the motorcycle body between their thighs and grip it with their knees. When this knee-grip is insufficient, the correct riding posture is sometimes lost when running over rough roads or accelerating and decelerating rapidly, making it challenging to stabilize the motorcycle body. When the shape of the contact area does not match that of the thighs during knee gripping, riders may feel some discomfort such as pain or localized pressure that prevents them from exerting sufficient gripping force. Conventionally, the knee-grip feeling was evaluated by having expert riders straddle a mock-up equivalent of the actual vehicle and confirm by subjective rating whether the shape enabled sufficient knee gripping without a feeling of discomfort in the thighs. However, there were issues such as mock-ups are easily damaged, so sufficient force could not be applied, making it a challenge to understand the degree of discomfort; or the seat and other mock-up parts are not deformed, making it a challenge to accurately understand the areas that the legs contact. In addition, even when using an actual vehicle, the areas that the legs contact depend on the leg length of the test subject, so there was also the issue of individual differences in the ratings. For these reasons, the construction of simulation technology is desired that enables quantitative and unified evaluation of the knee-grip feeling.

Regarding the subject of contact between the human body and machines, studies are being conducted on grasp and seated comfort. Koyano et al. were able to

computationally evaluate the fit feeling of the motorcycle seat by reproducing the body pressure distribution when sitting on the seat through nonlinear finite element method analysis⁽¹⁾. It is expected that knee grip can also be evaluated using similar techniques. However, nonlinear finite element method analysis requires much time to create special mesh models for analysis and to perform the calculations. The knee-grip surface shape has a strong influence on the motorcycle body styling and the internal structure such as the gasoline tank, so it needs to be decided in the early stage of development. Therefore, it is challenging to apply nonlinear finite element method analysis under the present situation.

When dealing with an elastic body such as a seat, deformation due to the load from the human body should also be taken into account. On the other hand, the areas contacted during knee gripping are mainly the gasoline tank and the side covers, which are comparatively resistant to deformation. The authors investigated whether the knee-grip surface shape can be evaluated in a short period of time by assuming these parts to be rigid bodies and using a standardized human body model.

This study uses a computer mannequin, which can simulate various rider physiques, and ergonomic techniques^{(2), (3)} to obtain the leg posture that enables riders to exert sufficient gripping force, and to derive the appropriate relative positions of the foot peg and knee contact area. In addition, a method is proposed to evaluate mock-up surface shape data in a short time by using the numerical evaluation parameters and the threshold values that express the feeling of discomfort during knee gripping.

* Motorcycle R&D Center

2. Rider Model

The feeling of discomfort during knee gripping refers to the difficulty of exerting force and the feeling of localized pressure on the thighs. Motorcycle body characteristics that produce a feeling of discomfort were investigated based on conventional riding tests by expert riders, and it was found that these characteristics can be broadly grouped into the following two factors.

- (1) Sufficient gripping force: Relative positions of foot peg and knee contact area
- (2) Feeling of discomfort produced by the contact surface:
Thigh contact surface shape

This study constructed a rider model that enables investigation of these two factors.

2.1. Creation of a Rider Model

A knee-grip simulation model was constructed using a model that simulates a mock-up surface and a rider in order to derive the locations that the rider's legs contact and the leg posture using a computer. The surface data of the locations that the legs contact was imported to CAD, and a rider model was created using the computer mannequin "RAMSIS" made by Human Solutions GmbH.

The RAMSIS software does not include information such as the internal tissues of the human body, but the physique can be changed and natural poses can be created that contact the given constraint points without exceeding the range of motion of each joint. In addition, the permanently exorable maximum isometric force (the force that can be exerted without changing the muscle length) corresponding to each posture can be output. RAMSIS used the body dimension data for various countries^{(4), (5)} as the input to express physiques corresponding to the target users.

To express the riding posture, RAMSIS constrained the hands to the handle grips, the soles of the shoes to the foot pegs, and the center point between the ischial tubercles to the hip point (HP) in consideration of seat deformation. In addition, the hip joints were rotated inward so that the knees

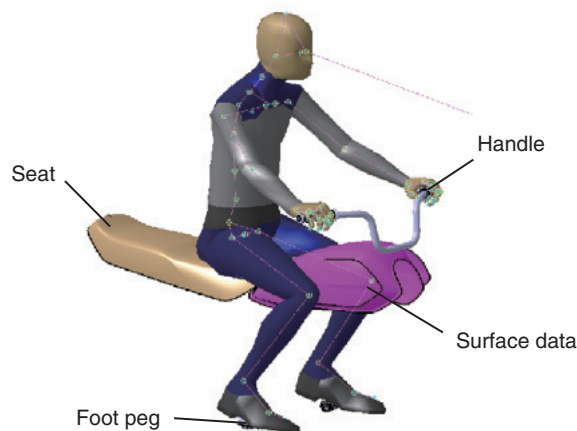


Fig. 1 Rider model

contacted the mock-up surface data. This was defined as the rider model (Fig. 1).

3. Relative Positions of Foot Peg and Knee Contact Area

3.1. Hip Joint Angles and Gripping Force

The ability to exert sufficient gripping force during knee gripping depends on the leg posture. Figure 2 shows the definitions of the human body joint angles. First, the gripping force of the legs for different hip joint abduction angles and internal rotation angles was obtained using the RAMSIS function that outputs the exorable maximum isometric force. The gripping force is the sum of the Y-axis forces that can be exerted by the knees and the ankles. Figure 3 shows the results.

Next, a simplified model was used to derive the

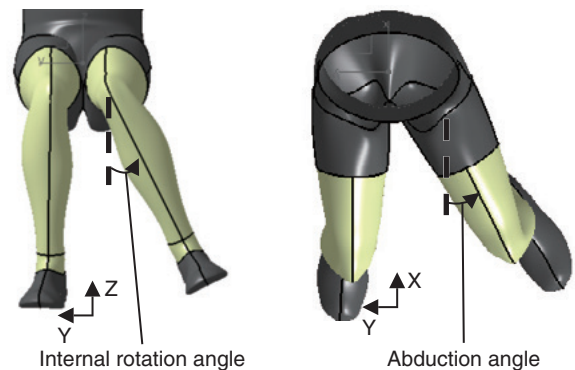


Fig. 2 Definition of angles around hip joint

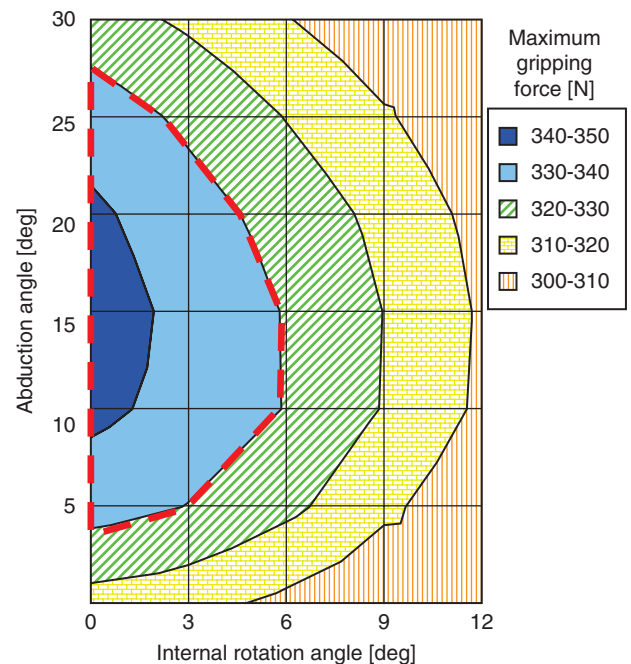


Fig. 3 Maximum gripping force in relation to hip joint angles

relational expression between acceleration and the gripping force T of a single leg required to support the upper body using only the legs. Here, M is the distributed mass of the upper body, α is the acceleration produced by braking or other factors, g is the acceleration of gravity, μ_1 is the coefficient of static friction between the clothing and the knee-grip surface, and μ_2 is the coefficient of static friction between the clothing and the seat. As shown by Eq. (1), it is hypothesized that to prevent the body from shifting forward or back due to deceleration or acceleration, the total force of the friction force $2T\mu_1$ generated between the knee-grip surfaces and the knees and the friction force $Mg\mu_2$ generated between the seat and the buttocks should be equal to or greater than the force $M\alpha$ applied to the upper body.

$$M\alpha \leq 2T\mu_1 + Mg\mu_2 \quad (1)$$

Converting the above equation yields Eq. (2).

$$T \geq M(\alpha - g\mu_2) / 2\mu_1 \quad (2)$$

Using the hypothetical case of a rider with a typical physique performing sudden braking, $M = 42$ kg (JM50%tile) and $\alpha = 5.6$ m/s² (corresponding to sudden braking as specified by driving schools: stopping in 11 m from a velocity of 40 km/h) were substituted to Eq. (2). In addition, the coefficient of static friction of cloth depends on the fiber direction, water content, temperature, and other factors. Therefore, 0.22 was substituted to μ_1 and μ_2 as reference values based on cases of measuring the coefficient of static friction of cotton⁽⁶⁾. In case of the aforementioned conditions, Eq. (2) indicates that a gripping force T of 330 N or more is required. Thus, it is thought that the leg posture should be set within the range indicated by the red broken line in Fig. 3.

3.2. Relationship between Foot Peg Width and Knee Contact Area Width

The leg posture when riding a motorcycle is determined by the relative positions of the center point of the foot peg and the knee contact area on the knee-grip surface. Therefore, a rider model (JM50%tile) that simulates a typical motorcycle riding posture was used, and the internal rotation angle and abduction angle of the hip joint were changed within the range in which gripping force of 330 N or more can be exerted. The relative positions of the shoe sole center point and the tibia point of the rider model at this time were investigated. The center of the shoe sole contacts the center of the foot peg and the tibia point contacts the knee-grip surface, so these results are shown in Fig. 4 as the relative positions of the foot peg center width and the knee contact area width (width: distance from vehicle's center plane). Sufficient gripping force can be exerted within the red-colored range shown in Fig. 4, so this was established as the recommended area. However, in consideration of the engine and frame size, it is not realistic to set a foot peg center width of 200 mm or less for a saddle-type motorcycle. Therefore, the range of 200 mm or less was excluded from the recommended area. The relative

positions of 20 existing motorcycle models judged to enable sufficient knee gripping were investigated and single regression analysis was performed to confirm the validity of this range. In addition, the 95% prediction interval of the values predicted by regression was derived. Figure 5 shows the results, which confirmed that the recommended area is within the prediction interval. That is to say, the set recommended area is within the prediction error margin of the regression formula, indicating that the recommended area is not too large. In addition, approximately 95% of the measured models were within the recommended area. These results indicate that the range shown in Fig. 4 is valid, and it can be said that the relative positions of the foot peg and knee contact area should be set within this range to enable exertion of sufficient gripping force.

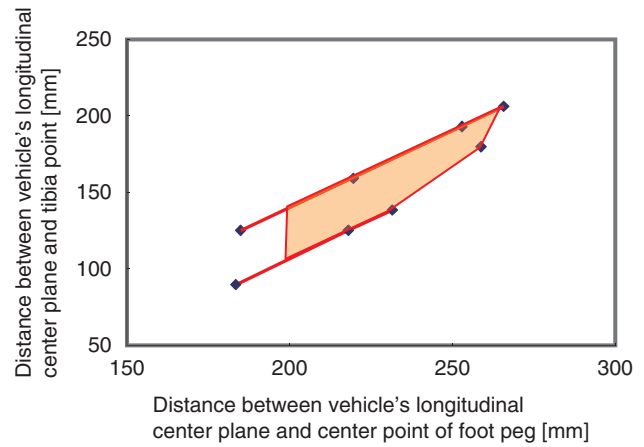


Fig. 4 Relative positions of foot peg and knee contact area that enable exertion of necessary force for gripping

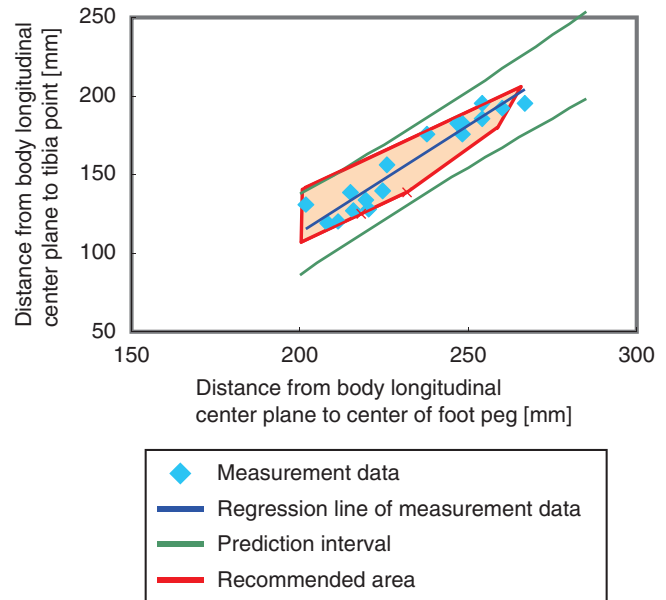


Fig. 5 Verification of appropriateness of recommended area

4. Evaluation of Thigh Contact Surface Shape

The relative positions of the thigh and knee-grip surface of the rider model were quantified through simulations, and the relationship between these values and the subjective ratings was investigated to evaluate the thigh contact surface shape.

4.1. Creation of State of Thigh Interference Model

The contact between the thigh and knee-grip surface of the rider model was checked on the computer and found to have a state of interference. The rider model is expressed using only surfaces, but the subcutaneous internal structure of the human body consists of tissues with different properties. The tissues near the skin surface in particular are soft tissues such as subcutaneous fat, so interference on the computer indicates yielding of the thigh area in the actual condition. It was thought that parameters expressing the feeling of discomfort that takes into account the degree of thigh yielding could be obtained by investigating this state of interference.

Therefore, the state of interference in the cross-section at the longitudinal center of the thigh was expressed numerically. As shown in Fig. 6, these parameters are the area over which the thigh interferes with the knee-grip surface (interference area), the ratio of the interference area to the thigh area (interference area ratio) and the depth of the interference area at the thigh center axis height (interference depth).

4.2. Investigation of Parameters Expressing a Feeling of Discomfort

4.2.1. Test subjects

The test subjects were 23 healthy men and women (19 men, 4 women) with experience riding motorcycles. Table 1 shows the physical characteristics values (average values and standard deviations) of the test subjects.

4.2.2. Experimental method

Five motorcycles with different knee-grip surface shapes were prepared (Table 2). The test subjects sat on the motorcycles in the stationary upright condition and rated the knee-grip feeling. Two seated positions were used: the position where the center point between the ischial tubercles matches the design HP (normal position), and the position seated as far forward as possible that produces the greatest

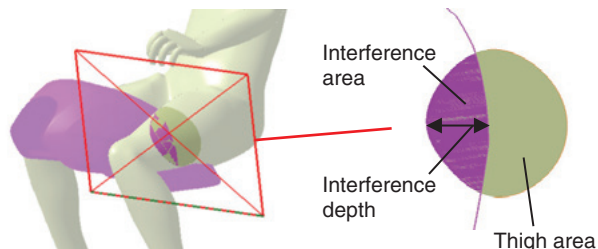


Fig. 6 State of interference of rider model

Table 1 Body sizes

	Male		Female	
	Ave.	S.D.	Ave.	S.D.
Height	171.2	5.9	156.3	6.2
Thigh length	393.3	38.0	371.3	22.9
Distance between tibia and groin	351.1	26.3	322.5	35.9
Lower leg length	407.6	28.0	342.5	33.0

(mm)

Table 2 Specifications of test motorcycles

Name	Category	Engine displacement [cm ³]	Wheelbase [mm]	Seat height [mm]	Mass [kg]
A	Sports	125	1355	818	127
B	Street	125	1318	730	123
C	Street	150	1311	771	127
D	Touring	800	1464	816	238
E	Naked	1000	1493	796	245

Table 3 Simulation results and subjective ratings

		Interference area [mm ²]	Interference area ratio [%]	Interference depth [mm]	Subjective rating [Point]
Normal position	A	2615	19.0	35.3	3.0
	B	150	1.3	4.5	4.1
	C	1628	12.4	23.9	3.6
	D	282	2.5	6.2	4.3
	E	184	1.6	4.7	4.8
Front position	A	3334	22.0	39.9	3.2
	B	3036	21.0	36.4	3.9
	C	8578	58.0	63.3	2.6
	D	2568	17.8	33.6	3.9
	E	2323	15.3	28.3	4.0
R		0.66	0.67	0.73	-

*R: Correlation coefficient with subjective rating

feeling of knee-grip discomfort as known from experience (front position).

- (1) The normal position and front position were subjectively rated, assigning one point for a strong feeling of discomfort in the thigh area or knee and five points for no discomfort whatsoever. In addition, surface data was prepared for motorcycles A through E, and the state of interference with the thigh of the rider model was investigated using the method described in Section 4.1 above.
- (2) The test subjects gradually rotated their hip joints outward (abduction) from the knee-grip conditions described above, stopping at the position where they felt that further abduction of the hip joint would result in insufficient knee grip. The length of the thigh area in the direction from the tibia point to the groin that did not contact the knee-grip surface at this time was defined as the minimum non-contacting length, and measured.

4.2.3. Results

- (1) Table 3 shows the subjective rating and simulation results. Note that the simulation values noted here are

the values derived using the JM50%tile rider model that is approximately equal to the average of the test subjects. The average subjective rating values of the test subjects showed that motorcycle A received the lowest normal position rating of 3.0 points, and motorcycle C received the lowest front position rating of 2.6 points.

In addition, comparison of the correlation coefficient R between the simulation values and the subjective rating values shows that interference depth exhibits the highest R . Therefore, a graph was plotted showing the interference depth for motorcycles A through E. As shown in Fig. 7, the graph coordinates use the tibia point at the cross-section of the thigh center axis as the origin, the line connecting the tibia point and the groin as the X axis, and the axis perpendicular to that line as the Y axis. Figure 8 shows a graph of the interference depth. Reading the characteristics of motorcycles A and C, which received low ratings, from Fig. 8 shows that motorcycle C exhibited the largest interference depth value of all the motorcycles, and motorcycle A exhibited a pointed shape compared to the other motorcycles. To express these characteristics numerically, an index was defined as follows based on the results in Fig. 8. Table 4 shows the calculation results for motorcycles A through E.

Interference depth: $F(x)$, Maximum interference depth: $F(a)$

Degree of protrusion: $\{F(a) - F(a + 30) + F(a) - F(a - 30)\} / 2$
(Here, a is the value of x when $F(x)$ is maximum)

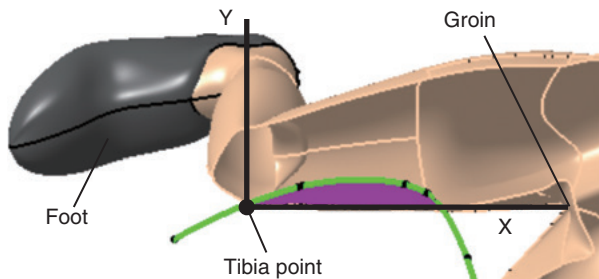


Fig. 7 Definition of axes

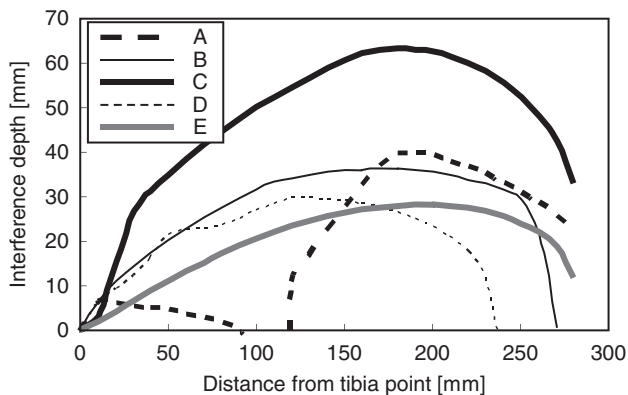


Fig. 8 Interference depth of motorcycles A-E (when seated at front position)

Table 4 Maximum interference depth and degree of protrusion (front position)

		$F(a)$ [mm]	Degree of protrusion
Front position	A	39.9	19.1
	B	36.4	0.9
	C	63.3	2.2
	D	29.9	2.2
	E	28.3	1.0

- (2) The average minimum non-contacting length between the test subjects' thighs and the knee-grip surface was 80.3 mm, and the standard deviation was 5.6 mm. These results are thought to indicate that when the minimum non-contacting length is 80 mm or more from the tibia point, knee gripping becomes insufficient and the rider feels uneasy. Therefore, the minimum non-contacting length threshold value was set at 80 mm.

4.3. Derivation of Interference Limit of Thigh

The thigh area of the rider model can interfere to any degree in simulations. However, bones and tendons are hardly transformed in the actual condition, and there are also cases where the tibia point cannot make contact depending on the shape of the knee-grip surface (Fig. 9). Therefore, the thigh yielding amount was measured and the interference depth limit value was derived in order to predict the location on the thigh where contact with the knee-grip surface begins in the actual condition.

4.3.1. Test subjects

The test subjects were the same as those in Section 4.2.

4.3.2. Experimental method

The thigh yielding amount was measured using an indentation hardness tester (TK-HS100) made by Tokushu Keisoku Co., Ltd.⁽⁷⁾ The seated posture was used for measurement, and the seat height was adjusted so that the knee joint was bent by 90° in the flexion/extension direction. In addition, a rigid barrier was set along the outside of the thigh to prevent abduction of the hip joint when applying pressure. To help ensure a constant pressing direction, the measuring instrument height was variable as

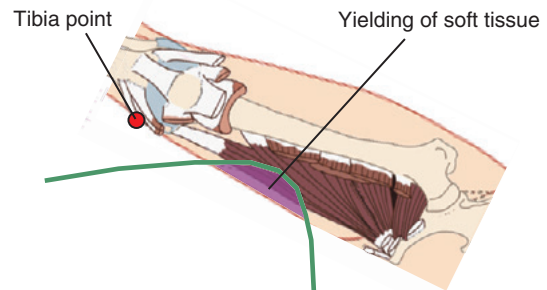


Fig. 9 Thigh posture in actual condition

shown in Fig. 10, and the measuring instrument was fixed on a stand with wheels that moved in only a single axial direction.

Markings were made at 10 mm intervals starting from the tibia point on the center axis on the inside of the thigh, and pressure was applied by the indentation hardness tester pressing in toward the femur at each marking point. The applied pressure was such that test subjects did not feel strong pain.

4.3.3. Results

Figure 11 shows a graph plotting the average values of all test subjects on the same coordinate axes as those shown in Fig. 8. These results are defined as the interference limit $G(x)$. This graph also shows the results for motorcycle C, which had the largest interference depth $F(a)$ in Section 4.2, and motorcycle E, which had a good rating. In addition, even for the same pointed shape, pain was less likely felt as the difference between the interference limit $G(x)$ and the interference depth $F(a)$ increased, so $G(a) - F(a)$ was defined as the interference tolerance $\delta(a)$.

Motorcycle C greatly exceeded the interference limit $G(x)$. When substituted to the actual condition, the tibia point did not contact the knee-grip surface, and the knee was detached. To express the actual condition using Fig. 11, the interference tolerance $\delta(a)$ should be 0 or more, i.e., $\delta(a) \geq 0$, so the hip joint of the rider model was abducted to satisfy this condition. Figure 12 shows a graph plotting

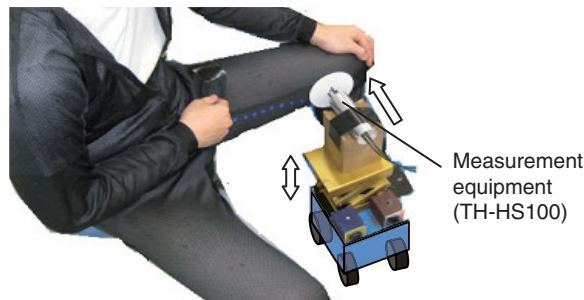


Fig. 10 Measurement of yielding depth

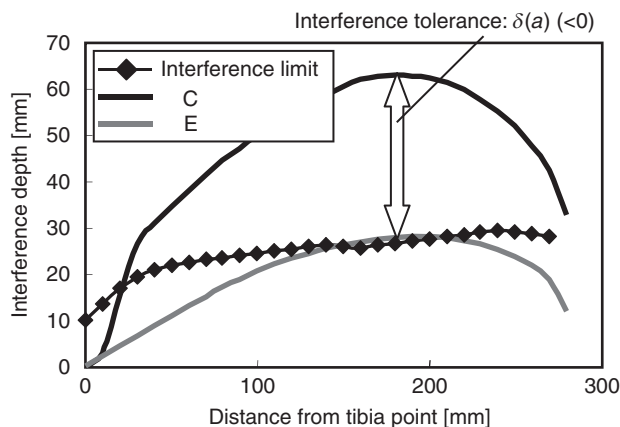


Fig. 11 Results of interference limit

the interference depth at this time. As shown in Fig. 12, the position where the thigh began to contact the knee-grip surface (minimum non-contacting length) was 100 mm from the tibia point in the direction of the groin. According to the results in Section 4.2, the minimum non-contacting length threshold value is 80 mm, so it is thought that motorcycle C received a low rating as a result of exceeding this threshold.

4.4. Derivation of Multiple Regression Formulas by Subjective Rating of Model Surface Shapes

Subjective rating was performed using model surface shapes, and multiple regression analysis was performed to determine the parameters that express a feeling of discomfort in the thigh during knee gripping and to derive a predictive formula that expresses the discomfort level.

4.4.1. Test subjects

The test subjects were the same as those in Section 4.2.

4.4.2. Experimental method

Model surface shapes were created that can express the difference between the maximum interference depth and the degree of protrusion. The model surface shapes had a saucer shape created by cutting a sphere. A total of nine different model surface shapes were prepared using three height specifications (10 mm, 12.5 mm, 15 mm) enabling the maximum interference depth to be changed, and three radius specifications (60 mm, 80 mm, 90 mm) enabling the degree of protrusion to be changed.

The test subjects sat on a motorcycle with a flat plate attached to the knee-grip surface to eliminate dependence on the existing knee-grip surface shape. The saucer-shaped model surface was attached to the flat plate so that the tip of the model surface contacted the specified thigh position when the test subjects gripped with their knees. Figure 13 shows the experimental conditions. This method was used to subjectively rate each of the total 27 patterns comprising different thigh positions (tibia point, middle of thigh, groin area), different maximum heights (3 types) and different radii (3 types), arranged in random order using a random number table.

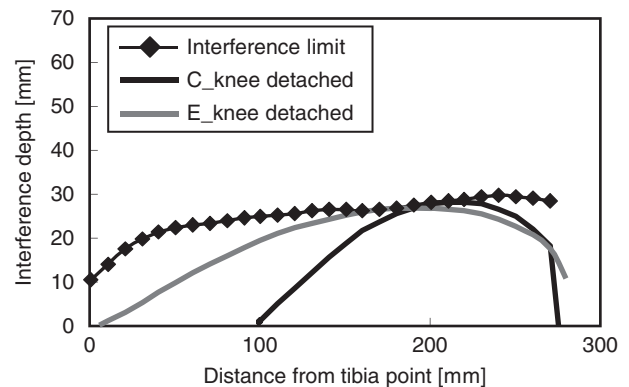


Fig. 12 Results adjusted by interference limit

Table 5 Verification of multiple regression formulae

Position on thigh		Tibia	Middle of thigh	Groin
Correlation coefficient R		0.93	0.99	0.95
Significance level F		0.01 **	0.00 **	0.03 *
P-value	Interference tolerance	-	0.02	0.02
	Degree of protrusion	0.01	0.00	0.04

Table 6 Multiple regression formulae for each position

Position	Multiple regression formulae
Tibia $0 \leq a \leq 60$	Point = $-2.3 \times [\text{Degree of protrusion}] + 6.02$
Middle of thigh $60 < a \leq 160$	Point = $-0.56 \times [\text{Degree of protrusion}] + 0.14 \times [\text{Interference tolerance}] + 3.4$
Groin $160 < a$	Point = $-0.35 \times [\text{Degree of protrusion}] + 0.1 \times [\text{Interference tolerance}] + 3.1$

4.4.3. Results

The test conditions were reproduced on a computer, and the interference tolerance $\delta(a)$ and the degree of protrusion were derived for each of the 27 patterns using the method described in Chapter 4 above. Table 5 shows the results of performing multiple regression analysis for each thigh position using the subjective rating values as the dependent variable and the interference tolerance $\delta(a)$ and degree of protrusion as independent variables. The multiple correlation coefficients showed an extremely strong correlation of 0.9 or more. In addition, P-values were calculated in order to check the significance of independent variables, obtaining both interference tolerance and degree of protrusion of less than 5%, so these variables are significant at the 5% level of significance. Therefore, it can be said that interference tolerance and degree of protrusion are necessary and sufficient as evaluation parameters that express the feeling of thigh discomfort. In addition, subcutaneous fat and other tissues differ according to the thigh position, so a multiple regression formula was obtained for each position. Therefore, the multiple regression formula should be selected according to the point “a” value that results in the maximum interference depth. Table 6 shows the obtained multiple regression formulae.

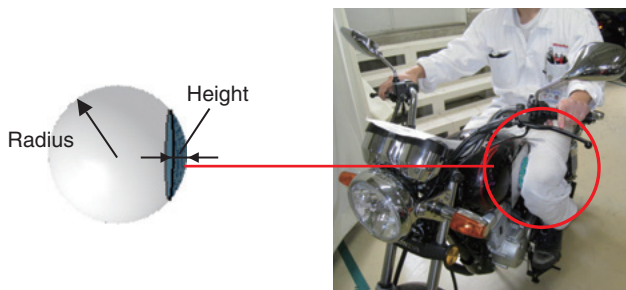


Fig. 13 Test to determine multiple regression formulae

5. Verification of Validity

To confirm whether the multiple regression formulae derived from the experiment results in Chapter 4 and minimum non-contacting length judgment using the interference limit correctly express the feeling of thigh discomfort during knee gripping, verification was performed using the shape data for 15 different motorcycles that had been rated by running tests.

5.1. Verification Method

Five motorcycles rated in running tests as producing a feeling of thigh discomfort, five motorcycles rated as producing some thigh discomfort, and five motorcycles rated as producing no thigh discomfort were selected as the 15 motorcycles.

The mock-up surface data of each motorcycle was imported, and rider models were added to the data using the method described in Chapter 2. The rider model seated positions were the normal position and front position, and the rider physiques used the JM50%tile, JM95%tile and JF5%tile models to simulate different heights.

The interference tolerance and degree of protrusion were derived from these models and substituted into the multiple regression formula corresponding to the maximum point “a” value (Table 6). In addition, when the interference depth exceeded the interference limit, the minimum non-contacting length was derived.

5.2. Verification Results

Figure 14 shows the predicted thigh discomfort values obtained as a result of calculations for the JM50%tile rider model in the front position. Note that when the minimum non-contacting length exceeded the threshold value of 80 mm, one point was assigned as the predicted value. Likewise, comparison of the results for each physique confirmed that the feeling of thigh discomfort could be estimated computationally with 93% accuracy by using the derived multiple regression formula and minimum non-contacting length.

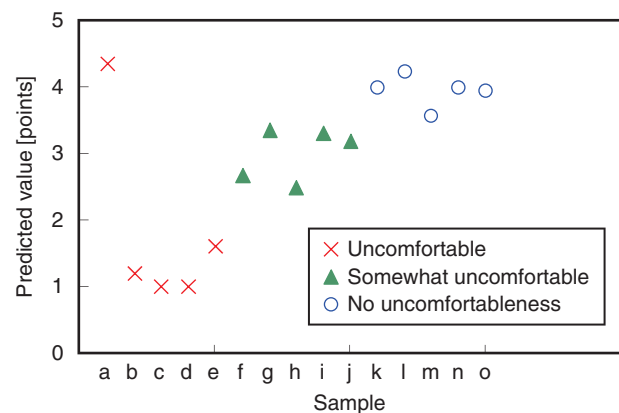


Fig. 14 Results of validity check

The results for motorcycle model “a” diverged from the predicted value, and a survey of the feeling of discomfort produced the response of “loose feeling”. Motorcycle model “a” is within the recommended area established in Chapter 3, and the leg posture should enable exertion of sufficient gripping force. However, there is insufficient interference near the groin, which is surmised to have lowered the secure feeling. The predictive formula derived in this study enables judgment when the knee-grip surface interferes excessively with the thigh, but is less reliable when there is insufficient interference. This is thought to be the reason why the result diverged from the predicted value.

6. Conclusion

Motorcycle knee-grip simulation technology was constructed using a computer mannequin and ergonomic techniques, and the following was achieved.

- The recommended relative positions of the center of the foot peg and the knee contact area on the knee-grip surface were determined that result in a leg posture that enables exertion of sufficient gripping force.
- Interference tolerance and degree of protrusion were defined as parameters that express the feeling of knee-grip discomfort, and a predictive formula was derived. This enabled computational evaluation of the feeling of knee-grip discomfort from the mock-up surface data.

References

- (1) Koyano, M., Nakamura, T., Sunaga, T.: Seating comfort anticipation of the motorcycle seat by body pressure distribution simulation, JSAE Annual Congress (Autumn), 20045648 (2004)
- (2) Yoshikawa, S., Kato, I.: Mechanical characteristics of human body surface and its measurement method, Ningenkogaku, Vol. 11, No. 1, p. 23-28 (1975)
- (3) Yonemoto, K.: Anthropometry for car design, Ningenkogaku, Vol. 6, No. 1, p. 15-18 (1970)
- (4) Research institute of human engineering for quality life: Japanese body size data book 2004-2006
- (5) Open Ergonomics Ltd.: ADULTDATA
- (6) Ishikawa, K., Onuma, G., Usuzumi, T., Tokunaga, K., Nakashima, M.: Santeiban Seni, Tokyo Denki Daigaku Shuppankyoku (1969) (in Japanese)
- (7) Murayama, M., Nosaka, K., Yoneda, T., Minamitani, K.: Change in hardness of the human elbow flexor muscles after eccentric exercise, Eur J Appl Physiol, Vol. 82, p. 361-367 (2000)

■ Author ■



Kobue IWATA



Koki KUDO

Method for Welding Condition Settings with Forecasted Distortion Applied for Two-wheeled Vehicle Body Preproduction by Automatic Welding Machine

Hidenori OKAJIMA*

Kenichi FURUDATE*

Kenta FUKUDA*

ABSTRACT

Welding experiments were conducted using test pieces in order to determine the arc current, arc voltage, and forecasted amount of distortion, which are conditions for the automatic welding process when fabricating two-wheeled vehicle bodies for preproduction. A database was created from the experimental results. Empirical formulas were derived from this database and used to calculate arc current and arc voltage. These calculation results were taken as the basis for establishing a method of forecasting heat shrink. These methods enabled the setting of processing conditions without the use of a two-wheeled vehicle body for welding tests. They further enabled the implementation of development testing using mass-production equivalents in the initial stages of development. As a result, early decisions on the design specifications for welded parts, shortening of preproduction vehicle fabrication schedules, and reduction of development costs were achieved.

1. Introduction

There is a need to determine which welding conditions can be applied during mass production in order to enable earlier determination of design specifications for welded parts on two-wheeled vehicle bodies for preproduction.

Welding processes in preproduction used to be performed manually, and settings for the processing conditions of arc current, arc voltage, and welding speed, as well as the forecasted amount of distortion due to heat shrink, depended on the experience and intuition of the worker. Because this method involves individual variations in processing condition settings, the processing conditions from preproduction could not be incorporated in mass production, which mainly employs automatic welding. The processing condition settings therefore had to be reset when the transition to mass production was made, which was inefficient.

This issue was addressed by introducing preproduction methods that include automatic welding equipment and takt time that are on a par with mass-production lines. This method becomes important in that the welding process conditions that have been set for preproduction are applied consistently to mass production.

Processing conditions for automatic welding are ordinarily set using welding tests with actual vehicle parts.

It is no simple matter to set conditions by means of a single test, however, and the consequent need for a trial-and-error approach necessitates the fabrication of numerous vehicle parts for testing as well as the expenditure of time and labor for the testing.

This research is intended to provide for greater efficiency in setting the processing conditions for automatic welding. To that end, correlations between welding process conditions and quality derived from experiments with test pieces were entered in a database, and a method was created to enable setting welding conditions and forecasting the amount of distortion from that database.

2. Two-wheeled Vehicle Body Welding Process

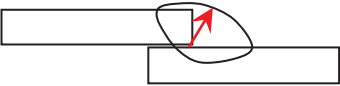
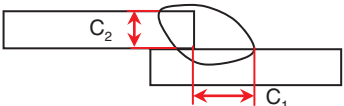
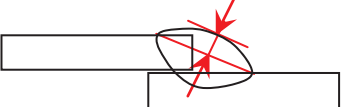
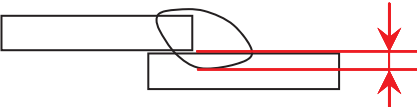
2.1. Two-wheeled Vehicle Body Welding

Figure 1 shows a two-wheeled vehicle body and the welded joints that are used. The joints in the welded parts mainly employ lap joints, T-joints, and pipe joints. These are made up of a maximum of over 200 parts. The total weld length can reach a maximum of 10 m or more.

Welding process machines mainly operate by metal active gas (MAG) welding, metal inert gas (MIG) welding, and other such forms of arc welding, which automatically feed in filler metal to the weld zone that is heated and

* Motorcycle R&D Center

Table 1 Weld quality evaluation items

Control items	Section	Explanation
Throat thickness		Minimum thickness of welding bead
Leg length		Distance from the root of joint to the toe of welding bead
Weld reinforcement		Height of the weld metal lying outside the plane containing the weld toes
Penetration		Depth of fusion into the parent metal from its surface

melted by an electric arc generated by the electrode of a tool tip known as a torch⁽¹⁾.

2.2. Weld Quality Control Items

2.2.1. Quality of welded parts

Fulfillment of the design requirements for strength is an important part of weld quality. Weld quality is evaluated by cutting across the welded part and examining the cross-section to check the quality inside the weld (Fig. 2). The areas evaluated are the leg length, throat thickness, weld reinforcement, and penetration. The quality control consists of judgment whether the requirements for these control items are fulfilled (Table 1).

2.2.2. Dimensional accuracy

It is necessary for the completed two-wheeled vehicle body to fit within the allowable dimensions given in the drawings. However, deformation occurs because of heat shrink of deposited metal (welding bead) when cooling after welding, and this affects dimensional accuracy. This makes

it necessary to forecast the amount of deformation before the welding is done⁽²⁾(Fig. 3).

2.3. Welding Process Conditions

Conditions for the welding process with two-wheeled vehicle bodies are the arc current, arc voltage, forecasted welding distortion, and welding speed.

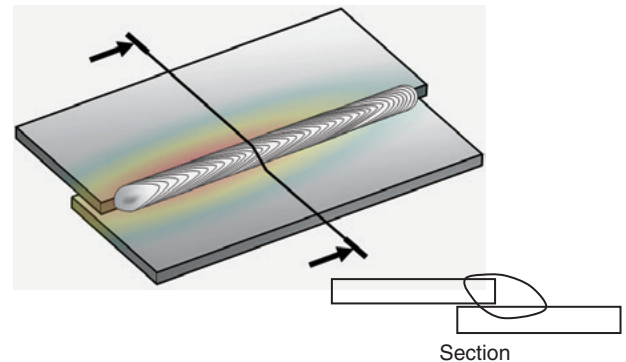


Fig. 2 Cross section of welding bead

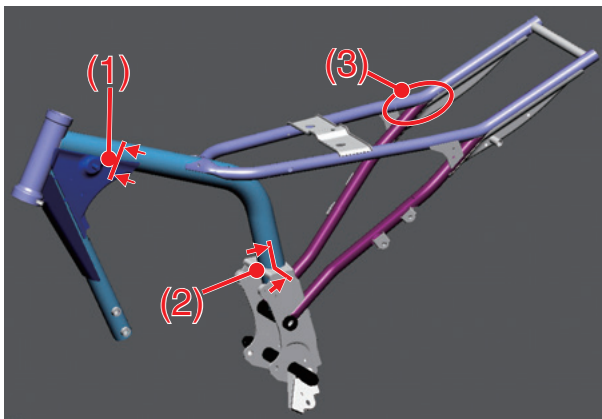
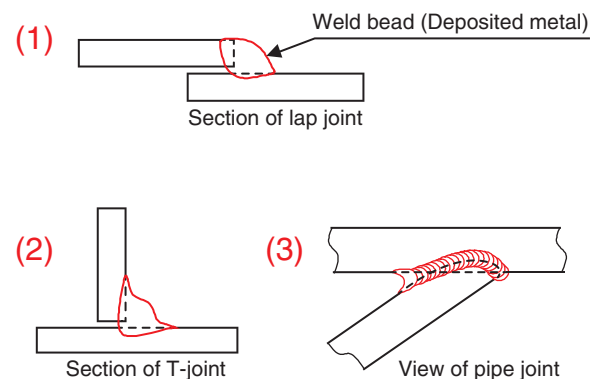


Fig. 1 Configuration of motorcycle body frame and type of welded joint



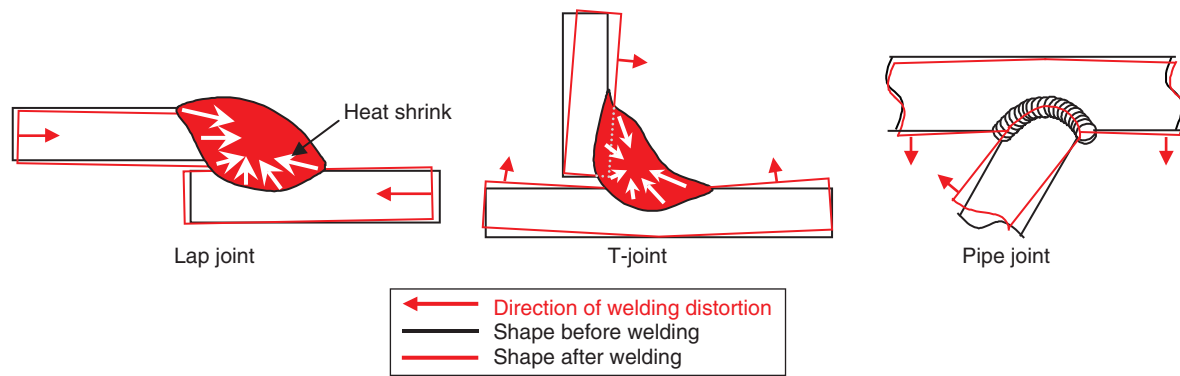


Fig. 3 Welding distortions

In the case of MIG welding and MAG welding, the arc current is used to set the filler metal feed amount and the arc voltage is used to set the amount of filler metal that is melted.

The setting of predicted welding distortion is carried out by implementing the value that includes the prospective amount of heat shrinkage to make the dimensions to be fitted within dimensional tolerance after welding.

The welding speed is the speed of welding torch movement. The welding torch is moved while filler metal is being fed in, so that a change in the speed also causes a change in the deposited metal volume. The strength of the welded joint is affected as a result, making it necessary to feed an optimal amount of filler metal for each speed range.

Those settings are to be determined to fulfill the strength requirements with the material properties, thickness and the joint shapes of the welding pieces in consideration.

3. Creation of the Database

Experiments were conducted using test pieces that recreated the shapes of welded joints used in two-wheeled vehicle bodies, and a database was created from the results.

3.1. Experiment Overview

The shapes of the welded joints, properties of the materials, and plate thicknesses classified by the model of two-wheeled vehicle body yielded 540 types of steel and 380 types of aluminum alloy welded joints. These were used in test pieces that were subjected to experiment under combinations of 17 arc current and arc voltage conditions and eight welding speed conditions. Data was collected on the welded joint deformation resulting from welding bead quality and heat shrink under each processing condition.

3.2. Experiment Conditions

Due to the necessity for considering mass producability when applying automatic welding processes to preproduction of the body, the welding speed as determined from the takt time in mass-production was set in advance.

Experiments were conducted using combinations of arc current and arc voltage conditions, as described above, based upon the set welding speed. Data on weld quality was then collected.

Pressed pieces and cast pieces are used in welded parts. When multiple parts are combined, the accumulated error can result in gaps in welded joints. Consequently, gaps were made in advance in welded joints and welding was performed (Fig. 4). The sizes of the gaps were set in view of the general tolerance of each part.

3.3. Experiment Results

To begin with, the experiments on throat thickness, leg length, penetration and weld reinforcement, which are crucial quality control items to the strength requirements, were carried out to collect the data while performing the quality evaluations. Regarding the distortions, as shown in Fig. 5, the differences in dimensions between pre- and post-welding were measured as the amount of distortions. Data on the rate of weld reinforcement was also collected as necessary to determine the deposited metal volume (Fig. 6).

Test pieces made of cold rolled steel plate (SPCC: JIS G 3141) with a plate thickness of 2 mm were used for experiments under four welding speed conditions and three arc current and arc voltage conditions with a high frequency of application in actual use. The results are shown below as one example.

3.3.1. Throat thickness

Figure 7 shows the relationships between arc current, arc voltage, and throat thickness at each welding speed. The results indicate that the throat thickness diminishes in proportion to the increase in welding speed. This is because the deposited metal volume diminishes. It is generally thought to be preferable to have the throat thickness be equal to or greater than the plate thickness in order to fulfill strength requirements. The experiments yielded figures equal to or greater than plate thickness under all the conditions used, so that the results fulfilled strength requirements.

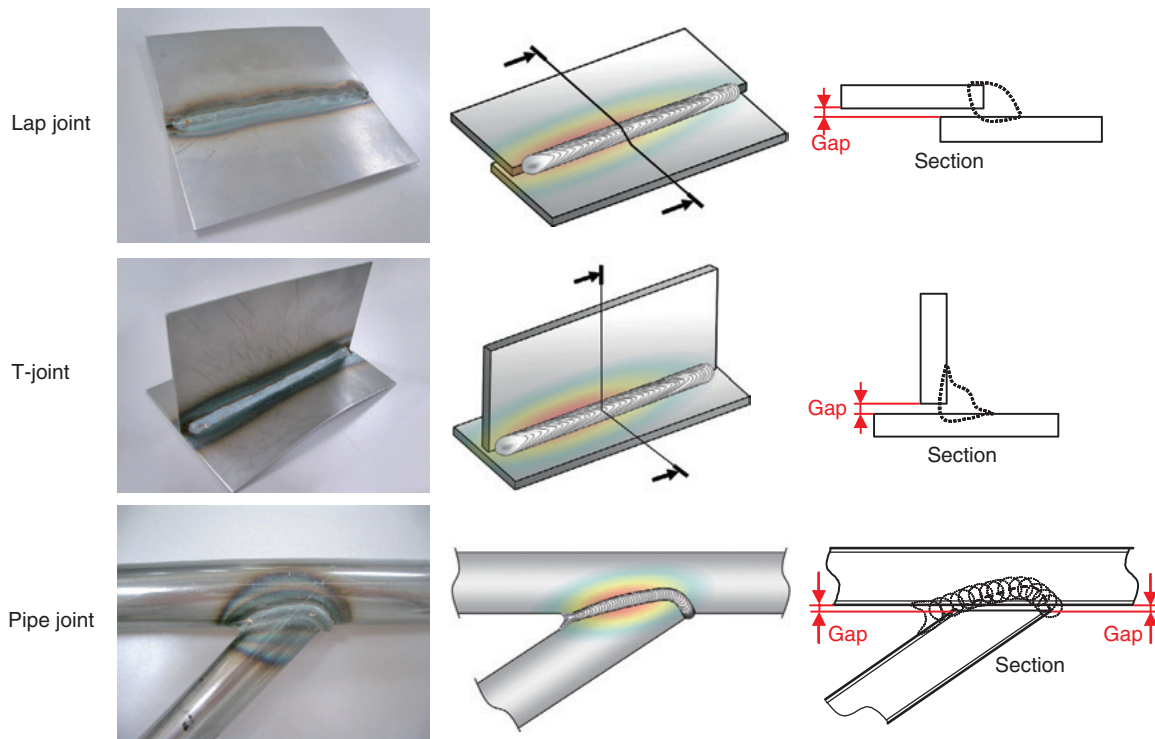


Fig. 4 Test pieces and welded joint gap settings

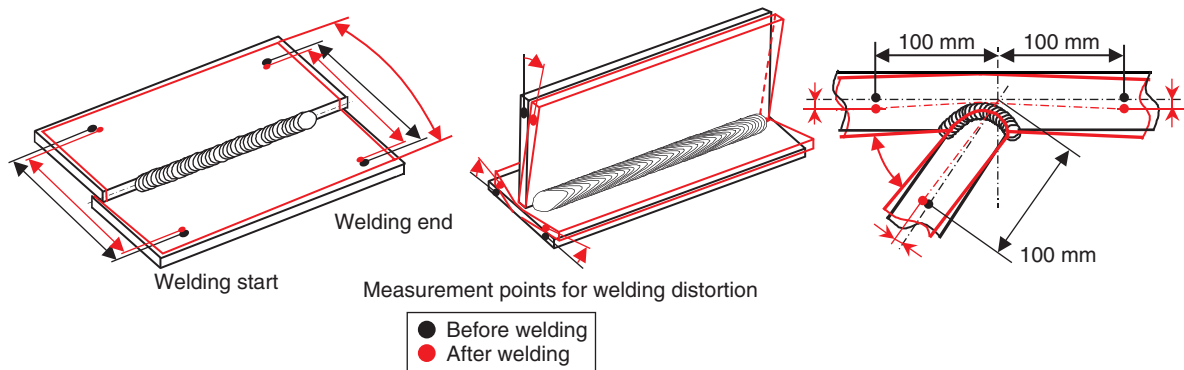


Fig. 5 Measurements of welding distortion

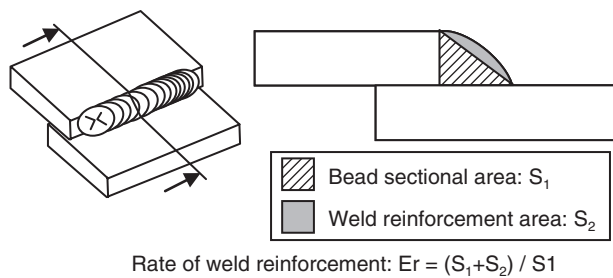


Fig. 6 Rate of weld reinforcement

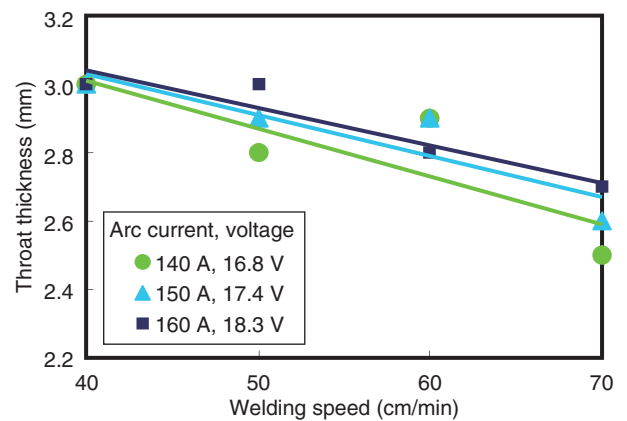


Fig. 7 Relation between welding speed and throat thickness for each arc current (SPCC, t2.0)

3.3.2. Leg length

Figure 8 shows the relationships between arc current, arc voltage, and leg length C_1 at each welding speed. It is generally thought to be preferable to have the leg length be equal to or greater than the plate thickness in order to fulfill strength requirements. The experiments yielded figures equal to or greater than plate thickness under all the conditions used, so that the results fulfilled strength requirements.

3.3.3. Weld reinforcement

Figure 9 shows the relationships between arc current, arc voltage, and weld reinforcement at each welding speed. Since weld reinforcement depends on the throat thickness, the weld reinforcement diminishes as the volume of the deposit diminishes in the higher speed ranges at all arc currents. Excessive reinforcement places the weld toe under high stress, and so is not preferable. The upper limit of height is set in accordance with the joint shape, the material, and the structure.

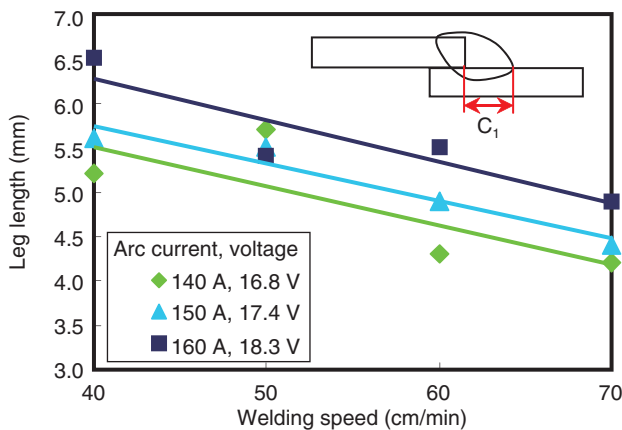


Fig. 8 Relation between welding speed and leg length for each arc current (SPCC, t2.0)

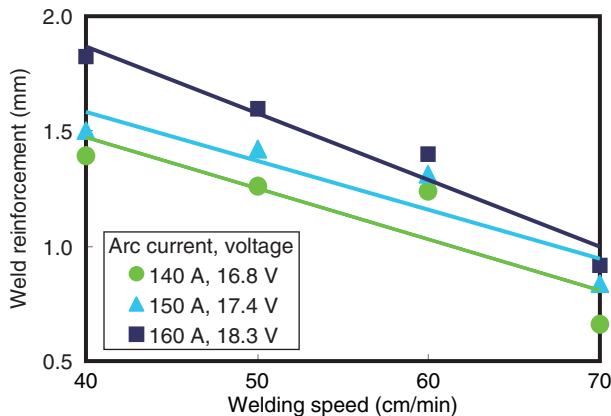


Fig. 9 Relation between welding speed and weld reinforcement for each arc current (SPCC, t2.0)

3.3.4. Penetration

Figure 10 shows the relationship with penetration at each welding speed. The results indicate that penetration was achieved in all speed ranges that were set. However, excess penetration results in deposited metal protruding from the reverse side of the plate, which can tend to cause oxidation or other such defects, and it is therefore not preferable.

3.3.5. Amount of distortion

Figure 11 shows the relationships between arc current, arc voltage, and distortion amount at each welding speed. The heat input per unit weld length increases in direct proportion to the product of arc current and arc voltage, and in inverse proportion to the welding speed. The increase in heat input also causes greater heat shrink, which is thought to result in greater distortion.

3.3.6. Rate of weld reinforcement

Figure 12 shows the relationships between arc current, arc voltage, and rate of weld reinforcement at each welding

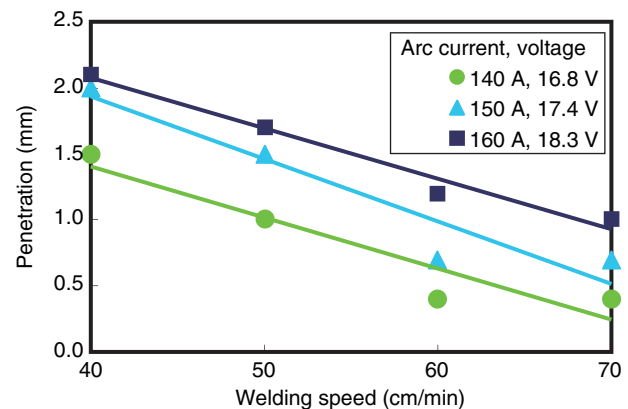


Fig. 10 Relation between welding speed and penetration for each arc current (SPCC, t2.0)

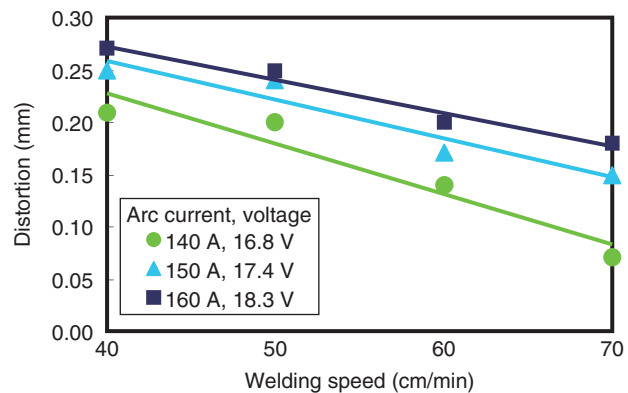


Fig. 11 Relation between welding speed and distortion for each arc current (SPCC, t2.0)

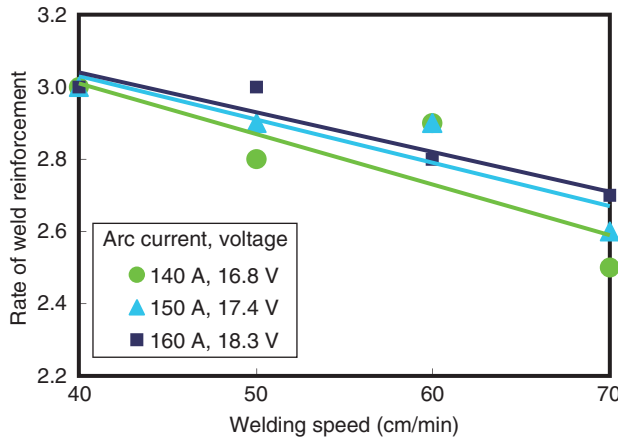


Fig. 12 Relation between welding speed and rate of weld reinforcement for each arc current (SPCC, t2.0)

speed. The rate of weld reinforcement is dependent on weld reinforcement, and the weld reinforcement diminishes with the reduction in the volume of the deposit in the higher speed ranges. The deposited metal volume can be found from the results in accordance with the welding speed, arc current, and arc voltage.

4. Derivation of Welding Conditions

The welding conditions of arc current and arc voltage are calculated by the following procedure:

1. Determine the deposited metal volume that satisfies the strength requirement suited to welding speed and plate thickness.
2. Determine the filler metal feed amount (hereafter expressed as the wire feed speed) from the deposited metal volume.
3. Convert the wire feed speed into arc current and arc voltage.

4.1. Determination of Deposited Metal Volume

Figure 13 shows typical dimensions for a welding bead section.

The deposited metal volume D is the volume of metal to be deposited per unit time. It is expressed as the product of the total bead sectional area and the welding speed. The total bead sectional area is expressed as the product of the bead sectional area S_1 and the rate of weld reinforcement

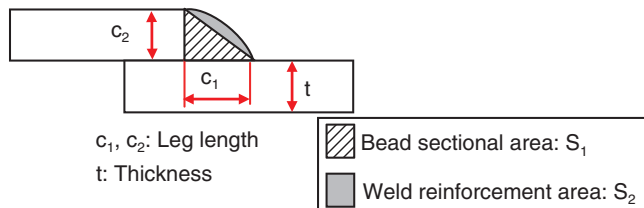


Fig. 13 Theoretical section of welding bead shape

E_r . The deposited metal volume D is expressed by Eq. (1)⁽³⁾. In general, a value of 1.15 is assigned as the rate of weld reinforcement and the leg length necessary for strength is made equal to the plate thickness.

$$D = (S_1 + S_2) v = E_r \cdot S_1 \cdot v$$

$$S_1 = \frac{(c_1 \cdot c_2)}{2} \quad (1)$$

$$E_r = \frac{S_1 + S_2}{S_1}$$

c_1, c_2 : Leg length

The experimental results in Section 3, however, indicate that the rate of weld reinforcement E_r and the leg length c_1 and c_2 vary with the welding speed v . The leg length c_1 is equal to or greater than the plate thickness. Using Eq. (1), therefore, the regression formula for each plate thickness is determined from the experimental data obtained in the practical use range, and the rate of weld reinforcement E_r' and the leg length c_1' and c_2' that depend on the welding speed are calculated. The joint gap G was then added to c_2' to derive Eq. (2), which determines the deposited metal volume D' (Fig. 14).

$$D' = E_r' \frac{c_1'(c_2' + G)}{2} v \quad (2)$$

The terms here are:

$$E_r' = -a_1 v + b_1$$

$$c_1' = -a_2 v + b_2$$

$$c_2' = -a_3 v + b_3$$

Figure 15 shows the results from calculation for each plate thickness using Eq. (2). This confirms that the figures come out close to the experimental results for deposited metal volume.

4.2. Determination of the Wire Feed Speed from the Deposited Metal Volume

The filler metal feed speed (hereafter wire feed speed)

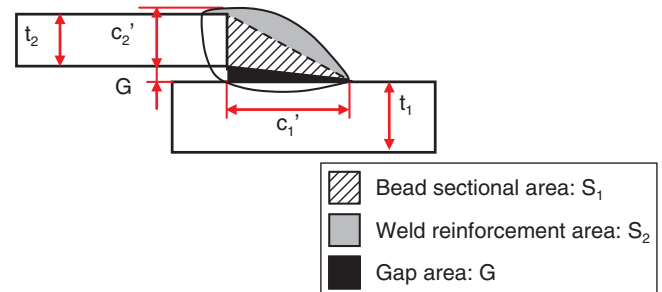


Fig. 14 Actual section of welding bead shape

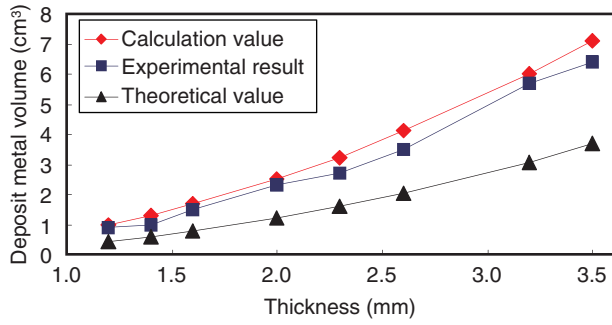


Fig. 15 Relation between deposited metal volume and thickness

W required per unit weld length is determined from the calculation results described above. The wire feed speed W is generally expressed by Eq. (3)⁽³⁾.

$$W = \frac{D}{\pi(d/2)^2} \quad (3)$$

d : Filler metal diameter

4.3. Conversion of Wire Feed Speed to Arc Current and Arc Voltage

Figure 16 shows the measurement results for the wire feed speed and arc current, and Fig. 17 the measurements

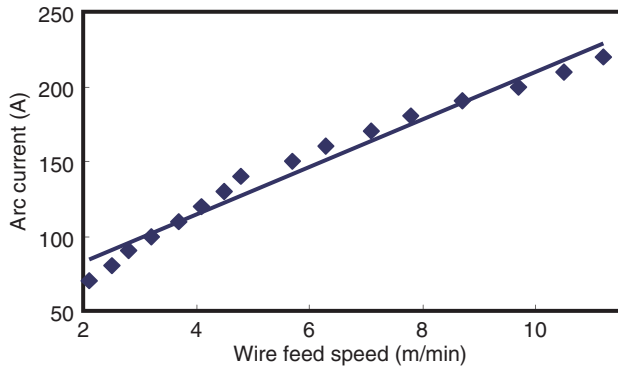


Fig. 16 Relation between wire feed speed and arc current

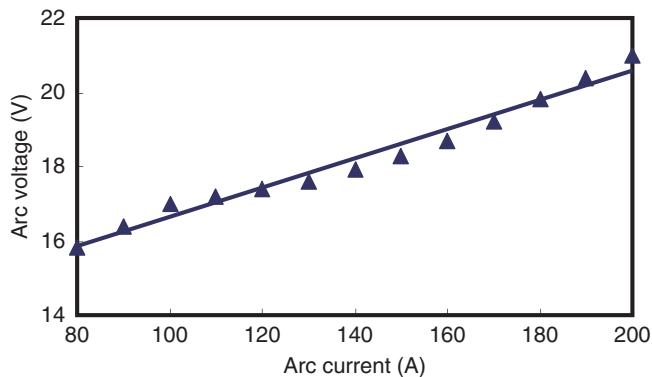


Fig. 17 Relation between arc current and voltage

results for arc current and arc voltage. The regression formula was determined from the database of actual wire feed speed measurements for each arc current condition in the experiment in Section 3.

$$I = a_4 W - b_4 \quad (4)$$

The arc voltage V was determined from the database of actual arc current measurements using the regression formula.

$$V = a_5 I - b_5 \quad (5)$$

The arc current and arc voltage are then calculated in a similar manner by material and welded joint shape.

5. Derivation of a Formula for Forecasted Distortion Amount

Calculation of the amount of distortion is performed by the following procedure:

1. Calculate the heat input.
2. Calculate the bead shrinkage of the welding start portion and the welding end portion.
3. Calculate the change in angle.

5.1. Calculation of Heat Input

The heat input Q (J/cm) per unit weld length can be determined from the arc current, arc voltage, and welding speed.

The heat input Q is generally expressed by Eq. (6)⁽¹⁾.

$$Q = \frac{I \cdot V}{v} \quad (6)$$

I : Arc current

V : Arc voltage

v : Welding speed

5.2. Calculation of Bead Shrinkage

The heat input Q obtained above is taken as the basis for determining a regression formula for welding bead shrinkage from the database of actual measurements of welding start portions and welding end portions.

$$\varepsilon_{st} = a_6 Q + b_6 \quad (7)$$

$$\varepsilon_{ed} = a_7 Q + b_7 \quad (8)$$

ε_{st} : Shrinkage of welding start portion

ε_{ed} : Shrinkage of welding end portion

5.3. Calculation of the Change of Angle

The temperature of the parent metal rises during the passage of time from the start of welding to the end of welding. The penetration therefore increases, and the

difference in the shrinkage at the two ends causes a change in the angle (Fig. 18). The change in angle θ that occurs per unit weld length can be determined using Eq. (9).

$$\theta = 2 \frac{\tan^{-1}[(\varepsilon_{ed} - \varepsilon_{st})/2]}{L} \quad (9)$$

L : Weld length (mm)

The above calculation results are used for the forecasted amount setting of the welding jig.

6. Verification of Practicability and Results of Application

The two-wheeled vehicle body is fabricated using an automatic welding machine (robot welding) with arc current and arc voltage set by the formula shown in Section 4 as well as a welding jig that incorporates the forecasted welding distortion calculated by the formula shown in Section 5. The weld quality and dimensional accuracy of the body were then verified (Fig. 19).

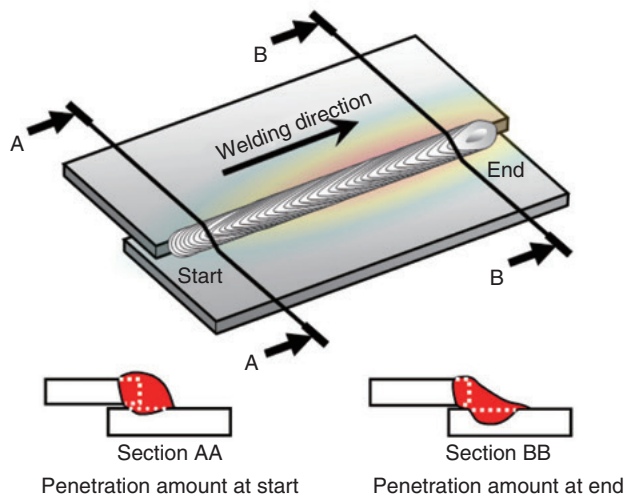
6.1. Verification Results

Weld Quality

Figure 20 shows the results from comparison of the deposited metal volume values relative to plate thickness as calculated and as actually measured. The two were in close correlation.

Dimensional Accuracy

Figure 21 shows the dimensional datum point and dimensional measurement points including the engine mounting points and suspension system attachment points.



Amount of shrinkage increases with increasing penetration into base metal

Fig. 18 Angle change due to welding distortion

Figure 22 shows the results for dimensional accuracy. The horizontal axis shows the distance of each measurement point from the dimensional datum point. The vertical axis shows the rate of dimensional error in the X, Y, and Z directions of each measured

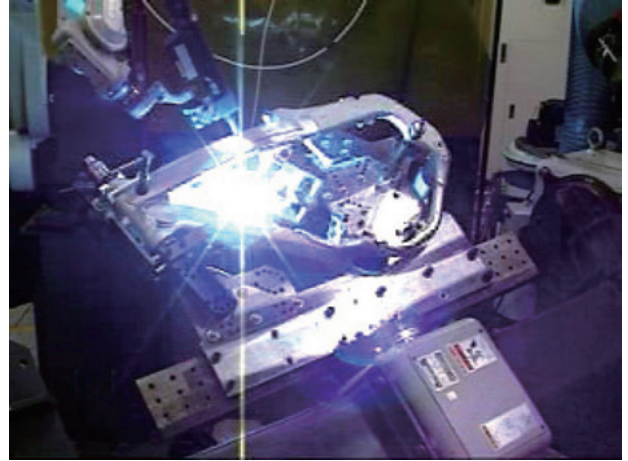


Fig. 19 Automatic welding machine

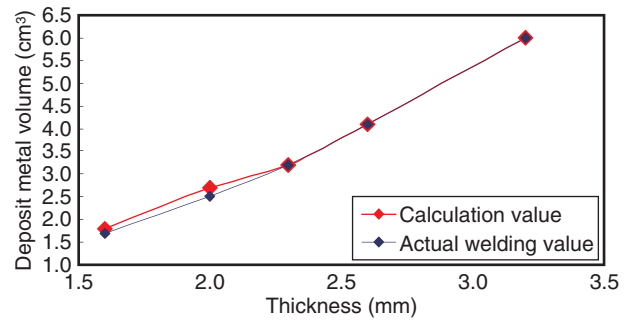
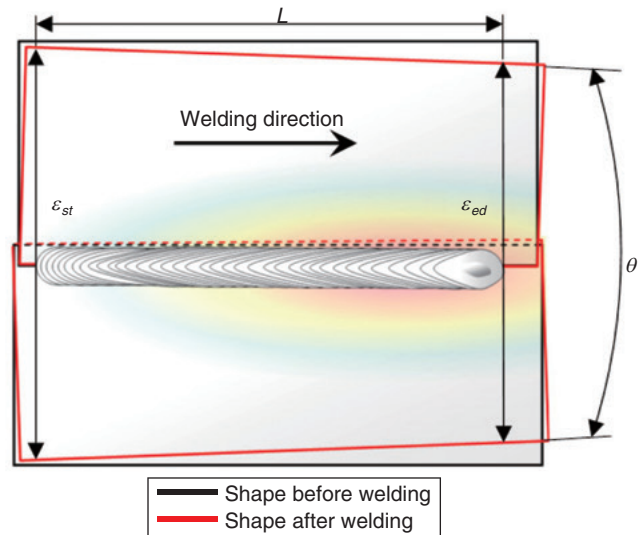


Fig. 20 Verification of calculation results



dimension as a percentage of the design tolerance. The dimensional error rate stayed at 50% or under for all the measurement points, indicating that stable accuracy had been achieved.

6.2. Effectiveness of Application to Developed Models

It was judged from the above verification results that this method could be applied to model development, and a preproduction two-wheeled vehicle body was fabricated. One effect of this was to eliminate the

necessity for the process of setting conditions by means of welding tests using two-wheeled vehicle parts. This eliminated the two days' time and labor for establishing conditions and enabled an approximately 40% reduction in preproduction costs (average value after database application) (Fig. 23). This also enabled implementation of two-wheeled vehicle body endurance tests and other such activities using mass-production equivalents in the initial stages of development, which contributed to earlier determination of design specifications.

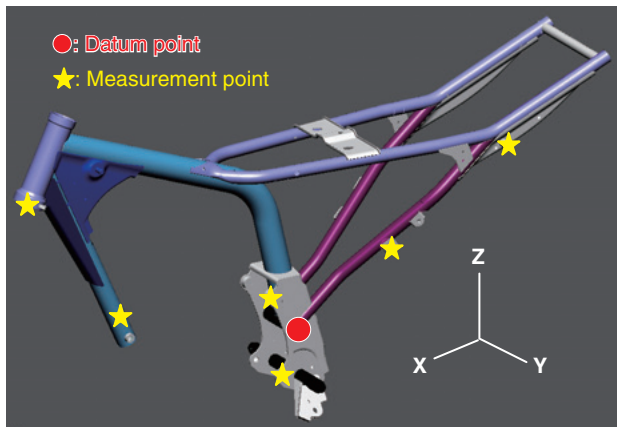


Fig. 21 Measurement points

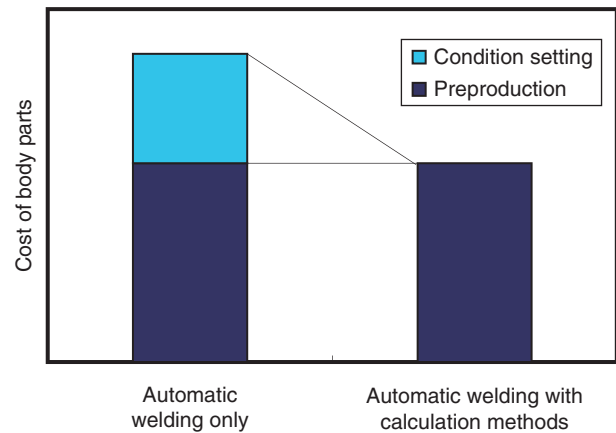


Fig. 23 Cost reduction of body parts

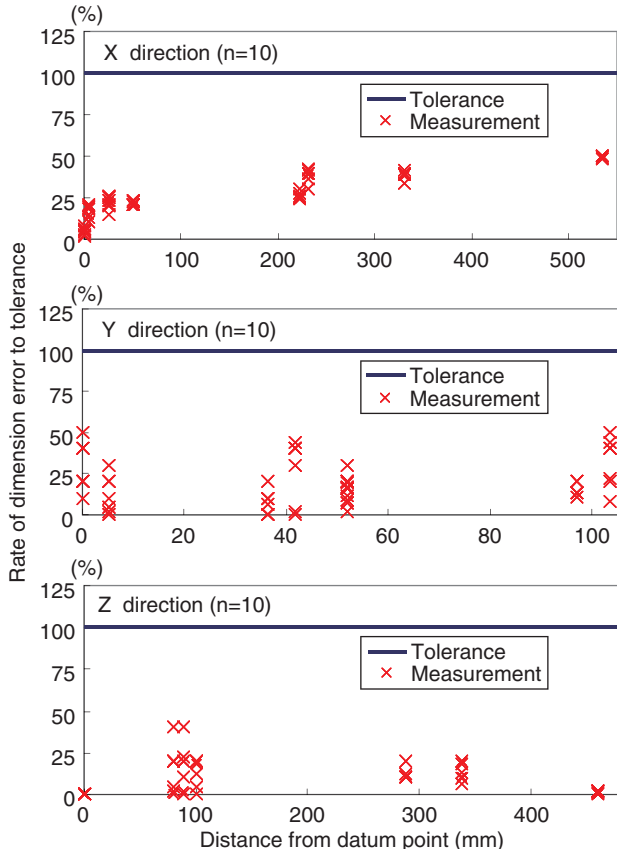


Fig. 22 Results of dimensional accuracy verification

7. Conclusion

In order to achieve efficient setting of welding process conditions for automatic welding machines used in fabricating preproduction two-wheeled vehicle bodies, experiments were conducted using test pieces, and those results were entered in a database. This database was used to create methods for setting the processing conditions of arc current, arc voltage, and welding speed, as well as to forecast the amount of heat shrink. The following results were obtained:

- (1) The necessity for the step of setting process conditions by means of welding tests using two-wheeled vehicle parts was eliminated, enabling reduction in preproduction time and labor and costs.
- (2) The highly accurate setting of processing conditions enabled stabilization of welding quality and dimensional accuracy.

The above enabled implementation of development testing using mass-production equivalents in the initial stages of development, contributing also to the earlier determination of design specifications for welded parts.

References

- (1) Japan Welding Society: Yousetsu Binran, Maruzen Publishing, p. 150-159, p. 331-340, p. 1357 (1985) (in Japanese)

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- (2) Tokumaru, Y., Okano, S., et al.: Kikai Kousaku 1, Kaiteiban, Jikkyo Shuppan, p. 209-210 (1989) (in Japanese)
- (3) Ueda, K.: Yousetsu, Shinkou welding service, p. 98-99 (2010) (in Japanese)

■ Author ■



Hidenori OKAJIMA



Kenichi FURUDATE



Kenta FUKUDA

Enhancement of Multi-Body Simulation Data Sharing

Hideyuki YOSHIMOTO*
Bryan SLABACH*
Dean TENER*

Kyle PEARCE*
Charles GAGLIANO*

Todd RODRICK*
Hernan MERCADO-CORUJO*

ABSTRACT

Obtaining accurate input data from CAD (Computer Aided Design) geometry and reliable measurement data from suppliers is the main challenge of Multi-Body Simulation (MBS). Standards, processes and software were developed to enhance the process of creating an MBS model. A data extraction process from CAD was applied to extract only necessary data and combine the geometry information. Application of the Association for Standardization of Automation and Measuring Systems (ASAM) Open Data Services (ODS) format in conjunction with suppliers provided a common, reliable and transferable format. Key benefits are a time reduction for simulation feedback and independence from specific suppliers and MBS data structures.

1. Introduction

Safety requirements, environmental commitments, and the complexity of automotive vehicles are growing. However, development time, budgets, and manpower are typically limited. A key strategy in automotive development is to shorten development time, build fewer physical prototypes, and increase development in early stages. Digital manufacturing is a technology that every automotive maker has worked to develop during the past two decades. One of the improved digital methodologies is a numerical simulation called Computer Aided Engineering (CAE), which helps reduce product development cost and time while enhancing the safety, comfort, and durability of vehicles.

Finite Element Method (FEM) is one of the world's most widely used CAE solutions as implemented for crash simulation. FEM mainly originated from the need for solving small displacement elasticity and structural problems. In contrast, Multi-Body Simulation (MBS) was used in the early 1990s for large translational and rotational displacements, such as suspension stroke analysis (Fig. 1). More recently, MBS allows for predicting highly advanced phenomena such as elastic deformation⁽¹⁾ and rough road analysis⁽²⁾.

In spite of advances in this technology, MBS pre- and post-environments were still not well-organized. It was challenging to share input data between different

solvers due to the small number of analysts and lack of a common data format. Moreover, to identify parameters, MBS requires actual measurement data from suppliers. For example, bushings require nonlinear, frequency-dependent, and displacement-dependent measurement data. Since there was no standardized measurement format for suppliers, each supplier had a different file and data format. Consequently, each CAE analyst had to gather the necessary input data and modify it to fit their own MBS input data. This led to redundant requests to design engineers and suppliers. Due to increased data requirements and reduced development time, the need to establish a common, shareable data format became necessary (Fig. 2).

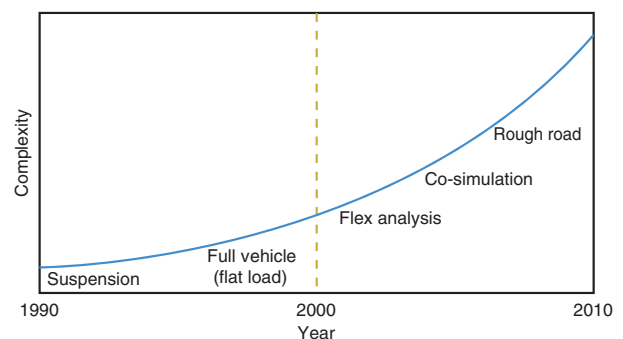


Fig. 1 Multi-body simulation technology

* HONDA R&D AMERICAS, INC.

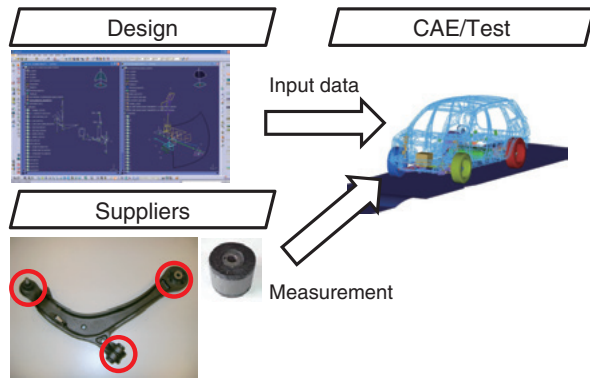


Fig. 2 Relation for Multi-body simulation input data

2. Construction of Multi-Body Simulation Data

MBS models are fundamentally constructed with geometric and non-geometric data (Fig. 3).

2.1. Geometric Data

Design engineers are responsible to confirm their own component designs and issue drawings for mass production. This is accomplished through component CAE and CAD applications during vehicle development. Design engineers can predict component characteristics by using component CAE at an early stage in development. Once design engineers evaluate their own concept after component performance is evaluated, CAD data is built and clearance with other components is confirmed. The purpose of component CAE is mainly static analysis such as suspension stroke. This static analysis is only dependent on the rough geometric information, but the full MBS model requires accurate mass, C.G. (Center of Gravity), and inertia. When a proper material with density is applied to the geometry, the MBS input data can be calculated. In addition to the shape of the parts, suspension/mount points and orientation information are required to satisfy the geometric data requirements of the MBS input model. All of this was accomplished by design engineers providing detailed CAD models.

2.2. Non-Geometric Data

Most measurement data comes from suppliers for

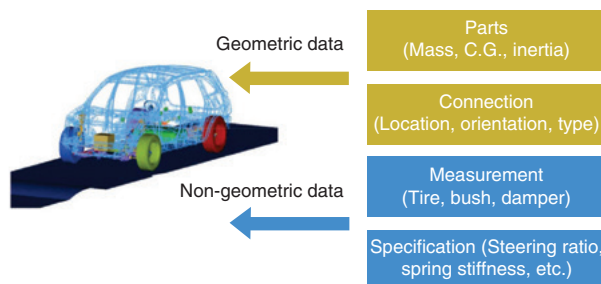


Fig. 3 Construction of MBS model

several components such as bushings, tires, and dampers. Chassis development demands several kinds of bushing data including orientation, displacement, frequency, preload condition and others. Suppliers may spend several weeks measuring this data. Previously, data requests to suppliers were inconsistent and the returned measurement data formats were unique for each one of the suppliers. Consequently, it took extra time to confirm the accuracy of the data and extract MBS numerical data.

3. Extraction Process of Design Data

3.1. Master Work Sheet

A Master Work Sheet (Fig. 4) contains all the MBS input information and lists each sub-system on a separate spreadsheet. Being independent of a specific CAE format, the Work Sheet is easy to understand and use and can be easily expanded to include other sub-systems.

Generally, the variety of suspension types is limited due to cost, performance, and manufacturing. Providing pre-defined templates for each suspension type enables a reduction in input workload. Through this research, design engineers defined the naming of the point and orientation; therefore, by using this naming rule and a predefined suspension type, each connection type can be automatically defined.

3.2. CAD Property / Naming Rule

To automate the process of geometric data extraction from the CAD data for use by MBS, the following three-part process is required:

- (1) Properties used for extraction
- (2) DPM/part construction
- (3) Naming rules of CAD hard point data

Property extraction from CAD requires standardization and correlation between test and design. The agreed upon properties include the part numbers and names that comply with the Honda Engineering Standards. In addition, material naming and mass values are required for determining the mass, C.G., and inertia values of the individual parts and MBS bodies.

The Digital Prototype Model (DPM) is used for the digital prototyping process before a prototype is actually built. The DPM construction further aids in automating data extraction. In order to guarantee accurate mass, C.G., and inertia values, each part should be constructed as a single

Front Suspension Connections											
Connection Name	Part No.	F - Section	Active Connection	Stem	Honda ID	Active Part	Active Node #	Passive Part	Passive Node #	Origin: P	
STABIL_R	F04	X	X	RLT1	STABIL_R	1011		DAMP	2011		
LOWER_ARM_R	F04	X	X	ALC1	LOWER_ARM_R	1011		KNuckle_R	2011		
SHOCKER_R	F04	X	X	SHO1	DAMP_LOWER_R	2010		KNuckle_R	4010		
STABIL_R	F04	X	X	RLT1	STABIL_R	1011		STABIL_R	2017		
STABIL_L	F04	X	X	RLT1	DAMP_LOWER_L	1009		STABIL_R	2009		
THE_RIDE_R	F04	X	X	RLT1	KNuckle_R	1010		THE_RIDE_R	2010		
THE_RIDE_L	F04	X	X	RLT1	THE_RIDE_L	1011		THE_RIDE_L	2011		
CONSTRAINT_FRONT_CIP	F04	X	X	SPS1							
SPS1_UPPER	F04	X	X	SPS1_UPPER1	FR SPRING			DAMP_UPPER_R			
SPS1_LOWER	F04	X	X	SPS1_LOWER1	FR SPRING			DAMP_LOWER_R			
WHEEL_CTR_R	F04	X	X	FLY1							

Fig. 4 Master work sheet format

body consisting of a single material. This method of part design should be used for all the vehicle parts required for simulation. Additionally, the construction and placement of the CAD hard point data file (Fig. 5) should also be standardized. It represents each critical control point, such as the bushing and ball joint centers of the suspension. This file's location was set to a specific area of the DPM to provide a consistent location for all users.

The CAD hard point data file should also follow a consistent construction format. This format is required for the automated extraction of the individual point locations. Additionally, the naming of these points was also standardized. This standardization was needed for the Master Work Sheet and involved design correlation to define the rules for point names and orientations. A sample of this standard is shown in Fig. 6.

Using required properties, a standardized structure, and naming rules allows for automation capability that leads to an increase in simulation efficiency.

3.3. CAD Extraction Process

Once the Master Work Sheet, CAD geometry, and CAD hard point data are prepared, necessary design data can be retrieved with a CAD Extraction tool (Fig. 7).

This tool can read both CAD and spreadsheet data. When the CAD point and orientation names are matched with the spreadsheet data, the point information of CAD

can be mapped to the spreadsheet. By using this process, large amounts of data can be extracted without errors. This operation is very simple in that the exact same CAD parts can be shown on the CAD extraction tool (Fig. 8). After completing every part assignment, the spreadsheet file can be exported.

Previously, the extraction process (Fig. 9) was

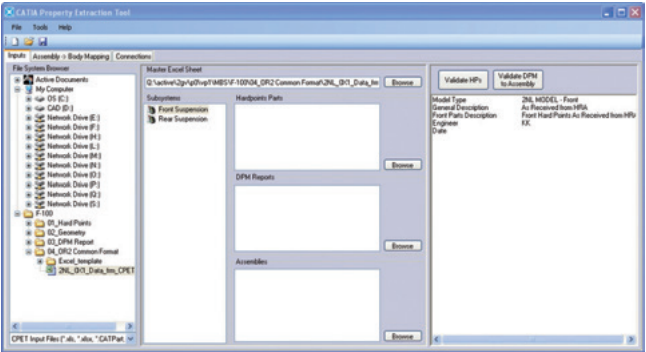


Fig. 7 CAD extraction tool (GUI)

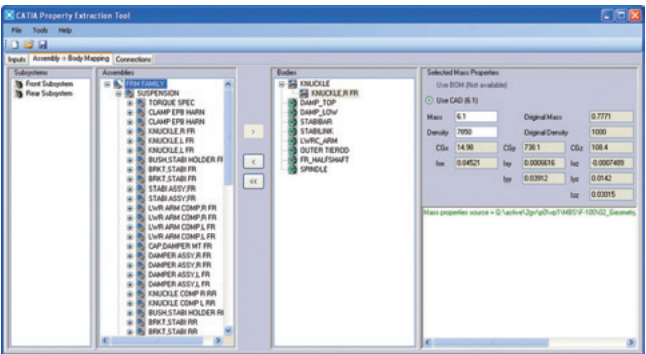


Fig. 8 CAD extraction tool (GUI)

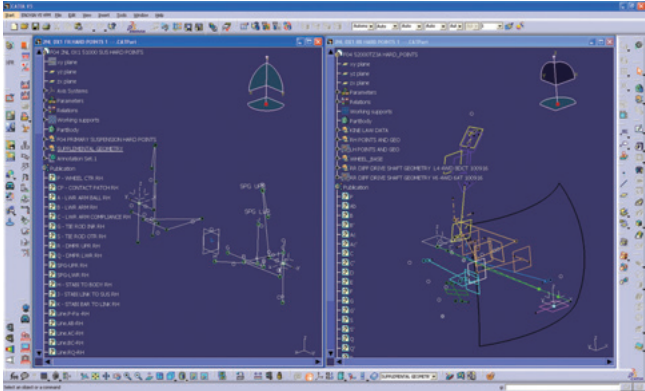


Fig. 5 Sample of CAD hard point data file

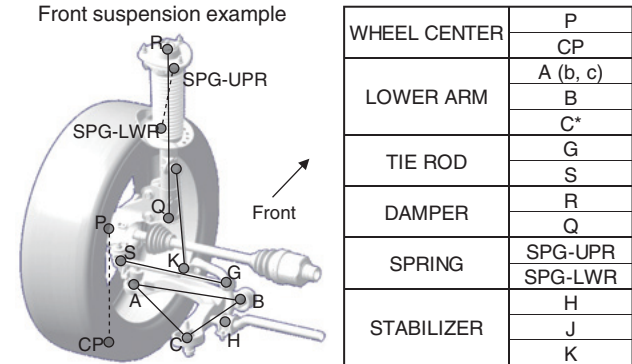


Fig. 6 Sample of suspension naming rule

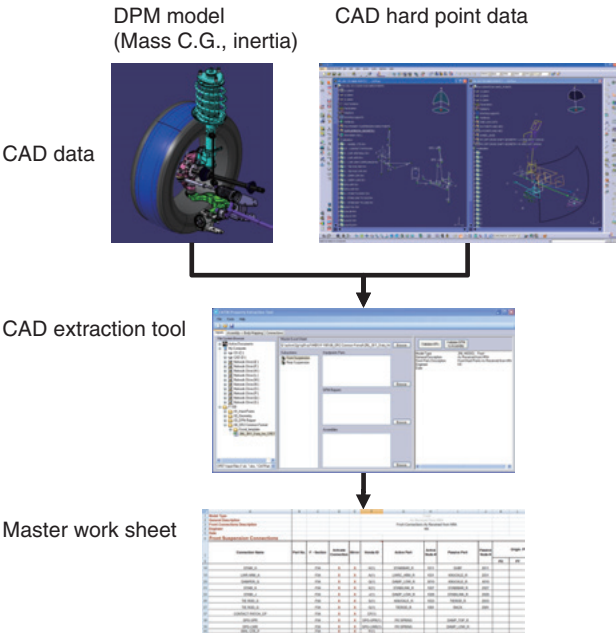


Fig. 9 CAD extraction process

4. Supplier Data Sharing

Similarly, it has been shown that an improved representation of the non-linear, dynamic characteristics of chassis bushings and mounts can provide improved MBS model accuracy. This is especially true in the areas of ride and NVH⁽³⁾. However, just like a single, computationally-intensive tire model is not always the best solution for a particular dynamics concern, the same can be said for bushing models. NVH prediction requires a component representation that is based on very small input displacements or forces. This small input is due to the nature of visco-elastic rubber behavior and is highly dependent on displacement and frequency of excitation⁽⁴⁾. Conversely, durability-related prediction requires characterization of the same parts over a much greater load range and the stiffness and damping characteristics are typically not as frequency dependent. Thus, for a single component, many different characterization tests in multiple degrees of freedom may be required. Maintaining a catalog of bushing data presents an opportunity for increasing efficiency not only for MBS but also at an enterprise level by decreasing the number of redundant tests requested of the suppliers and by making the measurement data more widely available and accessible.

The first step was to create a data template that captures the broadest array of possible test specifications with the smallest number of specified parameters (e.g., static, dynamic: min/max load, Hz, preload, etc.) (Fig. 10). This information should be clearly and easily communicated to suppliers in a package that maintains a record of the test component(s) and both the requested and actual test procedures. This template also helps to establish a minimal set of metadata to be captured and reported with every set of test results. The metadata aids in interpreting the data by having a record of the conditions under which the part data was recorded.

The next step of this project was to create a template for exchanging the actual measurement data between the OEM and suppliers (Fig. 11). This template is an extension of the Master Work Sheet and is extended to include the raw measurement data as well as functions for error-trapping and basic data confirmation, such as key reference data, plotting, rate calculations, and others.

Fig. 10 Simple request form

Fig. 11 Detailed specifications in request sheet

The ISO-22720 standards defined by ASAM-ODS describe the structure and rules for a relational data model for measurement data. ASAM was founded in December 1998 as an initiative of the German Car manufacturers Audi, BMW, Daimler, Porsche, and Volkswagen and is now in use at OEMs and suppliers globally. This standard was considered ideal for this project because it is hardware-, software-, and vendor-independent, and is supported by many of the largest and most common analysis software vendors. With the bushing test data templates matured and in frequent use by suppliers, the next step was to create a bushing-specific ADM.

5.1. Basis for Application Data Model

The purpose of this template is to set the basis for the development of a bushing-specific Application Data Model (ADM) that is used to form the basis of the database and Web interface to be used to exchange data with suppliers. Once the data is stored in the ADM, which is a customized version derived from the ASAM Open Data Services (ODS) base model, the data can be exchanged directly with the ODS database or exported into XML text files according to the ATF-XML standard.

5.2. Data Converter

Within ASAM ODS, there is defined a base model with a set of defined based elements and a set of base relations between them. An application model was developed specifically for bushings and specifies which elements and relations from the base model are actually in use (Fig. 12). The actual measured data is stored in data elements and related to test parameter elements (e.g., preload, duration, rate, etc.) as described in the Test Request Sheet. The Data Converter uses the application model to map the Excel data to an ODS ATFX file or ODS database.

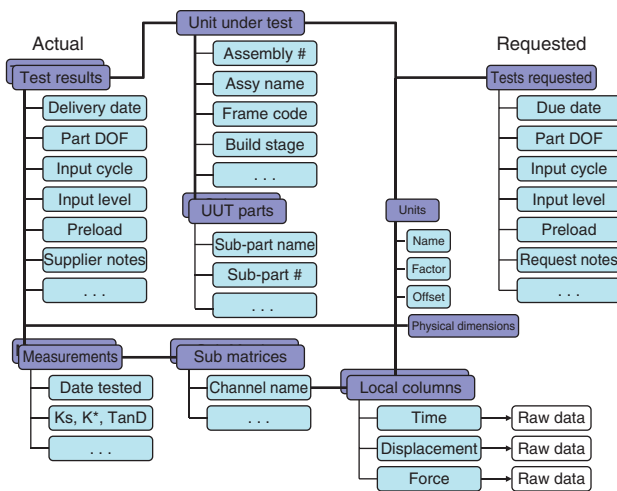


Fig. 12 Bushing application data model

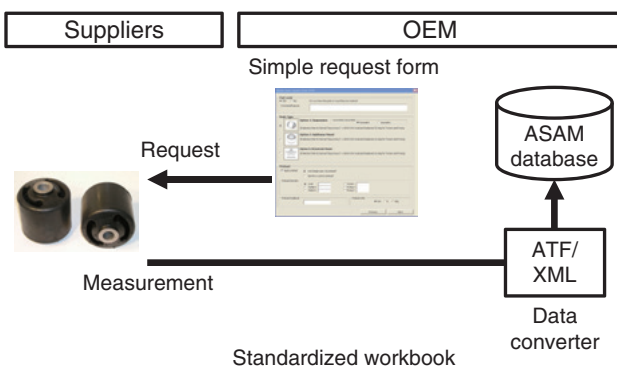


Fig. 13 Bush request flow

The biggest challenge is to create the ADM and the applications necessary to interpret and verify the data in the spreadsheets. After these steps are achieved, it is much easier to store the data into an ATF/XML file or directly upload it to an ASAM database.

Figure 13 shows the overall bush data sharing flow. This process was created through discussion and understanding between suppliers, design engineers, and Information Technology (IT) engineers.

6. Case Study

The process described in this paper was tested with a suspension subsystem. The new rules and tools reduced data gathering time from one week to one day. Three suppliers who supported the direction towards standardization were introduced to the test data standard format. In total, the time to request, receive, and review the data was reduced.

7. Conclusion

Through this research, an effective MBS data sharing process was developed. The key items can be summarized as follows:

- (1) Standardized CAD properties and measurement data are important in reducing the MBS feedback time by allowing for automated data extraction and MBS model creation.
- (2) A flexible and extendable data format for non-geometric data is necessary to accommodate future CAE methods.
- (3) While the initial trial and benefits were limited to suspension, it is clear that extending this process to the other parts and systems will further enhance the simulation-design development loop.

Finally, through this research, there were opportunities to meet with other OEMs to discuss their systems and methodologies related to MBS. This provided a better understanding of the different strategies and systems implemented across a range of OEMs' CAE maturity and scale.

References

- (1) Tsukagoshi, M., Shudo, T.: Analysis of Suspension Characteristics Considering the Elastic Deformation of Parts, Honda R&D Technical Review, Vol.15, No.1, p.195-202
- (2) Haga, H.: Evaluation of Road Load Simulation on Rough Road Using a Full Vehicle Model - Load Prediction for Durability Using a Tire Model -, JSAE Paper, No.123-07(20075868), p.21-24 (2007)
- (3) He, S., Singh, R.: Improved Estimation of Linear and Nonlinear Hydraulic Mount Models for Transient Responses, SAE Technical Paper, 2005-01-2411 (2005)

- (4) Diglio, P., Cibrario, V., Van J.: Accurate Harshness Modeling and Analysis Using Multi-Body CAE Models, JSAE Paper, No.74-05(20055476), p.9-14 (2005)

■ Author ■



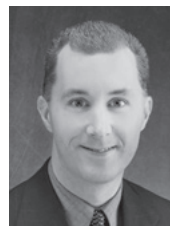
Hideyuki YOSHIMOTO



Kyle PEARCE



Todd RODRICK



Bryan SLABACH



Charles GAGLIANO



Herman MERCADO-CORUJO



Dean TENER

Research on Inertia Force Characteristics and Its Reduction in Extended Expansion Linkage Engine

Gaku NAOE*

Kota TOKUBI*

Sei WATANABE*

ABSTRACT

An extended expansion engine (EXlink engine) using a multiple linkage system was studied as a system that realizes an extended expansion thermal cycle that raises the expansion ratio higher than the compression ratio in order to enhance the thermal efficiency of a general-purpose engine. An initial prototype EXlink engine increased the EPA exhaust emission control operating mode fuel economy by 13% compared to a conventional engine with the same output, but the maximum value of the inertia force produced by the multiple linkage system also increased by 93% compared to the conventional engine. This is because the maximum piston acceleration during the expansion stroke of the initial EXlink engine is 2.7 times that of the conventional engine.

A new linkage layout that controls the maximum piston acceleration was investigated to reduce the inertia force. By using a linkage configuration that allots a longer crank angle to the expansion and exhaust strokes, the piston acceleration is smoothed, enabling reduction of the inertia force. Compared to the initial linkage layout, the newly designed linkage layout reduced the calculated maximum piston acceleration by 42% and the calculated maximum inertia force value by 61%, and the expected vibration reduction effects were also confirmed in actual engine tests.

1. Introduction

The compact general-purpose engines mounted in agricultural and construction equipment have outputs corresponding to the work load, so they are often used at medium to high loads. Amid the growing concern over global warming and depletion of fossil fuels, there are demands to enhance the thermal efficiency of these compact general-purpose engines. However, many compact general-purpose engines are air-cooled and used in the stationary state, so increasing the compression ratio can lead to knocking, and there is a limit to how much the thermal efficiency can be enhanced simply by increasing the compression ratio.

The authors focused on an extended expansion thermal cycle (Atkinson cycle) in which the expansion ratio exceeds the compression ratio as a means of enhancing the thermal efficiency of compact general-purpose gasoline engines. Use of the piston system using a multiple linkage system (EXlink) shown in Fig. 1 was proposed as a means of realizing this extended expansion thermal cycle⁽¹⁾. This system realizes an extended expansion thermal cycle by setting a longer expansion and exhaust stroke length compared to the intake and compression strokes as shown in Fig. 2. Operation tests using an actual prototype engine showed that the EXlink engine has the same friction as the

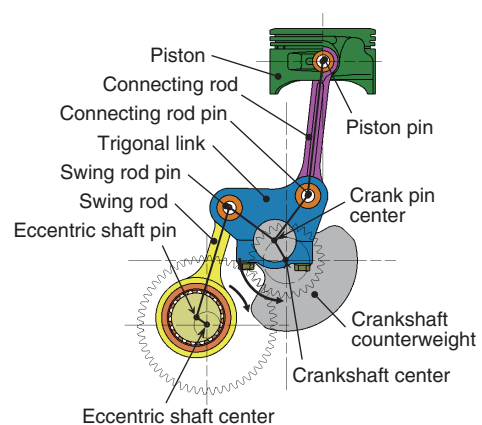


Fig. 1 Extended expansion linkage system (EXlink)

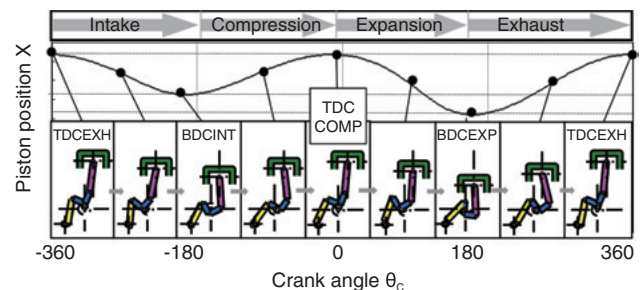


Fig. 2 Piston movement through one cycle

* Power Products R&D Center

conventional engine, and enhances fuel economy in the EPA emission control operating mode⁽²⁾ by 13% compared to a conventional engine⁽³⁾. However, operation tests using this initial EXlink engine clarified that vibrations originating in the inertia excitation force are larger than those of a conventional engine. This is because the EXlink engine system differs greatly from a conventional engine.

Recently, user requests to reduce engine vibration are increasing not only for cars and motorcycles, but also for work equipment that uses compact general-purpose engines. As a result, regulations concerning work equipment vibration are being enacted in various countries on a trial basis⁽⁴⁾. This makes the reduction of engine vibration an important technical issue for practical application of the EXlink engine. This paper presents numerical calculations describing an approach to reducing the inertia excitation force using a new linkage layout, and reports on verification of the effects by actual engine operation tests.

2. Inertia Excitation Force Characteristics

2.1. Dynamic Characteristics of Multiple Linkage System

This chapter describes the formulation of equations for the acceleration and inertia force of each component part of the extended expansion engine using multiple linkage system to investigate the dynamic characteristics of the entire system. Figure 3 shows the main dimensional parameters of the multiple linkage system, and Table 1 gives the specifications of the initial EXlink engine proposed in a previous report⁽³⁾ and the conventional engine used for comparison. The stated quantities of the various parts of the multiple linkage system are expressed by the following equations, with the crank angle as $\theta_c^{(1)}$.

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

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

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

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

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

Equation (5) expresses the driving of the eccentric shaft by the crankshaft, and $\eta = -1/2$ is given in the case of gear reduction ratio of 1/2. This results in a phase difference with a cycle of $\theta_c = 720^\circ$ between the crankshaft and the eccentric shaft, which produces a difference in the length of each stroke in piston operation. By applying an appropriate initial phase difference γ , the compression top dead center and the intake/exhaust top dead center can be set to the same piston position. This enables obtaining an extended expansion thermal cycle in which the expansion and exhaust strokes are longer than the intake and compression strokes, and the expansion ratio is larger than the compression ratio.

Table 1 Engine specifications

			EXlink engine	Conventional engine
Engine type			Air cooled, 4 stroke, single cylinder, horizontal shaft	
Bore		[mm]	64	68
Stroke	Intake /Compression	[mm]	42.1	45
	Expansion /Exhaust	[mm]	63.3	
Displacement	Intake /Compression	[cm³]	135.4	163.4
	Expansion /Exhaust	[cm³]	203.6	
Compression ratio			8.5	8.5
Expansion ratio			12.3	
Fuel			Gasoline	
Maximum power		[kW]	3.2	3.1
Valve mechanism			OHC	OHV

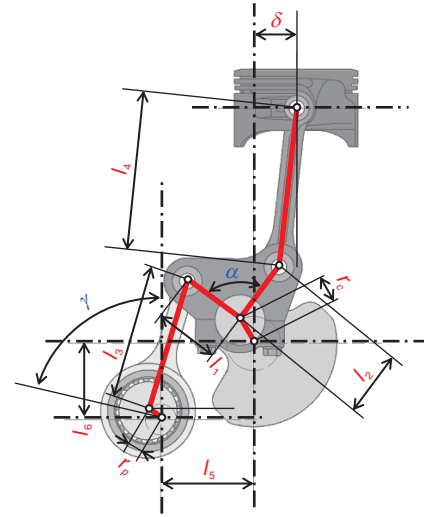


Fig. 3(a) Dimensional parameters of multiple linkage system

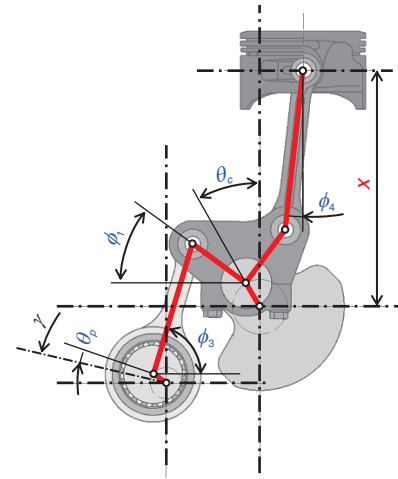


Fig. 3(b) Characteristic state quantities of multiple linkage system

The acceleration of each component part of the multiple linkage system can be obtained by solving the following relational equations, which are derived by the second-order differential of Eq. (1) to Eq. (4) relative to time.

$$\begin{aligned} & \frac{d^2 x}{dt^2} + l_4 \left(\frac{d\phi_4}{dt} \right)^2 \cos \phi_4 + l_4 \frac{d^2 \phi_4}{dt^2} \sin \phi_4 \\ &= -l_2 \left(\frac{d\phi_1}{dt} \right)^2 \cos(\phi_1 + \alpha) - l_2 \frac{d^2 \phi_1}{dt^2} \sin(\phi_1 + \alpha) \\ &+ r_c \frac{d^2 \theta_c}{dt^2} \cos \theta_c - r_c \left(\frac{d\theta_c}{dt} \right)^2 \sin \theta_c \end{aligned} \quad (6)$$

$$\begin{aligned} & r_c \frac{d^2 \theta_c}{dt^2} \sin \theta_c + r_c \left(\frac{d\theta_c}{dt} \right)^2 \cos \theta_c \\ &+ r_p \left(\frac{d\theta_p}{dt} \right)^2 \cos(\theta_p + \gamma) - r_p \frac{d^2 \theta_p}{dt^2} \cos(\theta_p + \gamma) \\ &= -l_1 \left(\frac{d\phi_1}{dt} \right)^2 \cos \phi_1 - l_1 \frac{d^2 \phi_1}{dt^2} \sin \phi_1 \\ &- l_3 \left(\frac{d\phi_3}{dt} \right)^2 \sin \phi_3 - l_3 \frac{d^2 \phi_3}{dt^2} \sin \phi_3 \end{aligned} \quad (7)$$

Next, the inertia force acting on the multiple linkage system was evaluated. Equations describing the inertia force of the entire multiple linkage system were formulated by viewing the entire system as consisting of multiple concentrated masses⁽⁵⁾. The equivalent mass at the node of each part of the multiple linkage system can be calculated from the gravity center and distance to the node of each part, and this equivalent mass was allocated to the six concentrated masses shown in Fig. 4 and Table 2.

The inertia force acting on each concentrated mass is expressed as follows as a function of the crank angle by multiplying the equivalent mass of these concentrated masses with the previously obtained acceleration of each part.

$$\begin{aligned} Fix_k(\theta) &= -m_k \cdot Ax_k \\ Fiy_k(\theta) &= -m_k \cdot Ay_k \end{aligned} \quad (8)$$

In addition, the size of the force is given by the following equation.

$$Fi_k(\theta) = \sqrt{Fix_k(\theta)^2 + Fiy_k(\theta)^2} \quad (9)$$

In this manner, equations describing the dynamic characteristics of the multiple linkage system were obtained.

2.2. Inertia Excitation Force Characteristics

The values of the summation of inertia force and maximum summation of inertia force described below were used as indices to evaluate the inertia excitation force characteristics of the entire system. The summation of

Table 2 Equivalent mass of linkage system

Equivalent mass	Mass of component parts	Corresponding mass
m_1	Piston, Piston ring, Piston pin	Connecting rod (small end)
m_2	Connecting rod pin	Connecting rod (big end) Trigonal link (m_{T2})
m_3	Crank pin	Trigonal link (m_{T3})
m_4	Swing rod pin	Trigonal link (m_{T4}) Swing rod (small end)
m_5	Eccentric shaft pin	Swing rod (big end)
m_6	Crankshaft counter weight	

inertia force is obtained by adding the inertia force of each concentrated mass, given by Eq. (10), for each component of the entire system at an arbitrary crank angle, and is a vector quantity having the components described by the following equations.

$$\begin{aligned} Fix(\theta) &= \sum_k Fix_k(\theta) \\ Fiy(\theta) &= \sum_k Fiy_k(\theta) \end{aligned} \quad (10)$$

In addition, the size of the vector quantity is given by the following equation.

$$Fi(\theta) = \sqrt{Fix(\theta)^2 + Fiy(\theta)^2} \quad (11)$$

The maximum $Fi(\theta)$ value during one cycle (720 degCA) is called the maximum summation of inertia force.

The summation of inertia force is determined according to the engine speed, the system dimensions, and the mass of each part, and the size and characteristics of the inertia excitation force generated in the system can be easily evaluated by analyzing the change in the summation of inertia force relative to the crank angle. This value can also be defined for the piston and crankshaft system of a conventional engine.

The counterweight of the conventional engine used to compare the characteristics was the mass that results in the minimum calculated value for the maximum summation of

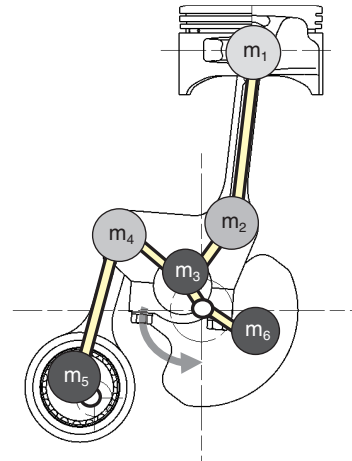


Fig. 4 Equivalent mass system

inertia force noted above. Figure 6 uses contour lines to show the size of the maximum summation of inertia force when the counterweight specifications given in Fig. 5 are varied in the test engine. The mass and the gravity center phase of the test engine counterweight were set to achieve the minimum calculated value for the maximum summation of inertia force.

Figure 7 expresses the summation of inertia force vector relative to the crank angle as a vector with the center of the crank shaft as the origin, and shows the track of the vector tip (Local curve)⁽⁶⁾. The engine speed here is 3000 rpm, which has a high frequency of use for general purpose engines. The results in this figure show that the track for the test engine traces a different shape and the maximum summation of inertia force is 93% higher than that of the conventional engine.

Next, FFT analysis was performed for the inertia force component in the F_x (cylinder axis) direction of the test engine, given by Eq. (10), relative to the crankshaft rotational speed. Figure 8 shows these results. The test engine generated inertia force components of orders in 0.5 increments relative to the crankshaft revolutions, which are not generated in a conventional piston crankshaft

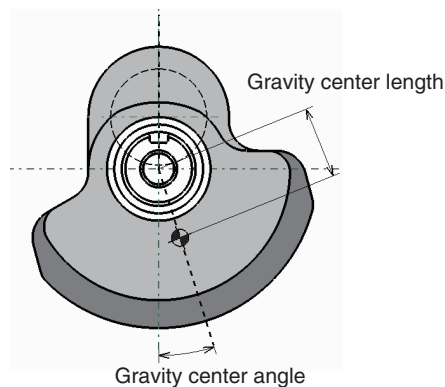


Fig. 5 Crankshaft gravity center position

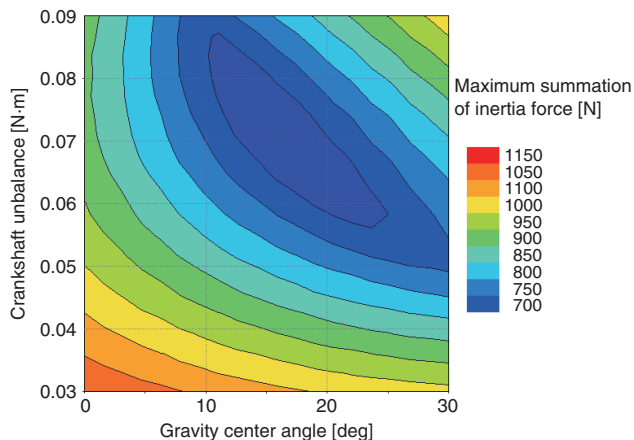


Fig. 6 Specification of counter weight and maximum summation of inertia force

system. This is because the intake and compression stroke lengths differ from the expansion and exhaust stroke lengths, which generates inertia force components with a cycle of two crankshaft revolutions. In addition, these results show that while the first-order component decreases compared to the conventional engine, the maximum summation of inertia force increases due to the second- and higher-order components. These characteristics reduce the effectiveness of rotation balancers and other measures, so an approach that concentrates the components into the main orders is thought to be better.

Figure 9 shows the results of analyzing the timing that increases the higher-order inertia force by breaking down the 2.5th- and higher-order components of the summation of inertia force in the F_x direction by concentrated mass. This shows that there are three high and low peak values from the expansion stroke to the exhaust stroke, and that the reciprocating-part inertial mass m_1 contributes greatly to each of these values. The factors for this are presumed to be that the piston acceleration of the test engine changes

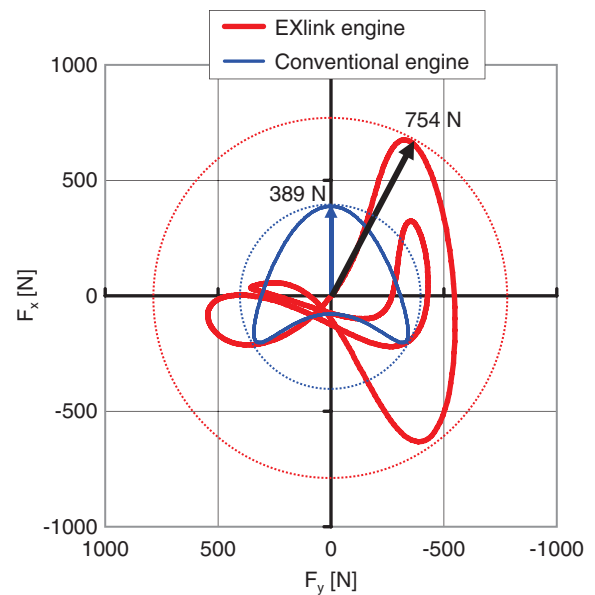


Fig. 7 Local curve

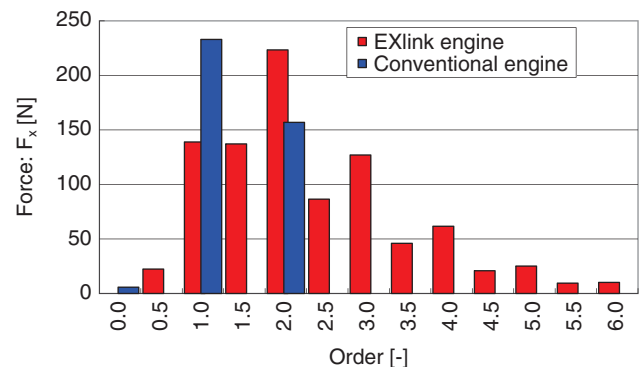


Fig. 8 FFT Analysis result (F_x)

suddenly following 90 degCA ATDC, and the maximum value is 2.7 times that of the conventional engine as shown in Fig. 10. A possible reason for the excessive piston acceleration of the test engine is that the intervals between the top and bottom dead centers of the multiple linkage system are not necessarily 180 degCA, and have unequal lengths. The expansion stroke period of the test engine is 173 degCA, which is 7 degCA shorter than the 180 degCA of the conventional engine. This means that in the test engine, the piston moves over the long expansion stroke in a short time, which is thought to result in a higher maximum acceleration.

In addition, the concentration mass m_2 that corresponds to the big end of the connecting rod also has a comparatively high degree of contribution. As shown

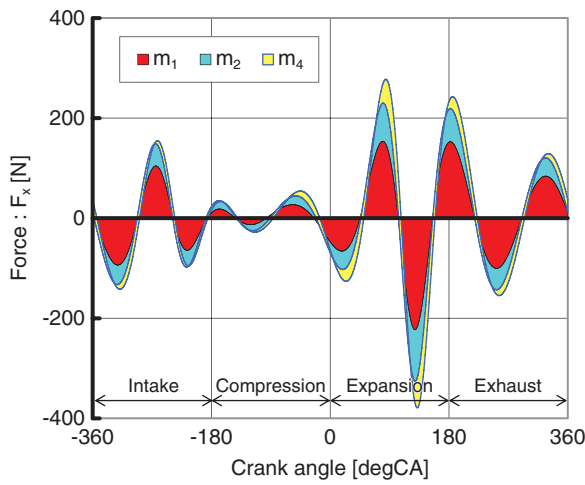


Fig. 9 Summation of inertia force for 2.5 th and higher orders (F_x)

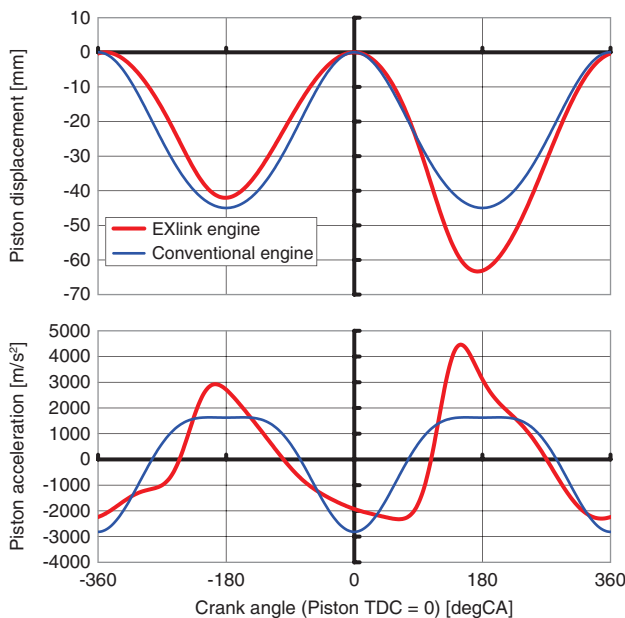


Fig. 10 Piston acceleration and displacement

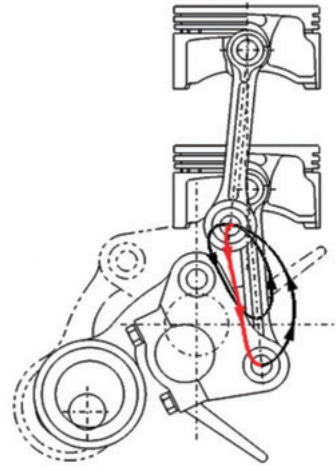


Fig. 11 Connecting rod behavior

in Fig. 11, the big end of the connecting rod traces an elliptical track lengthwise in the cylinder axis direction. For this reason, both the piston and the connecting rod mainly perform reciprocating motion in the cylinder axis direction. Therefore, the inertia force generated from m_1 and m_2 is thought to have a large influence on the piston acceleration.

3. Reduction of Inertia Excitation Force

3.1. Reduction of Piston Acceleration by New Linkage Layout

The authors thought it might be possible to reduce piston acceleration by making use of the above-noted characteristic that the intervals between the top and bottom dead centers of the multiple linkage system have unequal lengths. This was done by applying a longer crank angle to the expansion and exhaust strokes, which have longer stroke lengths.

Figure 12 and Table 3 show the newly designed

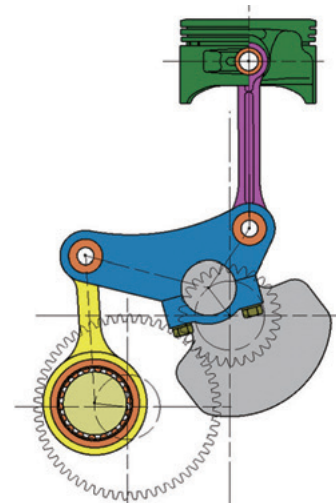


Fig. 12 Newly designed linkage system

Table 3 New and former linkage engine specifications

			Newly designed linkage engine		Former designed linkage engine
Maximum piston acceleration		[m/s²]	2575	-42%	4460
Maximum summation of inertia force		[N]	293	-61%	754
Expansion stroke term		[degCA]	190	10%	173
Bore		[mm]	64		
Stroke	Intake /Compression	[mm]	38.8	-8%	42.1
	Expansion /Exhaust	[mm]	55.7	-12%	63.3
Compression ratio			8.5	-	8.5
Expansion ratio			11.8	-4%	12.3

linkage layout. Compared to the initial linkage layout, this new linkage layout that aims to reduce vibration shortens the intake and compression strokes by -8% and the expansion and exhaust strokes by -12%, and extends the crank angle of the expansion stroke, which has a long stroke length, by 17 degCA to 190 degCA. Figure 13 shows the calculated piston acceleration at 3000 rpm. The maximum acceleration is 2575 m/s², which is a reduction of 42% compared to the initial linkage layout. In addition, the difference in the peak piston acceleration values of each stroke is also controlled, indicating that the reciprocating-part acceleration of each stroke has been smoothed.

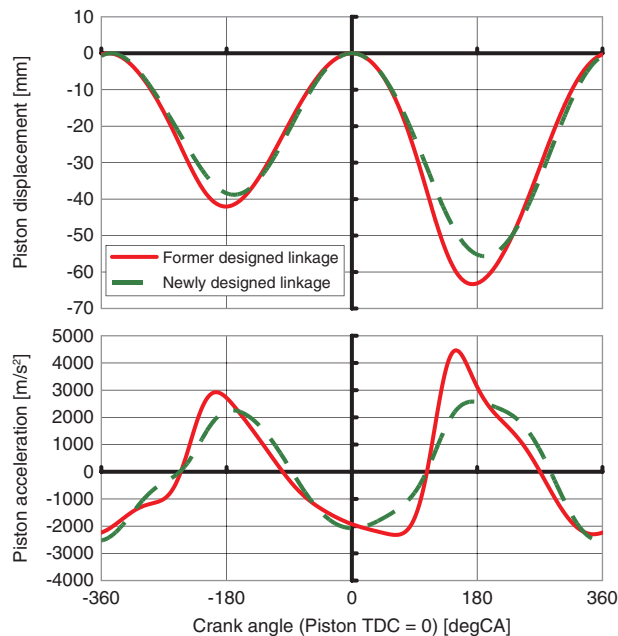


Fig. 13 Piston acceleration and displacement (New and former linkage system)

Figure 14 compares the summation of inertia force of the newly designed linkage layout with that of the initial linkage layout. The maximum summation of inertia force is 293 N, which is a reduction of 61% compared to the initial linkage layout. Figure 15 shows the FFT analysis results of component of the summation of inertia force in the cylinder axis direction, F_x . The newly designed linkage layout does not exhibit the rapid change in acceleration during the expansion stroke seen with the initial linkage layout, which indicates that the second- and higher-order components have been reduced.

3.2. Verification of Effects by Actual Engine Operation Test

The vibration of an actual engine was measured to confirm the vibration reduction effects of the newly designed linkage layout. In this test, the engine shown in Fig. 16 was operated no-load under Free-Free conditions hoisted via rubber, at the same 3000 rpm engine speed as used in the calculation conditions. The engine acceleration was measured by an acceleration sensor located on the

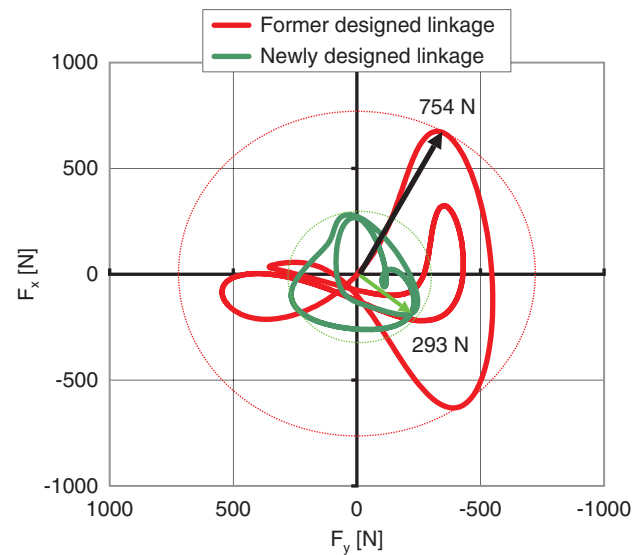


Fig. 14 Local curve (New and former linkage system)

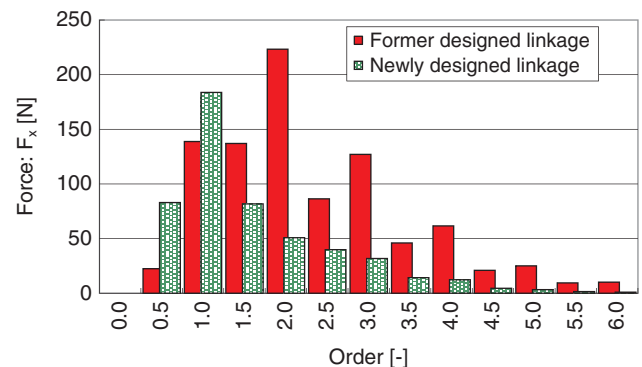


Fig. 15 Frequency characteristics

engine mount position as shown in Fig. 16, and the cylinder axis direction component was used as the typical value. The measurement time was 10 s, and the various order components relative to the crankshaft revolutions were calculated by FFT analysis.

Figure 17 shows the actual engine operation test results. The acceleration of all first- and higher-order components in the newly designed linkage layout decreased, and the overall value decreased by 43.5% compared to the initial linkage layout.

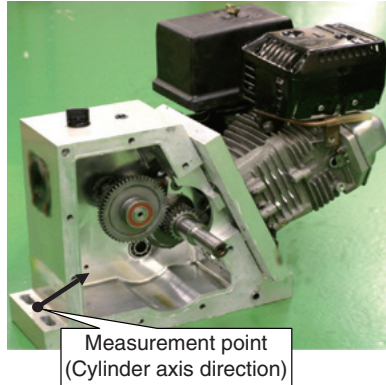


Fig. 16 Acceleration measurement point

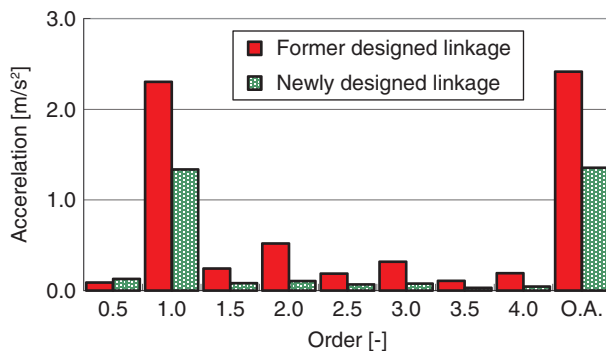


Fig. 17 Measurement results

4. Conclusion

The following conclusions were reached regarding the inertia excitation force characteristics of an extended expansion engine using multiple linkage system and methods for reducing this inertia excitation force.

- (1) The maximum summation of inertia force of the initially designed multiple linkage system was 93% higher than that of a conventional engine with the same output. This is because the crank angle during the expansion stroke, which has a long stroke length, was short at 173 degCA, and the maximum piston acceleration during the expansion stroke was 2.7 times that of a conventional engine.
- (2) The maximum calculated piston acceleration value can

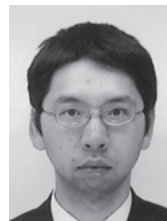
be controlled and the change in acceleration during each stroke can be smoothed by applying a longer crank angle to the expansion and exhaust strokes, which have long stroke lengths.

- (3) A newly designed linkage layout with an expansion stroke period of 190 degCA based on the above approach reduces the calculated maximum piston acceleration by 42% and the calculated maximum summation of inertia force by 61% compared to the initial linkage layout.
- (4) A prototype extended expansion engine was made using the newly designed linkage layout, and actual engine operation tests were performed. The acceleration component in the cylinder axis direction was 43.5% lower than that of the initial linkage layout, proving that linkage design based on the above approach is an effective means of reducing vibration.

References

- (1) Watanabe, S., Kono, S., Kurata, M., Koga, H.: Research on Extended Expansion General-Purpose Engine -Theoretical Analysis of Multiple Linkage System and Improvement of Thermal Efficiency-, Honda R&D Technical Review, Vol. 18, No. 1, p. 81-87
- (2) Environmental Protection Agency: Title 40 CFR, Part 90, Subpart E, p. 219-247 (2000)
- (3) Koga, H., Watanabe, S.: Research on Extended Expansion General-Purpose Engine (Part2) -Heat Release and Brake Performance-, Honda R&D Technical Review, April, 2007, p. 53-59
- (4) 2002/44/EC, Official Journal of the European Communities: L177/13
- (5) Naoe, G., Watanabe, S.: Research on Extended Expansion General-Purpose Engine -Characteristic of Vibration-, SAE Paper, 2008-32-0012 (2008)
- (6) Basshuysen, R. V., et al; Modern Engine technology from A to Z, SAE International, p. 42 (2007)

■ Author ■



Gaku NAOE



Kota TOKUBI



Sei WATANABE

Construction of Knock Detection Logic by Pattern Recognition Using Short-time Fourier Transform

Kenji AKIMOTO*

Hirotaka KOMATSU*

Atsushi KURAUCHI*

ABSTRACT

The project discussed in this paper focused on the frequency attenuation characteristic of knock and mechanical noise in order to develop a more accurate method of distinguishing between the two phenomena. Short-time Fourier transform (STFT), which makes it possible to identify changes in frequency over time, was studied as a method of analyzing this characteristic. The duration of knocking is 80-90 deg in terms of crank angle. The frequencies of the multiple vibration modes that knock generates decline as time elapses, with attenuation greater in the high-frequency vibration modes, until finally only the 1,0th mode remains. Mechanical noise occurs over a wide frequency band, and its attenuation is rapid. Based on these results, the concept of time was introduced in order to help enable the attenuation of the frequencies to be identified, and a knock detection logic that judges knock on the basis of pattern recognition was developed. This logic is able to distinguish knock and mechanical noise with a high degree of accuracy even when the timing and frequency of the two phenomena match. As a result, it has been possible to increase robustness in relation to mechanical noise in comparison to conventional knock detection systems that judge knock based on the intensity of the frequency at which it occurs, helping to ensure stable functioning of the system even when mechanical noise occurs. This has made it possible to set the threshold value for judgment of knock lower than is the case in conventional systems, contributing to the reduction of knock noise.

1. Introduction

In order to increase the fuel efficiency of gasoline engines, high compression ratios are being employed in today's engines, resulting in an expansion of the range in which knock occurs, and at the same time a diverse range of devices are coming into use. Because many of these devices generate mechanical noise, a high-accuracy method of distinguishing mechanical noise is essential for knock detection systems using vibration sensors. It is challenging for conventional knock detection systems to distinguish between knock and mechanical noise when the frequency, intensity, and timing of the two phenomena match. A great deal of research focusing on the analysis of knock has been conducted in order to respond to this issue⁽¹⁾⁻⁽⁷⁾. Methods employed to assist in preventing an effect from mechanical noise in knock detection systems to date include the restriction of the range of detection⁽⁸⁾, the use, as a base, of a non-knock signal determined by deviation from a moving average obtained when no knock is occurring, the use of statistical methods⁽⁹⁾, and the use of trends in changes in sensor output⁽¹⁰⁾. However, because the mechanical noise generated by increasingly diverse devices, as typified by valve seating noise, displays a variety of frequencies,

intensities, and timings, it is challenging to detect knock using conventional methods.

The project discussed in this paper therefore attempted to enhance the method of distinguishing between knock and mechanical noise by newly introducing the concept of time and developing a knock detection logic employing pattern recognition. This paper provides an overview of this knock detection logic, considers methods employed in order to increase the accuracy of detection, and discusses the results of testing in an actual engine.

2. Issues of Knock Detection

Figure 1 shows a comparison of frequency and spectrum intensity characteristics when fast Fourier transform (FFT) processing is applied to non-resonance knock sensor output for knock and mechanical noise. When the frequencies and spectrum intensities match at the same timing, as shown by the circled areas in the graph, conventional knock detection logic employing band pass filters cannot distinguish between the knock and mechanical noise. It is therefore necessary to construct a knock detection logic that uses a method other than simple knock judgment when frequencies and spectrum intensities match.

* Automobile R&D Center

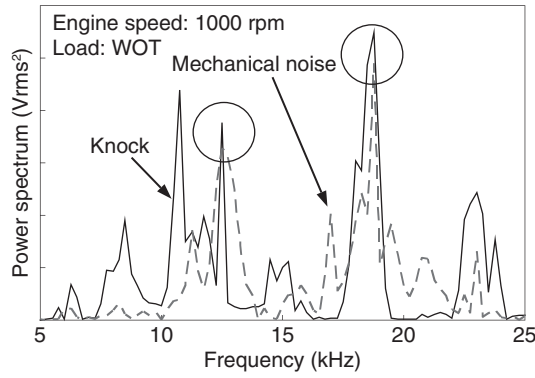


Fig. 1 Comparison of FFT processing results for knock and mechanical noise

3. Characteristics of Knock and Mechanical Noise

The difference in frequency attenuation characteristics between knock and mechanical noise was focused on in order to develop a more accurate method of distinguishing between the two phenomena. Short-time Fourier transform (STFT), which makes it possible to grasp changes in frequency over time, was studied as a potential method of analyzing these characteristics. STFT processing was applied to non-resonance knock sensor output when knock and mechanical noise were occurring, with results indicating that it was possible to clarify the difference between the two cases. The results of each analysis are discussed below.

3.1. Results of Knock Analysis

3.1.1. Duration of knock vibration

Figure 2 shows the non-resonance knock sensor output when knock occurs. As shown here, the amplitude of the knock waveform steadily declines with the passage of time. Figure 3 shows the results obtained when STFT processing is applied to the non-resonance knock sensor output when knock occurs. The X axis shows the crank angle, and the Y axis shows the frequency. The spectrum intensity is expressed by shade of the colors, with darker colors indicating stronger spectrum intensity.

Figure 3 indicates that knock vibration continues from the point of commencement of knock to approximately

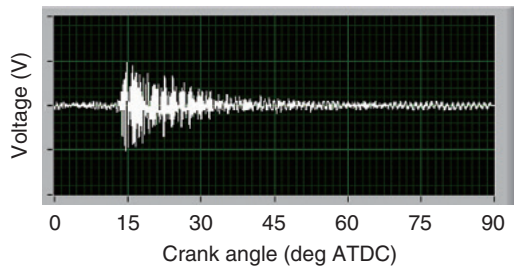


Fig. 2 Non-resonance knock sensor output when knock occurs

Table 1 Comparison of knock vibration duration for three different types of engine

Engine specifications	Displacement (cm ³)	1,339	1,799	2,354
	Bore x Stroke (mm)	73 x 80	81 x 87.3	87 x 99
Engine speed (rpm)	Intake manifold pressure (kPa (G))	Knock vibration duration (deg)		
1000	0 (WOT*)	95	82	90
	-20	90	85	92
2000	0 (WOT)	92	80	84
	-20	90	85	90
3000	0 (WOT)	89	90	91
	-20	89	83	89

* Wide-Open Throttle

ATDC 90 deg. This is thought to be because repeated air column vibration occurs due to the knock pressure wave. Table 1 compares the duration of knock vibration measured on a crank-angle basis using engines of differing displacements. These results indicate that knock vibration continued from the point of commencement of knock to approximately 80 to 90 deg, irrespective of displacement or load.

3.1.2. Phenomenon of frequency drop in vibration mode

Figure 4 shows the in-cylinder vibration modes generated by knock, and Eq. (1) shows Draper's equation⁽¹¹⁾. Figure 5 shows the results of STFT processing of the non-resonance knock sensor output when knock occurs, and frequencies calculated using Eq. (1) (shown as broken lines). Figure 5 indicates that the multiple vibration modes shown in Fig. 4 occur in the results of STFT processing of the non-resonance knock sensor output when knock occurs. The speed of sound when knock occurs in Fig. 5 was calculated from in-cylinder pressure using an equation of state as approximately 1000 m/s. Table 2 shows the results of obtaining the frequency of each vibration mode from these values using Eq. (1). The vibration mode frequencies shown in Table 2 display a high correlation with the frequencies obtained as a result of STFT processing of the sensor output when knock occurs shown in Fig. 5. In addition, the vibration mode frequencies in the STFT results display a tendency to decline as the expansion stroke progresses. From Eq. (1), this can be conjectured to represent a decline in

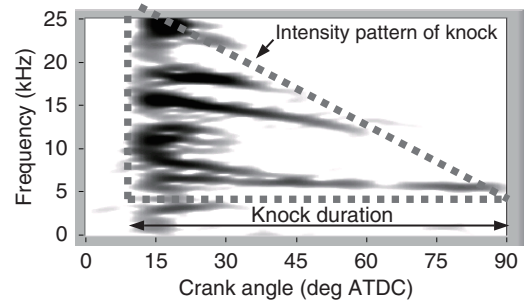


Fig. 3 STFT results when knock occurs

Table 2 Vibration mode frequencies immediately after knock occurs

Calculation condition: C=1000 m/sec D=87 mm					
	1,0 mode	2,0 mode	0,1 mode	3,0 mode	1,1 mode
Frequency (kHz)	6.8	11.3	14.2	15.6	19.8

the frequency of each vibration mode due to a decline in the speed of sound. The speed of sound was therefore determined from the output of a pressure sensor during the expansion stroke, and Eq. (1) was used to determine frequencies. The results of this procedure displayed the identical tendency as the results of STFT processing. Based on the results discussed above, it was judged that the frequencies of the multiple vibration modes generated by knock tend to decline because the speed of sound declines as the expansion stroke progresses.

$$f_{m,n} = \beta_{m,n} \frac{C}{\pi D} \quad (1)$$

Here,

$f_{m,n}$: Frequency

$\beta_{m,n}$: Vibration mode coefficient

C : Speed of sound

D : Bore diameter

3.1.3. Attenuation characteristics of vibration modes

The frequency of each vibration mode becomes attenuated as the expansion stroke proceeds, and at close to a crank angle of 90 deg only the 1,0th mode remains. This is thought to be because the pressure waves in the high-

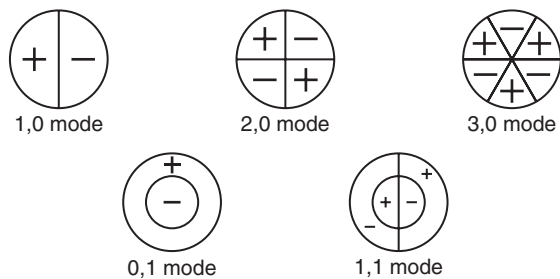


Fig. 4 In-cylinder vibration modes

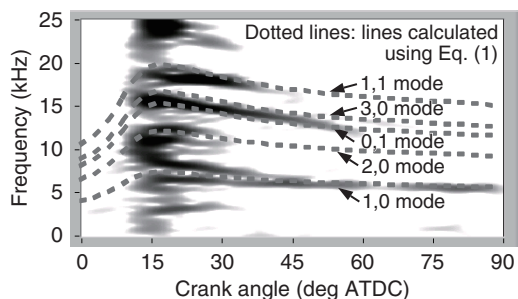


Fig. 5 Comparison of STFT results and curves calculated using Draper's equation

frequency vibration modes combine and become attenuated due to reflection and interference inside the cylinder.

3.2. Results of Analysis of Mechanical Noise

The major factor in most mechanical noise is impact. Typical mechanical noise includes the valve seating noise generated when intake and exhaust valves close. Figure 6 shows the non-resonance knock sensor output during valve seating. Figure 7 shows STFT processing results for valve seating noise, indicating that valve seating noise is generated over a wide frequency band. In addition, valve seating noise attenuates rapidly, and the duration of the vibration is approximately 1/5 that of knock, as discussed above. This is because the closing action of the valve, which is the vibration source for valve seating noise, undergoes forced convergence due to the valve train mechanism. This characteristic does not change even when the timing at which seating noise is generated changes due to the operation of continuously variable valve timing mechanisms.

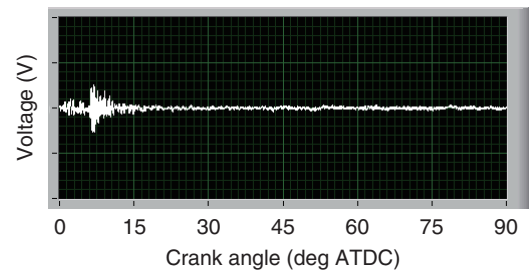


Fig. 6 Non-resonance knock sensor output when valve-seating noise occurs

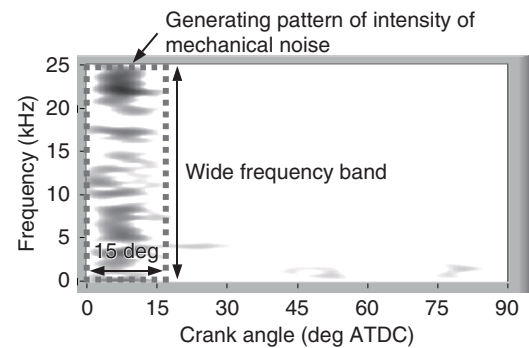


Fig. 7 STFT processing results for valve-seating noise

4. Methods of Distinguishing between Knock and Mechanical Noise

Based on the characteristics of knock and mechanical noise, a concept of time was newly introduced to make it possible to determine the attenuation of frequencies, and a new knock detection logic employing pattern recognition to identify knock was developed.

As shown in Fig. 3, because the results of STFT processing of sensor output when knock occurs demonstrate that the level of attenuation is high for high frequencies, and that in terms of crank angle the 1,0th mode continues over approximately 90 deg, the spectrum intensity displays a triangular distribution. In addition, as shown in Fig. 7, because the results of STFT processing of sensor output when mechanical noise occurs indicate that the noise occurs over a wide frequency band and its spectrum intensity attenuates rapidly, the spectrum intensity displays a rectangular distribution. These results are thought to indicate that the spectrum intensity distributions in the STFT results for knock and mechanical noise can be converted into patterns, and the two phenomena can be distinguished based on the differences between these patterns. Figure 8 shows pattern diagrams of the spectrum intensity distributions for knock and mechanical noise.

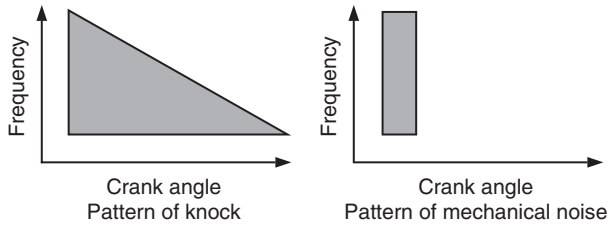


Fig. 8 Knock and mechanical noise intensity patterns

5. Knock Detection Logic

Figure 9 shows the processing flow for the STFT knock detection logic (“STFT logic” below).

- (1) STFT processing is applied in order to introduce the concept of time to the knock sensor output
 - (2) The results are stored in the frequency and crank angle map
 - (3) The noise component is removed in order to produce a map showing only knock
 - (4) Results are compared with the threshold values for judgment of knock, and binarization processing is applied
 - (5) Results are compared with the knock pattern, and knock is judged to occur where a correlation of 50% or more exists
 - (6) When knock is judged to occur, ignition timing is retarded in proportion to the conformance rate
 - (7) When knock is not judged to occur, the logic learns the conditions as a case of mechanical noise, and this judgment is employed in the next cycle
- These steps are discussed in greater detail below.

5.1. STFT Processing

STFT processing is applied to the non-resonance knock sensor output in the 12-78 deg ATDC interval in which knock occurs. Based on sampling theorem, from the

frequencies analyzed sampling is conducted at 50 kHz and above. In order to analyze frequency for each 1 kHz, the time constant is set at 1 msec, and frequency components are calculated for every 6 deg. The results of this STFT processing are stored on an STFT map of the type shown in Fig. 10, composed of 66 squares, with frequency from 5-23 kHz divided into six groups and crank angle from 12-78 deg divided into 11 groups, one for each 6 deg. The map on which the knock spectrum intensity distribution is arranged into patterns is termed the master map; this is shown as the shaded section in Fig. 10. The figures on the master map show coefficients for raising the threshold value for knock judgment, which will be discussed below.

5.2. Isolation of Vibration Modes

In order to identify the vibration mode frequencies, which display a decline with the passage of time, the frequencies in the STFT processing results calculated for each 1 kHz are divided into groups as shown in the STFT map in Fig. 10, and the maximum spectrum intensity for each group is selected. By means of this processing,

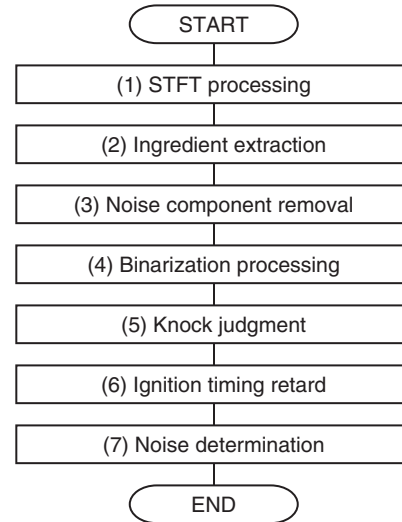


Fig. 9 Knock detection logic processing flow

		Crank angle (deg ATDC)										
		12	18	24	30	36	42	48	54	60	66	72
Frequency band (kHz)	20 - 23	1	1									
	17 - 19	1	1	1								
	14 - 16	1	1	1	1							
	11 - 13	1	1	1	1	1						
	8 - 10	1.25	1.5	2	2	1.5	1.25					
	5 - 7	1.25	1.5	2	2	1.5	1.25	1	1	1	1	1

Master map
Numbers in the master map show coefficient of boosting knock judgment threshold value

Fig. 10 STFT map, master map, coefficient of boosting knock judgment threshold value

it is possible to grasp the phenomenon of the decline in frequency of each of the vibration modes generated by knock. The calculated results are termed the maximum intensities map.

5.3. Removal of Noise Components

When knock is not judged to occur, a noise learning map based on the system's previous learning of continuously generated mechanical noise is employed. The difference between this map and the aforementioned maximum intensities map is calculated for each grid point. By means of this processing, a map that expresses only knock is produced. This map is termed an intensity distribution map.

5.4. Binarization Processing

The values on the intensity distribution map are compared with the threshold value for knocking judgment. When the value is greater than the threshold value it is converted to "1," and when it is lower it is converted to "0." This processing converts the intensity distribution map into a map that shows the intensity generation pattern. This is termed the pattern map.

Figure 11 shows the threshold value for knock judgment. This value is set to help enable detection of trace knock. In addition, taking the frequency band on the master map in which the intensity becomes stronger when knock is produced and the knock attenuation characteristic into consideration, a coefficient for raising the threshold value for knock judgment (as shown in Fig. 10) is applied in order to partly raise the threshold value. This method increases the robustness of the system, helping it to maintain stable functioning even when mechanical noise is being generated. Non-continuous mechanical noise, such as piston slap, is also generated when water temperature is low, and the threshold value for knock judgment is also raised in order to increase robustness in relation to mechanical noise at low water temperatures.

5.5. Judgment of Knocking

The intensity generation pattern on the pattern map calculated using binarization processing and a previously determined knock intensity generation pattern are compared, and the level of correlation between the two is calculated. The calculated value is termed the conformance

rate. Knock is judged to occur when the conformance rate is 50% and above.

The timing of knock generation changes depending on the ignition timing and the speed of combustion. In Fig. 10, the master map commenced from 12 deg, but the start position of the master map is adjusted depending on ignition timing and whether or not exhaust gas recirculation (EGR) is performed. Figure 12 shows the relationship between ignition timing, EGR, and the start position of the master map.

5.6. Retardation of Ignition Timing

Following judgment of knocking, ignition timing is retarded the necessary minimum amount in relation to the conformance rate, which indicates the intensity of knock. Figure 13 shows the relation between the conformance rate and the degree of retardation of ignition timing. In the previous system, ignition timing was retarded by a fixed degree when knock was judged to occur, but the newly developed system determines the optimum degree of retardation of ignition timing in response to the conformance rate. This helps to prevent excessive or insufficient retardation of ignition timing, and makes it possible to reduce knock without causing fuel efficiency to decline.

5.7. Noise Learning

Valve seating noise is a representative example of mechanical noise. Valve seating noise occurs frequently, and is stable because the energy involved is determined by

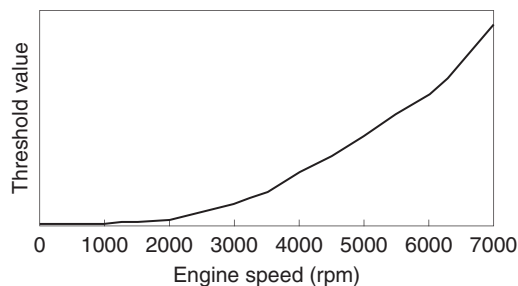


Fig. 11 Knock judgment threshold value

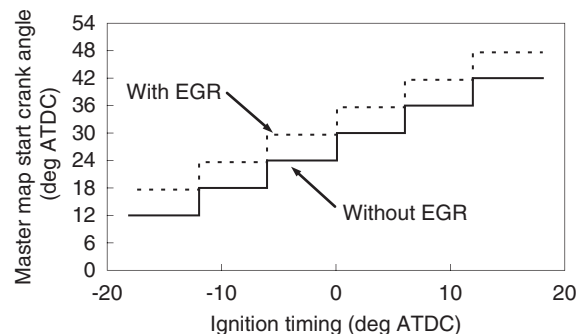


Fig. 12 Start timing of master map

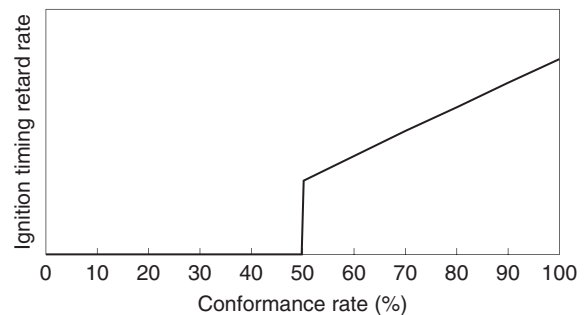


Fig. 13 Knock retard for conformance rate

the mass of the valve body and the spring coefficient of the valve spring. When knock is not judged to occur, the logic learns the conditions as a case of mechanical noise, and this is employed as a noise learning map in the next cycle. The noise learning map corrects for changes in engine speed, and is calculated using weighted averages with past values.

6. Verification of STFT Logic

6.1. Test Engine Specifications

A Honda 2.4-liter DOHC-VTEC-L4 engine equipped with a Variable valve Timing and lift Electronic Control (VTEC) mechanism and a VTC system for the intake valve camshaft was employed as the test engine. Table 3 shows the engine specifications.

Table 3 Engine specifications

Cylinder configuration	In-line 4-cylinder
Bore x Stroke (mm)	87 x 99
Displacement (cm ³)	2354
Compression ratio	10.5
Valve train	DOHC (inlet: valve timing control)
Number of valves	4 valves / cylinder
Fuel injection system	Port injection

Table 4 Operating conditions

Engine speed (rpm)	Intake manifold pressure (kPa (G))
1000	-20
1000	0 (WOT)
3000	-20
3000	0 (WOT)
6000	-20
6000	0 (WOT)

6.2. Check of STFT Logic when Knock is Detected

Table 4 shows the engine speed and load conditions used in the tests. The operation of the STFT logic was checked by analyzing the STFT logic results for a cycle in which knock occurred under each test condition.

Figure 14 shows STFT processing results when knock occurred. The X axis of the graph shows the crank angle, and the Y axis shows frequency. The spectrum intensity is expressed by the shade of the colors, with lighter colors indicating stronger spectrum intensity.

The white areas in Fig. 14 show the spectrum intensity produced by knock. The same knock vibration modes, and the frequency drop in those modes, as shown in Fig. 5 can be observed in Fig. 14. In addition, this figure shows that attenuation as the expansion stroke progresses is greater in the high-frequency vibration modes. Based on these results, the STFT logic was judged as being able to accurately identify the characteristics of knock. Figure 15 shows the intensity distribution map calculated by the STFT logic from the STFT processing results in Fig. 14. This figure indicates that the calculated intensity distribution map is able to isolate the knock vibration modes in which frequency declines over time from the STFT calculation results shown in Fig. 14. Figure 16 shows pattern map calculation results when knock occurs. Here, the intensity distribution map when knock occurs is subjected to binarization processing and converted into a pattern displaying the knock features shown in Fig. 8. Calculation of the conformance rate for Fig. 16 produced a result of 93%; the conformance rate exceeds 50%, and knocking is judged to occur. Based on these results, the STFT logic was judged as correctly reproducing the knock pattern.

6.3. Verification of Knock Detection Accuracy

An in-cylinder pressure sensor was fitted in an engine, and the detection accuracy of the STFT logic was verified by comparison with the pressure sensor output. In order to judge knock in the pressure sensor output, the output was processed using a band pass filter (5 kHz to 10 kHz) and calculated values were compared with a threshold value for

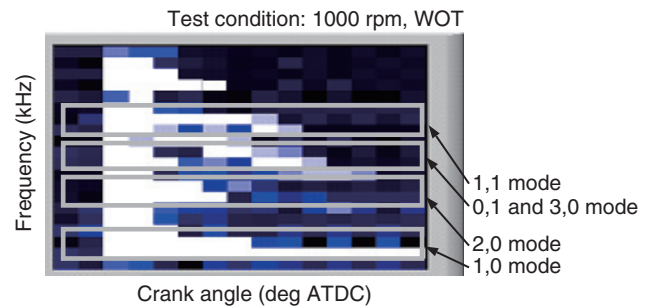


Fig. 14 STFT calculation results

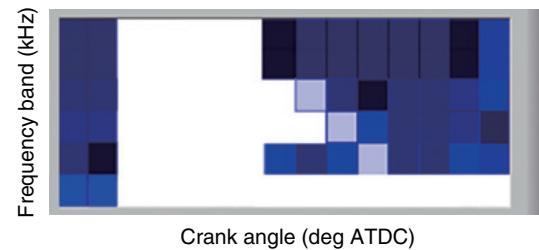


Fig. 15 Knock energy calculation results

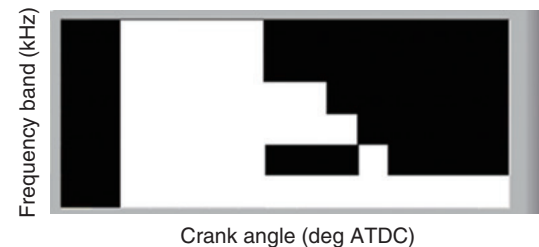


Fig. 16 Pattern map calculation results when knock occurs

Table 5 Comparison of knock detection logic detection count for varying conditions

Engine speed (rpm)	Intake manifold pressure (kPa (G))	Number of times of knock detection in 50 cycles	
		STFT logic	In-cylinder pressure logic
1000	-20	6	6
1000	WOT	8	8
3000	-20	8	8
3000	WOT	7	7
6000	-20	11	11
6000	WOT	8	8

judgment of knock. Figure 17 shows the configuration of the measurement system.

Figure 18 shows time charts of the knock output obtained by the in-cylinder pressure sensors under conditions of almost continuous knock and the conformance rate calculated by the STFT logic. These results confirm that both knock detection logics detected knock in the same cycle. Table 5 shows the knock detection count for each logic under different engine speed and load conditions. Even under different engine speed and load conditions, both knock detection logics displayed equivalent detection

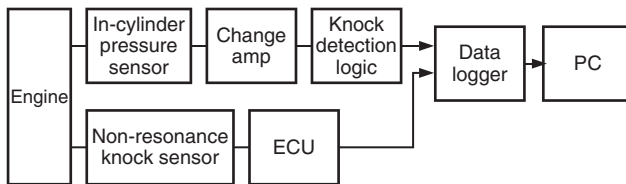


Fig. 17 Measuring system configuration

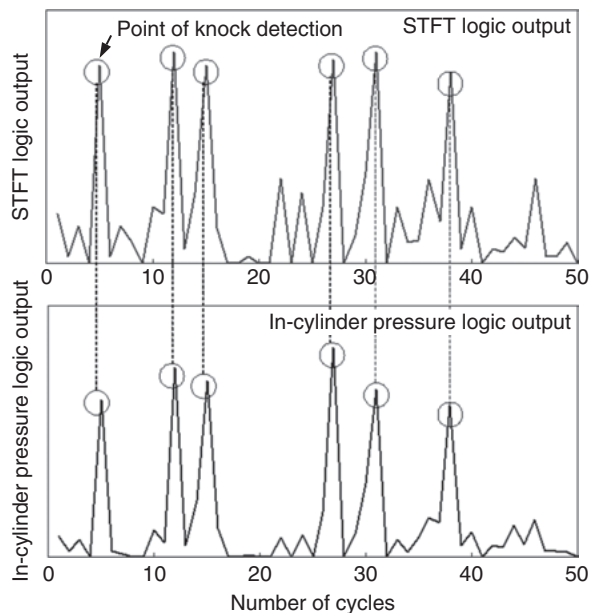


Fig. 18 Comparison of in-cylinder pressure sensor knock output and conformance rate

performance. The STFT logic was therefore judged as being able to accurately detect knock.

6.4. Verification of Robustness in relation to Mechanical Noise

Valve seating noise was generated within the knock detection gate, and knock detection was performed to verify the robustness (stable system functioning) of the STFT logic in relation to mechanical noise.

The upper graph in Fig. 19 shows knock output for the band pass filter logic and the threshold value for judgment of knock. The lower graph in Fig. 19 shows the conformance rate for the STFT logic and the conformance rate of 50% at which knock is judged to occur.

In the case of the band pass filter logic, when valve seating noise occurs within the knock detection gate, knock output increases compared to the output when valve seating noise is not generated, and exceeds the threshold value for knock detection at some points. This indicates that the band pass filter logic misjudges mechanical noise as knock. By contrast, the conformance rate calculated by the STFT logic does not increase significantly when valve seating noise is generated within the knock detection gate, indicating that mechanical noise is not misjudged as knock.

Figure 20 shows the pattern map calculated by the STFT logic when valve seating noise occurs. This figure

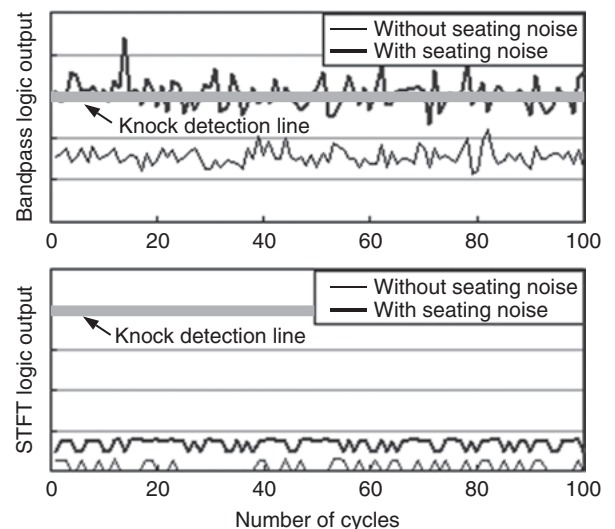


Fig. 19 Knock judgment output results for each logic when seating noise occurs

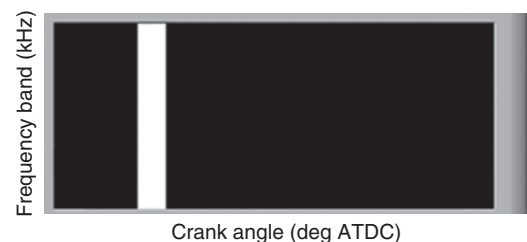


Fig. 20 Pattern map when valve-seating noise occurs

demonstrates that the valve seating noise pattern calculated by the STFT logic displays a wide frequency band. This valve seating noise pattern clearly differs from the knock pattern shown in Fig. 16. Therefore, the conformance rate calculated by the STFT logic does not increase, and does not lead to misjudgment of knocking. Based on these results, pattern processing by the STFT logic is judged as providing the system with greater robustness in relation to mechanical noise than conventional knock detection systems that judge knock based on the intensity of the frequency at which it occurs.

The results discussed above demonstrated that the newly developed system is able to distinguish knock and mechanical noise with a high degree of accuracy even when the timing and the frequency of the phenomena match. In addition, increasing the robustness of the logic in relation to mechanical noise has made it possible to set the threshold value for judgment of knock lower than is the case in conventional systems, and this contributes to the reduction of knock noise.

7. Conclusion

A knock detection logic able to distinguish knock and mechanical noise with a high degree of accuracy was developed. The following results were obtained from this development.

- (1) Using STFT, the concept of time was introduced in order to help enable the attenuation of frequencies to be identified, and a knock detection logic that judges knock on the basis of pattern recognition was developed. This logic is able to distinguish between knock and mechanical noise with a high degree of accuracy even when the timing and frequency of the two phenomena match, and has made it possible to increase robustness in relation to mechanical noise against conventional knock detection systems that judge knock based on the intensity of the frequency at which it occurs.
- (2) Increasing the robustness of the logic in relation to mechanical noise has made it possible to set the threshold value for judgment of knock lower than is the case in conventional systems, and this contributes to the reduction of knock noise.

References

- (1) Zadnik, M., Vincent, F., Vingerhoeds, R., Galtier, F.: SI Engine Knock Detection Method Robust to Resonance Frequency Changes, SAE Paper, 2007-24-0054 (2007)
- (2) Lonari, Y., Polonowski, C., Naber, J., Chen, B.: Stochastic Knock Detection Model for Spark Ignited Engines, SAE Paper, 2011-01-1421 (2011)
- (3) Ham, Y. Y., Chun, K. M., Lee, J. H., Chang, K. S.: Spark-ignition engine knock control and threshold value determination, SAE Paper, 960496 (1996)

- (4) Borg, J. M., Cheok, K. C., Oho, S. T., Saikalas, G.: Knock Signal Analysis Using the Discrete Wavelet Transform, SAE Paper, 2006-01-0226 (2006)
- (5) Zhang, Z., Tomita, E.: Diagnostic of Knocking by Wavelet Transform Method Utilizing Real Signal as Mother Wavelet, SAE Paper, 2001-01-3546 (2001)
- (6) Zhang, Z., Tomita, E.: A New Diagnostic Method of Knocking in a Spark-ignition Engine using the Wavelet Transform, SAE Paper, 2000-01-1801 (2000)
- (7) Xiaofeng, G., Stone, R., Hudson, C., Bradbury, I.: The detection and quantification of knock in spark ignition engines, SAE Paper, 932759 (1993)
- (8) Kearney, M. J.: Knock Signal Conditioning Using the Discrete Fourier Transform and Variable Detection Window Length, SAE Paper, 2007-01-1509 (2007)
- (9) Wu, G.: A Real Time Statistical Method for Engine Knock Detection, SAE Paper, 2007-01-1507 (2007)
- (10) Samimy, B., Rizzoni, G.: Engine knock analysis and detection using time-frequency analysis, SAE Paper, 960618 (1996)
- (11) Draper, C. S.: The physical effects of detonation in a closed cylindrical chamber, N.A.C.A., p. 493 (1934)

■ Author ■



Kenji AKIMOTO



Hirotaka KOMATSU



Atsushi KURAUCHI

Power Circuit Technology using Next-generation SiC Power Semiconductors for Power Electronics Devices in Electric Powertrain

Tomoyo EGOSHI*

Sadao SHINOHARA*

ABSTRACT

Next-generation silicon carbide (SiC) power semiconductors are expected to offer size reductions and increased efficiency. Optimized design is necessary for the power circuits employing these semiconductors in order to exploit their performance potential. The identification of each circuit element related to transient response during high-speed switching, a characteristic of SiC power semiconductors, and the correlation of circuit simulation models with the results of actual measurements, made it possible to obtain voltage and current waveforms to within a 4% level of error.

Using this circuit simulation technology, the most important circuit elements to power circuit design were identified. The employment of optimized values in sections of the circuit in which parasitic impedance affected efficiency and electromagnetic interference (EMI) performance was shown to reduce switching loss and noise.

1. Introduction

With the transition to the use of higher levels of power, it becomes necessary to make the power electronics devices such as inverters and converters used in electric vehicles and similar vehicles smaller, more efficient, and less prone to electromagnetic interference (EMI). Power semiconductors are one of the key technologies employed in power electronics devices. Most semiconductors today are based on silicon (Si), but the performance limits of the material have been reached, and research and development efforts are proceeding in the area of SiC semiconductors, a type of next-generation power semiconductor expected to offer ultra-low levels of loss. However, because SiC power semiconductors are capable of high-speed switching, they present issues with regard to their use in automotive applications, including reduced efficiency and increased EMI, making optimal power circuit design essential.

Parasitic impedance in power semiconductor chips, electrical wiring, and other components is an important parameter in realizing faithful reproduction of circuits by the circuit simulation technologies used at the design stage. The development of a circuit simulation technology that reflected this parameter would make it possible to (1) simulate a power circuit able to reproduce the high-speed switching operation of an SiC power semiconductor, and (2) analyze and evaluate the voltage and current when EMI originating in high-speed switching by an SiC power

semiconductor occurs.

This paper discusses the development of a circuit simulation that reproduces the high-speed switching operation of an SiC power semiconductor, the identification of areas of circuits with a significant effect on EMI due to voltage and current waveforms, and achieved noise reduction results. The achievement of good reproduction of voltage and current waveforms was an important factor in realizing this development, in respect of their relationship to loss in devices and the efficiency of the system as a whole.

2. SiC Power Semiconductors: Characteristics and Issues of Application

Figure 1 shows the theoretical limit line for the on-resistance of Si and SiC. As this graph indicates, a dramatic increase in on-resistance performance can be expected from the use of SiC against Si.

To provide an indication of projected performance increases, Table 1 shows a comparison of the characteristics of new semiconductor materials with those of the conventionally used Si⁽¹⁾.

On-resistance, $R_{on}A$, an important parameter for power semiconductors, can be expressed by Eq. (1)⁽²⁾.

$$R_{on}A = \frac{4V_B^2}{\mu\epsilon E_c^3} \quad (1)$$

* Automobile R&D Center

Table 1 Comparative characteristics of new semiconductor material and Si

Material Physicality	New semiconductor			Conventional semiconductor
	SiC (4H)	Diamond	GaN	Si
Breakdown strength (10^6 V/cm)	2.5	10.0	3.3	0.3
Energy bandgap (eV)	3.26	5.47	3.39	1.12
Electronic saturation velocity (10^7 cm/s)	2.2	2.7	2.7	1.0
Thermal conductivity (W/cm·K)	4.9	20.9	2.0	1.5
Stage of wafer manufacturing technology	Mass production in part	Research	Development	Mass production

V_B : Breakdown voltage

μ : Bulk migration

ε : Permittivity

E_c : Breakdown strength

From Eq. (1), $R_{on}A$ is inversely proportional to the cube of E_c . Because the E_c of new semiconductor materials is at least eight times that of Si, as shown in Table 1, it is possible to expect that the use of these materials will help enable chips to be reduced in thickness, thus reducing on-resistance.

The use of insulated gate bipolar transistors (IGBT), which are unipolar and bipolar compound devices and make it possible to realize higher withstand voltage than in the case of unipolar metal oxide semiconductor field-effect transistors (MOSFET), is common in the field of high withstand voltage power semiconductors. Employing high withstand voltage SiC power semiconductors will make it possible to use unipolar device configurations, helping to enable high-speed switching.

Diamond and GaN display a similar level of potential to SiC as next-generation semiconductor materials. However, in terms of the stage of advancement of wafer manufacturing technology, wafers using diamond materials are still at the research stage and wafers using GaN materials are under development, while wafers using SiC materials are at the stage of partial mass production. At present, therefore, the highest expectations are on devices using SiC.

However, some issues arise in the application of SiC power semiconductors. Because SiC is capable of realizing high-speed switching, increases in voltage and

current surges and ringing and common mode current due to parasitic impedance make it a challenge to adequately exploit the high performance of which SiC is capable. The design of power circuits that are optimized for the complete vehicle is important in responding to these issues, and this makes a circuit simulation technology able to be applied to the development and analysis of optimal power circuits essential.

3. Circuit Simulation Technology

In order to formulate circuit simulation models, the research discussed here selected SiC power MOSFET and Schottky barrier diodes (SBD), which can be applied in hybrid, electric, and similar vehicles, for the power semiconductor chip configuration. Figure 2 shows the graphical symbols for each. Here, V_{ds} is the voltage between the drain and the source, I_d is the drain current, V_{gs} is the voltage between the gate and the source, V_f is the voltage between the anode and the cathode, and I_f is the anode current.

Figure 3 shows an example of circuits as employed in the powerplant system of an electric vehicle. Here, C_b is parasitic capacitance between the battery and the body, C_i is parasitic capacitance between the inverter and the body, C_l is parasitic capacitance between the three-phase line and the body, and C_m is parasitic capacitance between the motor and the body. For the simulation model system, a single-ended push-pull circuit of the type shown in Fig. 4, which forms the basis of devices including inverters and converters, was employed. C_i' is the parasitic capacitance determined by the SiC MOSFET/SBD power module configuration. With regard to common mode current, while parasitic capacitance exists between the body and all the components including

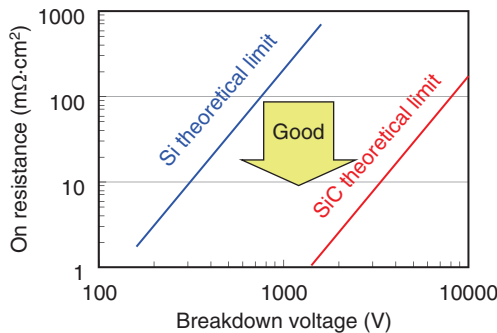


Fig. 1 Si and SiC theoretical limit line

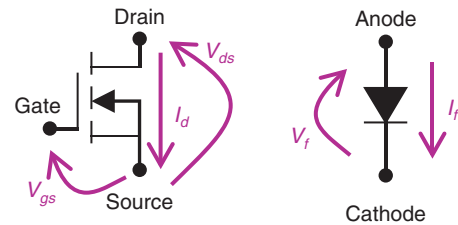


Fig. 2 MOSFET and SBD symbol

the motor, here, for the sake of simplicity, the parasitic capacitance between the body and the motor, the battery, and the three-phase line is summed, and the parasitic capacitance divided between PN is taken as C_0 , creating an equivalent circuit that makes it possible to conduct a comparative analysis of the effect of common mode current.

The flow of procedures for the circuit simulation technology is as follows:

- (1) Listing of circuit parameters
- (2) Fitting of models of power semiconductor chips
- (3) Calculation of parasitic impedance
- (4) Formulation of circuit model
- (5) Verification using results of tests

3.1. Listing of Circuit Parameters

Table 2 shows the parameters that are essential to the formulation of the circuit simulation. Figure 5 shows the capacitance parameters for the power semiconductor chips.

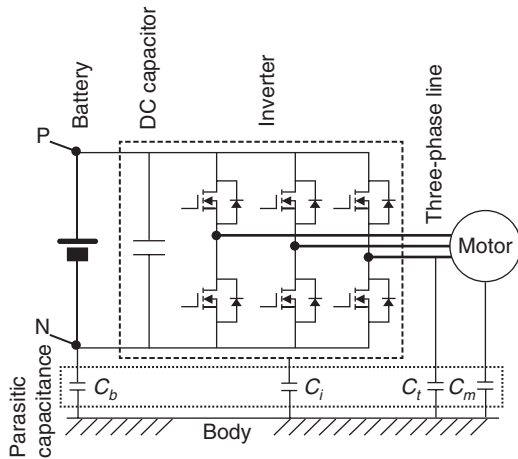


Fig. 3 System model

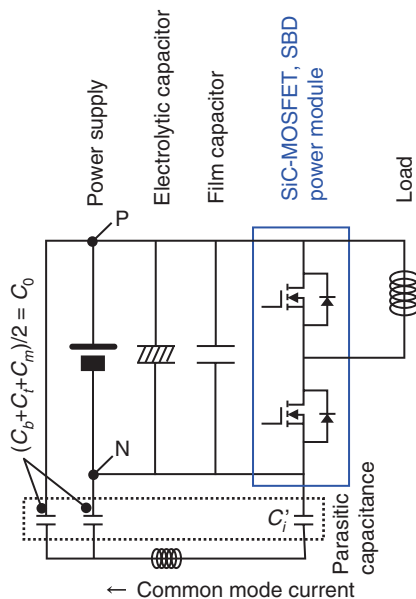


Fig. 4 Simulation model system

Table 2 Parameter list

Division	Subdivision	Parameter
Chip	SiCFET	I_d - V_{ds} Characteristic
		C_{dg} - V Characteristic
		C_{ds} - V Characteristic
		C_{gs} - V Characteristic
	SiCSBD	I_f - V_f Characteristic
Other parts	Bus bar	Parasitic inductance/resistance
	Electrode plate	
	Wire	
	Insulating substrate	Parasitic capacitance

C_{dg} is the capacitance between the drain and the gate, C_{ds} is the capacitance between the drain and the source, C_{gs} is the capacitance between the gate and the source, and C_{ac} is the capacitance between the anode and the cathode. The parameters listed are electrical characteristics (static and dynamic characteristics) for the chips and all parasitic impedance originating in the structural parts such as the bus bar for parasitic impedance.

3.2. Fitting of Models of Power Semiconductor Chips

The models of power semiconductor chips are fitted by setting the parameters so that the characteristics expressed by approximation equations closely match the characteristics determined by actual measurements.

3.2.1. SiC FET

(1) I_d - V_{ds} characteristic

The approximation equations for I_d are

$$I_d = I_{sat} (1 + \lambda \cdot V_{ds}) \left(2 - \frac{V_{ds}}{V_{sat}} \right) \frac{V_{ds}}{V_{sat}} \quad (2)$$

for the linear domain, and

$$I_d = I_{sat} (1 + \lambda \cdot V_{ds}) \quad (3)$$

for the saturation domain.

Here, the saturation voltage, V_{sat} , and the saturation current, I_{sat} , are expressed as

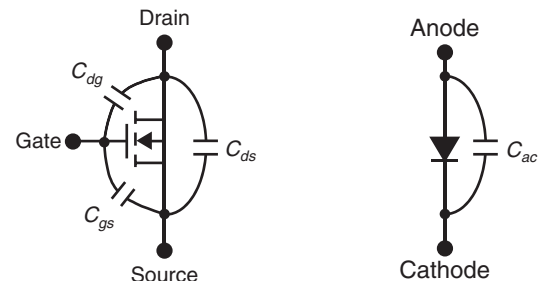


Fig. 5 Chip C parameter

$$V_{sat} = A(V_{gs} - V_p)^M \quad (4)$$

$$I_{sat} = \frac{K}{2}(V_{gs} - V_p)^N \quad (5)$$

λ : Channel length modulation parameter

V_p : Pinch-off voltage

K : Transfer conductance coefficient

A : Saturation voltage coefficient

M : Saturation voltage exponent

N : Saturation current exponent

Figure 6 shows the fitted I_d - V_{ds} characteristic with each parameter set.

(2) C - V_{ds} characteristic

The approximation equation for C is as follows:

$$C = C_c \left[\delta + \frac{1-\delta}{(1+V_{ds}/V_{Diff})^\alpha} \right] \quad (6)$$

C_c : Capacitance constant

α : Junction grading coefficient for C

δ : Junction correction factor for C

V_{Diff} : Diffusion voltage

Figure 7 shows the fitted C - V_{ds} characteristic with each parameter set.

3.2.2. SiC SBD

(1) I_f - V_f characteristic

The approximation equation for I_f is as follows:

$$I_f = I_{sat_fwd} \left[\exp \left(\frac{V_f}{M_{fwd} \cdot V_T} \right) - 1 \right] \quad (7)$$

M_{fwd} : Carrier correction value

I_{sat_fwd} : Operating point saturation current

R_{b_fwd} : Junction bulk resistance

V_T : Thermal voltage (25.5 mV in a 300 K environment)

Figure 8 shows the fitted I_f - V_f characteristic with each

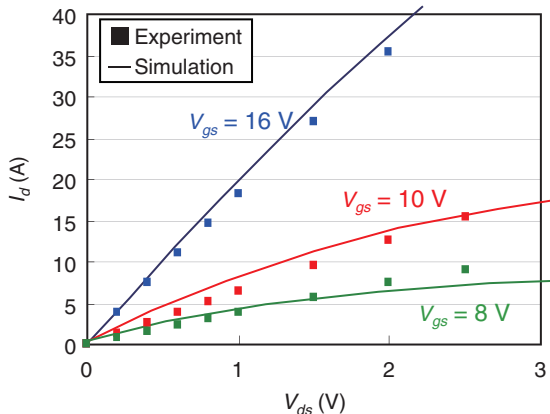


Fig. 6 I_d - V_{ds} fitting diagram

parameter set.

(2) C - V_r characteristic

The approximation equation for C is as follows:

$$C = C_{j0} \left[\delta + \frac{1-\delta}{(1-V_r/V_{Diff})^\alpha} \right] \quad (8)$$

C_{j0} : Zero bias junction capacitance

$V_r = -V_f$

Figure 9 shows the C - V_r characteristic with each parameter set.

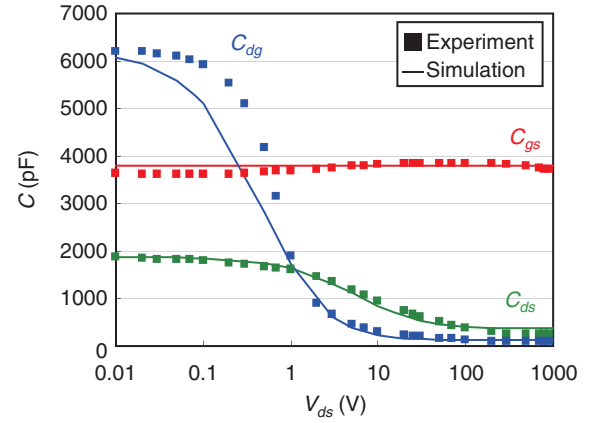


Fig. 7 C - V_{ds} fitting diagram

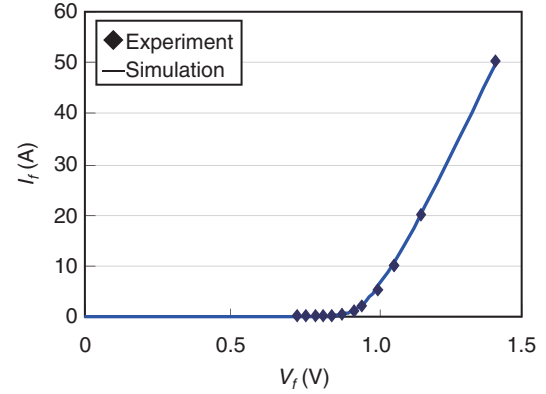


Fig. 8 I_f - V_f fitting diagram

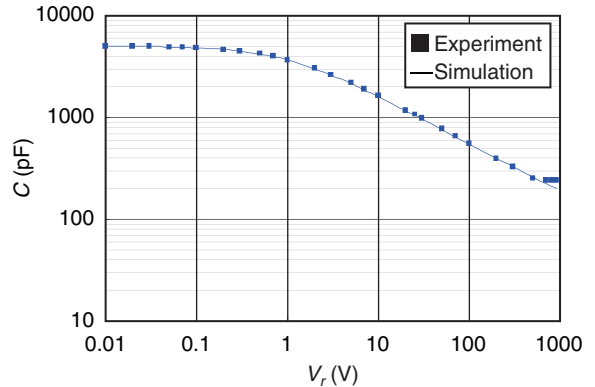


Fig. 9 C - V_r fitting diagram

3.3. Calculation of Parasitic Impedance

Figure 10 shows a diagram of the SiC MOSFET/SBD power module used in the simulation circuit model. The values of parasitic impedance for the constituent components of the module were determined using three-dimensional LCR parameter identification software, based on the configuration shown in Fig. 10. Here, parasitic inductance, a parameter with a strong relationship with surge voltage, was calculated with consideration of the mutual inductance generated by the overlapping of the current paths.

3.4. Formulation of Circuit Model

Equivalent circuit models were formulated by incorporating the power semiconductor chip models fitted in Section 3.2 and the parasitic impedances calculated in Section 3.3. In order to help ensure greater accuracy in reproduction, distributed constant models, as represented by C_1 and C_2 , were employed for the direct brazed copper (DBC) substrate model. Figure 11 shows a diagram of the circuit simulation.

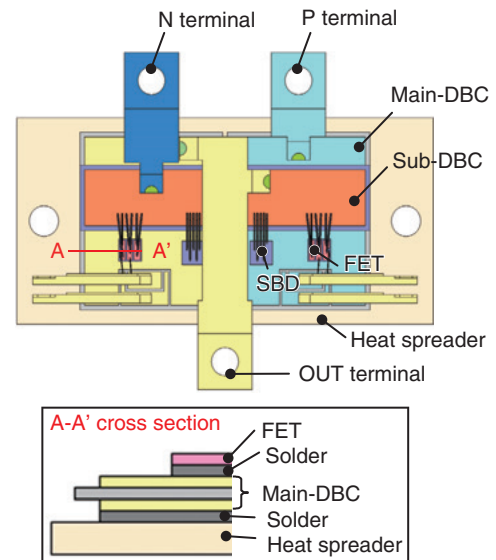


Fig. 10 SiC-MOSFET, SBD power module diagram

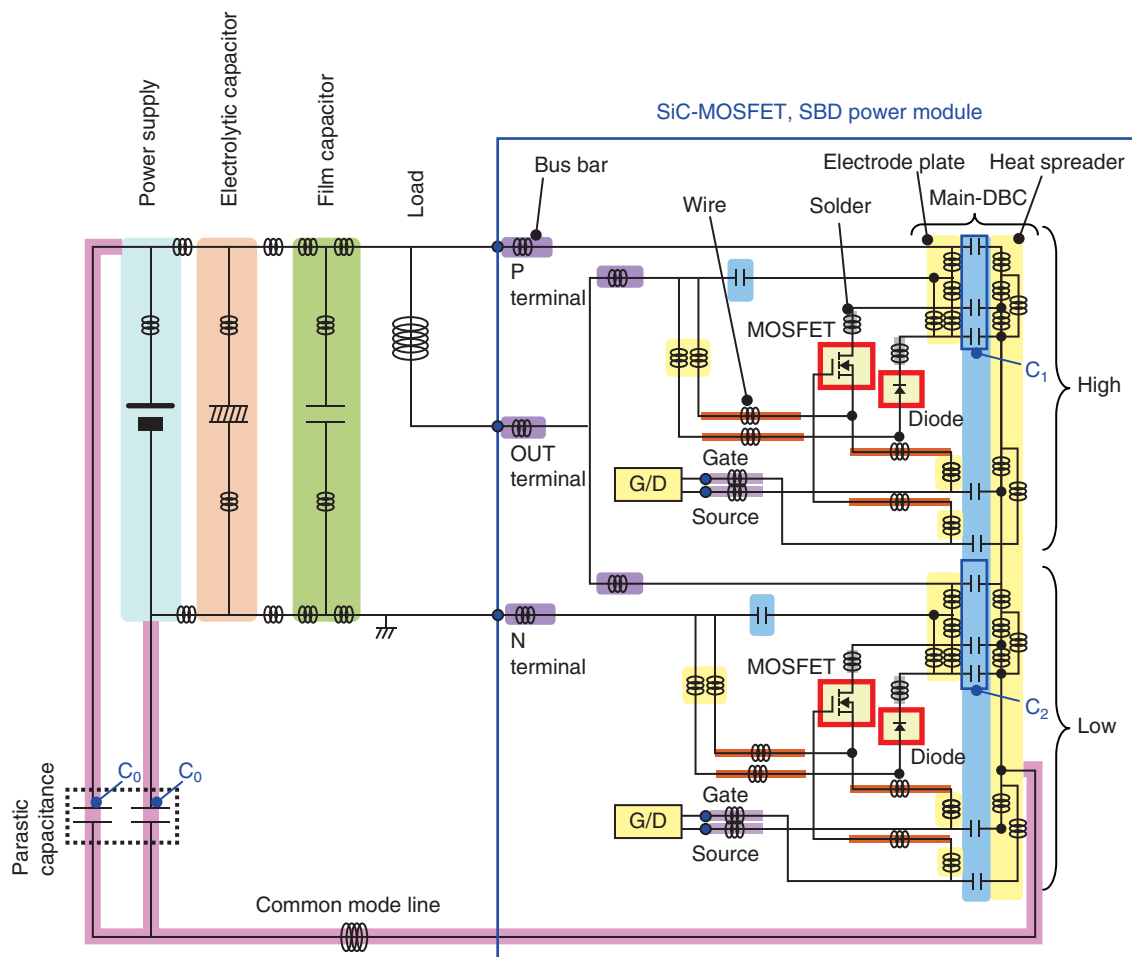


Fig. 11 Simulation diagram (resistance abbreviated)

Table 3 Comparison between experiment and simulation

				Gap
I_d (at turn on)	Peak (A)	Experiment	68	- 3 %
		Simulation	67	
	Frequency (MHz)	Experiment	11.2	+ 18 %
		Simulation	13.2	
V_{ds} (at turn off)	Peak (V)	Experiment	283	- 4 %
		Simulation	280	
	Frequency (MHz)	Experiment	26.3	- 39 %
		Simulation	18.9	

3.5. Verification using Results of Tests

Figures 12 and 13 respectively show the turn-on and turn-off waveforms for the SiC MOSFET. Table 3 shows the peak value for I_d and the ringing frequency at turn-on, and the peak value for V_{ds} and the ringing frequency at turn-off. The test conditions employed were a source voltage of 200 V and drain currents of 30 A (at turn-on) and 40 A (at turn-off).

Results were obtained for peak values of voltage and current within an error of 4% at both turn-on and turn-off. Results for ringing frequency were within 40% error, necessitating the realization of a greater degree of

correspondence in the model through correlation with parasitic impedance. However, because a match on the MHz order was obtained in the EMI analysis, qualitative judgments were considered possible, and the analysis proceeded using the developed circuit simulation.

4. Identification of Important Parts in Power Circuit and Results

Using the circuit simulation technology discussed in Chapter 3, a study was conducted to identify the circuit parameters in a power circuit network that are important in relation to EMI generated by resonance frequencies. These parameters were shown to be C_1 , C_2 , and C_0 in Fig. 11.

Figure 14 shows results for common mode noise obtained by optimizing these parameters. Common mode current increases by 12.8% in the case of an SiC MOSFET (without optimized design of constants) against a conventional Si IGBT, but optimized design of values reduces the figure for the SiC MOSFET by 81.8%. Figure 15 shows the common mode current waveforms for the circuits. Figures 16 and 17 compare normal mode noise and switching loss for the circuits. Because the SiC power semiconductor offers high-speed switching, normal mode noise increases by 125% against the Si IGBT element, but optimized design of constants results in a 41.5% reduction of noise. With regard to switching loss, results indicate that further reductions can be expected with optimized design of constants.

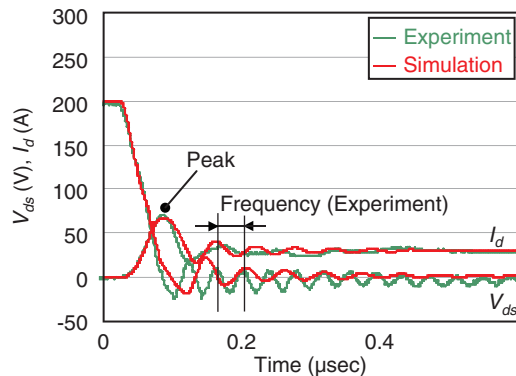


Fig. 12 Turn-on waveform

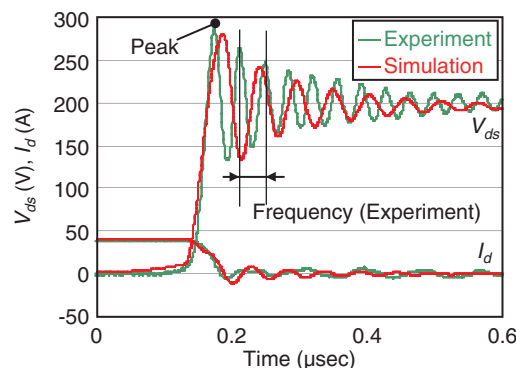


Fig. 13 Turn-off waveform

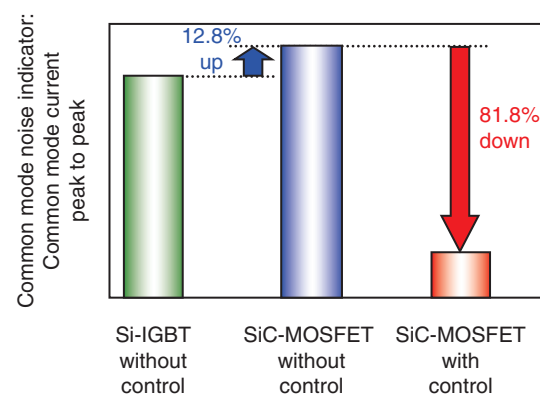


Fig. 14 Common mode current comparison

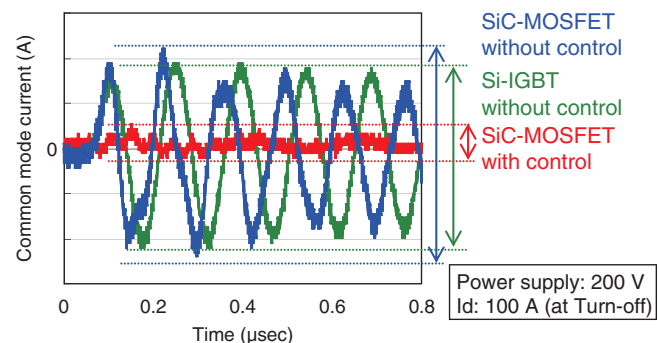


Fig. 15 Common mode current waveform

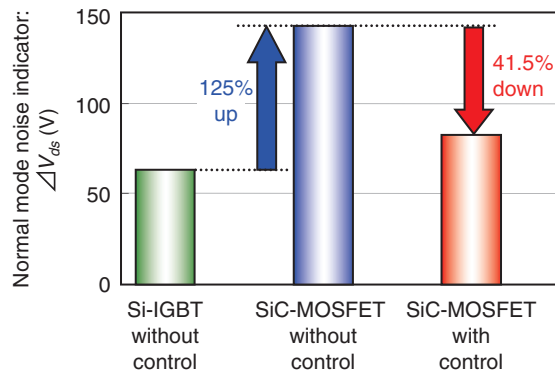
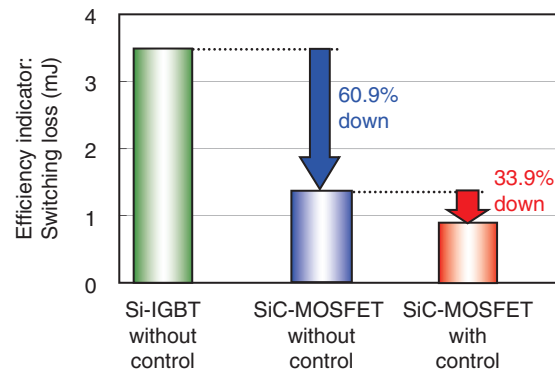
Fig. 16 ΔV_{ds} comparison

Fig. 17 Switching loss comparison

competitive power electronics devices. In future, in relation to EMI, it will be necessary to achieve further detailed correspondence of the parasitic impedance around the main interconnections related to normal mode noise.

References

- (1) Shinohe, T.: SiC Power Devices, Toshiba Review, Vol. 59, No. 2, p. 49-53 (2004)
- (2) Power electronics handbook editorial board: Power electronics handbook, Ohmsha, 288 p. (2010)

5. Conclusion

The development of a simulation technology for high-speed switching power circuits employing SiC power semiconductors produced the following results:

- (1) Three-dimensional LCR parameter identification software was used to determine power semiconductor chip parameters and parasitic impedance between the wiring and interconnections of structural parts; correlation with actual measurements and the further correction of parameters involved in mutual inductance helped to enable the setting of highly accurate power circuit parameters.
- (2) The results of correlation with voltage and current waveforms and frequencies and the evaluation of performance as power circuits indicated that figures for surge voltage and current error were within 4%. Using the circuit simulation technology described above, the following results were obtained for efficiency and EMI performance.
- (3) Circuit parameters closely related to efficiency and EMI performance were identified, and reductions of 81.8% in common mode noise and 33.9% in switching loss were achieved through optimized design of constants.

The results produced by the technology considered in this research indicate that the application of SiC power semiconductors will help enable the design of highly

■ Author ■



Tomoyo EGOSHI



Sadao SHINOHARA

Dynamic Simulation of Soil Cultivation using Particle Method

Yosuke YAMAMOTO*

Takayuki SATO*

Genki ANRAKU*

ABSTRACT

It is desirable for the rotors used in tillers to be shaped appropriately for the specific characteristics of soils in different regions. A technology for the prediction of the dynamic behavior of soils that makes it possible to study soils with varying characteristics on a computer is therefore demanded for the efficient design of tiller rotors.

However, soil cultivation is a dynamic phenomenon involving the deformation, breakage, and agitation of the soil, and it is challenging for methods that use grids, such as the finite element methods and finite volume methods generally used in conventional CAE technologies, to treat phenomena of this type. The research discussed in this paper therefore attempted to develop a simulator for three-dimensional fluid-structure analysis using the particle method, a method suited to the analysis of large-scale deformation and breaking phenomena. In conducting this development, analytical accuracy was increased by reproducing the boundaries between soil and rigid bodies using polygons. In addition, to make it possible to faithfully reproduce the break-up behavior of soil, a model of soil break-up conditions that considered the effect of soil consolidation was formulated, and measurements of soil properties were conducted in order to provide input conditions for the simulation.

Comparison of the results of simulations of a single tiller tine and an entire rotor with test results demonstrated that results matched for the break-up behavior of the soil and the shape of the plough pan. The simulator developed in the research discussed here has made it possible to reproduce soil cultivation on the desktop, and it is believed that it will be effective in helping to enable the efficient development of tillers.

1. Introduction

The rotors used in tillers rotate in order to till soil. The fact that soil characteristics differ from region to region necessitates the development of rotors suited to specific soils. The efficient development of tiller rotors demands technologies for the prediction of the dynamic behavior of soil under a variety of soil conditions.

CAE technologies for the analysis of strength and stiffness are conventionally employed in tiller development. However, the study by computer of the condition of the soil following tilling represents a challenge. This is because it is necessary to reproduce the deformation and breakage of the soil by tilling, but the amount of deformation is high in conventional methods employing grids, and the computations fail when the distortions in the grid become too great.

Particle methods have recently attracted attention as methods able to resolve the issues described above. By contrast with conventional methods using grids, particle methods can discretize a continuous body using particles able to move freely through space. Because the particles

do not carry data concerning connections with contiguous points, unlike in the case of grids, calculations will not fail even when particles move a significant distance from their initial positions. This means that free surface problems and phenomena associated with large-scale deformation or breaking can be handled with comparative ease, leading to the application of the methods in dynamic analyses of fluids and structures⁽¹⁾. Calculation methods classified as particle methods include the discrete element method (DEM), the smoothed particle hydrodynamics (SPH) method, and the moving particle semi-implicit (MPS) method⁽²⁾.

In order to reproduce the tilling of soil on the desktop, the research discussed in this paper attempted to develop an original three-dimensional fluid-structure simulator for the Power Products R&D Center using an MPS method^{(2),(3)}. To increase the accuracy of the simulator, modifications were made to the treatment of rigid boundaries. In addition, because the effects of consolidation result in localized changes in the strength of the soil, a model of soil-breaking conditions able to faithfully reproduce the mode of break-up of the soil was formulated.

* Power Products R&D Center

2. Overview of Simulator

2.1. Overview of Moving Particle Semi-implicit Method (MPS Method)

The research discussed here employed the MPS calculation method, a type of particle method. The method was developed in 1995 to provide semi-implicit algorithms for use in incompressible fluid analyses, and is known for its ability to be applied in structural analyses and fluid-structure coupled analyses⁽²⁾⁻⁽⁴⁾. The MPS method discretizes equations formulated using Lagrange methods. In specific terms, using this method continuous bodies are expressed as particles, and partial differential equations are discretized using interparticle interaction models. The weighting function shown as Eq. (1) is used in the interparticle interaction model.

$$w_{ij}(r) = \begin{cases} \frac{r_e - r}{r_e} & r \leq r_e \\ 0 & r > r_e \end{cases} \quad (1)$$

r is the distance between particle i and particle j , and r_e is termed the effective radius, expressing the range of effect of interparticle interaction. Figure 1 shows a conceptual diagram of the effective radius. The interparticle interaction model is applicable only to particles existing within the effective radius for any specific particle. The larger the effective radius, the higher the number of neighboring particles; calculation accuracy increases as a result, but calculation time also increases. Because of this, it is necessary to set appropriate values based on a balance between calculation accuracy and calculation time.

The MPS method uses the interparticle interaction model to discretize gradient, Laplacian, divergence and other operators employed in the equations. Using the weighting function, gradient, Laplacian, and divergence models are defined for a variable ϕ or a vector $\vec{\phi}$ as follows:

$$\langle \nabla \phi \rangle_i = \frac{d}{n^0} \sum_{j \neq i} \frac{\phi_j - \phi_i}{|\vec{r}_j - \vec{r}_i|} \frac{(\vec{r}_j - \vec{r}_i)}{|\vec{r}_j - \vec{r}_i|} w_{ij} \quad (2)$$

$$\langle \nabla^2 \phi \rangle_i = \frac{2d}{n^0 \Lambda} \sum_{j \neq i} (\phi_j - \phi_i) w_{ij} \quad (3)$$

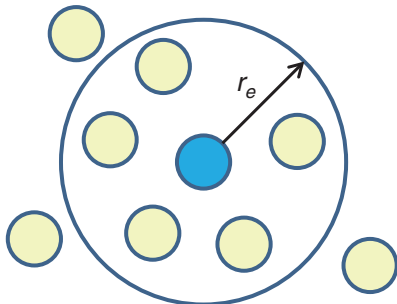


Fig. 1 Effective radius for MPS method

$$\langle \nabla \cdot \vec{\phi} \rangle = \frac{d}{n^0} \sum_{j \neq i} \frac{(\vec{\phi}_j - \vec{\phi}_i) \cdot (\vec{r}_j - \vec{r}_i)}{|\vec{r}_j - \vec{r}_i|^2} w_{ij} \quad (4)$$

Here, the position vector of the particles is termed $\vec{r}$, and the number of spatial dimensions is termed d . Λ is defined as follows:

$$\Lambda = \frac{\sum_{j \neq i} w_{ij} |\vec{r}_j - \vec{r}_i|}{\sum_{j \neq i} w_{ij}} \quad (5)$$

The particle number density at the position of particle i is defined as follows:

$$\langle n \rangle_i = \sum_{j \neq i} w_{ij} \quad (6)$$

This is the sum of the weighting functions for all the interparticle interactions in the vicinity of particle i . n^0 in Eqs. (2), (3), and (4) represents the initial state of $\langle n \rangle_i$, and the same value is employed for this parameter throughout the calculations.

When treating fluids using the MPS method, Navier-Stokes equations for incompressible flows shown as Eqs. (7) and (8) are used as the governing equations.

$$\frac{D\rho}{Dt} = 0 \quad (7)$$

$$\frac{D\vec{v}}{Dt} = -\frac{\nabla P}{\rho} + \nu \nabla^2 \vec{v} + \vec{f} \quad (8)$$

Here, t is time, $\vec{v}$ is velocity, P is pressure, ρ is density, ν is the kinematic viscosity coefficient, and $\vec{f}$ is external force. Equations (7) and (8) are discretized using the gradient model (Eq. (2)) and the Laplacian model (Eq. (3)) for the interparticle interaction model. Explicit calculations are conducted for the MPS method algorithms with the exception of the pressure term in Eq. (8), and a provisional velocity for the next step is determined. Following this, implicit calculations are conducted for pressure in order to equate the particle number density for each particle in Eq. (5) to n^0 , and the velocity is corrected.

When treating structures using the MPS method, the governing equation for elastic bodies shown as Eq. (9) is employed.

$$\rho \frac{\partial v_\alpha}{\partial t} = \frac{\partial}{\partial x_\beta} (\lambda \varepsilon_{\gamma\gamma} \delta_{\alpha\beta} + 2\mu \varepsilon_{\alpha\beta}) \quad (9)$$

Here, ρ is density, v is velocity, x is position, ε is a strain tensor, and δ is the Kronecker delta. λ and μ are Lamé's elasticity constants. These constants can be expressed as follows using Young's modulus, E , and the Poisson ratio, ν :

$$\lambda = \frac{E\nu}{(1+\nu)(1-2\nu)} \quad (10)$$

$$\mu = \frac{E}{2(1+\nu)} \quad (11)$$

The strain tensor in Eq. (9) is defined using the relative interparticle displacement shown in Fig. 2. The initial relative position vector between particle i and particle j in Step 1, $\vec{r}_{ij}^0$, becomes $\vec{r}_{ij}$ when particle j moves to its position in Step 2. The difference between the two position vectors is considered as the relative interparticle displacement vector, $\vec{u}_{ij}$. In relation to vector $\vec{r}_{ij}$, this vector is further divided into the vertical component $\vec{u}_{ij}^n$ and the non-vertical component $\vec{u}_{ij}^s$. The product obtained by dividing this relative interparticle displacement by the initial interparticle distance is defined as the strain tensor in the MPS method. Because divergence of the strain tensor is necessary for the calculation of the right side of Eq. (9), the divergence model, Eq. (4), for the interparticle interaction model is used for discretization and the velocity and position of the particles are explicitly calculated for the next step.

The MPS method also considers interparticle rotational motion. The force acting on a particle in the shear direction, $\vec{F}_{ij}$, and the moment produced by this force, $\vec{M}_{ij}$, are defined as follows:

$$\vec{F}_{ij} = \frac{2dm}{\rho n^0} \frac{\vec{\tau}_{ij}}{|\vec{r}_{ij}^0|} w_{ij} \quad (12)$$

$$\vec{M}_{ij} = \left| \vec{r}_{ij} \times \vec{F}_{ij} \right| \quad (13)$$

Here, m is the mass of a single particle, and $\vec{\tau}_{ij}$ is the shear stress acting between particles i and j . $\vec{\tau}_{ij}$ is calculated from the second term within the parentheses on the right-hand side of Eq. (9). To conserve angular momentum, a half-torque is applied to particles i and j in order to cancel the resultant moment.

$$I \frac{\partial \omega_i}{\partial t} = -\frac{1}{2} \sum_{j \neq i} \vec{M}_{ij} \quad (14)$$

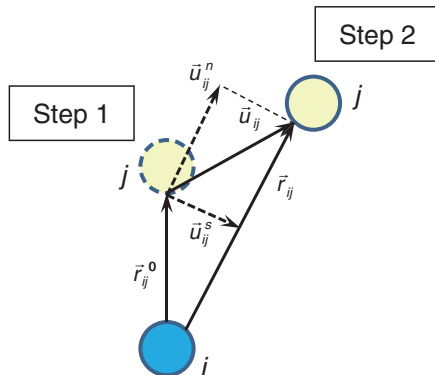


Fig. 2 Relative interparticle displacement

Here, I is the moment of inertia for a single particle, and ω is angular velocity. By imparting degrees of freedom to the rotation of each particle, the MPS method makes it possible to remove the rigid rotational components from the relative interparticle displacement vectors and calculate stress even in the case of large-scale deformation.

2.2. Modifications to Increase Accuracy

2.2.1. Treatment of boundaries with rigid bodies

Boundaries with rigid bodies are generally expressed using particles when applying the particle method. In the present study, “rigid body” refers to the tines of a tiller. When the boundary has a complex configuration, it is necessary to increase particle resolution in order to accurately express the configuration, which can increase calculation time. Altering the particle resolution locally or not employing particles at boundaries have been proposed as methods of resolving this issue⁽⁵⁾⁻⁽⁷⁾. The use of a method in which polygon models rather than particles are employed at boundaries reduces the number of particles necessary for the calculation, thus reducing the calculation load. However, it is necessary to take care with regard to the interaction between the polygons and the particles in order to accurately reproduce curvature.

In the present research, polygon models were used to express boundaries, and the polygons affecting any specific particle were determined using the velocity vector of that particle. Figure 3 shows the relationship between a particle and a polygon model. The boundary is composed of an infinitesimal triangular element (a polygon); the polygon acts as a boundary that imparts a repelling force when a particle approaches to within a given distance. Because a polygon that exists in the direction of the velocity vector of the particle and at the shortest distance from the particle is selected as the boundary, two or more polygons will not exert a repelling force on the particle even in areas of high curvature.

2.2.2. Model of breaking conditions for reproduction of soil behavior

On the microscopic level, soil is composed of the elements soil particles, water, and air (Fig. 4), and its properties change depending on the ratio and geometrical configuration of these elements^{(8),(9)}. When soil is deformed

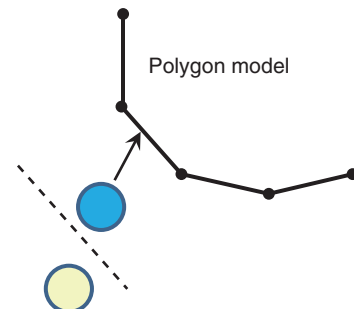


Fig. 3 Particle and polygon models

by the action of tensile and compressive forces, its porosity changes, and this results in changes in its hardness and strength. Because the numerical analysis assumes a homogeneous continuous body obeying the governing equation shown as Eq. (9), the faithful reproduction of properties with consideration of the elements composing the soil represents a challenge. The present research, therefore, attempted to modify the breaking conditions employed in order to reproduce the break-up of soil.

Normally, the breakage of structures in the MPS method is determined by the interparticle distance, and the equivalent stress is defined as follows, using the interparticle stress tensor⁽¹⁰⁾:

$$\bar{\sigma} = \sqrt{\frac{3}{2}(\sigma^n^2 + \sigma^s^2)} \quad (15)$$

σ^n is interparticle deviatoric stress in the vertical direction, and σ^s is deviatoric stress other than in the vertical direction. This is identical to Von Mises equivalent stress and is not affected by mean vertical stress. However, soil properties are thought to change with consolidation due to soil structure. The equivalent stress was, therefore, corrected as follows:

$$\bar{\sigma}' = \bar{\sigma} - \alpha\sigma_m \quad (16)$$

σ_m is mean vertical stress and α is the coefficient for the adjustment of the degree of influence of the mean vertical stress. Breakage is considered to occur when the corrected equivalent stress exceeds breaking strength. Positive values for σ_m express the action of tensile forces, while negative values express the action of compressive forces. Because the corrected equivalent stress declines when tensile forces are acting, strength declines. By contrast, the corrected equivalent stress increases when compressive forces are acting, and strength thus increases. The consolidation of the soil can be reproduced through the changes in soil strength due to tensile and compressive forces.

2.3. The Simulator

The present research employed Fortran90 as the development language for the simulator. Approximately

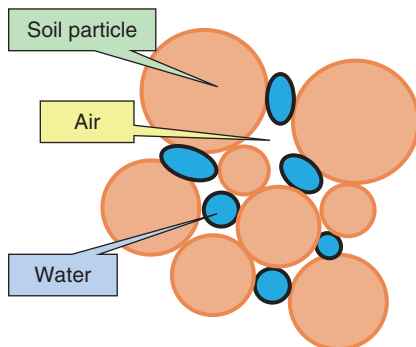


Fig. 4 Microscopic structure of soil

7000 lines were used in the program. OpenMP was used for parallelization in order to increase calculation speed.

As indicated in Section 2.1., it is possible to conduct dynamic fluid analyses, structural analyses, or fluid-structure coupled analyses. At present Newtonian fluids and elastic bodies can be treated, and customization is possible by means of the program.

With regard to the input of shapes, three-dimensional point sequence data is formulated for the fluids and structures to be treated as particles, and this data is input. For the polygon models, STL models formulated from 3D models can be employed without modification as input data. It is not necessary to formulate high-quality mesh data from 3D models produced using a grid or other method. It is possible to set parallel translation and rotational motion at fixed speeds for the polygon models, and more complex motion can also be realized by means of the program.

3. Analysis of Soil Cultivation

3.1. The Tiller

Figure 5 shows an external view of a self-propelled tiller. A rotor to enable tilling of the soil is attached to the bottom section of the tiller. Because the condition of the soil following cultivation is an important element in the raising of crops, it is essential to evaluate the performance of these rotors.

3.2. Measurement of Soil Properties

In order to conduct a simulation of soil cultivation, it is necessary to conduct measurements of the properties of the soil under study. The research discussed here used unconfined compression tests in order to conduct measurements of the modulus of deformation and breaking strength of the soil to be simulated, and these measurements were reflected in the simulation. Tests for particle size



Fig. 5 External appearance of tiller

distribution were also conducted in order to help enable the quantitative classification of the soil for simulation.

The tests were conducted following the methods specified in JIS A 1216⁽¹¹⁾. The relationship between strain and compressive stress shown in Fig. 6 was obtained when loads were applied using an Autograph to soil compressed into a cylinder. The maximum value of compressive stress is equivalent to breaking strength, and the modulus of deformation can be found from the gradient at the point at which 50% of maximum compressive stress occurs. A clay soil was used in the tests. This soil possessed a high level of hardness, and was resistant to breakage. Test specimens were formulated that reproduced the water content and density of the soil in an actual field, and unconfined compression tests were conducted. Variations were observed between the test specimens, but the soil under study was found to have a modulus of deformation of 0.7-1.5 MPa and a breaking strength of 0.06-0.10 MPa.

In addition, because the composition of the size of the particles making up the soil has a significant effect on its properties, tests for particle size distribution based on JGS 0131 were conducted⁽¹²⁾ in order to determine the size of the particles. For these tests, a dispersant is applied to soil samples in order to disperse the soil particles into water and create suspension liquids, which are then left untouched. As time elapses, the dispersed particles settle, and the density of the suspension liquids changes. Because the speed of settling is faster the larger the particles and slower the smaller the particles, the composition of the size of the particles can be determined from the ratio of the change in density to the elapsed time. Figure 7 shows a particle size distribution curve obtained by measuring density at specific points in time using a hydrometer and applying Stokes' theorem. Tests for particle size distribution were conducted on the clay soil forming the subject of the present study and a black soil possessing different properties. The smaller the size of the particles making up the black soil, the closer its properties were to those of clay soil. From Fig. 7, it is possible to quantitatively verify the fact that the clay soil under study is composed of finer particles than the black soil that was tested.

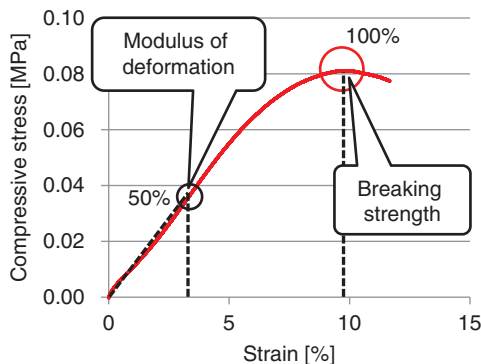


Fig. 6 Stress-strain diagram for soil

3.3. Unit Analysis of Tiller Tines

First, a simulation of soil break-up was conducted on a parallelepiped soil model composed of soil particles using a single tiller tine rotating at 10 rpm. Approximately 50000 particles with an interparticle distance of 5 mm, treated as elastic bodies, were subject to analysis. Comparisons were conducted between two cases with differing breaking conditions, as described in Section 2.2.2. For Case 1, α in Eq. (16) was set at 0 without consideration of the effect of consolidation, and for Case 2, α in Eq. (16) was set at 1 with consideration of the effect of consolidation. Other than the soil-breaking conditions, all conditions were identical for the two cases. Values for the physical properties of the soil were determined on the basis of the values for the clay soil measured in Section 3.2. In order to compare the simulation results with the break-up behavior of actual soil, a parallelepiped clay soil test specimen was subjected to tests involving the rotation of a single tine (Fig. 8), and the break-up behavior of the soil in the tests and in the analysis results was compared.

Figure 9 shows soil behavior during a test using actual soil. Break-up of the soil occurred in areas distant from the tine, and scraping that raised the soil across a wide area was observed. The results of the simulations of soil cultivation are shown in Fig. 10. In Case 1, because the

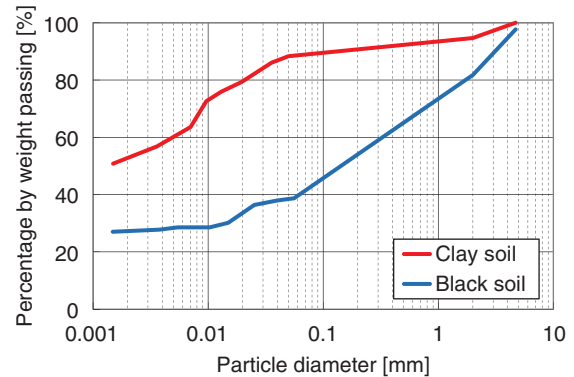


Fig. 7 Particle size distribution curve

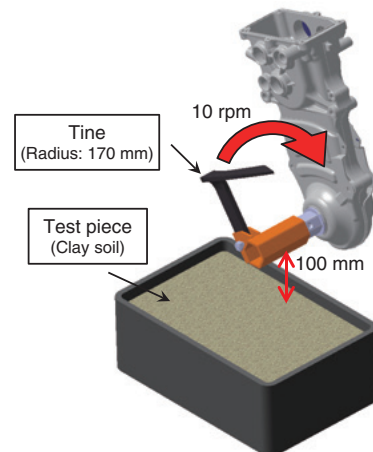


Fig. 8 Experimental apparatus for single-tine unit

break-up of the soil occurred close to the tine, the shape of the scraped-up clods of soil was similar to the width of the tine. By contrast, in Case 2, because the break-up of the soil occurred farther away from the tine, the scraped-up clods were larger than those in Case 1. These results indicate that the behavior displayed by the soil in Case 2, which considered the effect of consolidation, was most similar to the break-up behavior of actual soil. Quantification of the break-up behavior of the soil and evaluations in which loads are applied to the tines, to make it possible to conduct quantitative comparisons, will be issues for the future.

3.4. Analysis of Entire Tiller Rotor

Next, an analysis of an entire rotor was conducted, and the results were compared with the shape of the plough pan produced by tilling with an actual tiller. As Fig. 11 shows, multiple tines were attached to two rotors by means

of holders, and the rotors were attached to each side of the transmission case as a left-right pair. Figure 12 shows an example of the analysis of the entire rotor. Analyses were conducted using rotors of two different shapes, and the shapes of the resulting plough pans were compared. As Fig. 13 shows, in the case of Type A some soil remains in the center of the plough pan, while the plough pan for Type B is flat. For both Type A and B, the tine tips are the furthest points from the center of rotation, but in the case of Type A the tips of the tines are distant from the transmission case at the center of the tiller, while in the case of Type B there are tine tips that are positioned close to the transmission case. It is believed that it is because of this that soil remains in the center of the plough pan in the case of Type A, while the soil in the center of the plough pan is broken up in the case of Type B. Figure 14 shows the shapes of plough



Fig. 9 Soil behavior using actual machine

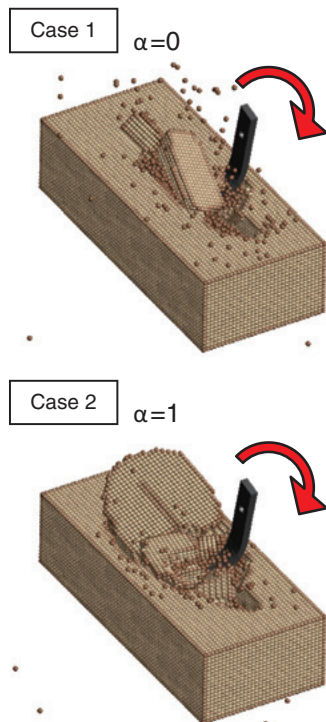


Fig. 10 Analytical results for single-tine unit

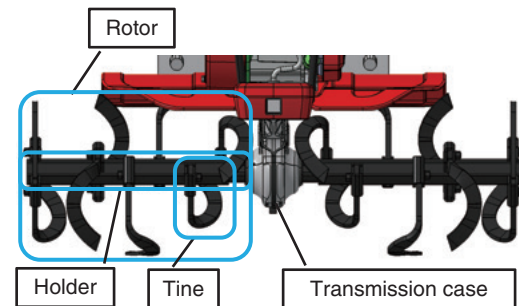


Fig. 11 External appearance of rotor



Fig. 12 Analysis of entire rotor

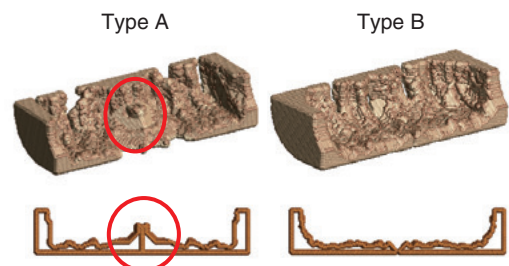


Fig. 13 Analytical results for soil cultivation

pans following cultivation by an actual tiller. As in the simulations, soil remains in the center of the plough pan in the case of Type A, while no soil remains in the case of Type B. For the entire rotor also, a qualitative match was achieved between test and simulation results.

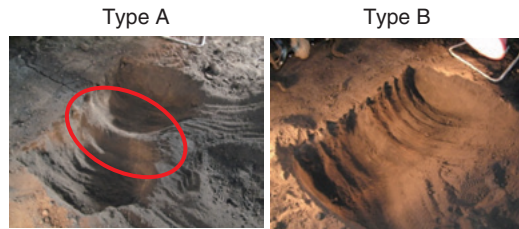


Fig. 14 Results for soil cultivation by actual machine

4. Conclusion

The development of a technology for the simulation of soil cultivation by a tiller using the particle method generated the following conclusions.

- (1) The reproduction of soil as particles using the MPS method and the expression of boundaries between soil and rigid bodies as polygons made it possible to calculate dynamic phenomena such as the break-up and agitation of soil by the tines of a tiller with comparative stability.
- (2) In order to faithfully reproduce the break-up of soil, a breakage model that considered consolidation was formulated and measurements of the properties of actual soils were conducted. The results indicated that a qualitative match was achieved between simulation and test results for the break-up behavior of soil under a single tine and the shape of the plough pan following cultivation.
- (3) The results of the research discussed in this paper have made it possible to reproduce the cultivation of soil on the desktop, and it is believed that the technology will be effective in helping to enable efficient development of tiller rotors.

References

- (1) Yamamoto, Y., Sato, T. Anraku G.: Dynamic Simulation of Water and Soil Using the Particle Method, SAE paper, 2011-32-0563 (2011)
- (2) Koshizuka, S.: Particle Methods (Computational dynamics lecture series), The Japan Society for Computational Engineering and Science, p. 1-139 (2005)
- (3) Koshizuka, S.: Numerical Analysis of Flow using Particle Method, Flow 21, p. 230-239 (2002)
- (4) Song, S. M., Koshizuka, S., Oka, Y.: Dynamic Analysis of Elastic Solids by MPS Method, Transactions of the Japan Society of Mechanical Engineers, Series A, Vol.

71, No. 701 (2005-1), p. 16-22 (2005)

- (5) Tanaka, M., Masunaga, T., Nakagawa, Y.: Multi-resolution MPS method, Transactions of JSCES, Paper No. 20090001 (2009)
- (6) Iida, K., Akimoto, H.: Water Slide Simulation using MPS Method, 17th Numerical Fluid Symposium D4-3 (2004)
- (7) Harada, T., Koshizuka, S.: Wall Boundary Calculation Model using MPS Method, Transactions of JSCES, Paper No. 20080006 (2008)
- (8) Murayama, S.: Dynamic Behavior of Soil, Gihodo Shuppan, p. 1-23, p. 119-221 (1990)
- (9) Yamaguchi, H.: Soil Mechanics (All revisions), Gihodo Shuppan, p. 1-32, p. 91-106, p. 141-194 (1969)
- (10) Tomita, Y.: Foundation and Application for Theory of Elasto-Plasticity, Morikita Publishing Co., Ltd., p. 25-64 (1995)
- (11) JIS 1216: Method for unconfined compression test of soils, p. 1-5 (2009)
- (12) JGS 0131: Test method for particle size distribution of soils, p. 23-33 (2008)

■ Author ■



Yosuke YAMAMOTO



Takayuki SATO



Genki ANRAKU

Technology to Enhance Deep-drawability by Strain Dispersion Using Stress Relaxation Phenomenon

Hiroyuki YAMASHITA*

Hiroaki UENO*

Hiroyuki NAKAI*

Takahiro HIGAKI*

ABSTRACT

When the introduction of strain is temporarily stopped during tensile testing of a metal, a stress relaxation phenomenon is known to occur whereby the stress diminishes with the passage of time. This phenomenon has been explained as the change of elastic strain into plastic strain.

The present report suggests the application of the stress relaxation phenomenon to stamping for the purpose of strain dispersion. A new step motion that pauses the die during forming was devised and succeeded in modifying the deep-draw forming limit.

Verification was conducted by tensile testing in addition to actual stamping tests. As a result, it was concluded that the mechanism that modifies the deep-draw forming limit arises from the use of step motion for the dispersion of strain, and this was also confirmed to be dependent on the stop time and the temperature.

1. Introduction

Since the 1990s, every automobile manufacturer has been actively advancing the application of high tensile-strength steel in order to achieve a balance of weight reduction and safety in their vehicles. Japan's steel manufacturers have been in the vanguard of high tensile-strength steel development. Their technology development has advanced to produce materials of greater strength and greater formability, and the application of those materials has reached as high as 60% or more on a body-in-white weight basis⁽¹⁾.

In general, increasing the strength of steel material involves a tradeoff in terms of reduced elongation. Consequently, high tensile-strength steel is essentially less formable than mild steel sheet, and stamped parts could not readily be obtained from materials with a tensile strength exceeding 780 MPa.

Materials technology addressed this with the development of materials such as low yield-point steel, which has low deformation resistance and superior elongation. This material is based on dual phase steel (DP steel) with a two-phase structure of ferrite for superior formability and martensite to assure strength. Another such material is transformation induced plasticity steel (TRIP steel) using stress-induced martensitic transformation,

which yields high strength after forming while preventing strain concentrations. More recently, it has been found that the formability of high tensile-strength steel can be increased effectively by enhancing hole expandability. Advances are being made in developing materials with local ductility enhanced for superior hole expandability by having the internal structure of the material combine structures that have smaller differences in hardness between them.

Meanwhile, measures have also been taken to enhance formability by means of stamping technology. Such approaches include increasing the lubricity between the material and the die and devising multi-stepped processing to avoid the concentration of processing strain. The appearance of servo press machinery in recent years now also allows press motions of unprecedented complexity. Tamai et al.⁽²⁾ used complex motions of the servo press in order to return the slide during forming and repour lubricating oil in between the blank material and the die. This succeeded in increasing lubricity during the latter part of forming, thus reducing friction with the die, reducing the amount of strain introduced to the material, and so increasing formability. However, repeating the action of returning the slide several times during forming tends to increase the total time taken in stamping, and there is concern that the diminished productivity could

* Automobile R&D Center

cause costs to rise.

Metal materials are known to experience a stress relaxation phenomenon whereby stress is reduced under a certain load at room temperature. Stress relaxation testing to evaluate the reduction in stress has also been standardized⁽³⁾. Initially, much of the research into this phenomenon was used in metal physics analysis, and most of it was related to pure metals, but there have also been recent reports on study of the phenomenon in structural steel⁽⁴⁾. Changes in stress when the crosshead was paused during tensile testing at room temperature were observed, and the results confirm that a significant reduction in stress occurs in an extremely short time. The mechanism of stress relaxation has been explained⁽⁵⁾ using Maxwell's model of viscoelasticity (Fig. 1). Due to viscous displacement of moving dislocation, a portion of the elastic strain that occurs during the initial deformation in the material is said to be substituted by the plastic strain that occurs with room temperature creep, and this is how stress relaxation is thought to occur. Viscous displacement of the dislocation is also accompanied by internal friction, and the moving speed is greater where there is less internal friction. This internal friction is of unrelaxed and relaxed types. The internal friction resulting from elastic strain changing into plastic strain, as can be explained by the Maxwell model, is the relaxed type. The magnitude of the internal friction is referred to as the Bordoni peak, and it is known to display temperature dependence⁽⁶⁾.

Given these circumstances, the present research sought to realize a previously unprecedented high level of deep-drawability using high-strength steel. To that end, a study was made of technology for extending the material forming limit by means of a servo press for complex press motions, in expectation that it would be feasible to control the process of plastic strain introduction through use of the stress relaxation phenomenon.

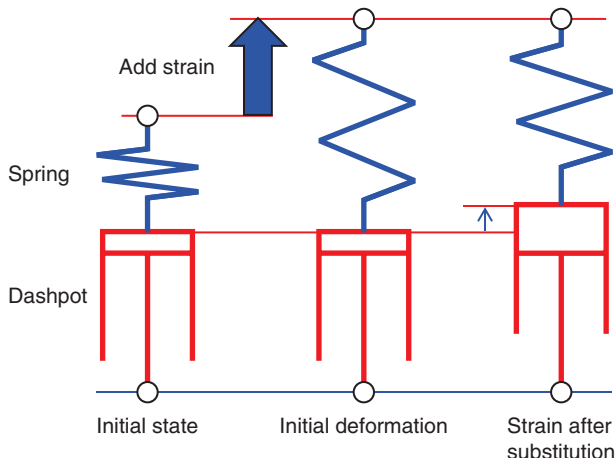


Fig. 1 Maxwell's model

2. Principles of the Method for High-level Deep-draw Processing by Strain Dispersion

The process by which stamping defects such as rupturing or necking occur in ordinary stamping is as follows. First, elastic deformation on a microscopic scale takes place during the initial processing because of heterogeneity in the material. When stress continues to concentrate in the deformed portions, these portions locally exceed the elasticity limit of the material so that plastic deformation occurs. Eventually, the constriction known as necking occurs, which is a deformation on a macroscopic scale. When processing is continued, stress concentrates progressively as a result of macroscopic deformation at this necking location and ultimately reaches the point of rupture.

The authors therefore devised a method to delay the occurrence of macroscopic deformation caused by microscopic elastic deformation that occurs in the material during initial processing, and so to make use of the deformability of the material to enhance its deep-drawability. The principle of this method is to introduce to the stamping process the substitution of plastic strain for elastic strain caused by the stress relaxation phenomenon. The microscopic areas of weakness in the material that would undergo the initial elastic deformation are strengthened by the introduction of plastic strain, and transferring the deformation to other areas causes dispersion of the strain throughout the entire material. In other words, the microscopic areas of elastic deformation that are initially areas of weakness in the material have plastic strain introduced to them by the stress relaxation phenomenon that has been brought about in advance. The dislocation density of these areas is thus increased and their yield strength is enhanced. That plastic deformation is transferred to other areas with lower yield strength in order to disperse the strain, and delaying the macroscopic local deformation thus enhances the stampability.

The principle of this processing method will be explained with stress-time curves from tensile testing of the material (Fig. 2). In the case of tensile testing conducted in the ordinary manner at a constant speed, elastic strain is first introduced (1). As tension continues to be applied, the elastic strain in areas of stress concentration partially turns into plastic strain (2). When this happens, the presence of heterogeneities in the material structure causes the concentrated introduction of plastic strain in some portions and of elastic strain in microscopic areas of weakness. Further strain is subsequently concentrated in the areas of weakness in the material, macroscopic constriction occurs as a result, and the concentration of stress advances further (3). The constriction ultimately advances to the point of rupture (4).

A forming method that works through strain dispersion, by contrast, follows a course like the following. From (1) to (2) is the same as in ordinary tensile testing. At (2), the movement of the die is temporarily stopped when plastic

strain is being introduced in some portions and elastic strain is being introduced into the microscopic areas of weakness. This causes a portion of the elastic strain to be substituted by plastic strain due to the stress relaxation phenomenon so that the microscopic areas of weakness are made stronger by work hardening (A) as a result. This phenomenon corresponds to the part of Fig. 1 from initial deformation to the strain after substitution. Next, when strain is reapplied (B), the dislocation density increases in areas where elastic strain has been substituted by plastic strain, thus increasing the strength more in these areas than in others. Consequently, no advance is made in introducing further plastic strain in these areas, and the introduction of elastic strain and in some parts plastic strain begins in other areas where the strength is relatively low (new areas of weakness) (C).

The authors therefore formulated the hypothesis that the repetition of this strain substitution would enable the dispersion of strain. Then, by suppressing local deformation, stress concentrations would be relaxed so that the occurrence of ruptures would be delayed.

It was reasoned that the application of this hypothesis to stamping and temporarily stopping processing could be anticipated to cause strain dispersion to occur, with the result that the forming limit would be extended.

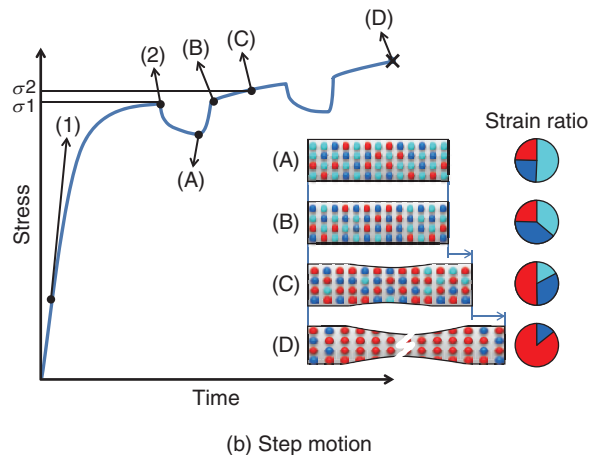
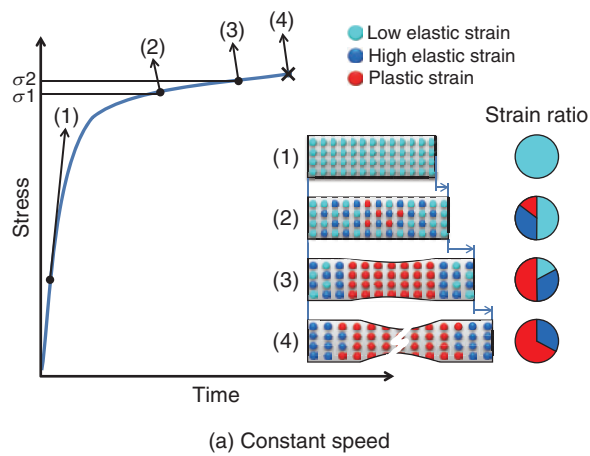


Fig. 2 Concept of forming

Table 1 Test conditions

Item	Data
Blank holding force	100 [kN]
Strokes per minute	1
Blank material	JAC440W 1.4t
	JAC590Y 1.4t

3. Test Method

In order to confirm the effectiveness of strain dispersion, a die with the deep-draw shape was actually used to carry out stamping tests.

3.1. Test Conditions

The test conditions are shown in Table 1.

In ordinary stamping (crank motion), as shown in Fig. 3, the slide position traces a sine wave over time. For the present research, the movement of the die was temporarily stopped during forming in order to promote substitution of elastic strain by plastic strain through the stress relaxation phenomenon. The relationship between time and the slide position is as shown in Fig. 4. Making advantageous use of servo press strengths, the press continued to apply a load even when the die movement was temporarily stopped, and it still performed position control. This stamping method is

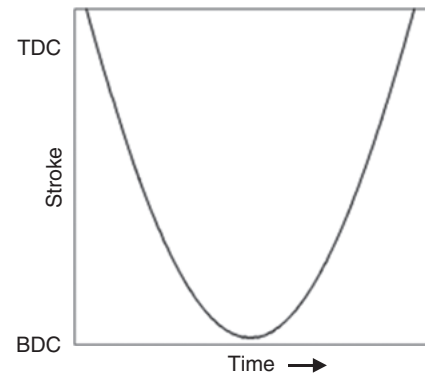


Fig. 3 Motion profile of crank motion

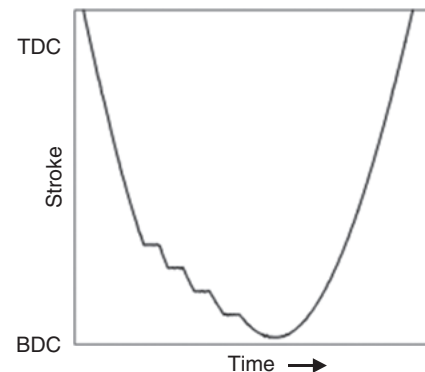


Fig. 4 Motion profile of step motion

referred to in the following as step motion.

The basic stamping stop time was set as 1.0 second. The strain substitution progresses with time, however, so differences in the effects due to stop time were also verified.

There is also the matter of the number of stop iterations. Although the effect can be expected even after one iteration, multiple stop iterations were made (at 5-mm intervals from the start of processing at 10 mm to bottom dead center) in order to make the effect clearer.

The forming depth evaluation method was to increase the forming depth in 5-mm increments. The limit depth where rupture or necking did not occur was identified as the forming limit depth.

Two test materials were used to check the effects of different strengthening mechanisms. JAC440W⁽⁷⁾ was used as a solid solution-strengthened steel, and the DP steel JAC590Y⁽⁷⁾ was used as a structure-strengthened steel.

3.2. Testing Device

Evaluation of deep-draw performance was conducted using the die shown in Fig. 5. The round columns at the bottom of the punch have strain gages attached. Three gages were used to measure the forming load.

Figure 6 shows the blank shape. The sheet is 1.4 mm thick.

The press machine used was an Aida Engineering, Ltd., NC1 2500(D) 2500 kN. This press machine has the advantage that its slide motion can be freely controlled by means of a numerical control (NC) servo mechanism.

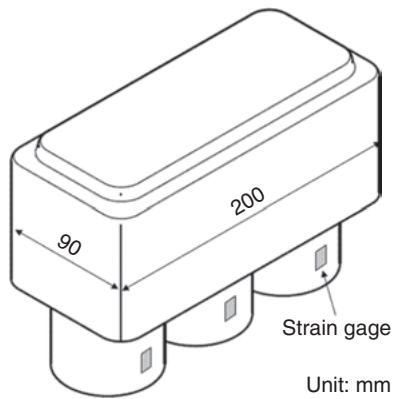


Fig. 5 Punch

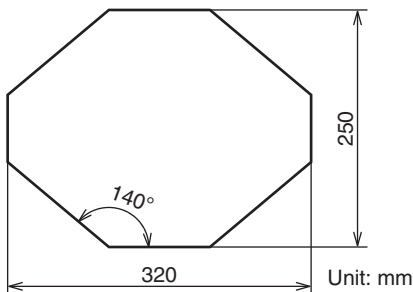


Fig. 6 Blank shape

4. Results and Discussion

4.1. Effects from Application of Step Motion

The result of forming with crank motion using JAC440W are shown in Fig. 7(a), and the result of forming with step motion in Fig. 7(b). With crank motion, forming could proceed to a depth of 60 mm, but a rupture was observed to occur in a corner area at 65 mm. Given the above, the forming limit with crank motion was judged to be 60 mm. When forming was carried out similarly with step motion, forming was able to proceed to a forming depth of 70 mm without any rupture occurring. Given the above results, the application of step motion was found to have the effect of extending the forming limit by approximately 10 mm.

Figure 8 shows forming load with respect to slide position (stroke length) when crank motion and step motion are used. It is apparent from the figure that both crank motion and step motion have generally the same forming load at every slide position. Figure 9, however, shows the changes in forming load with respect to time when using step motion, and it was found from examination of this graph that the forming load relaxes while the press is stopped temporarily during the course of forming. Then, when the stamping die starts moving again, the load recovers to generally the same value it had before stopping, after which the load describes generally the same curve as with crank motion processing.

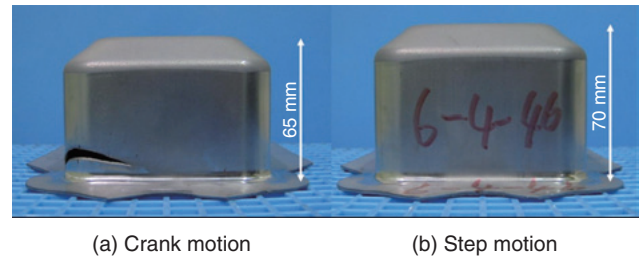


Fig. 7 Results of forming test

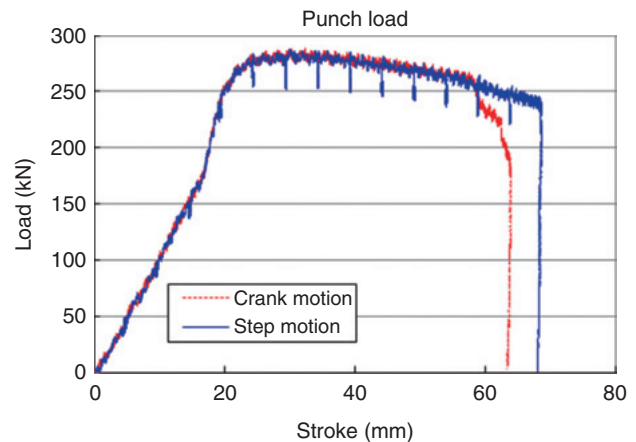


Fig. 8 Forming load in crank motion and step motion

The results of forming tests conducted using JAC590Y show a forming limit of 50 mm with crank motion. With step motion, however, forming was able to proceed to 70 mm. Testing confirmed that formability was enhanced with step motion even in the case of materials with different strengthening mechanisms.

4.2. Influence of Step Motion Stop Time

As discussed above, the strain substitution that occurs with the stress relaxation phenomenon progresses with time. The present results from tests of step motion application show that the load relaxes over time, as well, and it is conceivable that this influences the stop time effect of formability enhancement. In the interest of productivity, it is desirable to shorten stop time to the maximum extent, so verification tests were conducted to determine the shortest necessary stop time that would yield a stampability enhancement effect. Varying stop times of 0, 0.1, 0.3, 0.5, and 1.0 seconds were used, as shown in Fig. 10. The stop time of 0 seconds represents crank motion. Rupture was found to occur with crank motion, but ruptures ceased to occur while necking as a result of localized sheet thinning did occur with stop times of 0.1 and 0.3 seconds. When the stop time was extended to 0.5 seconds and longer, the necking also ceased to occur and forming was able to produce quality parts. It was determined from the above

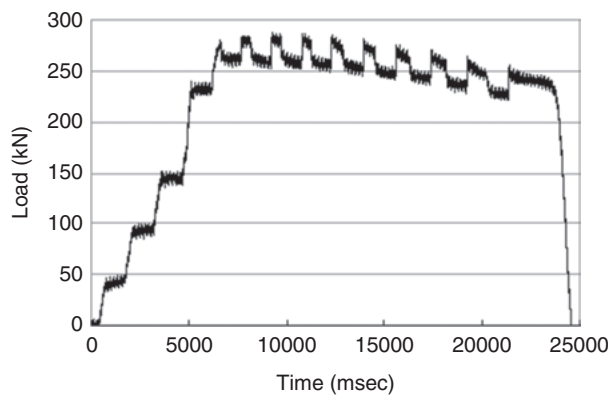


Fig. 9 Forming load in step motion (Time-Load)

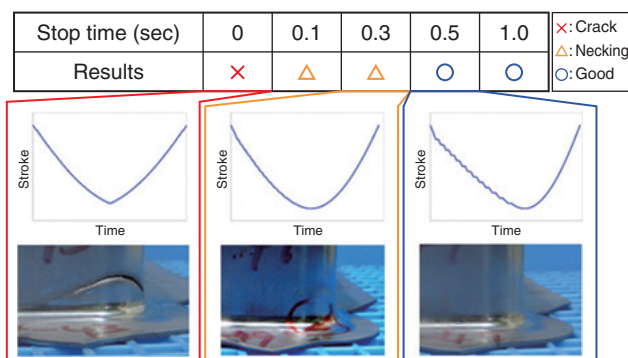


Fig. 10 Influence of stop time

results that the effect of step motion is dependent on stop time, and, with the stamped part shape used here, a die stop time of 0.5 seconds or longer produced stamping products of favorable quality.

4.3. Examination of the Stampability Enhancement Mechanism

Tensile testing with JIS13B test pieces⁽⁸⁾ was carried out in order to verify the influence of step motion on deformation behavior in the material. The test method was to employ two patterns of constant tension speed and step motion in order to confirm the effect of stopping in step motion. In step motion, processing was stopped for 10 seconds for every $\Delta l=1$ or $\Delta l=2$.

Figure 11 shows the stress-strain curves for the two patterns that were tested. As is apparent from the figure, the application of step motion in tensile testing caused the total elongation to increase. The corresponding stresses for the same amounts of strain have the same values here with both constant speed and step motion, and no difference was found between the two.

The sheet thickness distribution in the tensile test pieces was therefore checked in order to verify the influence on rupture of the material. Tensile testing was performed with the two patterns. Test pieces that had the load on them released when the elongation reached $\Delta l=11$ mm with respect to the distance between reference points (50 mm) were used. The thickness distribution in these test pieces was measured in the longitudinal direction and the results were compared.

As shown in Fig. 12, the pieces tested at a constant speed showed an extreme reduction in thickness only near the middle of the test pieces (maximum rate of reduction in sheet thickness 11.3%). On the other hand, pieces processed using step motion do not show as extreme a reduction in thickness, and even in those parts where the greatest thinning occurred, the rate of reduction in sheet thickness was 9.5%. This thinning occurred in a relatively average manner. A further comparison was then

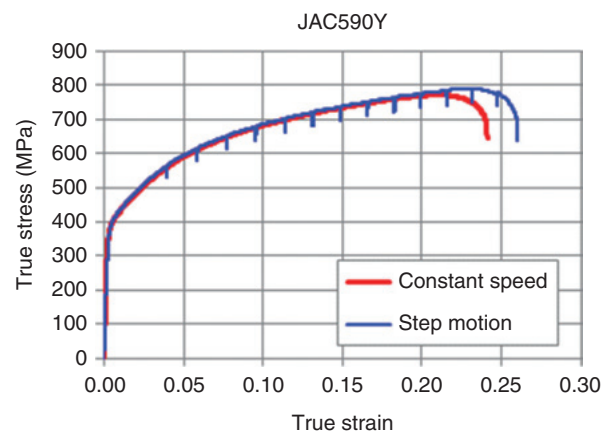


Fig. 11 Strain-stress curve

conducted according to the difference that results when the minimum thinning rate is subtracted from the maximum thinning rate within a range of ± 25 mm from the test piece center. This showed that the application of step motion rather than constant speed in testing suppressed localized deformation to approximately one-third, and confirmed that strain dispersion does have the effect of suppressing local deformation. It can be considered that this change in the type of deformation enabled use of the material's intrinsic deformability to good advantage and increased the total elongation of the material.

4.4. Examination of the Stop Time Dependence of Step Motion

Tensile testing applying step motion was carried out with stop times of 1, 5, and 10 seconds and the thickness distribution was checked in order to verify the time dependence of the deep-drawability enhancement effect found through the stamping tests. First, as shown in Fig. 13, testing with a stop time of 1 second resulted in

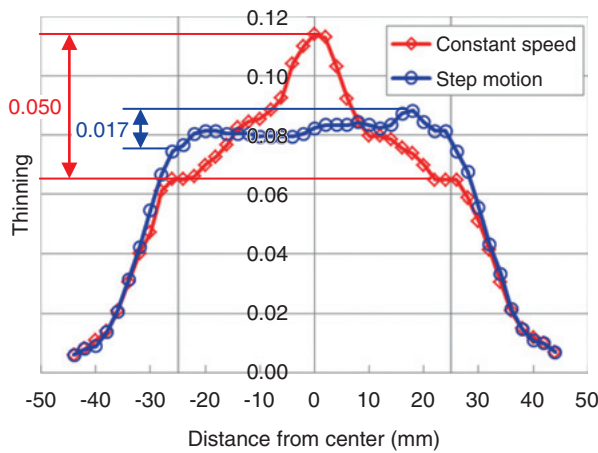


Fig. 12 Comparison of thickness distribution in constant speed and step motion ($\epsilon=22\%$)

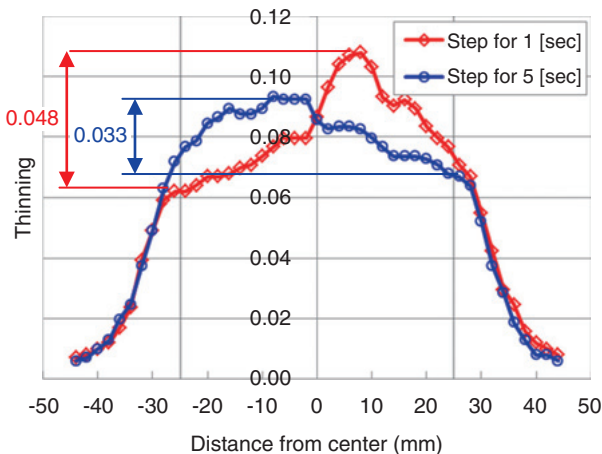


Fig. 13 Comparison of thickness distribution in 1 second stop and 5 second stop ($\epsilon=22\%$)

localized thinning similar to that found when testing with constant speed. The maximum rate of reduction in sheet thickness was 10.8%. Next, when tested with a stop time of 5 seconds, the rate of reduction in sheet thickness was suppressed to 9.2%, and there was a tendency toward suppression of the localized reduction in sheet thickness in comparison with the testing with a stop time of 1 second. Here again, a comparison was conducted according to the difference that results when the minimum thinning rate is subtracted from the maximum thinning rate within a range of ± 25 mm from the test piece center. This showed that local deformation with a stop time of 1 second was generally the same as in testing at a constant speed. With a stop time of 5 seconds, however, the thinning was suppressed to approximately two-thirds. These results confirm that the effect from application of step motion in tensile tests is dependent on the stop time much as it is in stamping tests.

A formability enhancement effect was confirmed in stamping tests with stop times of 0.5 seconds or more, but no such effect was apparent in tensile testing even at 1.0 second. Figure 14 shows the results from comparison of the two in terms of changes in stress when stopped. In the stamping tests, the relaxation of load occurred in an extremely short time by comparison with the tensile testing. With a stop time of 1.0 second, the load relaxation rate was approximately doubled or more.

This difference between the two can be considered to result from the influence of temperature differences or strain speed differences that arise because, while stamping is a high-speed process, tensile testing is conducted at extremely low speed so as to avoid processing heat generation. The influence of strain speed appears unlikely to be a direct factor in this case because the stress relaxation phenomenon is manifested after a temporary stop. Consequently, the influence of temperature was examined as follows.

The magnitude of internal friction with respect to temperature is expressed by F_e in the schematic view

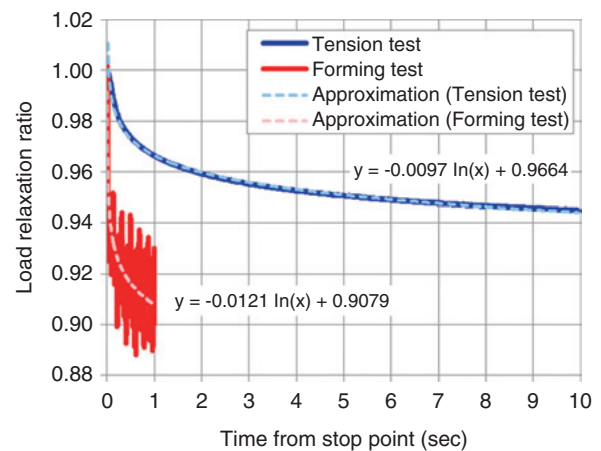


Fig. 14 Time per load relaxation behavior

of Bordoni peaks shown in Fig. 15⁽⁵⁾. It is dependent on temperature and it has two peaks. In order to examine the temperature dependency of the stamping and tensile testing described here, the temperatures of the respective materials used in these tensile tests and in deep-draw stamping were checked against each other, and significant differences in internal frictional force were found to exist, as shown in the figure. It may be inferred specifically that these differences in internal frictional force are closely involved in the stress relaxation time.

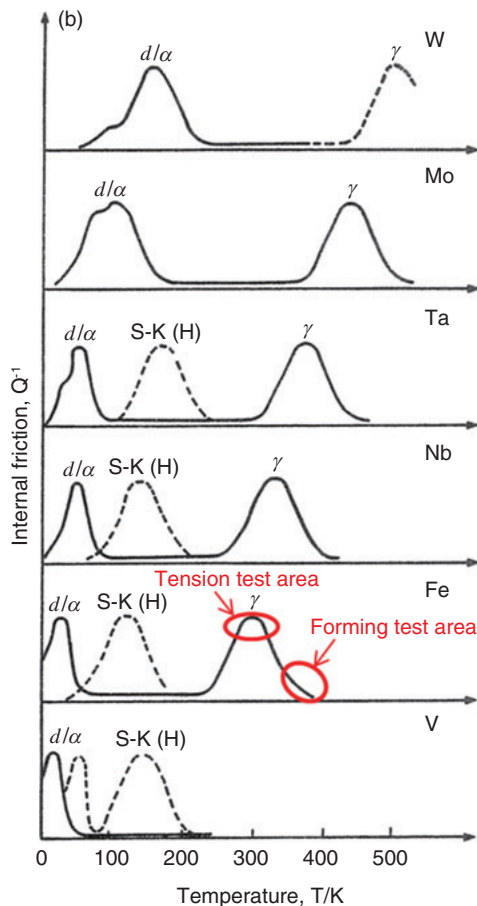


Fig. 15 Image of Bordoni peaks

5. Conclusion

An examination was made of the enhancement of stampability by means of strain dispersion that makes use of the stress relaxation phenomenon. The following conclusions were reached as a result:

- (1) The introduction of step motion in stamping so as to generate stress relaxation by temporarily stopping the die during processing caused a 40% increase in the forming limit depth.
- (2) The suppression of local deformation was confirmed to take place in tensile testing conducted to recreate a step motion, in contrast with tensile testing at constant speed.

- (3) Tensile tests and stamping tests confirmed that step motion has the effect of extending the forming limit and that this effect is dependent on the temporary stop time.

Given the above, the dispersion of strain using the stress relaxation phenomenon, which was presented as a hypothesis, was found to be effective in enhancing formability.

References

- (1) Azuma, Y., Nakagawa, H.: Sustainable na Zidousya Zairyou Gizyutsu, Forum Text, No. 11 FORUM-9, JSAE (2011) (in Japanese)
- (2) Tamai, Y., Yamasaki, Y., Yoshitake, A., Imura, T.: Technology of improving press formability applying press motion control - Advanced multi-stage forming with servo press -, Proc.IDDRG2006, p.403-408 (2006)
- (3) ASTM E328-02, Standard test methods for stress relaxation for materials and structures
- (4) Tagawa, T., Haramishi, Y., Minami, F.: Stress Relaxation Behavior of Low Carbon Structural Steels, Quarterly Journal of the Japan Welding Society, Vol. 29, No. 1, p. 48-54 (2011)
- (5) The Japan Institute of Metals, Zairyou Kyoudo no Genshiron, Maruzen Publishing, p. 71 (1985) (in Japanese)
- (6) Minier, C., Lauzier, J.: Proc. of Yamada Conf., Point Defects and Defects Interactions in Metals, Kyoto, (1981), Univ. Tokyo Press, Tokyo, p.749 (1982)
- (7) The Japan Iron and Steel Federation: Hot-dip galvanized steel sheet and strip for automobile use, JFS A 3011 (2008)
- (8) JIS Z 2201, Test pieces for tensile test for metallic materials

Author



Hiroyuki YAMASHITA



Hiroaki UENO



Hiroyuki NAKAI



Takahiro HIGAKI

Development of Multi-scale Simulation Methods for Polymer Electrolyte Membrane Fuel Cell

Tetsuya KOIDO*

Katsunori MAKINO*

Yuki ITO*

Nobuhiro KUSUMI*

Mahito CHIBA*

ABSTRACT

It is expected that polymer electrolyte membrane fuel cells (PEMFC) will be increasingly used as the power plants of automobiles in the future due to their high thermodynamic efficiency utilizing electrochemical reaction. PEMFC have a detailed internal structure including electrode catalysts, and various phenomena interact in a complex manner across a wide range of scales within this structure. Particularly important phenomena are thought to be (1) the interaction between the elementary reactions and the adsorbates on the electrode catalysts, (2) proton hopping within the polymer electrolyte, and (3) condensation of water droplets within the micro-porous electrolyte. However, these phenomena are rooted in nano-scale mechanisms, so they have not been investigated in sufficient detail in past studies. This paper focused on these issues from a computational science perspective, created appropriate models of the PEMFC internal environment, and applied the dynamic Monte Carlo method to phenomenon (1), a quantum molecular dynamics method based on tight binding approximation to phenomenon (2), and a molecular dynamics with constant chemical potential method to phenomenon (3). The results confirmed that each computational approach effectively simulates the respective issue. In addition, the various effects were calculated in a coupled manner as a multi-scale model to enable these nano-scale phenomena to be reflected when calculating the PEMFC unit cell performance. The results showed that the above phenomena interact and influence the unit cell current-voltage characteristics.

1. Introduction

Interest in energy issues and reducing environmental loads is increasing, and efforts are being made around the world to develop and popularize fuel cell vehicles as a solution to addressing these issues. Further enhancement of performance is desired in terms of cost and durability to help realize mass popularization of fuel cell vehicles in the future⁽¹⁾. Increasing PEMFC efficiency is an effective means of achieving this, which requires clarification of internal phenomena. PEMFC have a detailed internal structure including electrode catalysts, and various phenomena interact in a complex manner within this structure; therefore, it was thought that a computational approach would be effective. This paper reports the construction of numerical computation methods that accurately describe essential mechanisms, which have not been addressed thus far among the various PEMFC internal phenomena.

2. Development Goals

While issues facing the popularization of automobiles that use PEMFC span a wide range, the most important are

reducing cost and enhancing durability, and increasing the fuel cell efficiency is essential for addressing these issues. This study aimed to predict generating performance via the detailed phenomena occurring within the unit cell, which is the minimum generating unit inside the PEMFC stack. **Figure 1** shows a simplified schematic of the PEMFC unit cell structure. The structure consists of the cathode and anode electrodes, a polymer electrolyte membrane located between the cathode and anode, and conductive gas diffusion layers, gas channels and conductive current collectors outside the cathode and anode. The dimensions of the various components are generally on the order of approximately 10 μm thick for each of the electrodes and the electrolyte membrane, 100 μm thick for the gas diffusion layer, and 1 mm deep and 10 cm long for the gas channels.

The typical format for each electrode, which is called the electrocatalytic layer, consists of a precious metal catalyst powder coated onto a conductive support and impregnated with an electrolyte. As shown in Fig. 1, the hydrogen dissociated at the anode migrates inside the polymer electrolyte as protons, passes through the external circuit as electrons, and reacts with the oxygen at the cathode to

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form water. The hydrogen and oxygen pass through the gas diffusion layers on the outsides of the respective electrodes, and are transferred via the gas channels. The physical quantities that express these phenomena are thought to be, at the locations indicated in the figure, the respective molecule concentrations (C), the fluid density and velocity (ρ , $\mathbf{u}$), and the electrolyte and electron potentials (Φ). This study aimed to develop techniques for predicting these physical quantities by retracing the mechanisms by which the phenomena manifest.

The electrocatalytic layer is an important location where the actual generation is performed, and Fig. 2 shows an enlarged image of the actual catalyst layer made for

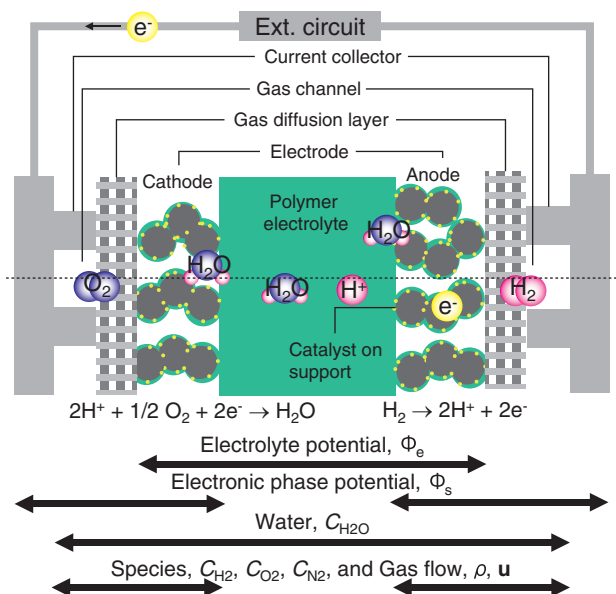


Fig. 1 Typical structure with transported species of unit cell of PEMFC. Related transport phenomena are also indicated.

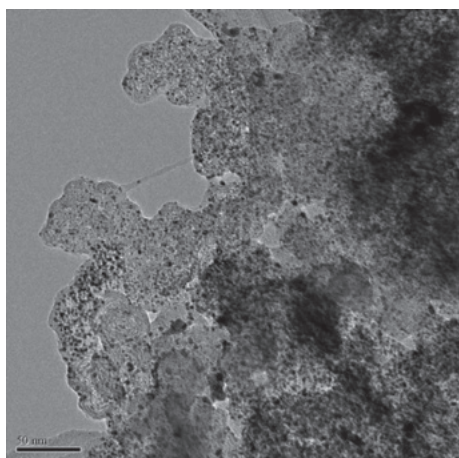


Fig. 2 TEM image of a catalyst layer of polymer electrolyte fuel cell. Grey objects are carbon support and polymer electrolyte. Black dots are platinum particles.

this study. The carbon and the electrolyte appear as gray. Platinum particles appear as black dots, and are supported on carbon (Ketjen Black) with a diameter of several 10 nm. This catalyst layer was made by spraying a liquid mixture of the polymer electrolyte and the catalyst onto the gas diffusion layer. The coating amounts were 0.5 mg/cm^2 for both the platinum and the carbon, and Nafion[®] was coated at 0.4 mg/cm^2 as the polymer electrolyte.

The important issues for enhancing the current-voltage characteristics are to reduce (1) the activation overpotential that derives from the activation energy needed to initiate the reaction at the electrocatalyst, (2) the resistance overpotential that mainly originates in the proton and other ion transfer resistance, and (3) the concentration overpotential produced by a lack of reactive matter at the reaction surface. Respectively, (1) the activation overpotential can be reduced by constructing a cathode electrocatalytic layer that facilitates the electrochemical reaction, (2) the resistance overpotential can be reduced by developing an electrolyte membrane with high proton conductivity, and (3) the concentration overpotential can be reduced by increasing the diffusivity of the reaction gas transfer paths that lead from the gas channels to the electrode catalysts, and by reducing obstruction of transfer by liquid water, particularly at high-current density operation.

Since the mechanisms that produce these phenomena need to be investigated down to the nano scale, they had not been sufficiently clarified thus far. Therefore, this paper attempted a computational science approach using a multi-scale model (a model that links phenomena of differing scales) that enables reflection of nano-scale mechanisms to the unit cell scale (several cm). The main goals when developing and applying the computational techniques were to take into account the following three effects:

- (1) Influence of proton hopping on proton conductivity;
- (2) Adsorbate status on the catalyst surface depending on electrode potential; and
- (3) Blocking of nano-pores in the catalyst layer by water droplets.

These correspond respectively to the three overpotentials (1) to (3) described above. Appropriate simplified models based on many assumptions need to be made in order to simulate these complex phenomena using computational science. Therefore, the validity of the various models was also verified.

In addition, techniques were also developed to calculate the various effects in a coupled manner to understand how individual phenomena manifest as fuel cell performance.

3. Computational Methods

3.1. Calculation of Surface Electrochemical Reaction Using the Dynamic Monte Carlo Method

In the electrode catalyst surface reactions, various elementary reactions involving adsorption and desorption

of chemical species occur in succession. For this reason, electrode potential should be taken into account when calculating adsorption, reaction and desorption on the reaction surface and the interaction between the adsorbates. The vector that represents the state of all sites in the adsorption site lattice on the catalyst surface is labeled $\mathbf{c}$. The equation that expresses the change over time of the probability $P_{\mathbf{c}}(t)$ that the status $\mathbf{c}$ will manifest at time t can generally be expressed as follows.

$$dP_{\mathbf{c}}(t)/dt = \sum_{\mathbf{c}' \neq \mathbf{c}} [k_{\mathbf{c}\mathbf{c}'} P_{\mathbf{c}'}(t) - k_{\mathbf{c}'\mathbf{c}} P_{\mathbf{c}}(t)] \quad (1)$$

Here, $k_{\mathbf{c}\mathbf{c}'}$ is the rate constant of the elementary reaction in which the state $\mathbf{c}'$ changes to $\mathbf{c}$. Since this equation is a challenge to solve analytically, this study used the dynamic Monte Carlo method (DMC)⁽²⁾, which is a probabilistic method. The DMC method first investigates the states of adjacent sites around all the reaction sites in status $\mathbf{c}$, and extracts the sites where the elementary reaction i can occur. Next, the probability of reaction i occurring is determined from the number of sites n_i where this reaction can occur and the reaction rate k_i , and this result is used to probabilistically determine which reaction will occur and initiate the reaction. In addition, the time required for a single reaction is determined probabilistically from the sum total of the times for all possible reactions to occur in this state. The reaction progress can be clarified by repeating this process.

The types of oxygen-reduction elementary reactions taken into account in this paper are described in the following chapter. Each elementary reaction rate is given by the Arrhenius equation expressed by Eq. (2).

$$k = A \exp(-E_{\text{act}} / (k_B T)) \quad (2)$$

Here, k_B is the Boltzmann constant, T is the absolute temperature, A is the frequency factor, and E_{act} is the activation energy. When quantifying the elementary reaction rates, they are considered as divided into reactions that include electron transfer and normal chemical reactions. In case of electron transfer reactions of adsorbed chemical species on platinum atoms, E_{act} is thought to depend on the electrode potential V , which uses the equilibrium electrode potential V_{rev} as the reference, in the manner of $E_{\text{act}} = f(V - V_{\text{rev}})$. Therefore, this was formulated as a conditional minimization problem of the energy obtained by first-principles calculation, and E_{act} was obtained by the method of Lagrange multipliers⁽³⁾. This condition is based on the assumption that the electron affinity in the reaction transition state is equal to the work function of the electrode at that potential and the Born-Oppenheimer (BO) approximation, which assumes that the motion of the atomic nucleus during electron exchange is sufficiently slow relative to the change in the electron

state. The equilibrium electrode potential V_{rev} can be calculated thermodynamically using the Nernst equation⁽⁴⁾. The frequency factor A is calculated based on a model that adds the protons in the electrolyte to the adsorbed oxygen atoms as described in the following chapter. In the case of normal chemical reactions, each factor in Eq. (2) is determined according to transient state theory⁽⁵⁾. In this case, the activation energy is the constant for each elementary reaction.

3.2. Tight Binding Molecular Dynamics (TBMD)

The proton transfer phenomena within the electrolyte are said to be due to two competing mechanisms; the vehicular mechanism and the Grotthuss mechanism⁽⁶⁾. Therefore, both mechanisms should be taken into account to predict the transfer rate. Of the two, the vehicular mechanism transfer phenomenon can be simulated by classical molecular dynamics calculations⁽⁷⁾. However, the Grotthuss mechanism is also known as proton hopping, which is the phenomenon where the water molecule adjacent to a hydronium ion (H_3O^+) changes to H_3O^+ as if a proton in the H_3O^+ is transported as shown in Fig. 3, which results in high-speed charge transfer. The Grotthuss mechanism involves reconfiguration of the chemical bonds, therefore the electronic state cannot be simulated using molecular dynamics calculation based on classical dynamics, and needs to be handled using quantum dynamics. The first-principles molecular dynamics approach supports this; however, an extremely large calculation cost is required to handle polymers. Therefore, in this study we developed a new computational code using the tight-binding molecular dynamics⁽⁸⁾ (TBMD) method that takes into account quantum chemistry. This method calculates the system energy and the electronic state of the molecules by solving a Schrödinger equation based on linear combination of atomic orbitals (LCAO) approximation. In this method, the calculation speed is increased by omitting the electron repulsion integral (ERI) calculations that have a high calculation cost. The equation of motion for the atomic nucleus A assumes the BO approximation described above, and is expressed as follows.

$$\mathbf{F}_A = \frac{\partial E}{\partial \mathbf{R}_A} = - \sum_{B \neq A} \sum_i^{\text{atoms MO}} n_i \sum_{rs}^{\text{AO}} C_{ir}^T \left(\frac{\partial H_{rs}}{\partial \mathbf{R}_A} - \frac{\partial S_{rs}}{\partial \mathbf{R}_A} \right) C_{is} - \sum_{B \neq A}^{\text{atoms}} \frac{\partial E_{\text{rep}}(\mathbf{R}_{AB})}{\partial \mathbf{R}_A} \quad (3)$$

Here, C_{ir} is a coefficient indicating the contribution of the basis function of the r th atomic orbital relative to the i th

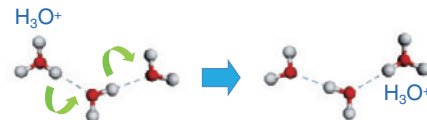


Fig. 3 Proton transfer by Grotthuss mechanism

molecular orbital, n is the number of electrons present in the i th molecular orbital, H is the Hamiltonian, S is the overlap integral, $\mathbf{R}$ is the position of the atomic nucleus, and the subscripts A and B indicate the atoms. To correct the short-range forces, this study enhanced the quantitativity by adding a term for the repulsion energy between the nuclei and the electrons to the second item on the right side of the equation. By omitting the ERI calculations as described above, the Hamiltonian matrix elements become adjustment parameters, and are determined in a manner to reproduce the results of separate first-principles calculation. In these calculations, the relationship between the change in energy and the O-H bond length in the H_3O^+ ions that affects the proton hopping frequency was adjusted to reproduce the results of accurate first-principles calculation using the MP2 method⁽⁹⁾. This method enabled performance of the quantum molecular dynamics calculations for approximately 1000 atoms using a single 8-core workstation.

3.3. Molecular Dynamics with Constant Chemical Potential

The gas-liquid phase transition point is known to change inside nm-order pores due to the phenomenon known as capillary condensation. Recently it has been confirmed that some phenomena cannot be adequately expressed by the Kelvin equation conventionally used to predict this phenomenon⁽¹⁰⁾. An effective means of accurately predicting the above phenomena within the PEMFC electrocatalytic layer is to perform direct calculations using the molecular dynamics method, taking into account the characteristics of the electrocatalytic layer wall surfaces. In the state where both the gas and liquid phases coexist within the pores, the temperatures and chemical potentials (Gibbs free energy per molecule) of both phases are equal, although the pressures differ. Although the molecular dynamics method generally performs calculations with the temperature and pressure fixed, this study used the Grand Molecular Dynamics (GMD) method⁽¹¹⁾ to perform calculations with the chemical potentials of both phases fixed to be equal. This method handles the issue of particle transfer into and out of the system by solving the equation of particle motion of a virtual particle that takes the value of 0 to 1 as the number of the particle in addition to the normal particles handled by molecular dynamics. This enables control of the chemical potential of the system to be calculated. Isochoric calculations that control the chemical potential and temperature were performed by introducing the Nose-Hoover method⁽⁷⁾ as the temperature control method to this equation of motion that takes into account virtual particles.

4. Modeling of PEMFC and Calculation Results

A target system was modeled to apply the methods described in the previous chapter to PEMFC, and the various calculation results were verified. The method used

to couple these results and calculate the current-voltage characteristics is described below.

4.1. Calculation of Electrochemical Reaction Rates

4.1.1. Modeling

The PEMFC cathode normally uses platinum catalyst particles, which have high oxygen reduction activity, coated on a carbon support and then covered by the electrolyte component. For the oxygen reduction reaction it was hypothesized that the oxygen dissolved in the water that exists together with the electrolyte is adsorbed to the platinum surface and reacts with the protons in the water. This reflects the fact that the diffusion coefficient of oxygen is larger in water than in Nafion[®], and that positively charged protons do not easily adsorb to the oxygen reduction electrode, which has a high potential.

The elementary reactions calculated in this study were narrowed down to the following reactions based on reference (12).

Adsorption and desorption of oxygen molecules and liquid water



Reduction of adsorbed oxygen molecules (Because the dissociation reaction after proton addition occurs easily, it is included in this reaction.)



Reduction of adsorbed oxygen atoms



Reduction of adsorbed hydroxyl groups



Desorption of adsorbed water



The asterisks (*) in the equations represent the adsorbed state to the metal surface. Here, the reaction activation energy in the electrochemical reactions expressed by Eq.

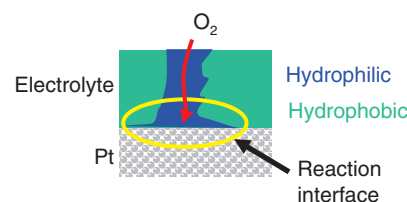


Fig. 4 Image of reaction interface

(6), (7) and (8) depends on the potential as described in the previous chapter. Assuming BO approximation, the energy in each adsorbed state in the reaction equations is calculated by first-principles calculations using DFT, and the equilibrium electrode potential is converted to the potential difference using the Nernst equation, based on this energy.

The platinum catalyst particles are assumed to consist of (111) facets and (100) facets with low surface energy and to exist in the stable state. The typical particle shapes are cubes and truncated octahedrons. In addition, all platinum surfaces are assumed to be electrochemically effective.

Two types of adsorbed states were confirmed for the water molecules on the platinum surface. One has two O-H bonds oriented parallel to the platinum surface and exists near the surface, and the other has a single O-H bond oriented perpendicular to the platinum surface and exists farther away from the surface. In addition, it has also been reported that an ordered structure results due to the hydrogen bonding that occurs between these states⁽¹³⁾. In consideration of these findings, this study calculated the state of the water adsorbed with an ordered structure on the platinum surface using DMC calculations beforehand, and used these results as the initial state to calculate the surface reaction. At this time the parallel-oriented water molecules described above have high adsorption energy; therefore, they are considered to be a substance not included in the oxygen reduction reaction system taken into account in these calculations, and it is hypothesized that only the perpendicular-oriented water molecules react.

In addition, the frequency factor based on transient state theory was corrected according to the proton concentration per unit volume in order to reproduce the water electrolysis tests using platinum electrodes⁽¹⁴⁾.

4.1.2. Calculation results

Figure 5 shows the activity of Pt(111) and Pt(100). The higher activity of Pt(111) was also exhibited in tests⁽¹⁵⁾, and matches qualitatively with the results of these calculations.

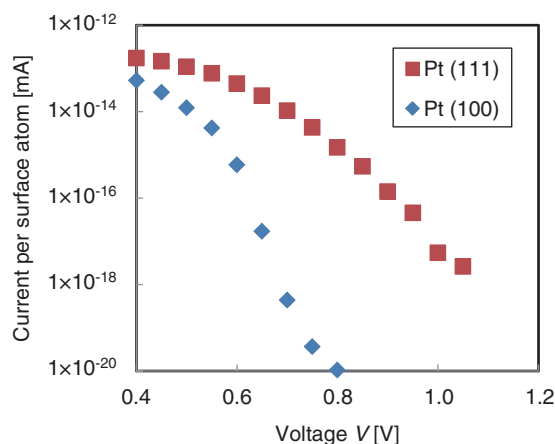


Fig. 5 Current per surface atom against cell voltage

Table 1 Change of Gibbs free energy ΔG , in eV

Reaction	Pt (111)	Pt (100)
OO $\rightarrow$ OOH	-0.82	-0.79
O $\rightarrow$ OH	-0.94	-1.27
OH $\rightarrow$ H ₂ O	-1.17	-0.53

Figure 6 shows snapshots of the adsorbed species states on the platinum surfaces at a voltage of 0.8 V. The Pt(100) is covered by OH groups and water molecules adsorbed parallel to the surface that do not easily contribute to the reaction as described previously, indicating that the OH-to-water reaction expressed by Eq. (9) is stagnating.

To help consider the reasons for this, Table 1 shows the change in the Gibbs free energy obtained by DFT calculation in the electrochemical reactions in the rightward directions of Eq. (6), (7) and (8). Here, the reaction equation (8) for Pt(100) suggests that the reaction does not proceed easily compared to Pt(111).

Next, the dependency of the activity on particle size was calculated for the typical shapes. The particle current density was estimated by adding the respective activity levels shown in Fig. 5 corresponding to the area of each particle surface orientation. Here, the adsorption energy of separate chemical species was calculated for the joints between surfaces (edges and apexes). Because the joints were found to have extremely high adsorption energy, they were considered to have low electrochemical activity, and were excluded from the reaction calculations. Figure 7 shows these results. This shows that the catalyst particle size has an optimum value in terms of activity, and that a regular octahedron is better as the shape.

4.2. Calculation of Proton Transfer within the Electrolyte

4.2.1. Modeling

This study targeted Nafion[®], which is generally used as the PEMFC electrolyte. Nafion[®] consists of a main chain made of carbon and fluorine, and a side chain with a sulfone group at the tip. Since the molecular weight

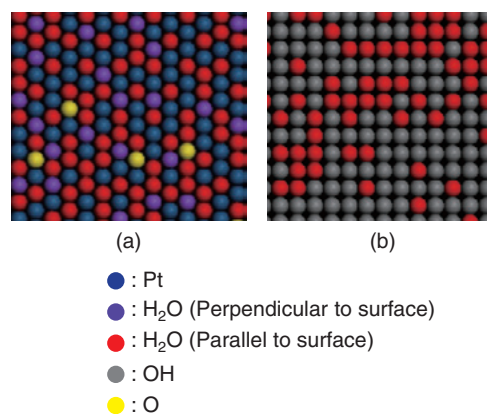


Fig. 6 Adsorption of species at 0.8V on Pt (111) ((a)) and Pt (100) ((b))

exceeds 10000 - 100000, it cannot be handled as is by the developed TBMD code in this study. Therefore, this study performed calculations with the degree of main chain polymerization, which is actually between 100 and 1000, set to 4, and the side chain simplified as shown in Fig. 8. TBMD calculations were performed for a sufficient time in the equilibrium state with a time increment of $\Delta t = 0.05$ fs, while controlling to a constant volume and a constant temperature of 300 K, and the results were used to calculate the H_3O^+ ion migration distance. The lengths of all O-H bonds were recorded during the calculations in order to track the H_3O^+ ion reconfiguration due to proton hopping. In this calculation, the ion exchange capacity (*IEC*, mmol/g) was used as the side chain density parameter, and the water content λ (number of water molecules per sulfone group) was used as the water concentration parameter. Figure 9 shows the calculated state when $\lambda = 12$ and *IEC* = 0.87 mmol/g.

During PEMFC operation, a nonequilibrium state called potential gradient occurs within the electrolyte, resulting in macro transport of the charge (H^+) accompanied by the electro-osmosis phenomenon whereby the water molecules also migrate. However, in these calculations the proton conductivity is calculated from the proton diffusion coefficient in the equilibrium state. Water transfer due to electro-osmosis is described separately in Sub-section 4.4.2.

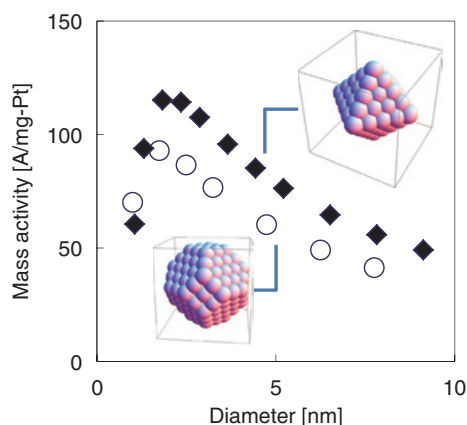


Fig. 7 Effect of catalyst particle size and shape on electrochemical activity

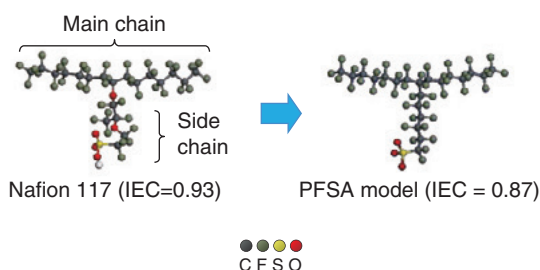


Fig. 8 Simplification of chemical structure

4.2.2. Calculation results

Figure 10 shows the calculated proton conductivity. The proton conductivity was calculated as follows. Each self-diffusion coefficient is calculated based on the mean-square displacement of the H_3O^+ , and this is then converted to the proton conductivity according to the Nernst-Einstein relation. Because the TBMD method is unable to take into account long-distance interactions, the hydrogen bonding energy cannot be quantitatively calculated. Therefore, this computational method evaluates the ratio (calculated value/measured value) of the water self-diffusion coefficient separately, and uses this to correct the proton conductivity. Figure 10 shows the change in proton conductivity due to the change in the side chain density obtained by this calculation, converted to the ion exchange capacity (*IEC*). The tendency of the proton conductivity to rise together with the change in water content closely matched the Nafion[®] proton conductivity measurement results reported by Zawodzinski et al.⁽¹⁷⁾. Since this tendency cannot be adequately reproduced by classical molecular dynamics calculations⁽¹⁷⁾, it is thought that this method enables understanding of the dependency on the water content by using quantum dynamics to solve proton hopping. In addition, based on these calculation results, increasing the *IEC* is predicted to greatly increase the proton conductivity, especially in the low-water-content range where $\lambda < 6$.

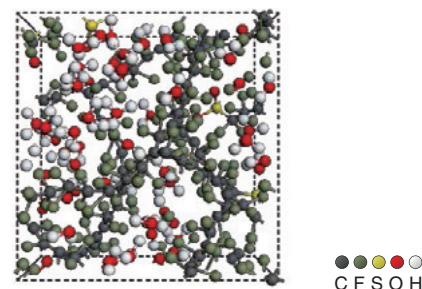


Fig. 9 Snapshot of TBMD calculation ($\lambda = 12$, *IEC* = 0.87 mmol/g)

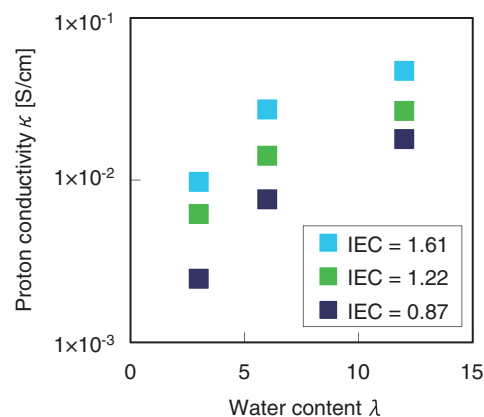


Fig. 10 Predicted conductivity

4.3. Calculation of Water Blockage inside Nano-pores

4.3.1. Modeling

Normally the PEMFC electrode catalyst consists of precious metal particles having a diameter of several nm coated onto carbon support having a diameter on the order of several tens of nm, and impregnated with electrolyte components. The remaining areas become gaps on the order of several nm as shown in Fig. 2. This study focused on the capillary condensation of water in these gaps.

Although the PEMFC is operated at a nonequilibrium state where water generation occurs continuously in the catalyst layer, this computational method assumed that this could be expressed by equilibrium state calculations. The calculations neglected the potential field within the catalyst layer, and focused on only the carbon wall surface, modeling it as follows. The gap was modeled as the space between two infinitely wide parallel flat plates, and the behavior of the water molecules inside this space was calculated using the chemical potential and the temperature as parameters. The calculations assumed a fixed interatomic distance within the water molecules, and the atoms comprising the wall surfaces were also fixed. The gap size is represented by the distance between the flat plates, and the wall surfaces are represented by graphite (graphene sheets). Figure 11 shows these parallel flat plates and the initial state for the calculations. The interaction between the atoms in the water is expressed by the SPC/E potential, the interaction between the graphite and the water is expressed by the LJ potential in which the parameters are set to reproduce the angle of contact between the flat plate surfaces and the water droplets of 86° , which is weakly hydrophilic⁽¹⁸⁾.

The change in the boiling point of the water inside the nano-pores was calculated to investigate the validity of the molecular dynamics calculations with constant chemical potential developed by this study. Since the catalyst layer surface was represented by a graphene sheet, the carbon potential was applied. In the case of a hydrophilic surface such as carbon, the boiling point inside the pores rises. The developed computational method calculated a rise in the boiling point of 5.8 K for an equivalent pore radius of 4 nm. In contrast, the rise in the boiling point for mesoporous carbon, which has the same modal pore diameter in the gas adsorption method, was 8.7 K measured by the DSC method. On the other hand, the Kelvin equation that has generally been used to predict the phase-change temperature inside the pores produced a value of 1.5 K. Based on these

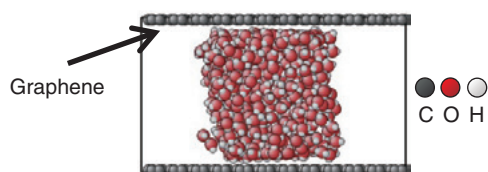


Fig. 11 Computational model of pore and initial state

results, it is thought that the developed computational method, which takes into account the behavior of the water molecules relative to the wall surface potential, is able to more accurately predict the actual phenomena.

4.3.2. Calculation results

The developed computation method was used to calculate the pressure at which the liquid phase evaporates at a certain temperature relative to the distance between the flat plates. The pressure at which the phase change occurs was obtained by performing GMD calculations multiple times with different chemical potentials, searching for the point at which the number of molecules decreases rapidly, and using the pressure value at that point. The temperature-pressure-pore diameter (distance between the flat plates) characteristics for the gas-liquid phase change can be obtained from this result. Next, the pore diameter distribution of an electrocatalytic layer created by the method described in Fig. 2 was measured using the gas adsorption method. The results were applied to the gas-liquid phase change characteristics described above, and the degree to which the electrocatalytic layer was blocked by liquid water according to the temperature and relative humidity was calculated. The pressure used was the water vapor pressure at the relative humidity. Figure 12 shows these results. For example, in the case of the created catalyst layer, when the temperature is 75°C and the relative humidity is 70%, it is predicted that approximately 10% of the pore volume will be blocked by water.

4.4. Coupled Computation of Multi-physics

4.4.1. Computational method

It is necessary to calculate the various phenomena described thus far simultaneously in a coupled manner to confirm the influence of the various phenomena on device performance. The relationship between the PEMFC current and voltage is determined from the governing equations for each chemical species, the behavior of the mixed gas consisting of these chemical species, the electrolyte phase potential, and the solid phase potential. As shown in Fig. 1, the dependent variables in these

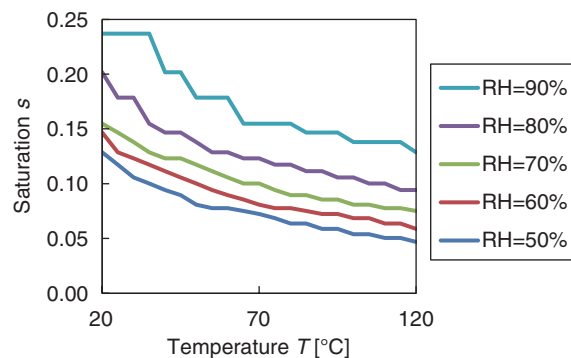


Fig. 12 Estimated pore volume fraction occupied by liquid water at various relative humidities

equations are the oxygen concentration C_{O_2} , hydrogen concentration C_{H_2} , water concentration C_{H_2O} , mixed gas velocity vector $\mathbf{u}$, electrolyte phase potential Φ_e , and solid phase potential Φ_s . The mixed gas density ρ is determined from the concentration of each chemical species and the state equation of the ideal gas. The electrocatalytic layer is considered to be a homogenous porous medium consisting of the three phases of the electrolyte phase that diffuses the protons, the gaseous phase inside the pores, and the support phase that transfers the electrons. The electrolyte phase and the gaseous phase water are assumed to be in the equilibrium state. The electric current generation per unit volume was calculated using a constitutive equation and given as a source term. Labeling the variable to be solved for all of these governing equations as ϕ , this can be expressed in the steady state in the form of the following convective diffusion equation⁽¹⁹⁾.

$$\rho \mathbf{u} \phi = \nabla \cdot (\Gamma \nabla \phi) + S \quad (10)$$

Here, ρ is the density, Γ is the diffusion coefficient, and S is the source term. The various governing equations can be expressed by replacing ϕ with the respective variables described above, and replacement with “1” results in an equation of continuity. Therefore, all governing equations can be solved by applying computational fluid dynamics techniques⁽²⁰⁾. The boundary conditions are the flow rate and composition of the gas flowing into the gas channels, and the current collector potential of both electrodes.

4.4.2. Multi-scale model

Based on the results obtained in the previous section, the various nano-scale phenomena were expressed as a multi-scale model (a model linking phenomena of differing scales) to enable incorporation into unit cell-scale (several cm) computations.

The reaction rate at the cathode was obtained as follows. The current value per platinum atom according to the potential, oxygen concentration and temperature was calculated by DMC calculations, and this current value was then converted to the reaction rate per unit volume $j = j_{MS}(V, T, C_{O_2})$ in accordance with the platinum coated mass per unit area, the area per unit platinum mass, and the catalyst layer thickness. Here, the catalyst effective utilization rate was set at 0.8 as suggested by the reference (21). The oxygen concentration of the catalyst surface used in the DMC calculations was calculated from the gas phase oxygen concentration C_{O_2} using Henry's constant. The effects of the oxygen concentration and the temperature appear via frequency factor A of the reaction rate in the reaction formula (Eq. (2)).

Based on the results in Fig. 12, the degree of catalyst layer pore blockage due to the influence of nano-pores was determined to be a function of the temperature and water concentration, and defined as $s = s_{MS}(T, C_{H_2O})$. Due to this effect, the catalyst surface area was thought to decrease,

Table 2 Models incorporated into the governing equation, Eq. (10). Listed modifications are applied to polymer electrolyte part including electrodes indicated in Fig. 1.

Variable, ϕ	Modified terms
Species, C_m	$\Gamma = D_m^{\text{eff}}$, only to electrodes.
	$S = S_d = \nabla \cdot [(n_d/F)\mathbf{i}_e]$ if $m = H_2O$
	$S = -j^{\text{eff}}/(2F) + S_d$, only to cathode if $m = H_2O$
	$S = j^{\text{eff}}/(4F)$, only to cathode if $m = O_2$ $S = -j^{\text{eff}}/(2F)$, only to anode if $m = H_2$
Electrolyte potential, Φ_e	$\Gamma = \kappa^{\text{eff}}(\lambda)$. $S = j^{\text{eff}}$, only to electrodes.
Electronic phase potential, Φ_s	$S = -j^{\text{eff}}$, only to electrodes.

and the reaction rate j was corrected to $j^{\text{eff}} = (1-s)j$. In addition, the diffusion coefficients for water, oxygen and hydrogen were also corrected to $D_m^{\text{eff}} = (1-s)^\tau D_m$ ($m = H_2O, H_2, O_2$). Here, τ is the tortuosity of the porous medium. The calculations assumed that the degree of blockage s did not increase further at a relative humidity of 99% or more.

In the electrocatalytic layer, the transfer rate of each conserved quantity with respect to the electrolyte phase water and potential, the gaseous phase chemical species in the pores, and the support phase electron potential is thought to be equivalent to the transfer rate multiplied by the volume fraction of the respective phases taken to the tortuosity power.

Based on the IEC dependency obtained by the TBMD calculations, the proton conductivity of the electrolyte membrane was corrected from the reference document value⁽¹⁹⁾ to $\kappa^{\text{eff}}(IEC, \lambda) = f(IEC, \lambda) \kappa_{\text{ref}}(\lambda)$. Various models have been proposed for the number of water molecules that migrate (electro-osmotic coefficient n_d) incidental to the current vector $\mathbf{i}_e$ due to H_3O^+ transfer within the membrane. However, from the MRI measurement results of the proton concentration inside the electrolyte membrane during operation, a water content dependent model was thought to be appropriate⁽²²⁾, and the model given in reference (23) was used.

Arbitrary models used the models given in reference (19), and arbitrary physical property values used generally known values. The diffusion coefficient Γ and source item S in Eq. (10) were modified as shown in Table 2, and calculations were performed with the above models applied to the locations indicated in Fig. 1. F in the table is the Faraday constant. Although the transfer of liquid phase water in the gas diffusion layers and gas channels is also an important issue, this study focused in particular on phenomena in the electrocatalytic layer. Therefore, only blockage of nano-scale pores inside the catalyst layer due to the liquid phase was taken into account. The calculation subject cell used 112 Nafion® as the electrolyte

membrane, simulated the electrocatalytic layer described in Chapter 1, and used three-dimensional shapes that simulate the JARI standard cell for other parts. Since appropriate thermophysical properties were not available, the calculations assumed that there was no temperature distribution and the temperature was fixed to 75°C. The dew point temperature of the inflow gas to the gas channels was set to 75°C at the anode and 55°C at the cathode, and the stoichiometry was set to 4 at the anode and 3 at the cathode. The inlet pressure was set to two times atmospheric pressure. The anode and cathode gas channels flow in parallel to each other on the respective sides of the membrane.

4.4.3. Calculation results

Figure 13 shows the calculated current density-voltage characteristics per unit electrode coated area. These results show a rapid drop in voltage in the low-current range and a drop in voltage in the high-current range, which are well known from experiments. Of particular note is the fact that this computational method calculates the reaction rate at the cathode from only the elementary reaction rate based on first-principles calculations and the measured characteristics. That is to say, arbitrary parameter fitting is not performed.

Next, Fig. 14 shows the calculated internal physical quantities at the cell voltage $V = 0.5$ V. This shows the distributions passing through the membrane in the direction from the cathode to the anode in the channel center section (cross-section along the dashed line shown in Fig. 1) near the inlet, center and outlet of the gas channels. This figure shows that the proton conductivity is widely distributed. This is due to the internal water content λ distribution. The current density per unit volume at the cathode is lower on the electrolyte membrane side at the inlet; however, this relationship reverses at the outlet. The reason is explained as follows: On the inlet side, the water content is low and the proton conductivity is also low accordingly. The electrolyte potential difference increases within the catalytic layer, causing the current density to increase near the electrolyte membrane. On the contrary, on the outlet side, the potential

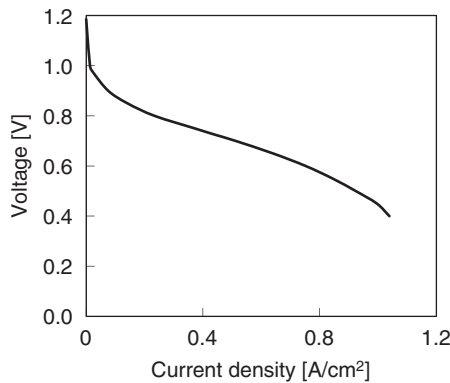


Fig. 13 Predicted voltage-current density curve with multi-scale models

difference is equalized and the oxygen concentration near the membrane drops, so the current density near the membrane drops. In addition, since the relative humidity exceeds 100% partway through the gas channels of both electrodes, the degree of catalytic layer blockage becomes saturated at this time. The fact that the computational model assumed that the degree of blockage s does not rise further at a relative humidity of 99% or more as mentioned previously is thought to have some influence. However, the degree of blockage reached nearly 30%, which is thought to contribute to the drop in oxygen concentration inside the catalyst layer near the outlet. Thus, the phenomena targeted by this study were found to interact and influence performance.

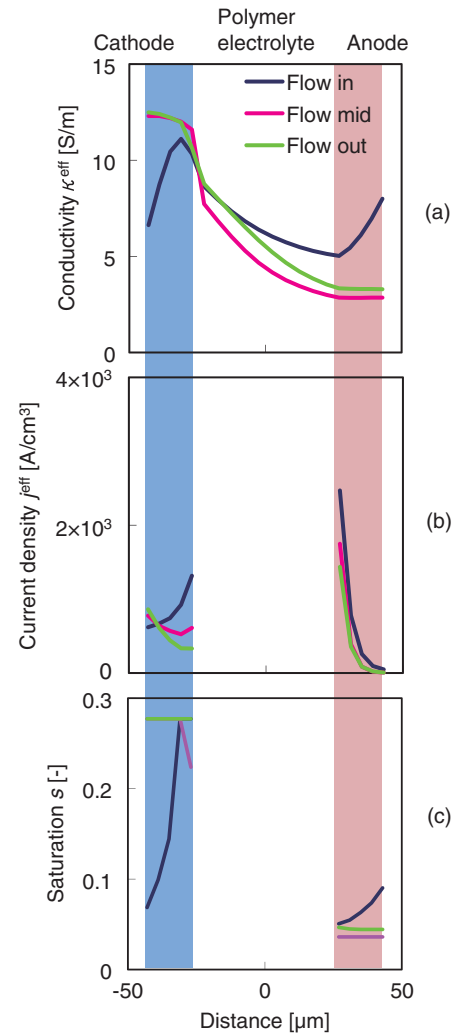


Fig. 14 Predicted distribution of physical quantities through polymer electrolyte membrane, (a) proton conductivity, (b) current density, (c) liquid saturation. The curves indicate values on the plane cut in the middle of the gas channel: Dashed line in Fig. 1. Anode and cathode are shaded with red and blue respectively.

5. Conclusion

Computational techniques that take into account nano-scale mechanisms were constructed for phenomena that had not been given sufficient consideration thus far in predictions of PEMFC performance, and the following knowledge was gained.

- (1) Calculations were performed that take into account the influence of proton hopping on proton conductivity, and clarified that there is a relationship between the ion exchange capacity and the conductivity.
- (2) Calculations were performed that take into account the states of the adsorbates on the catalyst surface in consideration of the potential, and clarified that the catalyst particle size influences the activity per unit catalyst mass.
- (3) Drop-wise condensation was calculated taking into account the influence of nm-order pores, and it was shown that capillary condensation inside the catalyst layer pores cannot be ignored.
- (4) A computational technique was also developed that calculates the effects of individual phenomena in a coupled manner as a multi-scale model that can be reflected to PEMFC unit cell performance. The results showed that the interaction between the above phenomena appears in the performance.

References

- (1) Wang, Y., Chen, K. S., Mishler, J., Cho, S. C., Adroher, X. C.: A review of polymer electrolyte membrane fuel cells: Technology, applications, and needs on fundamental research, *Applied Energy*, Vol. 88, p. 981–1007 (2011)
- (2) Rai, V., Pitsch, H.: First-principles analysis of oxygen-containing adsorbates formed from the electrochemical discharge of water on Pt(111), *Journal of Physical Chemistry C*, Vol. 112, p. 9760–9768 (2008)
- (3) Aryanpour, M. M., Rai, V., Pitsch, H.: Convergent iterative constrained variation algorithm for calculation of electron-transfer transition states, *Journal of the Electrochemical Society*, Vol. 153 (3), p. E52–E57 (2006)
- (4) Rossmeisl, J., Nørskov, J. K.: Calculated Phase Diagrams for the Electrochemical Oxidation and Reduction of Water over Pt(111), *Journal of Physical Chemistry B*, Vol. 110, p. 21833–21839 (2006)
- (5) Glasstone, S., Laidler, K., Eyring, H.: *The theory of rate processes*, McGraw-hill (1941)
- (6) Agmon, N.: The Grotthuss Mechanism, *Chemical Physics Letter*, Vol. 244, p. 456 (1995)
- (7) Ueda, A.: *Molecular simulation, - From classical to quantum methods -*, Shokabo, Tokyo, 446 p. (2003) (in Japanese)
- (8) Laasonen, K., Nieminen, R. M.: Molecular dynamics using the tight-binding approximation, *Journal of Physics: Condens. Matter.*, Vol. 2, p. 1509–1520 (1990)
- (9) Pople, J. A., J. Binkley, J. S., Seeger, R.: Theoretical models incorporating electron correlation, *International Journal of Quantum Chemistry*, Vol. 10, S10, p. 1–19 (1976)
- (10) Klein, J., Kumacheva, E.: Confinement-Induced Phase Transitions in Simple Liquids, *Science*, Vol. 269, No. 5225, p. 816–819 (1995)
- (11) Çağın, T., Pettitt, B. M.: Grand molecular dynamics: A method for open systems, *Molecular Simulation*, Vol. 6, No. 1, p. 5–26 (1991)
- (12) Kim, S. H., Rai, V., Pitsch, H.: Multi-Scale Modeling of Catalyst Layers in PEM Fuel Cells Based on Dynamic Monte Carlo Method for Surface Reactions, 212th ECS Meeting, Abstract #512 (2007)
- (13) Meng, S., Wang, E. G., Gao, S.: Water adsorption on metal surfaces: A general picture from density functional theory studies, *Phys. Rev. B*, Vol. 69, p. 195404 (2004)
- (14) Wang, J. X., Markovic, N. M., Adzic, R. R.: Kinetic Analysis of Oxygen Reduction on Pt(111) in Acid Solutions: Intrinsic Kinetic Parameters and Anion Adsorption Effects, *Journal of Physical Chemistry B*, Vol. 108, p. 4127–4133 (2004)
- (15) Markovic, N. M., Adzic, R. R., Cahan, B. D., Yeager, E. B.: Structural effects in electrocatalysis: oxygen reduction on platinum low index single-crystal surfaces in perchloric acid solutions, *Journal of Electroanalytical Chemistry*, Vol. 377, p. 249–259 (1994)
- (16) Zawodzinski, T. A., Neeman, M., Sillerud, L. O., Gottesfeld, S.: Determination of water diffusion coefficients in perfluorosulfonate ionomeric membranes, *Journal of Physical Chemistry*, Vol. 95, p. 6040–6044 (1991)
- (17) Zhou, X., Chen, Z., Delgado, F., Brenner, D., Srivastava, R.: Atomistic Simulation of Conduction and Diffusion Processes in Nafion Polymer Electrolyte and Experimental Validation, *Journal of the Electrochemical Society*, Vol. 154, p. B82–B87 (2007)
- (18) Werder, T., Walther, J. H., Jaffe, R. L., Halicioglu, T., Koumoutsakos, P.: On the Water- Carbon Interaction for Use in Molecular Dynamics Simulations of Graphite and Carbon Nanotubes, *J. Phys. Chem. B*, Vol. 107, p. 1345–1352 (2003)
- (19) Wang, C. Y.: Fundamental Models for Fuel Cell Engineering, *Chemical Reviews*, Vol. 104, p. 4727–4766 (2004)
- (20) Ferziger, J. H., Peric, M.: *Computational Methods for Fluid Dynamics*, Springer (1997)
- (21) Gasteiger, H. A., Kocha, S. S., Sompalli, B., Wagner, F. T.: Activity benchmarks and requirements for Pt-Pt-alloy, and non-Pt oxygen reduction catalysts for PEMFCs, *Applied Catalysis B: Environmental*, Vol. 56, p. 9–35 (2005)
- (22) Tsushima, S., Ikeda, T., Koido, T., Hirai, S.:

Investigation of Water Distribution in a Membrane in an Operating PEMFC by Environmental MRI, Journal of the Electrochemical Society, Vol. 157, Iss. 12, p. B1814-B1818 (2010)

- (23) Springer, T. E., Zawodzinski, T. A., Gottesfeld, S.: Polymer Electrolyte Fuel Cell Model, J. Electrochem. Soc., Vol. 138, No. 8, p. 2334-2342 (1991)

■ Author ■



Tetsuya KOIDO



Katsunori MAKINO



Yuki ITO



Nobuhiro KUSUMI



Mahito CHIBA

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Editorial Afterword

When the first Honda R&D Technical Review came out in 1989, Nobuhiko Kawamoto, then-President of Honda R&D Co. Ltd., described it in the foreword as a medium for “Proving Yourself as One of the Best.” The current issue will be the 39th, including the two F-1 special issues. During this time, we have published a total of 1 040 technical papers and introductions of new technologies. Browsing through them again, I see how these articles distilled the technical issues and trends of their times in each area, how they captured the ardent feeling and determination of the authors facing those issues, and how they summed up the many accomplishments arising from their passion and spirit of inquiry. Truly these comprise one of the foundations of Honda today.

Starting with this issue, Vol. 24 No.1, the Honda R&D Technical Review has reached a turning point by publishing as an electronic book. I hope first of all that this will make it a technology communication tool that all HG members can share with ease, and that in the near future engineers around the world can take part, too.

I hope that this medium will make us all more familiar with many valuable insights and facilitate technical exchanges with the outside world. I hope that you and your fellow members will use it effectively. Moreover, if you have a chance to write your own paper, please do not hesitate to step forward. On my part, I will endeavor to ensure that the Honda R&D Technical Review remains a cornerstone for building new value.

Tomoo Shiozaki

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