



# Honda R&D

## Technical Review

October 2011  
Vol.23 No.2



---

## Comment on the Cover

---

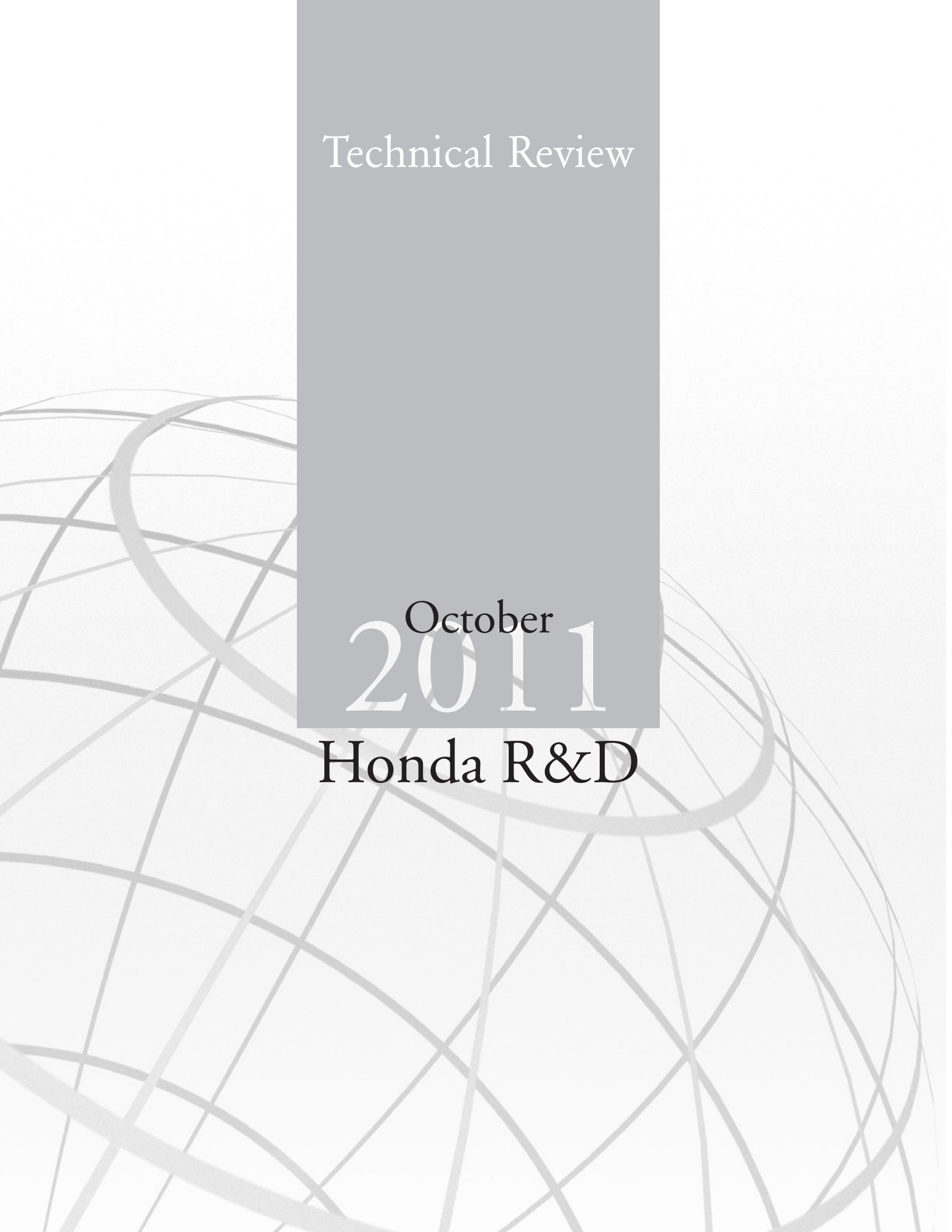
Toys are among the first ways that children encounter society and come to understand it. The excitement that a child feels playing with toys remains when he or she is grown.

This issue's cover features a toy box to evoke that sense of excitement.



Poewr Products R&D Center  
Fumiya Isono

---

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.

Technical Review

October  
2011

Honda R&D

---

# CONTENTS

## Introduction of new technologies

|                                                                                                                |                                                                                                                                                         |    |
|----------------------------------------------------------------------------------------------------------------|---------------------------------------------------------------------------------------------------------------------------------------------------------|----|
| Styling Design of CBR250R. ....                                                                                | Omi IIDA . . . . .                                                                                                                                      | 1  |
| Styling Design of BRIO . . . . .                                                                               | Akihiko OHHASHI•Koichi TANAKA<br>Tetsuo NARITA•Maki ONIZUKA . . . . .                                                                                   | 7  |
| Development of EV-neo, Electrical Motorcycle for Business Applications<br>.....                                | Toshiyuki CHO•Koichiro HONDA•Masahide MIMURA . . . . .                                                                                                  | 13 |
| Development of Midrange Outboard Motor BF60 with Equipment for User Comfort<br>.....                           | Mitsuharu TANAKA•Norikazu SHIMIZU•Hiroshi YAMAMOTO<br>Shigekazu SAKATA•Koji YASUDA•Masahiko YAMAGISHI . . . . .                                         | 22 |
| Development of Micro Combined Heat and Power Generation Unit with Extended Expansion Linkage Engine<br>.....   | Yoshiharu TAKITA•Shohei KONO•Hideaki TAKADA<br>Kunihiro KITSU•Atsumu NAOI•Koichi TSUNO . . . . .                                                        | 30 |
| 2011 Odyssey Fuel Economy Improvement. ....                                                                    | Ryan CHAPMAN•Paul DeHART•Patrick SHAFER<br>Spyridon BLATSEAS•Chase LIN•Chris HARTWIG•Dennis PLANK<br>Evan MARTIN•Scott PAUL•Brandon OLLANKETO . . . . . | 39 |
| Development of Intake Air Pressure Sensor-less Fuel Injection System for Small Motorcycles<br>.....            | Katsuhiro UTSUGI•Yuki TAKANO<br>Kazuhiko SAKAGUCHI•Kenta ONISHI . . . . .                                                                               | 49 |
| Development of Axial Motor . . . . .                                                                           | Shoei ABE•Junji INOUE•Satoshi ISHIKAWA . . . . .                                                                                                        | 60 |
| Operability Enhancement of Steering Wheel Switch for Audio Remote Control . . . . .                            | Masaaki ABE . . . . .                                                                                                                                   | 66 |
| Development of Nitrocarburized High-strength Crankshaft with No Rare Metals Added<br>.....                     | Yoshihiro TAKITANI•Isamu SAITO<br>Takahiro MIYAZAKI•Keiichiro KAMIYA . . . . .                                                                          | 70 |
| Visualization of Oil Behavior when Piston Pin Noise Occurs . . . . .                                           | Takashi KONDO•Hisayuki OHBAYASHI . . . . .                                                                                                              | 76 |
| Development of Noise-reducing Wheel . . . . .                                                                  | Youichi KAMIYAMA•Katsushi ISHII<br>Hisamitsu TAKAGI•Mikio KASHIWAI . . . . .                                                                            | 83 |
| Development of CAE Standardization System for Motorcycle Parts with Navigating Function for Designers<br>..... | Mijo ITO•Daisuke ISHIHARA•Miki TSUCHIYA<br>Masato OTSUKA•Kenji TSUCHIMOTO•Yasuhiro MIYAZAKI . . . . .                                                   | 90 |

**Technical papers**

|                                                                                                                                                                                               |                                                                              |
|-----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|------------------------------------------------------------------------------|
| Mechanical Structure and Performance Evaluation of High Differential Pressure Water Electrolysis Cell<br>..... Eiji HARYU•Koji NAKAZAWA•Kenji TARUYA<br>Hiroyuki ISHIKAWA•Masanori OKABE..... | 97                                                                           |
| Influence of Chassis Frictions on Vehicle Dynamic Behavior: Analytical Study .....                                                                                                            | Jinghong YU..... 106                                                         |
| Influence of Chassis Frictions on Vehicle Dynamic Behavior: Vehicle Experimental Investigation<br>.....                                                                                       | Jinghong YU•Edward HOSNI..... 114                                            |
| Integrated Active Noise Control System for Low-frequency Noise in Automobiles<br>.....                                                                                                        | Toshio INOUE•Akira TAKAHASHI<br>Kosuke SAKAMOTO•Yasunori KOBAYASHI..... 121  |
| Development of SRS Module Damping Structure for Vibration Enhancement and Weight Reduction<br>.....                                                                                           | Takashi KONDO•Masashi TERADA<br>Yukihiro KUNITAKE•Kunitomo MIYAHARA..... 129 |

# Styling Design of CBR250R

Omi IIDA\*

---

## ABSTRACT

The CBR250R was developed as a top-end model for the rapidly growing motorcycle markets of Southeast Asia and India and an entry-level model for the mature markets of Japan, North America and Europe. This development was undertaken under the banner of offering more customers one of motorcycling's attractions, namely the "joy of driving." The model has been launched in 22 countries as a motorcycle with a "sporty full-fairing exterior," the "joy of operating" and a "reasonable price."

With a model concept during development of "sports quarter for one world," design proceeded on several fronts, including concept and determination of specifications. The purpose was to develop a package that is lightweight, compact and easy to handle so that sporty cruising is enjoyable, and to do it all with globally common styling at a reasonable price.

Carrying on the styling theme of "mass concentrate proportion" that underlies Honda's sport motorcycles, designers from Japan, Europe and Asia discussed the styling that evolves to fit the times, and working together, they created such a styling.

---

## 1. Introduction

The CBR250R is a top-end model in Southeast Asia and India, where there is marked growth in the market for small-sized motorcycles in the 100 cm<sup>3</sup> – 150 cm<sup>3</sup> class. Its role is to develop new markets for users who want not only the "pleasure of driving" but also the "joy of ownership." A simultaneous role is to invigorate the more mature markets for motorcycles in Japan, North America and Europe as an entry-level sport model.

Thus, the issue in developing this model was to meet the market needs of vastly differing parts of the world and bring a single concept and a single styling into a global sport model that is a standard around the world.

This development was undertaken under the banner of offering more customers one of motorcycling's attractions, namely the "pleasure of driving." The development targets were to provide an expression of "sporty styling that gives the customer the joy of ownership," as well as to achieve a pleasure of operation that "anyone from beginner to experienced riders can enjoy" and a wide range of user friendliness that "accommodates everything from city cruising to sporty touring."

In the development of this model, it was assumed that the production would be done in Thailand where high quality yet low cost production for a 250 cm<sup>3</sup> sporty model could be achieved. Therefore, all departments concerned, such as development, marketing, production, procurement and service, collaborated on development from the

planning stage.

The CBR250R was launched in 22 countries in November 2010 and has been favorably received in them all (sales in Japan began in March 2011).

## 2. Concept

At the early stages of planning, Honda began a study with the grand purpose of developing markets for sport motorcycles of 250 cm<sup>3</sup>, the largest engine class in the markets of Southeast Asia and India.

Starting with the assumption that the target user was a sports-oriented male in his 20s or 30s seeking an upper-end motorcycle, the study members took the customer's point of view to answer the question, "What product values must this model have to appeal to this user?" As a result, they determined that in order to embody the development targets, the characteristics of this model would be a "full-fairing exterior" that is sporty and gives the customer the joy of ownership, a pleasure of operation that anyone from beginner to veteran riders can experience, and a "high-torque single-cylinder engine in a lightweight, compact body that is easy to handle" to ensure user-friendliness for everything from city cruising to sporty touring. An additional objective was to achieve the satisfaction for wide range of customers making it our top-end model for the part of Asian market while keeping easy handling as an entry-level model as well.

As for the price, although it would be situated in the

---

\* Motorcycle R&D Center

higher price range in the markets of Southeast Asia and India, Honda researched the purchasing power of the target users and set a proper price accordingly.

In the process of establishing the basic specifications, the marketability has been evaluated closely for the markets in other areas but Southeast Asia and India, such as Japan, U.S. and Europe where the sporty motorcycle category market has been already matured, to examine the product validity in the market as a full-faired sporty entry-model with attractions of reasonable price and high quality. This investigation had been concluded to find that the model had the potential for developing a market with many entry-level customers and the decision was made to start the development. Thus, development for the global market ultimately began with the model concept of “Sports quarter for one world.”

### 3. Package Design

The development team had a strong desire that “We want people to enjoy motorcycles with their sensations, regardless of their traffic environment or differences in physical sizes.” To realize this desire, we had set several design criteria that had something to do with functional performances. One of them was about the shape of cowlings that could accommodate a variety of riders ranging from Asian riders with smaller average height to Western riders with larger average height. The second criterion was about the maximum steering angle to get the minimum turning radius that satisfied good handling for light-weighted and compact body while considering good maneuverability suited for both city cruising and sporty driving. The third criterion was about riding posture which is to be upright posture to the vehicle (Figs. 1 and 2).

However, a steering angle that meets these functional requirements, in other words, one that ensures user-friendliness in the city, and the steering handle shape that provides an upright riding position, were expected to

significantly limit modeling freedom of critical elements that are the very essence of full-fairing styling, especially the low, momentarily-striking proportions of the upper cowl and the relaxed character of the middle cowl from front to back.

Therefore the team conducted the configuration studies with 3D virtual models to determine the basic dimensions (Figs. 3 through 5). This also made it possible to match the arrangement of auxiliary equipments and wiring around the

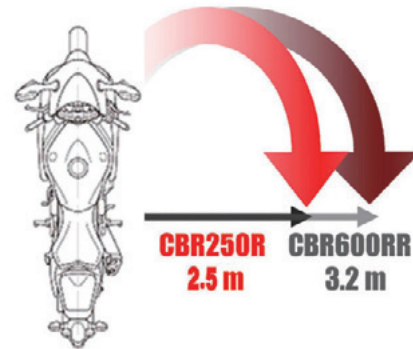


Fig. 2 Minimum turning radius

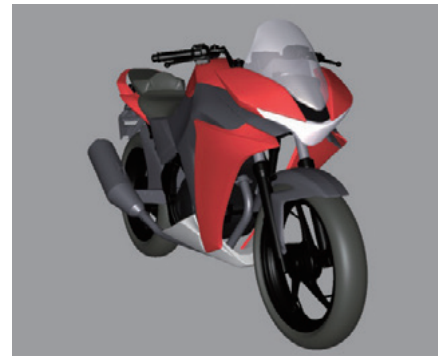


Fig. 3 3D model: Front quarter view



Fig. 4 3D model: Overhead view

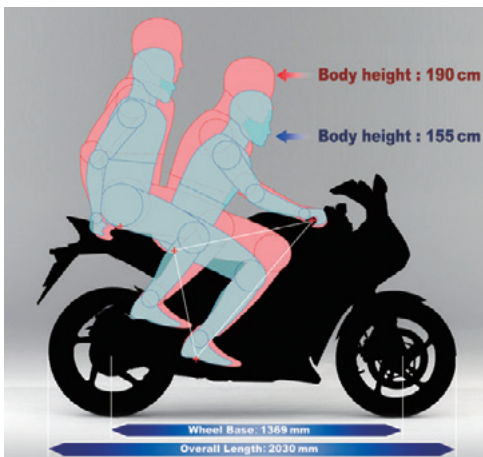


Fig. 1 Riding posture (body height: 155-190 cm)



Fig. 5 3D model: Gauges and switches

steering handle with the outlines that appeared in the design sketches at an early stage.

Such prior simulations enabled the team to find, in advance, places where modeling limitations existed, helping to identify the possible increase in the degree of freedom for examining styling.

#### 4. Styling Design

Design work for the CBR250R started with the goal of creating a benchmark for middle-class sport models, with basic proportions that provide a strong sense of Honda sport model identity together with modern styling.

First, “mass concentrate proportion,” which has been a basic theme of Honda’s sport category styling since the RC211V, an HRC racer that debuted in 2002, was chosen for the main keywords (Fig. 6). Next, the design techniques of “multiplex structure” and “sculptural surface,” which were used in the VFR1200F<sup>(1)</sup> and CBR1000RR developed in 2008, were chosen as sub-keywords, and then the styling examination began.

Specific design techniques referred to by these keywords are described below.

**Mass concentrate proportion:** A design technique that puts heavy parts like the engine, and the exterior parts that cover them, in the center of the layout and makes the overall proportions compact.

**Multiplex structure:** A design technique that enhances functionality and provides visual novelty by giving every functional components of cowling (air deflectors, air regulators and heat exhaustion components) an expression of shapes as separate components, and by overlapping them when building the overall outlines.

**Sculptural surface:** A technique with which models are created just like the ones by sculpting with boldly chiseled surfaces from a larger mass, emphasizing large modulation and deep clear cuts.

##### 4.1. Pursuit of the Styling Concept, and Sketch Collaboration among Designers from Research Labs in Various Countries

In developing the sketches for this model considering

the differing positioning in those markets, as noted above, the design team had two opinions: to proceed with a focus on the “high-class look and status of the VFR” as a top-end sport motorcycle model, or, instead, to focus on “the youngest brother of CBR series with the entry-level sporty image” as one of the super sport models (Figs. 7 and 8).

The size and external design of a motorcycle vary greatly depending on its purposes and how it is to be enjoyed, such as models intended for long-distance travel on a highway, and those for a driving in a circuit. The aforementioned VFR1200F was designed with a large cowling to provide traveling comfort to minimize fatigue for cruising long distances at high speed, as well as to bear the image of property of a matured adult.

The CBR1000RR, on the other hand, was designed to be lightweight and compact and express simplicity and a sense of condensing everything into the center of the body, primarily so individual riders could enjoy sport cruising.

In the current model, the team considered visual elements that satisfy customers’ desire to own a motorcycle (such as the high-class look and status that come from the coloring and a form that has modulation and a sense of presence, as in the VFR1200F) to be necessary elements for a top-end model in the Southeast Asia and India markets.

On the other hand, it was also to be a highly universal entry-level sport model as the youngest brother of the



Fig. 7 2010 VFR1200F



Fig. 8 2010 CBR1000RR

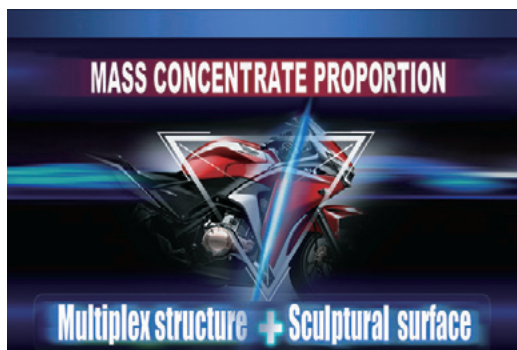


Fig. 6 Styling concept

CBR series, and therefore the proportions, which are basic to styling, were set according to the main keywords “mass concentrate proportion” (in other words, everything appears condensed into the center of the body), to preserve the simple and decisive styling of a super sport motorcycle.

This fuses two styling elements – “a form that has modulation and a sense of presence” and “simple and decisive styling” – that at first glance appear contradictory to each other, and finding the optimal solution to this was seen as the main modeling issue.

The design team, which brought together designers from research centers in Japan, Europe and Asia to solve the above issue, developed the sketches.

In response to the above issue, the designers’ differing cultural backgrounds enabled them to express their individual ideas in sketches, and allowed them to expand their scope of styling expression in trying to create something original while simultaneously inheriting the image of higher-level models.

The European proposal (Fig. 9) and the Japanese one (Fig. 10), which survived into the final round of considerations, each expressed the CBR Series’ sense of condensing everything into the center of the body, and at the same time divided the functional parts of the upper, middle and lower cowls into independent elements, giving an expression of visual novelty with the nimbleness of a sport bike. The process then went to the final sketch based on these two proposals.



Fig. 9 Concept sketch from Honda R&D Europe



Fig. 10 Concept sketch from Honda Motorcycle R&D Center Japan

## 4.2. Final Sketch Development

In the final sketch, there was an awareness of the high-class look and status that were lacking in the previously mentioned sketches. The surface’s modulation and stress, and the continuous transition between sharp, hard forms and soft forms, express tension and change in the overall form.

The surface that extends from the tank, with which the rider has direct physical contact, to the middle cowl comprises continuous surfaces that are organic and modulated; in contrast, the section from the front cowl to the middle cowl, which regulates the air stream, is made to appear bold with simple, large surfaces. A major characteristic in the design of this model was the respective use of these surfaces with different characters and their fusion into a single form.

Moreover, the undercowl (which covers the exhaust pipe and the engine that produces great heat) is continuously connected to the muffler cover. This helps to cool the engine area and forms a configuration of hard, mechanical surfaces, so that the area is expressed as if it were part of the engine components and the nimble overall look is enhanced.

The result is that the high-class look that gives the customer the joy of ownership and the nimbleness of a sport model have been packaged together in a single form (Figs. 11 and 12).



Fig. 11 Final sketch: side view



Fig. 12 Final sketch: front view and rear view

### 4.3. Functioning Parts

#### 4.3.1. Engine design

The 250 cm<sup>3</sup> water-cooled DOHC single-cylinder engine with next-generation environmental and power performance combines a simple, non-affected side view and modeling around compact but powerful cylinder heads.

Moreover, the black metallic paint emphasizes a bold exterior that projects functional beauty (Fig. 13).

#### 4.3.2. Meter design

The design of the meter and indicator area is sporty and functional. The analog tachometer is placed in the middle of the instrument panel, and when the ignition is turned on, its needle sweeps all the way to full. As the needle returns, the liquid crystal speed display counts down along with it so the rider can make sure there are no irregularities in the meters.

To give the area a functional and modern expression, silver garnish with geometrical and mechanical styling is placed around the liquid crystal display and indicator area and the blue liquid crystal backlight adds to the effect (Fig. 14).

#### 4.3.3. Headlight design

To express sportiness and a unique look, the main reflector was extended to be shared with the position lights and the lighting was arranged in a Y-shape, creating a feature of the front view.



Fig. 13 Engine sketch



Fig. 14 Meter view sketch

Also, the surface slanting from the screen is driven toward the back of the vehicle body like a person drawing his chin downward, avoiding visual weight around the front.

Moreover, the inner lenses that covers the position lights emphasizes the Y-shaped character of the overall lighting, expressing a mechanical and high-precision luxury feel (Figs. 15 and 16).



Fig. 15 Front view



Fig. 16 Front quarter view

## 5. Color Design

The configuration of the exterior components of the CBR250R was innovated to allow two color variations that project very different images, allowing the model to suit the color preferences of a wide range of areas. There are many models of motorcycles with characteristic paint patterns, so color is a very effective means of differentiation.

In the example of Fig. 17, though the overall form is a simple structure, with the use of a chic monotone color emphasizes the middle cowl's organic and modulated continuous surfaces that extend from the tank. On this basis, this is the model chosen to express a high-class look and status as the successor to the VFR image, targeted at Southeast Asia and India. Figure 18, on the other hand, shows the result of combining bright and sporty tricolor



Fig. 17 Monotone color



Fig. 18 Tricolor

paint with the sharp wing pattern stripe that directly inherits the image of the CBR Series. This expresses the simple and nimble sport model that is sought in Japan, North America and Europe.

## 6. Conclusion

During the development of the CBR250R, the following techniques were used to deal with the design issue of determining one styling that meets the needs of different areas with completely different markets, fusing a flagship sport motorcycle for Southeast Asia and India with an entry level sport model for Japan, North America and Europe.

- (1) For the package design, 3D virtual models and other techniques were used to achieve both an upright riding position and low, momentarily-striking full-fairing exterior, allowing riders to experience the pleasures of motorcycling regardless of differences in transportation environments and body types.
- (2) For the styling design, the team fused two styling elements – “a shape that has modulation and a sense of presence” and “simple and decisive styling” – that initially appear contradictory, and packaged the high-class look that gives the customer the joy of ownership with the nimbleness of a sport bike in a single form.
- (3) For the color design, the configuration of the exterior components was innovated to allow two variations – a monotone color that emphasizes a high-class look and

a sporty color that is sharp and nimble – to ensure the model would suit the color preferences of a wide range of areas.

Through these design developments, Honda achieved its original development goals, namely “a sporty full-fairing exterior that gives the customer the joy of ownership,” “a pleasure of operation that anyone from beginner to veteran riders can experience” and “a wide range of user-friendliness that accommodates everything from city cruising to sporty touring.”

## Reference

- (1) Kishi, T.: Styling Design of VFR1200F, Honda R&D Technical Review, October, 2010, P. 15-19

## ■ Author ■



Omi IIDA

# Styling Design of BRIO

Akihiko OHHASHI\*

Koichi TANAKA\*

Tetsuo NARITA\*

Maki ONIZUKA\*

---

## ABSTRACT

Wishing to build a new small car suited to the coming age, Honda started with the grand concept of a “Micro Metropolitan Jet” and developed the BRIO. The size of the Brio makes it easy to use in urban areas, makes it is more efficient, and offers a striking appearance and spacious interior.

The exterior design creates a presence with an advanced form based on the “Efficient Energetic Exterior” concept. The interior design concept of “Efficient Impressive Interior,” achieves a refreshing, high-quality ambiance. Furthermore, colors and fabrics were developed to better bring out those characteristics.

---

## 1. Introduction

To meet the growing need for cars in Asia, Honda launched the CITY in 1996 as an exclusive model for Asian markets. More than a decade later those Asian markets, and the user awareness in them, have changed so that people have more diverse needs from cars.

Honda began the current development process by taking another look back at our roots with hopes of build a new small car that is inexpensive, more suited for the coming age, and featuring strong environmental performance thereby offering more customers the joy of owning a car.

As part of the process, the development staff observed various Asian nations and performed thorough research on each from various angles such as their traffic conditions and user lifestyles. As a result they discovered three important themes: presence, a spacious interior, and good fuel economy. Following these themes, they set about to design a next-generation small car, with optimal packaging and quality as well as fuel efficiency, as a means of urban transportation.

Figure 1 shows the grand concept of the BRIO.

In order to achieve this goal they chose the platform of the JAZZ<sup>(1)</sup> which has a solid reputation as a compact car. In addition, the design uses a short body for easy turning while achieving a roomy interior and an outstanding presence thanks to putting ingenuity into every detail (Fig. 2).

Image not displayed for copyright reasons.

Fig. 1 Grand concept



Fig. 2 2011 Honda Brio

---

\* Automobile R&D Center

## 2. Package Layout

With entry users as the main target, the development staff sought a high-efficiency package layout that would both provide a short body with good turning performance (in consideration of urban traffic conditions) and a roomy interior.

Taking the JAZZ platform as the basis, the wheelbase of the BRIO was shortened to 2,345 mm (155 mm shorter than the JAZZ), which reduced the minimum turning radius. It was also given a shorter overall length of 3,610 mm (290 mm shorter than the JAZZ), to distinguish the BRIO from the JAZZ.

Moreover, to give the form presence in Asia where there are many space-oriented small cars, the developers were particular about developing a low and wide form. To ensure sufficient headroom, they set a height of 1,485 mm (40 mm shorter than the JAZZ) and a width of 1,680 mm. In India, to accommodate poor road conditions, the height off the ground was raised 15 mm to make the total height 1,500 mm (Fig. 3).

One major issue was how to provide a spacious interior within a compact body size. The interior space was established not to maximize the interior overall, but to “emphasize roominess in places where people can actually sense it.”

The market research indicated that people in India have long legs relative to their total height. Therefore, the knee clearance of the front passenger seat was increased (60 mm more than the JAZZ) by making the instrument panel thinner, and back seat knee space was increased by narrowing the front seat back (Fig. 4).

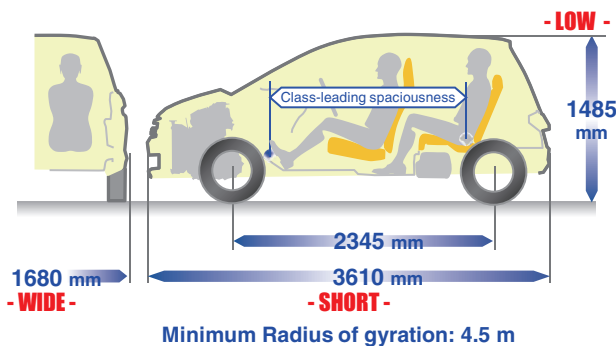


Fig. 3 Package layout

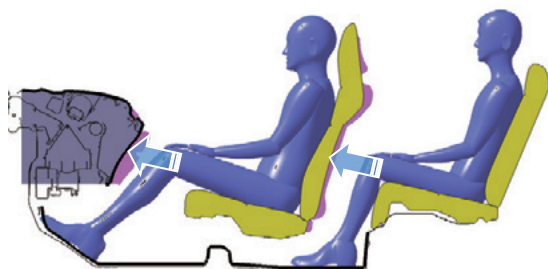


Fig. 4 Interior space

Additionally, comments from women who wear the traditional sari indicated a need for a wide back door opening to improve the back seat comfort. This was achieved by moving the center pillar forward without making it harder to access the front seat. In addition to the easy turning afforded by a short body, good visibility is ensured to increase driver confidence. By making the front pillars narrower in cross-section and lowering the bottom of the side windows, visibility to the front diagonals and farther down on the sides was expanded making it easier to make turns and drive on narrow roads (Fig. 5). In addition, a large glass hatch was used to ensure good visibility to the rear.

The luggage compartment provides sufficient capacity for daily needs. As a module, it can hold one large suitcase (750 × 540 × 280 mm). With the seat down it can accommodate a folding bicycle (Fig. 6). In addition, the glass hatch is lightweight and easy to open and close which makes it easier to load cargo.

By incorporating the above ideas the development staff was able to build an interior more spacious than one might imagine from the outside, given the low, wide and short body appearance.

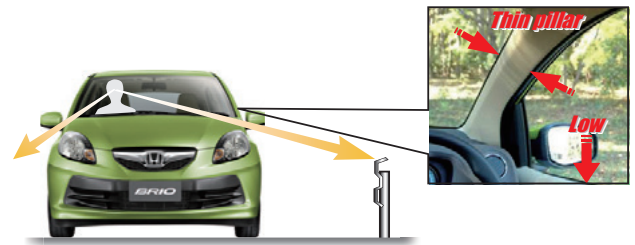


Fig. 5 Good visibility



Fig. 6 Luggage compartment

## 3. Exterior Design

The BRIO's exterior design concept is “Efficient Energetic Exterior” emphasizing its presence with an advanced form (Fig. 7). Figure 8 shows the concept sketch of the exterior design.

Under the theme of “Double Triangle Form” (Fig. 9) the design features a low and wide framework and projecting fenders. Within this, the design consists of two triangles representative of dynamic and sporty characteristics that are bounded by the side's character lines. The exterior design theme is apparent in the side view.

The “Dynamic Triangle” is a forceful mass that appears to sweep toward the nose, while the “Sporty Triangle” mass appears to be jumping back toward the rear fender. These two masses, piled one atop the other, seem to be sliding past each other toward the front and rear creating a pulsing side view suggesting a brisk and energetic drive. The surface structure joining the two triangles also projects an organic muscularity and increases presence.

Figure 9 shows the final rendering of the exterior design.

For the front view, a wide design imparting a sense of cruising stability was sought. To accomplish this, the fender was made to project from the short but sturdy, thick and strong front mask to bring out the front view’s dynamic character.

The headlights have a wide, cylindrical design presenting a bright and youthful expression. The front bumper has a wide opening better emphasizing the wide look and the stable stance. The front grill, additionally, has the feel of high quality and presence made prominent by using an accentuated surface structure suited to the body, and finished with high-quality chrome plating.

The rear view emphasizes the wide look by placing the round red lenses of the tail lights as far to the outside as possible. The form of the outside of the outer lens has

been given an edge to optimize the air flow and increase aerodynamic performance. The glass area of the glass hatch was maximized to take advantage of its ease of use from its light handling and its excellent visibility. On top of that, shaping the rear so that it looks like cut-off masses of material creates a rear view that is wide and evokes an advanced feel and an image of excellent aerodynamic performance (Fig. 10).

The door mirror is long from front to back making it look advanced. The surface is shaped to have a sharp edge which unifies it with the body giving a look of sharpness and speed. The design was improved through repeated wind tunnel testing ensuring that the door mirror both contributes to aerodynamic performance and adds product value to the exterior (Fig. 11).

Figure 12 shows the emblem design.

The character line uses a bold style with appealing motion as expressed by the arrows. Surfacing is by chrome plating and the accentuated surface structure, with both positive and negative faces, unites with the body and catches light to create a sparkle.

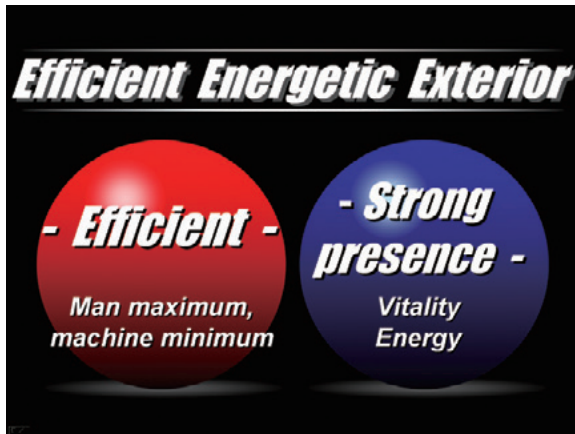


Fig. 7 Exterior design concept



Fig. 9 Final rendering

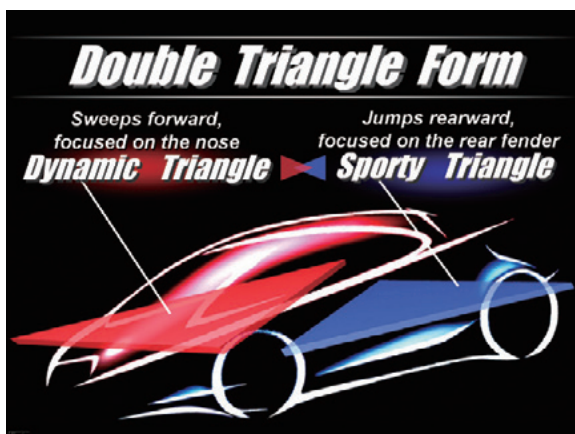


Fig. 8 Exterior design concept sketch



Fig. 10 Tail light

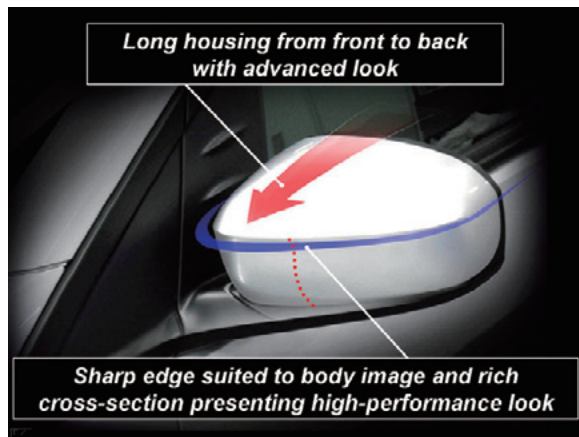


Fig. 11 Door mirror

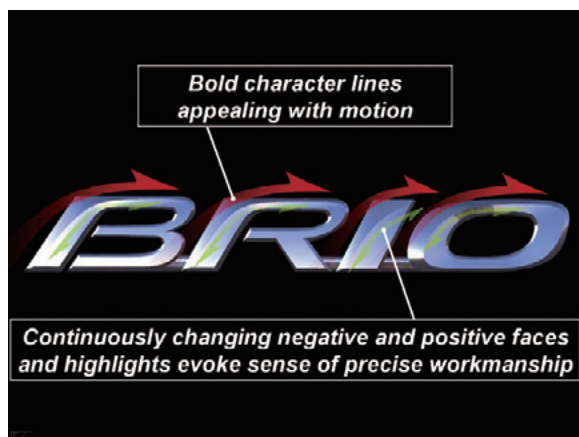


Fig. 12 Emblem design

#### 4. Interior Design

The interior design was developed under the concept “Efficient Impressive Interior” and seeks to provide comfort and high satisfaction to those within. The attractiveness of a package seeking to create spaciousness within a compact body was maximized. Wasteful parts were removed and functional value was condensed into the cockpit area (Fig. 13).

The development staff worked with the designers to simplify the structural materials inside the instrument panel and reduce the number of components and lighten the BRIO. The aim was to achieve more spaciousness than the eye could see by eliminating as many things as possible without losing functionality like user-friendliness and storage. This was in response to the dissatisfaction of many Indian customers with long legs toward the lack of knee room in earlier generations of small cars built by other companies that had based their designs on Japanese kei cars.

Customers desire the joy of car ownership. Therefore, the functional value was condensed into the cockpit area of BRIO so that customers could experience for

themselves the great satisfaction that comes from driving a car (Fig. 14).

The independent three-eye meters emphasize three-dimensionality with ring illumination both day and night, and a 2 DIN sized USB audio system exclusive to the BRIO is provided. The piano black finishing of the audio system imparts an advanced feel with high quality and a feeling of integration with the instrument panel (Fig. 15).

Front seats have a sporty design. Their headrests provide firm support and are an integral part of the seats, while the seat backs are thinner to ensure knee clearance for rear seat passengers. Thick cushions in both front and rear seats provide a high quality, comfortable ride (Fig. 16).



Fig. 13 Interior design concept



Fig. 14 Final rendering



Fig. 15 Drivers cockpit



Fig. 16 Seat design

## 5. Color and Fabric Design

### 5.1. Exterior

Character colors were chosen to express a proud presence that would not be lost amongst other cars even though the BRIO body is small. Taking “Energetic Vivid Color” as the theme, high-impact, high-chroma colors were chosen with crisply sparkling highlights and vivid brilliance. Two character colors were based on the destination market (Fig. 17).

The focus was on a highly chromatic expression that asserts presence even with a compact body. Since metallic colors generally have low chroma in shaded areas, a paint was developed that can retain high chroma even in shaded areas.

Fresh Lime Metallic, a color for the Thai market, is a thoroughly transparent lime green that evokes the image of fresh green leaves. Rich highlights are expressed with high-luminance aluminum flakes. Organic yellow pigment and titanium oxide express a transparent brilliance and accumulation of color in the shade characterized by a color shift from yellow in the highlights to green in the shade.

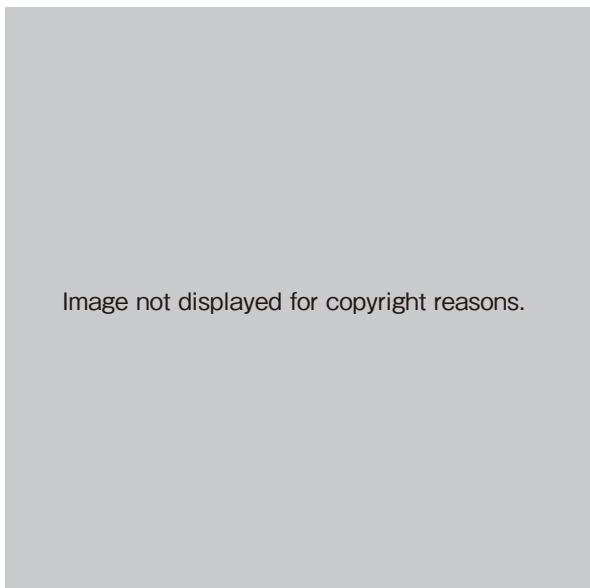


Fig. 17 Exterior character color

Energetic Blue Metallic, the color for the Indian market, is a high-chroma blue evocative of the clear, blue sky. The combination of high-luminance aluminum and super-high-chroma blue pigment makes it possible to have both highlight brightness and as much chroma as with solid colors. The color is characterized by refreshing silver highlights in a vivid blue.

The wide color lineup includes popular existing colors such as white, silver and black as well as colors suited to the tastes of the destination market (Fig. 18).

### 5.2. Interior

The interior color expresses the kind of quality the owner can take pride in, as well as excellent user-friendliness.

With “Efficient, Proud Coordinate” as the theme, it uses fine quality beige as a base and coordinates in a refreshing and rich way. The instrument panel has a two-tone color scheme of black and beige, and the points are elegantly decorated with a combination of brown coloring and vertical crepe surface texturing (Fig. 19).

A sporty check pattern resembling carbon cloth is used for the seat surface. Depending on the destination market and grade of the car, a monotone type and sportier two-tone type are available (Fig. 20). The most efficient method of weaving was studied and a simple pattern was selected that can be woven quickly with a dobby weaving machine. To reduce dyeing processes, thread of the cloth color was used effectively in some parts, making it possible to lower costs



Fig. 18 Exterior color lineup



Fig. 19 Interior color coordinate

significantly while producing a sporty and durable high quality surface material.

The fabric also has a smooth and pleasant feel, and excellent usability, being easy to clean of dust and pet hair (Fig. 21).

In India, where many women wear sari, this fabric makes it easier for them to get in and out of the car since sari are less likely to catch on the seat surface.



Fig. 20 Seat fabric (1)

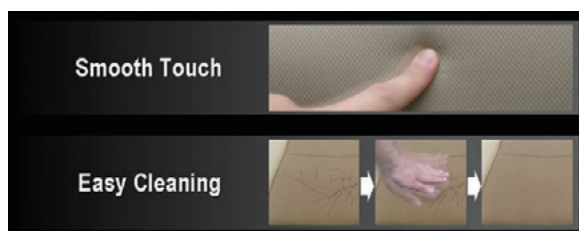


Fig. 21 Seat fabric (2)

## 6. Conclusion

The BRIO was developed under the concepts of “Efficient Energetic Exterior” and “Efficient Impressive Interior.” As a result, value that provides a sense of presence and advanced feel was condensed into the body to satisfy the needs of the owner.

Moreover, the development staff was particular about the aerodynamics of the entire body while enhancing fuel economy, thus they completed a small car with outstanding ease of use in urban areas and excellent drivability.

The name BRIO means “robust” or “vigorous” in Italian, and it is the hope of the development staff that the BRIO will be popular with many people thereby expanding the joy and vigor of car ownership.

## Reference

- (1) <http://world.honda.com/Fit-Jazz/>

## ■ Author ■



Akihiko OHHASHI



Koichi TANAKA



Tetsuo NARITA



Maki ONIZUKA

# Development of EV-neo, Electrical Motorcycle for Business Applications

Toshiyuki CHO\*

Koichiro HONDA\*

Masahide MIMURA\*

---

## ABSTRACT

The acceleration of attempts to reduce CO<sub>2</sub> emissions in recent years in order to deal with global environmental issues has heightened expectations for the electric vehicle (EV). In general, however, an EV needs to be loaded with a larger amount of battery to realize sufficient range. This ends up making the EV heavier and more costly. Taking these circumstances into consideration, EV-neo was developed as an electric motorcycle that satisfied the requirements for business applications while keeping the amount of loaded battery minimum.

A study of usability in business applications was first conducted. Classification of travel and operation patterns as well as objectives for performance were set.

In order to achieve the performance objectives for business applications, the EV was designed to be light in weight and compact by keeping the on-board batteries to the minimum necessary and having a power unit with an integrated configuration. The electric motor was made to be at least 75% efficient throughout the practical operating range. The vehicle was given the required driving performance defined as 12-degree hill-climbing departure performance when carrying one person and a load of 30 kg, and it also realized a range of 34 km (traveling at 30 km/h on level ground).

The package configuration of lithium-ion batteries was designed to realize both of battery cooling performance and waterproof performance to stand up to actual use. A battery monitoring system was also included for control of charging and discharging. These features enable use of the vehicle in ambient temperature requirement range from -10°C to 35°C for business applications.

Furthermore, the quick charger capable of charging a battery from 0% to 80% of capacity in 20 minutes, in addition to a standard charger compact enough to be stored in the utility box, was developed to support various business users.

---

## 1. Introduction

Recent years have seen measures taken in a variety of fields to mitigate environmental impacts. There are heightened expectations for electrically powered motorcycles, as well, because they generate no CO<sub>2</sub> emission when they are in operation. A current issue, however, is that the vehicle has to have large capacity of battery to realize an adequate range, making the motorcycle heavy and expensive as a result.

In light of this issue, an investigation was carried out to determine the system configuration that minimized the onboard battery capacity while considering the most appropriate application patterns for the system. As a result, it was found that there would be potential in business applications such as delivery operations in which range per single trip is relatively limited and charging time is expected to be spared.

This paper introduces the technology adopted in EV-

neo, the electric motorcycle, which was developed based on these investigation results. Figure 1 shows an external view of this motorcycle.



Fig. 1 External view of EV-neo

---

\* Motorcycle R&D Center

## 2. Overview of Business Applications

### 2.1. Classification of Business Applications

Studies were conducted on how vehicles are used in business applications by businesses such as pizza delivery, newspaper delivery, and other product delivery services currently using motorcycles. This included studies on traveled distance on a single delivery, stand-by time spent between deliveries, and so on. On the basis of those studies, applications were classified, in terms of travel patterns and operation patterns, and that classification had been concluded as that of two types shown in Fig. 2.

The first travel pattern, type one, is the short round trip-type of delivery services. This is the travel pattern typical of pizza and other delivery services from shops or business stations to nearby destinations. It involves mainly short-distance travels. A distinctive feature of this travel pattern is that the vehicle returns to a shop or a business station after each delivery. This operation pattern has the distinguishing characteristic of allowing time for recharging between one delivery and the next.

The second travel pattern, type two is the sequential multi-destination type of delivery services represented by, for example, newspaper delivery services, in which a vehicle returns to a shop or a business station after delivering things to a number of predetermined delivery destinations, and the total traveling distance can be preplanned. This operation pattern has the distinguishing characteristic of providing an interval of four or more hours ahead of the next operation after a previous delivery operation.

Both of these types are methods of use in which a shop or a business station can be considered the base, so that the place where the vehicles are charged is specified<sup>(1)</sup>.

### 2.2. Performance Objectives of Business Applications

The following performance objectives were defined for an electric motorcycle with business applications based on

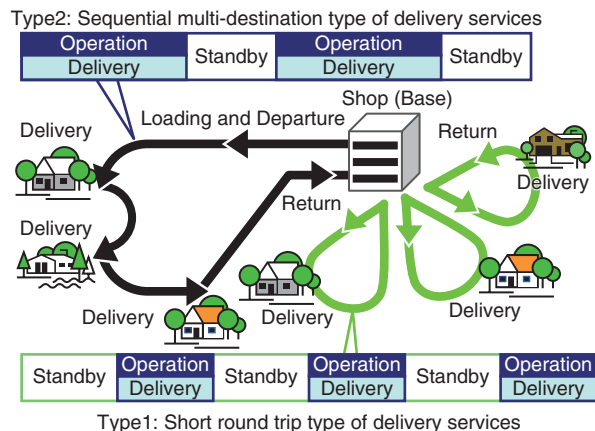


Fig. 2 Classification by operation patterns and travel cycles

the data collected on travel modes and the results of surveys on usage.

#### (1) Range

It is highly important to identify the necessary range of travel when the system configuration with a minimum amount of loaded battery is under consideration. Figures 3 and 4 show the distributions of travel distances derived from the data on the distances traveled in a single delivery by two types of users aforementioned.

Based on the distribution data of traveled range of those two application types, for either type, more than a half of the users, who use the vehicle for business applications targeted in the present development, can be satisfied if the vehicle meets the travel range of 15 km or more for each charging.

The sequential multi-destination type of delivery services involves frequent departing and stopping and travels over short ranges. Power consumption rates were

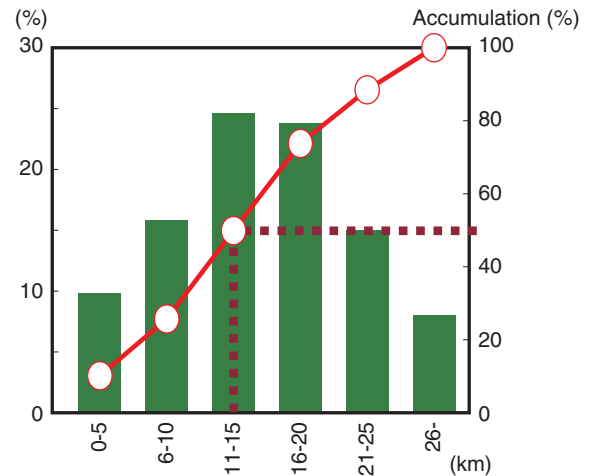


Fig. 3 Distance traveled per one operation in type 1

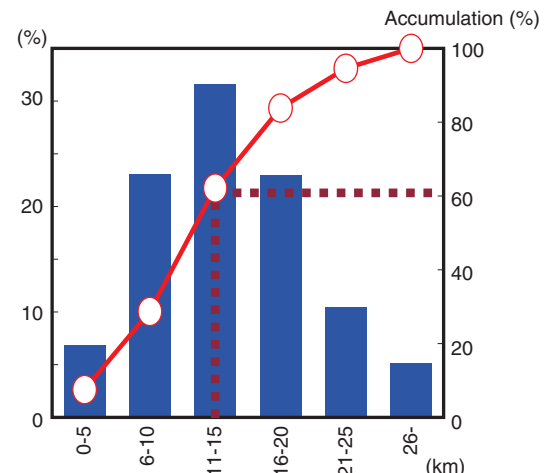


Fig. 4 Distance traveled per one operation in type 2

calculated from the travel data for users of this type, and it was found that they are approximately twice as high as the power consumption rate at constant speed of 30 km/h. Accordingly, the range objective was set as 30 km or more at constant speed of 30 km/h.

#### (2) Driving Performance

It was found, from the travel data of users aforementioned, that the departure capability on slope is important, in terms of usability, as departing and stopping occur frequently in business applications. Therefore, in the present development, the hill-climbing departure performance of 12 degrees, with maximum loading of 30 kg within the regulation for class 1 motorcycle in Japan, was set as one of the requirements for driving performances, which is the same performance level as conventional gasoline powered motorcycles.

#### (3) Environmental Durability

The vehicle requires environmental durability such that it can cope with weather and temperature differences between morning and evening or summer and winter. The objective was set as usability in ambient temperatures from -10°C to 35°C, and the ability to be driven through puddles and in the rain.

#### (4) Charging Time

Based on the operation patterns aforementioned, for the short roundtrip type of applications, it was considered that the scope of application could be expanded, beyond the application areas normally expected from the travel range for a single charging, if supplemental charging could be done in the intervals between deliveries. Accordingly, the charging time requirement of 20 minutes, for charging from 0% to 80% of the battery capacity, was set to make it possible to be charged when the vehicle is standby status between deliveries.

### 3. Overview of an Electric Motorcycle for Business Applications

#### 3.1. Major Specifications

Table 1 shows the major specifications of the EV-neo. The dimensions (length × width × height) are 1.830 × 0.695 × 1.605 m and the vehicle weight is 106 kg, equivalent to those specifications for conventional motorcycles with gasoline engines. The nominal voltage of the battery was set to 72 V and the nominal capacity to 12.6 Ah. The rated power of the motor was set to 0.58 kW and the maximum power to 2.8 kW.

#### 3.2. Layout of the Completed Vehicle

The EV-neo development concept was “a tough and competent business EV.” Ease of use as a business motorcycle was pursued not only in range and driving performances, but also in the system configuration of the vehicle. Figure 5 shows the installation arrangement of the main components.

The vehicle was designed so as to have the center of

Table 1 Specifications of EV-neo

|                                |                  |                       |
|--------------------------------|------------------|-----------------------|
| Dimensions L x W x H (m)       |                  | 1.830 × 0.695 × 1.065 |
| Wheelbase (m)                  |                  | 1.25                  |
| Ground clearance (m)           |                  | 0.120                 |
| Seat height (m)                |                  | 0.756                 |
| Rear deck height (m)           |                  | 0.630                 |
| Vehicle curb mass (kg)         |                  | 106                   |
| Rated power (kW)               |                  | 0.58                  |
| Maximum power (kW/rpm)         |                  | 2.8/5000              |
| Maximum torque (Nm/rpm)        |                  | 11/2000               |
| Cruising range at 30 km/h (km) |                  | 34                    |
| Main battery                   | Type             | Lithium ion           |
|                                | Battery (V)/(Ah) | 72 V/12.6 Ah          |
| Tire size                      | Front            | 90/90-12 44J          |
|                                | Rear             | 100/80-12 56J         |

gravity lowered by carrying the battery pack, which is the heaviest component, underneath the floor to maintain the stability of driving even when the vehicle was loaded. Consequently, the height of the center of gravity when the vehicle is loaded has become approximately 6% lower than that of gasoline engine motorcycles with an equivalent size.

In addition to that, the enhancement of energy efficiency and the compact design were achieved with the integrated drive train unit, located behind the battery pack, in which the power drive unit (PDU), the motor, the automatic centrifugal clutch and the reduction gears were incorporated into one. The charger was not designed as an onboard type for weight reduction of the vehicle considering the circumstances of business applications where the charging would take place only within the designated places. To cope with the differences in the requirement for charging devices among users of various application patterns, two types

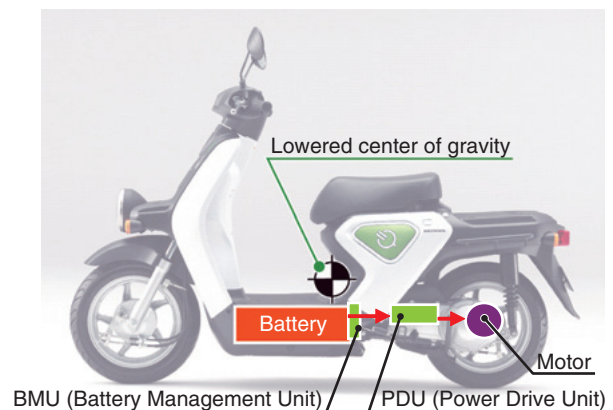


Fig. 5 Layout of components

of chargers as shown in Fig. 6, standard type and quick charging type, were developed allowing the users to select either or both of those types, of any numbers, depending on the vehicle operation pattern, the needs for sharing one charger with multiple vehicles or the needs for combined use of those two types.

The charging connector, which serves as the interface when charging the battery, was placed on the left side of the vehicle body where users are more likely to stand when getting on and off the motorcycle. A compact connector suited to the size of the motorcycle body was developed. For security and prevention of tampering, a lock mechanism was installed on the lid of the connector receptacle, and the lock manipulation knob was located inside the luggage box, which was designed to be locked by a key. Figure 7 shows



Fig. 6 External view of chargers (standard type and quick type)



Fig. 7 Charging connector and lock of charging receptacle lid

the connector configuration.

The meter unit shown in Fig. 8 was positioned in the center of the steering handle on the body. On the meter unit, together with a digital speedometer, the battery charge indicator, which consisted of five display segments, was installed to linearly display the remainder of the battery charge. When the battery remainder goes down to 10%, one of the segments starts to blink with a buzz, inducing awareness of the user with visual and auditory stimuli. The user is also notified when restricted power (to be explained later) is applied by illumination of the restricted power status indicator located inside the meter unit.

### 3.3. System Overview

Figure 9 shows the block diagram of the system. The basic operations are designed as same as gasoline-fueled motorcycles so as to avoid confusions by operators who are accustomed to use conventional motorcycles. When

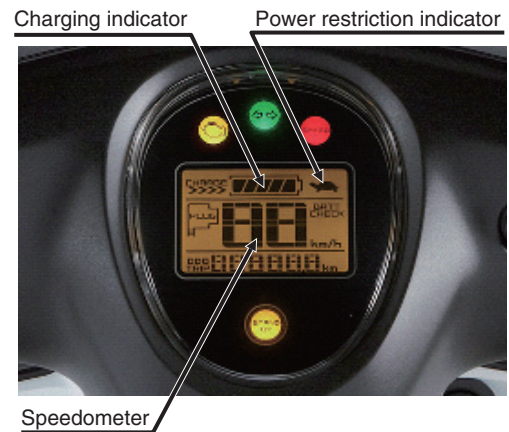


Fig. 8 External view of meter

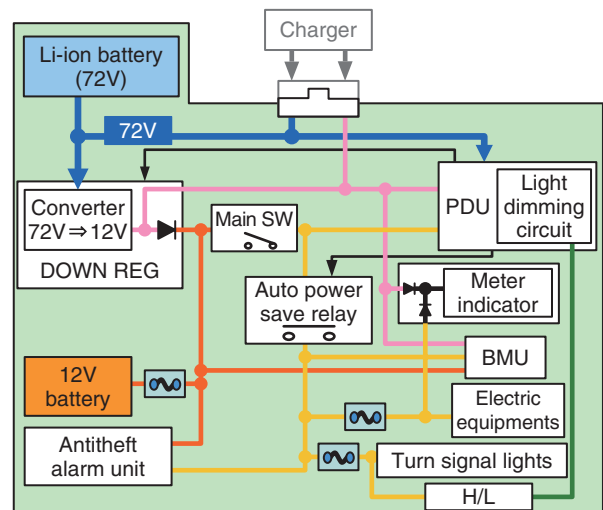


Fig. 9 System diagram

the main switch is turned on, PDU is supplied with the voltage from the sub-battery followed by activation of the whole system. A signal from the PDU activates the down-regulator, which supplies electrical components with 12 V system power. This makes it possible to share electrical components with conventional gasoline powered motorcycles for lighting devices, meters and other such devices. Rider's intention delivered through the throttle grip operations is converted into voltage signals by the accelerator position sensor (hereafter APS) and transferred to the PDU. The battery management unit (hereafter BMU), which is integrated into the battery pack, transfers the battery information such as voltage, current, temperature, etc. to the PDU via Controller Area Network (CAN). The PDU calculates an optimum output to drive the motor based on the APS signal, the motor speed and the battery information. The PDU dims lighting devices by Pulse Width Modulation (PWM) when a vehicle is at a stop to efficiently use the limited power. Furthermore, when the vehicle is in standstill for more than 3 minutes, the lighting system is automatically shut down by turning off the output signal to the automatic power save relay.

#### 4. Integrated Drive Unit

The drive train unit was designed with the primary goal of converting energy without waste from the energy in battery into the energy for propulsion.

In order to minimize the electrical power loss, parts were positioned to have the high-voltage wiring to a minimum length. The battery, PDU, and motor were wired together at the shortest distance, and the total length of high-voltage line from the battery to the motor is 895 mm.

##### 4.1. Motor

Figure 10 shows the integrated drive unit mechanism. The motor employs a three-phase brushless system with eight poles on the rotor and 12 slots on the stator. It is built with an interior permanent magnet (IPM) that uses rare earth magnets. The use of a high-speed motor with reduction gears enables generation of the rear wheel torque of 150 Nm, which is required for business applications, with

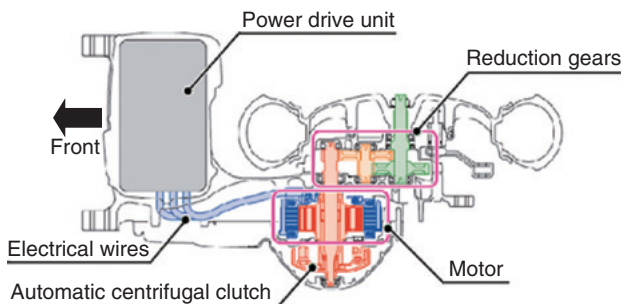


Fig. 10 Layout of drive train unit

a compact size of 149 mm in outside diameter and 35 mm in cumulative thickness.

##### 4.2. Automatic Centrifugal Clutch

The following two benefits have been obtained by arranging the automatic centrifugal clutch on the same axis as the motor.

###### (1) Reduction of vehicle handling effort

For motorcycle users it is very common, apart from driving astride on a vehicle, to move the vehicle around manually. This maneuverability is a crucial matter for business applications as well. By installing an automatic centrifugal clutch, the motor spindle is left apart without rotational movement when the vehicle is manually moved and the necessary power for maneuvering has been reduced to the level equal to or smaller than the one for a gasoline-fueled scooter.

###### (2) Enhanced efficiency

Efficiency map of the motor within the drive train unit is shown in Fig. 11. The efficiency of a motor ordinarily goes down in the low revolution speed range with low torque, but this range has been successfully avoided in vehicle operations by installing the automatic centrifugal clutch and by designing the stall revolution speed at 1900 r/min. Together with the motor structure aforementioned, the maximum efficiency of 92.7% and the efficiency above 75% for the whole conventional operation range have been achieved.

##### 4.3. Motor Control

The PDU decides the torque instruction value based on the APS signal and the motor speed, and calculates the duty and the advance angle of PWM control to get the highest efficiency. The advance angle was set to utilize

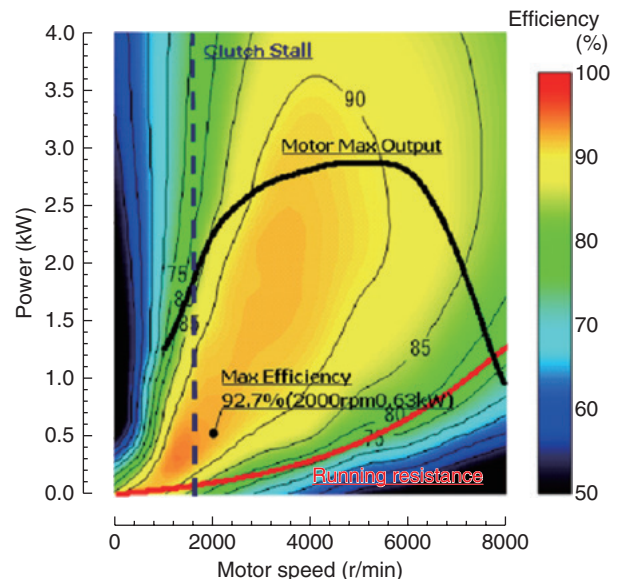


Fig. 11 Power and efficiency of motor

the benefit of weak field control and wide rotational range characteristic of the motor was realized to cover from 0 to 9000 r/min. Consequently, the gear transmission has been made unnecessary and the efficiency has been enhanced by eliminating the mechanical loss derived from the transmission. Figure 12 shows the transmission efficiency of a scooter of the same class<sup>(2)</sup>.

Eliminating the transmission also allowed the PDU to be placed where the crank drive pulley is located on a gasoline engine scooter. The space that would have been occupied by the cylinder head and air intake system became available for stowing the standard type of battery charger, which will be explained later. Figure 13 shows a comparison of the contours of the EV-neo drive unit and the drive unit of Honda Today, which is one of the Japanese class 1 scooters.

Energy regeneration control is activated responding to the rotation speed of the motor, when the throttle grip angle is small enough to retrieve energy back to the battery, while generating an equivalent level of braking force as the engine brake in conventional gasoline-fueled motorcycles, resulting in the contribution for longer range.

With these configurations in the integrated power train unit, the targeted power unit performances for business applications, which are hill-climbing departure capability for 12 degree slope loaded with a rider plus 30 kg load and 34 km range for a single full charging, at 30 km/h constant speed, have been achieved.

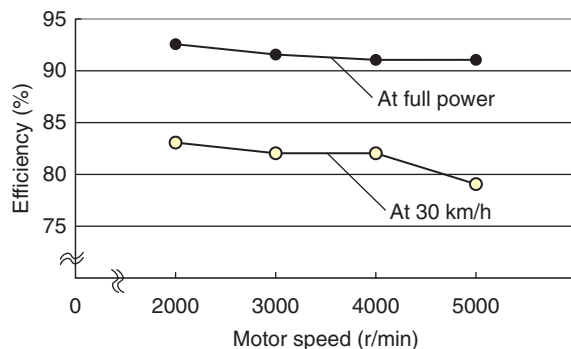


Fig. 12 Belt torque converter efficiency

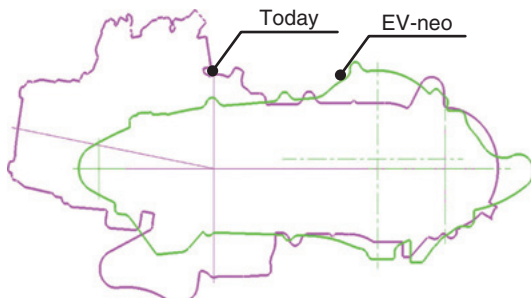


Fig. 13 Comparison of contours

## 5. Battery

### 5.1. Selection of Battery

To satisfy the target performances in the business applications, a lithium ion battery made by Toshiba, SCiB™ has been selected because the deterioration in discharging capacity at low temperature is small and the charging current acceptance is large. The unit cells, with the nominal voltage of 2.4 V and the nominal capacity of 4.2 Ah each, are connected in parallel by 3, and by connecting those groups of cells in series by 10 the unit module is formed. The battery pack is made of 3 modules connected in series and has the nominal voltage of 72 V and the nominal capacity of 12.6 Ah.

### 5.2. Battery Pack Structure

Figure 14 shows the structure of the battery pack. BMU located within the battery pack monitors the battery status information, the voltage and the temperature, constantly collected by Voltage/Thermal Monitoring Board (VTM board), which is incorporated into the battery module. The battery pack is equipped, in its upper portion, with the cooling fan and the contactor unit, which turns off the high voltage circuit when the main switch is off. With these structures, the specifications required for business applications, shown below, have been achieved.

#### (1) Battery cooling performance

When a lithium ion battery is exposed to a high temperature, the battery capacity goes down. To satisfy enough battery life for business applications, the cooling system with the structure shown in Fig. 15 has been developed. The cooling fan operates when the temperature of the battery exceeds 42°C, and the air is blown into the battery pack through the filter that prevents the invasions of dust, for example, from the intake vent, which is installed under the steering handle. The air goes through the gaps between the cells located so as to have clearances among them and is discharged out of the battery pack. With this cooling system, it has been achieved that the maximum temperature of the battery is suppressed under 55°C even

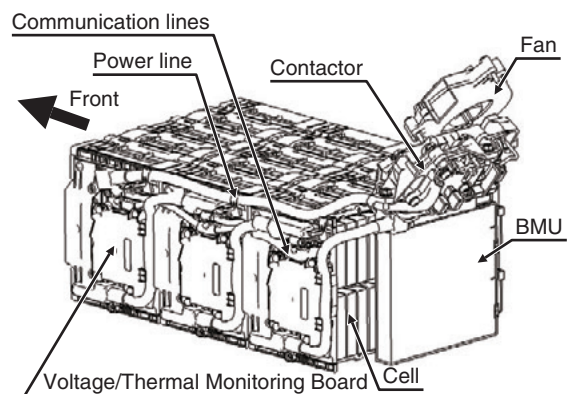


Fig. 14 Battery pack

at the condition of continuous hill climbing in the ambient temperature of 35°C.

### (2) Environmental durability

The battery package is structurally sealed and the intake vent and the outflow opening are located where they do not get drenched directly by water. With this measure, it has been realized to have sufficient environmental durability performances against running in puddles and driving in rains.

### (3) Battery output restriction control

It is characteristic of a lithium-ion battery that the internal resistance increases when the battery is at a low temperature, resulting in a decline in output voltage according to the discharge current. This can cause the vehicle to stop operating even though there is battery capacity still remaining. The present system implements battery output restriction control as shown below (MAP1–MAP4 in Fig. 16) so that the battery capacity can be used effectively.

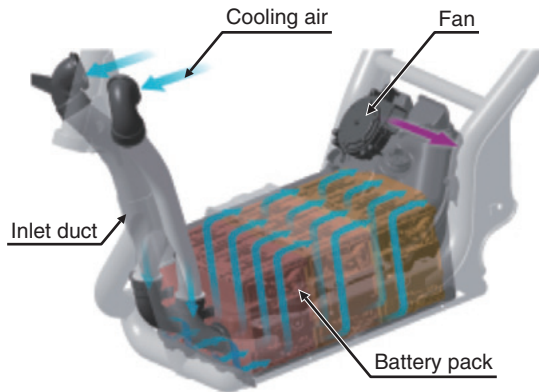


Fig. 15 Battery cooling system

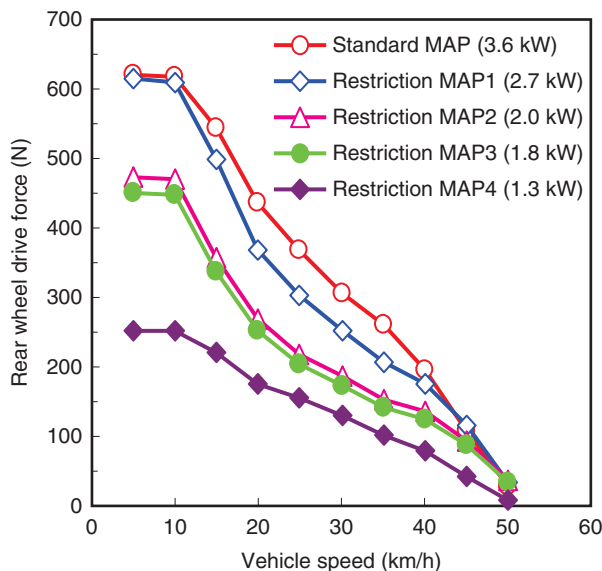


Fig. 16 Characteristics of output power

- After receiving voltage information of all cells from the BMU, the PDU is switched to use the output restriction MAP1 when the lowest voltage becomes 1.95 V or less.
- While driving using the output restriction MAP1, if the lowest voltage goes down to 1.95 V or less, it makes the next transition to the output restriction MAP2 which is one step down to the MAP1.
- There are 4 patterns of output restriction MAPs, the transition is made to the next lower MAPs in sequence.
- When the vehicle stops, it is reset to use the unrestricted MAP because the internal resistance goes down with the rise of battery temperature, which is increased by the heat generation while driving.
- However, the shift of output restriction status to MAP4, which is to be the most restricted output condition, has been designed to happen only when the lowest voltage goes down to 1.90 V or less. This status is considered as the situation in which the State of Charge (SOC) is close to 0% and the current output is terminated when the integrated current of the output exceeds a predetermined value without recovering to the upper levels of MAPs.

Applying those measures to the battery pack structure and the output restriction controls, it has been made possible to satisfy the target specifications for business applications, which are the operation temperature range from -10°C to 35°C and the environmental durability against driving in puddles and rain.

## 6. Charging System

Figure 17 is a schematic drawing of the charging system and the battery remainder management system. For charging, the AC plug on the charger is connected to a commercial AC power source and the output connector from the charger is plugged into the receptacle on the vehicle located on its side. Battery information that is monitored by the BMU is sent to the PDU through CAN communication. The PDU controls the charger and sends signals to the battery charge indicator located in the meter.

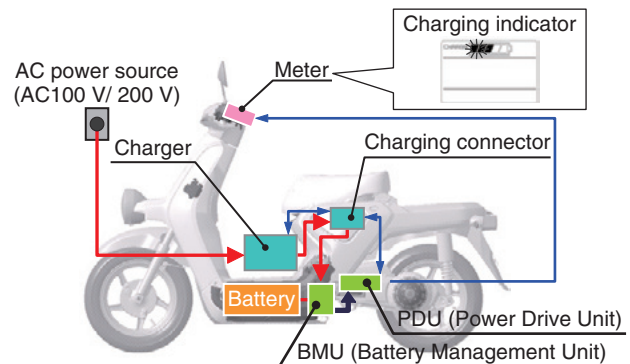


Fig. 17 Schematic drawings of charging system

Table 2 Quick charger specifications

|                  |                                           |
|------------------|-------------------------------------------|
| Size             | W526 mm × H400 mm × D274 mm               |
| Weight           | 18.8 kg                                   |
| Charging time    | Approx. 30 min (at ambient temp. of 25°C) |
| AC power source  | Single-phase 200 V                        |
| Charging current | 1st stage: 28.4 A ⇒ 2nd stage: 3.2 A      |
| Input            | 200 V 2900 VA 50/60 Hz                    |
| Output           | DC86 V: 14.2 A × 2 DC14 V: 2 A            |

### 6.1 Quick Charger

The principal specifications are shown in Table 2. The input power is provided from single-phase 200 V power source to provide the output power at DC 86 V with the output current of 28.4 A, and it is capable of charging the battery from 0% to 80% within 20 minutes.

To ensure to reach the full-charged status (SOC 100%), two-step current charging method has been applied, in which the output current is switched from 28.4 A to 3.2 A when it gets close to the full-charge status. The relation between the charging time and the capacity of the battery with two-step current charging method is shown in Fig. 18.

In order to obtain current output of 28.4 A, the charger is configured with two converters, each of which has the output capacity of 14.2 A. Compact size and light weight, 526 mm in width, 400 mm in height, 274 mm in depth and the weight of 18.8 kg, are realized with optimized arrangement of the circuit boards and the cooling mechanism. In addition, it has a holding grip to make it portable and handy without any specific location needed for its operation.

As shown in Fig. 19, the output connector has two sets of charger output terminals to use both of them for the quick charger which needs larger charging current and one of them for the standard charger, described below, which needs only small charging current. That allows sharing the same charging cable to realize the convenience of using it without care of the charger type.

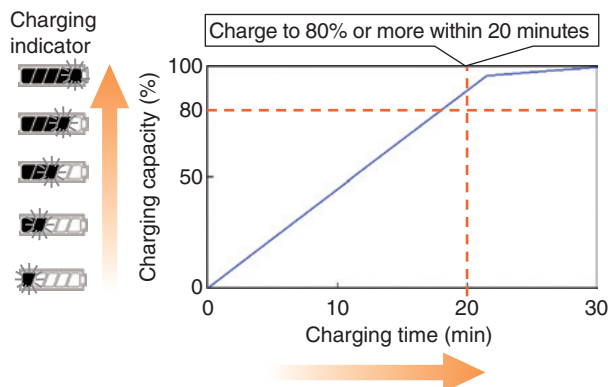


Fig. 18 Quick charge characteristics

Table 3 Standard charger specification

|                  |                                  |
|------------------|----------------------------------|
| Size             | W244 mm × H189 mm × D172 mm      |
| Weight           | 3.2 kg                           |
| Charging time    | 3.5 h (at ambient temp. of 25°C) |
| AC power source  | Single-phase 100 V               |
| Charging current | 3.2 A                            |
| Input            | 100 V 275 VA 50/60 Hz            |
| Output           | DC86 V: 3.2 A                    |

### 6.2. Standard Charger

The main specifications of the standard charger are shown in Table 3. The input is single-phase 100 V power and the output voltage is 86 V with output current of 3.2 A. It was made capable of charging a battery from 0% to 100% of capacity in 3.5 hours.

The optimized positioning of the circuit boards and cooling mechanisms realizes a compact size and light weight with 244 mm in width, 189 mm in height, 172 mm in depth, and 3.2 kg in weight. As shown in Fig. 20, it can be stored in the utility box.

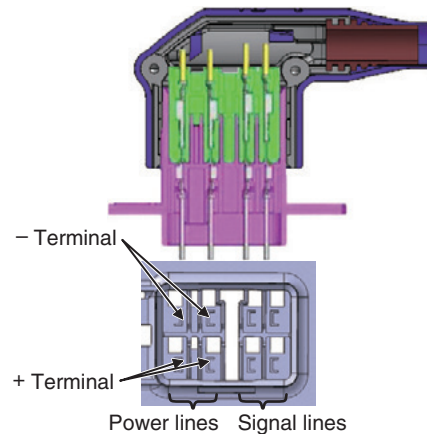


Fig. 19 Charger connector



Fig. 20 Storage of standard charger

## 7. Conclusion

The target performances of an electric motorcycle that satisfied the requirements of the driving patterns and operational patterns in business applications were determined based on the survey results on the usage patterns in those applications, which could be divided into two types, i.e. the short round trip type of delivery services and the sequential multi-destination type of delivery services.

With these processes, an electric motorcycle, EV-neo, was developed to meet the aforementioned performance objectives with a minimum amount of onboard battery and the following results had been achieved:

- (1) Compact size and the efficiency of 75% or more for all the operation range of practical use were achieved by applying the IPM motor with rare earth magnets, the automatic centrifugal clutch and the integrated power train unit with the PDU, and performances of hill-climbing departure capability of 12 degrees with a rider plus 30 kg load and range of 34 km per one charging at constant speed of 30 km/h was realized.
- (2) The operational capability for ambient temperature from -10°C to 35°C together with the durability that withstands the environmental conditions in practical use including driving in puddles and rains was achieved by applying the aforementioned battery package structure, which fulfilled a good cooling capability and a good impact resistance, and the battery output restriction controls.
- (3) With the quick charging system, which adopted two-step charging current control, the quick charging from 0% to 80% within 20 minutes was realized that satisfied the performance requirements for business applications with the minimum battery loaded on the vehicle.

## References

- (1) Shinmura, H., Nabeya, M., Cho, T.: The Development of “EV-neo,” a Tough and Useful Electric Motorcycle for Business Use, Journal of Society of Automotive Engineers of Japan, Vol. 65, No. 3, p. 114-115 (2011)
- (2) Nakazawa, Y., Kumagai, C.: Development of Electric Scooter for Practical Use, Honda R&D Technical Review, Vol. 6, p. 17-23

### ■ Author ■



Toshiyuki CHO



Koichiro HONDA



Masahide MIMURA

# Development of Midrange Outboard Motor BF60 with Equipment for User Comfort

Mitsuharu TANAKA\*

Norikazu SHIMIZU\*

Hiroshi YAMAMOTO\*

Shigekazu SAKATA\*

Koji YASUDA\*

Masahiko YAMAGISHI\*

## ABSTRACT

Midrange outboard motors are installed on a variety of different boat types and used for a range of different applications, and customer demands are diverse. The newly developed midrange outboard motor BF60 was given a top speed 8.6% greater than outboard motors of the same class by increasing the engine power output and reducing the drag of the underwater portion of the motor. Fuel economy was also enhanced by 13% by expanding the range of situations where lean-burn control operates. Handling was also enhanced by devising the first wide-angle steering system for a midrange outboard motor, and the product was made highly convenient by incorporating such technologies as the newly configured multi-functional tiller handle with enhanced operability, the configuration with added Power Thrust gear case that yields higher thrust force, and the expanded range of options for attaching outboard motor gauges by complying with the NMEA 2000 standards. The adoption of the Wedge Shape form also gave material form to the concept of beauty, which is a theme common to Honda power products design.

## 1. Introduction

Honda has developed the BF60 to follow the BF75/90<sup>(1)</sup> and the BF40/50 in its new lineup of midrange outboard motors. Figure 1 shows an external view of the BF60.

### 1.1. Various Uses of the Midrange Outboard Motor

The midrange outboard motor has a variety of different

uses, from leisure uses like angling and pontooning to commercial uses like boat rentals and fishing. The boat types on which it is installed are also numerous, from aluminum light-weight boats (mass of 100-500 kg) to pontoon boats (mass of 400-1200 kg). The required functionality and performance vary with the boat type. The characteristics of major boat types are described below.

#### 1.1.1. Aluminum fishing boat

The aluminum fishing boat in Fig. 2 is operated primarily for the purpose of fishing on lakes and rivers. In



Fig. 1 External view of BF60



Fig. 2 Aluminum fishing boat

\* Power Products R&D Center

addition to long voyage range (low fuel consumption) and speedy movement from point to point in order not to miss the timing for fishing, this boat is also required to offer precise low-speed control for trolling.

The type of tiller handle shown in Fig. 3 may also be used, depending on the region and the application. The tiller handle is equipped not only for steering but comprises a centralized boat control system with throttle, shift, and so on.

#### 1.1.2. Rigid inflatable boat

The rigid inflatable boat shown in Fig. 4 is used mainly for enjoyment of power performance on lakes, rivers, and the ocean. The bottom of the boat is given a deep V angle, which provides enhanced turning performance. The board



Fig. 3 Tiller handle type



Fig. 4 Rigid inflatable boat



Fig. 5 Pontoon boat

is also enclosed by floats, giving the boat high levels of stand-still stability and anti-overturn performance, so that these craft are commonly used for diving support, rescue, and other such applications. In addition to requiring high levels of power performance and steering stability, there are also strong demands for throttle linearity that provides a response from every point across the speed range.

#### 1.1.3. Pontoon boat

The pontoon boat shown in Fig. 5 is a major boat type in North America. It has a flat deck with sofas and tables affixed to it, and is used on lakes, rivers, and bays with few waves. Its purpose is to serve as a waterborne base for parties or swimming, and for enjoyment of fishing and other such leisure activities. There is strong demand for this type as a rental boat in tourist resort areas, and it is not uncommon for it to be used by customers who are not accustomed to operating boats. The bottom of the boat is formed from a combination of two or three large aluminum tubes, giving the craft superior stability, but it is also unsuited to tight turns.

### 1.2. Concept

The BF60 was developed in line with three basic concepts: “Enjoyable to use: Fun,” “Economical to use: Eco,” and “Simple to use: Easy.” Not only do the specifications for this model place it at the top of the class for basic performance in terms of the power performance and high fuel economy characteristic of Honda outboard motors, but a variety of “comfortable equipment” was also developed with the aim of providing customers with satisfaction in their diverse kinds and patterns of use. The Wedge Shape was adopted as a new design that represents these enhanced elements of performance in order to position this as a premium midrange outboard motor.

## 2. Technology Employed in the BF60

The four key points on which the product appeal of the BF60 were determined were power performance, environmental performance and fuel economy, comfortable equipment, and design.

### 2.1. Power Performance

The BF60 does not simply realize increased power output in the engine. It also achieves the top level of power performance for outboard motors in its class by various means, including reducing the drag in the water of the gear case, propeller, and other underwater sections.

The compact gear case on the standard model was newly designed to achieve increased maximum speed and enhanced fuel economy by holding down the size of the front projection area of the underwater section, and further reducing water resistance by using fluid analysis to smooth out pressure variations in the water flow. A somewhat high gear ratio (29:14) for this class was adopted in order

to heighten propeller efficiency with the greatest priority placed on maximum speed.

As a result of the above, in combination with the output power characteristics of the engine, the BF60 achieved maximum speed more than 8.6% (4.5 km/h) greater than an outboard motor of the same class on a 16-ft aluminum fishing boat. This model also adopted an ignition timing control interlocking air-fuel ratio (a boosted low-speed torque system) that enhances acceleration performance by increasing the mass of the fuel injection and advancing ignition timing up to the limit before knocking occurs during rapid acceleration.

Figure 6 shows the pressure distribution diagram produced by fluid analysis. Figure 7 shows the maximum speed. Figure 8 shows the acceleration time.

The BF60 was configured not only with the standard model gear case as mentioned above, but also with the Power Thrust model gear case. The purpose of these gear

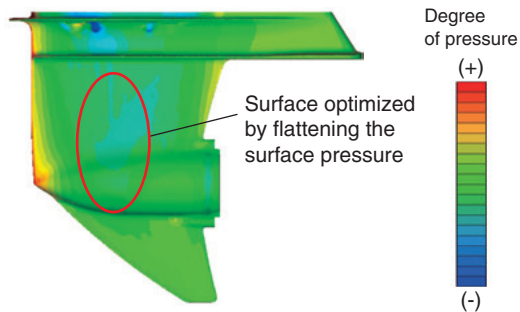


Fig. 6 Pressure distribution on the gear case

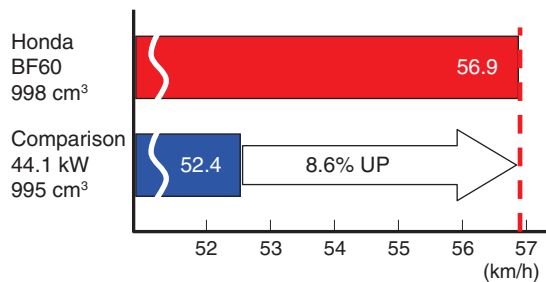


Fig. 7 Maximum speed

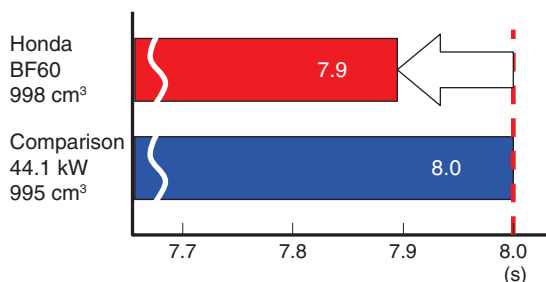


Fig. 8 0-50 m acceleration time

cases is to enhance compatibility with a variety of boat types, and its characteristics are described below.

The Power Thrust model utilizes the gear case of the next-higher model, the BF90. This specification turns a larger-diameter propeller at slow speed so that the propeller thrust force is increased without changing the engine displacement. Placing the propeller at an appropriate depth underwater limits the reduction in thrust caused by ventilation at the water surface, while also limiting the front projection area of the underwater section in consideration of the influence on maximum speed.

The Power Thrust model is frequently installed on heavy boats, where reverse thrust force also is an important performance element to change direction or stop a boat that is coasting forward (moving by inertia). Reverse thrust force was therefore considered important for the BF60, and the below exhaust system was adopted.

Ordinarily, outboard motors discharge exhaust gases through the center of the propeller out into the water in order to enhance exhaust efficiency during forward movement. During reverse movement, however, these exhaust gases can readily be ventilated around the propeller and it becomes difficult to obtain a high reverse thrust force. In addition to the exhaust outlet in the center of the propeller, therefore, another exhaust outlet was also placed higher up. This reduced the exhaust ventilation onto the propeller during movement in reverse without adverse effect on efficiency during forward movement.

Figure 9 shows the exhaust system around the gear case.

As a result, this motor achieved reverse thrust force that exceeds not just comparison motors with the same displacement, but also comparison motors with

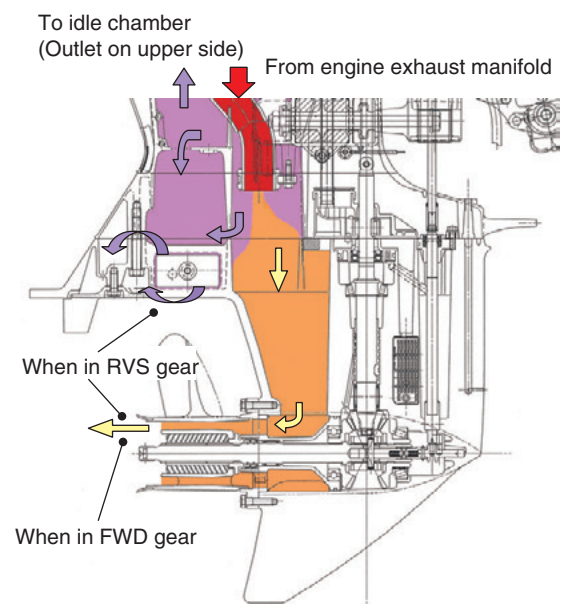


Fig. 9 Path of exhaust gas

displacement that is 1.3 times larger.

Figure 10 shows the comparison of propeller thrust force.

## 2.2. Environmental Performance and Fuel Economy

The BF60 achieves a balance between heightened fuel economy and enhanced environmental performance that complies with the new exhaust emission regulations of the U. S. Environmental Protection Agency (EPA).

The new exhaust emission regulations adopt the HC and NO<sub>x</sub> regulation values of the California Air Resources Board (CARB) starting with the 2010-year model, and also newly added CO regulation values.

Additional regulations on the Not-to-Exceed (NTE) zone and regulations on fuel permeation have also been added for new models.

Figure 11 shows the low-permeation fuel hose that was adopted to support the fuel permeation regulations. Figure 12 shows an overview of the NTE zone regulations.

The BF60 embodies further refinements to the lean-burn control<sup>(2)</sup> that is a characteristic of Honda outboard motors. The range of situations where lean-burn control operates was expanded, the control map was made more fine-grained, and thus a balance of environmental performance and fuel economy was realized. Through these measures, the BF60 achieved a 13% enhancement in fuel economy in practical use relative to outboard motors of the same class on a 16-ft aluminum fishing boat.

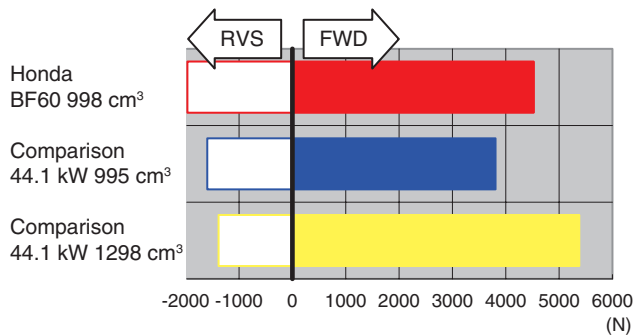


Fig. 10 Maximum thrust force of propeller

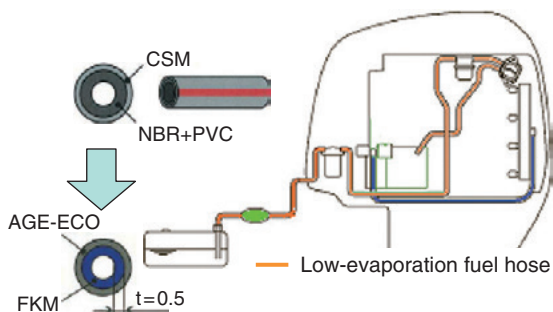


Fig. 11 Low-permeation fuel hose

Figure 13 shows fuel consumption during practical use of the motor. Figure 14 shows a conceptual map of boat load characteristics and air-fuel ratio.

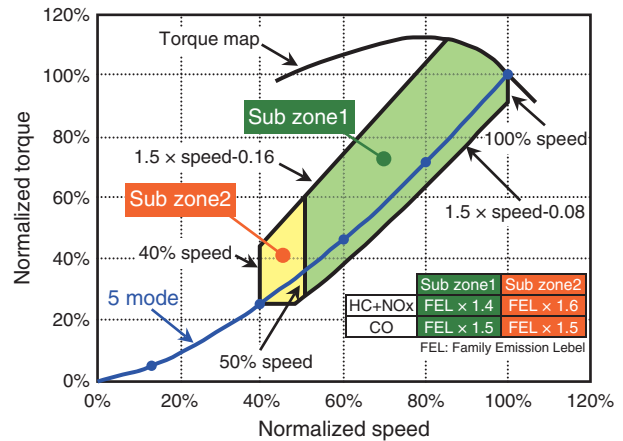


Fig. 12 NTE zone map under EPA regulations

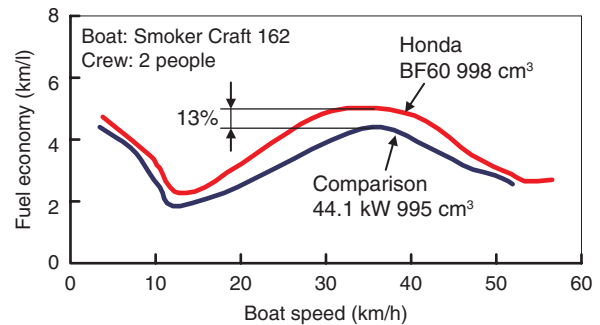


Fig. 13 Mileage in practical use

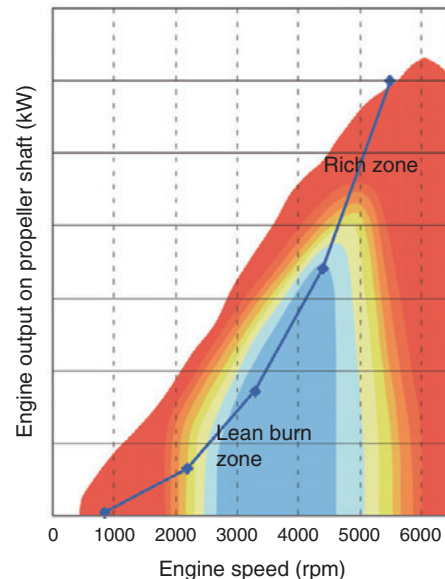


Fig. 14 Map of A/F ratio

## 2.3. Comfortable Equipment

### 2.3.1. Steering with wide steering angles

The pontoon boats that are commonly used in North America are a boat type with high cruising resistance. Consequently, even the installation of a high-powered outboard motor was not expected to yield very much greater power performance, and these boats can often be found, therefore, with underpowered outboard motors installed on them. When changing direction or in docking scenarios, for example, when the capability to make tight turns is required, it would be necessary to do repeated backing and turning with inadequate propeller thrust. It was in order to free users from this kind of troublesome maneuvering that ways were devised and adopted to give the BF60 a wider steering angle, increased from the industry standard of 30° to left and right to 50°.

These measures enable the BF60 to make turns with a minimum turning radius that is half or less than half that of outboard motors of the same class. This has yielded a significant enhancement in maneuverability.

Figure 15 shows a comparison of minimum turning diameters.

The outboard motor steering system has been configured from industry technical standards issued by such groups as the International Council of Marine Industry Associations and the American Boat and Yacht Council. The movement distance of the steering pivot is limited accordingly. This pivot and the outboard motor are connected by a steering rod that completes the outboard motor steering system. The steering angle is therefore determined by the distance between the center of the steering axis and the connection point of the steering rod. When this distance is made shorter, the outboard motor steering angle can be made wider.

The steering rod connection point is ordinarily located on the center line when the outboard motor is in a neutral

steering situation. If the layout is simply configured this way for 50° steering, however, the link ratio during steering will be 84% when steering left and 39% when steering right. This means that the operation load of steering will be conspicuously greater when steering right.

In the BF60, the steering rod connection point is offset from the center line. It is configured to make for a link ratio of 71% for both steering left and steering right.

These measures were taken to keep the steering load from becoming extremely heavy while also minimizing the load difference between steering left and right in order to enhance the steering feeling.

Figure 16 shows the layout of the steering system. Figure 17 shows the link ratio.

### 2.3.2. Multi-functional tiller handle

The outboard motor tiller handle is used to control the steering, throttle, and shift operations. In addition, it controls the power trim and tilt (a mechanism that changes the vertical angle of the outboard motor, thereby adjusting the boat's cruising posture and speed), which are distinctive

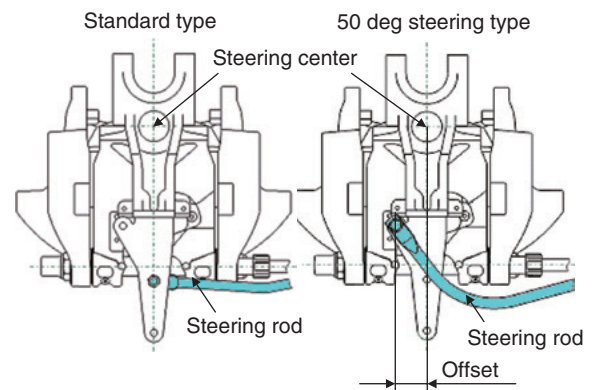


Fig. 16 Layout of steering system

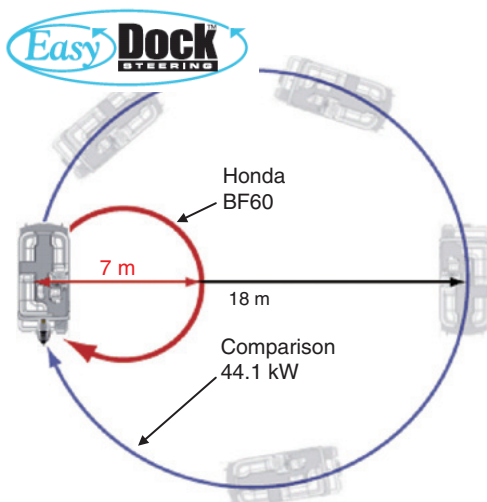


Fig. 15 Minimum turning diameters

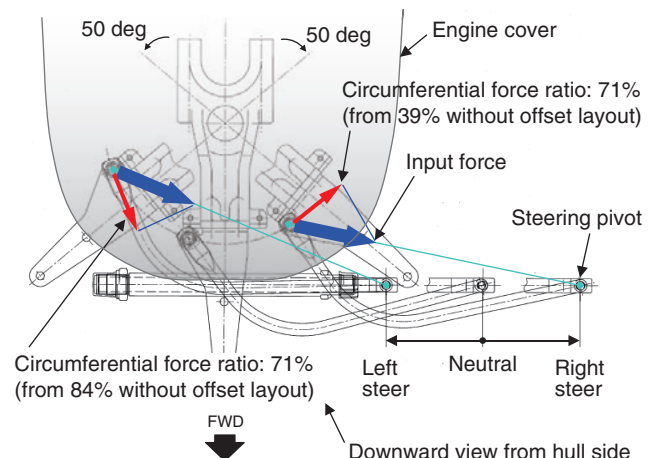


Fig. 17 Link efficiency

adjustment features of an outboard motor. As the tiller handle brings together these various functions, its ease of use is important. The BF60 tiller handle was newly developed with thoroughgoing pursuit of ease of use as an aim, and the shape and layout of the control systems were decided accordingly.

On craft with tiller handles, the operators ordinarily sit on the starboard side and operate the tiller with their left hands. The controls on the tiller handle are therefore configured on the assumption that the operator will be seated to starboard. Unlike automobiles and other such vehicles, however, the steering position taken in the case of an outboard motor is not defined by the seat. Depending on the shape of the craft or the handedness of the operator, the operator may in some cases sit on the port side. A reversible shift lever structure was therefore adopted for the BF60 tiller handle so that the operator can choose to attach the shift lever either to the left or to the right. This makes the shift lever easy to operate with either the left hand or the right hand.

When adjusting the maximum speed while cruising at full throttle, it is also common for operators to operate the throttle and steer with one hand while using the other hand to hold on to some part of the hull so as not to be thrown from the boat. Under these circumstances, throttle operation and trimming are performed simultaneously, and it is desirable, therefore, that the trim switch allows operation with the hand that is holding the throttle grip. The BF60 tiller handle has the power trim and tilt switch built into the grip near the end of the throttle grip. This places the switch in a constant fixed position relative to the hand that is holding the throttle, so that operators can readily operate the switch without changing their grip on the throttle.

Figure 18 shows an external view of the tiller handle.

On the other hand, a craft can be affected by wind and tidal flow during troll fishing and other extremely low-speed movement, so that fine adjustment of boat speed is required. The BF60 was designed to enhance this kind of use by configuring a trolling control system that can adjust the engine speed by 50-rpm increments when operating at low speed between 750 and 1000 rpm.

Figure 19 shows a conceptual image of speed adjustment using the trolling control.

This allows the trolling speed of a 16-ft aluminum boat using a 15-inch-pitch propeller to be adjusted at 0.3-km/h increments from 3.5 km/h to 5.0 km/h by the operation of a single switch. This enables easy fine adjustment of boat speed for troll fishing.

This functionality can also be provided with the remote control handle specification by the use of an added panel switch.

### 2.3.3. Expanded selection of gauges for use with outboard motors

In the case of outboard motor craft, there is a strongly

rooted practice of having dealers provide finished boats as a customized package with rigging, including fuel gauge, battery voltmeter, and other electronic instruments, from different manufacturers than the outboard motor. However, choices of speed gauges, tachometers, and other instruments have come to be practically limited to genuine branded products from the same manufacturers as the outboard motors due to issues of compatibility with the engines. In other words, there has been a narrowing of the choices whereby customers could exercise their preferences. This was addressed by the NMEA 2000 standards, adopted in 2001, that unify the communication protocols for marine electronic instruments, making it easier to connect all types of electronic instruments.

NMEA 2000 standards<sup>(3)</sup> are unified communication protocols for marine electronic instruments defined by the National Marine Electronics Association in the United States. Worldwide, 500 or more companies that are manufacturers and dealers subscribe to the standards.

Making the BF60 compliant with the NMEA 2000 standards has enabled it to use a wide range of marine



Fig. 18 Multi-functional tiller handle

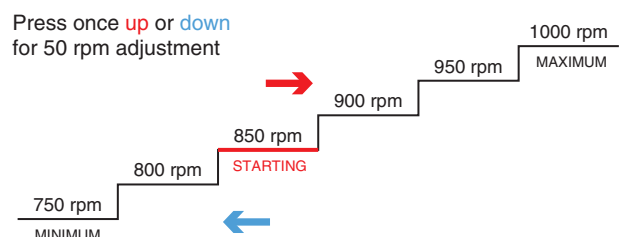


Fig. 19 Variable trolling speed control

Table 1 Specifications

|                                         |                                             |
|-----------------------------------------|---------------------------------------------|
| Code name                               | BF60A L-type                                |
| Overall length (mm)                     | 777                                         |
| Overall width (mm)                      | 417                                         |
| Overall height (mm)                     | 1397                                        |
| Transom height (mm)                     | 521                                         |
| Dry mass (kg)                           | 110                                         |
| Engine type                             | 4 stroke water cooled<br>in-line 3 cylinder |
| Displacement (cm <sup>3</sup> )         | 998                                         |
| Bore (mm) X Stroke (mm)                 | 73.0 X 79.5                                 |
| Compression ratio                       | 9.2 : 1                                     |
| Valve train                             | SOHC 12 valves                              |
| Maximum power (kW/rpm)                  | 44.1 / 5500                                 |
| Maximum torque (N·m)                    | 79                                          |
| Recommended engine<br>speed range (rpm) | 5000 - 6000                                 |
| Trolling engine speed (rpm)             | 850 (750 - 1000)* <sup>1</sup>              |
| Gear ratio                              | 2.07:1, 2.33:1* <sup>2</sup>                |
| Fuel supply system                      | Programmed fuel<br>injection                |
| Ignition system                         | Programmed ignition                         |
| Charge output (A)                       | 22                                          |
| Exhaust system                          | Through propeller hub                       |

\*1 : ( ) under variable trolling speed control

\*2 : Power thrust model

electronic instruments of various types that support this standard in addition to genuine Honda brand gauges. This has greatly expanded the range of choices in electronic instruments open to users.

The NMEA 2000 standards do not cover only those items of engine information that support the standards. It includes domains of information that can be defined at each maker's discretion, and in the BF60, these domains are used for output of Eco Lamp signals. NMEA 2000-compliant instruments that support the Eco Lamp signals have also been configured for Honda use.

Eco Lamp signals belong to the system that causes a green lamp to illuminate on NMEA 2000 instruments that support the Eco Lamp when the engine satisfies lean-burn control conditions. This enables users to visually confirm the range of situations where lean-burn control operates,

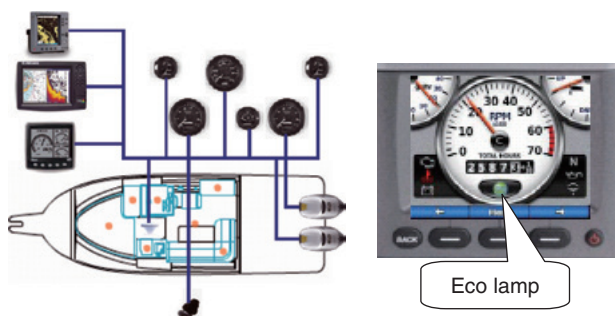


Fig. 20 NMEA 2000 instrument network

which had been difficult to perceive up to this time. This capability allows users to sense more easily when they are cruising under positive fuel economy conditions.

Figure 20 shows a conceptual image of electronic instruments networked under the NMEA 2000 standards.

## 2.4. Main Specifications

The BF60 engine was newly designed as a three-cylinder version of the 1496 cm<sup>3</sup> in-line four-cylinder engine for the Fit, making it a 998 cm<sup>3</sup> dedicated outboard engine. The mainstream of outboard motors in the same class are four-cylinder engines. The choice of a three-cylinder format allowed the engine to be made lighter and more compact.

The main specifications of the BF60 are shown in Table 1.

## 3. Wedge Shape Design

The concept for the exterior design was the dynamic Wedge Shape design that alludes to power performance that leads its class. This builds the product's identity as a new generation of Honda outboard motor. The key appeal made with the BF60 to differentiate it from other companies' products was its premium quality, and it was formed with persistent attention to quality down to the smallest details of design. Specifically, the top cover was divided into two parts, the Wedge Shape character line was highlighted, and the water through-gate on the front was sculpted using the motif of an animal's eye, adding an individualistic expression. The splash line on the lower cover was modeled in relief, which heightened the variety of the surface and also contributed to prevention of fouling resulting from splash run-up on the upper part of the main body.

The incorporation of these methods realized the creation of a premium-quality midrange outboard motor that gives material form to the concept of *beautility*, which is a theme common to Honda power products design.

Figure 21 shows the BF60 design concept. Figure 22 shows the top cover.

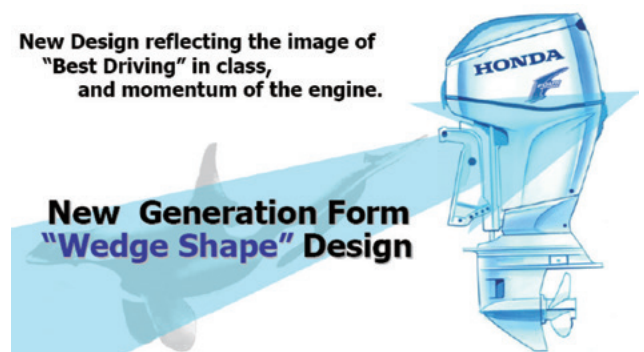


Fig. 21 The BF60 design concept



Fig. 22 The top cover of the BF60

#### 4. Conclusion

The development of the BF60 not only sought enhanced driving performance and environmental performance. It also involved developing comfortable equipment in order to address the variety of demands made of a midrange outboard motor. This gave concrete form to the following:

- (1) The maximum speed was increased by 8.6% relative to outboard motors in the same class.
- (2) A fuel economy enhancement of 13% was achieved relative to outboard motors in the same class.
- (3) Steering with a 50° steering angle left and right was devised and adopted, enhancing maneuverability.
- (4) Two types of gear case were configured, enhancing suitability to boat types.
- (5) The new configuration of a multi-functional tiller handle enhanced operability.
- (6) The range of choices of marine electronic instruments that comply with the NMEA 2000 standards was expanded.
- (7) The adoption of the Wedge Shape design realized *beautility*.

#### 5. Epilogue

The developers intend to make use of the technology and know-how acquired through development of the new model BF60 in developing the outboard motors of the future, and hope to continue building products that embody the customer's desires.

#### References

- (1) Yoshimura, H., Fujima, A., Tsubouchi, M., Miyauchi, T., Kimura, T.: Development of New BF90 Outboard

Motor, Honda R&D Technical Review, Vol. 19, No. 1, p. 21-26

- (2) Yonezawa, M., Matsuda, Y., Tawa, H., Shidara, S., Bungo, K., Suzuki, Y.: Development of High-power, High Fuel Economy BF150 Outboard Motor, Honda R&D Technical Review, Vol. 16, No.1, p. 29-38

- (3) NMEA2000 Main Document Version 1.200

#### ■ Author ■



Mitsuharu TANAKA



Norikazu SHIMIZU



Hiroshi YAMAMOTO



Shigekazu SAKATA



Koji YASUDA



Masahiko YAMAGISHI

# Development of Micro Combined Heat and Power Generation Unit with Extended Expansion Linkage Engine

Yoshiharu TAKITA\*

Shohei KONO\*

Hideaki TAKADA\*

Kunihiro KITSU\*

Atsumu NAOI\*

Koichi TSUNO\*

## ABSTRACT

A micro combined heat and power generation (CHP) unit that uses an extended expansion linkage engine (EXlink) has been developed with the aim of simultaneously enhancing fuel efficiency and reducing the unit size. This is the first mass production known to date of a compact engine that uses an original multiple linkage system to realize an Atkinson cycle.

EXlink increases the power output efficiency by using a linkage system to extend the expansion and exhaust stroke lengths relative to the intake and compression stroke lengths. Use of EXlink in a micro CHP unit increased the power generating efficiency by 3.8 points to 26.3% (lower heating value: LHV), and optimization of the heat flow design also increased the overall efficiency by 6.5 points to 92% (LHV), compared to the 2006 model.

The increased engine efficiency enabled reduction of the intake air amount which reduced the intake and exhaust noise and achieved a CHP unit noise value of 43 dB(A)/1m (Japanese specifications).

A depth of 298 mm was achieved in consideration of installation in the narrow spaces around a typical Japanese house. In addition to revision of the parts layout, this realized a 33% reduction in chassis volume and an 11-kg weight reduction compared to the 2006 model.

## 1. Introduction

Environmental awareness continues to increase due to global climate fluctuation and energy issues, and various efforts are being made in each sector to reduce CO<sub>2</sub> emissions. Energy consumption in the civil sector (households) also continues to grow steadily around the world, and has become a major factor behind the nonstop growth in overall energy consumption.

Combined heat and power generation (CHP) systems use a generator to generate electricity, while simultaneously recovering and using the heat produced by the power generation process. CHP systems are viewed as an effective means of reducing CO<sub>2</sub> emissions, and have been widely used for some time centering on large-scale private industrial power generating facilities that also serve as heat sources.

Honda applied its engine and inverter generator technologies developed for power products, and released a compact CHP unit for Japanese households in 2003<sup>(1)</sup>. This was followed by the release of a second-generation model (hereafter “2006 model”) in 2006, and had achieved cumulative shipments of 110000 units as of June 2011.

Honda has now developed a third-generation micro CHP unit that provides further enhanced environmental performance in an even more compact size. This unit is the first known to date to employ an extended expansion linkage engine (hereafter “EXlink”) that uses an original multiple linkage system to realize an Atkinson cycle.

## 2. Development Goals

Most industrial CHP systems are mainly used to supply electrical power, and employ a “power first, heat second” operating pattern that constantly generates electrical power. The heat produced at that time is also used, and excess heat is disposed. On the other hand, households can be expected to demand not only electricity, but also a certain amount of heat in the form of hot water and heating. Therefore, Honda systems employ “heat first, power second” operation that stores heat in the form of hot water in a hot water storage tank, and operates until the hot water reaches a certain temperature and amount. Operation is controlled according to predicted high power and heat usage time periods based on learned time periods, realizing a low-waste household system.

\* Power Products R&D Center

Figure 1 illustrates micro CHP system installation in a typical Japanese household.

Electric power generated by the CHP unit installed outdoors passes through a power distribution board and is consumed by the electric appliances in the household. Heat is stored as hot water in the hot water storage tank, and used as necessary for hot water supply, underfloor heating, bathroom heating and other heating uses. A supplementary boiler is installed in the hot water storage tank to prevent the system from running out of hot water.

Like the previous systems, the newly developed model also employs heat first, power second operation, and generates power of 1 kW while using both a hot water storage tank and supplementary boiler. Product appeal was enhanced as follows.

(1) Increased power generating efficiency

Electricity has a higher economic value compared to hot water with the same amount of energy, and is a form of energy that can be converted to various applications. CHP systems produce high-value electrical energy, and the remaining energy is also not wasted and can be used as hot water. This means that CHP systems can be said to be more economic when compared to systems such as hot water boilers that convert the energy of fuel into hot water. As a result, higher power generating efficiency is demanded of CHP systems as power generating facilities.

(2) Increased heat recovery efficiency

Micro CHP systems are both power generating facilities and hot water supply facilities, so the heat recovery efficiency is extremely important. When only the power generating efficiency is increased without changing the amount of energy discarded from a micro CHP unit, the amount of recovered heat decreases. Therefore, efforts were made to increase the heat recovery efficiency by minimizing the ventilation air flow. In addition, an oil cooler heat exchanger integrated with an oil pan was used to increase the amount of heat recovered from the engine oil.

(3) Reduced vibration and noise

Japanese specification micro CHP units are installed on residential land and are also anticipated to operate in the early morning and evening hours when there is heat demand, so a higher level of quietness is demanded compared to other products with engines. In addition, it is

assumed that the reduced size will result in installation in even narrower outdoor spaces compared to the 2006 model, so development aimed to both reduce the overall noise and enhance the sound quality.

(4) More compact size

The goal was to develop a unit with a width of 300 mm or less that could be installed in spaces as small as 650 mm, including the maintenance space, to enable installation and the necessary maintenance in narrow spaces between houses and property lines (Fig. 2). In addition, the overall height was also reviewed to facilitate installation below bay windows, with the ultimate goal of a 33% reduction in volume compared to the 2006 model.



Fig. 2 Example of 650 mm exterior space

### 3. Introduction of Developed Technologies

#### 3.1. Increased Power Generating Efficiency

##### 3.1.1. Use of EXlink

The EXlink engine was employed to increase the power generating efficiency.

EXlink adds trigonal link, swing rod and eccentric shaft linkage systems to the connecting rod and crankshaft of a normal engine, and rotates the eccentric shaft synchronized at half of the crankshaft rotations. This realizes an Atkinson cycle in which the lengths of the intake and compression strokes differ from those of the expansion and exhaust strokes (Fig. 3)<sup>(2)-(9)</sup>.

Setting shorter intake and compression stroke lengths and longer expansion and exhaust stroke lengths enables more work to be performed with less intake, which increases the thermal efficiency compared to a normal engine.

Table 1 lists the specifications of the engine (GE160V) mounted in the 2006 model and the engine (GFV110) mounted in the newly developed model (hereafter “2011 model”). The compression ratio was set to the same as the 2006 model, and the expansion ratio was set at 17.6. In addition, reducing the bore diameter enabled reduction of the combustion chamber size, and arranging the intake and exhaust valves in a radial shape around the combustion chamber with the spark plugs located closer to the center

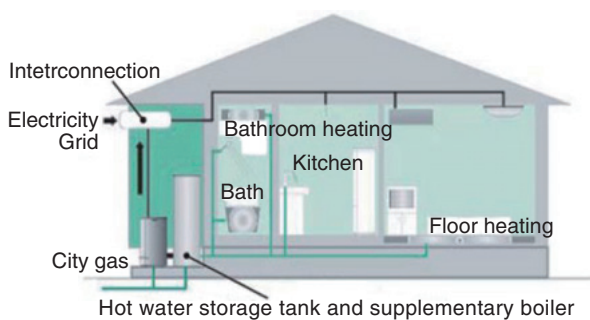


Fig. 1 Micro combined heat and power generation system

Table 1 Comparison of 2006 model and 2011 model engine specifications

|                                              | 2006 model | 2011 model |
|----------------------------------------------|------------|------------|
| Engine name                                  | GE160V     | GFV110     |
| Bore diameter (mm)                           | φ62        | φ53        |
| Intake/compression stroke (mm)               | 54.0       | 49.8       |
| Expansion/exhaust stroke (mm)                | 54.0       | 74.1       |
| Intake/compression volume (cm <sup>3</sup> ) | 163        | 110        |
| Expansion/exhaust volume (cm <sup>3</sup> )  | 163        | 163        |
| Compression ratio                            | 12.0       | 12.2       |
| Expansion ratio                              | 12.0       | 17.6       |
| Rated speed (rpm)                            | 1,950      | 1,950      |

created a hemispherical combustion chamber with good combustion efficiency. Figure 4 shows the engine structure.

In addition to the higher expansion ratio and modified combustion chamber shape, the intake stroke volume was also optimized. This increased the engine brake thermal efficiency at rated operation by 16.4% compared to the 2006 model.

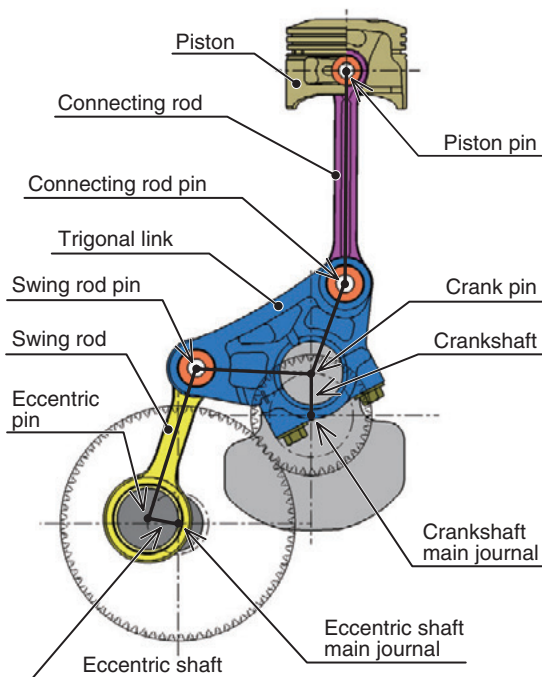


Fig. 3 Configuration of EXlink

### 3.1.2. Increased electrical conversion efficiency

The conversion efficiency of the alternator that converts the axial power output generated by the engine into electrical power output and the inverter unit that converts that electrical power output into interconnectable electric power was also increased in order to increase the power generating efficiency. Figure 5 shows the interconnection power generating block diagram.

### 3.1.3. Technologies to increase alternator efficiency

Alternator loss consists of iron loss that depends on the magnetic materials and copper loss that depends on the windings. Iron loss is divided into eddy current loss, which is the Joule heat loss when eddy current is generated by a magnetic field passing through the iron core, and hysteresis loss, which is generated according to the hysteresis curve (B-H) of the magnetic body. Equation 1 shows the typical formula for eddy current loss  $We$ , and Eq. (2) shows the typical formula for hysteresis loss  $Wh$ .

$$We = af^2 Bm^2 \quad (1)$$

$$Wh = bfBm^{1.6} \quad (2)$$

$a$ : Constant determined by the magnetic body plate thickness

$b$ : Constant determined by the magnetic body material

$f$ : Alternator frequency

$Bm$ : Magnetic flux density

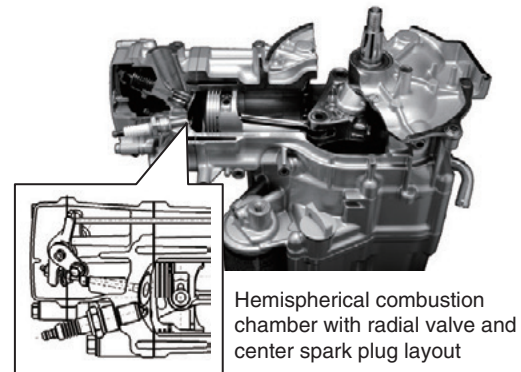


Fig. 4 External and sectional view of 2011 model engine

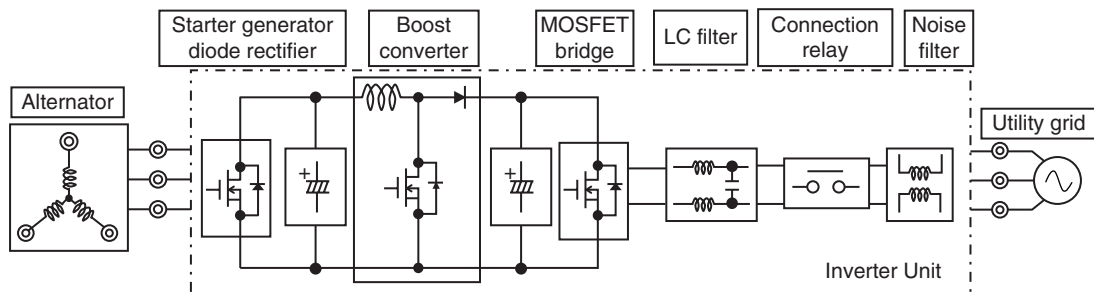


Fig. 5 Block diagram of interconnection inverter and alternator

This development focused on iron loss, and the following three technical elements were used in the 2011 model to reduce loss.

(1) Reducing iron loss by lowering the frequency

Equations (1) and (2) indicate that reducing the frequency can effectively reduce iron loss. Figure 6 shows the alternator used in the 2011 model. The frequency was lowered by reducing the number of stator slots and rotor poles. Normally this would also lower the rated voltage, but the rated voltage was kept the same as the 2006 model by increasing the core layer thickness.

(2) Reducing eddy current loss by reducing the eddy current in the core

The constant  $a$  in Eq. (1) is proportional to the square of the magnetic body plate thickness. Eddy current loss in the 2011 model was reduced by reducing the stator core plate thickness.

(3) Reducing the rate of magnetic characteristic deterioration due to blanking strain

The constant  $b$  in Eq. (2) can be reduced by eliminating factors resulting in deterioration of the magnetic characteristics of the magnetic body. In the 2011 model, the strain produced by blanking and forming was reduced by reducing the stator core plate thickness, and hysteresis loss was reduced by reducing the deterioration in magnetic characteristics.

Application of the efficiency enhancing technologies (1) to (3) above achieved an increase in the conversion efficiency of the alternator assembly.

### 3.1.4. Technology to increase inverter efficiency

Efficiency was increased in the 2006 model by carefully selecting the switching elements and the choke coil core materials to realize a high efficiency interconnection inverter<sup>(10)</sup>. In the 2011 model, loss was reduced by focusing on the switching frequencies in the boost converter block that boosts the DC voltage generated by the alternator and diode rectifier shown in Fig. 5 to a DC voltage that enables interconnection with the electric grid, and the MOSFET bridge block that converts the boosted DC voltage into an AC voltage in accordance with the grid voltage and frequency. The switching element circuit configuration and the LC filter constants were reviewed, and the switching frequency was reduced. This reduced both the switching

loss of the boost converter and MOSFET bridge blocks and the iron loss of the choke coil in the LC filter block, thus achieving an increase in the conversion efficiency of the inverter unit.

### 3.1.5. Increased power generating efficiency

By increasing the thermal efficiency of the engine and the conversion efficiency of the alternator and inverter, the power generating efficiency of the unit was increased by 3.8 points from 22.5% (lower heating value, hereafter “LHV”) for the 2006 model to 26.3% (LHV).

## 3.2. Increased Heat Recovery Efficiency

CHP units extract the energy possessed by fuel as electric power via an engine, alternator and inverter, and recover the heat generated by that process via cooling water using an exhaust heat exchanger and an engine cooling system. Low-temperature heat is absorbed by the ventilation air and exhaust, and exhausted to the outside of the system as disposed heat.

Ventilation cools by stealing the heat from the surface of each part such as the engine and electronic parts, and the air flow is set so that all components are cooled to the appropriate temperature or less. In order to increase the heat recovery efficiency, a reduction in the amount of the ventilation air was sought by ventilating effectively.

Increasing the conversion efficiency reduced the amount of heat generated from heat-generative elements, and enabled reducing the air flow required to cool electronic parts.

The ventilation air flow and the engine cooling system were enhanced to enable cooling of the entire unit by the reduced air flow.

### 3.2.1. Optimization of ventilation air flow

The locations inside the unit that require active cooling by ventilation are as follows.

- Inverter (electronics room)
- ECU (electronics room)
- Alternator (coupled to the engine)
- Engine mount (engine room)

The engine and hoses are cooled by cooling water.

In the 2006 model, a high amount of air was required because the alternator coupled with the engine was cooled using a fan mounted on the alternator by means of the heated ventilation air that had ventilated the electronics room and engine room. In the new unit, the ventilation air that cooled the electronics room is first used to cool the alternator and then ventilates the engine room (Fig. 7).

The order in which the engine mounts in the engine room and the alternator are cooled is reversed, so the average temperature might rise inside the engine room. However, this temperature rise around the mounts is prevented by enhancing the velocity and direction of the ventilation air that flows into the engine room.

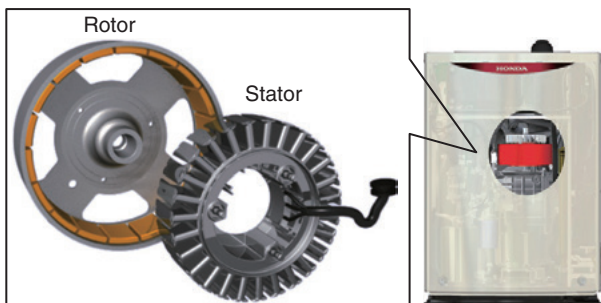


Fig. 6 External view of alternator

Figure 8 shows an example of the simulation results.

The ventilation air is exhausted to the outside of the unit via a ventilation outlet duct with a structure that prevents noise from leaking to the outside of the unit.

This realized efficient internal ventilation, and enabled ultimately reducing the cooling air flow from the previous 35 m<sup>3</sup>/h to 13 m<sup>3</sup>/h.

### 3.2.2. Engine oil cooling system

In the heat recovery system of the 2011 model, the cooling water first performs heat exchange with the engine oil before reaching the exhaust heat exchanger.

The 2006 model had a coil-shaped oil cooler (tank coil-type heat exchanger) inside the oil tank, but the 2011 model uses a configuration that integrates the oil cooler with the oil pan (tank jacket-type heat exchanger) (Fig. 9). This realized efficient heat recovery from the high-temperature

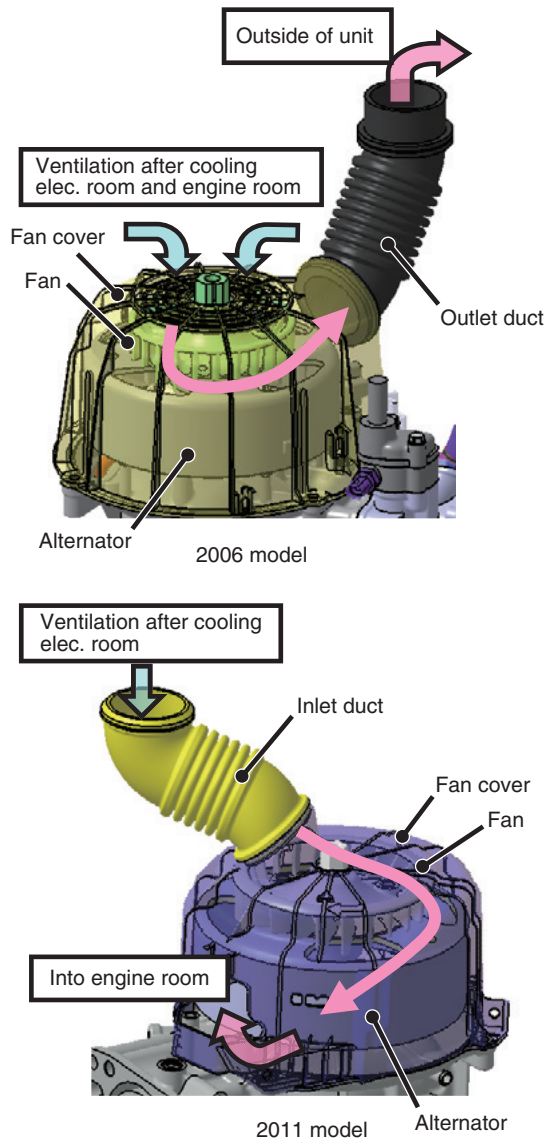


Fig. 7 Comparison of ventilation air flow in 2006 model and 2011 model

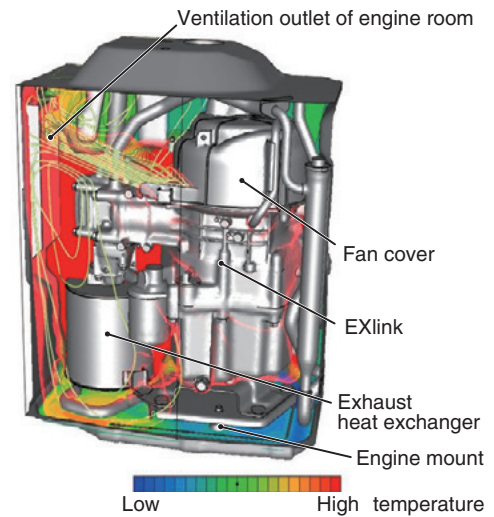


Fig. 8 Calculated flowlines and temperature distribution in engine room

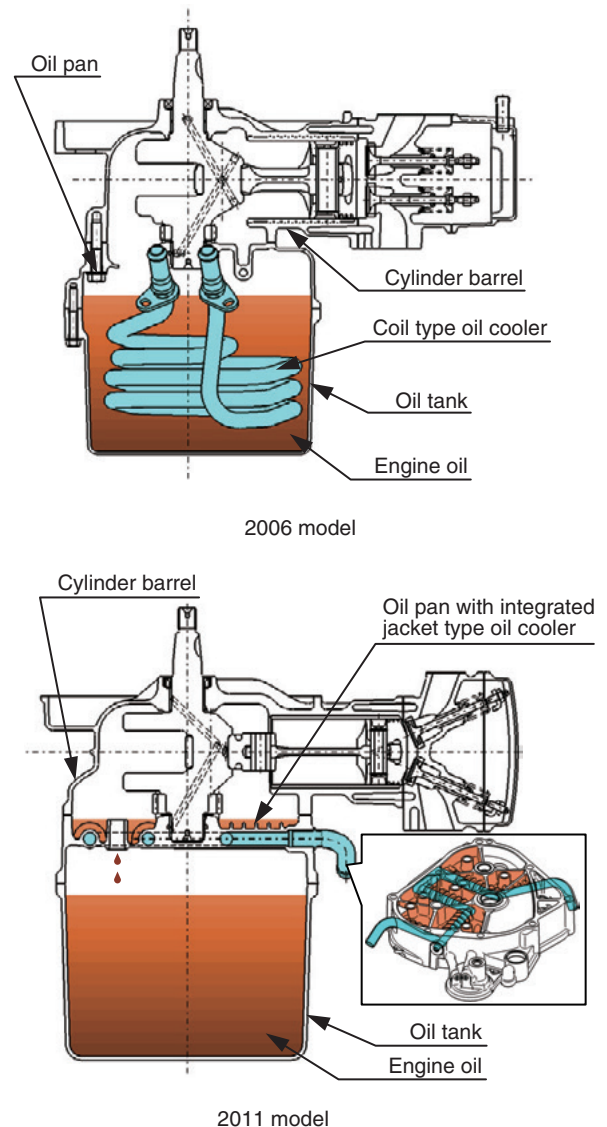


Fig. 9 Oil cooler structure

engine oil immediately after it lubricated the inside of the engine.

### 3.2.3. Increased heat recovery efficiency and increased overall efficiency

Efforts were made to increase the heat recovery efficiency by reducing the heat energy discarded to the outside of the unit. This realized a 2.7-point increase in the heat recovery efficiency from 63% (LHV) for the 2006 model to 65.5% (LHV), which increased the overall efficiency from 85.5% (LHV) for the 2006 model to 92% (LHV).

## 3.3. Technologies to Reduce Vibration and Noise

### 3.3.1. Reducing noise by using EXlink

Use of the high-efficiency EXlink reduced the intake air amount, which enabled reduction of the cubic volume of the intake silencer by 45% compared to the 2006 model. In addition, the reduction in the intake air amount is also advantageous for lowering the exhaust noise. The sound quality was enhanced by reviewing the packaging, and by increasing the muffler volume by 15% compared to the 2006 model.

### 3.3.2. Reducing vibration by optimization of the linkage layout

Increasing the expansion ratio changes the piston stroke length, so the inertia force due to piston acceleration also increases, which increases the vibration. As a countermeasure, the linkage layout applied a crank rotation angle shorter than  $180^\circ$  to the intake and compression strokes that have short stroke lengths, and a crank rotation angle longer than  $180^\circ$  to the expansion and exhaust strokes that have long stroke lengths<sup>(6)-(7)</sup>. This reduced the calculated maximum inertia force due to piston reciprocating motion by 23% compared to the 2006 model. In addition, changes were also made to the engine layout, and together with other enhancements this reduced the exciting force from the unit to the installation foundation by 40% compared to the 2006 model.

### 3.3.3. Countermeasures for knocking noise due to shaft thrust loads

Gear-meshing noise was reduced by using a helical gear for the cylindrical gear that synchronize the rotation rate of the eccentric shaft at half, and by supplying an oil jet to the meshing parts. Use of a helical gear generated thrust force in the axial directions of the crankshaft and the eccentric shaft due to the torque reaction force. On the other hand, the torque generated by the cylinder internal pressure and the inertia force around the piston fluctuates over the course of a single cycle, so the gear driving-driven relationship of each shaft alternates. This means that the thrust force alternately inverts, which gives rise to concerns about knocking noise. To address this thrust-knocking noise, a structure was used that applies hydraulic pressure to the crankshaft

in the direction to cancel the thrust force produced by the crankshaft weight and the helical gear, and to both ends of the eccentric shaft. In this way, the direction in which the shaft receives loads does not change even when the gear driving-driven relationship inverts (Fig. 10). Figure 11 shows the relationship between the engine assembly noise and the crankshaft thrust force generated by the hydraulic pressure. As a result, the 2011 model achieved the same engine assembly noise as the 2006 model, which together with the reduced intake and exhaust noise described above, achieved a 1-dB(A) reduction in the overall unit noise to 43 dB(A)/1m.

## 3.4. Technologies to Reduce Size

### 3.4.1. Technologies to reduce the engine size

There were concerns that the following points would increase the EXlink engine size.

- Addition of a multiple linkage system and linkage driving gears

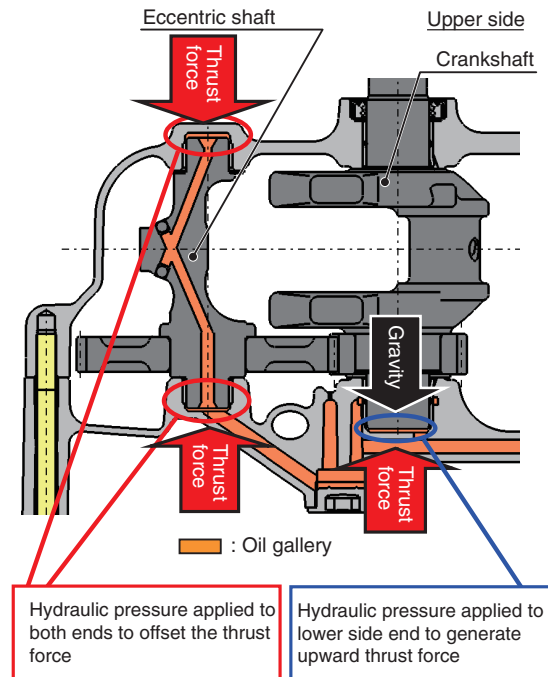


Fig. 10 Oil damping system

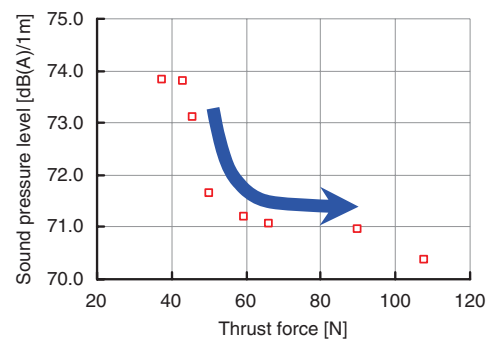


Fig. 11 Relative crankshaft thrust force (by hydraulic pressure) and engine noise

- An apparent super-long stroke due to extension of the expansion ratio

Formerly, the cylinder barrel and the oil tank were separately tightened to the oil pan. However, in order to house this engine in a compact chassis, the structure was modified so that all these parts are jointly tightened, and the crank case external dimensions were kept the same as in the previous model (Fig. 12).

### 3.4.2. Technologies to reduce the unit size

The engine cylinder layout was changed from diagonal to sideways as viewed from above. This made the unit width dimension the sum of the engine crank case thickness, the sound insulator thickness, and the clearance. The sound insulator was reviewed to achieve the target unit width of 300 mm or less.

The previous engine room sound insulator consisted of a cut piece of soft urethane foam with a constant thickness. Large external dimensions were set to secure the necessary

clearance between the sound insulator and the engine, and holes were opened in the sound insulator in locations where sufficient clearance could not be secured.

Therefore, the materials were changed to a molded felt insulator with a compression molded surface. This secured the necessary clearance in accordance with the unevenness of the engine, and enabled minimizing the holes from which sound can escape, which simultaneously achieved both high quietness and compactness (Fig. 13).

The 2006 model had a unit width of 380 mm excluding protrusions. This was reduced by 82 mm to 298 mm for the 2011 model.

In addition, the intake system parts layout was also reviewed. As a result, the unit height excluding protrusions was reduced by 130 mm from 880 mm to 750 mm, and the volume was also reduced by 33% compared to the 2006 model.

The reduced unit surface area enabled reducing the number of large-sized parts such as panels.

Reducing the unit size increased the chassis stiffness, which enabled elimination of the left side frame and reduction of the number of fastening parts. Ultimately, the number of large-sized pressed parts was reduced from ten to eight, and the overall unit weight was reduced by 11 kg (Fig. 14).

The compact unit size and reduced weight enhanced installation performance, and also provided merits such as reduced heat radiation from the outer panels, shorter wiring inside the unit, and the ability to load more units during transport.

The maintenance directions and the wiring and piping directions during installation were gathered on both sides of the unit. This enabled a reduction in the width of the required space for installation from 950 mm to 650 mm, including the maintenance space, achieving considerable instability.

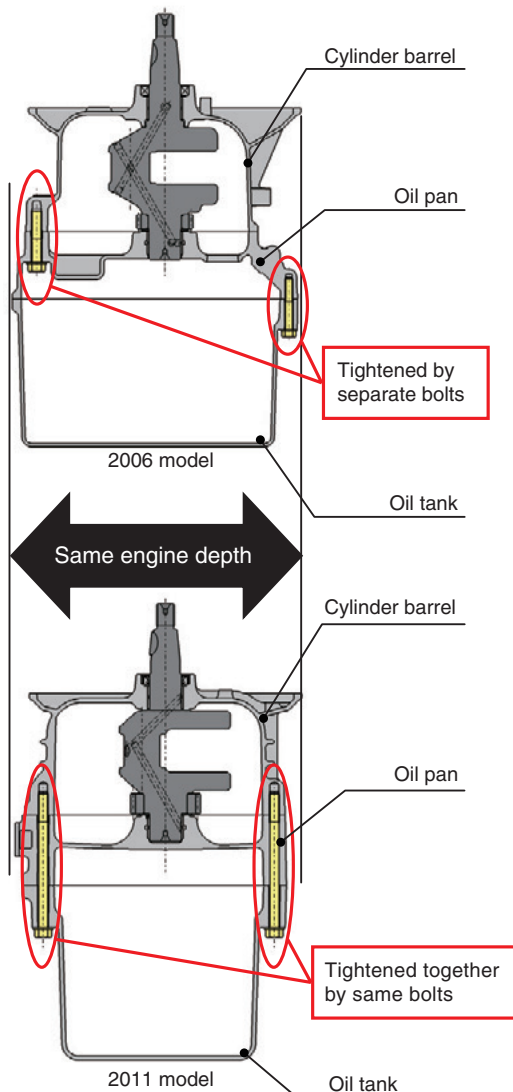


Fig. 12 Engine case tightening structure

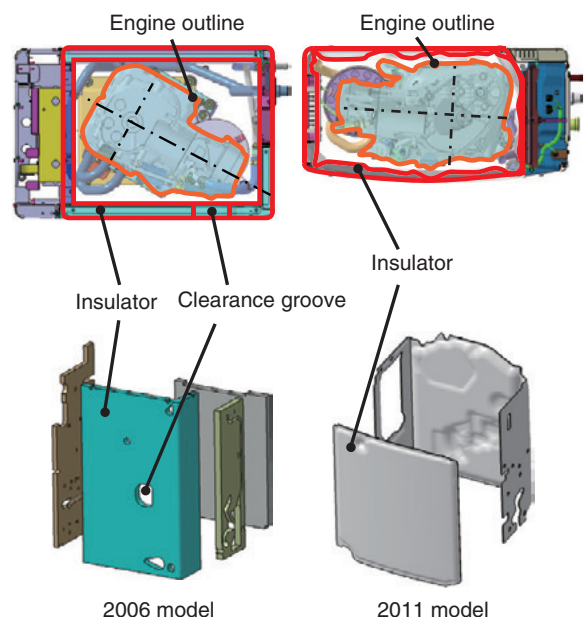


Fig. 13 Comparison of insulator structure

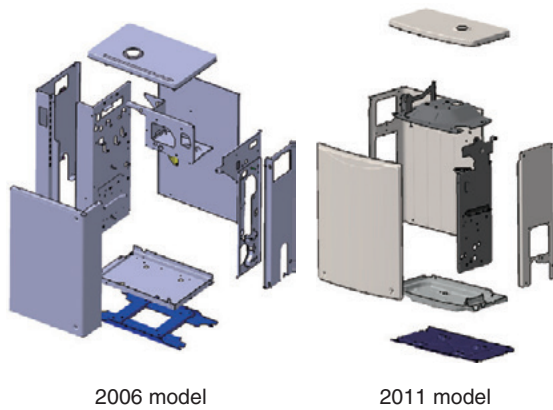


Fig. 14 Comparison of frame and panel structure

#### 4. Conclusion

A micro combined heat and power generation (CHP) unit that uses the EXlink engine was developed. The external view (Fig. 15), product specifications (Table 2) and achieved values are shown below.

- (1) A micro CHP unit equipped with EXlink was developed, and achieved an increase in power generating efficiency of 3.8 points to 26.3% (LHV) and an increase in heat recovery efficiency of 2.7 points to 65.7% (LHV) compared to the 2006 model. The overall efficiency increased by 6.5 points to 92% (LHV), and it was shown that EXlink is useful for enhancing the environmental performance of a micro gas engine CHP system.
- (2) Enhancement of the engine packaging method achieved a smaller chassis size. Specifically, the width was reduced by 82 mm to 298 mm, the height was reduced by 130 mm to 750 mm, and the volume was reduced by 33%, compared to the 2006 model. This realized enhanced installation performance.
- (3) With the use of EXlink, the intake and exhaust systems were also reviewed and gear noise was reduced, which achieved a noise value of 43 dB(A)/1m.



Fig. 15 External view of 2011 model

Table 2 2011 model specifications

|                    |             |                                         |
|--------------------|-------------|-----------------------------------------|
| Output             | Electricity | 1 $\phi$ 3W 202V/101V 50/60Hz<br>1.0 kW |
|                    | Heat        | 2.5 kW                                  |
| Power supply       |             | Grid connection by inverter             |
| Heat recovery      |             | 75°C hot water                          |
| Efficiencies (LHV) | Electricity | 26.3%                                   |
|                    | Heat        | 65.7%                                   |
| Dimensions (LxWxH) |             | 580 mm x 298 mm x 750 mm                |
| Mass               |             | 74 kg                                   |
| Noise              |             | 43 dB(A)/1m (JIS B 8627-1)              |
| Starting system    |             | Starter generator                       |

- (4) The compact size enabled integration of large-sized frame parts, and achieved a weight reduction of 11 kg compared to the 2006 model.

#### References

- (1) Takaishi, T., Ara, M., Togawa, K.: Introduction of Micro Combined Heat and Power Generation Unit for Residential Use, Honda R&D Technical Review, Vol. 15, No. 1, p. 53-58
- (2) 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
- (3) Koga, H., Watanabe, S.: Research on Extended Expansion General Purpose Engine (Part 2) –Heat Release and Brake Performance–, Honda R&D Technical Review, Vol. 19, No. 1, p. 53-59
- (4) Watanabe, S., Koga, H., Kono, S.: Research on Extended Expansion General-Purpose Engine Theoretical Analysis of Multiple Linkage System and Improvement of Thermal Efficiency, SAE paper, 2006-32-0101 (2006)
- (5) Koga, H., Watanabe, S.: Research on Extended Expansion General-Purpose Engine –Heat Release and Friction–, SAE paper, 2007-32-0003 (2007)
- (6) Naoe, G., Watanabe, S.: Research on Extended Expansion General-Purpose Engine –Characteristic of Vibration–, SAE paper, 2008-32-0012 (2008)
- (7) Watanabe, S., Naoe, G., Sato, Y.: Research on Extended Expansion General-Purpose Engine –A Numerical Approach to Reduce Vibration–, SAE paper, 2009-01-1006 (2009)
- (8) Naoe, G., Watanabe, S.: Research on Extended Expansion General-Purpose Engine –Noise Characteristics Caused by Multiple Linkage System and Reduction of the Noise–, SAE paper, 2009-32-0042 (2009)
- (9) Kono, S., Koga, H., Watanabe, S.: Research on Extended Expansion General-Purpose Engine –Efficiency Enhancement by Natural Gas Operation–, SAE paper, 2010-32-0007 (2010)

- (10) Ueno, M., Eguchi, H., Enomoto, T., Ogawa, M.:  
Study of Power Generation Loss Decrease in Small  
Gas Engine Cogeneration, SAE Paper, 2008-32-0044  
(2008)

■ Author ■



Yoshiharu TAKITA



Shohei KONO



Hideaki TAKADA



Kunihiro KITSU



Atsumu NAOI



Koichi TSUNO

# 2011 Odyssey Fuel Economy Improvement

Ryan CHAPMAN\*

Paul DeHART\*

Patrick SHAFER\*

Spyridon BLATSEAS\*

Chase LIN\*

Chris HARTWIG\*

Dennis PLANK\*

Evan MARTIN\*

Scott PAUL\*

Brandon OLLANKETO\*

## ABSTRACT

The 2011 Odyssey development team targeted achieving the “Best Family Vehicle” in terms of functionality, safety, and efficiency. In the area of efficiency, the overall goal was set to achieve “best-in-class” fuel economy. This goal was achieved through a total vehicle approach that involved efficiency gains in the following areas: engine operation, engine accessories and accessory belt drive system, chassis running resistance, and aerodynamics. In addition to hardware efficiency gains, aggressive weight reduction activities were conducted across all vehicle areas that contributed to the overall fuel economy improvement while meeting the functionality and performance targets.

## 1. Introduction

For the 2011 Odyssey, the development team was tasked with increasing fuel economy while maintaining and refining the minivan characteristics that consistently made it a top seller. During the planning stages, the need to increase fuel economy and maintain Odyssey’s marketability was apparent as fuel prices increased to record levels and consumers began seeking smaller vehicles. In addition, the Odyssey’s improved fuel economy was necessary to help meet increasing CAFE requirements. To maintain overall marketability and achieve CAFE, the development team

targeted “best-in-class” fuel economy while providing minivan functionality with V6 passenger car level fuel economy.

As shown in Fig. 1, label fuel economy targets were set at 18/27 miles per gallon on 5-speed automatic transmission (5AT) models (LX~EX-L) and 19/28 miles per gallon on 6-speed automatic transmission (6AT) models (Touring and Touring Elite). Achieving the label and CAFE fuel economy targets required a total vehicle effort that encompassed the engine, transmission, chassis, and body-related areas as shown in Fig. 2.

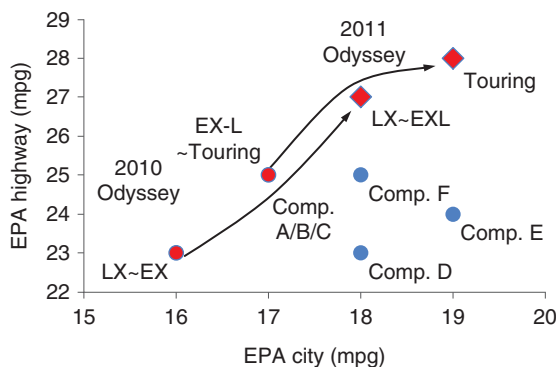


Fig. 1 Fuel economy competitiveness – EPA city versus EPA highway (mpg)

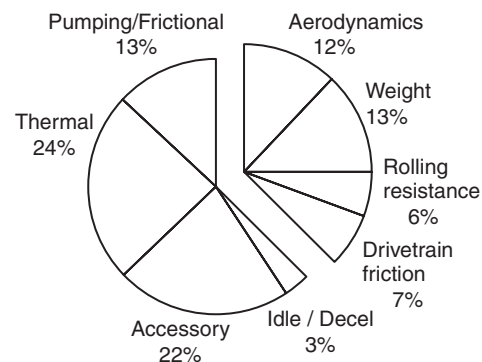


Fig. 2 Fuel economy improvement breakdown – Touring model highway mode

## 2. Engine

The power train development focused on “Smart Efficiency.” Based on this concept, the targets were set for “best-in-class” fuel economy and “best drivability.” Best drivability was retained by increasing power and mid-range torque. Fuel economy improvements started with applying the 3.5L V6 engine with Variable Cylinder Management (VCM) on all grades<sup>(1)</sup>. Engine improvements were made in the following areas: engine pumping loss reduction, internal friction reduction, thermal efficiency, accessory load reduction, and idle/deceleration losses.

### 2.1. Pumping Loss Reduction

To minimize pumping losses, the Odyssey uses Honda’s VCM technology, which allows for dynamic downsizing by deactivating two or three cylinders to allow the engine to run on three or four cylinders instead of six as shown in Fig. 3. As displacement decreases, throttling is reduced to achieve the same torque, which reduces pumping losses.

The VCM operation area was expanded through enhancements to Honda’s Active Control Engine Mount (ACM)<sup>(2)</sup> and Active Noise Cancellation systems (ANC)<sup>(3)</sup>.

When the engine operates in 3- and 4-cylinder VCM modes, the engine mount system must inhibit vibration transfer to the chassis in addition to normal 6-cylinder mode. The ACM system compensates for operation on reduced cylinders by adding movement back into the engine to create a balance similar to 6-cylinder mode. The engine mount displacement has been increased by as much as 20%, which allows for higher torque operation in 3- and 4-cylinder modes.

For the 2011 Odyssey, the subwoofer size was increased and the low frequency performance (gain and phase) of the ANC microphone improved as shown in Fig. 4. Making these changes improved cancellation of the low frequency noise and enabled expansion of 3- and 4-cylinder VCM operation at lower engine speeds.

In addition to producing more torque and expanding low engine speed operation in 3- and 4-cylinder modes, engine losses and vehicle running resistance were reduced. These changes reduce the gross engine power required to move the vehicle. Due to the reduced power requirements, the 2011 Odyssey is able to operate in the 3- and 4-cylinder VCM modes approximately 50% more often. Overall, VCM (as enabled by all the other reductions in losses) is the single

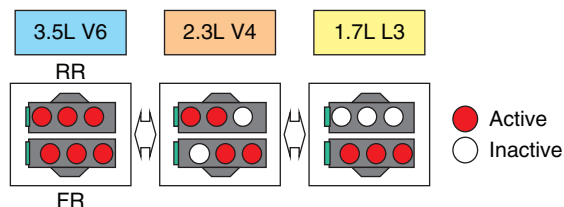


Fig. 3 VCM system operation: 3-4-6 cylinder modes

largest contributor to the Odyssey’s improved fuel economy and key to balancing performance and fuel economy.

Another major improvement for the Odyssey’s reduction in pumping losses is an increase in Exhaust Gas Recirculation (EGR). While EGR is often considered an emissions control device, it can have a considerable effect on fuel economy. By allowing exhaust gas recirculation back into the cylinder, pumping losses can be reduced in two ways. First, since the exhaust gas bypasses the throttle and flows directly into the intake manifold from the high pressure exhaust side, the flow is not throttled and pumping is not required. Second, in order to maintain torque, more fresh air must be admitted into the cylinder to compensate for air that was displaced by the exhaust gas. This means that the throttling must be reduced, which reduces pumping losses.

For the new Odyssey, the mass ratio of EGR to fresh air was increased by up to 10%, which reduced pumping losses and improved fuel economy. The primary enabler for this increase was the application of Dual-Finite-Electrode spark plugs (Fig. 5). The dual finite electrodes guarantee a strong, clean spark, allowing an increased EGR ratio without introducing combustion stability concerns.

Overall, efforts taken to reduce pumping losses through enhancements to the VCM (engine, transmission, ACM, ANC) and EGR systems netted over 7% of the Odyssey’s highway fuel economy improvement.

### 2.2. Internal Friction Reductions

Reducing engine internal friction was another strategy for improving fuel economy. This was accomplished by

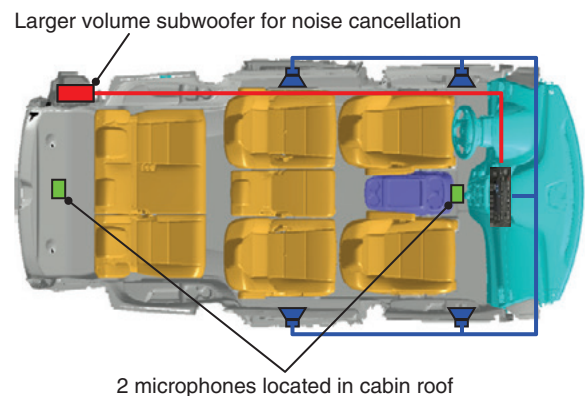


Fig. 4 Active Noise Cancellation (ANC) system

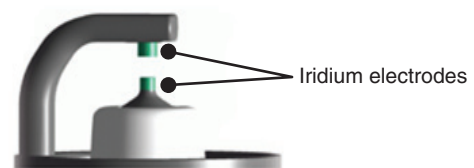


Fig. 5 Dual finite electrode spark plug

reducing piston sliding force and using 0W-20 engine oil.

To reduce piston sliding force, the two main piston sliding interfaces were analyzed and improved. These interfaces are the piston ring-to-cylinder liner and the piston skirt-to-cylinder liner.

The sliding friction in the piston ring-to-cylinder liner interface was reduced by changing the piston ring coating and the cylinder liner honing process. To achieve a smooth sliding surface, an ion plating hardening treatment was applied to the piston oil ring to help maintain the ring's smoothness and reduce wear. To reduce cylinder wall friction, a secondary honing process was added to remove any high peaks from the initial honing process, which creates "plateaus" with valleys in-between (Fig. 6). The plateaus maintain a smooth sliding surface while the valleys hold oil to lubricate the rings as they slide by.

Figure 7 shows modifications made to the piston skirt-to-cylinder wall interface for reduced friction by removing small "dots" of molybdenum coating from the piston's thrust surfaces on the skirt. These "dots" hold oil, helping to distribute the oil up and down the cylinder wall to further lower the friction between the molybdenum coating and cylinder wall.

Changing the piston sliding friction reduced engine friction by 4% and improved fuel economy by over 5% in both city and highway modes. To reduce friction under cold conditions, the engine oil was switched from 5W-20 to 0W-20 viscosity. This change contributed to over 10% of the engine's fuel economy improvement in the city mode.

### 2.3. Thermal Loss Reduction

In addition to reducing pumping losses and internal friction, changes were made to reduce thermal losses in the

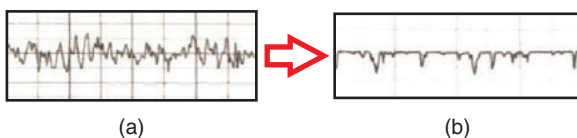


Fig. 6 Cylinder honing showing normal hone (a) and plateau hone (b)



Fig. 7 "Moly-Dot" piston skirt coating

engine. As shown in Fig. 8, the first change addressed the air entering the engine by applying a cold air intake system, which reduced intake air temperature by 21°C during the highway fuel economy test mode. Application of the cold air intake system improved highway mode fuel economy by 16%.

Second, EGR lowered combustion temperatures by displacing some of the oxygen in the cylinder with inert exhaust gas and netted an 8% increase in highway fuel economy. Overall, both of these changes advanced ignition timing, which converts more of the combustion energy into moving the piston versus escaping as heat through the exhaust gas.

### 2.4. Accessory Drive System Reduction

The accessory drive system was another focus area for reducing engine pumping losses. Reducing losses in each of the engine's accessories allowed for an overall reduction in accessory belt tension. As shown in Fig. 9, the first and most fundamental change was made to the alternating current generator (ACG) by adding an Override ACG Decoupler (OAD). This device replaces the standard ACG drive pulley with one that incorporates a spring to absorb engine torque fluctuations, which yields a smoother operation and more consistent speed. As a result, the overall tension of the accessory drive belt was reduced by approximately 50%. Together, the OAD and reduced belt tension yielded a 9% improvement in highway mode fuel economy.

The largest benefit to the accessory drive system reductions came from changes made to the power steering pump. While Electric Power Steering (EPS) was investigated, applying a Variable Displacement power steering Pump (VDP) emerged as a solution<sup>(4)</sup>. The VDP system came at no additional cost from the previous Odyssey's power steering pump and provided most of the fuel economy benefit of an EPS system. This VDP contains an internal cam ring around the central vane pump. This cam ring is balanced by a spring and the pump pressure. When flow is reduced, the internal pressure shifts the cam

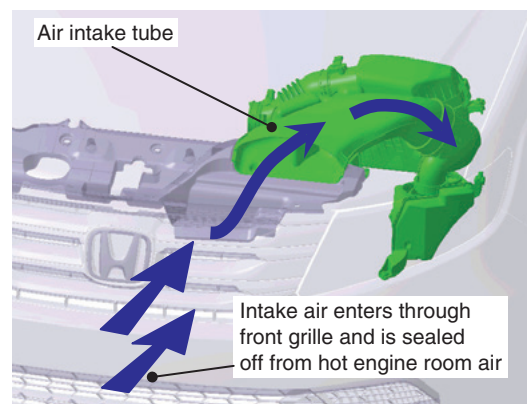


Fig. 8 Cold air intake system

ring closer to the centerline of the pump, thereby reducing the fluid displaced. By reducing the pump displacement, pump torque is decreased, thereby lowering engine parasitic losses. Overall, the VDP accounted for a 13% highway mode fuel economy improvement. As an added weight reduction benefit, a power steering cooler was not required on the towing package due to lower power steering fluid temperatures.

The final accessory on the belt drive system is the Air Conditioning (AC) Compressor. While the compressor remained essentially common to the outgoing model, changes were made in the AC control logic to reduce load on the compressor and hence the engine. This control system monitors a variety of parameters, such as humidity, and controls the evaporator temperature, air recirculation, and condenser fan speed for optimal performance and reduced compressor run time. This system yields fuel economy benefits of up to 5% in real-world driving.

### 2.5. Idle and Deceleration Reduction

The final area addressed in the engine was idle and deceleration losses. When the engine is idling and the vehicle is not moving, fuel economy is zero mpg. Conversely, when decelerating, if the vehicle is moving and fuel is shut-off to the cylinders, fuel economy is infinite. By reducing the idle speed, the impact of zero mpg operation

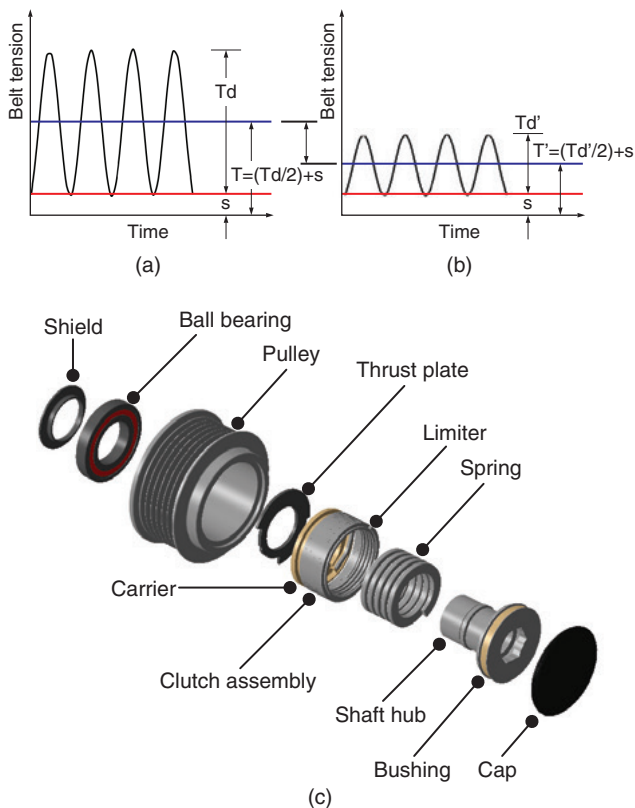


Fig. 9 Accessory belt tension and fluctuation shown without OAD (a), with OAD (b), and OAD components (c)

was reduced, and optimizing the Deceleration Fuel Cut system, increased the time of operation at infinite mpg. During deceleration fuel cut, the Odyssey further applies a regenerative charging and cooling system to make use of the energy produced during deceleration to charge the battery and cool the engine. These changes netted a 9% increase in city mode fuel economy and a 3% increase in highway mode fuel economy.

## 3. Transmission

In addition to the already optimized 5AT, the application of the 6AT further improved fuel economy in multiple aspects<sup>(5)</sup>. The 6AT improves city mode fuel economy by over 4% and highway mode fuel economy by over 1% through reduced drivetrain losses.

### 3.1. Close Gear Ratio

Figure 10 compares the 5AT and 6AT gear ratios. Applying 6AT allowed for closer ratio gaps between gears, which resulted in better use of the engine's power during acceleration in city driving by keeping the engine speed in an optimum operating range. As a result, 3-cylinder VCM operation in the city fuel economy mode was dramatically increased.

For the 6AT, the 6<sup>th</sup> gear ratio is higher than the 5AT's 5<sup>th</sup> gear; however, due to increased mid range torque, there is ample driving force during the highway mode to allow for 3-cylinder VCM operation while lowering the engine speed to reduce overall friction.

### 3.2. Multi-Disk Lock-Up Converter

Figure 11 shows the higher capacity Multi-Disk Lock-Up Converter (LC), which allowed for expanding the LC feedback control area. This expansion reduced unnecessary torque converter (TC) slip; which was not possible with the 5AT single plate LC. The expanded LC area reduced wasted

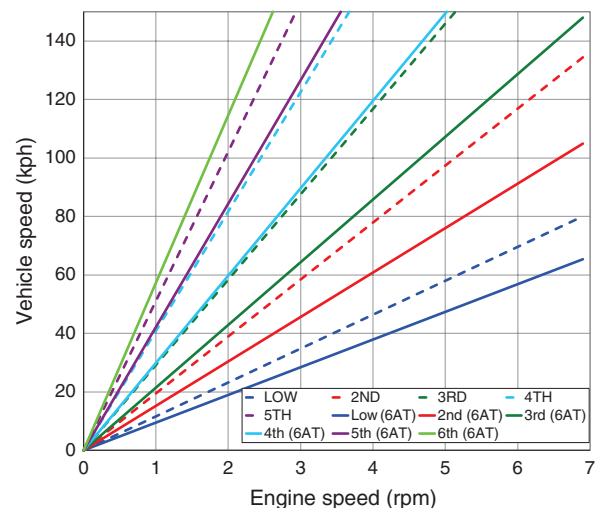


Fig. 10 5AT and 6AT gear ratio comparison

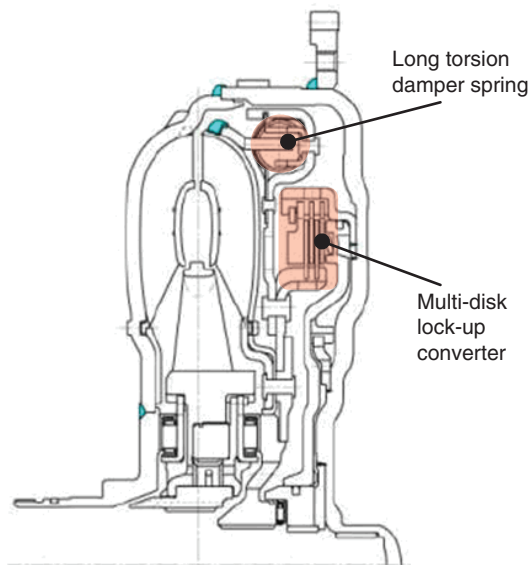


Fig. 11 Multi-disk lock-up torque converter

energy and provided the majority of the city fuel economy increase and enhanced dynamic driving feel.

### 3.3. Long Torsion Damper Spring

Figure 11 shows the special torsion damper springs inside the TC that was selected to increase vibration absorption during VCM operation. This allowed for further expansion of VCM areas without compromising overall vehicle quality in terms of noise, vibration, and harshness (NVH) and improves fuel economy through reduced pumping losses.

### 3.4. Low-Viscosity Automatic Transmission Fluid

Applying low-viscosity automatic transmission fluid (ATF) reduced fluid friction during the critical warm-up phase, which in turn improved overall efficiency.

### 3.5. Transmission Calibration

Transmission calibration focused on more aggressive transient tip-in control to decrease unnecessary TC slippage. Shiftmap tuning was also optimized to maximize VCM operation before and after shifts. Continuous fuel cut (F/C) during natural deceleration downshifts also contributed to overall fuel economy by increasing F/C time.

## 4. Aerodynamics

Aerodynamic drag ( $C_dA$ ) is a function of the vehicle's coefficient of drag ( $C_d$ ) and frontal area ( $A$ ) and has a direct impact on highway fuel economy. Reduced running resistance through reduced aerodynamic drag lessens the required engine torque to maintain highway speeds. The 2011 Odyssey's overall  $C_dA$  was reduced by 5% compared to the previous model and contributed to 12% of the Touring model's highway fuel economy benefit.

Enhancing the functionality and usability of the interior packaging increased the vehicle's frontal area, even with a lowered roof profile. The main contributor to the increase was reduced tumblehome to increase head clearance for the second row seat's wide mode position. Also, adding turn signals to the side mirrors on Touring grades increased frontal area. To overcome the frontal area increase, the entire vehicle exterior was considered for reducing the coefficient of drag ( $C_d$ ).

To improve engine room and underbody flow, air guide plates were added to increase the airflow efficiency to the cooling system. The cooling open area was then strategically minimized to reduce drag while maintaining cooling and heat resistance performance as shown in Fig. 12. To control the exit flow from the engine room, the front inner fender side wall was optimized by extending rearward.

Under the vehicle, the front lip spoiler and front inner fender strakes were designed to work in concert. As shown in Fig. 12, the lip spoiler and strake position and shape were optimized using computational fluid dynamics simulation. The airflow was diverted lower and toward the inside of the vehicle. This airflow change reduced the high pressure zone on the front tires. A rear undercover was added on the left side to reduce the drag of the rear fascia.

As shown in Fig. 13, the upper cabin and tail lights

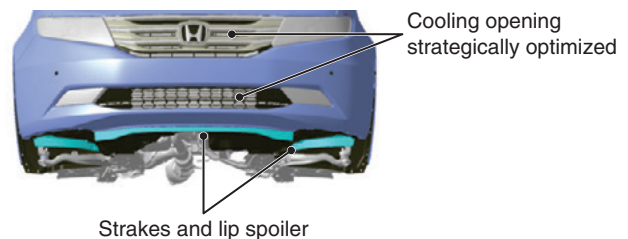


Fig. 12 Front aerodynamic components



Fig. 13 Wind tunnel development of rear flow separation

were shaped through extensive wind tunnel testing to control the airflow separation. The tailgate spoiler and side spoilers were shaped specifically for aerodynamic drag reduction. The Touring grade side sill garnish shape was styled to push air out and around the rear wheels.

The Touring grade's side mirror is larger than the base grades' side mirror due to adding an integrated side turn signal. Consequently, extensive wind tunnel testing was conducted to reduce drag induced by the mirror wake. Wind noise reduction was also completed in parallel during the testing.

Finally, the seals on the front and rear quarter windows were designed with a ramp shape to avoid separation and reduce wind noise.

## 5. Drivetrain Friction Reduction

Traditionally, chassis-related efforts to improve fuel economy performance were limited to reducing tire rolling resistance. While the contribution to overall vehicle fuel consumption among chassis parts is clearly dominated by tire energy loss, every part's influence was considered during vehicle development.

### 5.1. Bearings

As total vehicle coast-down testing has improved, the bearing drag reduction benefit to fuel economy has been quantified. In an effort to improve fuel efficiency wherever possible, an effort was made to reduce bearing drag, even though its effect on fuel economy is on the order of 0.01 mpg. Two key design factors influencing bearing drag are sealing against water and the required preload needed to achieve bearing stiffness. In the past, improvements for sealing and preload have directly resulted in a drag increase. For the 2011 Odyssey, the chassis design team targeted an 18% drag reduction without affecting sealing or stiffness.

The front wheel bearing drag was reduced by changing the bearing type and optimizing the seal lip geometry. The front bearing was changed from a GEN1 to GEN3 structure, which reduced the bearing preload by eliminating the variation from the outer race-knuckle interface and the inner race-hub interface. Optimizing the seal lip involved changing the lip material and modifying the seal geometry to reduce contact pressure while ensuring equivalent sealing performance. These changes reduced front bearing drag by an average of 24% and increased bearing mass by 9.7%.

The rear wheel bearing drag was reduced by modifying the bearing layout and optimizing the seal lips. The bearing layout was changed from a symmetric GEN3 layout to an asymmetric GEN3. This change reduced the bearing weight and increased the geometrical stiffness, which allowed the preload to be reduced while maintaining overall bearing stiffness. Reducing the preload reduced the rolling element contribution to drag. The seal lip optimization strategy

matched the front bearing approach and also reduced the overall bearing drag. These changes resulted in an average rear bearing drag reduction of 48% and a mass reduction of 6%.

From a total vehicle standpoint, wheel bearing drag was reduced by 37% with a nominal mass increase of only 2% and contributed to over 2% of the highway fuel economy increase.

### 5.2. Brake Drag Reduction

The 2011 Odyssey increased the brake package size from 16" to 17" to enhance brake feeling and performance without increasing mechanical drag. Several countermeasures were required to overcome the increase in effective radius. The target was achieved by changing pin A and B locations, applying V-springs to the pads, changing the pad retainer shape, and applying grease between the pad abutments and retainer.

Pin A and B positions were changed to better control the caliper body tilt angle. The 2010 Odyssey previously applied the tighter fit Pin A to the trailing side and the guide Pin B to the leading side. During the 2011 Odyssey testing, it was determined that due to the additional weight and center-of-gravity (CG) location of the caliper body, the pins needed to be swapped to better control caliper body position during retraction as shown in Fig. 14. It was also decided that the body position could benefit from increasing the guide Pin B bushing volume.

Figure 15 shows the pad v-springs that were added to separate the pads from the disk during off brake to reduce parasitic drag. The v-spring set force was determined by checking the post durability vehicle test brake pad sliding force to determine the minimum force required to separate the pads from the disk. However, due to the application of the v-springs, it was determined that the pad rattle noise rating exceeded the development target.

To offset the rattle noise caused by adding v-springs and to maintain a good drag level, it was necessary to change the pad retainer shape. The pad retainer was changed from a P-type to a C-type as shown in Fig. 16. The P-type retainer was initially applied due to its ability to aid in pad retraction; however, it was not able to meet the pad rattle

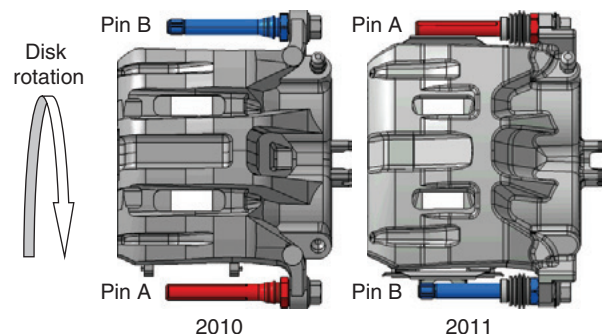


Fig. 14 2010 and 2011 Odyssey pin A and B locations

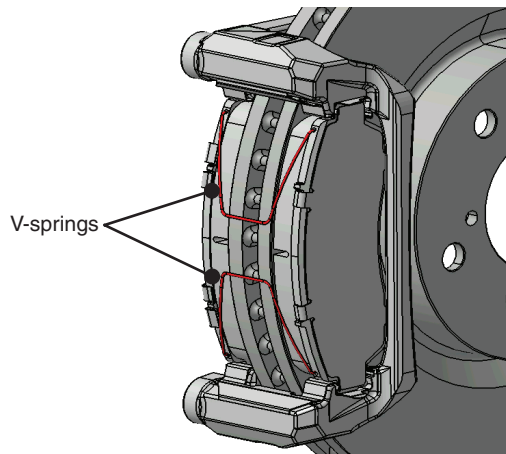


Fig. 15 2011 Odyssey V-springs

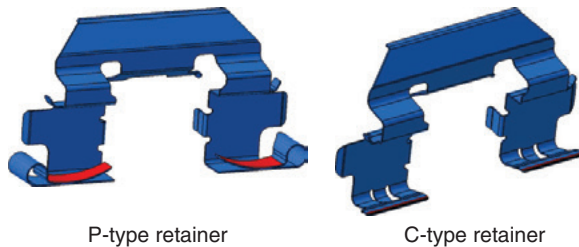


Fig. 16 2010 Odyssey P-type and 2011 C-type retainers

target. To meet the pad rattle target, the C-type retainer was implemented to maintain enough spring force on the pad to reduce pad rattle while reducing pad sliding force by approximately 32%. Pad sliding force was reduced by limiting the contact surface area between the retainer and pad interfaces. In addition, Molycoat M77 grease was applied between the pad and retainer interfaces to reduce the frictional forces and further reduce the pad sliding force.

After applying the aforementioned brake drag reduction items, the vehicle met the brake drag target. At 55 mph, the vehicle has an average 5% reduction in brake drag allowing enhanced brake feeling and performance without sacrificing fuel economy.

## 6. Rolling Resistance

The 2011 Odyssey uses 17" and 18" wheel fitments, which are one-inch larger over the previous model's wheel sizes. The 2010 Touring grade P235/60R17 tire was chosen as a reference tire for 2011 model tire development. The target was to reduce rolling resistance 6% versus the reference tire without degrading important performance areas, such as braking and snow performance.

For the 2011 Odyssey, the LX through EX-L grades applied P235/65R17 tires while the Touring grades applied P235/60R18 tires. An iterative approach was used to

develop each tire's cavity shape, tread pattern, compounds, and materials to optimize the tire performance.

Compared to the reference tire, each tire improved rolling resistance by 7 to 8%, which contributed to 6% of the Touring model's highway fuel economy increase. In addition, each tire developed for the 2011 Odyssey achieved all dry braking, wet braking, and subjective performance targets.

## 7. Weight Reduction

For US Environmental Protection Agency (EPA) Fuel Economy, vehicle weight is critical in determining vehicle running resistance and EPA test conditions. Vehicle weight is categorized into equivalent inertia weight ranks (IWR) based on vehicle curb weight. Furthermore, the certification inertia weight rank is set based on power train and the model grade comprising the majority of vehicle sales. The certification IWR for the 2010 Odyssey was based on the EX-L grade, which allowed all variations to fall within the 4750 IWR.

The 2011 Odyssey required a separate weight certification based on 5AT application on LX through EX-L grades and 6AT application on Touring grades. Achieving the targeted fuel economy required all grades to fall within the 4750 IWR. This was especially important on the high feature content Touring grades. Overall, the vehicle weight was reduced by 47 kg and the 4750 IWR was achieved for all certification models offsetting a 13% reduction in highway mode fuel economy.

### 7.1. Engine Weight Reduction

Engine weight was reduced by 5.5 kg while reducing cost, increasing horsepower, and contributing to improved fuel economy. The main design changes responsible for this weight reduction involved converting to lightweight materials, such as magnesium and plastic; optimizing material thickness and size for stamped, cast, and molded parts; and improving system efficiency.

Intake system weight reduced by 2 kg through structural optimization and system efficiency improvements while improving performance and fuel economy. The overall number of parts in the intake system was reduced by integrating parts between the air cleaner case and intake tube and between the resonator and cold air intake tube.

By applying a lightweight radiator package, the overall cooling system weight was reduced by 1.4 kg while increasing cooling performance. This was achieved by combining a new thin radiator core design with dual ring fans and changing the sub shroud material from steel to lightweight nylon.

### 7.2. White Body Weight Reduction

The 2011 Odyssey body is a conventional steel body with targeted use of advanced high strength steels (AHSS).

It is 12.5 kg lighter than its predecessor and has increased body rigidity and collision performance.

Each component of the body was reviewed during development with material grades and thicknesses chosen to meet the functional requirements within each body system. A collaborative approach between design and manufacturing groups was used to refine the white body design. Each structural change was evaluated through extensive use of CAE tools. Crash simulations, body rigidity, and spot weld durability simulations were conducted to help ensure structural changes met all of the functional requirements. Part thickness and grade changes were cross-checked by all simulation groups.

Another challenge was to enhance collision performance by reducing side impact intrusion and increasing roof strength. Both required structural changes from the previous design. A typical approach would be to increase the size of the body sections to increase strength; however, this can reduce passenger space. The team chose not to increase the overall section sizes and looked for opportunities to use AHSS grades and enhance the section efficiency.

The Rear Roof Rail is one example of section optimization. The rail spans across the vehicle from D-Pillar to D-Pillar and forms the top of the tailgate ring. This part has functional requirements for body rigidity and safety performance as well as providing the structural base for the Tailgate Hinges. As shown in Fig. 17, the previous design rotated the tailgate into the section during opening. For the 2011 Odyssey, the Tailgate Hinge design rotates the tailgate above the roof as shown in Fig. 18. This allowed the Rear Roof Rail section to be smaller and more efficient. The third row seat packaging was improved and the more compact design reduced the weight by over 0.35 kg.

The enhanced strength of AHSS allowed for body system construction changes that further reduced weight. As shown in Fig. 19, 59% of the Odyssey's white body structure is composed of steels with tensile strength over 340 MPa, which is a 68.5% increase in AHSS usage. Building upon Honda's previously-developed models, 780 MPa, 980 MPa, and 1500 MPa AHSS grades were used

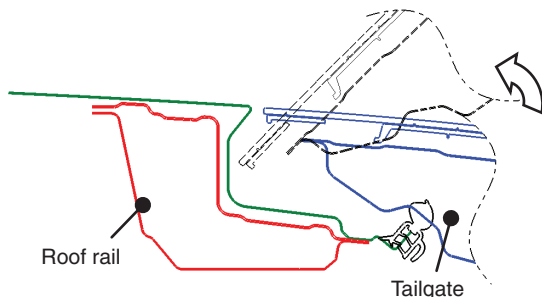


Fig. 17 2010 Odyssey rear roof rail section – tailgate closed and open positions

for the first time on the Odyssey platform. Examples of AHSS usage to reduce weight on the 2011 Odyssey include the front floor frame (-1.3 kg), second row seat mounting crossmember (-0.86 kg), and the roof side rail (-0.75 kg).

### 7.3. Interior Weight Reduction

The majority of interior weight can be attributed to the seating system. Seats in the 2010 Odyssey EX-L accounted for approximately 60% of the interior parts' total weight. Customers' desire for versatile eight-passenger seating arrangements along with higher seat performance targets can result in higher seat design complexity and additional weight. The challenge for the 2011 Odyssey development was to find ways to reduce the seat system weight, enhance the seating arrangement flexibility, and exceed existing functionality. With this in mind, the focus was to optimize the previous seat design and maintain a high degree of functionality and performance.

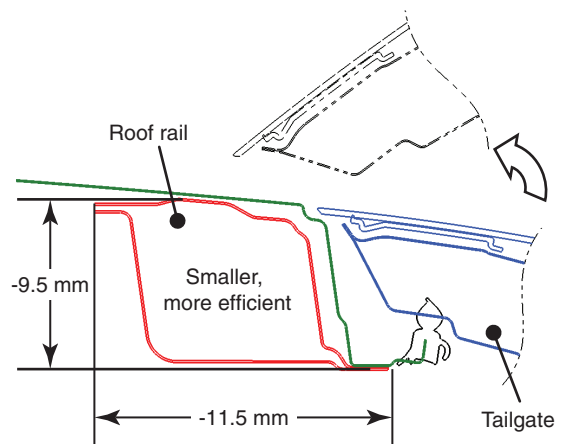


Fig. 18 2011 Odyssey rear roof rail section – tailgate closed and open positions

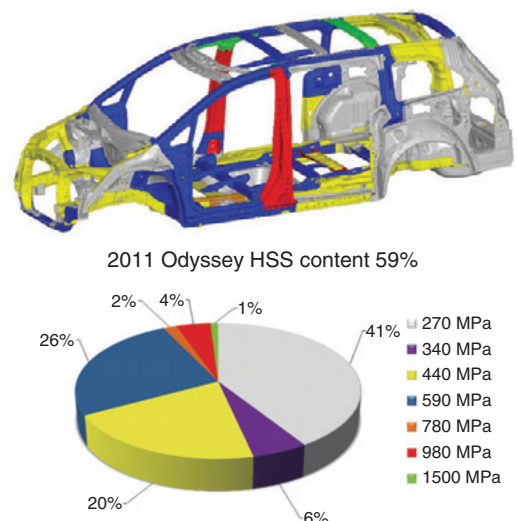


Fig. 19 2011 Odyssey high strength steel usage

Table 1 Seat assembly weight comparison

| Seat assembly weight comparison (EX-L)<br>(Excludes level-1 plastics and bolts) |        |        |                 |
|---------------------------------------------------------------------------------|--------|--------|-----------------|
| EX-L                                                                            | 10M    | 11M    | Difference (kg) |
| FR RH                                                                           | 35.85  | 28.58  | -7.27           |
| FR LH                                                                           | 33.65  | 30.39  | -3.26           |
| Mid RH                                                                          | 28.50  | 25.10  | -3.40           |
| Mid LH                                                                          | 27.00  | 25.10  | -1.90           |
| Mid CTR                                                                         | 15.10  | 18.60  | 3.50            |
| RR 40%                                                                          | 24.40  | 22.68  | -1.72           |
| RR 60%                                                                          | 36.60  | 31.98  | -4.62           |
| Total                                                                           | 201.10 | 182.43 | -18.67          |

As shown in Table 1, regarding the complete seat set, the seat weight was reduced by approximately 18.7 kg compared to the previous design. Considering the addition of new features and regulation changes, the weight savings is actually higher when compared to an equivalent design specification. For example, the 2011 Odyssey mid center seat adds a comfort slide adjustment, ISOFIX child seat attachment, and a dynamic damper, resulting in a weight increase of 3.5 kg. Mid row outboard seats added 1.66 kg due to dynamic dampers. The rear seat weight increased by 3.22 kg due to upgrading the outer headrests to comply with FMVSS 202a, adding an ISOFIX location to the 40% seat, and adding new cargo load floor flaps.

In the front row seating, weight was reduced by a total of 6.53 kg. This does not include a 4 kg weight transfer caused by replacing the passenger seat-mounted folding table with a body-mounted removable center console. The majority of the front seat weight reduction, which is approximately 5.05 kg, was achieved by replacing the complex welded stamping seat riser assembly with a more efficient design. The new seat mounting concept consisted of a body-integrated rear mount structure and a taller 3-piece stamping for the device front mounting foot. These changes eliminated the need for a separate seat riser altogether (Fig. 20). An additional 0.88 kg savings was achieved by applying the latest generation passenger seat occupant detection system. The remaining amount of front row seat weight reduction was obtained by applying the latest generation anti-whiplash front seat back frame and an 8-way/4-way power device.

In the mid row, the outboard seat combined weight was reduced by 5.3 kg. The majority of this mid outboard weight savings resulted from applying a new low-effort slide adjuster device and an optimized recliner and walk-in mechanism. For the 2011 Odyssey, a dynamic damper was added to each mid outboard seat; however, approximately 95% of this weight increase was offset by deleting the inboard arm rests and their mounting brackets. To compensate the inner armrest feature loss, the larger 2011 mid center seat applied an integrated arm rest and cup-holder, which is accessible when the center seat back is

folded down.

In the rear row seating, a total weight savings of 6.34 kg was achieved by optimizing the seat frame and mounting structure and reducing the lock quantity of the 60% foot comp to one using latest generation locks (Fig. 21).

For the 2011 Odyssey rear seat, the pivot mounting structure design was changed to reduce weight and simplify factory assembly (Fig. 22). For the center pivots, a lightweight welded steel stamping design was applied to replace the previous generation's bolt-on cast iron structure. For the outboard spring-loaded pivots, a lighter level-2 bolt-on design was applied, eliminating the need for the factory to install it to the body prior to the seat installation. Optimizing the rear seat pivot mounting structures reduced the rear seat weight by approximately 5.54 kg.

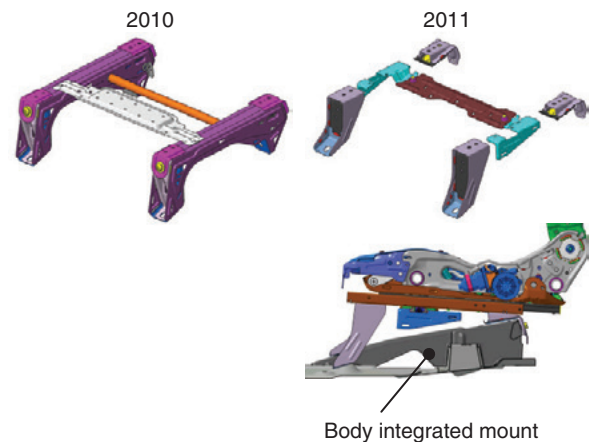
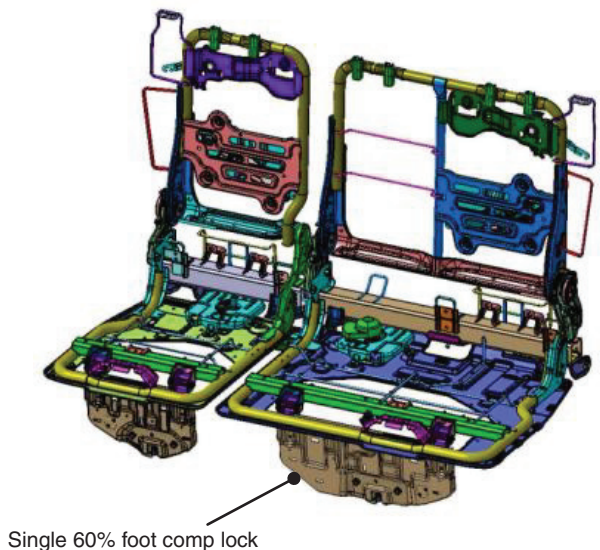


Fig. 20 2010 and 2011 front seat mounting system

Fig. 21 2011 Odyssey 3<sup>rd</sup> row seat structure and 60% foot comp single lock

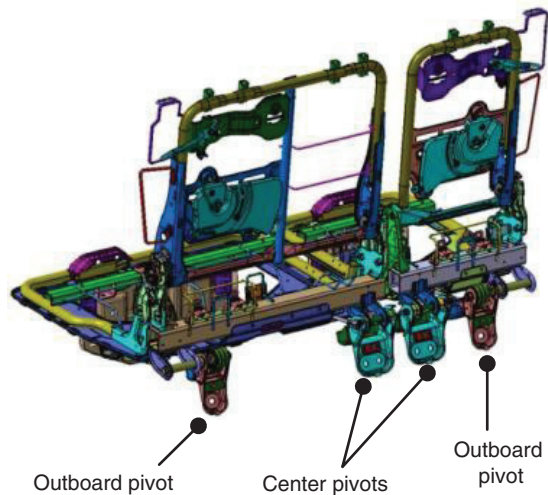


Fig. 22 2011 Odyssey rear seat integrated mounting pivot

## 8. Conclusion

The 2011 Odyssey's overall fuel economy achievement involved a total vehicle effort on a part-by-part and system-by-system basis that encompassed the engine, transmission, chassis, and body areas. Fuel economy improvements started with applying the 3.5L V6 engine with VCM to all grades. Changes to the engine design and calibration involved reduced pumping losses, increased EGR usage, air intake temperature management, reduced internal friction, reduced accessory drive losses, and reduced idle/deceleration losses. For the transmission, internal friction was reduced and gear ratios optimized for VCM usage. In addition, 6AT was applied to the Touring models to enhance drivability and further increase fuel economy. Changes to the chassis focused on reduced running resistance through tires, brakes, and bearings. Changes to the vehicle body focused on aerodynamics and weight reduction with increased structural rigidity and enhanced safety performance. Total vehicle weight reduction was 47 kg.

Fuel economy on the 2011 Odyssey improved by +2/+4 mpg on the LX~EX models, +1/+2 mpg on the EX-L model, and +2/+3 mpg on the Touring models in EPA city and highway modes. Through these fuel economy improvements, the 2011 Odyssey achieved the 18/27 mpg label target on the LX~EX-L models and 19/28 mpg label target on the Touring models. In addition, vehicle development achieved the CAFE strategy and maintained Odyssey's overall marketability with "best-in-class" fuel economy by a margin of 3 to 4 mpg over competitors.

## References

- (1) Segawa, M., Kumagai, K., Sato, R., Tamura, Y., Fujiwara, M.: Development of V6 Engine with New Variable Cylinder Management, Honda R&D Technical

Review, Vol. 20, No. 1, p. 35-42

- (2) Matsuoka, H., Mikasa, T., Nemoto, H.: Development of Active Control Engine Mount, Honda R&D Technical Review, Vol. 15, No. 2, p. 209-214
- (3) Inoue, T., Takahashi, A., Minowa, S., Sano, H.: Development of Active Noise Control System for Engine Booming Noise, Honda R&D Technical Review, Vol. 15, No. 2, p. 201-208
- (4) Yu, J., Haskell, J.: Fuel-Efficient and Low-Noise Steering Systems in Large and Heavy Vehicles using Variable Displacement Pump Technology, Honda R&D Technical Review, Vol. 22, No. 2, p. 182-188
- (5) Takei, H., Ishikawa, Y., Saito, Y., Tsubata, Y.: Fine Control System for Multi-plate Lock-up Clutch, Honda R&D Technical Review, Vol. 21, No. 2, p. 155-161

## ■ Author ■



Ryan CHAPMAN



Paul DeHART



Patrick SHAFER



Spyridon BLATSEAS



Chase LIN



Chris HARTWIG



Dennis PLANK



Evan MARTIN



Scott PAUL



Brandon OLLANKETO

# Development of Intake Air Pressure Sensor-less Fuel Injection System for Small Motorcycles

Katsuhiro UTSUGI\*

Yuki TAKANO\*

Kazuhiko SAKAGUCHI\*

Kenta ONISHI\*

---

## ABSTRACT

An electronically controlled fuel injection system for small motorcycles was developed without an intake air pressure sensor to reduce costs. In a fuel injection system, the intake air pressure sensor performs the three functions of measuring the amount of intake air in minute throttle-opening range, measuring the atmospheric pressure, and distinguishing the engine stroke. This study replaced these functions with a new learning control system that applies O<sub>2</sub> feedback control, and engine-stroke distinction control based on detection of crankshaft rotation speed fluctuation. This new learning control system defines the operational condition compensation coefficient  $K_{bu}$ , which covers operational conditions such as atmospheric pressure and conditional variations among completed engines, and uses this coefficient in the fuel injection quantity correction function. This enabled proper fuel injection quantity control without depending on intake air pressure sensor functions. In addition, storing  $K_{bu}$  in an Engine Control Unit (ECU) enabled proper air fuel ratio correction even when O<sub>2</sub> feedback control is inactive, such as when the engine is cold immediately after starting or during transient operation. The engine-stroke distinction function enabled stroke distinction equal to that of the conventional system by changing the method from measurement of the difference in intake air pressure between engine strokes to detection of crankshaft rotation speed fluctuation between engine strokes. This technology realized a simplified system that maintains the same performance for exhaust emissions and fuel economy without an intake air pressure sensor.

---

## 1. Introduction

As concern about environmental issues grows worldwide, demands are increasing to reduce exhaust emissions. Motorcycles are no exception to this, and the shift from carburetor-based fuel supply systems to electronically controlled fuel injection systems (hereafter “current FI systems”) is progressing as an effective means of reducing motorcycle exhaust emissions. Compared to carburetors, current FI systems can more accurately control the air fuel ratio in response to changes in the intake air temperature and atmospheric pressure. However, this type of control requires multiple sensors to measure factors such as the amount of intake air, intake air temperature, atmospheric pressure, cam phase, crankshaft phase, and concentration of residual oxygen in the exhaust gas. As a result, there is the issue that the system is more complex and expensive than a carburetor. Simple and low-cost systems are more strongly demanded in particular for small motorcycles compared to cars and large motorcycles.

Various studies thus far have sought to reduce costs by eliminating the secondary air supply system<sup>(1)</sup>, combining various sensors with the throttle body<sup>(2)</sup>, realizing heater-less compact O<sub>2</sub> sensors<sup>(3)</sup>, and reducing injector and fuel pump sizes<sup>(4), (5)</sup>. However, few studies have focused on simplifying the system by integrating sensor functions.

This study integrated sensor functions with the aim of simplifying the system. The functions of the intake air pressure sensor were instead performed using a new learning control system that uses O<sub>2</sub> feedback control and a stroke distinction function that detects crankshaft rotation speed fluctuation. This enabled maintaining the same performance for exhaust emissions and fuel economy without an intake air pressure sensor.

## 2. Overview of Test Motorcycle and Current FI System

Figure 1 and Table 1 show an external view and the main specifications of the test motorcycle WAVE110.

---

\* Motorcycle R&D Center

Table 1 Engine specifications of test motorcycle

| Item                                   |         | Specifications           |
|----------------------------------------|---------|--------------------------|
| Engine type                            |         | 4-stroke single-cylinder |
| Cooling system                         |         | Air-cooled               |
| Bore × Stroke (mm)                     |         | 50.0 × 55.6              |
| Engine displacement (cm <sup>3</sup> ) |         | 109.1                    |
| Compression ratio                      |         | 9.0                      |
| Valve train                            |         | SOHC                     |
| Valve diameter (mm)                    | Inlet   | φ 25.5 × 1               |
|                                        | Exhaust | φ 21.0 × 1               |
| Maximum Horsepower (kW/r/min)          |         | 6.18/7500                |
| Maximum torque (Nm/r/min)              |         | 8.15/5500                |

The WAVE110 is a small motorcycle equipped with an air-cooled 4-stroke single-cylinder 110 cm<sup>3</sup> engine that complies with Thailand's sixth exhaust emission standards, which are equivalent to EURO3 exhaust emission standards.

Figure 2 shows an external view of the throttle body. The intake air pressure sensor eliminated in this study is mounted inside a sensor unit integrated with the throttle body.

Figure 3 shows the current FI system configuration of the test motorcycle.

The current FI system mounted in the WAVE110 does not have a startup air device. It complies with Thailand's



Fig. 1 Test motorcycle

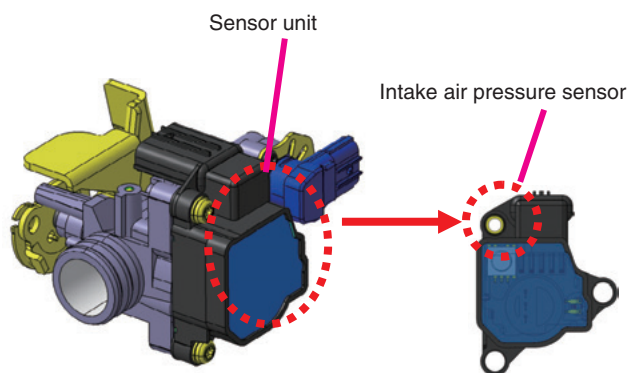


Fig. 2 External view of throttle body

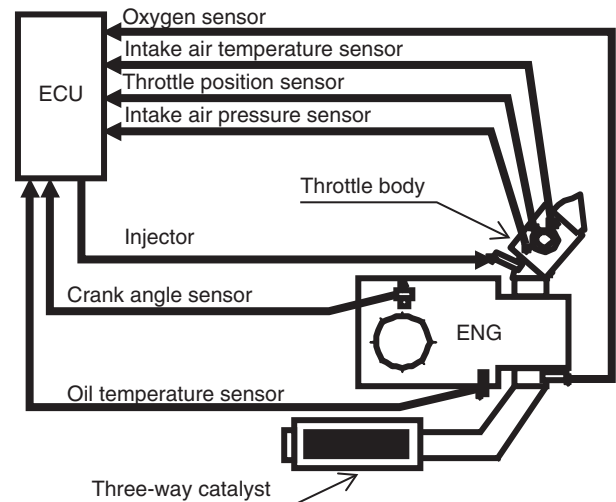


Fig. 3 Basic diagram of current FI system

sixth exhaust emission standards, which are equivalent to EURO3 exhaust emission standards, using a system that expands O<sub>2</sub> feedback control to the idle speed range and a three-way catalyst.

### 3. Investigation of Simplification

Elimination of sensors used in current FI systems was investigated as a means of simplification. Current FI systems determine the basic fuel injection quantity according to the amount of intake air obtained by direct measurement or indirect measurement. In the test motorcycle, the amount of intake air is not measured directly using an air flow sensor, and is instead measured indirectly as follows. In the minute throttle-opening range, the speed density method is used to calculate the amount of intake air from the intake air pressure and the engine rotation speed. Beyond the minute throttle-opening range, the throttle speed method is used to calculate the amount of intake air from the throttle-opening angle and the engine rotation speed<sup>(6)</sup>. In addition, multiple sensors are mounted to perform correction control in response to changes in the intake air temperature and atmospheric pressure, the engine warm-up state, and other factors. The intake air temperature sensor, O<sub>2</sub> sensor, engine oil temperature sensor, and crank angle sensor respectively measure the intake air temperature, concentration of residual oxygen in the exhaust gas, engine oil temperature, and crankshaft rotation speed. That is to say, each sensor measures a different subject, so replacement by other sensors is thought to be a challenge. On the other hand, while the throttle position sensor and intake air pressure sensor measure different subjects, these sensors perform the same function to indirectly measure the amount of intake air. Therefore, the feasibility was investigated of simplifying the system by eliminating one of these sensors and having the other sensor perform part of the eliminated sensor's functions.

#### 4. Determination of the Sensor to be Eliminated

The feasibility of eliminating the throttle position sensor or the intake air pressure sensor was investigated, focusing on the functions of each sensor. The main functions of the throttle position sensor are as follows.

- (1) Measurement of the amount of intake air beyond the minute throttle-opening range by the throttle speed method;
- (2) Determination of the throttle position (detection of full close, full open, and other operating states);
- (3) Acceleration decisions (correction to increase the fuel quantity when accelerating and correction to advance the ignition timing angle); and
- (4) Determination of ignition timing.

Investigation of the issues involved in eliminating the throttle position sensor found that:

- The accuracy drops somewhat, but the amount of intake air beyond the minute throttle-opening range can be measured by the intake air pressure sensor even in the current system.
- The throttle position judgment is used to determine idling, full open running and other operating states. Therefore, substitution requires the addition of a throttle switch, so the system cannot be simplified.
- Motorcycles require good acceleration response, so the throttle position sensor is the sensor most suited to detecting operating conditions. This makes it a challenge to construct a means of replacing the functions of acceleration decisions and determination of ignition timing.

As a result, it was decided not to eliminate the throttle position sensor.

On the other hand, the main functions of the intake air pressure sensor are as follows.

- (1) Measurement of the amount of intake air in the minute throttle-opening range by the speed density method;
- (2) Estimation of atmospheric pressure from the intake air pressure; and
- (3) Engine stroke distinction.

Similar investigation of the issues involved in eliminating the intake air pressure sensor found that:

- The accuracy drops somewhat, but the amount of intake air in the minute throttle-opening range can be measured by the throttle position sensor even in the current system.
- The atmospheric pressure is estimated to correct the fuel injection quantity in response to changes in the atmospheric pressure, but that correction can be replaced with other means for correcting fuel injection quantity.
- The engine-stroke distinction function can be replaced by detecting the angular velocity of the crankshaft, which fluctuates according to each stroke.

As a result, it was thought possible to construct a system without the intake air pressure sensor.

#### 5. Intake Air Amount Characteristics and Issues Involved in Eliminating the Intake Air Pressure Sensor

To eliminate the intake air pressure sensor it is necessary to replace the three functions performed by the intake air pressure sensor as described above. Investigations to extract the issues involved in this replacement first focused on the relationship between the throttle-opening angle and the amount of intake air.

Figure 4 shows a typical example of the amount of intake air and intake air pressure characteristics relative to the throttle-opening angle. In the minute throttle-opening range (throttle opening: 7.5 deg or less), there is little change in the amount of intake air but the intake air pressure changes greatly. On the other hand, beyond the minute throttle-opening range (throttle opening: 7.5 deg or more), the amount of intake air changes greatly but there is little change in the intake air pressure. This means that the speed density method can more accurately measure the amount of intake air in the minute throttle-opening range, and the throttle speed method can more accurately measure the amount of intake air beyond the minute throttle-opening range. Generally, both the speed density method and the throttle speed method are used in motorcycles, and various settings are made in accordance with these characteristics. Therefore, when eliminating the intake air pressure sensor, the first issue is to maintain accurate measurement of the amount of intake air in the minute throttle-opening range.

Next, issues related to adjustment of the engine idle speed are described.

Figure 5 shows the structure of a throttle body with a bypass. The base model throttle body has a bypass that enables manual adjustment of the amount of air passing through the throttle in order to adjust the engine idle speed. The air taken into the engine is divided into two passages: the main passage through the throttle valve and the bypass.

Figure 6 shows an example of the intake air

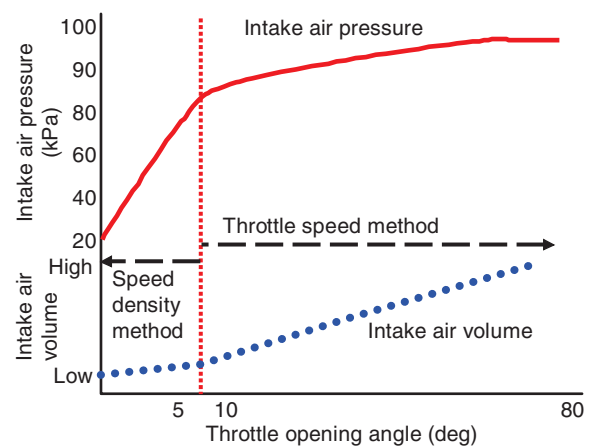


Fig. 4 Characteristics of intake air pressure and amount of intake air in relation to throttle opening angle

composition characteristics relative to the throttle-opening angle. At a throttle opening of 0 deg, most of the intake air passes through the bypass, and the amount of the bypass air decreases as the throttle valve opens further. The speed density method can measure changes in the amount of intake air due to adjusting the bypass as changes in the intake air pressure. This enables maintaining accurate measurement of the amount of intake air. On the other hand, changes in the amount of intake air do not appear as changes in the throttle-opening angle, so the throttle speed method is unable to maintain this measurement accuracy. Therefore, the second issue is how to measure changes in

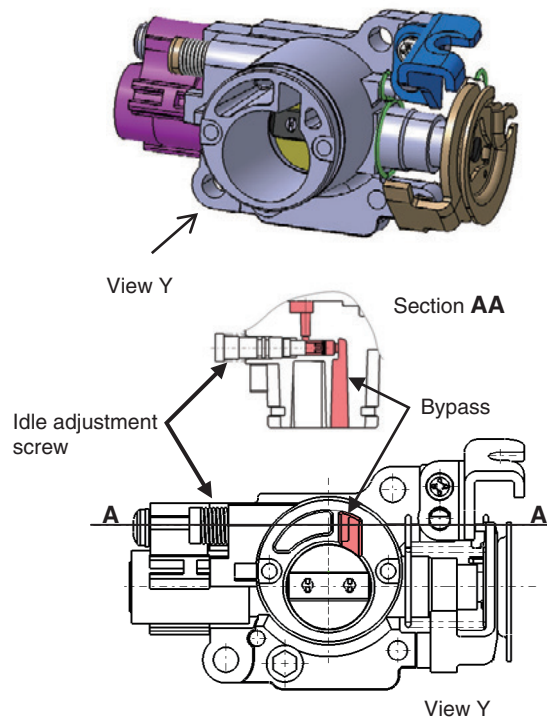


Fig. 5 Throttle body and bypass structure

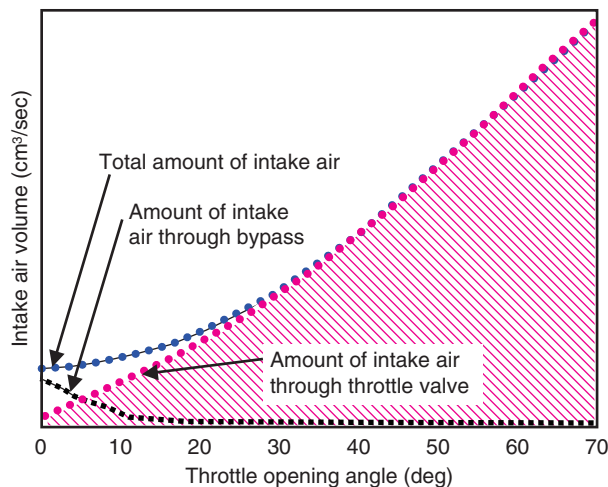


Fig. 6 Amount of intake air in relation to throttle opening angle

the amount of intake air due to adjustment of the engine idle speed.

Next, the second function of the intake air pressure sensor is described: corrections relative to changes in atmospheric pressure. The intake air density changes according to changes in atmospheric pressure due to weather conditions or the altitude at which the motorcycle is running. To perform control to the proper air fuel ratio, the fuel injection quantity should be corrected in response to changes in the intake air density. Most current FI systems estimate the atmospheric pressure from the intake air pressure in order to perform correction. Therefore, the third issue when eliminating the intake air pressure sensor is how to maintain the accuracy of fuel injection quantity corrections in response to changes in atmospheric pressure.

Finally, the third function of the intake air pressure sensor is described: engine stroke distinction. In a 4-stroke single-cylinder engine, stroke distinction is normally performed immediately after starting the engine in order to inject the fuel at the proper timing. The engine-stroke distinction function of the current system uses the intake air pressure sensor signal, so the fourth issue when eliminating the intake air pressure sensor is how to maintain this stroke distinction function.

As described above, in order to replace the three functions of the intake air pressure sensor, the following four issues need to be considered.

- Maintaining accurate measurement of the amount of intake air in the minute throttle-opening range
- Maintaining accurate measurement of the amount of intake air when adjusting the engine idle speed
- Maintaining accurate correction of the fuel injection quantity in response to changes in atmospheric pressure
- Maintaining the engine-stroke distinction function

These can be broadly grouped into the following two items:

- (1) Maintaining accurate measurement of the amount of intake air and accurate correction of the fuel injection quantity
- (2) Maintaining the engine-stroke distinction function

Alternate methods of accomplishing these items were constructed.

## 6. Alternate Method of Maintaining Accurate Measurement of the Amount of Intake Air and Accurate Correction of the Fuel Injection Quantity

First, an alternate method is described of maintaining accurate measurement of the amount of intake air and accurate correction of the fuel injection quantity, which is the first item mentioned above.

### 6.1. Measures Using O<sub>2</sub> Feedback Control

Construction of an alternate method focused on the O<sub>2</sub>

feedback control that is used to increase the purification performance of the three-way catalyst in current FI systems. In  $O_2$  feedback control, it is detected whether the air fuel ratio is richer or leaner than the theoretical air fuel ratio based on the output voltage of the  $O_2$  sensor which measures oxygen concentration in the exhaust emissions, and constantly makes small adjustments to the fuel injection quantity using the  $O_2$  feedback coefficient (hereafter “ $K_{O2}$ ”) to cancel out the rich and lean ratios and converge towards the theoretical air fuel ratio. Figure 7 shows an example of  $O_2$  feedback control operation.  $O_2$  feedback control enables correction of the fuel injection quantity using  $K_{O2}$  to supply the proper amount of fuel in response to changes in the amount of intake air in the minute throttle-opening range, changes in the amount of intake air due to engine idle speed adjustment, and changes in the air density due to changes in atmospheric pressure. That is to say,  $O_2$  feedback control basically enables maintaining accurate measurement of the amount of intake air and accurate correction of the fuel injection quantity.

## 6.2. Measures when $O_2$ Feedback Control is Inactive

### 6.2.1. Issues when $O_2$ feedback control is inactive

$O_2$  feedback control enables maintaining accurate measurement of the amount of intake air and accurate correction of the fuel injection quantity. However,  $O_2$  feedback control cannot be performed under all operating conditions.

$O_2$  feedback control cannot be performed under the following conditions:

- When the  $O_2$  sensor is cold;
- While the engine is warming up;
- While the fuel injection is cut;
- During rapid acceleration; and
- Immediately after restarting the engine.

These conditions occur frequently during actual operation, so  $O_2$  feedback control often cannot be

performed. At that time, the fuel injection quantity cannot be corrected in response to changes in the amount of intake air due to changes in engine conditions, adjustment of the engine idle speed, and changes in atmospheric pressure. Therefore, measures are needed for when  $O_2$  feedback control is inactive.

### 6.2.2. Measures of addressing issues when $O_2$ feedback control is off

To deal with these issues, learning control was constructed using the operational condition compensation coefficient (hereafter “ $K_{bu}$ ”) defined below. The basic approach of this learning control is as follows. The average value of  $K_{O2}$  during a given time period while  $O_2$  feedback control is operating in actual atmospheric conditions, after stabilizing to the theoretical air fuel ratio, is defined as  $K_{O2ave}$ . The  $K_{O2}$  value that results in the theoretical air fuel ratio in standard atmospheric conditions is defined as the reference value  $K_{O2std}$ , and  $K_{bu}$  is calculated as the ratio of  $K_{O2ave}$  to  $K_{O2std}$ . When  $K_{bu}$  is stable, it is stored in the ECU memory as the learned value (hereafter “learned”) and applied to the basic fuel injection quantity. This learning control uses the learned  $K_{bu}$  to perform correction that matches the engine demands, enabling constant supply of the proper fuel injection quantity even under conditions when  $O_2$  feedback control cannot be performed.

The  $K_{bu}$  calculation formula is shown below.

$$K_{bu} = K_{O2ave} / K_{O2std} \quad (1)$$

$K_{bu}$ : Operational condition compensation coefficient

$K_{O2}$ :  $O_2$  feedback correction coefficient ( $K_{O2} = 1.0$  when  $O_2$  feedback control is inactive)

$K_{O2ave}$ : Average value of  $O_2$  feedback correction coefficient  $K_{O2}$  at the actual atmospheric pressure

$K_{O2std}$ :  $O_2$  feedback correction coefficient that results in the theoretical air fuel ratio at the standard atmospheric pressure

The  $K_{bu}$  obtained by this calculation formula was reflected to the formula used to calculate the fuel injection quantity.

This fuel injection quantity calculation formula is as follows.

$$T_{out} = Map_{th} \times K_{bu} \times K_{to} \times K_{O2} \quad (2)$$

$T_{out}$ : Fuel injection quantity

$Map_{th}$ : Basic fuel injection quantity in the throttle speed method

$K_{to}$ : Oil temperature correction coefficient

When  $O_2$  feedback control is active,  $K_{O2}$  is used to adjust towards the theoretical air fuel ratio in response to changes in the amount of intake air. When the  $K_{O2ave}$  calculated at

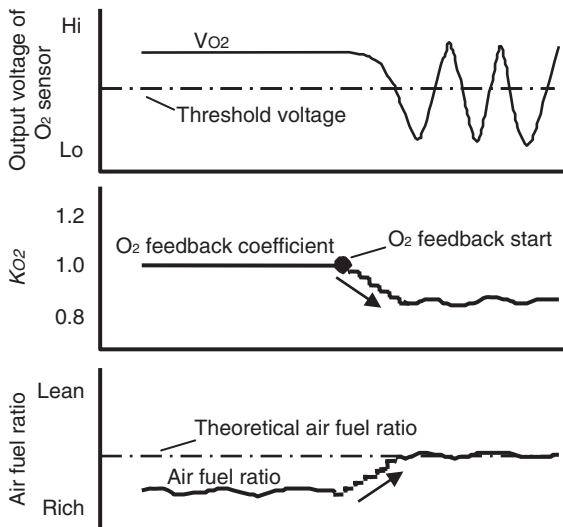


Fig. 7  $O_2$  feedback control operation

this time deviates from  $K_{O2std}$  by a certain amount or more,  $K_{bu}$  is updated. This operation keeps  $K_{O2}$  at around 1.0.

When  $O_2$  feedback control is inactive,  $K_{O2}$  is fixed to 1.0, but using the  $K_{bu}$  updated by the preceding operation enables supply of the same fuel injection quantity as when  $O_2$  feedback control is active.

Tests were conducted using a completed motorcycle to compare the changes in the air fuel ratio during strong acceleration before and after  $K_{bu}$  is calculated (Fig. 8). Before calculation (left figure), when  $K_{O2}$  becomes 1.0 while  $O_2$  feedback control is inactive, correction does not function and the difference from the target air fuel ratio increases. After calculation (right figure), correction is performed based on calculated  $K_{bu}$  and  $K_{to}$  values in response to changes in engine conditions, the amount of intake air, and atmospheric pressure, even when  $K_{O2} = 1.0$  while  $O_2$  feedback control is inactive. This results in small differences from the target air fuel ratio.

Points (1) to (7) in Fig. 9 show the process used by the developed system to calculate and store  $K_{bu}$  after starting the engine in the cold condition, and the process until  $O_2$  feedback control starts the next time the engine is operated.

- (1) The engine is cold when it starts, so the fuel injection quantity is generally increased as startup correction to stabilize the engine idle speed.  $K_{bu}(0)$  represents the initial value. When the altitude, weather or other environmental conditions differ from the conditions set for the initial value, the target air fuel ratio may not be achieved after this startup correction ends.

- (2) The increase in the fuel injection quantity ends, and  $O_2$  feedback control starts. The air fuel ratio is immediately corrected to the theoretical air fuel ratio, and at the same time  $K_{bu}(n)$  is calculated from the  $K_{O2}$  average value.  $K_{bu}$  is updated from  $K_{bu}(0)$  to  $K_{bu}(n)$ .
- (3) This  $K_{bu}(n)$  value is stored in the nonvolatile memory as the learned value.
- (4) The engine is stopped.  $K_{bu}(n)$  is retained because it is stored in the nonvolatile memory.
- (5) When the engine is restarted next, the  $K_{bu}(n)$  value is read from the nonvolatile memory and used as the operational condition correction coefficient  $K_{bu}(n)$  until  $O_2$  feedback control starts.  $K_{bu}(n)$  is updated at points (2) and (3), so control can be performed to a value near the target air fuel ratio even when  $O_2$  feedback control is inactive.
- (6) When  $K_{O2}$  differs after  $O_2$  feedback control starts, a new value  $K_{bu}(n+1)$  is calculated.
- (7) When the  $K_{bu}(n+1)$  value differs from  $K_{bu}(n)$ ,  $K_{bu}(n+1)$  is stored in the nonvolatile memory as the learned value.

### 6.3. Verification Results for the Effects of Correction by $K_{bu}$

Figure 10 shows the results of verifying the effects of fuel injection quantity correction by  $K_{bu}$  from the condition with the air fuel ratio shifted to the rich side during idling in a completed motorcycle.

- The left figure shows the condition before  $K_{bu}$  has been

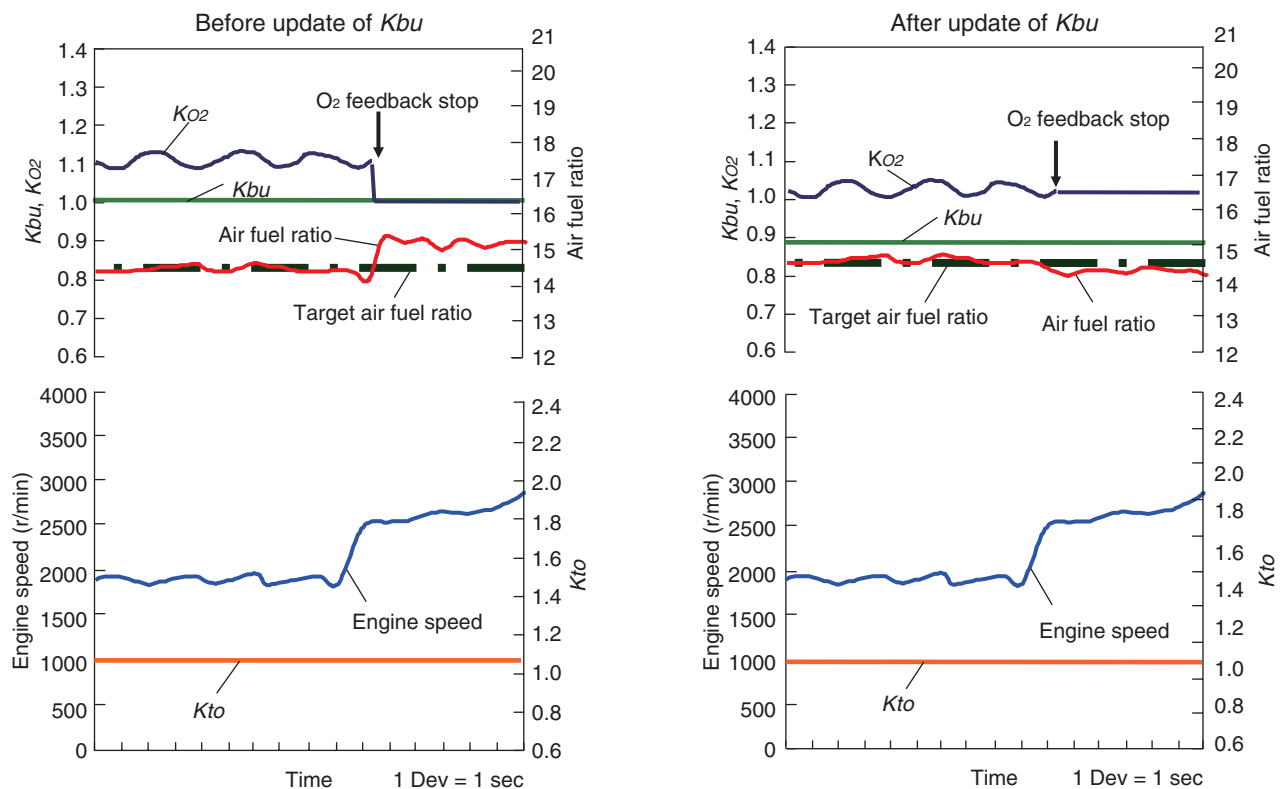


Fig. 8 Air-Fuel ratio correction during steep acceleration before and after  $K_{bu}$  is calculated

learned, with the actual air fuel ratio shifted from the target air fuel ratio (theoretical air fuel ratio).

- The center figure shows the condition after  $Kbu$  is learned, with the air fuel ratio approaching the target air fuel ratio (theoretical air fuel ratio).
- The right figure shows the start of  $O_2$  feedback control and convergence of the air fuel ratio to the theoretical air fuel ratio.

Figure 11 shows the changes in the  $Kbu$  value and the air fuel ratio when the atmospheric pressure is gradually lowered from approximately 100 kPa to approximately 65 kPa in the idling state. As the atmospheric pressure changes, a difference occurs between  $Ko_2$  and  $Ko_2std$ , and  $Kbu$  is

updated. This operation is repeated so that  $Kbu$  tracks the change in atmospheric pressure, enabling control of the proper air fuel ratio even when operating at high altitudes.

These results show that the function to measure the amount of intake air can be maintained even without the intake air pressure sensor.

#### 6.4. Enhancing the Accuracy of Correction by $Kbu$

This learning control system using  $Kbu$  is an effective control system that applies  $O_2$  feedback control to support various operating environments and conditions. However, some consideration must be given to the bypass characteristics described above.

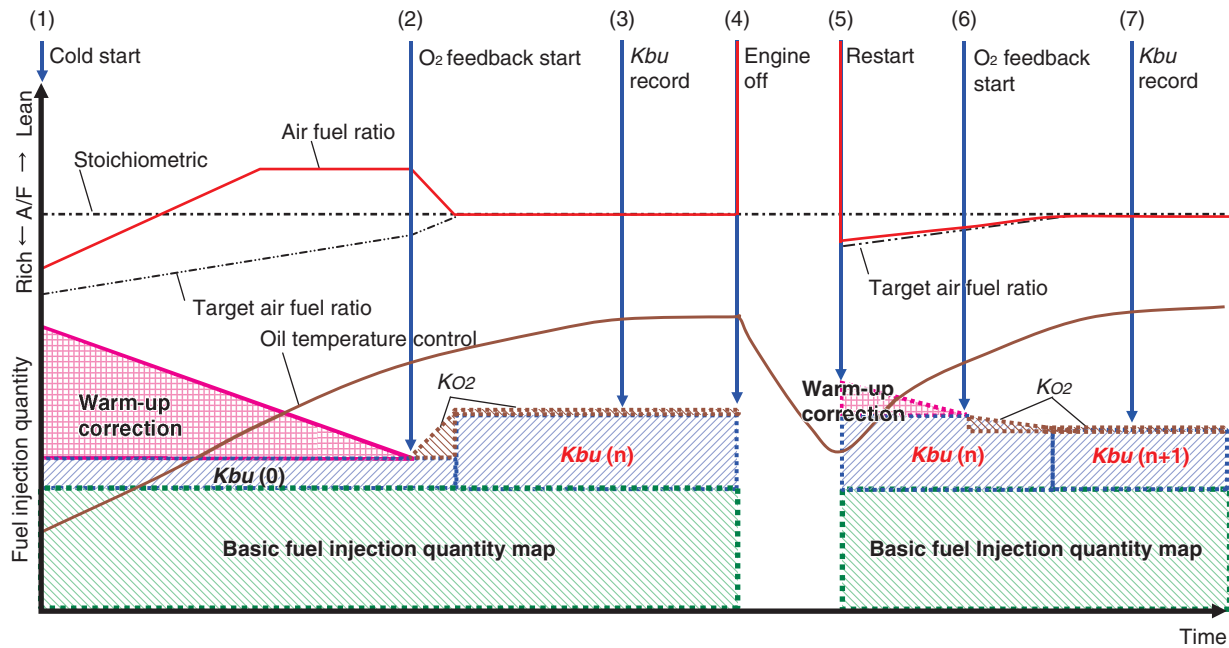


Fig. 9 Process to calculate and store  $Kbu$  and perform fuel injection control after cold start of engine

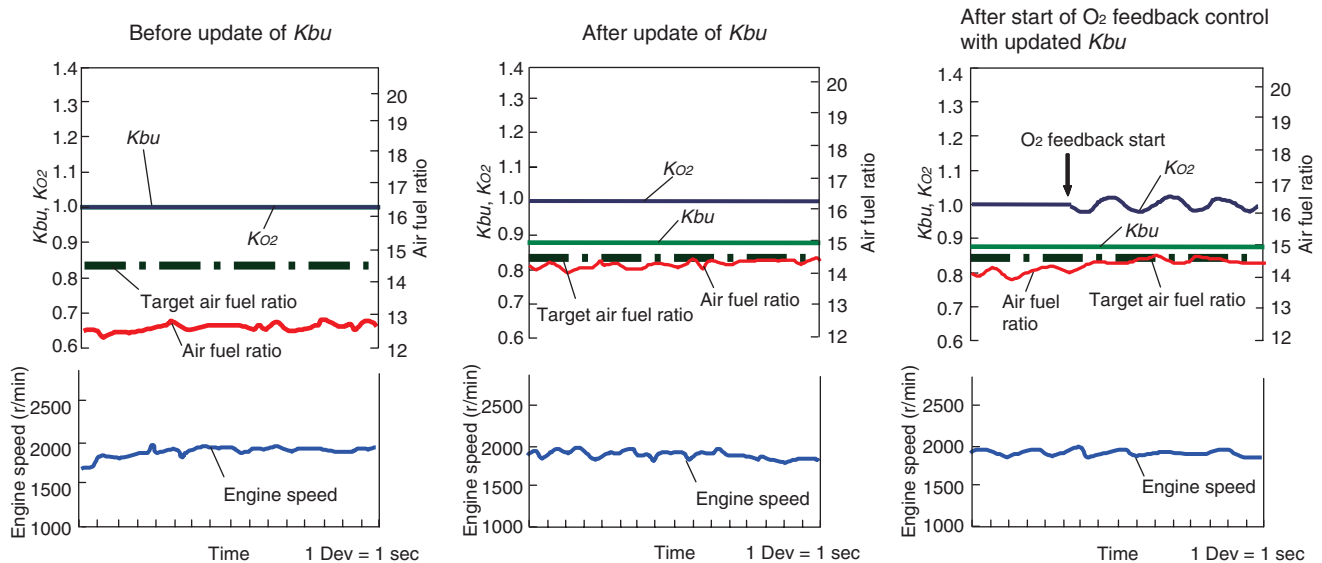


Fig. 10 Process to calculate and store  $Kbu$  and perform fuel injection control after cold start of engine

Figure 12 shows the change in the amount of intake air relative to the throttle-opening angle when the bypass is set to reduce the amount of intake air by 20% during idling. The 0% line is the baseline, and represents the state before adjusting the amount of idling intake air. When the amount of intake air during idling is reduced by 20%, the percentage of change in the amount of intake air relative to the total air increases as the throttle-opening angle approaches the minute throttle-opening range. Figure 13 shows the change in the  $K_{bu}$  value relative to the throttle-opening angle at that time. The 1.0 line is the baseline, and represents  $K_{bu}$  before adjusting the amount of idling intake air.  $K_{bu}$  represents

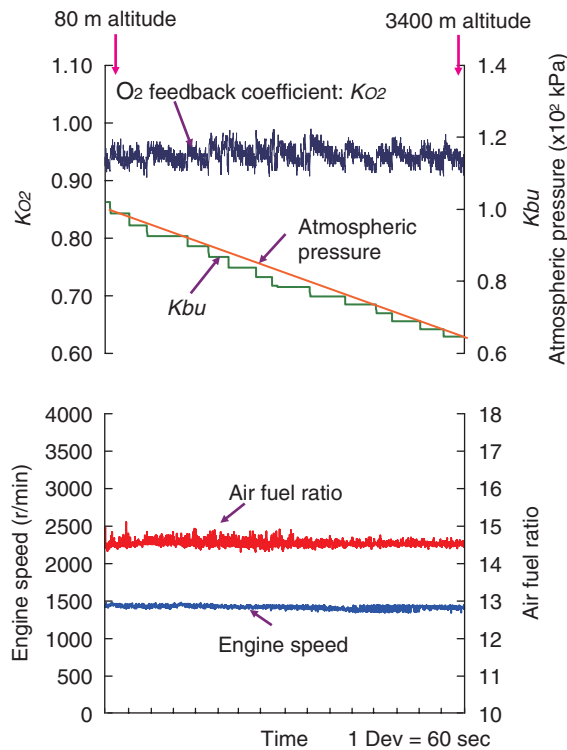


Fig. 11 Corrective effect by  $K_{bu}$  in response to changes in atmospheric pressure

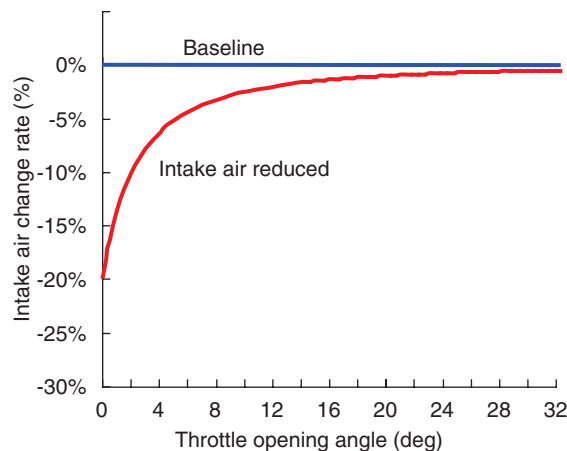


Fig. 12 Change rate of air intake when amount of intake air is reduced by 20% by bypass

the rate of change in the amount of intake air from its own definition, so it exhibits the same characteristics as the amount of intake air and shows the greatest change in the minute throttle-opening range. This means that accurate correction cannot be performed using only a single  $K_{bu}$  value when operating in the minute throttle-opening range. Therefore, to increase the accuracy of correction by  $K_{bu}$ , the  $K_{bu}$  calculation area was divided into multiple sections according to the throttle-opening angle.

Smaller  $K_{bu}$  calculation area divisions mean less change in  $K_{bu}$  within a single division, which enables minimizing the discrepancy between the theoretical air fuel ratio and the actual air fuel ratio. However, in transient operating states where the throttle-opening angle changes constantly, the time that the measurement point remains within a single  $K_{bu}$  calculation area shortens if the  $K_{bu}$  calculation area divisions become smaller and the throttle operation speed increases. When this time is shorter than the time required to calculate  $K_{bu}$ , the measurement point moves to a different  $K_{bu}$  calculation area before the calculation can be completed. As a result, an occasion to calculate  $K_{bu}$  is missed and the accuracy of air fuel ratio correction by  $K_{bu}$  drops. This makes it necessary to take into account the air fuel ratio correction accuracy and the occasion for  $K_{bu}$  calculations when setting the  $K_{bu}$  calculation area divisions. The approach taken to determine the  $K_{bu}$  calculation area divisions is described below using Fig. 14.

- The idling area is the area most affected by conditional variations among completed engines and intake air adjustment, so a narrow width is set (Throttle 0 deg - Border 1).
- In the minute throttle-opening range, the required  $K_{bu}$  value changes greatly in response to slight changes in the throttle-opening angle as shown in Fig. 13. When a wider division is set in this area, the occasions for  $K_{bu}$  calculation increase, but the  $K_{bu}$  deviation at the upper and lower boundaries of a single area also increases. This increases the difference in the air fuel ratio within a single area, which reduces drivability.

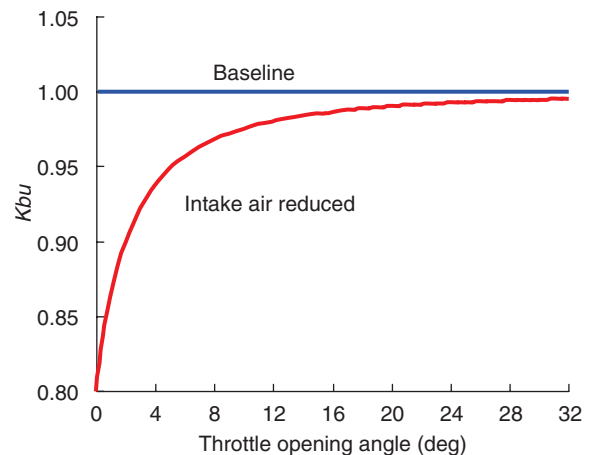


Fig. 13 Change in  $K_{bu}$  when amount of intake air is reduced by 20% by bypass

Conversely, setting narrower divisions reduces the effect on drivability within a single area, but the occasions for  $K_{bu}$  calculation decrease, so the air fuel ratio correction accuracy drops.

In consideration of these aspects and as a result of investigating the balance between occasions for  $K_{bu}$  calculation and drivability, it was confirmed through completed motorcycle tests that both occasions for  $K_{bu}$  calculation and drivability can be secured by keeping the

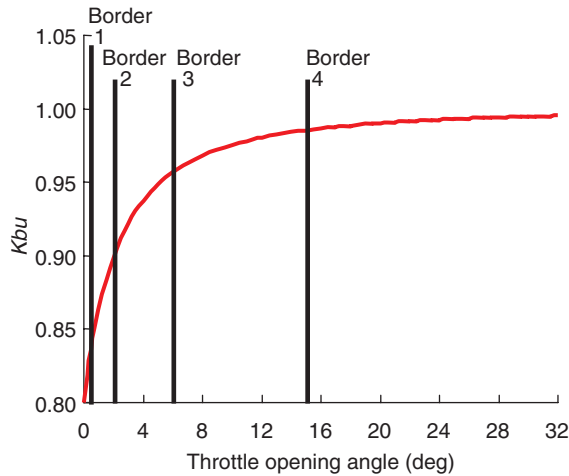


Fig. 14  $K_{bu}$  calculation area divisions

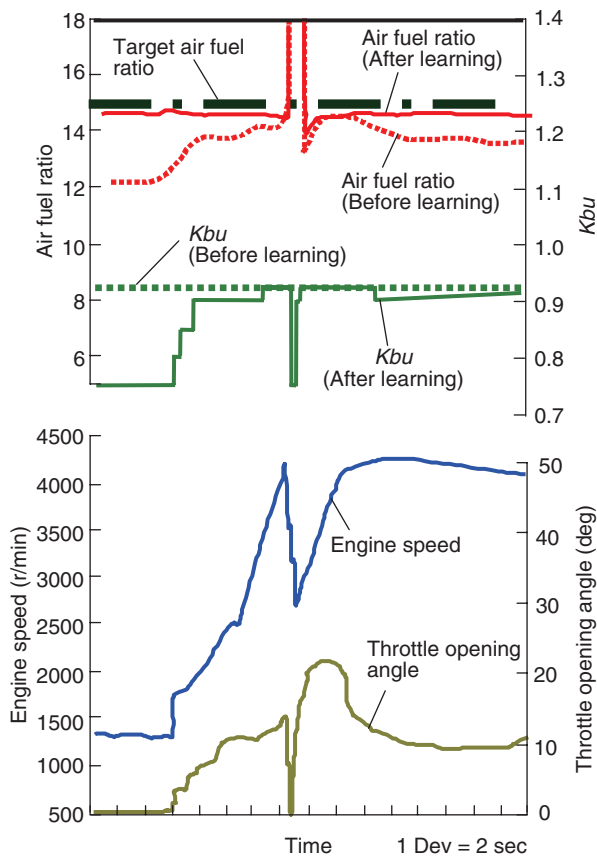


Fig. 15 Effect of  $K_{bu}$  calculation division setting on air fuel ratio correction

$K_{bu}$  deviation at the upper and lower boundaries of each area to within 10%. As a result, three  $K_{bu}$  calculation areas were set. (Border 1 - Border 4)

- The medium and higher throttle-opening range is relatively unaffected by conditional variations among completed engines, intake air adjustment during idling, and other factors, so a single calculation area was sufficient even in consideration of correction accuracy. (Border 4 - WOT)

Based on the above investigation and confirmation results, five area divisions were determined as shown in Fig. 14.

Figure 15 compares the air fuel ratio correction results for a single  $K_{bu}$  calculation area and five  $K_{bu}$  calculation areas when  $O_2$  feedback control is inactive. The dotted line shows the air fuel ratio correction results for a single  $K_{bu}$  calculation area, and the solid line shows the results when divided into five areas according to the throttle-opening angle.  $O_2$  feedback control is inactive, so  $K_{bu}$  is the value calculated under the conditions preceding the test. When performing calculation with a single division, there is only a single  $K_{bu}$  value, so there is continued large deviation from the  $K_{bu}$  required in response to changes in the throttle-opening angle. This means that correction cannot be performed to the target air fuel ratio. On the other hand, when using five divisions, the five  $K_{bu}$  values calculated for the respective divisions prior to the test are applied, so the air fuel ratio is corrected to a value close to the target in response to changes in the throttle-opening angle, even when  $O_2$  feedback control is inactive. This confirmed that the deviation between the target air fuel ratio and actual air fuel ratio can be minimized by setting the minimum necessary number of  $K_{bu}$  calculation area divisions in consideration of the change in the amount of intake air when the engine idle speed is adjusted.

## 7. Maintaining the Engine-Stroke Distinction Function

Next, an alternate method is described of maintaining the engine-stroke distinction function without the intake air pressure sensor, which is the other item mentioned above.

### 7.1. Conventional Engine-stroke Distinction Method

The conventional system does not have a camshaft position sensor, and distinguishes the engine stroke using the intake air pressure sensor. Figure 16 shows the intake air pressure waveform of a 4-stroke single-cylinder engine in the idling state.

The intake air pressure drops during the intake stroke, and rises gradually to approach atmospheric pressure during the compression, expansion and exhaust strokes. This characteristic is used to distinguish the strokes by measuring the intake air pressure of the intake stroke and the expansion stroke at the same phase. The stroke with the lower air pressure is identified as the intake stroke<sup>(7)</sup>.

## 7.2. Stroke Distinction Method Using Crankshaft Rotation Speed Fluctuation

In order to perform stroke distinction without using the intake air pressure sensor, this study focused on the characteristic that the crankshaft rotation speed of a 4-stroke single-cylinder engine becomes slower during the compression stroke.

Figure 17 shows the crankshaft rotation speed fluctuation in the idling state. The crankshaft rotation speed falls during the exhaust, intake and compression strokes, and rises during the expansion stroke. This means that the minimum crankshaft rotation speed occurs during the compression stroke. Therefore, control is performed to

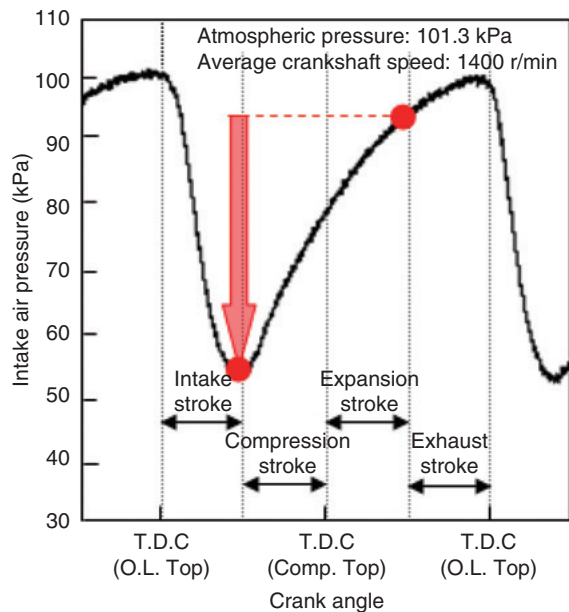


Fig. 16 Variation of intake air pressure at idling speed

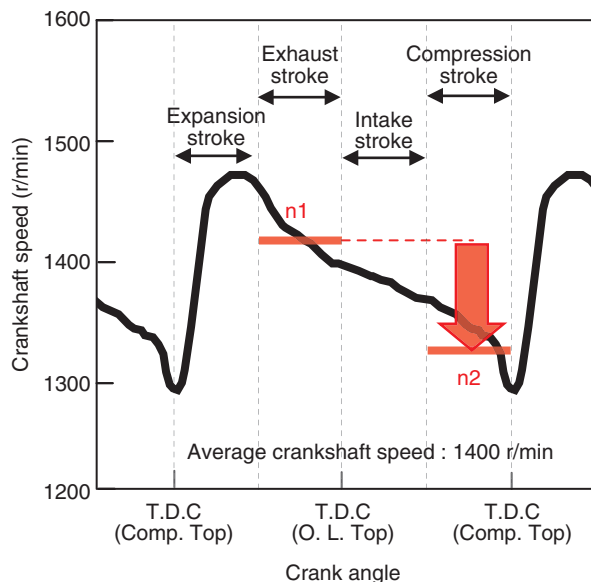


Fig. 17 Variation of crankshaft speed at idling speed

measure the average engine speed during the exhaust stroke  $n1$  and the average engine speed during the compression stroke  $n2$  based on the crank angle sensor signal, and the lower of the two is distinguished as the compression stroke. Note that not including the expansion stroke in the comparison of crankshaft rotation speeds increases resistance to the effects of combustion states such as misfiring.

Figure 18 compares the average engine speeds during the exhaust stroke and the compression stroke, relative to the average engine speed in the idling state. This shows that the average engine speed during the exhaust stroke  $n1$  is always lower than the average engine speed during the compression stroke  $n2$ , so stroke distinction is possible even when the idling speed changes. This enabled maintaining the engine-stroke distinction function even without the intake air pressure sensor.

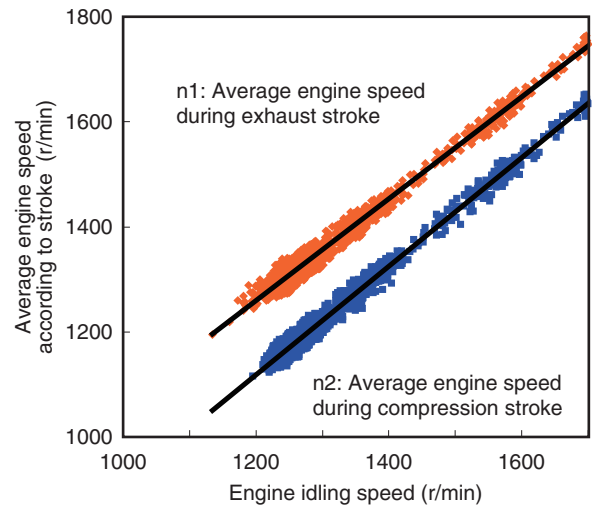


Fig. 18 Comparison of average engine speed between exhaust stroke and compression stroke

## 8. Results of Confirming Effects

### 8.1. Exhaust Emissions and Fuel Economy Measurement Results

Table 2 shows the exhaust emissions measurement results, and Table 3 shows the measurement results for mode fuel economy and fuel economy at constant speed. Exhaust emissions and mode fuel economy were measured in Thailand's sixth exhaust emission standards mode, which is a driving pattern equivalent to ECE R40 mode. A motorcycle equipped with the developed system exhibited the same performance for exhaust emissions, mode fuel economy, and fuel economy at constant speed as a motorcycle equipped with the current FI system. This confirmed that a simple system without an intake air pressure sensor can maintain performance equivalent to that of the current system.

Table 2 Comparison of exhaust emissions

|                     | CO<br>(g/km) | THC<br>(g/km) | NOx<br>(g/km) |
|---------------------|--------------|---------------|---------------|
| Current system      | 0.525        | 0.217         | 0.039         |
| Developed system    | 0.529        | 0.214         | 0.038         |
| Regulation values * | 2.0          | 0.8           | 0.15          |

\* Thailand's sixth exhaust emission standard

Table 3 Comparison of fuel economy performance

|                  | Mode<br>(ECE R40) | Constant speed<br>(50 km/h) |
|------------------|-------------------|-----------------------------|
| Current system   | 55.6 km/l         | 79.6 km/l                   |
| Developed system | 56.7 km/l         | 80.0 km/l                   |

## 9. Conclusion

A simple fuel injection system without an intake air pressure sensor was constructed using new learning control that applies O<sub>2</sub> feedback control and an engine-stroke distinction function based on detection of crankshaft rotation speed fluctuation. The following results were obtained.

- (1) O<sub>2</sub> feedback control enabled accurate measurement of the amount of intake air, which was an issue when eliminating the intake air pressure sensor.
- (2) Performing correction using the operational condition correction coefficient  $K_{bu}$  calculated beforehand when O<sub>2</sub> feedback control is active enabled accurate measurement of the amount of intake air even when O<sub>2</sub> feedback control is inactive. In addition, dividing the  $K_{bu}$  calculation area into five areas enabled precise correction of the air fuel ratio even when the engine idle speed was adjusted.
- (3) The engine stroke distinction could be maintained by comparing the average engine speed during the compression stroke and the average engine speed during the exhaust stroke.
- (4) The developed system maintained the same performance for exhaust emissions and fuel economy as the conventional system.

## References

- (1) Utsugi, K., Akamatsu, S.: Technology for Secondary Emission Regulation in Japan of 50 cm<sup>3</sup> Two-wheeled Vehicle, 2007 JSAE Annual Congress Proceedings, No. 29-07, p. 1-4 (2007)
- (2) Shimada, N., Akamatsu, S., Ueda, M., Hayashi, A., Hirakata, Y.: Development of an ECU Integrated Throttle Body System Module for a Compact Fuel Injection System Used in Small Motorcycles and Scooters, Honda R&D Technical Review, Vol. 15, No. 1, p. 83-88

- (3) Hayashi, T., Uraki, M., Makino, K., Okumura, T.: Development of Heaterless Oxygen Sensor for Small Motorcycles, 2006 JSAE Annual Congress Proceedings, No. 92-06, p. 21-24 (2006)
- (4) Ikeda, K., Akamatsu, S., Tanaka, H., Murakami, Y., Oki, K., Kurosaka, H.: Development of 50 cm<sup>3</sup> Four-stroke Four-valve Electronically Controlled Fuel-injection Engine for Two-wheeled Vehicle, 2004 JSAE Annual Congress Proceedings, No. 9-04, p. 15-18 (2004)
- (5) Ueno, M., Ueda, M., Udono, T., Akamatsu, S., Torikai, M., Fukuda, M.: Development of Small Fuel Pump Module for Motorcycles, Honda R&D Technical Review, April, 2008, p. 89-94
- (6) Horiike, S., Yamada, H., Ooyama, T., Kano, K., Abe, M., Iwata, Y.: Exhaust Emission Reduction System for "VFR800FI," HONDA R&D Technical Review, Vol. 10, p. 42-48
- (7) Akamatsu, S., Ishida, S., Hamauzu, A.: A Research of Fuel Injection System for Low Emission Motorcycle, 2000 JSAE Annual Congress Proceedings, No. 103-00, p. 17-20 (2000)

## ■ Author ■



Katsuhiro UTSUGI



Yuki TAKANO



Kazuhiko SAKAGUCHI



Kenta ONISHI

# Development of Axial Motor

Shoei ABE\*

Junji INOUE\*

Satoshi ISHIKAWA\*

## ABSTRACT

Research was conducted on an axial gap-type rotating machine with the aim of increasing the torque density. To expand the torque-generating surface, an interior permanent magnet (IPM) rotor was used in a double stator-type motor that enables effective use of both sides of the rotor. This IPM rotor combines main magnets with a magnetization direction in the axial direction, auxiliary magnets with a magnetization direction in the circumferential direction, and laminated electrical steel, and fixes these components in place using adhesive. The developed motor was confirmed to have a torque density per unit volume approximately 50% higher than that of a radial-type motor. In addition, to address the manufacturing issues for a rotor structure based on this bonded structure, the yoke was manufactured using the rolling laminate method, and a new structure was developed that inserts the magnets into the yoke, integrates them with the magnetic circuit, and then fixes them in place by interference fitting inner and outer rings that are nonmagnetic reinforcing components. This structure was confirmed to have sufficient strength and stiffness, and the motor efficiency was also approximately the same as that of a rotor manufactured using the adhesive method. The new structure was ultimately confirmed to achieve a volumetric torque density 1.34 times that of a radial-type motor.

## 1. Introduction

With increased awareness towards the global environment in recent years, demands are strengthening to reduce automobile emissions and the greenhouse gas CO<sub>2</sub>. To meet these expectations, Honda aims to contribute to the global environment by developing and popularizing environmentally friendly vehicles such as electric vehicles, hybrid vehicles, and fuel cell electric vehicles.

To increase application of electric vehicles, the motor that is the drive source is expected to manifest the desired torque and output power within a smaller design space. This means that even higher torque density and power density are required as motor performances. Power performance can be enhanced by increasing the motor speed and boosting the voltage. However, this involves an increase in ancillary parts such as a reduction gear and boost converter. In addition, the magnetic flux level or the current level must be increased to enhance torque performance, but magnetic flux density and current density are physical property limits of materials, so enhancement must rely on material advances. This study investigated a new structure and manufacturing method for an axial-type motor with a shape that increases the torque density by using existing materials without an increase in ancillary parts.

The torque generated by a motor is determined by the area of the torque-generating surface and the magnetic flux

density at the torque-generating surface. The torque to be generated can be increased even with the same volume by using a structure that increases the torque-generating area and the magnetic flux density within the limited space. Conventional motors have a structure called a radial gap type that locates the rotor inside the stator. This limits both the rotor surface area that generates torque and the arrangement of the magnets inside the rotor, so increasing the volumetric torque density is an issue. This study aimed to increase the volumetric torque density of the motor by combining the following two technologies:

- (1) Expansion of the torque-generating surface area by using a double stator axial-type motor
- (2) Use of a Halbach-type magnet arrangement that has a magnetic lens effect

## 2. Features of the Double Stator Axial Gap Motor

### 2.1. Difference from Radial Gap Type

Figure 1 shows the torque-generating surfaces of a conventional radial gap-type rotating machine and an axial gap-type rotating machine. In the radial gap-type motor, the interlinkage magnetic flux surface that generates the torque is the surface of the cylindrical rotor and has a ring shape. In contrast, in the axial gap-type motor, the stator and rotor are axially opposed, and the interlinkage magnetic flux surface

\* Automobile R&D Center

is shaped like a disc. In addition, the torque-generating surface area can be doubled by placing stators on both sides of a rotor or by placing opposing rotors on both sides of a stator. Further, in this case the rotor core back yoke or the stator core return pass can be eliminated. This means that the axial dimension can be shortened to less than double that of the conventional type, which enables increasing the torque density per unit volume<sup>(1)</sup>.

## 2.2. Double Halbach-type IPM Motor

As mentioned above, the motor torque is determined by the area of the torque-generating surface and the magnetic flux density at that surface. This study selected a double stator axial gap-type structure that increases the torque-generating surface area. In addition, a Halbach-type magnet arrangement was used to increase the magnetic flux density at the torque-generating surface and further increase the torque density<sup>(2)-(4)</sup>.

In an axial-type motor, the magnets are normally arranged so that the magnetization direction within a single pole matches the axial direction and the directions of adjacent poles alternate by 180 degrees. This is repeated to create a rotor with continuously alternating N and S poles. In contrast, arranging the magnets so that the magnetization directions are orthogonal as shown in Fig. 2 produces a magnetic lens effect that strengthens the magnetic flux in one direction. This was used to create a double Halbach-

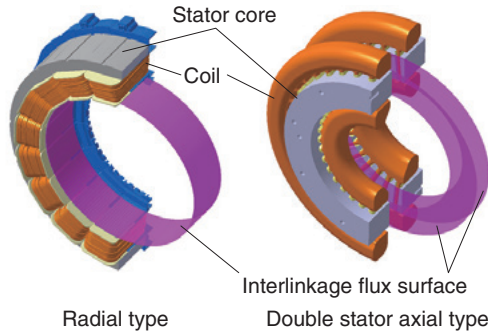


Fig. 1 Interlinkage flux surface

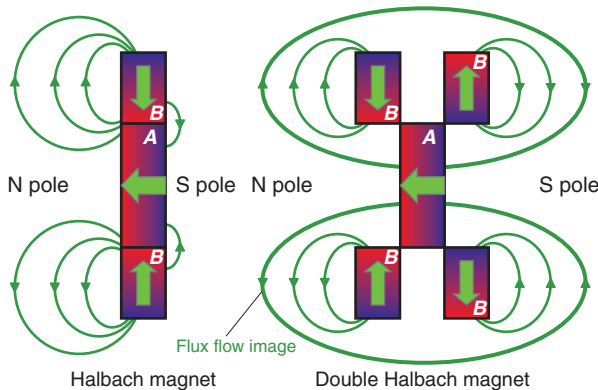


Fig. 2 Double Halbach magnet arrangement

Table 1 Specifications

|                      | IMA motor | Axial motor |
|----------------------|-----------|-------------|
| Number of poles      | 12        | 12          |
| Magnet               | Nd-Fe-B   | Nd-Fe-B     |
| Stator diameter [mm] | 258       | 258         |
| Rotor diameter [mm]  | 179       | 220         |
| Motor length [mm]    | 80        | 88          |
| Volume [L]           | 4.2       | 4.6         |

type magnet arrangement that manifests this magnetic lens effect on both surfaces of the rotor as shown in Fig. 3.

In addition, the structure has low magnetic resistance electrical steel in the axial direction on both sides of the main magnets that are opposed to the stator, and a rotor-holding component (frame) made of nonmagnetic materials in the spaces surrounded by the main magnets and auxiliary magnets. This strengthened the magnetic flux flow to the stator side and created an Interior Permanent Magnet (IPM) rotor with embedded magnets.

## 2.3. Verification of Torque Density

A prototype engine was designed and manufactured to confirm the effects of expanding the torque-acting surface area and increasing the magnetic flux density on increasing the torque density. As shown in Fig. 4, this prototype has the same six pole pairs and outer diameter size as the 2006 model year CIVIC IMA motor. Table 1 lists the detailed specifications.

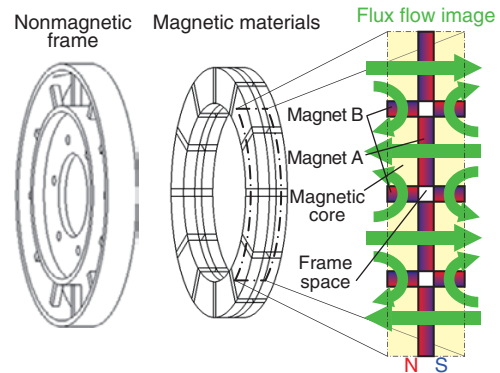


Fig. 3 Double Halbach-type rotor structure

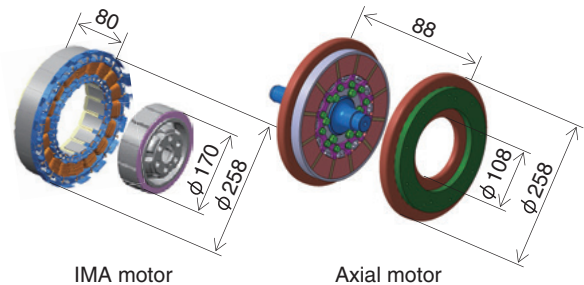


Fig. 4 Motor dimensions

Figure 5 shows the results of measuring the torque characteristics of the prototype motor. Comparing the generated torque confirmed that the prototype motor manifests 1.63 times the torque of the conventional motor. The volumetric torque density was 1.48 times that of the conventional radial gap motor, and 1.22 times even when converted to the torque per unit magnet mass for comparison. This confirmed that the double stator axial-type motor can greatly increase the torque density compared to a radial-type motor.

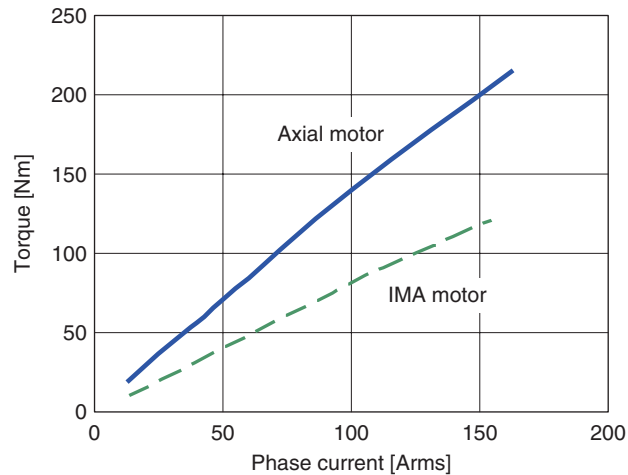


Fig. 5 Comparison of current-torque characteristics

### 3. Enhancement of Rotor Structure in Consideration of Manufacturing Method

#### 3.1. Rotor Issues

In a radial-type motor with magnets embedded in an IPM rotor, the magnetic circuit does not change in the axial direction and is two-dimensional. Therefore, the rotor yoke is generally made by punching and laminating thin sheets of electrical steel, and then press-fitting or otherwise fixing these laminates to the shaft that supports rotation and transfers the torque. The parts where the magnets are inserted are also punch-molded to create spaces that hold the magnets after lamination. This means that the electrical steel can be punch-molded into a shape that simultaneously realizes both the magnetic characteristics and mechanical strength, so there are no major manufacturing method issues.

In contrast, axial-type rotors have the following three issues:

- (1) Selection and molding of three-dimensional magnetic circuit yoke materials
- (2) Machinability of nonmagnetic reinforcing components (stainless materials)
- (3) Integration of magnetic circuit components and reinforcing components

Figure 6 shows the structure of the rotor manufactured to measure the torque characteristics. There are no particular issues concerning the molding and magnetization of the magnets used in an axial-type motor. However, the rotor magnetic circuit is not identical in the radial direction and is three-dimensional, so issue (1) is how to create the yoke that comprises the circuit. Soft magnetic composites and sintered material are generally used as the yoke materials for the three-dimensional magnetic circuit. However, the iron loss is greater than that of laminated electrical steel, so this study selected electrical steel. When using thin electrical steel sheets to configure a three-dimensional magnetic circuit for an axial-type rotor, a shape with the thin electrical steel sheets laminated in concentric rings most effectively suppresses the generation of eddy current. However, to achieve a fan shape similar to the main magnets that are magnetized in the axial direction, electrical steel pieces with different lengths and radii of curvature must be laminated sheet by sheet in the circumferential direction. This requires multiple cutting dies with different length dimensions and curvature-forming press machining. Another manufacturing method is to cut yoke materials wound and bonded in a hoop (quasi-concentric shape) into fan shapes using a machining method such as wire-cutting that can secure insulation between each layer. In any case, it must be said that these manufacturing methods are not suited to large-scale mass production compared to the yoke of a radial-type rotor.

Issue (2) is the machinability of the component (frame) used to secure strength and rigidity as a rotor. When magnetic flux flows easily through the frame component, a magnetic flux short-circuit occurs inside the rotor, and the magnetic flux that does not contribute to torque increases. Therefore, nonmagnetic materials must be used. Aluminum alloy is a general nonmagnetic material that is lightweight and easy to machine, but the required stiffness cannot be secured within a limited space (dimensions), so austenite stainless<sup>(5)</sup> was used. However, austenite stainless is a challenge to machine, so a machining process is required to accurately form a one-piece part with thin spokes such as

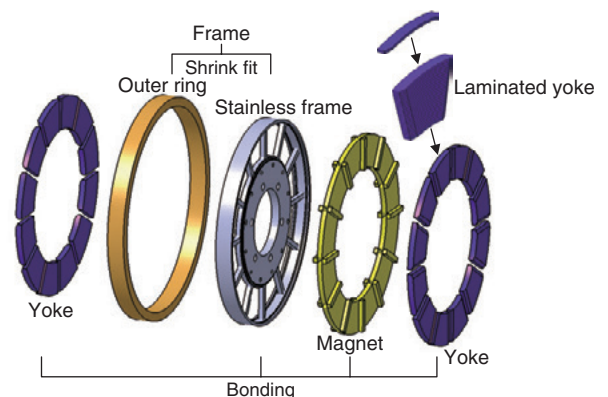


Fig. 6 Bonded fragment structure

shown in Fig. 6. Therefore, this manufacturing method is also not suited to large-scale mass production.

Issue (3) is integration of the magnets and the yoke with the stainless frame-reinforcing component in order to configure the rotor. To realize an assembly structure with dimensional tolerance, a layout that resembles parquetry with gaps must be used, so bonding is performed by filling the gaps with adhesive. However, there are currently no known adhesive materials suited to the conditions of elastic deformation of the frame when rotating at high speed and rotor temperature rise during operation. This might result in a motor that is unsuitable for automotive applications.

### 3.2. Method of Addressing Rotor Issues

To address these three issues, the rotor yoke was formed by rolling laminate using a notching press, and then the magnets were inserted and integrated with the magnetic circuit. In addition, a multi-ring press-fit structure was developed that secures the outer circumference by compression using aluminum die-casting, and then interference-fits the outer ring that provides the centrifugal force strength.

#### 3.2.1. Laminated steel roll yoke

The rotor yoke was changed from the prototype shape used to confirm the increase in torque density to a shape with chamfered corners that contact the auxiliary magnets so that the back EMF voltage waveform more closely approximated a sine wave. The notching press rolling-laminate manufacturing method was then used to create a shape based on this yoke shape that has the minimum necessary bonding locations and hooks to hold the auxiliary magnets (Fig. 7 right side).

As shown in Fig. 8, the yoke on both sides is formed in a single process by rolling up hoop-shaped materials of the rotor yoke width that have had the magnet and nonmagnetic pin insertion spaces cut out by a punch. The inner circumference is short and the outer circumference is long, and punching and rolling are performed continuously while precisely controlling the feed rate (pitch) for each punch so that the main magnet insertion spaces and the yoke have a fan shape. Figure 9 shows a photograph of the yoke formed by rolling laminate.

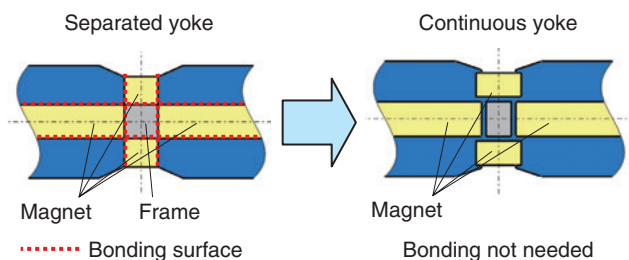


Fig. 7 Change in yoke structure

#### 3.2.2. Multi-ring press-fit structure

A multi-ring press-fit structure was designed that broadly comprises the three layers of the abovementioned yoke formed by the rolling laminate, and inner and outer rings made of nonmagnetic materials. This secured rotational strength and stiffness.

Figure 10 shows the assembly process. An aluminum layer is applied to the surface of the outer circumference of the laminated yoke by aluminum die-casting before the final process to press fit (shrink fit) the outer ring. This provides the following effects:

- (1) Fixing of the laminate by compression
- (2) Fixing of the magnets by deformation of the outermost layer (anchor effect)
- (3) Easy control of the outer diameter dimension by machining the aluminum layer

These effects enabled securing of rotational strength, thrust stiffness, and the surface pressure of each layer required for torque transmission. This enabled manufacture of a rotor that does not need to rely on adhesive material strength.

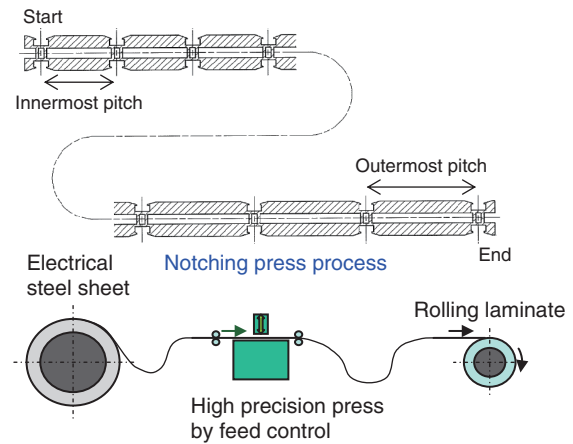


Fig. 8 Yoke manufacturing process using notching press

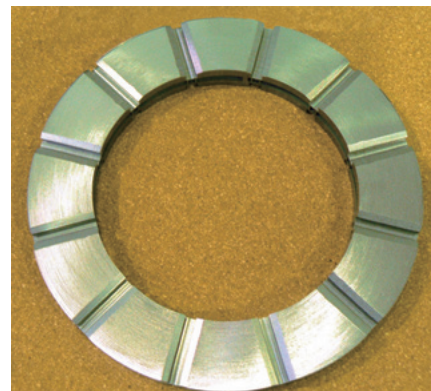


Fig. 9 Laminated roll yoke

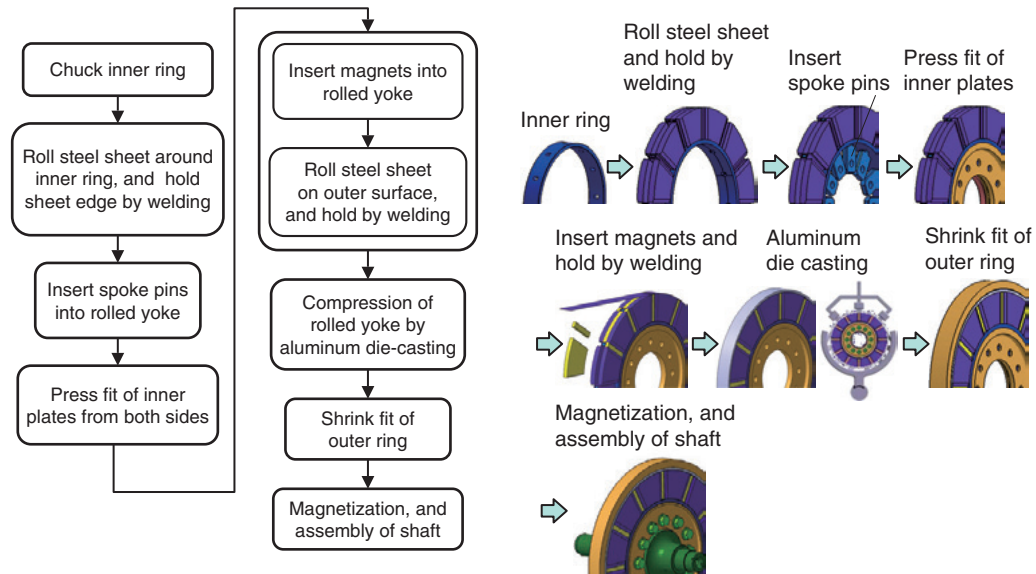


Fig. 10 Rotor assembly process

### 3.3. Confirmation of Strength and Stiffness

Figure 11 shows the rotational strength, Fig. 12 shows the thrust stiffness, and Fig. 13 shows the torsional stiffness, measured using the prototype multi-ring press-fit rotor.

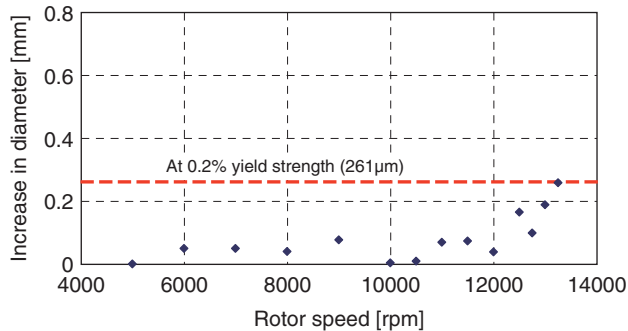


Fig. 11 Rotational strength

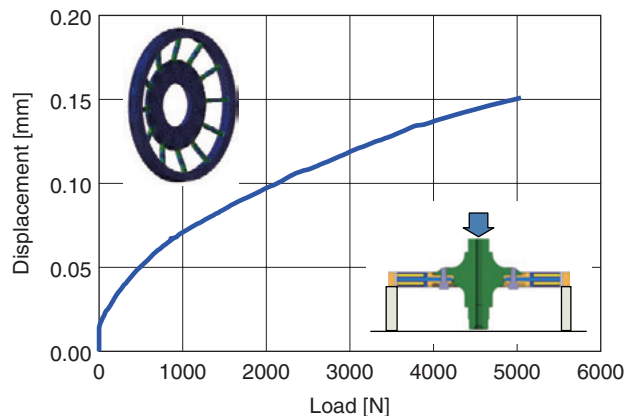


Fig. 12 Thrust stiffness

First, regarding the rotational strength, the amount of outer diameter deformation of the stainless outer ring was measured to confirm the rotor speed at which the elastic deformation range is not exceeded. The results showed that elastic deformation was within the allowable range even at 12000 rpm, which is 1.85 times the hypothesized maximum rotor speed of 6500 rpm. This confirmed that sufficient strength can be secured.

Next, the thrust stiffness was confirmed by measuring the displacement when the outer ring is fixed and thrust force is applied to the shaft (equivalent to a concentrated load). The displacement at the maximum thrust force of 5000 N was 0.15 mm, which is sufficiently small relative to the minimum air gap of 0.6 mm. In addition, the thrust load acting on the rotor due to the magnetic attraction

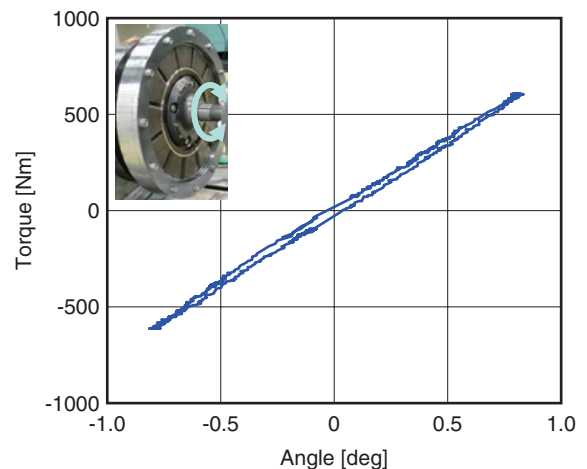


Fig. 13 Torsional stiffness

Table 2 Comparison of losses

|                 | Bonded rotor | Rolled rotor |
|-----------------|--------------|--------------|
| Iron loss [W]   | 692          | 630          |
| Copper loss [W] | 483          | 518          |
| Motor loss [W]  | 1175         | 1148         |

force acts as a distributed load on the yoke surface. Therefore, CAE calculations show that displacement can be suppressed to 62% of that when thrust force is applied to the shaft. This confirmed that sufficient stiffness is secured.

Regarding stiffness relative to torsional torque, discontinuous behavior did not occur even at alternating torque of 600 Nm, which is three times the maximum torque of 200 Nm. This confirmed that the new rotor also has sufficient stiffness and strength with respect to torque transmission.

#### 3.4. Confirmation of Torque and Efficiency

Figure 14 compares the torque characteristics per unit volume of developed multi-ring press-fit rotor with enhanced rotor structure reliability and the parquetry structure rotor. The back EMF voltage is 11% lower due to the magnetic flux short-circuit from the couplings used to create a rolling-laminate yoke and the reduced magnet surface area due to the smaller auxiliary magnet size. The torque characteristics vary approximately in proportion to this degree of voltage drop. Figure 14 shows that the double stator axial gap-type motor has a torque density 1.34 times that of the 2006 model year CIVIC IMA motor. In addition, Table 2 compares the various motor losses under typical operating conditions. The increase in the copper loss and the decrease in the iron loss due to the lower torque constant cancel each other out, confirming that the motor loss is approximately the same.

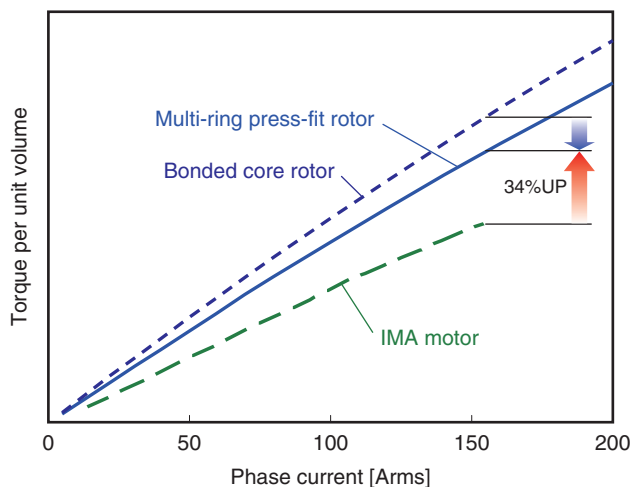


Fig. 14 Comparison of current-torque characteristics per unit volume

## 4. Conclusion

Research was conducted on a double stator axial gap-type motor that has two torque-generating surfaces with the aim of increasing the motor torque density. The rotor used an IPM-type structure like conventional radial gap-type motors, and an actual prototype motor was manufactured and performance was evaluated. The results confirmed that the torque density per unit volume increased compared to that of a radial gap-type motor.

In addition, axial gap-type motors have issues with rotor strength and stiffness. Therefore, a highly reliable rotor structure was designed by combining an electrical steel rolling-laminate yoke, aluminum die-casting, and nonmagnetic rings, and the effects were confirmed in an actual motor.

## References

- (1) Aydin, M., Huang, S., Lipo, T. A.: Axial Flux Permanent Magnet Disc Machines: A Review, In Conf. Record of SPEEDAM, p. 61-71 (2004)
- (2) Halbach, K.: IEEE, Trans., NS26 (1979), 3882, NIM169 (1989) 1 NIM 187 (1981) 109 NIM198 (1982) 213
- (3) [http://www.pasj.jp/web\\_publish/lam28/proceedings/WP-45.pdf](http://www.pasj.jp/web_publish/lam28/proceedings/WP-45.pdf)
- (4) Morimoto, S., Ueno, T., Takeda, Y.: Wide Speed Control of Interior Permanent Magnet Synchronous Motor, Trans. of IEEE Japan, Vol. 114-D, No. 6, p. 668-673 (1994)
- (5) JIS G 4304 : 2005, Hot-rolled stainless steel plate, sheet and strip

#### ■ Author ■



Shoei ABE



Junji INOUE



Satoshi ISHIKAWA

# Operability Enhancement of Steering Wheel Switch for Audio Remote Control

Masaaki ABE\*

## ABSTRACT

A steering wheel switch for audio remote control with a new shape has been developed with the aim of enhancing operability of the switches and addressing issues related to advancing functions of the on-board equipment in the new model CIVIC. The switch operation time and the magnitude of the effect on steering while driving were measured, and a comparison made with the former switch confirmed the operability of the new model switch. The effect on steering while driving was reduced, and the operating load of the new model switch is lower than that of the former switch.

## 1. Introduction

The functions of equipment installed on-board automobiles have been increasing rapidly in recent years, resulting in an increase in the number of switches that control operations. The question of how to make the kinds of switches that control these functions easier to operate has recently been a common issue in the development of automotive on-board equipment.

The new model CIVIC has introduced a new multi-information display (MID) that allows the driver to select display information from a variety of different kinds of information so as to make use of multiple functions. In order to operate the MID, a steering wheel switch for audio remote control was developed that included the previous audio remote control function with information display switching. The operability of the switch that makes it possible to operate it without looking was sought to enhance the operability of this operating switch during driving by focusing in particular on the human tactile capability and conveying the different operations to the driver by distinguishable shapes. This paper reports on the design concept and operability evaluation of the new switch.

## 2. Design Concept

In the course of this development, three aims were pursued in order to achieve momentary cognition and intuitive operation, which are the concept for Honda

interfaces: (1) Heightened operability during driving (allows for momentary operation of the switch and has little impact on driving); (2) keeps misoperation of audio devices down to a minimum; and (3) allows switch operation without confusion, even by users of previous systems.

### 2.1. Basic Structure and Target Requirements

The operating switch for the audio remote control has conventionally featured with three functions on the steering wheel: (1) volume adjustment, (2) changing channel/track, and (3) audio source switching. In order to help prevent confusion for users of previous systems, it was necessary to provide the new CIVIC with the same equipment. The ordinary switch configuration would allocate these three functions to three switches on the middle part of the steering wheel (Fig. 1(a)). The points to be considered when designing these switches are that they have little impact on driving operations with a shorter operation time and that misoperation of the switches is limited, while maintaining a similar form with conventional equipment.

It is important to place these switches at locations on the steering wheel where the fingers can reach them easily in order to shorten the operation time. When placing the switches, if they are located too close, then they will impinge on the steering wheel grip. This will not only have an impact on steering operation, it will also introduce the likelihood that the driver will press a switch inadvertently while steering. In this case, it becomes necessary to install a guard near the switch or use some other such means to help

\* Automobile R&D Center

keep the switch from being pressed mistakenly.

Clustering the individual switches together in one location can also be effective in shortening the operation time. However, this also introduces the likelihood that adjacent switches could be pressed at the same time, so it necessitates measures to prevent mistaken switch operation. The individual switches are ordinarily placed apart from each other, or separator ribs are placed between or near the switches to help keep them from being pressed mistakenly.

These methods of trying to help prevent misoperation will to some extent interfere with the range of switch operation from searching for the switch to pressing it, thus inviting some reduction in operability. Consequently, these measures to shorten the operation time and measures to help prevent misoperation are ordinarily in a trade-off relationship. The new model switch achieves a shorter operation time than previous switches while having a shape that also acts to help prevent misoperation.

Figure 1 shows enlarged images of the former and new model switches. The new model switch is placed so as to

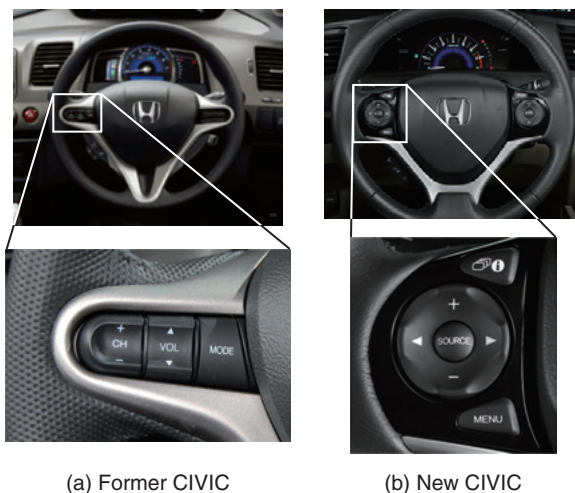


Fig. 1 Images of steering switch

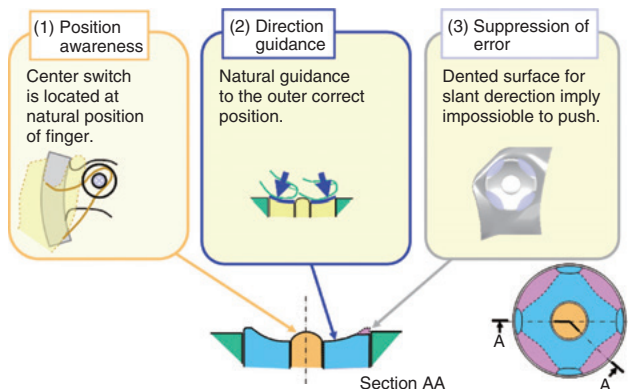


Fig. 2 Ergonomic design of new switch

be naturally at the center of the thumb's location when the steering wheel is held horizontally (in a neutral position), and the switch is given a spherical shape. This makes the center of the whole switch (the home position) clearly identifiable even without looking (Fig. 2(1)).

Of the switches at top and bottom and on the left and right sides, the top and bottom are for volume and the left and right are for channel/track selection. This layout makes it possible to readily determine the function by changing the movement of operation from the center to top-and-bottom or left-and-right. Moving from the center toward the periphery, the switch surface rises up higher. When the switch is pressed, therefore, the load naturally tends to be directed toward the periphery so that operability is enhanced while the likelihood of misoperation from pressing the center of the switch is limited (Fig. 2(2)).

It is a feature of the switch that the protrusion of the surface in the four directions and the depression in diagonal directions make it possible to distinguish momentarily by the tactile sense of the thumb placed on the switch in which directions the switch can be pressed and which directions it cannot (Fig. 2(3)). The way the switch shape is devised also made it possible to reduce the overall size of the switch and minimize impact on the steering grip shape.

### 3. Operation Time Measurement Method using Driving Simulator

In order to perform accurate measurements of the switch operation time that indicate operability performance, switch operation testing on a driving simulator that models the driving environment was utilized as the main performance measurement method.

Figure 3 shows the driving simulator used for these tests. It has a driver's seat installed indoors with a screen that shows a 180° field of view of driving scenes. It provides the test subject with simulated reaction force from the steering wheel and brakes.

Measurement of switch operation time while driving was done by the so-called dual-task technique<sup>(1)</sup> because driving tasks and switch operation tasks are constantly performed in an overlapping manner. Measurement by the dual-task technique takes the secondary task (in this case, the driving operation) to be an interference load with respect



Fig. 3 Image of driving simulator

to the main task (the switch operation), so that the balance of loads becomes important. If the two are not in balance, then the secondary task ceases to function as an interference task with respect to the main task, with the result that measurement cannot be considered dual-task technique measurement. Driving in a driving simulator generally tends to be taken as imposing a somewhat lighter driving load compared to actual driving because the driver receives a smaller amount of information. In dual-task designs with tasks involving essentially small operation loads, such as the operation of steering wheel switches, overcoming this kind of characteristic of the driving simulator becomes an issue. For the present research, the driving environment was configured to increase the driving load on the driving simulator so as to approximate the load of actual driving more closely by even a small margin. Specifically, the simulation was configured with the task of following a vehicle to the front, and the driver is given instructions such as to maintain a fixed distance from the vehicle in front.

While driving over the set course, the driver is given voice instructions once every six seconds to perform a random selection from five different tasks (raising volume, lowering volume, selecting next channel/track, selecting previous channel/track, switching sound source). The operation is initiated on hearing a buzzer that is sounded one second after the voice instruction, so that the time taken in hearing the voice instructions will not be included in the operation time. The time up until the designated switch was pressed was measured as the operation time. Single test subjects performed approximately 45 switch operations, the average of which was taken as the operation time.

#### 4. Test Results for Former and Developed Steering Switches

In order to confirm the operation time reduction effect of the developed switch, a comparison of operation time for the same task between the former and developed switches was made. Figure 4 shows the measurement results. Wilcoxon signed-rank test<sup>(2)</sup> results show that operation time was shortened approximately 0.15 sec, with 5% significance.

As concerns the influence on driving while operating the switch, the mean rate of variation in steering angle after operating the switch was taken as an index for the smoothness of steering operation and former and developed switches were compared. Figure 5 shows the comparison result of former and developed switches in terms of the mean rate of variation in steering angle while operating the switches. It is considered that, following the former switch operation, the drivers repeatedly perform sudden, minute steering corrections in brief periods of time to compensate for their departure from the intended running trajectory. In the case of the developed switch, on the other hand, the value goes down, indicating smoother steering operations are conducted.

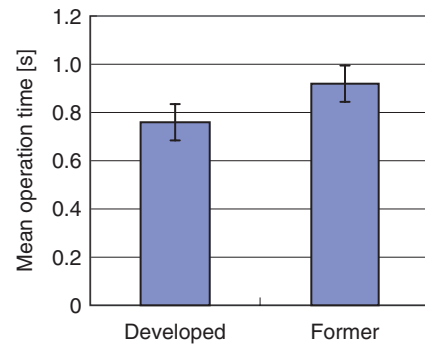


Fig. 4 Measurement result of mean operation time

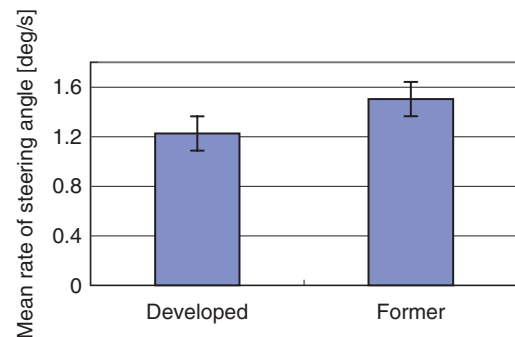


Fig. 5 Example of variation of steering angle

#### 5. Conclusion

A switch shape that enables momentary cognition using the tactile sense of the thumb and switch operation directions that achieve intuitive operation with a high functional distinguishability were used in enhancing the operability of steering wheel remote switches by designing a steering wheel switch with a new structure capable of shortening the operation time. This was confirmed to produce a reduction in the operation time by comparison with the former switch.

There are various different perspectives on the enhancement of automotive Human Machine Interface (HMI). In addition to operability while driving, there are the issues of ease of comprehending the switch when operating it for the first time, the layout that enables operation without confusion among multiple functions, and marking methods that indicate function content in an easily readable manner. Comprehensive enhancement of all these is necessary. The author hopes to proceed with HMI development by using human characteristics effectively in devising combinations of methods at multiple levels to reduce cognition and operation load and resolving these issues in an HMI that can be understood more intuitively and operated more instantaneously.

## References

- (1) Uchiyama, Y., Kojima, S., Hongo., T., Terashima, R., Wakita, T.: The Measurement of the Spare Mental Capacity of Automobile Drivers by a Dual Task Method, Proceedings of Human Interface Society, p. 617-620 (1999)
- (2) Wilcoxon, F.: Individual comparisons by ranking methods, Biometrics Bulletin, Vol. 1, p. 80-83 (1945)

■ Author ■



Masaaki ABE

# Development of Nitrocarburized High-strength Crankshaft with No Rare Metals Added

Yoshihiro TAKITANI\*

Isamu SAITO\*

Takahiro MIYAZAKI\*\*

Keiichiro KAMIYA\*\*

## ABSTRACT

Conventional nitrocarburized high-strength crankshafts secure high fatigue strength through a strengthening mechanism using the rare metal molybdenum (Mo). In contrast, a crankshaft with high fatigue strength equal to that of the conventional material was developed without the addition of rare metals by using copper (Cu) and chromium (Cr) and optimizing the material design.

Cr increases the hardness near the surface layers during nitrocarburization, which effectively increases the strength. However, in this case cracking easily occurs when plastic deformation is applied to straighten crankshaft bending that occurs in the nitrocarburization process, which has made it a challenge to actively use Cr. The developed material combines hardening by Cr with ferrite solid solution strengthening by Cu, and enhances straightenability by suppressing the Cr content required for strengthening. This simultaneously achieved both high fatigue strength and good straightenability without the addition of rare metals.

## 1. Introduction

Recently the conflicting characteristics of higher output power and environmental performance are being simultaneously demanded of automobile engines. In particular, growing environmental needs are increasing demands for enhanced fuel economy. As a result, crankshafts, which are engine components, are tending to become lighter in weight and shaft diameters are also decreasing in order to reduce the sliding resistance between the crankshaft and the bearings. This reduces stiffness, which increases the load on the crankshaft, so high fatigue strength is required. Crankshafts are generally formed by hot forging or casting, and hardening processes such as induction hardening, surface rolling, or nitrocarburization are used and residual compressive stress is applied to increase the fatigue strength<sup>(1)-(5)</sup>.

Various methods have been used to increase the hardness of nitrocarburized high-strength crankshafts. For example, the nitriding elements aluminum (Al) and chromium (Cr) were added to steel with a relatively high carbon content of 0.45 mass% or more on the assumption that normalization is performed. Currently, the rare metal molybdenum (Mo) is added to steel with a relatively low carbon content of 0.35 mass% in order to secure high fatigue strength and

straightenability without normalization, and also to enhance machinability by suppressing the material hardness<sup>(6)</sup>. Adding Mo enables the achievement of high fatigue strength and good straightenability even for non-normalized steel with a relatively low carbon content. However, rare metals easily become objects of speculation so prices are unstable, and expanding demand in developing countries is predicted to further destabilize supply<sup>(7)</sup>. Therefore, this development investigated a means of omitting Mo in the electric furnace manufacturing method.

## 2. Development Goals

The crankshafts used in Honda automotive gasoline engines are formed by forging, machined, and then nitrocarburized to enhance the fatigue strength characteristics. A crankshaft that satisfies the following characteristics was investigated with the target of realizing a high-strength material.

- (1) No need to add rare metals
- (2) High fatigue strength (equal to that of the conventional high-strength material)
- (3) Good straightenability (equal to that of the conventional high-strength material)

\* Automobile R&D Center

\*\* Daido Steel Co., Ltd.

### 3. Component Design

#### 3.1. Approach toward Component Design

Bending fatigue strength for nitrocarburized crankshafts that assume straightening is secured by increasing the hardness near the surface layers (hereafter “surface hardness”) during nitrocarburization. However, increasing the surface hardness reduces the straightenability, so it is important to create a component design that simultaneously realizes both bending fatigue strength and straightenability. The conventional high-strength material uses the rare metal Mo to increase the surface hardness to an appropriate level during nitrocarburization, and generation of Mo clusters also strengthens the ferrite. This simultaneously achieves both high fatigue strength and good straightenability. In addition to Mo, surface hardness can also be increased during nitrocarburization by adding elements such as Cr, vanadium (V) or Al. However, while these elements effectively increase strength, they also lower straightenability, which makes active use a challenge for crankshafts that assume straightening<sup>(6)</sup>. As an example of an element that increases surface hardness, Table 1 shows an actual case with 0.15 mass% Cr added instead of Mo to steel with a carbon content of 0.36 mass%, which is equal to that of conventional high-strength materials. This shows that while fatigue strength equal to that of the conventional high-strength material can be achieved, the straightenability drops. Straightenability here was evaluated by applying bending deformation to a test piece and measuring the strain value (cracking limit strain) at which cracking occurred<sup>(8)</sup>. This shows that high fatigue strength and good straightenability equal to those of the conventional material cannot be simultaneously achieved simply by adding Cr and performing nitrocarburization to increase the hardness. In order to use the strengthening effects of Cr, V and Al to omit Mo, it is thought that some means must be found to strengthen the ferrite, which is the other effect of Mo, and to suppress the required content of these elements. Therefore, the elimination of Mo and substitution of other elements with ferrite strengthening functions was investigated as a means of addressing these issues in the developed material.

#### 3.2. Ferrite Area Fraction Setting

Ferrite strengthening was investigated by setting the ferrite area fraction of the metallic structure of a crankshaft made of ferrite and pearlite, and then investigating strengthening methods. The ferrite area fraction is greatly influenced by the C and manganese (Mn) contents, so first the quantities of these two elements were set. Reducing the C content is said to be an effective means of enhancing straightenability without lowering the bending fatigue strength<sup>(6)</sup>. This is thought to mean that straightenability can be enhanced by increasing the ferrite area fraction, even at the same hardness. Figure 1 shows the results of investigating the straightenability of steel with the C and Mn contents adjusted so that surface hardness is

Table 1 Bending fatigue strength and cracking limit strain of Cr added steel after nitrocarburization (Conventional steel = 100)

|                | Bending fatigue strength | Cracking limit strain |
|----------------|--------------------------|-----------------------|
| Cr added steel | 106                      | 51                    |

approximately the same after nitrocarburization and the ferrite area fractions are adjusted to 63.2%, 41.5% and 11.1%, respectively. These results confirmed that even at approximately the same surface hardness, increasing the ferrite area fraction increases the cracking limit strain value and enhances straightenability. This means that setting a high ferrite area fraction is an effective means of obtaining good straightenability. However, crankshafts also need wear resistance against the oil seal, and reducing the C and Mn contents more than is necessary to increase the ferrite area fraction lowers the hardness, so wear resistance becomes an issue. Figure 2 shows the results of investigating the relationship between the internal hardness and wear resistance of crankshafts with different internal hardness manufactured by adjusting the C and Mn contents. The internal hardness needed to secure the same

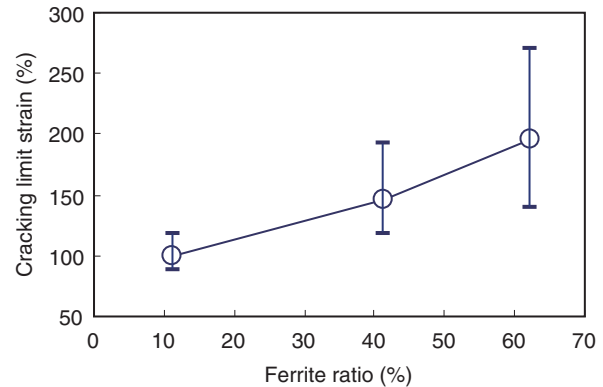


Fig. 1 Relationship between ferrite ratio and cracking limit strain (Ferrite ratio 11.1% data = 100)

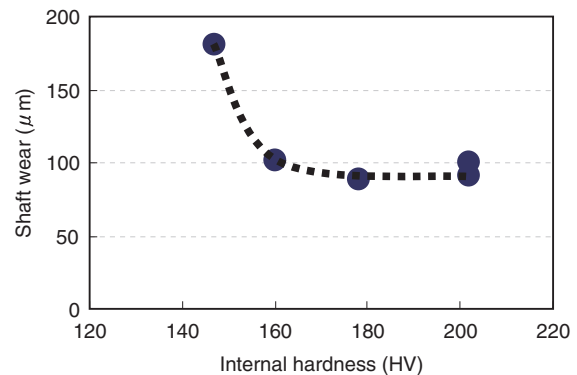


Fig. 2 Relationship between internal hardness and wear loss at oil seal section of crankshaft

wear resistance as the conventional crankshaft is HV 160 or more. Figure 3 shows the relationship between the C and Mn contents and the ferrite area fraction when the C and Mn contents are adjusted to change the ferrite area fraction. The aim is to reduce the C and Mn contents and secure the highest ferrite area fraction possible, but when the ferrite area fraction exceeds 70%, the hardness falls below the required hardness of HV 160. In consideration of these results, the ferrite area fraction was set between 50% and 70%, and 0.30 mass% of C and 0.80 mass% of Mn were used as the typical values for the developed material.

### 3.3. Ferrite Solid Solution Strengthening

As a substitute for the ferrite strengthening effects of Mo, the use of ferrite solid solution strengthening by other elements was investigated. The aim was to suppress the required content of Cr and the like and enhance straightenability. The cluster generation effects of Mo strengthen the ferrite evenly from the surface to the interior. Therefore, the substitute elements should also strengthen the ferrite evenly from the surface to the interior, without affecting the change in hardness before and after nitrocarburization, by generating nitrides and other compounds during nitrocarburization that increase the surface hardness and suppressing the hardened layer depth. Various elements known as typical ferrite solid solution strengthening elements were added to steel having the base composition shown in Table 2, and nitrocarburization was performed. Figure 4 plots the difference in the hardness at a depth of 50  $\mu\text{m}$  of the base composition steel and the steel with each element added. The amount of increase in hardness at a depth of 50  $\mu\text{m}$  did not depend on the added Cu and nickel (Ni) contents and was relatively constant, which indicates that these elements are unaffected by nitrocarburization and can be used to strengthen ferrite<sup>(9)</sup>. Compared to Ni, Cu is an element that can be easily used by adjusting the scrap materials used as raw materials. Therefore, since it is thought that the same effects can be obtained from either Cu or Ni, the use of Cu was investigated.

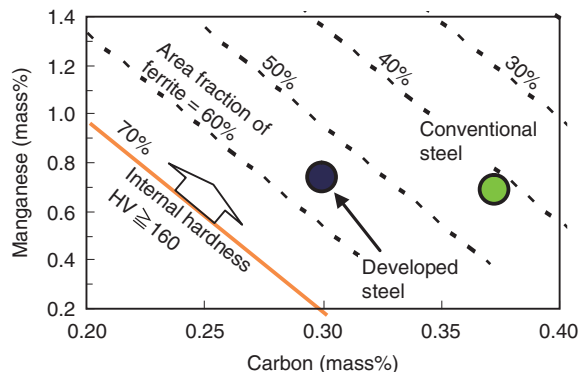


Fig. 3 Relationship between added alloying element contents and area fraction of ferrite

Table 2 Chemical composition of original test steel

|                     | C    | Si   | Mn   | Cu   | Ni   | Cr   |
|---------------------|------|------|------|------|------|------|
| Original test steel | 0.36 | 0.07 | 0.65 | 0.10 | 0.10 | 0.15 |

Figure 5 shows the relationship between surface hardness and bending fatigue strength when the surface hardness is varied by adjusting the C, Mn, Cr and other alloying element contents, assuming a ferrite area fraction between 50% and 70%. A hardness of HV 300 is required to secure bending fatigue strength equal to that of the conventional material. Figure 6 compares the

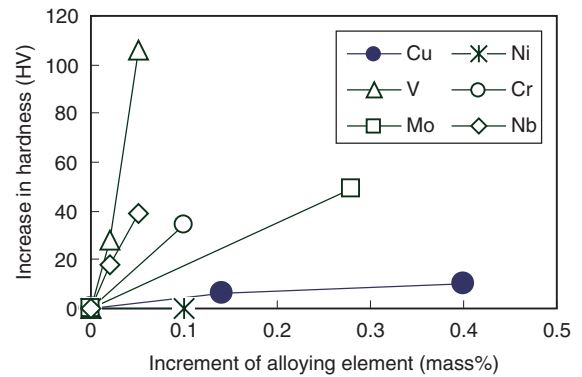


Fig. 4 Influence of alloying elements on surface hardness after nitrocarburization

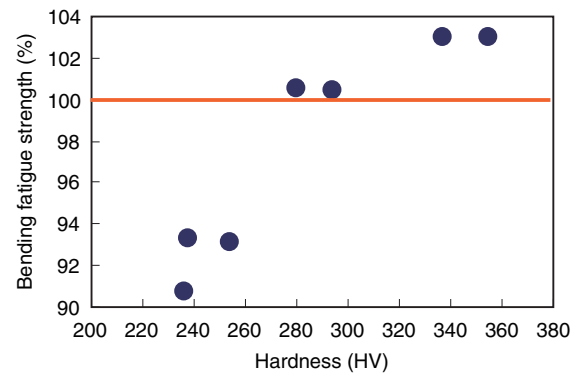


Fig. 5 Relationship between hardness and bending fatigue strength (Conventional steel = 100)

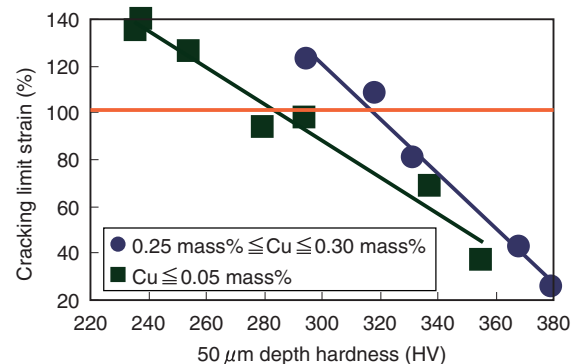


Fig. 6 Relationship between 50  $\mu\text{m}$  depth hardness and cracking limit strain

straightenability for a Cu content of 0.05 mass% or less and a Cu content between 0.25 mass% and 0.30 mass%. This shows that when the Cu content is between 0.25 mass% and 0.30 mass%, the straightenability at the same hardness is enhanced, and straightenability equal to or greater than that of the conventional material is secured at the required hardness of HV 300. In addition, the required Cu composition range can be secured without addition by adjusting the scrap materials. Therefore, Cu between 0.25 mass% and 0.30 mass% was used as the solid solution strengthening element.

### 3.4. Strengthening by Nitrocarburization

Cr, V, Al<sup>(6)</sup> and other elements are known to increase the surface hardness during nitrocarburization, and Cr can be secured without addition by adjusting the scrap materials used as the steel raw materials. Figure 7 shows the change in the surface hardness and the internal hardness after nitrocarburization for different Cr contents. Cr has almost no influence on internal hardness, so the surface hardness can be controlled during nitrocarburization without affecting the forgeability and machinability. For this reason, Cr was selected as the element used to increase the surface hardness during nitrocarburization. Figure 8 shows the relationship between the Cr content and the surface hardness assuming a Cu content between 0.25

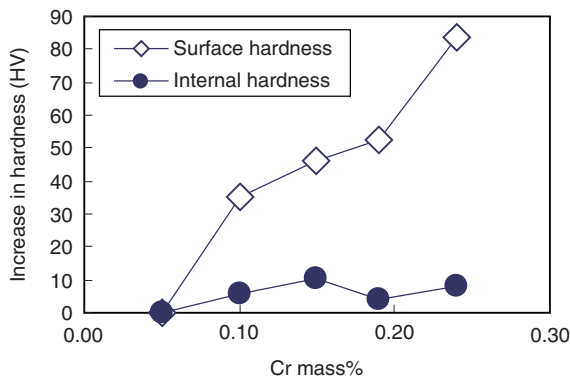


Fig. 7 Influence of Cr on surface hardness and internal hardness after nitrocarburization

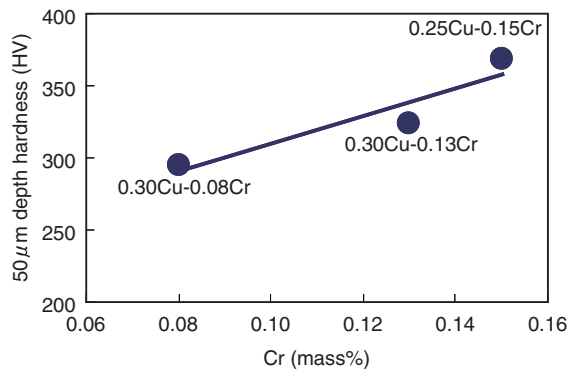


Fig. 8 Relationship between Cr mass% and 50 μm depth hardness

mass% and 0.30 mass%. This shows that a Cr content of 0.10 mass% or more is required to achieve the hardness of HV 300 needed to secure bending fatigue strength equal to that of the conventional material. A higher Cr content than is necessary reduces the straightenability, so a Cr content of 0.12 mass% was used as the typical value for the developed material in consideration of the balance between bending fatigue strength and straightenability.

## 4. Strengthening Mechanism

The ferrite strengthening mechanism of the developed material is solid solution strengthening by Cu<sup>(9)</sup>. In the case of carbon steel, the strengthening mechanism of nitrocarburization is generally said to be nitrogen solid solution strengthening, but Cr is also known as an element that forms nitrides<sup>(10)</sup>. Therefore, investigation was conducted to confirm whether the strengthening mechanism during nitrocarburization of steel with a relatively low Cr content of approximately 0.10 mass% such as the developed material is nitrogen solid solution strengthening or nitride precipitation strengthening. When the solid solution strengthening mechanism is dominant, the lattice constant of ferrite is thought to vary according to the solid solution nitrogen concentration. Figure 9 shows the results of measuring the nitrogen concentration using an electron probe microanalyzer and measuring the lattice constant of the ferrite using X-ray analysis, at the depths of 30 μm, 100 μm and 1000 μm from the surface. This shows that the lattice constant is nearly constant and independent of the nitrogen concentration, which infers that solid solution strengthening has little contribution. Next, Fig. 10 shows the analysis results at a depth of 50 μm from the surface obtained using a field emission transmission electron microscope. Here, large numbers of precipitates with a crystalline structure similar to chromium nitride (CrN) can be confirmed. Figure 11 shows the results of analyzing these precipitates using energy dispersive X-ray spectrometry. The precipitates contain Cr, and are thought to be CrN precipitates. Therefore, CrN precipitation strengthening is thought to be more dominant than nitrogen solid solution strengthening as the strengthening mechanism

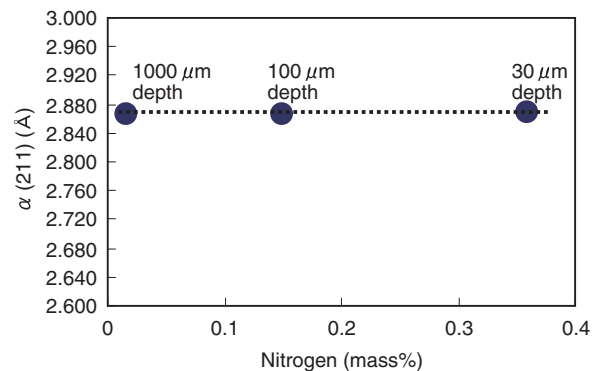


Fig. 9 Lattice constant of ferrite after nitrocarburization

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

|                    | C    | Si   | Mn   | P     | S    | Cu   | Cr   | Mo   |
|--------------------|------|------|------|-------|------|------|------|------|
| Developed steel    | 0.30 | 0.08 | 0.80 | 0.013 | 0.05 | 0.25 | 0.12 | -    |
| Conventional steel | 0.37 | 0.20 | 0.82 | 0.015 | 0.05 | -    | -    | 0.16 |

during nitrocarburization of steel with a relatively low Cr content of approximately 0.10 mass%. This precipitation strengthening mechanism is thought to be the reason why the straightenability drops when strengthening is performed using only Cr. Precipitation strengthening is thought to lower the crack propagation energy<sup>(11)</sup>, and this is thought to be the reason for the marked drop in straightenability. However, by using Cu instead of Mo to obtain ferrite strengthening effects for the developed material, it is thought that the Cr content required for strengthening could be reduced, which enabled simultaneous achievement of both high fatigue strength and straightenability.

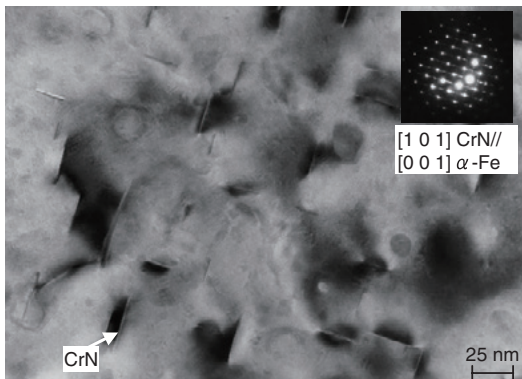


Fig. 10 Transmission electron micrograph taken from diffusion layer of steel containing 0.10 mass% Cr

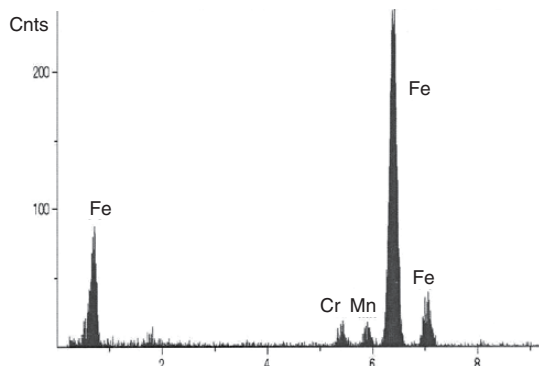


Fig. 11 Energy dispersive X-ray spectrometry analysis of precipitate

## 5. Crankshaft Characteristics

Table 3 lists the typical composition values for the developed material set according to the above investigation results and the typical composition values for a conventional high-strength crankshaft. Compared to the conventional

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

|                 | Bending fatigue strength | Cracking limit strain |
|-----------------|--------------------------|-----------------------|
| Developed steel | 102                      | 99                    |

material, the developed material has a chemical composition that omits Mo but also uses Cu and Cr within the ranges that can be secured by adjusting the scrap materials, which makes the addition of rare metals unnecessary. Next, a crankshaft for a 3.5-liter V6 engine was manufactured and the cross-section hardness was measured to confirm the crankshaft characteristics of bending fatigue strength and straightenability. Figure 12 shows that the crankshaft surface hardness satisfied the required hardness of HV 300, and Table 4 shows that bending fatigue strength and straightenability equal to those of the conventional crankshaft were achieved.

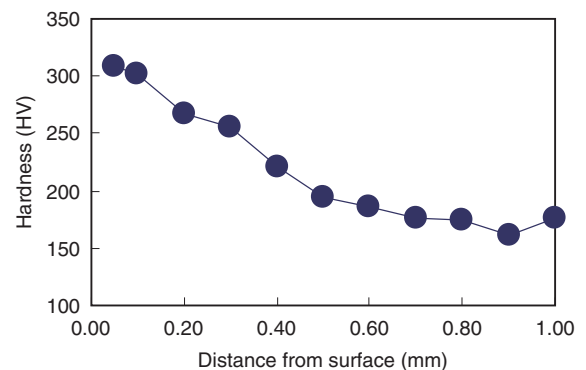


Fig. 12 Hardness profile after nitrocarburization

## 6. Conclusion

A high-strength crankshaft that does not require the addition of Mo or other rare metals was developed by manufacturing materials with a component design that optimizes the Cu and Cr contents, using the electric furnace manufacturing method. This reduced the cost by 2% and avoided the risk of future increases in the price of rare metals. This development clarified the following items.

- (1) The combination of precipitation strengthening by Cr during nitrocarburization and ferrite solid solution strengthening by Cu enables the achievement of high fatigue strength and good straightenability equal to those of the conventional high-strength material without

adding rare metals.

- (2) The Cu and Cr needed for strengthening are contained in the scrap materials used as raw materials, and the required quantities can be secured by adjusting the scrap materials.
- (3) CrN precipitation strengthening is the dominant strengthening mechanism instead of nitrogen solid solution strengthening when performing nitrocarburization with a Cr content of approximately 0.10 mass%.

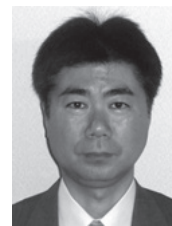
## References

- (1) Iwama, N.: Trend of Special Steel for Automotive Parts, JSAE, Vol. 55, No. 10, p. 64-68 (2001)
- (2) Suzuki, N., Tani, K.: Recent Trend of Nitrided Products, Journal of the Japan Society for Heat Treatment, Vol. 32, No. 5, p. 262-266 (1992)
- (3) Mori, M., Koumoto, T., Harikago, T., Owaki, S., Iwama, N.: Development of High Fatigue Strength Microalloyed Steel Crankshaft, JSAE Technical Paper, No. 47-00, p. 9-12 (2000)
- (4) Onda, H., Asai, T., Fujiki, K., Kudo, H., Murofushi, K.: Development of High-strength Induction Hardened Crankshaft, Honda R&D Technical Review, Vol. 15, No. 2, p. 175-182
- (5) Nagasaka, H., Mori, H., Araki, S., Ishibashi, Y., Hamada, T., Takami, M.: Fatigue Strength Improvement of Solid Type Crankshaft through Fillet Cold Rolling, Kobe Steel Engineering Reports, Vol. 48, No. 1, p. 68-71 (1998)
- (6) Asai, T., Takitani, Y., Sano, N., Matsumoto, H.: Development of Non-normalized Nitrocarburized High Strength Crankshaft, Honda R&D Technical Review, Vol. 19, No. 1, p. 89-96
- (7) Watanabe, Y.: Reserves and Production of Rare Metals and Prediction of Their Future Demands and Supplies, Journal of Society of Automotive Engineers of Japan, Vol. 63, No. 11, p. 14-18 (2009)
- (8) Matsumura, Y., Kurebayashi, Y., Konagaya, D., Mizuno, K.: Development of Nitrocarburizing Steel for Crankshafts, SAE Technical Papers, 1999-01-0601 (1999)
- (9) Hobo, M., Takada, K.: Effect of Small Amounts of Cu on Nitrocarburizing Characteristics for Medium-Carbon Steels, DENKI-SEIKO, Vol. 77, No. 1, p. 29-38 (2006)
- (10) Inoue, K., Matsumura, Y.: Influence of Alloying Elements on Hardness Distribution after Nitrocarburizing in Medium Carbon Steel, DENKI-SEIKO, Vol. 75, No. 1, p. 11-18 (2004)
- (11) Inoue, K., Nakamura, S.: Influence of C and V Contents on Mechanical Properties of Microalloyed Ferrite-Pearlite Steels, DENKI-SEIKO, Vol. 65, No. 1, p. 22-30 (1994)

## ■ Author ■



Yoshihiro TAKITANI



Isamu SAITO



Takahiro MIYAZAKI



Keiichi KAMIYA

# Visualization of Oil Behavior when Piston Pin Noise Occurs

Takashi KONDO\*    Hisayuki OHBAYASHI\*\*

## ABSTRACT

Piston pin behavior was measured on occurrence of conspicuous piston pin noise when using a semi-floating support system, and oil behavior around the piston pin was visualized.

Piston pin behavior was measured by the linkage method using a gap sensor. This clarified the behavior of the piston pin using a semi-floating support system when the noise occurred.

Visualization of the oil behavior was performed by the linkage method using optical fiber. Particle images of oil droplets were captured with a high-speed camera. Processing those images with particle tracking velocimetry (PTV) successfully vectorized the oil behavior inside the engine.

## 1. Introduction

As the development of hybrid, fuel cell, and other such new power plants progresses rapidly, development of conventional gasoline engines is also advancing in order to further enhance their performance. In addition to greater compactness and lighter weight, as well as greater fuel economy, higher power, and greater reliability performance, there is also demand for engines that are quieter and lower in cost. Major moving parts, in particular, are crucial parts that have a direct influence on those aspects of performance, and this is true of the pistons most of all. Piston support systems generally belong to two kinds, the full-floating support systems and semi-floating support systems. Mini-sized and compact vehicles more frequently adopt the semi-floating system due to its cost advantages. This support method, however, involves the issue of greater likelihood of piston pin noise occurring.

This issue is thought to arise because of differences in the clearance between each part in this support system. It is further known through experience, however, that lubrication conditions can also have an effect on the noise. Consequently, in order to reduce piston pin noise, it will be important to ascertain the mechanism that results in piston

pin noise and to grasp the oil behavior that is thought to be an influential factor.

## 2. Research Aims

It is crucial that the mechanism of piston pin noise be clarified in order to reduce the noise. As a first step to that end, the below actions were taken to grasp the phenomena that take place when piston pin noise occurs, which had not previously been made clear.

- (1) Measured piston and piston pin behavior
- (2) Visualized oil behavior

## 3. Test Engine

Table 1 shows the basic specifications of the test engine. A 1.5-L engine with the semi-floating support was used. In order to clarify the phenomenon of piston pin noise, the piston pin clearance was configured so that piston pin noise would be more likely to occur. The testing was then conducted with the engine operating in a state in which piston pin noise occurs. The test conditions, which are typical of that state, were that the engine was hot and running at 1850 rpm at no load.

\* Automobile R&D Center

\*\* Honda Foundry Co.,Ltd

Table 1 Engine specifications

| Item                            | Specifications                                |
|---------------------------------|-----------------------------------------------|
| Engine type                     | In-line 4-cylinder                            |
| Injection type                  | Port injection                                |
| Bore × stroke (mm)              | 73 × 89.4                                     |
| Displacement (cm <sup>3</sup> ) | 1497                                          |
| Compression ratio               | 10.4                                          |
| Valve train                     | SOHC i-VTEC                                   |
| Number of valves                | 4 per cylinder                                |
| Valve diameter (mm)             | IN: $\phi 28 \times 2$ EX: $\phi 23 \times 2$ |
| Gasoline                        | Regular (RON91)                               |
| Max. power (kW/rpm)             | 88/6600                                       |
| Max. torque (Nm/rpm)            | 145/4800                                      |

Table 2 Occurrence conditions of piston pin noise

|                  | (a) Steady operation | (b) After snap     |
|------------------|----------------------|--------------------|
| Frequency        | 2-3 kHz              | 2-3 kHz            |
| Engine load      | No load, Light load  | No load            |
| Timing           | Around BTDC20°       | Not only BTDC20°   |
| Expansion stroke | With combustion      | Without combustion |
| Imitative sounds | "Kata", "kara"       | "Jyara", "Jya-"    |

#### 4. Piston Support System and Piston Pin Noise

Figure 1 shows the piston support systems employed in Honda automobiles. There are two types, the full-floating and the semi-floating, and piston pin noise occurs primarily with the semi-floating type. The full-floating type has clearance between the piston pin and the connecting rod, which are both made of steel. The semi-floating type has clearance between a piston made of aluminum alloy and a piston pin made of steel. Consequently, the semi-floating type is considered to be more subject to piston pin noise because the clearance increases, especially when the engine is hot, due to the different linear thermal expansion coefficients.

Piston pin noise occurs during two engine operating states. The characteristics of each are shown in Table 2. Piston pin noise and the piston pin behavior accompanying

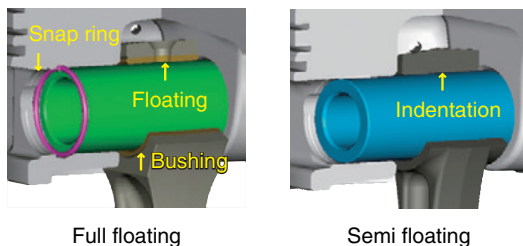


Fig. 1 Two types of floating support

it have been described in numerous reports of research and instances of measurement<sup>(1), (2)</sup>.

For this research, the focus was first placed on piston pin noise that occurs during steady-state operation (Table 2(a)). The various phenomena that occur when the noise occurs were measured and analyzed.

#### 5. Measurement of Piston and Piston Pin Behavior

##### 5.1. Overview of Measurement System

Figure 2 shows a diagram of the measurement system. The piston pin behavior was measured using two gap sensors (Model PU-02A by the Applied Electronics Corporation) affixed orthogonally to the piston pin boss. The sensors were of the eddy current type. They were made with a ceramic portion on the tip to allow a margin for trimming, enabling round chamfering along the piston pin hole after installation to the piston.

The speed and vibrational acceleration of the piston and piston pin were measured using accelerometers (Model 352C22 by PCB Piezotronics, Inc.). Since the piston pin is press-fitted into the connecting rod, the measurements were made on the connecting rod near the piston pin. The measurement output signal was fed to the outside for retrieval by a link mechanism<sup>(3)</sup>. The link had three joints and extended from the big end of the connecting rod with one end attached to the connecting rod and the other end to a case affixed to the engine. The path of the wiring leading out from the measurement point is indicated by the dotted arrow.

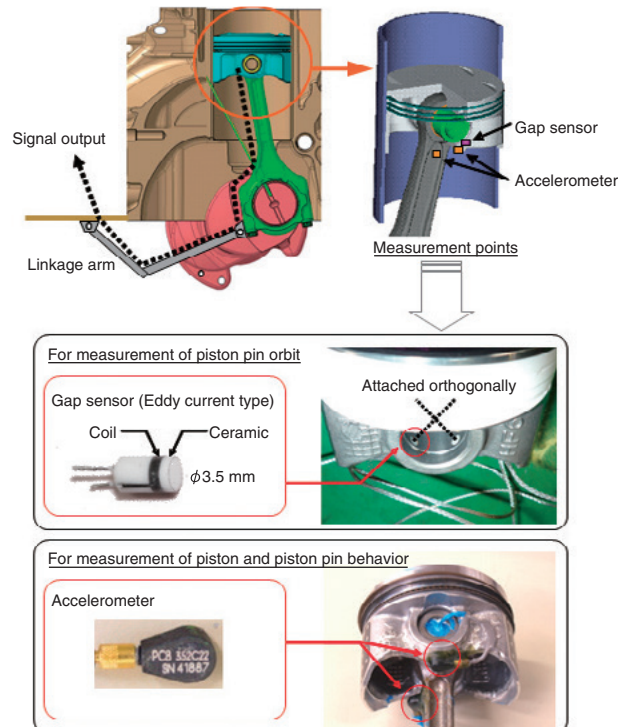


Fig. 2 Configuration of measurement system

## 5.2. Measurement of the Behaviors Involved

### 5.2.1. Piston and piston pin behavior in the vertical direction

Figure 3 shows the cylinder pressure together with the vertical component of the velocity and acceleration of the piston and piston pin when piston pin noise occurs. As shown in Table 2, piston pin noise occurs near BTDC 20°. Attention was therefore focused on this vicinity.

First of all, regarding the cylinder pressure, since the engine is operating under no load, the pressure changes from negative to positive near BTDC 40°. The pressure then increases further because of combustion.

At this time, the increase in cylinder pressure exerts a braking force against the vertical rise of the piston. Meanwhile, the relative velocity of the piston pin has increased so that the piston pin rises and hits the surface of the piston pin hole. The result is that the piston and piston pin impact against each other at a point from BTDC 30° to BTDC 20°. This impact phenomenon can also be discerned in the acceleration graph.

### 5.2.2. Piston pin axis center behavior

Next the behavior of the piston pin axis center inside the piston pin hole was measured in order to clarify the piston pin behavior when piston pin noise occurs. The results are shown in Fig. 4. The solid line drawn in the figure represents the axis center behavior from BTDC 90° to TDC. The open circles drawn on that line represent the position of the axis center at each 1° of crank angle.

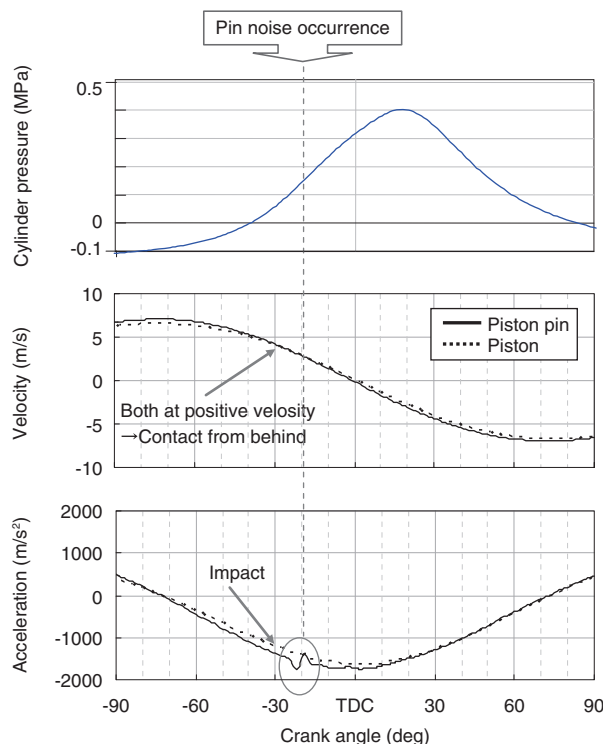


Fig. 3 Vertical behavior of piston and piston pin (1850 rpm, No load)

It can be inferred from the confirmation results in the previous paragraph that piston pin noise is a phenomenon of impact in a vertical direction between the piston and piston pin that occurs as the cylinder pressure increases. However, the results from measurement of piston pin axis center behavior made it clear that the piston pin is not just displaying simple vertical behavior.

It was observed that as the piston pin heads toward BTDC 20°, where piston pin noise occurs, it makes a large, circular motion inside the piston pin hole toward the counter-thrust side (this movement will be referred to hereafter as a circular motion) while rapidly traveling in an upward direction. Figure 5 shows the findings to this point organized in the form of a flow chart. The behavior of the

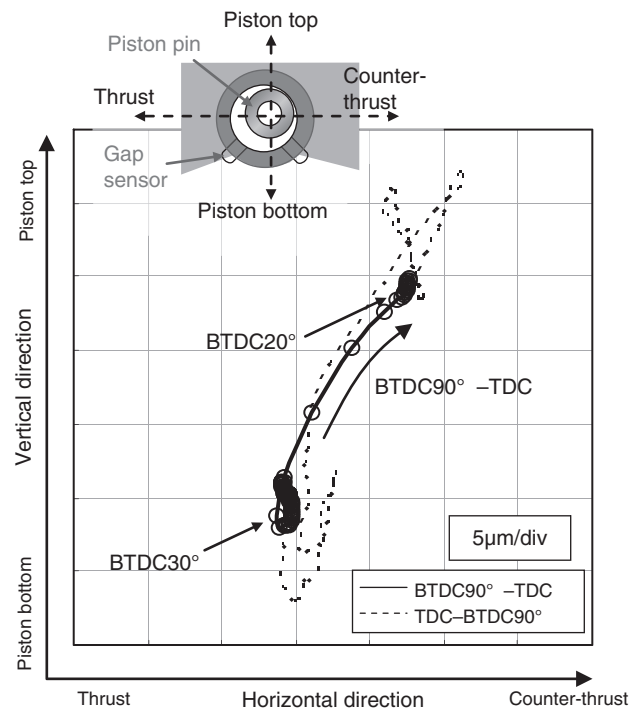


Fig. 4 Piston pin center orbit of semi floating system at steady operation (1850 rpm, No load)

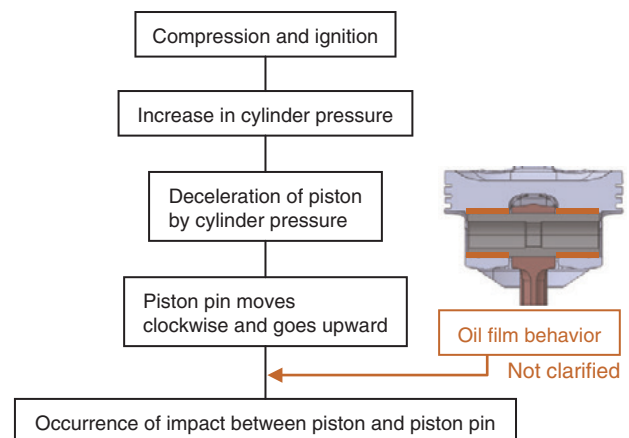


Fig. 5 Flow of piston pin noise phenomena

Table 3 Comparison of oil behavior visualizations

| Method               | Developed method                                                                                                                                                                                     | Conventional methods                                                                                                                                                                           |                                                                                                                                                                                                 |
|----------------------|------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|-------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
|                      | Direct view<br>by fiber scope PTV                                                                                                                                                                    | Direct view<br>(fluorescent paint)                                                                                                                                                             | Neutron radiography                                                                                                                                                                             |
| Observation point    | Inside of engine<br>Moving with linkage<br>Very near to target                                                                                                                                       | Outside of engine<br>View from static point<br>Visible via quartz/sapphire glass port                                                                                                          | Outside of engine<br>View from static point<br>Permeation by neutron beam                                                                                                                       |
| Difficulty           | Easy                                                                                                                                                                                                 | Easy                                                                                                                                                                                           | Difficult                                                                                                                                                                                       |
| Cost                 | Low cost                                                                                                                                                                                             | Low cost                                                                                                                                                                                       | Expensive                                                                                                                                                                                       |
| Safety consideration | None                                                                                                                                                                                                 | None                                                                                                                                                                                           | Radiation exposure                                                                                                                                                                              |
| Advantages           | <ul style="list-style-type: none"> <li>• Viewable from moving point</li> <li>• Viewable inside/reverse side of parts</li> <li>• No influence for flow</li> <li>• PTV possible by oil mist</li> </ul> | <ul style="list-style-type: none"> <li>• Easy to install</li> </ul>                                                                                                                            | <ul style="list-style-type: none"> <li>• Viewable wide area</li> </ul>                                                                                                                          |
| Disadvantages        | <ul style="list-style-type: none"> <li>• Depends on fiber scope spec (because of mass product model)</li> </ul>                                                                                      | <ul style="list-style-type: none"> <li>• Expensive fluorescent paint</li> <li>• Difficult to view inside/reverse side</li> <li>• Different deformation at window from actual engine</li> </ul> | <ul style="list-style-type: none"> <li>• Limited equipment only</li> <li>• Significant cost</li> <li>• Significant safety considerations</li> <li>• Depends on permeability of parts</li> </ul> |

piston and piston pin has become clear, but the state of the oil that is interposed between those parts has not been made clear. Therefore, this will be done in the next section.

## 6. Oil Behavior Visualization

The state of the lubrication when piston pin noise occurs will be made clear. It would be ideal to directly observe the oil inside the piston pin hole. Since there was no method for doing so, however, the state of the oil that flows into and out of the piston pin hole was observed by visualization and the state of the lubrication inside the piston pin hole was inferred from that.

### 6.1. Overview of the Measurement System

Particle tracking velocimetry (PTV) was used to measure oil behavior. Figure 6 shows the visualization of

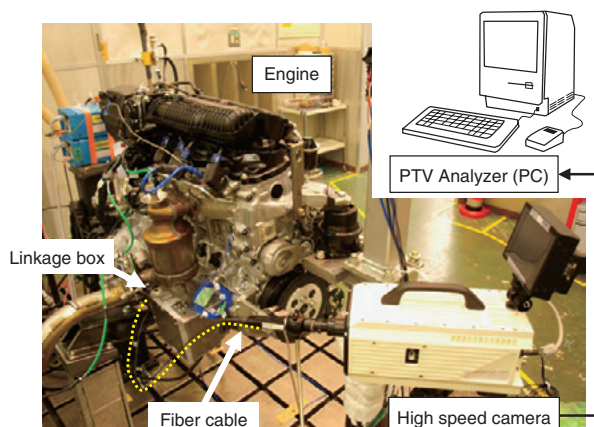


Fig. 6 Visualization and PTV system

oil behavior and the PTV system.

The interior of the engine was observed using a fiberscope mounted on the linkage mechanism. What was observed there was photographed with a high-speed camera (MEMRECAMfx K5 from nac Image Technology, Inc.) and made into particle images for use in PTV.

Table 3 presents a comparison of the measurements made by the fiberscope and linkage method applied here together with the PTV analysis and visualization methods of oil in the engine interior that have been reported to date<sup>(4), (5)</sup>.

The method adapted for use here has the following characteristics:

- (1) Having the fiberscope installed on the linkage enables visualization to be maintained unchanged, always following the visualization target at the same distance, even if the target is moving.
- (2) Imagery of visualized oil droplets can be speed-vectorized by PTV processing, allowing oil behavior in the engine interior to be ascertained.
- (3) It enables visualization of the reverse side of parts and other such areas that cannot be visualized through a viewing port installed in the cylinder block or other engine exterior wall.
- (4) Since no tracer is needed, engine damage due to deterioration of the oil or admixture of foreign matter can be avoided.
- (5) Lasers, optical systems, and other such equipment generally used for PTV are not needed, so the system is simple.

Since there were no instances of reports on this measurement method to date, the suitability of the device was considered.

Table 4 Specifications of fiber scope

| Consideration                   |          | Fiber specifications | Requirements        |
|---------------------------------|----------|----------------------|---------------------|
| (1) Oil proof                   |          | OK                   | Need oil proof      |
| (2) Heat resisting              |          | -10 to 80°C          | Up to 80°C          |
| (3) Minimum radius of curvature |          | R19 mm               | Less than R25 mm    |
| (4) Imaging area                |          | 60°                  | Enough to visualize |
| (5) Size                        | Diameter | φ2.4 mm              | Less than φ5 mm     |
|                                 | Length   | 1170 mm              | More than 450 mm    |

Table 5 Specifications of light source

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

## 6.2. Examination of the Fiberscope and Light Source

For the purpose of visualization, it is necessary to install the fiberscope inside an actual operating engine by means of an intermediate linkage. Since this was the initial attempt to do this, however, consideration was given to the question of whether a fiberscope is suitable for the measurement environment.

The points requiring consideration are as shown below. The results are given in Table 4.

- (1) Oil-proof quality
- (2) Heat resistance
- (3) Flexibility
- (4) Imaging area (angular field of view)
- (5) Size (diameter, length)

As a result of this consideration, it was determined that the fiberscope satisfies every requirement and can therefore be applied to this use. The fiberscope that was used was an off-the-shelf product for industrial applications.

Consideration was next given to the light source. A 250-W metal halide light source installed coaxially with the fiberscope was tried. Imagery taken with this equipment was verified at the frame rate of the equipment as installed in the engine interior. The imagery was determined to have sufficient brightness for PTV and this equipment was therefore adopted. Table 5 shows the specifications.

## 6.3. Visualization of the State of Oil Flowing into and out of the Piston Pin Hole

### 6.3.1. The state of flow into the piston pin hole

First, the state of inward oil flow was observed. This was done by PTV (speed vectorization) of the visualization imagery from the fiberscope attached to the linkage. The oil inside the piston is also affected by oil spatter from the

crankshaft as well as by forced oil feed from the connecting rod and oil jet. This time, therefore, attention was focused on the state of lubrication resulting from the natural feed of oil by spray from the outside of the piston pin.

The visualization imagery and vector imagery acquired by this method were shown in Fig. 7 together with the diagram of the oil behavior clarified by these imageries. The results were obtained as video imagery, and the vectors of oil spatter behavior from moment to moment were clear. The following results were consequently obtained:

- (1) Oil behavior in the vicinity of the piston pin, which does not allow static visualization from outside, was made clear.
- (2) Vectorization further clarified the oil behavior.
- (3) Oil that flowed through from the bottom of the piston lands actually did not reach the piston pin hole, but instead fell off partway there.

These results clarified the state of natural oil feed by spattering from the outside and enabled consideration of methods for enhancement.

Use of this visualization method also enables determination of oil behavior on the inner side of the piston

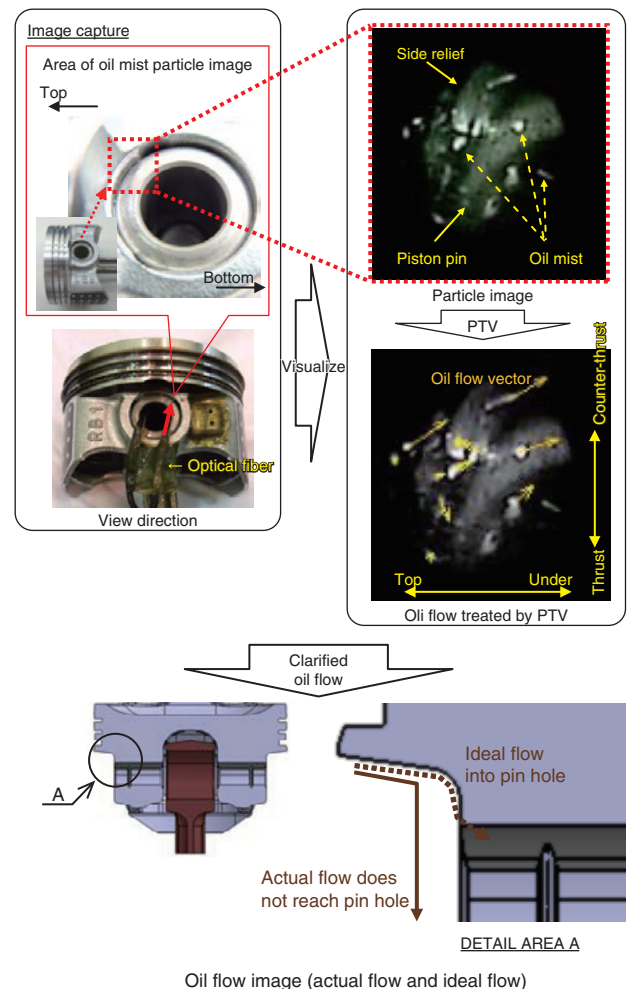


Fig. 7 Visualization of oil flow from piston to piston pin hole (Particle image: 640×560 pixel, 5000 fps)

pin. Further, it can also be used to judge flow volume quantitatively and ascertain oil behavior when forced oil feed occurs by oil jet or other such means. Also, with the use of a tracer, this method enables quantitative detection of oil flow inside the clearance spaces of each part.

This visualization technology thus offers possibilities for applications capable of ascertaining oil behavior in various locations inside the engine.

### 6.3.2. The state of flow out of the piston pin hole

Next, the state of oil flow out from the piston pin hole was clarified. Visualization of this phenomenon was performed with an engine that artificially recreated the circumstances in which piston pin noise occurs. Visualization was performed by direct observation using a high-speed camera. Figure 8 shows the oil behavior that occurred at that time.

As is clear from the visualization imagery, oil can be observed blowing out from the side relief following the impact of the piston and piston pin. This means that oil is leaking out from between the piston and piston pin.

Figure 9 shows the inferred course of events that lead

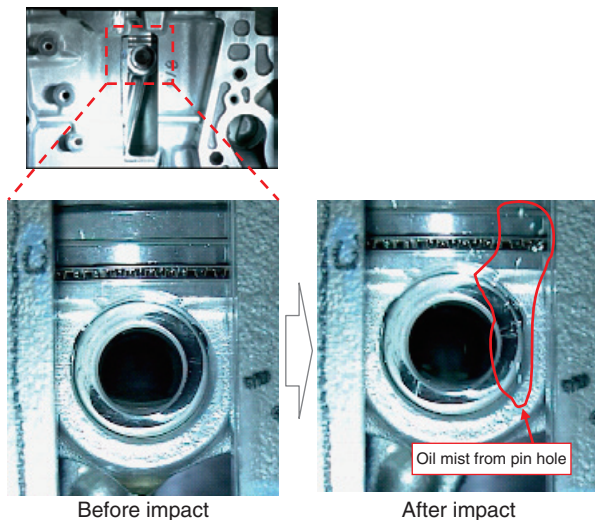


Fig. 8 Visualization of oil flow from piston pin hole to outside



Fig. 9 Oil transfer path in piston pin hole

up to oil being blown out as an end result of piston pin behavior. It is inferred that oil is pushed by the circular behavior of the piston pin and flows out through the passageway formed by the grooves around the inside circumference of the pin hole. Ultimately, it is thought to be expelled from the side relief.

### 6.4. Summary of Lubrication States

Visualization of the inward and outward flow states made clear that, in the state when piston pin noise is occurring, the oil at the bottom of the piston lands does not enter the piston pin hole, but instead falls off along the way and is blown out from the side relief. There is little oil inside the piston pin hole when this happens, so it can be inferred that the buffering effect when the piston pin impacts is reduced while the impact increases (Fig. 10). This is a crucial finding for the future clarification of the mechanism of piston pin noise as well as for the consideration of measures to counter it. In actuality, as well, the phenomena of being filled up and being blown out are interrelated, and their clarification will be another issue to be resolved.

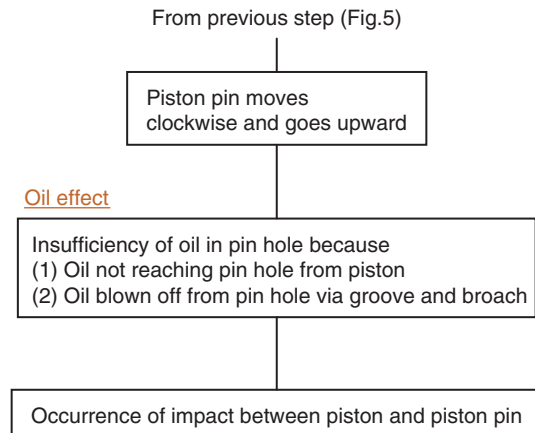


Fig. 10 Flow of piston pin noise phenomena with oil effect

## 7. Conclusions

- (1) Piston pin behavior when piston pin noise occurs with a semi-floating support system was clarified.
- (2) Methods of oil behavior visualization in the engine interior by means of the linkage method applying a fiberscope and of oil droplet vectorization by PTV were established.
- (3) Oil behavior around the piston pin hole when piston pin noise occurs during engine operation was clarified.

The visualization technology established by the present research is thought to be usable for ascertaining oil behavior in every part. It has future potential as a method for deciding specifications related to lubrication.

## Acknowledgements

The authors would like to take this opportunity to express their profound gratitude to Hideshi Hitosugi and Fumihiro Nakada of the Nippon Piston Ring Co., Ltd., who generously cooperated in the work of fabricating test equipment for the furtherance of the present research.

## References

- (1) Takiguchi, M., Aoki, Y., Anzai, S.: Relation between Lubricating Condition and Pin Slap Vibration at Pin Boss Bearing, JSAE Proceedings, No. 50-02, p. 5-8 (2002)
- (2) Kawano, K., Sue, S., Ishida, H.: Mechanism analysis of piston pin/pin boss bearing impact noise, JSAE Proceedings, No. 148-08, p. 5-10 (2008)
- (3) Kamiya, M., Kobayashi, T., Take, K., Takahashi, Y., Mihara, Y., Someya, T.: Measurement of Piston Skirt Oil-film Pressure when Piston Slap Noise Occurs, Honda R&D Technical Review, April, 2007, p. 149-155
- (4) Usui, M., Murayama, K., Sato, K., Yoshida, H.: Research in Lubricating Oil Flow and Visual Observation around Piston Ring, JSAE Proceeding, No.7 -06, p. 5-8 (2006)
- (5) [http://www.nissan-global.com/JP/NEWS/2008/\\_STORY/081110-03-j.html](http://www.nissan-global.com/JP/NEWS/2008/_STORY/081110-03-j.html)

### ■ Author ■



Takashi KONDO



Hisayuki OHBAYASHI

# Development of Noise-reducing Wheel

Youichi KAMIYAMA\*

Katsushi ISHII\*

Hisamitsu TAKAGI\*

Mikio KASHIWAI\*\*

---

## ABSTRACT

Tire cavity noise is a noise produced in the cabin by resonance in the hollow cavity inside the tire when it is excited by unevenness on the road surface, and this phenomenon is one factor of road noise.

A noise-reducing wheel was developed as a means of reducing tire cavity noise. In the proposed configuration, a thin, lightweight plastic Helmholtz resonator is fitted into grooves newly cut in the wheel well, balancing strength in relation to centrifugal force with fittability. The noise-reducing wheels have had the effect of reducing tire cavity noise to an inaudible level.

The newly developed noise-reducing wheels are used on the 2011 model HONDA LEGEND / ACURA-RL, mass-market vehicles.

---

## 1. Introduction

Tire cavity noise is a noise produced in the vehicle cabin when resonance in the circular cavity inside the tire excited by unevenness on the road surface causes the wheels to vibrate, and is transmitted to the body via the suspension as vibration. The resonance frequency is determined by the circumferential length of the circular cavity inside the tire, and is normally between 200 and 250 Hz in the case of standard passenger vehicle tires. Compared to other components of road noise, the sound pressure of tire cavity noise is high; in terms of hearing, it is close to a pure tone and it reverberates. These characteristics make it an abrasive sound for vehicle occupants, and it has long been one of the main road noise issues.

Because 1<sup>st</sup>-order resonance phenomena of the interior cavity of the tire, which are unavoidable in the case of air-filled tires, are the major factor in tire cavity noise, a fundamental solution to the issue represents a challenge unless resonance itself is dealt with. Taking a conventionally applied measure as an example, the structural vibration of the tires can be reduced, but when the balance between this measure and other tire functions is considered, the measure displays a low degree of freedom in implementation and a significant benefit cannot be expected. Looking at measures applied downstream from the tires, such as measures applied

in the suspension transmission paths or body-soundproofing measures, in many cases the parts involved are distributed over a wide area of the vehicle, and the measures result in weight increases.

In the future, the achievement of automotive weight savings will be more significant as part of more intensive responses to environmental issues, and it is predicted that the application of downstream measures that result in weight increases will become increasingly challenging. In addition, given that the contribution of tire input noise will become increasingly important because powerplant noise is being reduced by switching from the internal combustion engine to the motor, it is predicted that there will be increasing need for a noise reduction technology that acts directly on the tire input and is able to offer a fundamental solution with good weight efficiency.

In the past 30 years, more than 100 ideas for devices to reduce tire cavity noise have been patented or presented in reports<sup>(1)</sup>, but meeting market demands when realized as products has proven a challenge, and to date only a configuration in which sound-absorbent sponges are attached to the inner liners of the tires<sup>(2)</sup> has been developed as a commercially available product.

In developing a noise-reducing wheel, the project discussed here formulated a method in which a Helmholtz resonator, which would satisfy targets for

---

\* Automobile R&D Center

\*\* Motorcycle R&D Center

functionality, cost, and the necessary productivity for mass production, was attached to wheels. This method was able to reduce tire cavity noise to an inaudible level regardless of type of tire.

## 2. Development Goals

The goals for the development of a tire cavity noise-reducing device were as follows.

- (1) Realization of a 10 dB reduction in noise during operation of a complete vehicle, in order to reduce tire cavity noise to an inaudible level regardless of type of tire.
- (2) Realization of an equivalent level of strength and durability to conventional tires and wheels in relation to centrifugal force at maximum speed and high levels of input when driving on poor road surfaces, crossing curbs, and similar situations.
- (3) Realization of a device that is lightweight and that does not affect wheel balance, in order to reconcile noise reduction with passenger comfort (ride comfort and vibration).
- (4) Realization of an equivalent level of operability to conventional tires and wheels when fitting and removing the tires or in other maintenance situations.
- (5) Realization of a low-cost device configuration that takes the degree of challenge of production technologies and mass production into consideration and that does not affect the conventional tire and wheel manufacturing processes.

## 3. Basic Concept

The configuration was envisioned as a wheel assembly with a separate flat, lightweight plastic resonator fitted in the wheel well.

Figure 1 shows the basic configuration of the device. It displays the following characteristics:

- (1) Thin, lightweight and rigid plastic helps enable resonator configuration with flat cross-section
- (2) Fitted assembly configuration that helps prevent separation of the resonator due to catching on the grooves in the direction of the circumference of the wheel
- (3) A structure that fixes the vents of the resonator by means of notched grooves in the wheel

The basic concept of the design of this configuration was to help ensure an increase in the holding force of the resonator when centrifugal force is acting on it (Fig. 2).

High-rigidity polypropylene was selected as the plastic resonator material in order to control increases in its unsprung weight and help ensure strength in relation to centrifugal force.

Low-cost and high-productivity blow molding, which offers a high degree of freedom in molding, was selected as the method of forming the device. The project aimed

by this means to help enable the formation of a seamless resonator structure that was beneficial in terms of air-tightness.

In order to help ensure that the resonator would display

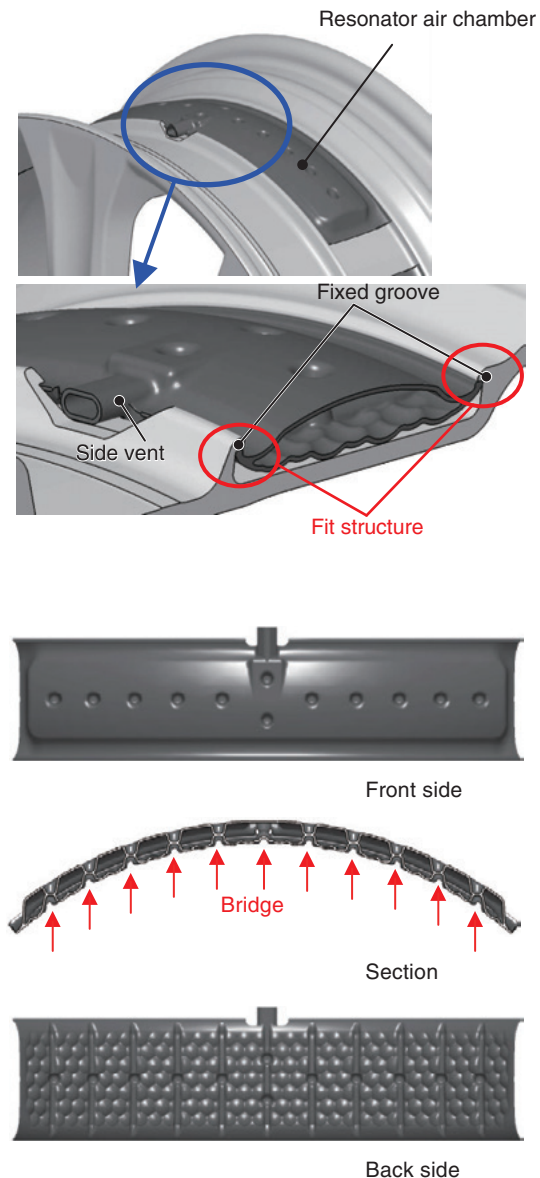


Fig. 1 Structure of noise-reducing wheel

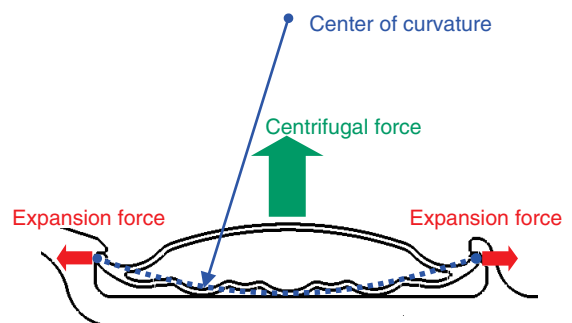


Fig. 2 Concept of structure under centrifugal force

an adequate level of noise attenuation, it was important to achieve air-tightness and realize sufficient resonator volume and vent area in a restricted space for the layout. Figure 3 shows the relationship between the volume of the resonator and noise attenuation, and between the cross-sectional area of the vent and noise attenuation.

The necessary basic specifications of the resonator were decided based on an understanding of the effect of the parameters discussed above.

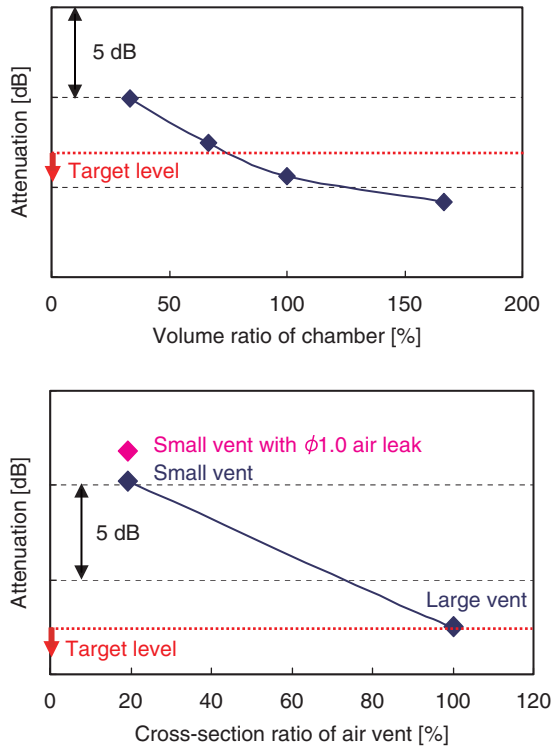


Fig. 3 Attenuation parameters of resonators

## 4. Main Development Details

This chapter will discuss the technological issues associated with realizing a lightweight, low-cost and high-productivity resonator structure displaying sufficient noise attenuation and strength and durability under all projected driving conditions, and the measures adopted in response to these issues.

### 4.1. Thin, Lightweight Structure and Noise Attenuation Performance

When a vehicle is driving at 300 km/h, the centrifugal force acting on the rim of the wheels reaches 1500 G. For a resonator to resist this maximum centrifugal force, its weight must be minimized. On the other hand, for a resonator to realize noise attenuation performance, it is necessary to attempt to ensure surface stiffness.

Taking the resonators used in automotive intake systems as an example, the thickness of polypropylene resonators is

generally between 2.5 and 3.0 mm. However, the weight of a resonator of this thickness would become too high when fitted to a wheel acted on by centrifugal force, and therefore could not be used. It was therefore necessary to develop a plastic resonator structure with thinness and low weight as prerequisites.

In order to study the surface stiffness necessary to achieving the target sound attenuation performance, tests were conducted in which the thickness of a simple hollow base structure was varied. These tests provided insight into the relationship between the sound attenuation performance and the necessary thickness of the structure.

As Fig. 4 shows, when the minimum thickness for blow molding (considered as 100%) is used, the resonator displays almost no noise attenuation performance. While the level of noise attenuation displays a slight increase at 150% of the minimum thickness, at 200% there is a significant decline in the volume of the resonator, and the level of noise attenuation also declines.

Given this, it was judged that the achievement of the target noise attenuation performance would represent a challenge using measures focusing on the thickness of the housing alone if a simple hollow structure were employed as the basic cross-sectional structure of the resonator. It would therefore be necessary to consider a structure able to realize adequate noise attenuation performance with a thin housing.

CAE computations were therefore employed to introduce the necessary number of bridges to make the upper and lower surfaces within the hollow section (Fig. 1) continuous, and in addition brambles (spherical projections in a honeycomb alignment) were added to the external surface of the low-stiffness lower surface in order to increase surface stiffness. The results of calculations of the degree of attenuation of noise produced by excitation of the tires indicated that this configuration achieved the target noise reduction of 10 dB or more with no increase in weight against the hollow structure of minimum thickness discussed above (Fig. 5).

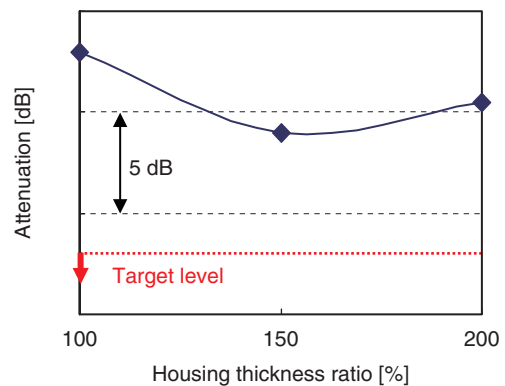


Fig. 4 Attenuation of resonator when thickness of housing is varied

#### 4.2. Structure Reconciling Strength in Relation to Centrifugal Force with Fittability

In order to help ensure a sufficient level of strength in the resonator to resist maximum centrifugal force, it was necessary to optimize the structure based on an understanding of the load conditions, frequency of input, environmental conditions, and other factors affecting a resonator attached to a wheel, under all driving conditions. But in addition, from the perspective of fittability, a structure able to be fitted at low loads would be desirable. A design that balanced these two demands would therefore be necessary.

The bending stiffness of the resonator has the greatest effect on its strength and fittability, and when plastic materials are employed this parameter varies significantly with the environmental temperature in which the resonator is placed. The maximum temperature of the surface of a resonator was measured. As Fig. 6 shows, the elastic modulus of plastic materials declines to approximately 20% of the figure at normal temperatures when the material is placed in temperatures close to the highest temperature reached by a wheel rim under severe conditions.

Because it would be challenging to realize a significant reduction in these temperature-dependent changes in the

elastic modulus by focusing on material characteristics alone, it was decided to increase the bending stiffness of the resonator by modifying its form, in order to help ensure the necessary bending stiffness for strength despite the low elastic modulus at the upper limit temperature. Fittability would be reconciled with this change by presupposing a shift from low-load fitting able to be performed by hand to high-load fitting by tool.

Using the CAE analysis of centrifugal force shown in Fig. 7, length adjustment of resonator arm section and lateral beads were added to the brambled lower surface of the hollow section to help ensure uniform bending of the resonator when acted upon by centrifugal force (Fig. 8), in an attempt to increase bending stiffness under centrifugal force while maintaining the surface stiffness important to noise attenuation performance.

The results of centrifugal force load tests using the structure described above showed that the structure displayed sufficient strength to resist the target maximum centrifugal force without any detachment or damage to the

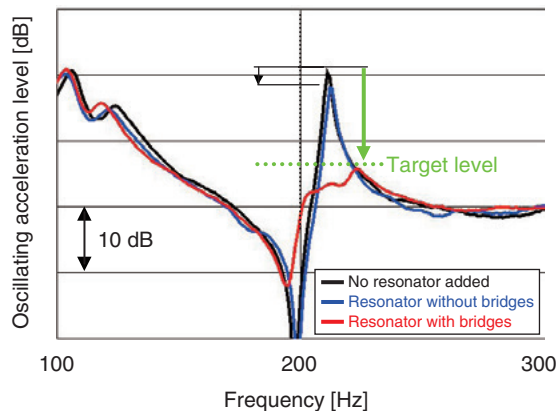


Fig. 5 Noise reduction results for tire tread hammering test on wheel rim

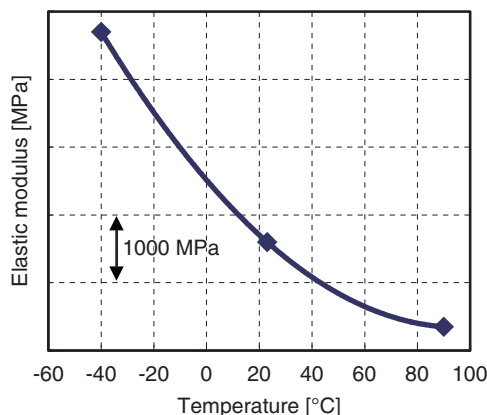


Fig. 6 Temperature-dependency of elastic modulus

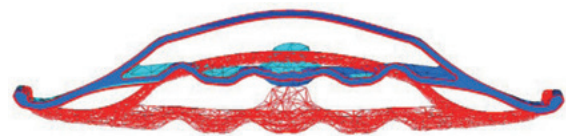
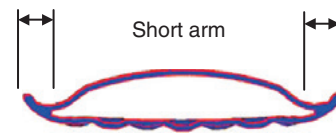


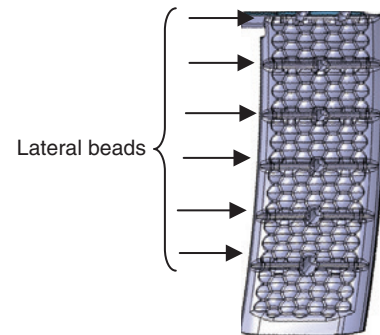
Fig. 7 CAE analysis result of centrifugal force



(a) Overview



(b) Section



(c) View of back side

Fig. 8 Structure for resistance to centrifugal force

resonator with regard to the tire and the wheel assembly, and without aggravating the wheel balance. These studies helped to enable the realization of a resonator structure that balanced strength against centrifugal force with fittability.

#### 4.3. Method of Fitting in Mass Production

In terms of production technologies, the greatest degree of challenge was represented by the process of fitting the resonator to the wheel. No examples existed of any parts other than TPMS sensors being fitted to wheels.

A consideration of the mass production of wheels with attached resonators indicated that the time available for the fitting of the resonator to the wheel would be restricted due to the line time necessary for the wheel manufacturing process and the fitting of the tire to the wheel. The issue was therefore the development of a method able to fit the resonator to the wheel within a restricted timeframe.

In addition, for fitting in the mass-production process, it was necessary to consider all combinations of the following product dimensional tolerances and environmental temperature (Fig. 9):

- (1) Interference between resonator and wheel
- (2) Stiffness of resonator (Thickness of blow molding)
- (3) Environmental temperature during fitting

This necessitated parameter setting that would help to ensure fittability under all projected environmental temperature conditions. For example, because the stiffness of the plastic material would change significantly with variations in temperature, it would be essential to consider the necessity for increasing the fitting load at low temperatures (Condition E in Fig. 9) and the fact that the resonator could not be fitted at high temperatures because it would bend excessively and the fitting stroke would be insufficient (Condition C in Fig. 9).

Because the stiffness of the resonators had been increased in order to increase strength in relation to centrifugal force, as discussed above, it was necessary to press the prototype resonators a number of times with a specialized tool (Fig. 10) in order to fit them. However,

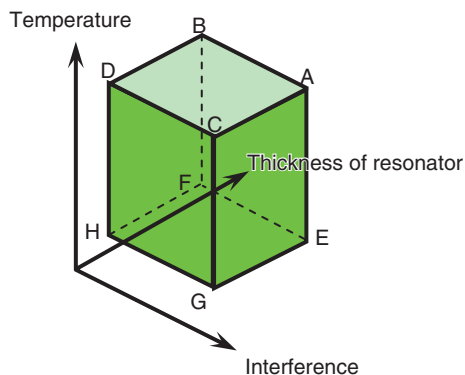


Fig. 9 Parameters for fitting

this fitting method was inefficient and impractical from the perspective of the mass-production process, as discussed above.

A method of fitting by means of an assembly device using an air cylinder was therefore studied. By making the shape of the “pusher” section of this machine (the section that presses the resonator) an arc that followed the shape of the arm section of the resonator, the project aimed to fit the resonators by pressing each one once.

The results of fitting tests on the resonators using a prototype machine demonstrated that it reduced fitting time by 80%, indicating the potential for use in mass production.

In addition, adjustment of the following items (Fig. 11) made fitting of the resonators possible in a wide range of temperatures projected for the mass-production process.

- (a) Fitting area for resonator arm sections in wheel grooves
- (b) Pusher position and stroke of assembly machine
- (c) Pushing load of assembly machine pusher

The optimization of these three items ultimately made it possible to fit the resonators in the mass-production processes under temperature conditions ranging from -5 deg C to 40 deg C.



Fig. 10 Tool for fitting (prototype)

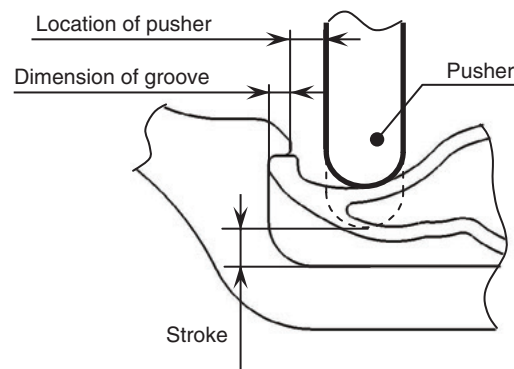


Fig. 11 Groove shape, pusher location and stroke optimization of resonator and wheel assembly machine

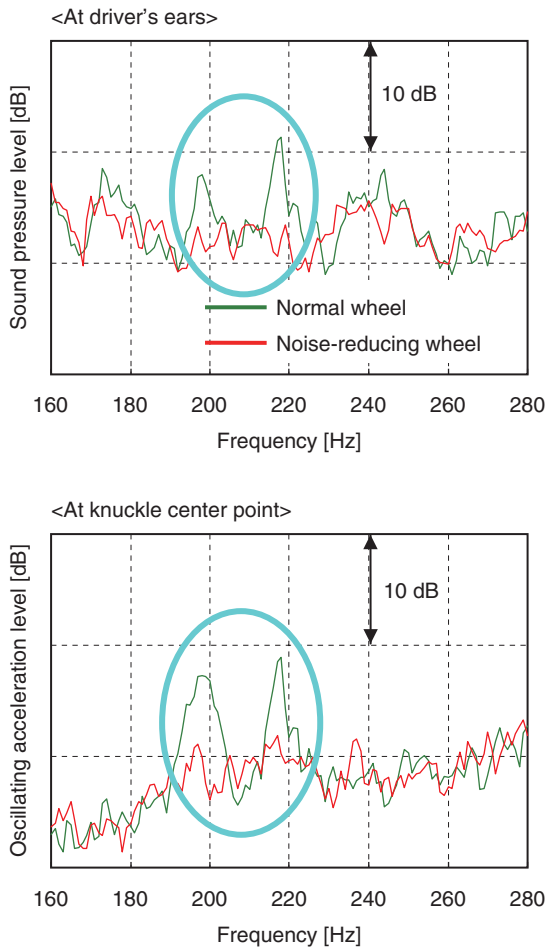


Fig. 12 Noise reduction effect when driving on rough road at 80 km/h

## 5. Effect of Use of Actual Resonators

Figure 12 shows results for in-cabin noise and knuckle vibration when driving on a rough road for a vehicle fitted with the noise-reducing wheels. There has been a 10 dB reduction in noise, and tire cavity noise has been reduced to an inaudible level.

The robustness of the effect of the noise-reducing wheels in relation to road surface roughness and vehicle speed conditions was tested next. Figure 13 shows the effect of the noise-reducing wheels on in-cabin noise when the vehicle was driven over road surfaces of varying roughness at different speeds. A reduction in noise of 10 dB was realized under all conditions (low input, high input, low vehicle speed, high vehicle speed), demonstrating that the noise-reducing wheels possess robustness in relation to road surface roughness and changes in vehicle speed.

## 6. Conclusion

A noise-reducing wheel structure fitted with a plastic resonator satisfying the conditions listed below was proposed, and realized in mass production for the first time in the world.

- (1) A thin, lightweight plastic resonator structure able to reduce noise by 10 dB at the minimum thickness for blow molding
- (2) A resonator structure and wheel fitting configuration able to resist the maximum centrifugal force of 1,500 G
- (3) A resonator fitting setting and mass-production fitting method realizing a high level of fittability

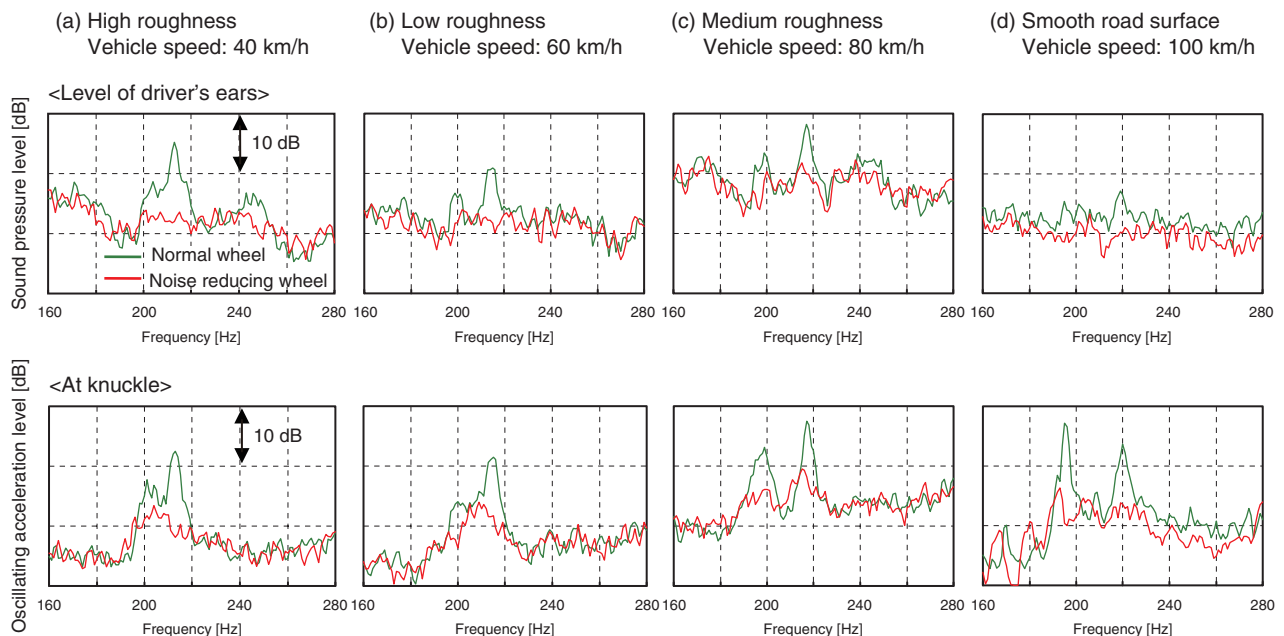


Fig. 13 Robustness of noise reduction in relation to road roughness and vehicle speed

## References

- (1) Yamauchi, U., Akiyoshi, Y.: Technique of Tire Acoustic-Cavity Noise Control, Transactions of Society of Automotive Engineers of Japan, Vol. 32, No. 2, p. 79-84 (2001)
- (2) Yukawa, N.: Engineering of Special Sound-Absorption Sponge, Japan Society of Mechanical Engineers, 2009. 5, Vol. 112, No. 1086, p. 426 (2009)

## ■ Author ■



Youichi KAMIYAMA



Katsushi ISHII



Hisamitsu TAKAGI



Mikio KASHIWAI

# Development of CAE Standardization System for Motorcycle Parts with Navigating Function for Designers

Mijo ITO\*

Daisuke ISHIHARA\*

Miki TSUCHIYA\*

Masato OTSUKA\*

Kenji TSUCHIMOTO\*

Yasuhiro MIYAZAKI\*

## ABSTRACT

“Motorcycle parts CAE standardization system” that focused on CAE for designers (hereafter termed “designers’ CAE”) has been developed. This system simplified the analytical condition settings by unique navigation function that was suitable for designers’ CAE, and realized prevention of redoing due to misunderstandings and operation errors. And also, the individual differences in analyses were eliminated by standardizing the methods for analysis and condition settings. In addition, the comparison of the analytical results was facilitated by the data accumulation using this system, and evaluations of the results and the fixed form report making were semi-automatically carried out by the function of “Analysis Tag.” Thus this system reduced the working hours of designers’ CAE by 75%, and enabled effective supports for design operations.

## 1. Introduction

With the progress of generally-applicable CAD tools in recent years, the CAE functions applicable on CAD systems have been enhanced along with the simplification of control operations. Consequently, the hurdle for designers’ CAE is lowered, and it is rapidly pervading along with an increased demand for maturation of designs on the desktop at the initial stage of the development process. In motorcycle design sections of Honda, the number of users of CAE for designers has increased approximately 2.5 times in these five years (Fig. 1).

On the other hand, there is an issue that the designers tend to forget the analysis processes and/or the operations because the majority of designers use CAE with a low level of frequency. Figure 2 shows the distribution of the frequency of use of CAE by designers. More than 50% of designers use CAE less than 20 times per year, and cases in which the designers did not use CAE for more than a year are not uncommon. For designers, CAE is a method of reviewing designs, and it is natural for them to use the method only when necessary. Such a trend is therefore a matter of course.

For designers’ CAE having such characteristics, a variety of attempts have been made such as the simplification and/or automation of operations<sup>(1), (2)</sup>.

However, there are such issues that a simple automation makes the analysis process into a black box and disturbs

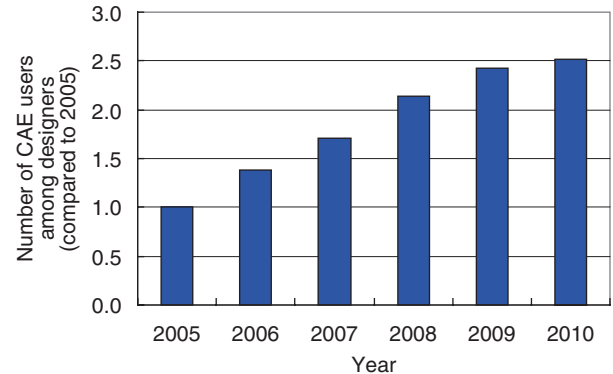


Fig. 1 Increase in number of CAE users among designers (compared to 2005)

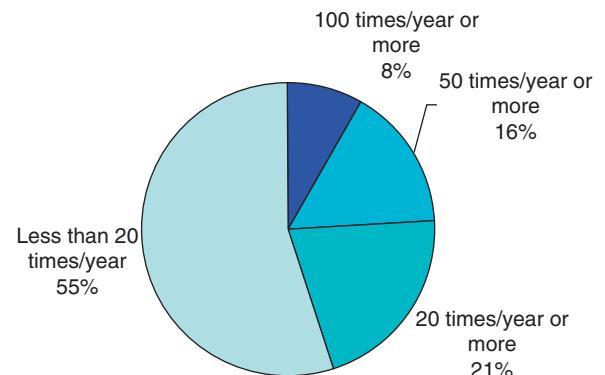


Fig. 2 Frequency of CAE use by designers

\* Motorcycle R&D Center

application of CAE to other parts. In view of that, we have developed the “CAE standardization system for motorcycle parts”, which is equipped with the navigation function for easy understanding of the analysis processes, for effective use of CAE by designers. This paper introduces the features of this system and discusses its effects.

## 2. Issues of CAE for Designers

The following issues have become conspicuous as a result of the rapid increase in the use of designers' CAE.

### 2.1. Redoing in Analytic Work

Figure 3 shows a breakdown of categories for requests for support from CAE users. It can be understood from the graph that there are considerable misunderstandings and operational errors. It is thought that many cases of redoing occur until acquisition of a proper analysis result due to mistakes in areas such as settings due to insufficient knowledge and unintended operating errors, especially in the case of designers who use CAE infrequently as mentioned earlier or in the case of beginning users.

### 2.2. Differences in Methods and Settings between Users

Because multiple analysis conditions (including load conditions, constraint conditions, coupling conditions, and mesh settings) can be set even for identical CAE analyses, differences in terms of analytic methods and settings can occur between different users. When these types of differences exist, issues can arise especially when comparing analysis results. Even when two sets of results based on different mesh sizes exist and each one is valid individually, a comparison of such two results is inappropriate in many cases.

### 2.3. Effective Use of Past Analysis Results

Due to the changes of computation environment such as the advancement of hardware and applications, the size of analysis such as the analyze-able number of meshes are changing year by year. The scale of the analyses

that can be conducted using standard computational environments has increased approximately 100 times in the past 10 years. When computational environments change like this, the optimal values of elements such as mesh size settings also change. Results obtained in the past cannot be compared with the latest results under the same conditions, and it is therefore not possible to effectively use accumulated data.

### 2.4. Increased Work Hours due to Cumbersome Analysis Condition-setting

With the evolution of CAD tools and hardware environments, as discussed above, designers also have become able to conduct analyses of complex, large-scale assemblies. Analyses of this type necessitate a large number of troublesome condition-setting procedures, thus increasing the associated work hours.

## 3. Aims of System Development

The targets of system development to cope with the abovementioned features and issues of design CAE can be outlined as follows.

(1) Rather than automation of procedures, development of a navigation system for analytic procedures

Automation of CAE offers significant benefits in terms of reducing work hours and helping to prevent operational errors. However, the investigation results of parts for which the designer used CAE in recent one year showed that the frequently analyzed 10 parts made up a share of approximately 50%, while the other 50% was covered by more than 300 parts with a low frequency of analyses. The automation of analyses for all of these parts requires tremendous development work hours, which is unrealistic. Further, when the analysis is automated only for the high-frequency parts, although the automated parts can be analyzed easily, as the settings become a black box (i.e., how the analysis procedures are set up will be invisible), an application to other part will be impossible.

This system navigates a series of operations from the setting of analysis conditions up to the evaluation while keeping the user conscious of the process. Accordingly, not only the system leads the user to accurate analyses and/or evaluations, but it effectively supports designers who tend to forget operations due to a low level of frequency of conducting analysis, and at the same time allows the user to acquire skills that can be applied to other analyses.

(2) Development of a system that prevent redoes and setting differences between designers

The redoing due to misunderstandings caused by insufficient knowledge or experience and differences in methods and settings depending on the user should be prevented by standardization of setting of analysis conditions and evaluation of results, making the system such that allows proper execution of the standardized method. In addition, the redoing due to operating errors that

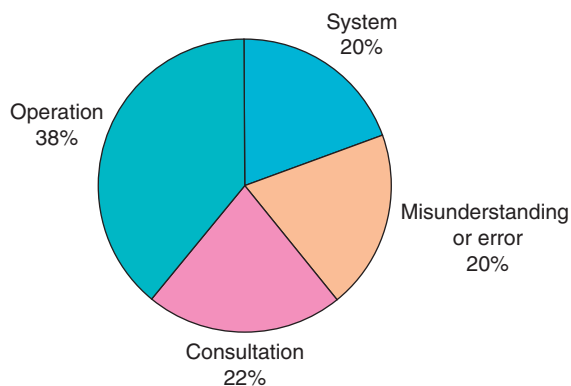


Fig. 3 Categories of requests for support from CAE users

occurs against the intention of the user should be prevented by simplification of troublesome condition settings that cause errors, and incorporation of a check tool that finds errors before execution of computation and such a process in which the user him/herself verifies the validity of the results after computation.

- (3) Development of a system that allows accumulation of analysis data under such conditions enabling its effective use

The analysis data and results should be accumulated in such a format that allows easy updating when necessary even when requiring a change of setting such as the mesh size due to changes of computation environment.

- (4) Development of a system able to reduce work hours for condition-setting

Because the increase in work hours for designers conducting CAE analyses represents a considerable burden, one of the aims for the system is to reduce work hours by simplifying troublesome and repetitive condition-setting procedures.

#### 4. Standardization of Analytic Methods

In order to realize the aims outlined above, 15 structural parts with a high level of demand for CAE analyses were selected on the basis of the investigation of the frequency with which CAE analyses were conducted, and methods for condition-setting and evaluation of results were standardized for each part. Because the technical discussion in this paper is conducted from the system perspective, discussion of the concrete details of this standardization will be omitted. Nevertheless, the standardization of analytic methods represented an essential precondition for the development of the system. However, because only by standardization, it is insufficient for everyone to execute the method surely, the development of a mechanism that ensure adherence to the standardized analytic process is important.

#### 5. System Configuration and Overview

##### 5.1. Basic System Configuration

Figure 4 shows a diagram of the system configuration. Generative Structural Analysis (GSA) is a CAE function of CATIA V5, a CAD system in use in Honda<sup>(3)</sup>. Because all settings are linked to 3D-CAD data in the GSA module, analytical conditions can be set on the basis of shape alone. Due to the change from the conventional mesh-based setting to the shape-based setting, it became easy for even designers with limited experience to conduct CAE.

The developed system is built on GSA using the configuration shown above. Because all the graphical user interfaces (GUI) used by the designers work on CATIA V5, designers are able to conduct analyses using an operating environment with which they have had experience on a daily basis. In addition, the creation of a

system based on GSA functions means that it is able to be used in combination with standard GSA functions, and thus possesses a high level of general applicability.

##### 5.2. System Overview

This section provides an overview of the system, with the footrest holder standardization system as an example. Figure 5 shows an outline of the system's navigation functions. By executing the steps in the menu from the top, the designer can properly conduct the standardized analysis and evaluate the result. In addition, as the flow of analysis itself is not peculiar for each part, the designer can naturally learn the general flow of analysis by practicing the analysis steps.

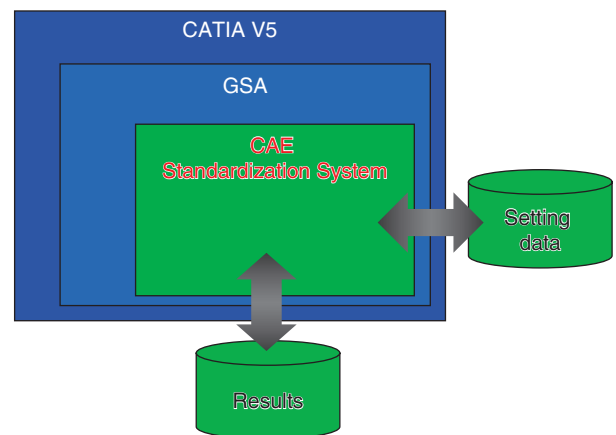


Fig. 4 System configuration

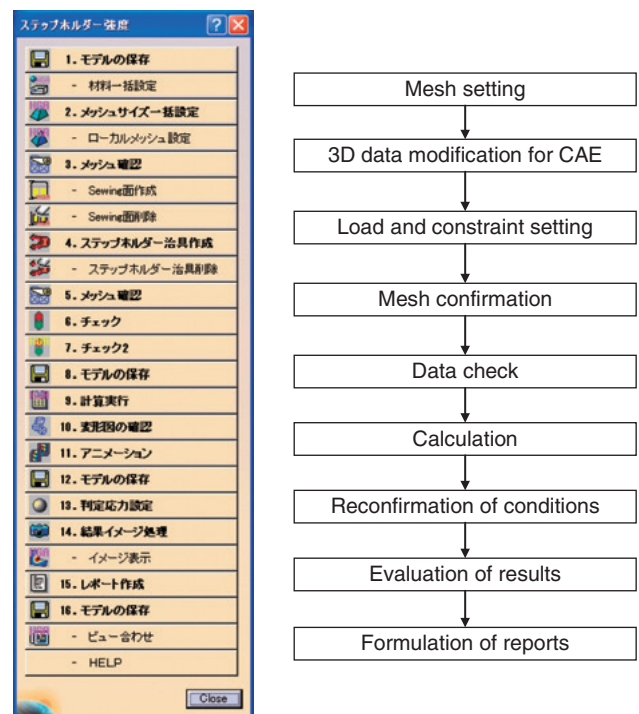


Fig. 5 Outline of navigation function

### 5.2.1. Mesh setting

Mesh settings such as mesh size and element types are the important condition settings when conducting CAE analysis. To eliminate the issue of difference of mesh setting depending on the user, the setting window in this system is designed in such a manner that the mesh setting determined by the standardization for each part is usually automatically input without allowing the user to make modifications (Fig. 6). However, because there are cases in which it is not possible to generate meshes for the specified settings using currently available meshers, a mechanism through which the correction of the setting was enabled by checking the modify check box, and change information from the standard was output to the report was introduced.

### 5.2.2. Correction of models for CAE

In recent years, simplification of shapes for use in CAE (elimination of unnecessary file, hole, etc.) is not frequently conducted, and in many cases 3D shapes for designs are used in analyses as it is<sup>(4)</sup>. However, there are cases in which it is necessary to correct shapes when setting conditions for CAE.

In the example of footrest holder shown by Fig. 7 as an example, while there is no necessity for 3D design drawings of the shape of the contact surfaces of the footrest bar stopper, it is necessary to set conditions for CAE.

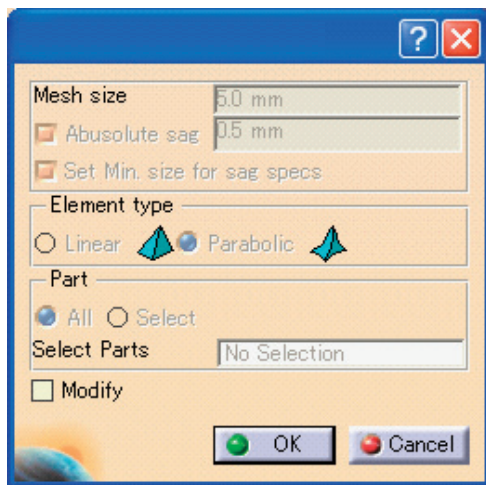


Fig. 6 Dialog box for mesh setting

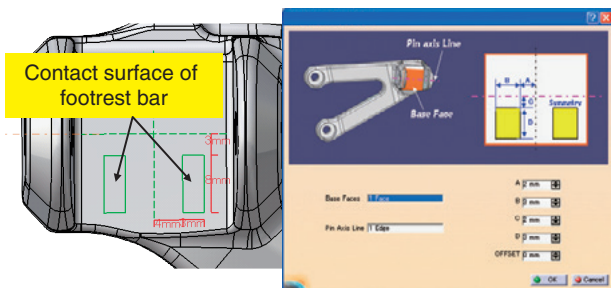


Fig. 7 Assist function for creation of contact surfaces for design of footrest bar

These contact surface shapes have to be created by a CAD function, and because this interrupts the flow of the CAE operations, the process involves a considerable amount of operating stress and a large number of work hours. The developed system permits practicing of this operation within the navigation menu without switching to the CAD function, thus reducing wasteful work hours.

### 5.2.3. Setting of load and constraint conditions

Figure 8 shows the window for setting load and constraint conditions. The setting of load and constraint conditions is the most important element in conducting CAE analyses, but at the same time differences between users tend to occur also in these operations. In the developed system, these operations have been simplified for correct setting of standardized conditions. With only the minimum necessary selection and input, the system permits creation of load, constraint and coupling conditions. For example, although no column for the input of load values is displayed on the window, fixed standardized values are automatically employed. The simplification not only allows observing of the standardized settings, but also reduces work hours and operation stress, and prevents mistakes.

### 5.2.4. Checking the mesh

The authors believe that the ultimate image of designers' CAE is that there is no need for the user to be aware of the mesh. However, at present auto-mesh functions and analytic scales are inadequate, and it is necessary to check aspects of the mesh including its density and shape prior to proceeding with computations. As indicated in Section 5.1, the current CAE function that permits setting only on the basis of the shape allows computation without viewing the mesh even once. In order to prevent such a situation, a mesh-check process was introduced to the navigation functions in the developed system.

Basically appropriate meshes are created in most cases if the standardized mesh setting is used. The deliberate addition of the mesh-check process in this system is not

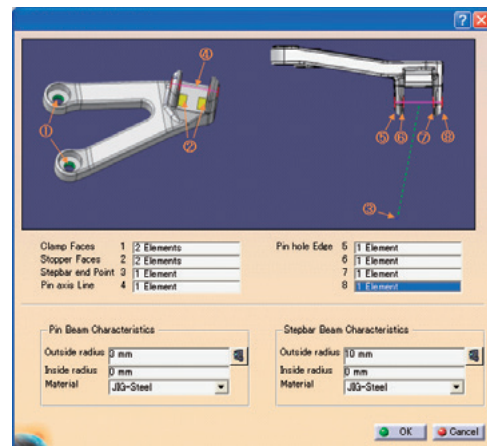


Fig. 8 Assist function for load and constraint condition setting

only for confirmation of actual operation, but also for acquisition of a regular practice for the user to make the mesh checking as a process of general analysis.

#### 5.2.5. Model check

In the check function shown in Fig. 9, frequent misunderstanding, setting that may cause an issue although not really an error, unique prohibition, etc. are automatically checked and made visible from the type of model setting. By this means, it makes users aware of misunderstandings and errors prior to execution of the computation, and thus prevents redoing after the computation.

#### 5.2.6. Conducting computations

When computations are conducted using this system, a use history is automatically recorded to allow for grasping of status of applications of the standardized system to each part. Based on this use history, the effects of standardization are checked, and the application of standardized procedures to new parts is reviewed.

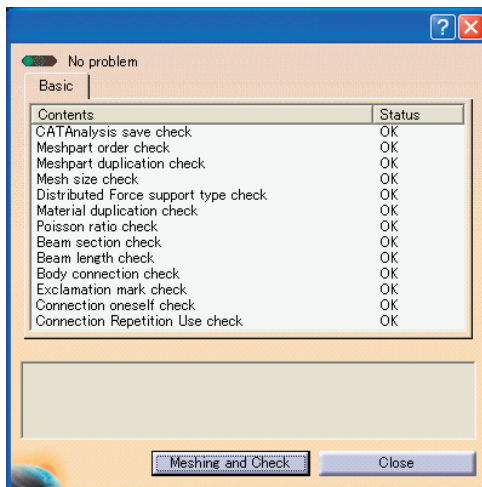


Fig. 9 Dialog box for data check function

#### 5.2.7. Rechecking analysis conditions

When the computation has been completed, the first thing that must be done is to recheck the correctness of the computation conditions. As the most simple and effective way, the process that displays magnified deformation animation is added in this system. This makes it possible for the user to verify visually that the computation conditions had been as they had imagined.

#### 5.2.8. Evaluation of results

In the evaluation of results of strength analysis, the maximum stress is used as the value for evaluation in many cases. Normally, a reading of the maximum value is taken by seeking the node where the maximum value is obtained in the proximity by manual operations after showing the stress distribution by a stress-contour diagram, etc. and determining where the high stress is occurring. Although some application software has a function for searching of a maximum value, as such software simply searches the maximum value of the entire model, a singularity not appropriate to use for evaluation or a value outside of the area subjected to the evaluation is searched in some cases. There is also such an issue that when the reading of a value is taken by a manual operation, it would be challenging to identify a value of what node is used for the evaluation when the model is viewed in a later day.

A unique Analysis Tag function was developed in order to resolve these issues (Fig. 10). This function allows saving of the node, value, comment, etc. used for the evaluation as model data. Even when a third-party views the model, the grounds of evaluation are clear. In addition, the implementation of a maximum value search function in areas of occurrence of high stress helps to prevent maximum values from being overlooked and reduces differences in evaluations due to different users.

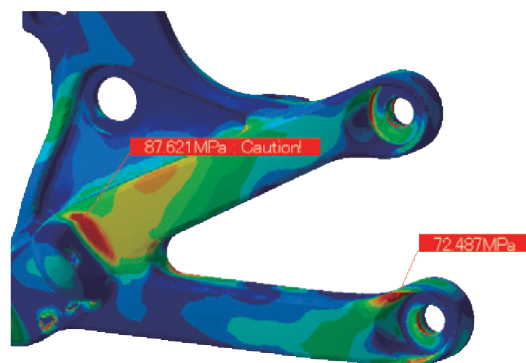
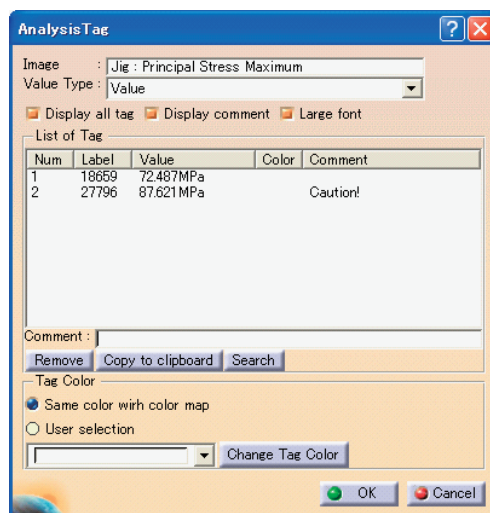


Fig. 10 Function of the analysis tag

### 5.2.9. Formulating reports

This system semi-automatically creates a regular-form report indicating information necessary for a third-party when viewing CAE results. Information including the model data name, details of the settings for the analysis, information on changes to standardized settings, evaluation values obtained from the abovementioned Analysis Tag, and the results of judgments are automatically incorporated in these reports. Based on this data, the designer is easily able to complete a report by simply adding a discussion.

## 6. Effect of the Developed System

### 6.1. Support for Designers' CAE

The offering of the navigation system that suits characteristics of designers' CAE permits proper analysis even by a designer using CAE infrequently. Also, by repeating analysis by this system, designers can learn the general analysis process and increase their skill to apply CAE for parts other than the standardized ones.

### 6.2. Reduction of Redoing in Analysis and Differences in Settings between Users

Practicing of the analysis method determined by the standardization on this system permits reduction of redoing caused by an improper setting of analysis condition, etc. and differences of setting between users. In addition, the system's simplification of troublesome condition-setting procedures and the introduction of a check function have reduced the incidence of redoes due to operational errors.

### 6.3. Effective Use of Past Analysis Data

The accumulation of analytic data by the developed system has made it possible to easily update results even when there are changes in mesh settings due to changes in the computational environment, preventing accumulated data from becoming unusable in later comparisons. In addition, a unique Analysis Tag function has made it possible to leave a clear record of the basis for evaluation in the accumulated model data.

### 6.4. Reduction of Work Hours

The system's simplification of troublesome condition-setting procedures and its use of a navigation function have reduced operational work hours for designers by an average of 75% (Fig. 11). This is chiefly an effect of the simplified formulation of load and constraint conditions and the supplementary elements necessary for condition-setting (pre-processing) and the semi-automatic generation of evaluations of analysis results and standard reports (post-processing). In addition, the developed system has also enabled analyses that were previously challenging for designers due to the complex condition-setting procedures for the parts.

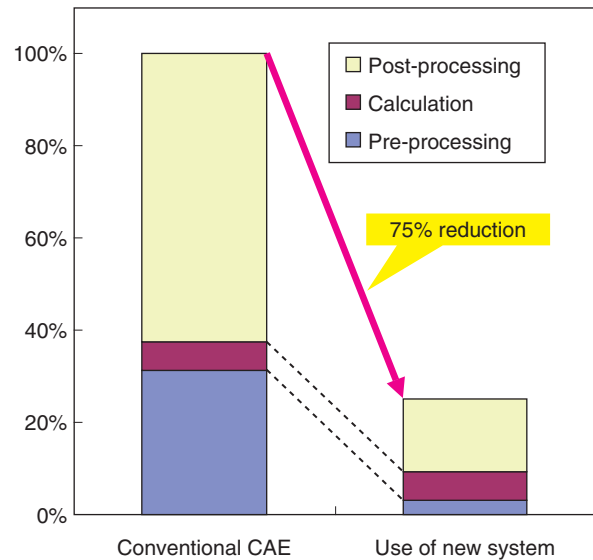


Fig. 11 Ratio of reduction of total work hours

## 7. Conclusion and Future Developments

The development of a CAE standardization system for motorcycle parts focusing on designers' CAE has reduced the incidence of redoes due to misunderstandings and of operational errors and differences in methods and settings between designers. In addition, by simplifying troublesome condition-setting procedures and the evaluation of results, the system has reduced operational work hours by an average of 75%. Because entire analytic procedures are available on a navigation menu, the system offers optimal support suited for the characteristics of designers' CAE, and at the same time, as designers can learn the general analysis process and increase their CAE skills, an application to those other than the standardized parts is enabled. In addition, the system also makes updating of accumulated data easy, allowing effective use of past analysis data.

The present system focuses exclusively on parts that can be reviewed using linear structural analyses. However, to fill the growing demand for the maturation of designs on the desktop, we will consider extension of application to areas including non-linear structure analysis in the future.

## References

- (1) Kojima, Y.: First Order Analysis as CAE for Design Engineers, R&D Review of Toyota CRDL, Vol. 37, No. 1, p. 1-3 (2002)
- (2) Umemura, K., Kondo, T.: Development and Deployment of SOL!BOX, Denso Technical Review, Vol. 7, No. 1, p. 136-140 (2002)
- (3) Dassault Systemes, <http://www.3ds.com/>

- (4) Uchida, T., Nagai, S., Koyama, S.: Practical Example and Future Directions for CAE-Based Development for Automobile Transmission by Designers, Honda R&D Technical Review, Vol. 15, No. 1, p. 109-114

■ Author ■



Mijo ITO



Daisuke ISHIHARA



Miki TSUCHIYA



Masato OTSUKA



Kenji TSUCHIMOTO



Yasuhiro MIYAZAKI

# Mechanical Structure and Performance Evaluation of High Differential Pressure Water Electrolysis Cell

Eiji HARYU\*

Koji NAKAZAWA\*

Kenji TARUYA\*

Hiroyuki ISHIKAWA\*

Masanori OKABE\*

## ABSTRACT

High differential pressure water electrolysis is a method that can replace the conventional high-pressure hydrogen production technology that uses normal pressure water electrolysis and a mechanical compressor. The differential pressure method requires a pressure resistant electrolysis cell, and the target hydrogen pressure is to be held by the proton exchange membrane. These issues were addressed by adjusting the strength and surfacial pores of the anode current collector and suppressing proton exchange membrane deformation, enabling the membrane to sustain a differential pressure of 35 MPa. Tests using a piston-type high differential pressure water electrolysis cell clarified that the optimum interfacial pressure between the current collector and the catalyst is 5 MPa. High-pressure hydrogen was successfully produced with a theoretical pressurization energy based on the concentration difference voltage by maintaining a constant interfacial pressure regardless of the hydrogen pressure inside the cell. In addition, it was confirmed that the current efficiency was 97%, the only loss was that due to hydrogen cross-permeation across the membrane, and that the cross-permeated hydrogen had no effect on other functions. Cross-permeation of oxygen was suppressed due to the differential pressure, and the oxygen concentration in the produced hydrogen was the specification value of 5 ppm or less. This means that the obtained hydrogen had a higher purity than that produced by normal pressure water electrolysis.

## 1. Introduction

Research is being conducted on a Solar Hydrogen Station (SHS) that combines renewable electricity generated by a solar cell with water electrolysis as a way of achieving zero emissions for the fuel production process of fuel cell electric vehicles (FCEV). Proving tests for the initial development-type SHS1 that used water electrolysis and a mechanical compressor showed that the power consumed by the mechanical compressor accounted for approximately 40% of the total of the system auxiliary equipment<sup>(1), (2)</sup>, and that eliminating the mechanical compressor would effectively further reduce size and increase efficiency for household use. Proton-exchange membrane-type high-pressure water electrolysis is an alternative technology for this, and there are two types: equal pressure and differential pressure. Of these, the equal pressure-type is generally used. Application of differential pressure to the proton exchange membrane can be avoided by making both the cathode and anode high pressure, and housing the water electrolysis cell in a high-pressure vessel. However, this is suited for commercial applications because the size increases. The differential pressure-type was therefore selected as the size can be reduced by making

the water electrolysis cell pressure resistant and making only the cathode high pressure. There are no known cases to date of using the proton exchange membrane to sustain the target pressure difference of 35 MPa. However, the Solar Hydrogen Station SHS2 applies technology that achieves this, and proving tests have been underway since 2010<sup>(3)</sup>. The basic configuration of the high differential pressure water electrolysis cell that is the core technology of the SHS2 was investigated, and this paper describes that cell structure and performance evaluation results.

## 2. Technical Issues for a High Differential Pressure Water Electrolysis Cell

Figure 1 shows a schematic of the high differential pressure water electrolysis cell. The water electrolysis principle is the same as in the case of the atmospheric pressure method, but high-pressure hydrogen is obtained by continuously storing the produced hydrogen in an enclosed space, and only the cathode becomes high pressure.

Equation (1) shows the formula used to obtain the electrolysis voltage  $E_{WE}$  required for high differential pressure water electrolysis.

\* Automobile R&D Center

$$E_{WE} = E_0 + \frac{RT}{2F} \ln \frac{P_1}{P_2} + ir \quad (1)$$

$E_0$ : Theoretical electrolysis voltage [V]

$T$ : Absolute temperature [K]

$R$ : Gas constant [J/(K·mol)]

$P_1$ : High-pressure side hydrogen partial pressure [MPa]

$P_2$ : Low-pressure side hydrogen partial pressure [MPa]

$ir$ : DC resistance voltage [V]

$F$ : Faraday constant [C/mol]

The electrolysis voltage required for high-pressure water electrolysis is obtained by adding the voltage derived from the hydrogen concentration difference to the voltage required for water electrolysis. The Nernst formula shown in the second term on the right side of the formula in Eq. (1) is the voltage from hydrogen concentration difference required by a gas concentration cell<sup>(4)</sup>. The pressurization energy to 35 MPa derived from this concentration difference is estimated to be approximately 180 Wh per 1 Nm<sup>3</sup> of hydrogen at 25°C, and this value does not depend on the cell size or current, that is to say the amount of hydrogen produced. This is a large advantage over compact mechanical compressors for household use.

However, there are three main technological issues that

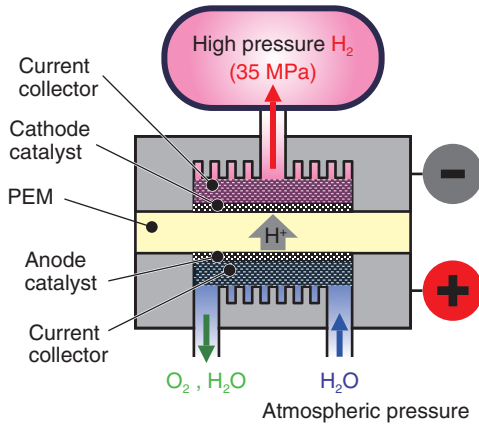


Fig. 1 Schematic of high differential pressure water electrolysis

must be addressed to realize high differential pressure water electrolysis. Figure 2 shows a schematic of these issues and the cell configuration.

The first issue is that the cell must be pressure resistant with respect to the differential pressure between the target produced hydrogen pressure and the anode. This means that the proton exchange membrane (PEM) is used to hold the differential pressure. To achieve this, deformation of the anode current collector should be avoided, and deformation of the proton exchange membrane into the pores present on the current collector surface should also be suppressed. The current collector is porous, and the strength is increased by reducing the surfacial pore and internal pore sizes to approach a bulk body. However, this reduces the fluid permeability required for water electrolysis, so the strength and surfacial pores must be adjusted to simultaneously realize both of these conflicting properties.

The second issue is that unlike water electrolysis, the hydrogen pressure inside the cell generates separation force (hereafter “counter direction load”) between the catalyst and the cathode current collector. Therefore, interfacial pressure should be secured to reduce the contact resistance between the catalyst and the cathode current collector, in accordance with the rise in the hydrogen pressure.

The third issue is an increase in the amount of hydrogen permeation to the anode due to the differential pressure, which is a phenomenon unique to differential pressure-type water electrolysis. Equation (2) shows the theoretical formula of the relationship between the hydrogen pressure and the hydrogen permeation amount<sup>(5)</sup>.

$$N = \alpha \cdot \left( \frac{P_1 - P_2}{d} \right) \cdot \exp \left( \frac{-E_a}{RT} \right) \quad (2)$$

$N$ : Gas permeation speed per unit area [m/sec]

$\alpha$ : Gas permeation coefficient [m<sup>3</sup>·m/(m<sup>2</sup>·sec·MPa)]

$d$ : Membrane thickness [m]

$P_1$ : High-pressure side pressure [MPa]

$P_2$ : Low-pressure side pressure [MPa]

$T$ : Absolute temperature [K]

$R$ : Gas constant [J/(K·mol)]

$E_a$ : Permeation activation energy [J/mol]

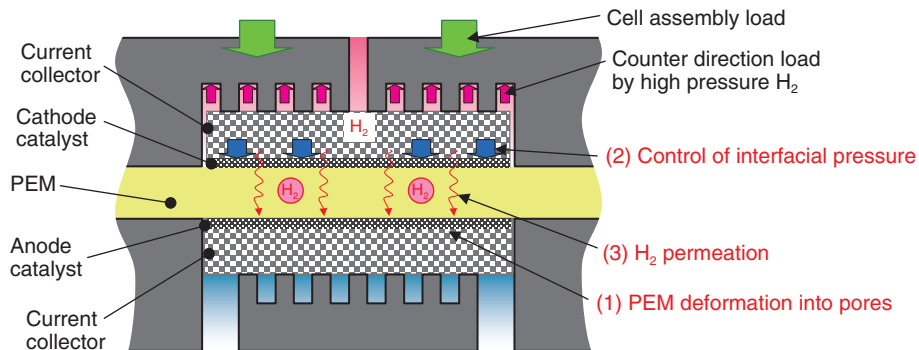


Fig. 2 Three technological issues for high differential pressure water electrolysis cell

According to this formula, the amount of gas cross-permeation across the proton exchange membrane is proportional to the pressure difference, so there is more hydrogen permeation compared to water electrolysis. This permeation reduces the amount of hydrogen that can be obtained relative to the current, and also gives rise to concerns of mixed gas with oxygen. Therefore, high differential pressure water electrolysis must also be investigated from the viewpoint of the gas permeation amount.

This study manufactured a cell that enables investigation of these issues. The goals were to successfully sustain the differential pressure using the proton exchange membrane and to evaluate electrolysis performance under a differential pressure of 35 MPa.

### 3. Experimental Method

#### 3.1. High Differential Pressure Water Electrolysis Cell Structure

##### 3.1.1. Test high-pressure water electrolysis cell

A piston-type high-pressure water electrolysis cell that enables adjustment of the cell internal pressure and the interfacial pressure was manufactured to investigate the issues described in the previous section. Figure 3 shows the sectional drawing.

The basic configuration is the same as that of a water electrolysis cell, but the high-pressure hydrogen generates additional stress, so a high-pressure vessel design was used for the overall cell. In addition, the cathode applied a piston structure to help ensure contact between the current collector and the catalyst. The piston diameter and the cathode current collector diameter are the same, so the pressure difference between the nitrogen gas introduced to the piston and the produced hydrogen is the interfacial pressure. The interfacial pressure was varied in accordance with the counter direction load by adjusting

this nitrogen pressure.

The anode catalyst used the same Ru-Ir-Fe complex oxide catalyst as that used for water electrolysis. A platinum black catalyst was used as the cathode catalyst. These catalysts were hot-pressed against a fluoride-based solid polymer proton exchange membrane with a thickness of 100  $\mu\text{m}$ . Titanium was used as the cell material as in the case of a water electrolysis cell.

##### 3.1.2. Current collectors

The anode current collector is a key part for sustaining the differential pressure, and spherical titanium powder sintered into a slab shape was used as this material. Figure 4 shows photographs of the external appearance and an enlarged view of the anode current collector. The sintering conditions were adjusted to increase the strength and fineness of this current collector while still retaining the fluid permeability required for water electrolysis.

The cell structure uses the proton exchange membrane to sustain the differential pressure of 35 MPa. Enhancing the surfacial pore fineness of the anode current collector enabled to suppress proton exchange membrane deformation, strengthen the current collector to withstand deformation due to the high-pressure gas load, and support the entire proton exchange membrane. These components

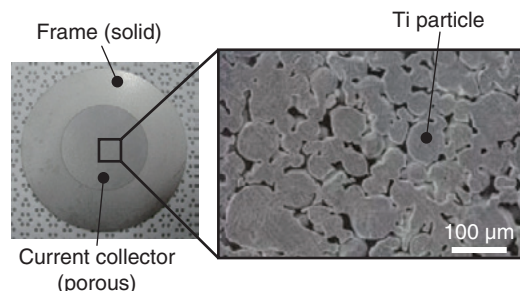


Fig. 4 Current collector

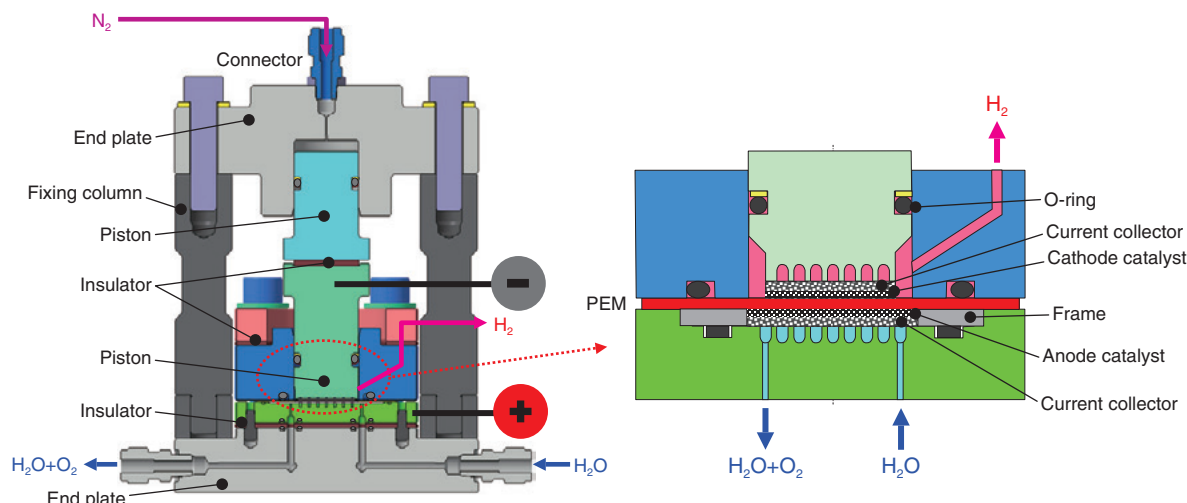


Fig. 3 Structure of experimental electrolysis cell

are stacked together to manifest the differential pressure sustaining function. The same structure was also used for the cathode current collector.

### 3.2. Confirmation of Differential Pressure Sustaining Function and Test Method for Measuring the Hydrogen Cross-permeation Amount

Figure 5 shows a schematic of the system used for the series of tests in this study.

The proton exchange membrane was soaked in purified water and placed on top of the current collector, then the components were bolted together to prevent gaps from occurring between each component due to the counter direction load of the high-pressure hydrogen. Purified water was circulated and the water temperature was adjusted using a heater. Nitrogen was introduced to the cathode by a compressor, and the internal pressure was regulated by a back-pressure valve. The internal pressure was measured using a pressure transmitter, and the pressure resistance was judged according to whether there was a rapid drop in pressure other than the drop in pressure due to cross-permeation across the membrane.

To measure the amount of hydrogen permeation, hydrogen was introduced by a compressor, the hydrogen that permeated across the membrane was collected by the circulation water, and the amount of hydrogen permeation per unit time was obtained from the product of the nitrogen dilution flow rate measured using a mass flow controller and the hydrogen concentration measured using a gas chromatograph. The permeation area was the anode current collector area.

### 3.3. Test Method for Evaluation of Electrolysis Performance

First, to determine the appropriate interfacial pressure between the current collector and the catalyst, the nitrogen

pressure was adjusted to vary the interfacial pressure from 0.2 MPa to 40 MPa while performing water electrolysis at atmospheric pressure, and the relationship between the interfacial pressure and the electrolysis voltage was evaluated.

Next, high-pressure water electrolysis was performed at 35 MPa with the appropriate interfacial pressure determined by the above operation, and the relationship between the hydrogen pressure and voltage efficiency was evaluated. The voltage efficiency is obtained by Eq. (3).

$$\eta_V = \frac{E_0}{E_{WE}} \times 100 \quad (3)$$

$\eta_V$  : Voltage efficiency [%]

$E_0$  : Theoretical electrolysis voltage = 1.48 [V]

$E_{WE}$  : Electrolysis voltage [V]

The generated hydrogen passes through a back-pressure valve and is then depressurized to atmospheric pressure and separated from the water by a hydrogen separator. The separated hydrogen flow rate was measured by a wet-type integrating flow meter. Equation (4) was used to calculate the current efficiency from the current and the integral value of the obtained hydrogen per unit time. Part of the hydrogen was dehumidified and the concentration of oxygen, which is an impurity, was measured using a gas chromatograph. The generated oxygen was separated using unreacted water and an oxygen separator, diluted by nitrogen, and exhausted.

$$\eta_A = Q \cdot \left( \frac{I_{WE} \cdot 3600}{F} \cdot \frac{1}{2} \right)^{-1} \times 100 \quad (4)$$

$\eta_A$  : Voltage efficiency [%]

$Q$  : Generated hydrogen amount [mol/hr]

$I_{WE}$  : Electrolysis current [A]

$F$  : Faraday constant [C/mol]

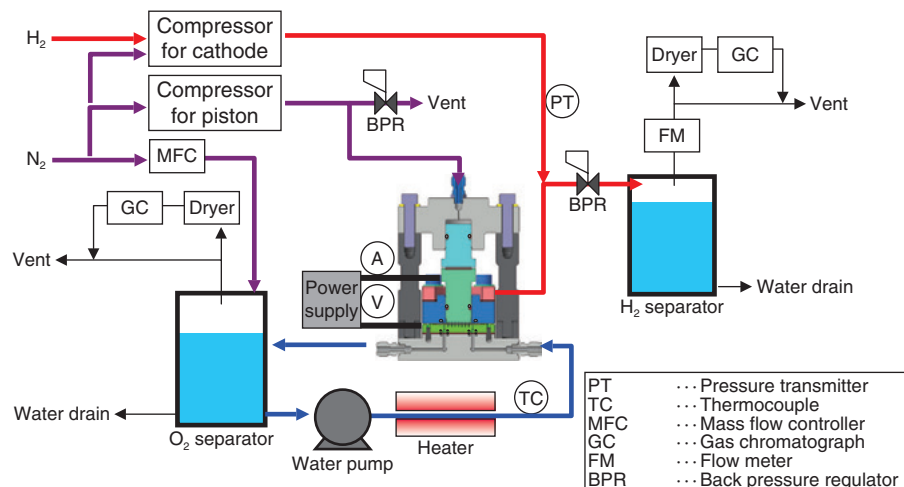


Fig. 5 Evaluation system

## 4. Results and Discussion

### 4.1. Test to Confirm Differential Pressure Sustaining Function and Test to Measure the Hydrogen Cross-permeation Amount

#### 4.1.1. Test to confirm differential pressure sustaining function

Tests to confirm that the differential pressure of 35 MPa was sustained were performed before performing actual high-pressure water electrolysis. Figure 6 shows the time series change in the cathode side pressure due to nitrogen pressurization by the compressor. Nitrogen introduction was stopped to check for pressure drops at 1, 5, 10, 15, 25 and 35 MPa. No drop in pressure other than the slight decrease due to cross-permeation across the membrane was observed during this operation, which confirmed that the differential pressure of 35 MPa was sustained by the proton exchange membrane.

Typical proton exchange membranes have a tensile strength of approximately 20 MPa to 30 MPa at room temperature, so they are unable to sustain this differential pressure. It is presumed that in the developed cell structure, reducing the size of the pores in the anode current collector that supports the membrane suppressed proton exchange membrane deformation so that break elongation was not reached, even in the state with pressure of 35 MPa applied.

#### 4.1.2. Test to measure the hydrogen cross-permeation amount

Next, the amount of hydrogen cross-permeation across the membrane during high differential pressure water electrolysis was confirmed. Figure 7 shows the relationship between the hydrogen pressure and the membrane permeation amount, and the estimated current efficiency values calculated from the membrane permeation amount. The membrane permeation amount increases linearly together with the rise in hydrogen pressure, and the trend matched the theoretical formula for the membrane permeation amount given in Eq. (2).

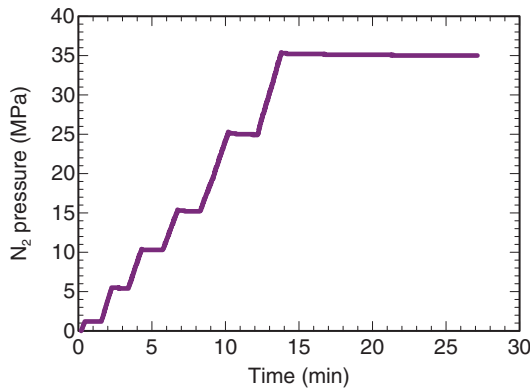


Fig. 6 Pressure sustaining characteristics relative to change in N<sub>2</sub> pressure over time (30°C)

Under conditions of 30°C and 35 MPa, the membrane permeation amount was 0.20 cm<sup>3</sup>/min·cm<sup>2</sup>. This is approximately 2.8% of the amount of hydrogen obtained at a current of 1 A/cm<sup>2</sup>, and becomes the loss due to cross-permeation to the anode, so a current efficiency of approximately 97.2% can be predicted. In addition, this cross-permeated hydrogen forms a mixed gas with a concentration of approximately 5.6% at the anode. This is a combustible concentration, so the reactivity inside the cell was investigated from the two points of the mixed gas state and the presence of an ignition source in order to confirm the effects of this mixed gas on high differential pressure water electrolysis cell function.

As shown in Fig. 5, in order to supply sufficient water to the proton exchange membrane, the high differential pressure water electrolysis cell circulates several times the amount of water used for the reaction. The oxygen and water in this state can be predicted to conform to slag flow in which the gas is separated as bubbles<sup>(6)</sup>, so even if an ignition source is present the reaction does not continue. This is confirmed by the method reported by Inoue et al. of suppressing combustion propagation by spraying water mist onto hydrogen combustion waves in the presence of inert gas in order to lower the temperature and pressure<sup>(7)</sup>. In addition, reactivity between the anode catalyst and the hydrogen-oxygen mixed gas was investigated to determine the possibility of the catalyst becoming an ignition source. However, there was no rise in the catalyst surface temperature even at a stoichiometric ratio concentration of 2:1.

The above results indicate that the drop in current efficiency due to the differential pressure is slight, and that water electrolysis cell function is not affected by the generated mixed gas.

### 4.2. Test to Evaluate Electrolysis Performance

#### 4.2.1. Determination of appropriate interfacial pressure by water electrolysis

First, the water electrolysis voltage was measured while varying the interfacial pressure to determine the target

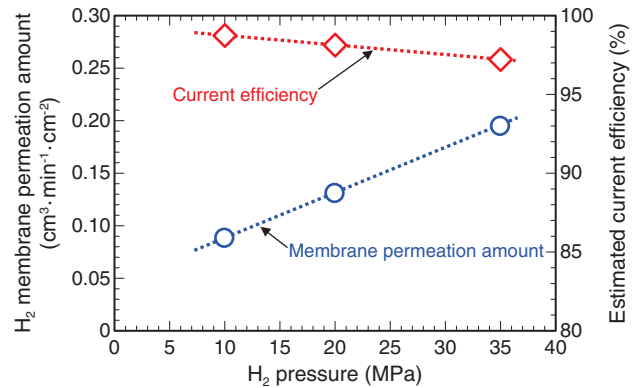


Fig. 7 Influence on H<sub>2</sub> permeation and estimated current efficiency (30°C)

interfacial pressure. Figure 8 shows these results. As the interfacial pressure was increased from around 0, the voltage dropped sharply, exhibited a minimum value in the 3 MPa to 10 MPa range, and then rose gradually towards 40 MPa.

The range with an interfacial pressure of 3 MPa or less is thought to be the range where the contact resistance increases between a current collector that has been smoothed for differential pressure water electrolysis and a catalyst. At high interfacial pressures of 10 MPa or more, other factors that reduce efficiency are present. This behavior is presumed to occur because the interfacial pressure ejects water from the moistened proton exchange membrane, which reduces the proton conductivity and increases the membrane resistance.

Based on these results, 5 MPa was selected as the appropriate interfacial pressure, as this pressure has the least influence on contact resistance and membrane resistance.

#### 4.2.2. Production of high-pressure hydrogen by self-pressurization and relationship between hydrogen pressure and voltage efficiency

Next, actual high-pressure water electrolysis was performed, and the relationship between the hydrogen pressure and the voltage was simulated and evaluated for the two cell types of the pre-set interfacial pressure-type and the constant interfacial pressure-type. Figure 9 shows schematics of each cell structure.

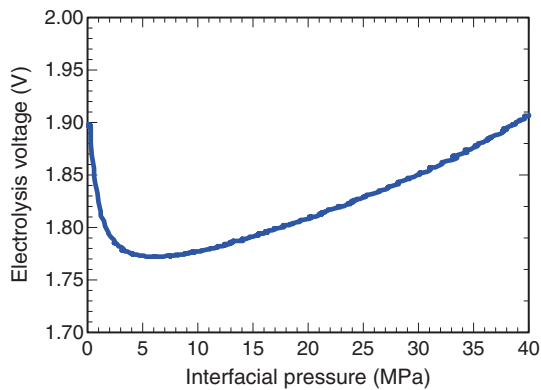


Fig. 8 Influence of interfacial pressure on electrolysis voltage (30°C, 1 A/cm<sup>2</sup>)

The pre-set interfacial pressure-type is also used for water electrolysis, and has a cell structure that clamps the electrodes from the edges of the cell using bolts or other fasteners. There is no need for a mechanism that applies interfacial pressure inside each cell, so there are the merits of a small number of parts and reduced thickness for stacking. However, when applied to a high differential pressure water electrolysis cell, the interfacial pressure drops as the hydrogen pressure inside the cell increases. Therefore, the sum of both loads of the hydrogen pressure of 35 MPa and the interfacial pressure must be applied during assembly. On the other hand, the constant interfacial pressure-type has a structure that maintains a constant interfacial pressure regardless of the hydrogen pressure, by adding an elastic part (Coned disc spring in Fig. 9) installed inside the cell. Figure 10 shows the results of evaluating the relationship between the voltage and hydrogen pressure while adjusting the piston internal pressure to apply interfacial pressure that simulates each structure.

As the hydrogen pressure changed from atmospheric pressure to 35 MPa, the interfacial pressure for the pre-set interfacial pressure-type changed from 40 MPa to the target 5 MPa. However, the constant interfacial pressure-type

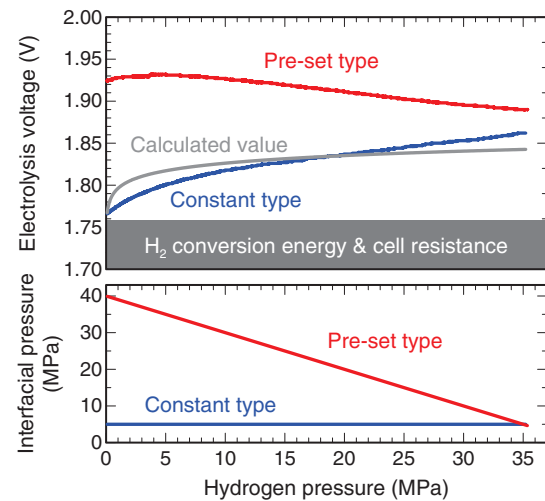


Fig. 10 Comparison of electrolysis voltage and interfacial pressure (30°C, 1 A/cm<sup>2</sup>)

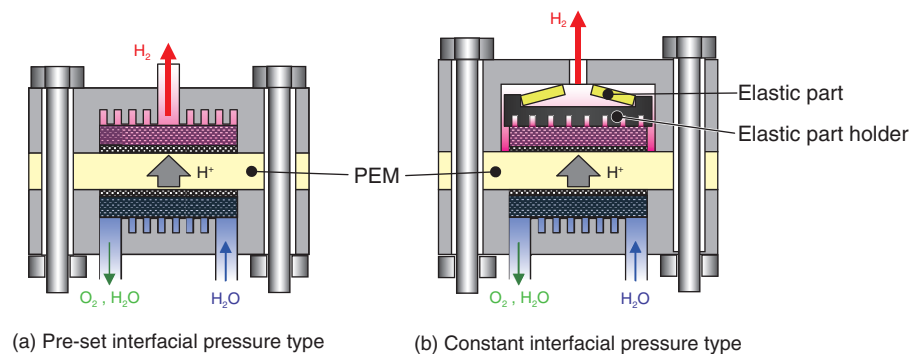


Fig. 9 Comparison of fastening mechanism

maintained a constant interfacial pressure of 5 MPa. During this operation, the pre-set interfacial pressure-type required a voltage of approximately 1.93 V at atmospheric pressure, and this voltage decreased as the pressure increased. The constant interfacial pressure-type required a voltage of approximately 1.76 V at atmospheric pressure, which is lower than that of the pre-set interfacial pressure-type, and this voltage then increased gradually as the pressure increased. The voltage for both types was approximately 1.9 V at 35 MPa, indicating that electrolysis could be performed at the target interfacial pressure.

The pre-set interfacial pressure-type experiences excessive interfacial pressure during pressurization from atmospheric pressure. This means that electrolysis is performed in the state with high membrane resistance, resulting in low voltage efficiency. This matched the tendency measured in the high interfacial pressure test shown in Fig. 8. The constant interfacial pressure-type maintains a constant interfacial pressure, so the effects of contact resistance and membrane resistance are small, and the behavior was close to the calculated concentration difference voltage values.

There is no major difference in voltage efficiency between both structures when steady-state electrolysis is performed at 35 MPa. However, during pressurization, the constant interfacial pressure-type that is not subject to a drop in voltage efficiency was found to be advantageous. In addition, the actual hydrogen pressurization energy was 183 Wh per 1 Nm<sup>3</sup> at 30°C, which means that energy consumption could be reduced to approximately 1/4 that of the mechanical compressor used in the SHS1<sup>(2)</sup>.

#### 4.2.3. Evaluation of electrolysis performance

Figure 11 shows the current-voltage performance curves (hereafter “IV curve”) from atmospheric pressure to 35 MPa, at 60°C. The IV curves from 10 MPa to 35 MPa trend in parallel to the atmospheric pressure IV curve, and there is almost no change in the resistance component

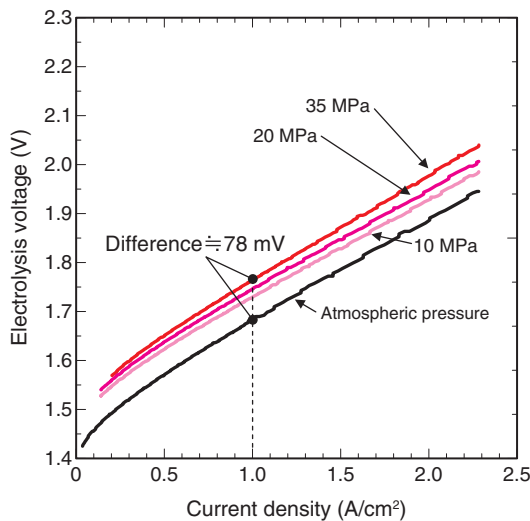


Fig. 11 Effect of H<sub>2</sub> pressure on IV curve (60°C)

due to hydrogen pressurization during high differential pressure water electrolysis. This behavior indicates that the relationship between the hydrogen pressure and voltage for the constant interfacial pressure-type shown in Fig. 10 matches that predicted by the theoretical formula for pressurization energy. The difference between the voltages at atmospheric pressure and 35 MPa was 78 mV at a current density of 1 A/cm<sup>2</sup>, and this value is close to the theoretical concentration difference voltage of 84 mV. In addition, the IV curve trended at a constant slope up to the high current density of 2.3 A/cm<sup>2</sup>, indicating that sufficient water is supplied to the catalyst in the high differential pressure water electrolysis cell.

Figure 12 shows the IV curve for each temperature at 35 MPa. The IV curve slope decreased as the temperature increased from 30°C to 60°C. The voltage efficiency at 1 A/cm<sup>2</sup> was 77% at 30°C, and increased to 84% at 60°C. This behavior was the same as that for water electrolysis<sup>(2)</sup>, indicating that the differential pressure had no effect on the relationship between temperature and membrane resistance.

Figure 13 shows the time series change in current

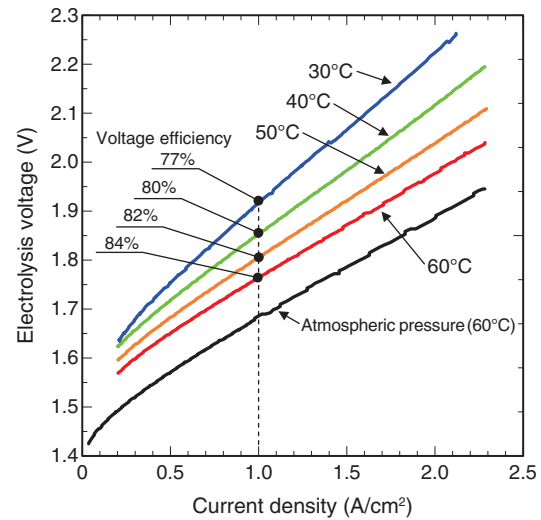


Fig. 12 Effect of temperature on IV curve (35 MPa)

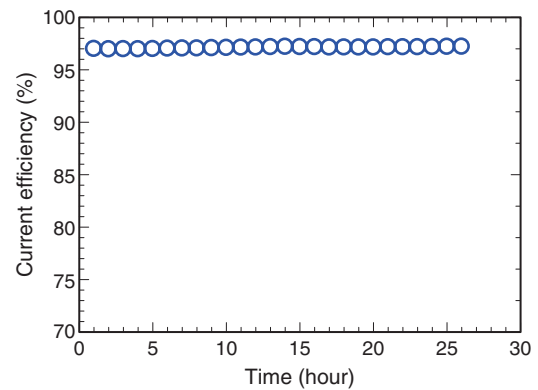


Fig. 13 Duration of current efficiency (integrated values for one hour, at 30°C)

efficiency. Water electrolysis has almost no hydrogen loss, so the current efficiency is approximately 100%. In contrast, the current efficiency at 35 MPa was approximately 97%. This matches the measurement results for the amount of hydrogen cross-permeation across the membrane, and confirmed that there is no loss other than cross-permeation across the membrane. In addition, there was also no drop in current efficiency over time, so it is thought that the proton exchange membrane shape was maintained and there was no deformation at a differential pressure of 35 MPa.

#### 4.2.4. Evaluation of hydrogen quality

Figure 14 shows the pressure dependency of the oxygen concentration in the produced hydrogen. The oxygen concentration was approximately 33 ppm at atmospheric pressure, and decreased rapidly as the pressure increased, reaching approximately 5 ppm at around 1 MPa.

This behavior indicates that movement of oxygen from the anode to the cathode was suppressed due to the pressure difference. According to ISO 14687-2 Hydrogen fuel - Product specification<sup>(8)</sup>, the oxygen concentration must be 5 ppm or less, and high differential pressure water electrolysis was found to satisfy this concentration condition under steady-state operating conditions. To increase the purity of the produced hydrogen, water electrolysis had to react oxygen in excess of the specified concentration with hydrogen and exhaust it as water. This meant that the system had to be equipped with a reactor, and a process to remove the generated water after the reaction was also necessary. However, these are not needed for high differential pressure water electrolysis.

In addition, the amount of saturated steam in the high-pressure gas is inversely proportional to the pressure, and the moisture concentration that is several ten thousand ppm at atmospheric pressure and 30°C decreases to approximately 135 ppm at 35 MPa. The abovementioned hydrogen fuel product specification also prescribes a water content in the produced hydrogen of 5 ppm or less. However, the dehumidification energy can also be greatly reduced compared to water electrolysis.

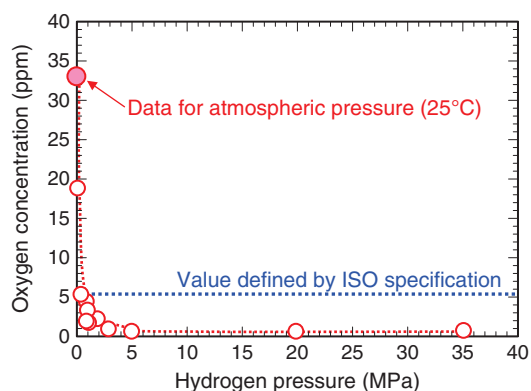


Fig. 14 Oxygen concentration in produced hydrogen (60°C)

These results confirm that a hydrogen refueling system using high differential pressure water electrolysis can simplify the device structure and increase efficiency compared to conventional systems.

## 5. Conclusion

The basic configuration of a high differential pressure water electrolysis cell was investigated, and tests were performed to confirm the differential pressure sustaining function and to evaluate electrolysis performance. The following conclusions were obtained.

- (1) Differential pressure of 35 MPa was sustained by the proton exchange membrane by reducing the surfacial pore size and using a powder sintered material with increased strength as the anode side current collector. In addition, hydrogen cross-permeating across the membrane due to the differential pressure was not reactive at the anode, so the basic configuration for a high differential pressure water electrolysis cell was established.
- (2) High differential pressure water electrolysis cell structures were investigated, and the constant interfacial pressure-type that maintains a constant interfacial pressure regardless of the hydrogen pressure inside the cell was found to have higher voltage efficiency than the pre-set interfacial pressure-type used by water electrolysis.
- (3) Hydrogen with a pressure of 35 MPa was produced using the theoretical concentration difference voltage. The current efficiency was 97% as a result of hydrogen cross-permeation across the membrane due to the differential pressure, and there was no change in this value over time.
- (4) It was found that when hydrogen with a pressure of 35 MPa is produced by high differential pressure water electrolysis, the oxygen concentration is much less than the ISO specification of 5 ppm, and an oxygen removal process is not necessary. Only dehumidification is required to achieve the specified hydrogen purity.

## References

- (1) Ueyama, M., Kita, M., Taguchi, S., Nakazawa, K., Nagaoka, H., Arakawa, S.: Introduction of Hydrogen Refueling Station with Photovoltaic Modules, Honda R&D Technical Review, Vol. 15, No. 2, p. 31-38
- (2) Okabe, M., Nakazawa, K., Taruya, K., Handa, K.: Verification Test of Hydrogen Refueling Station with Photovoltaic Modules, Honda R&D Technical Review, April, 2008, p. 67-73
- (3) Nakazawa, K., Nagaoka, H., Takeuchi, J., Moburg, F., Yamashita, I., Okabe, M.: Introduction of High Differential Pressure Water Electrolysis-type Solar Hydrogen Station, Honda R&D Technical Review, October, 2010, p. 28-35

- (4) The Electrochemical Society of Japan: Electrochemical handbook, Vol. 5, p. 155 (2000)
- (5) Sakai, K., et al. : Makubunri process no Riron to Sekkei, IPC, p. 196-197 (1993) (in Japanese)
- (6) The Society of Chemical Engineers, Japan: Chemical engineering handbook, Vol. 6, p. 298 (1999)
- (7) Inoue, M., Nakahara, M.: Research on the Control of the Propagation of Hydrogen Combustion by Water Spray, Hydrogen Energy Systems, Vol. 34, No. 3, p. 23-31 (2009)
- (8) ISO14687-2: Hydrogen fuel – Product specification – Part 2, Type 1, Grade D

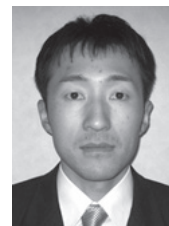
■ Author ■



Eiji HARYU



Koji NAKAZAWA



Kenji TARUYA



Hiroyuki ISHIKAWA



Masanori OKABE

# Influence of Chassis Frictions on Vehicle Dynamic Behavior: Analytical Study

Jinghong YU\*

## ABSTRACT

An extensive analytical study was performed to analyze the influence of chassis frictions on vehicle dynamic behavior and identify the relative importance of frictions in different locations of a chassis system. First, a multi-body simulation model of a full vehicle was developed that integrates customized friction functions in all major chassis joints. The model was correlated with measurement in chassis kinematics, compliance, system-level frictions, and vehicle dynamic performance. Using the simulation, along with the confirmation of two test vehicles, seven key vehicle performance areas affected by chassis frictions were identified. This was followed by a thorough sensitivity analysis to identify frictions that have high influence on each of those performance areas. Common influential frictions that have a considerable influence on multiple performance areas were identified.

## 1. Introduction

Vehicle dynamics and comfort have become two principal vehicle attributes that customers judge when deciding on a vehicle purchase. However, simultaneously achieving these attributes has been a challenging task for automotive engineers. It is even more challenging to achieve in passenger cars with high-power front-wheel drive and in vehicles with a high center of gravity, such as sport utility vehicles.

Friction exists in any mechanical joint in an automotive chassis system, and many frictions can have considerable effect on vehicle performance. Passively applying supplemental frictions at a chassis location, such as in the steering gearbox, is a common approach used to enhance system toughness and performance. Nevertheless, while one friction setting can enhance vehicle performance in one area, it may affect performance in other areas.

The influence of chassis frictions has not been studied thoroughly to determine the influence of a specific friction at a given location to a performance area; what level of friction is best for a given performance area; how a required friction level can be realized; and more importantly, how to appropriately control chassis frictions to enhance multiple performance areas without compromising others areas.

To answer these questions, systematic research was conducted. As the initial step in this research, an extensive analytical study was performed to quantify the influence of chassis frictions on various vehicle performance areas and to identify the relative importance of those frictions.

A broad spectrum of performance in dynamics and comfort was analyzed, including torque steering, vehicle drift and pull, brake judder, vehicle gross body motion control, vehicle body roll control, steering feel at both low and high speeds, steering kickback, impact harshness, driving vibration, and self-steer. This paper explains first the development of a full vehicle multi-body simulation model that integrates customized friction functions in all major chassis joints. The model was correlated with the measurement of chassis kinematics, compliance, system-level frictions, and vehicle dynamic performance. Using the simulation and a confirmation with two test vehicles, key vehicle performance areas affected by chassis frictions were then identified. Next, a virtual sensitivity analysis was done to find influential frictions for each of the affected performance areas. From this research, common influential frictions that have influence on multiple performance areas were identified.

## 2. System Modeling

### 2.1. Modeling Approach

A multi-body simulation model was developed for a sport utility vehicle equipped with a MacPherson front suspension, trailing-arm rear suspension, and a rack-and-pinion steering system. To build the model, an Adams/Car environment was used due to its ability to assemble different sub-systems.

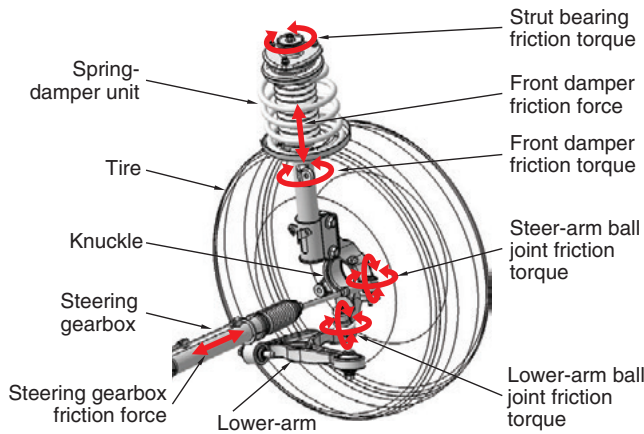
In order to capture critical effect of chassis frictions on vehicle performance, custom templates and sub-systems

\* HONDA R&D AMERICAS, INC.

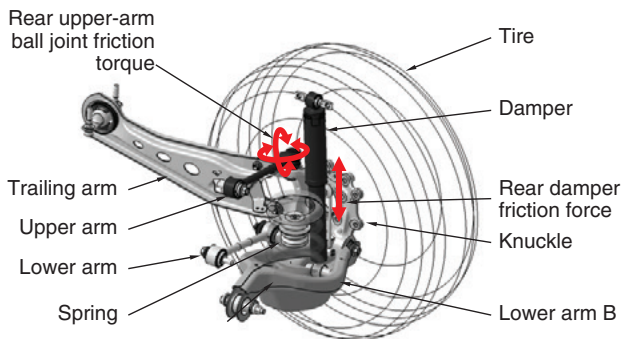
were created to include frictions at major chassis joints in translational, rotational, and rocking directions. Figure 1 shows the frictions that were considered in the study, which included strut bearing friction torque, front damper friction force, front damper friction torque, and lower-arm ball joint friction torque for the front suspension, steer-arm ball joint friction torque and steering gearbox friction force for the steering system, and upper-arm ball joint friction torque and rear damper friction force for the rear suspension.

In order to help achieve numerical simulation stability and replicate a realistic friction profile, velocity-dependent friction functions (Fig. 2) are used to describe the chassis frictions. The velocity-dependent friction profile allows the breakthrough friction to exponentially decay to the kinetic one while guarantying continuity at zero relative velocity. The breakthrough friction and the ratio of kinetic friction to breakthrough friction are measured by using related components for each defined chassis friction.

Figure 3 shows the full vehicle simulation model, which includes the front chassis, rear chassis, power train system, tires and wheels, rigid body, front and rear stabilizer bars, front and rear sub-frames, and chassis frictions. Displacement-dependent stiffness and damping characteristics were considered for all mount bushings. A



(a) MacPherson front suspension and steering system



(b) Trailing-arm rear suspension

Fig. 1 Frictions in vehicle chassis system

Pacejka tire model with relaxation consideration was used for the tires in order to capture both low-speed and high-speed behavior. The complete vehicle model functions as a single tool to analyze multiple dynamic phenomena.

## 2.2. Vehicle Model Validation

In order to systematically validate the vehicle simulation model, the simulation results of chassis kinematic, compliance, and friction characteristics were compared in sequence with the measurement data of the vehicle. The simulation model was then compared with the test vehicle in multiple performance areas.

The measured kinematic and compliance characteristics are captured on a suspension parameter measurement machine (SPMM). During the measurement of vertical friction of suspension systems, a quasi-static loading cycle in vertical direction on the SPMM was carried out, and the vertical friction was then determined as half of the hysteresis bandwidth of the suspension displacement-loading force curve. During the measurement of steering friction torque of the front suspension and steering system, the vehicle was supported at the front tire contact patches by a pair of friction-free bearings, and a quasi-static steering lock-to-lock cycle was performed. Half of the hysteresis bandwidth of the steering angle-torque was used as the steering friction torque of the chassis system.

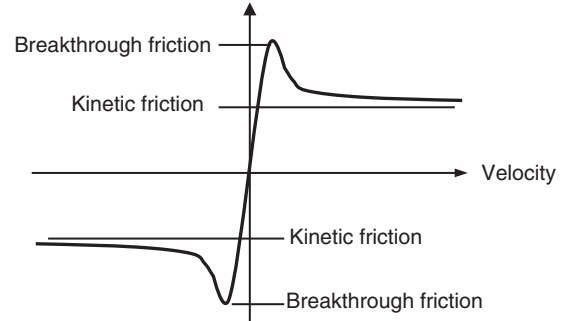


Fig. 2 Velocity-dependent friction profile used to define chassis frictions

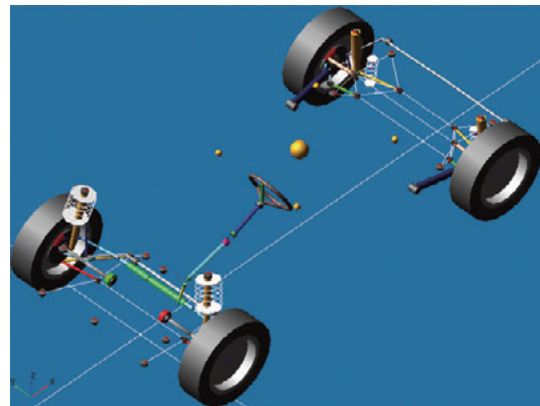


Fig. 3 Multi-body simulation model for complete vehicle

Table 1 Friction specifications of the Low-Friction and High-Friction vehicles

| Friction name                                  | Low-Friction | High-Friction |
|------------------------------------------------|--------------|---------------|
| Lower-arm ball joint friction torque (Nm)      | 7            | 21            |
| Steer-arm ball joint friction torque (Nm)      | 0.5          | 11            |
| Strut bearing friction torque (Nm)             | 1.7          | 8             |
| Steering gearbox friction force (N)            | 100          | 175           |
| Rear upper-arm ball joint friction torque (Nm) | 1            | 17.5          |

During the validation of compliance characteristics, a common stiffness scaling factor was included for all suspension bushings, sub-frame bushings, and steering gearbox mounting bushings, while during the friction correlation of the simulation model, a common friction scaling factor was used for all frictions in the chassis system.

Three vehicles (High-Friction vehicle, Low-Friction vehicle, and Production vehicle) were specially built to validate the model. The vehicles all had the same specifications except for different friction levels in the following chassis parts: strut bearing, lower-arm ball joint, steer-arm ball joint, steering gearbox, and the upper-arm ball joint of the rear suspension. These parts were chosen based on the production line capability for manufacturing parts with varying friction specifications. The differences in friction specifications between the Low-Friction and the High-Friction Vehicles are listed in Table 1. The ratio of system-level frictions between the High-Friction and Low-Friction vehicles achieved 1.3 in vertical direction of the front suspension and 1.6 in the steering rotational direction.

Whereas the kinematic and compliance correlation was mainly based on the Production vehicle, the friction correlation was performed on all three vehicles. Figure 4 illustrates an exemplary comparison between simulated and measured suspension kinematic and compliance characteristics. As shown, the simulated bump toe, bump camber, lateral compliance, and longitudinal compliance have fairly good agreement with the measurement. Similarly, good correlation results are obtained for other kinematic and compliance parameters, such as steering ratio, roll stiffness, wheel rate, and toe compliance.

Figure 5(a) demonstrates the measured and simulated suspension vertical frictions. As shown, the simulated system-level friction results closely match those from measuring all three vehicles and for both front and rear suspension systems. Figure 5(b) shows close correlation between simulated and measured steering friction torques of the chassis system in all three vehicles.

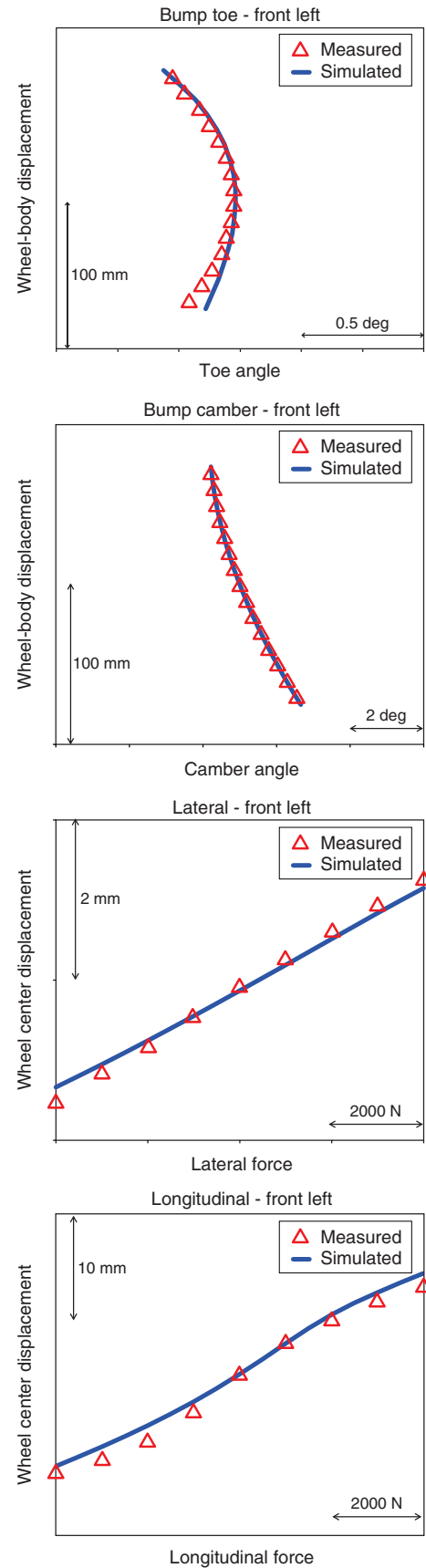


Fig. 4 Comparison of simulated and measured chassis kinematic and compliance characteristics

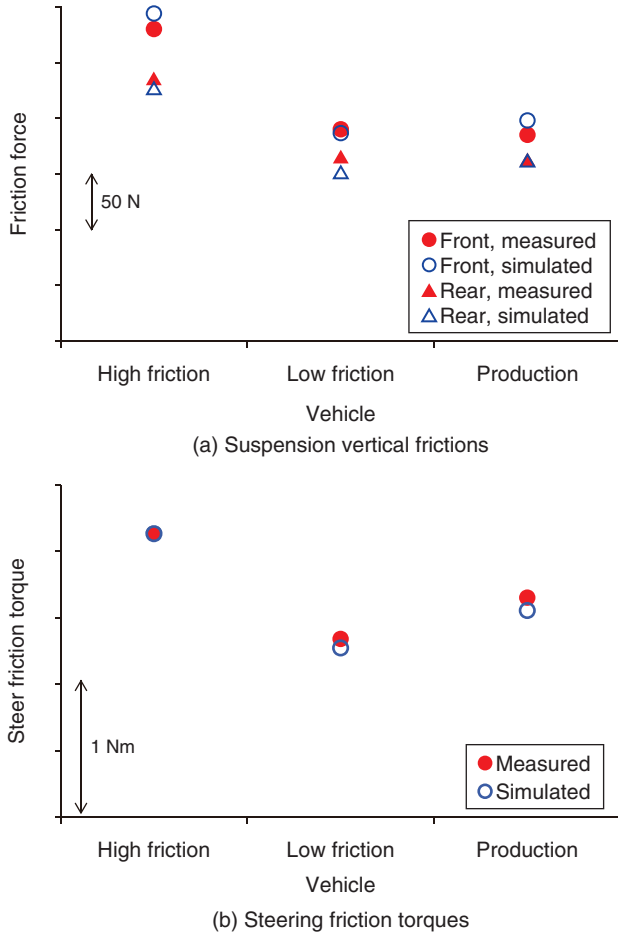


Fig. 5 Comparison of simulated and measured system-level friction characteristics

### 3. Performance Areas Affected by Frictions

#### 3.1. Quantifying the Influence of Frictions

The High-Friction and Low-Friction vehicles were used to evaluate the relative performance change in each area and the evaluation results were used to further validate the simulation model.

The performance areas covered in the investigation include high-speed steering feel, low-speed steering feel, torque steer, vehicle drift and pull, brake judder sensitivity, gross body motion, body roll motion, steering shimmy, steering kickback, impact harshness, driving vibration caused by the tire-wheel's second order imbalance, and self-steer. High-speed steering feel is focused on the on-center steering angle-torque characteristics that can be presented by a steering damping factor and a spring coefficient<sup>(1)-(4)</sup>, while low-speed steering feel is evaluated by the nimble feeling and maneuverability at low speeds. One of the key low-speed steering feel attributes is the self-centering ability of the steering wheel when it is released during a steady-state turning event. It is quantified by the steering wheel returning angle from a given angle in a certain period of time after the steering wheel is released.

Torque steer is when a vehicle drifts to one side during hard acceleration from zero speed on a split- $\mu$  surface, which can be judged by the steering angle change during a hands-free acceleration event. Self-steer is the extent of steering effort degradation with the increase of steering input during cornering. Vehicle drift (and pull) is normally experienced during middle- to high-speed straight-line driving.

On the other hand, chassis sensitivity of brake judder and steering shimmy can be quantified as the transmitted vibration at the steering wheel when the front knuckles are excited by a given level of brake torque fluctuation or tire-wheel imbalance<sup>(5), (6)</sup>. Whereas gross body motion is judged by the vehicle body motion in vertical direction, body roll motion is evaluated by the combined effect of the first and second overshoots of roll angle, roll rate, and yaw rate when there is an abrupt change in steering angular input. Steering kickback is measured as the steering effort fluctuation while impact harshness is quantified by a specially-designed index for the impact to vehicle seats, floor, and steering wheel, when the vehicle runs over an obstacle on a road.

Extensive simulation and measurement have been performed for the performance areas described above. Figure 6 shows the simulated and measured results of the

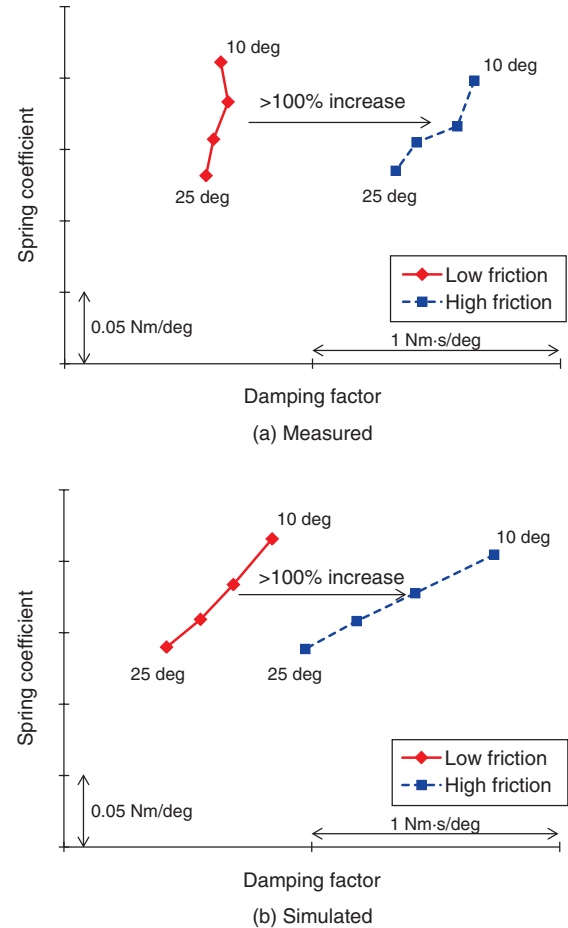


Fig. 6 Measured and simulated on-center steering feel characteristics (at high speed)

high-speed steering feel characteristics under a sinusoid steering input (at 0.5 Hz). It suggests that chassis frictions can affect the steering damping factor but maintain steering spring coefficient for various amplitudes of steering input. It was also confirmed that the simulation can accurately predict the trend.

Figure 7 compares the torque steer performance between the High-Friction and Low-Friction vehicles during hard acceleration from zero speed. As shown, the steering wheel angle change is reduced about 80% from the Low-Friction to High-Friction vehicle, and the simulation model accurately predicts the maximum steering angular rotation.

Figure 8 shows how body roll behavior changes between the Low-Friction and High-Friction vehicles when the steering wheel is released after a steady-state steering input of 60 degrees. As shown, increasing chassis frictions results in a reduction in the second overshoots and a mild decrease in the first overshoots of roll angle, roll rate, and yaw rate. It is further shown that the friction change from the Low-Friction vehicle to the High-Friction vehicle is able to reduce the number of roll rate and yaw rate overshoots from four to two. This can enhance a stable performance

feel during abrupt maneuvers, such as making an emergency lane change or avoiding an obstacle.

These examples suggest that the higher the chassis frictions, the better the vehicle performance in torque steer, body roll control, and high-speed steering damping. However, this trend does not hold true for low-speed steering feel. During low-speed maneuvers, such as in

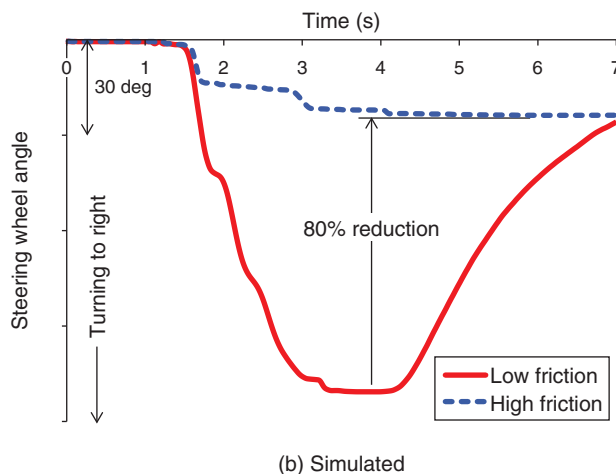
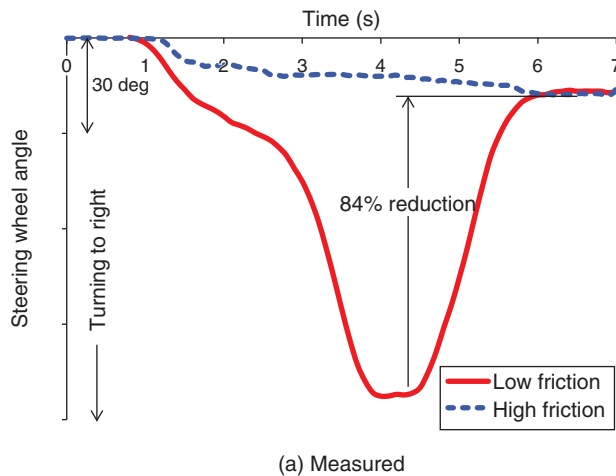
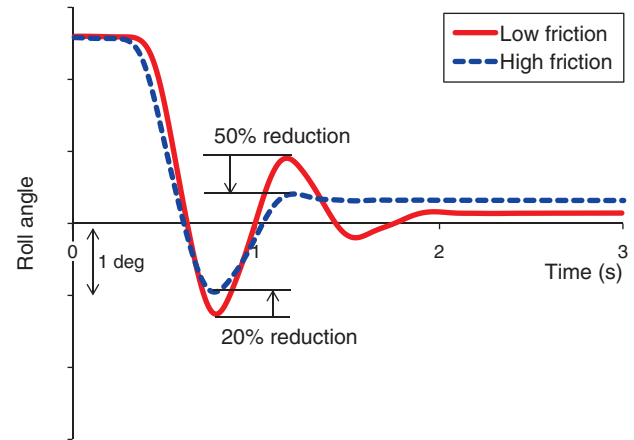
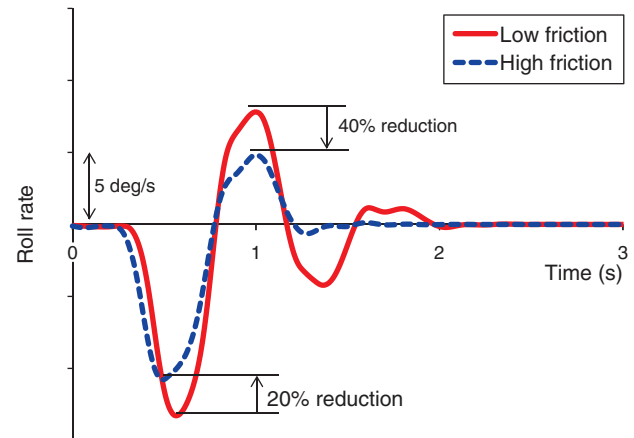


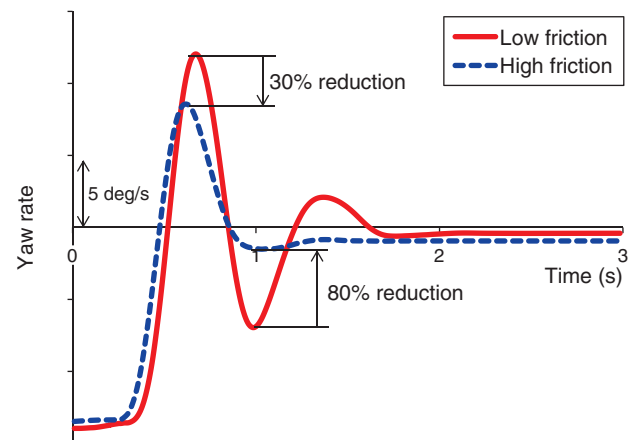
Fig. 7 Torque steer during hard acceleration from zero speed



(a) Roll angle



(b) Roll rate



(c) Yaw rate

Fig. 8 Simulated vehicle roll motion control

city driving or parking, lower levels of chassis frictions are desired to achieve a nimble and clear steering feel. Figure 9 shows the effect of frictions on low-speed steering feel by comparing the steering wheel self-centering ability of the High-Friction vehicle versus Low-Friction vehicle. Due to the increase in frictions, the self-centering ability of the High-Friction vehicle is reduced by almost 50%.

### 3.2. Performance Areas Influenced by Frictions

In order to compare the impact of chassis frictions on different vehicle performance areas, the percentage of performance change from the Low-Friction vehicle to the High-Friction vehicle is calculated for each performance area. The absolute values of the percentage results are then summarized as relative changes in performance in Fig. 10.

In general, one quarter (or 25%) change in performance can be considered as significant enough for normal drivers to feel the difference. Thus, as shown in Fig. 10, chassis frictions can considerably affect the following performance areas: high-speed steering feel, torque steer, vehicle drift, low-speed steering feel, body roll motion, brake judder, and gross body motion.

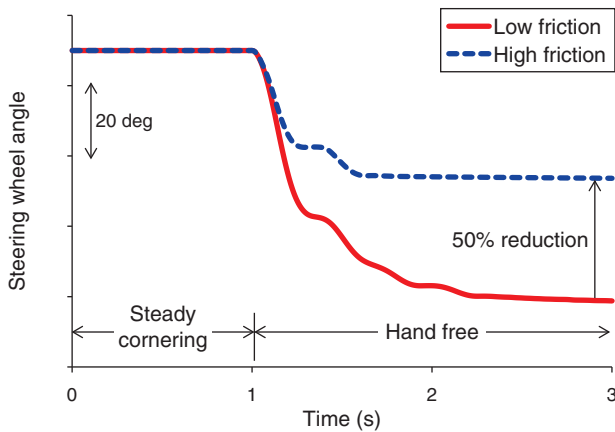


Fig. 9 Simulated steering wheel self-centering ability

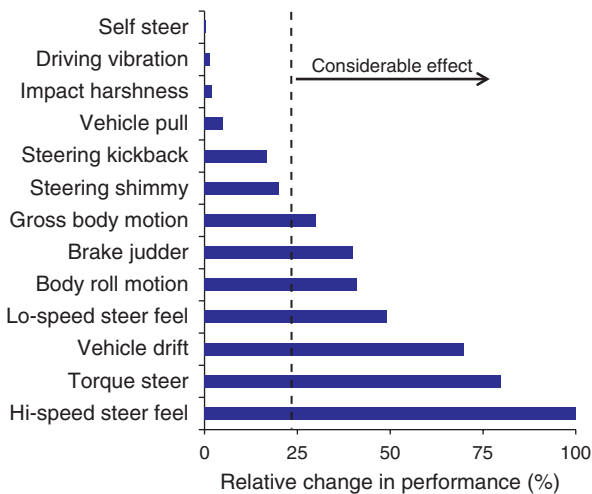


Fig. 10 Influence of chassis frictions on vehicle performance areas

## 4. Sensitivity of Performance to Frictions

### 4.1. Most Influential Frictions

A virtual sensitivity analysis was conducted to identify which frictions have more influence. During the sensitivity analysis, the vehicle model with half of production specification levels of chassis frictions was used as a baseline model. Next, one derivative model was created for each chassis friction in such a way that the breakthrough and kinetic values of that specific chassis friction in the baseline model was increased up to five times as much as the corresponding production specification, while other chassis frictions remained unchanged. Because there are eight chassis frictions included in the baseline model, there are eight derivative models in total. The baseline model and the derivative models were used to predict individual performance areas; the relative sensitivity of each performance area to the changes of chassis frictions was analyzed, and most influential frictions for each performance area were determined.

Figure 11 shows that for torque steer, steer-arm ball joint friction torque, steering gearbox friction force, and lower-arm ball joint friction torque have high influence; strut bearing friction torque has some influence; while front damper friction force, front damper friction torque, rear damper friction force, and rear upper-arm ball joint friction torque have no influence. Conversely, for gross body motion, Figure 12 suggests that front damper friction force has high influence; lower-arm ball joint friction torque and steer-arm ball joint friction torque have some influence; rear damper friction force has low influence; and strut bearing friction torque, front damper friction torque, steering gearbox friction force, and rear upper-arm ball joint friction torque have no influence.

Similarly, Figs. 13 and 14 show that with respect to low-speed and high-speed steering feel, steer-arm ball joint friction torque and steering gearbox friction force have high or some influence, lower-arm ball joint friction torque and strut bearing friction torque have low influence, and all other frictions have no influence.

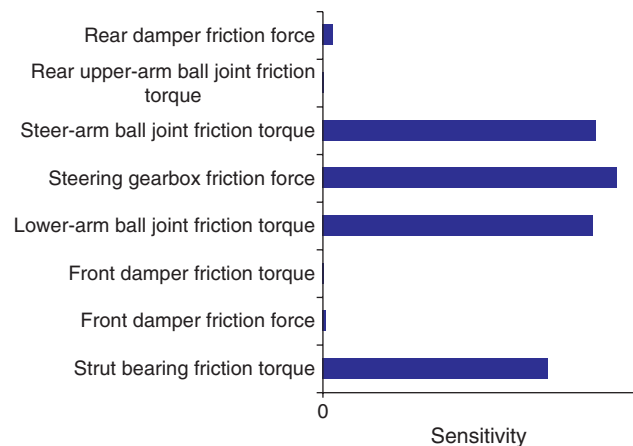


Fig. 11 Sensitivity of torque steer to chassis frictions

#### 4.2. Common Influential Frictions

Table 2 summarizes the findings of the high-influence chassis frictions for each performance area. As shown, steer-arm ball joint friction torque and lower-arm ball joint friction torque are the only chassis frictions that affect all

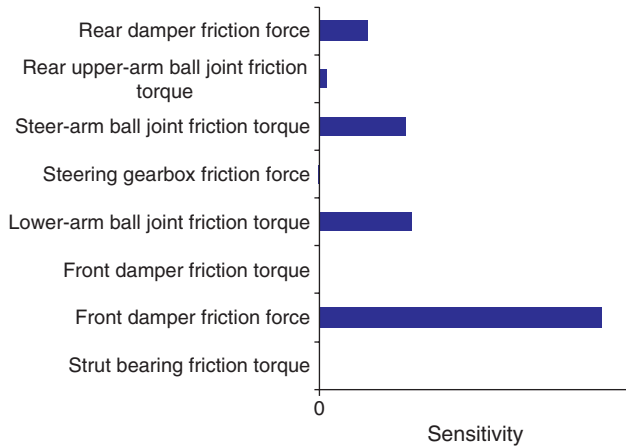


Fig. 12 Sensitivity of gross body motion to chassis frictions

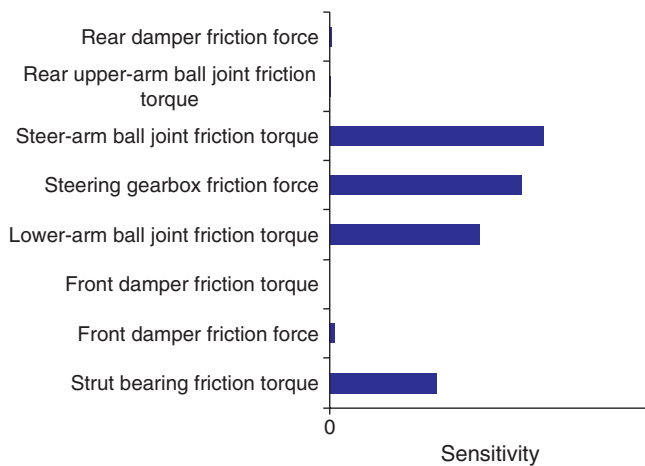


Fig. 13 Sensitivity of low-speed steering feel to chassis frictions

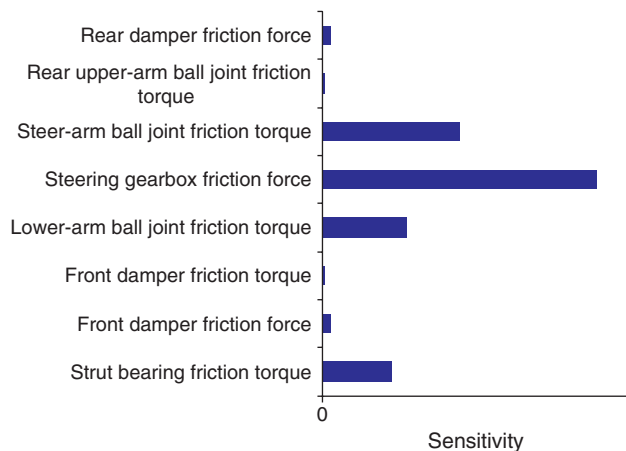


Fig. 14 Sensitivity of high-speed steering feel to chassis frictions

Table 2 Summary of the sensitivity of vehicle performance to chassis frictions

⊙ Most influential ○ Influential △ Less influential × Not influential

| Friction \ Performance | Strut bearing friction torque | Front damper friction force | Front damper friction torque | Lower-arm ball joint friction torque | Steer-arm ball joint friction torque | Steering gearbox friction force | Rear upper-arm ball joint friction torque | Rear damper friction force |
|------------------------|-------------------------------|-----------------------------|------------------------------|--------------------------------------|--------------------------------------|---------------------------------|-------------------------------------------|----------------------------|
| Hi-speed steer feel    | △                             | ×                           | ×                            | △                                    | ○                                    | ⊙                               | ×                                         | ×                          |
| Torque steer           | ○                             | ×                           | ×                            | ⊙                                    | ⊙                                    | ⊙                               | ×                                         | ×                          |
| Vehicle drift          | ⊙                             | ×                           | ×                            | ⊙                                    | ○                                    | ○                               | ×                                         | ×                          |
| Lo-speed steer feel    | △                             | ×                           | ×                            | ○                                    | ⊙                                    | ⊙                               | ×                                         | ×                          |
| Body roll motion       | ○                             | ○                           | ×                            | ○                                    | ○                                    | ⊙                               | ×                                         | △                          |
| Brake judder           | ⊙                             | ×                           | ×                            | ○                                    | ○                                    | ⊙                               | ×                                         | ×                          |
| Gross body motion      | ×                             | ⊙                           | ×                            | ○                                    | ○                                    | ×                               | ×                                         | △                          |

seven performance areas, while steering gearbox friction force is influential on most, but not all, performance areas. The data also imply that steer-arm ball joint friction torque, lower-arm ball joint friction torque, and steering gearbox friction force are principal chassis friction elements that need close attention during vehicle development.

## 5. Conclusion

An extensive analytical study was conducted to quantify the influence of chassis frictions on vehicle dynamic performance areas and to identify the relative importance of frictions in different locations in a chassis system. A multi-body simulation model of a full vehicle was developed with the integration of customized friction functions in all major chassis joints and the correlation with measurement in chassis kinematics, compliance, system-level frictions, and vehicle dynamic behavior. Using vehicle simulation and actual measurement, it was determined that chassis frictions can affect seven key vehicle performance areas, which include high-speed steering feel, torque steer, vehicle drift, low-speed steering feel, body roll motion, brake judder, and gross body motion. This was followed by a thorough sensitivity analysis to identify the high-influence frictions for each of those performance areas. As a result, steer-arm ball joint friction torque, lower-arm ball joint friction torque, and steering gearbox friction force were identified as the common influential chassis frictions that exert considerable effect on almost all of the seven vehicle performance areas.

The study provides valuable information for understanding the influence of chassis frictions on vehicle dynamic behavior and for developing an effective friction control strategy for vehicle development. Further experimental investigation is conducted<sup>(7)</sup> to understand how each of the seven key performance areas can be affected by the different friction levels at a selected location.

## References

- (1) Misaji, K., Tokunaga, H., Takimoto, S., Shimizu, Y., Shibata, K.: Vehicle Dynamics Evaluation by Analytical Method of Equivalent Linear System using the Restoring Force Model of Power Function Type, Proceedings of 6<sup>th</sup> International Symposium on Advanced Vehicle Control, AVEC'02, Hiroshima, Japan (2002)
- (2) Tokunaga, H., Misaji, K., Takimoto, S., Shimizu, Y., Shibata, K.: Steer Feel Evaluation Method Based on 'Analytical Method of Equivalent Linear System using the Restoring Force Model of Power Function Type', Proceedings of 6<sup>th</sup> International Symposium on Advanced Vehicle Control, AVEC'02, Hiroshima, Japan (2002)
- (3) Yu, J., Aston, J., Gilsinger, C., Shutway, J., Tokunaga, H.: Vehicle Dynamic Feeling Study with a Focus on the On-Center Steering Feeling of North American Highway Driving, Proceedings of the 7th International Symposium on Advanced Vehicle Control, AVEC'04, HAN University, Netherlands (2004)
- (4) Yu, J., Goolsbey, S.: Modeling and Simulation of Vehicles with Hydraulic Power Steering for High-Speed On-Center Dynamic Steering Feel Study, Proceedings of 8th International Symposium on Advanced Vehicle Control, AVEC'06, Taipei, Taiwan (2006)
- (5) Yu, J., Nutwell, B., Brickner, B.: Analysis of Vehicle Chassis Transmissibility of Steering Shimmy and Brake Judder: System Modeling and Validation, SAE Technical Paper 2007-01-2341 (2007)
- (6) Yu, J., Brickner, B., Nutwell, B., Johnson, M.: Analysis of Vehicle Chassis Transmissibility of Steering Shimmy and Brake Judder: Mechanism Study and Virtual Design of Experiment, SAE Technical Paper 2007-01-2342 (2007)
- (7) Yu, J., Hosni, E.: Influence of Chassis Frictions on Vehicle Dynamic Behavior: Vehicle Experimental Investigation, Honda R&D Technical Review Vol. 23, No. 2, p.114-120

## ■ Author ■



Jinghong YU

# Influence of Chassis Frictions on Vehicle Dynamic Behavior: Vehicle Experimental Investigation

Jinghong YU\*     Edward HOSNI\*

## ABSTRACT

To study the influence of chassis frictions on vehicle dynamic behavior, an experimental investigation is done using a test vehicle with a specially-designed friction control device integrated with the steer-arm ball joints. Extensive vehicle on-road tests were conducted to understand the relationship between the chassis friction level and the following seven performance areas: torque steer, high-speed steering feel, low-speed steering feel, vehicle drift, body roll motion, gross body motion, and brake judder. The research demonstrated that vehicle performance in these areas is highly dependent on the level of friction torque at the steer-arm ball joints in a linear or nonlinear manner. Each performance area can be enhanced by an appropriate level of the friction torque, but different performance areas may require different settings of the friction torque and some settings may contradict each other.

## 1. Introduction

Frictions exist at multiple locations in an automotive chassis system, including suspensions and steering. With unique nonlinear characteristics, frictions in a chassis system may affect vehicle dynamic behavior.

Passively applying supplemental frictions at a chassis location, such as in the steering gearbox, is a common approach used to tune steering feel or to enhance system toughness for vehicle performance. However, while one friction setting can increase vehicle performance in one area, it may affect performance in other areas. To help design a chassis system that enhances various dynamic and comfort factors, research was done on the influence of chassis frictions.

Related research<sup>(1)</sup> by the first author reveals that the following seven key vehicle performance areas can be highly affected by chassis frictions: torque steer, high-speed steering feel, low-speed steering feel, vehicle drift, body roll motion, gross body motion, and brake judder. Furthermore, steer-arm ball joint friction torque and lower-arm ball joint friction torque have been identified as the frictions that exert high influence on all of these seven vehicle performance areas.

This paper explains an experimental study on the influence of chassis frictions on vehicle dynamic behavior. The study involved a test vehicle with a specially-designed on-board friction control device integrated with the steer-arm ball joints. Extensive vehicle on-road tests were conducted to understand the relationship between the friction torque level at the steer-arm ball joints and each of the seven key vehicle performance areas.

## 2. Experimental Approach

### 2.1. Test Vehicle

The test vehicle used in the investigation was a sport utility vehicle equipped with a MacPherson front suspension, trailing-arm rear suspension, and rack-and-pinion steering system. In addition, two modifications were made. First, the front lower-arm ball joints, strut bearings, and steering gearbox were replaced with parts of low-limit specifications for friction. Second, at the location of each steer-arm ball joint, a specially-designed friction control device (FCD) was installed as shown in Fig. 1. The design of the FCD consists of a ball joint with movable ball socket and a force actuator that acts on the socket. By the control of the input (in hexadecimal) of the force actuator, the

\* HONDA R&D AMERICAS, INC.

friction torque of the ball joint can be controlled. The design intention of the FCD is to have an on-board device to vary the friction level of the steer-arm ball joint as needed. With the FCD, real-time control of the friction torque can also be possible.

As demonstrated in other research<sup>(1)</sup>, the steer-arm ball joint friction torque is one common friction that exerts high influence on the seven vehicle performance areas. By changing input to the FCD, the friction torque of the steer-arm ball joints can be varied without physical part change, and the investigation on the relationship between the level of the friction torque and each of the seven vehicle performance areas can be conducted.

## 2.2. Tool Used to Vary Frictions

The FCD is designed to generate sufficient and smooth friction torques. Friction torques of the FCD can occur in different directions, where the rotational direction is around the axial line of the ball stud and the rocking direction is the direction in which the ball stud swings back and forth relative to the ball socket. Kinetic (or dynamic) friction occurs when the ball stud is rotating or rocking relative to the ball socket, while breakthrough friction is the maximum friction between the ball stud and the ball socket when they are not moving relative to each other (i.e. zero relative velocity). Figure 2 shows an example of the measured FCD friction torques under a given input. As shown, the FCD can create smooth and linear friction torques with little increase from kinetic to breakthrough frictions in both rotational and rocking directions.

Figure 3 shows the measured kinetic FCD friction torques with different device input. It was confirmed that friction torques in both rotational and rocking directions increase linearly with the increase of the device input, the ratio between the friction torques in rotational and rocking directions remains nearly constant regardless of input value,

and the variation range of friction torque can achieve 70 Nm maximum in rotational direction and 50 Nm maximum in rocking direction.

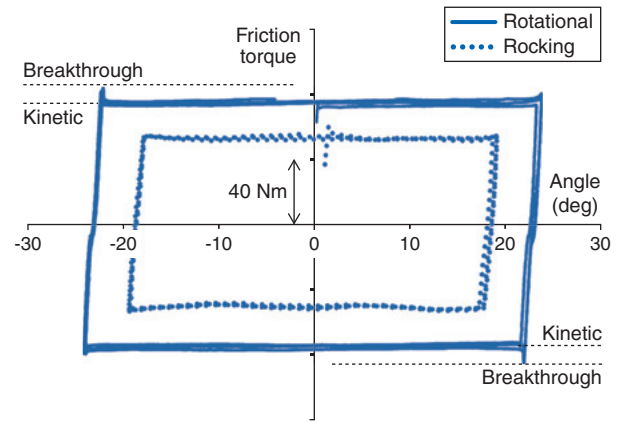


Fig. 2 Rotational and rocking friction torques of FCD at one given input

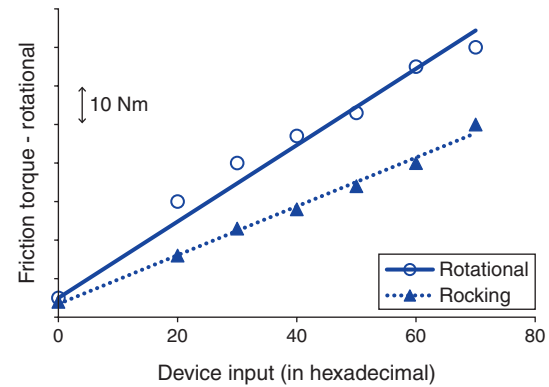


Fig. 3 Friction torque characteristics of FCD with respect to its input

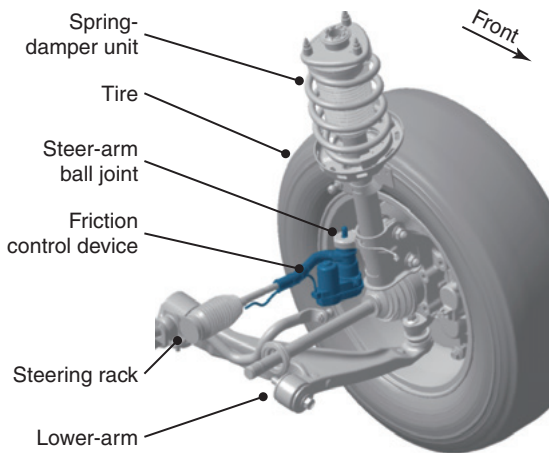


Fig. 1 Friction control device integrated with steer-arm ball joint

## 3. Performance Dependency on Friction Level

### 3.1. Torque Steer

Torque steer occurs normally during hard acceleration from zero speed on split- $\mu$  surface, which can be quantified by the steering angle change during the hard acceleration event when the driver's hands are free from the steering wheel. Torque steer performance of the test vehicle is investigated against a series of FCD friction torque settings.

Figure 4 illustrates the variation of maximum steering wheel angle with the change of the friction torque during hands-free, hard acceleration events under a given unbalanced vehicle loading condition. As shown, the FCD friction torque has high influence on torque steer

performance. Steering wheel rotation and vehicle lateral drift caused by torque steer can be improved by about 87% to an unnoticeable level when the friction torque is increased from the lowest setting to the highest setting.

The data also demonstrate clearly a nonlinear relationship between the torque steer performance and the FCD friction torque. Maximum steering angle change decreases steadily with the initial increase of the friction torque, but beyond a critical level of friction torque, further increase of the friction torque does not further enhance torque steer performance significantly. This implies that the critical level of friction torque can be considered as the sufficient setting for enhancing torque steer performance.

### 3.2. Steering Feel

Steering feel is evaluated with various FCD friction torques at different vehicle speeds so that the dependency of steering feel on the friction torque can be understood thoroughly. Steering feel for mid to high speeds is focused on an on-center region (small steering input) and characterized by a steering damping factor and a spring coefficient<sup>(2)-(5)</sup>, while steering feel for low speeds is quantified by the self-centering ability of the steering wheel when the steering wheel is released during a steady-state turning event.

Figure 5 shows on-center steering feel characteristics at a middle speed and a high speed under different FCD friction settings. It is shown that at the middle vehicle speed, the increase of the FCD friction torque from 5 Nm to 30 Nm can increase the steering damping factor while maintaining the level of the spring coefficient for both smaller sinusoid steering input (15 deg) and a relatively larger steering input (30 deg). However, further increasing the friction torque does not considerably increase the damping factor; it actually reduces the effect and raises the spring coefficient for smaller steering inputs. The results indicate that if the goal is to achieve maximum impact on the damping factor for both smaller and larger steering

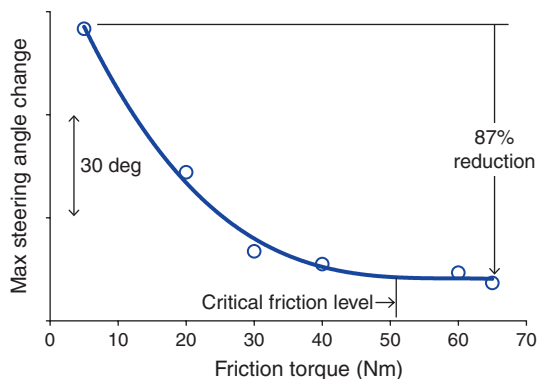


Fig. 4 Variation of maximum steering wheel rotation angle with change of FCD friction setting during torque steer tests

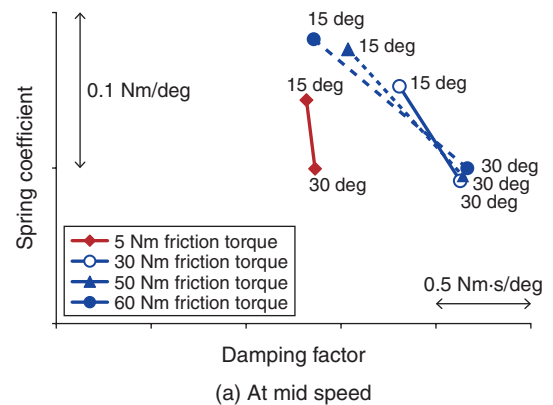
inputs, there exists an optimal friction level for the vehicle speed being discussed.

As shown in Fig. 5(b), the experimental results for high speeds reveal an interesting trend. With a lower level of friction torque, the steering damping factor not only is lower but also has a decreasing trend with the increase of steering input amplitude. This is an undesired situation that causes an insecure steering feel. As evidenced by the figure, this situation can be completely reversed by applying a higher friction torque setting.

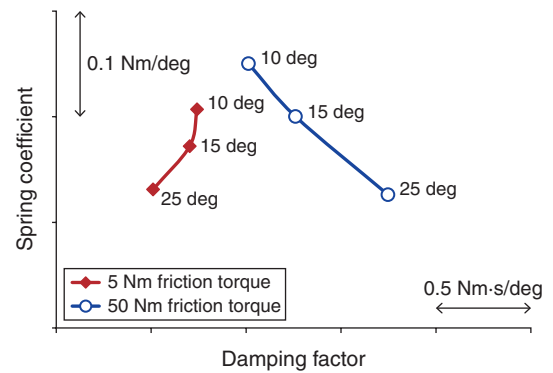
On the other hand, Fig. 6 shows that increasing the FCD friction torque quickly decreases the steering wheel's self-centering ability. Since a high level of steering wheel self-centering ability is desired for achieving a nimble feel, lower FCD friction torque is necessary at lower speeds. This suggests that the vehicle needs a completely different friction setting for enhancing low-speed steering feel from the one required for achieving a good high-speed steering feel.

### 3.3. Vehicle Drift

Vehicle drift is normally experienced during mid-to high-speed straight-line driving. In order to obtain



(a) At mid speed



(b) At high speed

Fig. 5 Variation of on-center steering feel characteristics with changes in FCD friction torque (at mid to high speeds)

repeatable results, the test vehicle was set with split camber angles and unbalanced tire conicity forces between left and right tires. With the driver's hands free from the steering wheel, the vehicle was driven at a constant speed along a straight line on a flat surface. Vehicle drift performance is determined by the lateral displacement of the vehicle within a predefined distance.

As shown in Fig. 7, vehicle lateral displacement during drift tests decreases almost proportionally with the increase of the FCD friction torque, and the lateral displacement can be improved by at least 60% from the lowest to the highest friction torque settings applied in the investigation. The results show that the higher the FCD friction torque, the better the vehicle drift performance.

### 3.4. Body Roll Motion Control

Body roll motion is evaluated by the combined effect of vehicle roll angle, roll rate, and yaw rate when there is an abrupt change in steering angular input (i.e., the steering wheel is released after a steady-state steering input). Since yaw and roll are coupled during a vehicle roll event, the

feeling of body roll motion control is related to both yaw and roll response.

Figure 8 shows the differences in the body roll motion between a lower setting and a higher setting of FCD friction torque. As shown, increasing FCD friction torque lowers the second overshoots in roll angle, roll rate, and yaw rate and reduces maximum roll rate and maximum yaw rate. The trends closely match analytical results<sup>(1)</sup>.

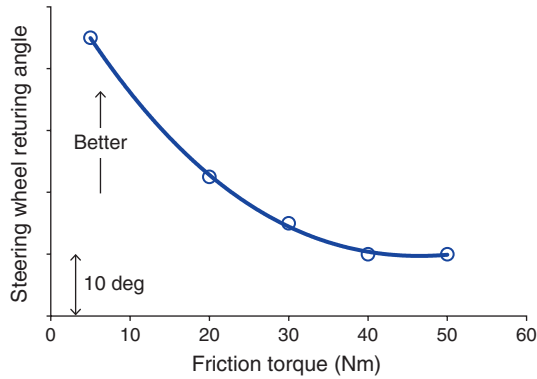


Fig. 6 Variation of steering wheel self-centering characteristics with changes in FCD friction torque (at low vehicle speeds)

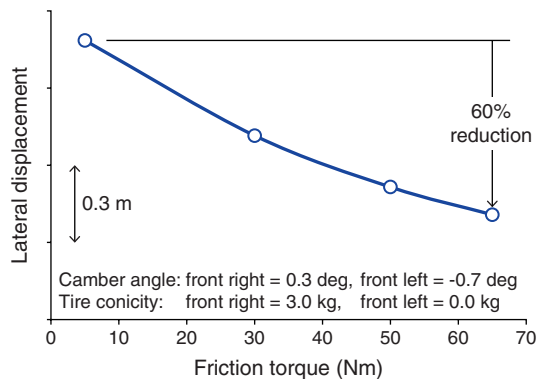
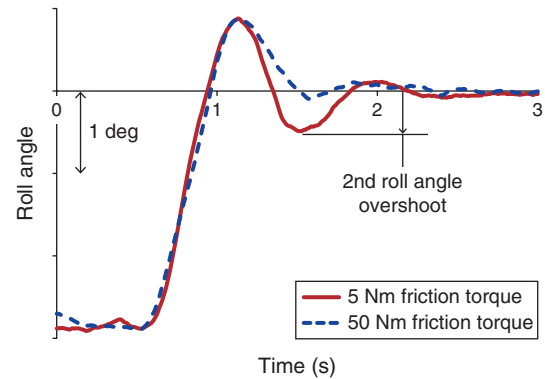
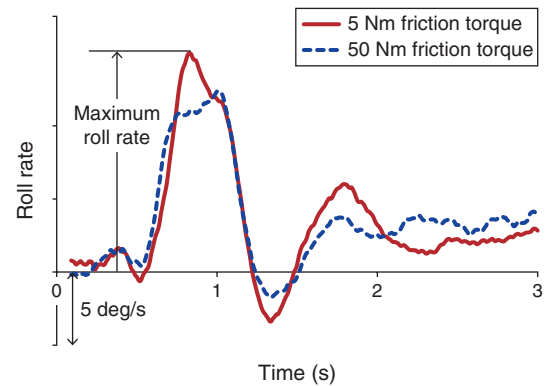


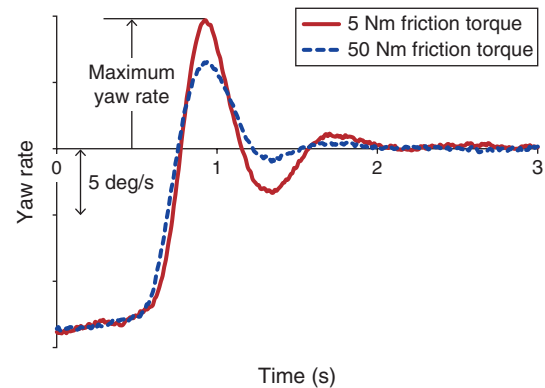
Fig. 7 Variation of vehicle lateral drift with changes in FCD friction torque



(a) Roll angle



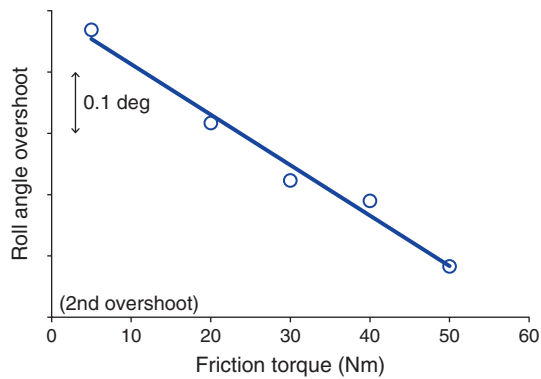
(b) Roll rate



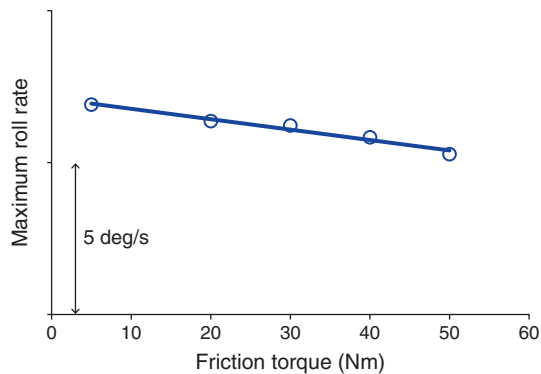
(c) Yaw rate

Fig. 8 Comparison of vehicle roll behaviors between lower and higher settings of FCD friction torque during abrupt change in steering input

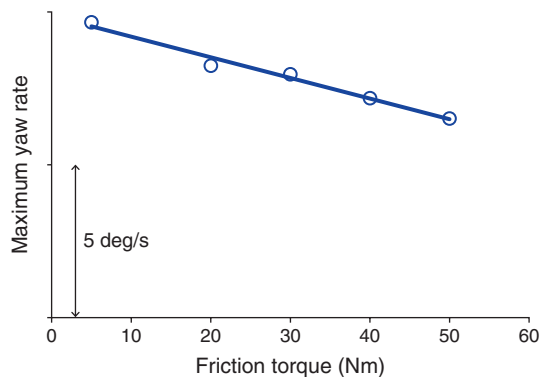
Body roll motion control was evaluated under several FCD friction torque settings. Figure 9 shows the variation of body roll characteristics with changes in friction torque. As shown, the second roll angle overshoot, maximum roll rate, and maximum yaw rate all demonstrate a purely linear relationship with the FCD friction torque. The friction torque has considerable effect on the roll angle and yaw rate and a mild effect on roll rate. The higher the friction torque, the lower the roll angle overshoot, maximum roll rate, and maximum yaw rate. Thus, a high FCD friction torque can create a more-damped and stable roll motion.



(a) Second overshoot of roll angle



(b) Maximum roll rate



(c) Maximum yaw rate

Fig. 9 Variation of body roll characteristics with changes in FCD friction torque

### 3.5. Gross Body Motion Control

Gross body motion was examined by the vehicle body motion in a vertical direction when the vehicle is tested both on a laboratory four-post tester and on a flat ride road.

Figure 10 shows the sprung mass (body) response under a larger excitation amplitude and the front unsprung mass (front suspension) response under a smaller excitation amplitude at the tire contact patches, when the FCD friction torque is set to different levels. As shown, with the increase of the FCD friction torque, the peak gain of the frequency response function (FRF) between the sprung mass and the pad of the four-post tester demonstrates a nonlinear decrease trend, while the peak gain of the FRF between the unsprung mass and the pad of the four-post tester exhibits a decrease trend initially and then a subtle increase trend.

Figure 11 shows the nonlinear dependency of seat-rail vibration, body vertical displacement, and steering effort fluctuation on the FCD friction torque when the vehicle is driven on a flat ride road with a sinusoid profile in vertical direction. Similar to the trend of the sprung mass FRF shown in Fig. 10, the amplitudes of seat-rail vibration, body vertical displacement, and steering effort fluctuation decrease slightly with initial increase of the friction torque but then decrease with further increase of the friction torque. For best ride comfort under the flat ride road condition, the highest possible FCD friction torque is needed. In addition, the characteristics of steering effort fluctuation shown in Fig. 11(c) suggest that increasing the FCD friction torque can reduce the busyness of the steering wheel and further enhance the driver's comfort.

### 3.6. Chassis Sensitivity of Brake Judder

Chassis sensitivity of brake judder is quantified as the steering wheel rotational vibration when the front knuckles are excited by a given level of brake torque fluctuation<sup>(6), (7)</sup>. A brake rotor with known, exaggerated rotor thickness variation is installed in the front right brake system of the

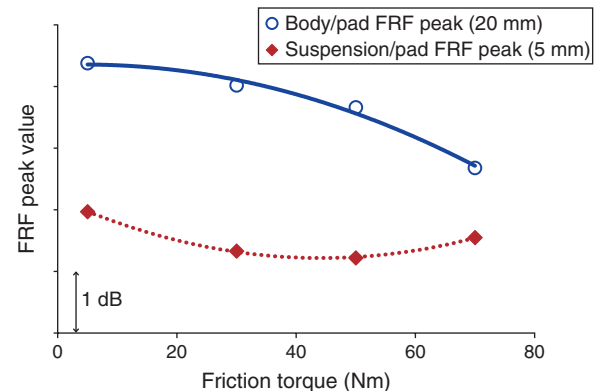
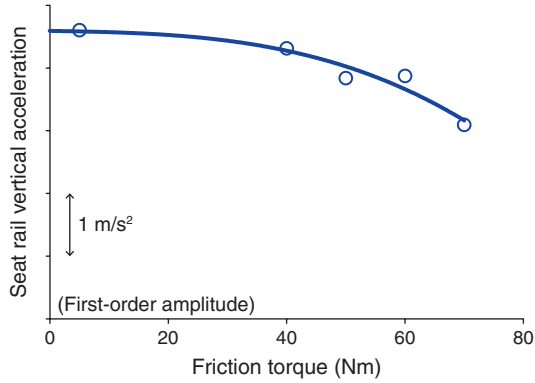


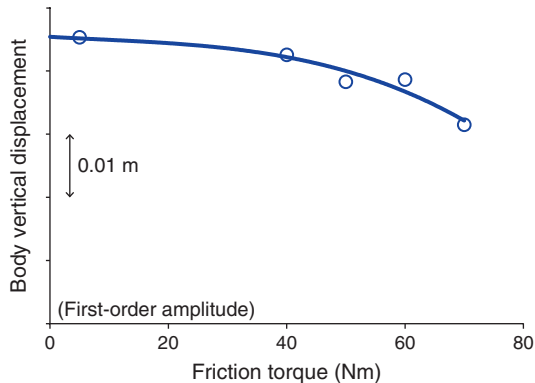
Fig. 10 Variation of FRF peak value with change of FCD friction torque during laboratory four-post excitation tests

test vehicle. During the brake judder tests, the vehicle is decelerated from 140 kph at a deceleration of  $2 \text{ m/s}^2$ .

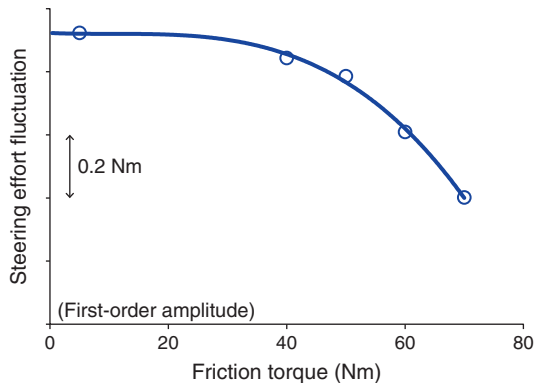
Figure 12 shows examples of brake judder test results when different FCD friction torques are applied. As shown, the FCD friction torque has considerable effect on the peak amplitude of steering wheel vibration during deceleration. Brake judder can be reduced linearly by increasing FCD friction torque and be improved by as much as  $6 \text{ m/s}^2$  within the limits of the FCD.



(a) Seat rail vertical vibration

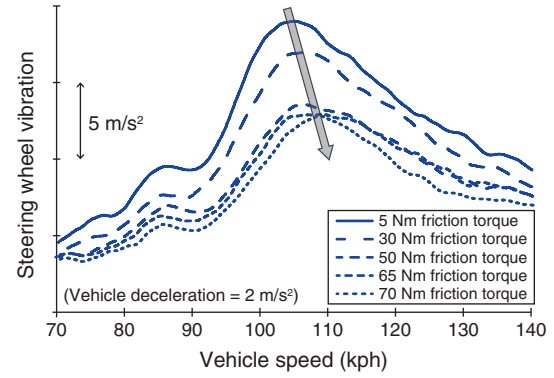


(b) Body vertical motion

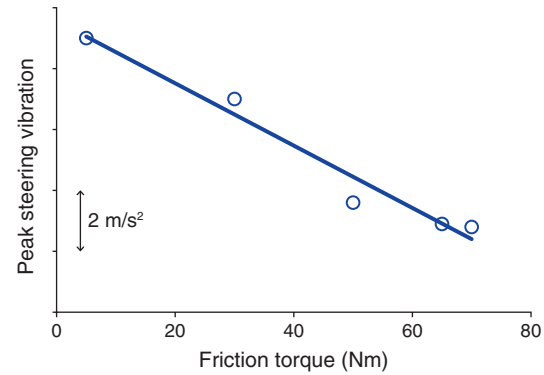


(c) Steering effort fluctuation

Fig. 11 Variation of flat ride characteristics with changes in FCD friction torque



(a) Steering wheel rotational vibration with respect to vehicle speed



(b) Peak value of steering wheel vibration

Fig. 12 Variation of brake judder sensitivity with changes in FCD friction torque

## 4. Conclusion

Building on other research into the influence of chassis frictions on vehicle dynamic behavior<sup>(1)</sup>, an experimental investigation was done using a test vehicle with a specially-designed friction control device integrated with the steer-arm ball joints. Extensive vehicle on-road tests were conducted to understand the relationship between the friction torque level at the steer-arm ball joints and the following seven key performance areas: torque steer, high-speed steering feel, low-speed steering feel, vehicle drift, body roll motion, gross body motion, and brake judder.

It was determined that these vehicle performance areas are considerably affected by the level of friction torque at the steer-arm ball joints in either a linear or nonlinear manner. Each performance area can be enhanced by an appropriate level of friction torque, but different performance areas may require different settings of friction torque and some settings may contradict each other.

This study extends the understanding of chassis frictions' influence on vehicle dynamic behavior. Further investigation is planned to take the advantage of the frictions' effects to simultaneously enhance multiple performance areas.

## References

- (1) Yu, J.: Influence of Chassis Frictions on Vehicle Dynamic Behavior: Analytical Study, Honda R&D Technical Review Vol. 23, No. 2, p.106-113
- (2) Tokunaga, H., Misaji, K., Takimoto, S., Shimizu, Y., Shibata, K.: Steer Feel Evaluation Method Based on 'Analytical Method of Equivalent Linear System using the Restoring Force Model of Power Function Type', Proceedings of 6<sup>th</sup> International Symposium on Advanced Vehicle Control, AVEC'02, Hiroshima, Japan (2002)
- (3) Misaji, K., Tokunaga, H., Takimoto, S., Shimizu, Y., Shibata, K.: Vehicle Dynamics Evaluation by Analytical Method of Equivalent Linear System using the Restoring Force Model of Power Function Type, Proceedings of 6<sup>th</sup> International Symposium on Advanced Vehicle Control, AVEC'02, Hiroshima, Japan (2002)
- (4) Yu, J., Goolsbey, S.: Modeling and Simulation of Vehicles with Hydraulic Power Steering for High-Speed On-Center Dynamic Steering Feel Study, Proceedings of 8<sup>th</sup> International Symposium on Advanced Vehicle Control, AVEC'06, Taipei, Taiwan (2006)
- (5) Yu, J., Aston, J., Gilsinger, C., Shutway, J., Tokunaga, H.: Vehicle Dynamic Feeling Study with a Focus on the On-Center Steering Feeling of North American Highway Driving, Proceedings of 7<sup>th</sup> International Symposium on Advanced Vehicle Control, AVEC'04, HAN University, Netherlands (2004)
- (6) Yu, J., Nutwell, B., Brickner, B.: Analysis of Vehicle Chassis Transmissibility of Steering Shimmy and Brake Judder: System Modeling and Validation, SAE Technical Paper 2007-01-2341 (2007)
- (7) Yu, J., Brickner, B., Nutwell, B., Johnson, M.: Analysis of Vehicle Chassis Transmissibility of Steering Shimmy and Brake Judder: Mechanism Study and Virtual Design of Experiment, SAE Technical Paper 2007-01-2342 (2007)

### ■ Author ■



Jinghong YU



Edward HOSNI

# Integrated Active Noise Control System for Low-frequency Noise in Automobiles

Toshio INOUE\*

Kosuke SAKAMOTO\*

Akira TAKAHASHI\*

Yasunori KOBAYASHI\*

## ABSTRACT

Honda has developed two Active Noise Control (ANC) technologies, and has employed them as standard in a large number of vehicles. One of these technologies is a drumming reduction technology that employs feedback control, and the other is an engine booming noise control technology that employs feed forward control, which uses the engine speed signal as a reference signal. These technologies have previously not been combined to produce an integrated system due to issues of mutual interference between the control systems and stability in the feedback control. The research discussed in this paper created a new drumming control technology that resolved these issues, and developed an integrated ANC system. The new ANC system developed in the research has been employed in a number of vehicles.

## 1. Introduction

Today, awareness of the need to protect the global environment is intensifying demands for increased automotive environmental performance and fuel efficiency. Together with technologies to achieve weight savings, numerous technologies to increase fuel efficiency are being applied in automotive powertrains. At the same time, demands are also intensifying for increased noise and vibration performance, which represents a factor in the commercial competitiveness of the vehicle. However, because measures to combat noise and vibration, in particular low-frequency phenomena, entail increased weight, in many cases they are in an antagonistic relationship with the aforementioned demand for increased fuel efficiency. These measures are antagonistic because many of the technologies used to increase fuel efficiency use the low-engine-speed range with a greater frequency than is normally the case. Because of this, these technologies often increase intake and exhaust noise and engine vibration in the low-frequency range. In many cases, the passive technologies that have traditionally been adopted to counteract this increase in noise and vibration, such as increasing the noise-attenuating volume of the intake and exhaust or attaching stiff members or damping members to the vehicle body, and measures such as the use of dynamic dampers to reduce vibration in specific frequency ranges, result in increased vehicle weight. This necessitates the development of technologies to counteract noise and vibration in the low-frequency range that entail minimal

weight increase. Honda worked towards the development of two types of ANC technology that would balance fuel efficiency with noise and vibration performance. The developed technologies have been employed as standard in a large number of Honda vehicles. One of these technologies reduces drumming<sup>(1)</sup>, a type of low-frequency road noise; the other reduces booming noise in the engine<sup>(2)-(4)</sup>. Figure 1 shows the in-cabin sound spectrum for a vehicle driven over an undulating road at a fixed speed of 50 km/h. Drumming and booming noise are produced simultaneously in the cabin. This situation demands the development of a control technology able to control these two phenomena simultaneously. In addition, the integration of the control systems would be desirable in order to minimize system costs. However, when ANC systems are

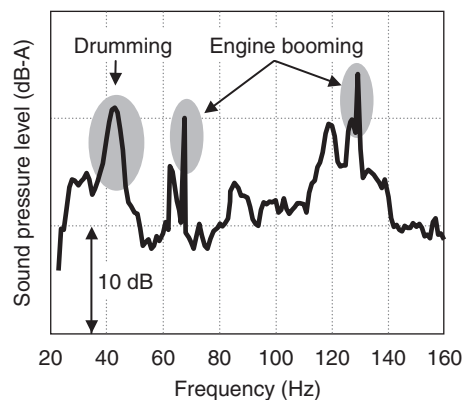


Fig. 1 Frequency characteristic for drumming and engine booming noise

\* Automobile R&D Center

integrated, differences in their methods of control result in interference between the control devices and stability issues, and it has therefore not been possible to create an integrated system to date. This paper presents an overview of a new ANC technology for drumming and an integrated ANC technology produced by fusing this technology with a booming noise control technology, developed to resolve these technological issues.

## 2. Overview of Existing Technologies

### 2.1. ANC for Drumming

#### 2.1.1. Characteristics of drumming

Figure 2 shows an acoustic eigenmode obtained by eigenvalue analysis of an in-cabin sound field modeled using the finite element method. When the vehicle is being driven, input from the road surface is transmitted to the body via the suspension, and drumming is produced by the excitation of the eigenmode. This phenomenon occurs at a frequency at which the length of the cabin is one-half the length of the sound waves, which is approximately 40 Hz in passenger vehicles. Figure 3 shows a sound pressure spectrum near the front seats when the vehicle is driving on a rough road surface. As this figure shows, drumming is a phenomenon that occurs in a narrow frequency band, occurring in a frequency width of around 8 to 10 Hz.

#### 2.1.2. Existing drumming ANC system

Figure 4 shows the configuration of the drumming ANC system. As the diagram shows, the control unit (termed “ECU” below) is positioned underneath the driver’s seat.

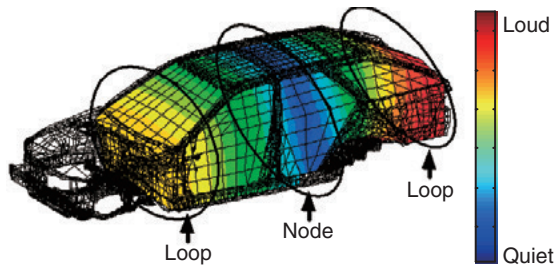


Fig. 2 Acoustic modes of station wagon

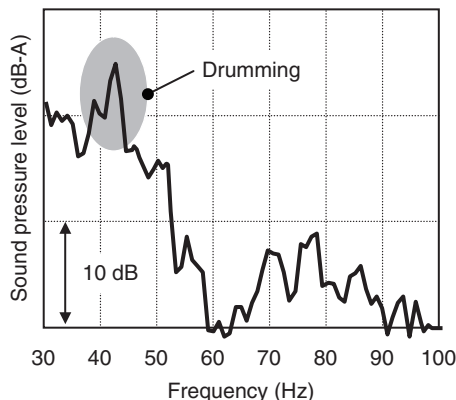


Fig. 3 Frequency characteristic for drumming

The system employs a feedback control method in which a control signal is output based on the sound in the cabin. The ECU is made up of a drumming control filter that comprises a number of analogue circuits, a speaker-drive amplifier, and a microphone. The in-cabin sound signal detected by the microphone is converted to a control signal to cancel out the drumming by passing through the drumming filter. This signal is amplified by the amplifier, and emitted into the cabin via the speakers.

#### 2.1.3. Control technology of existing drumming ANC

Figure 5 shows a block diagram of feedback control (termed “FB control” below) for drumming control, using a microphone signal and control filter, and speakers that are shared with the audio system. The FB-ANC corresponds to external disturbance suppression control in control engineering. This is to say that the purpose of the control is, when an external disturbance (noise  $d$ ) is added to the sound field of the cabin (the plant,  $C$ ), which is the transfer function from the speakers to the microphone, to reduce this noise by means of the controller,  $G$ . If the signal remaining after control has been applied is termed  $e$ ,  $e/d$  gives  $S$ , the sensitivity function. An  $S$  value of 1 or less indicates that noise can be reduced.  $S$  is expressed by Eq. (1). In order to reduce  $S$ , the controller,  $G$ , is adjusted to make the gain of the open loop function at the control target frequency  $\omega$  [given by  $L(j\omega)=C(j\omega)G(j\omega)$ ] as high as possible and its phase  $0^\circ$ . For example, when  $|L(j\omega)|=1$  and  $\angle L(j\omega)=0^\circ$ , Eq. (1) gives an  $S$  value of 1/2, and noise can be reduced by 6 dB.

$$S = \frac{1}{1 + C \cdot G} \quad (1)$$

Because the microphone signal includes all frequency components, in addition to being adjusted to make the

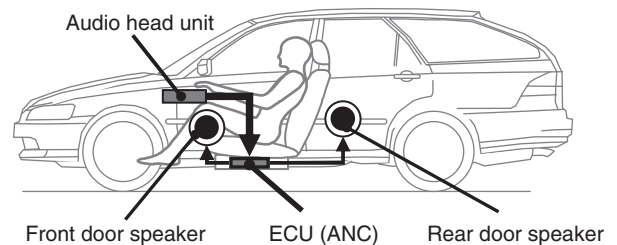


Fig. 4 On-board system configuration of drumming ANC

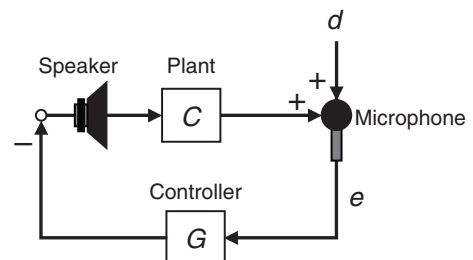


Fig. 5 Block diagram of feedback ANC

gain of the open loop function  $L$  in the control target frequency band as high as possible and its phase to  $0^\circ$ , the controller,  $G$ , is adjusted to reduce the high-frequency range, i.e., to display low pass characteristics, for the sake of stability. Figure 6 shows an example of the results of shaping the open loop function. Because the gain level is approximately 6 dB in the target frequency band (close to 40 Hz), a reduction in drumming of approximately 10 dB can be expected from Eq. (1); at the same time because the phase margin is approximately  $90^\circ$  and the gain margin is approximately 12 dB, it can be seen that stability has been achieved.

## 2.2. ANC for Engine Booming Noise

### 2.2.1. Characteristics of engine booming noise

Figure 7 shows cabin noise when the vehicle is accelerating. In this figure, the noise level is highest where the color is most intense. The area of high noise at the center of the figure shows engine booming noise. As this figure indicates, the engine booming noise component is synchronous with engine speed and occurs at high frequencies, and at the same time is a single-frequency phenomenon. That is, because booming noise has a noise waveform close to a sine wave, the canceling sound generated to reduce the noise has a waveform close to a sine wave of opposite phase.

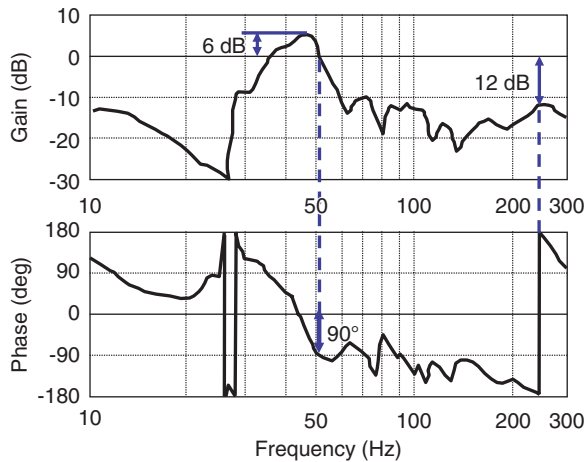


Fig. 6 Bode plot of open loop transfer function

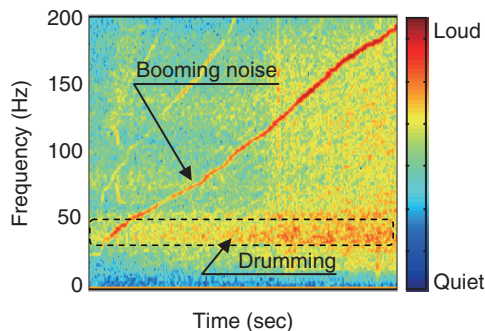


Fig. 7 Contour map of noise inside cabin

### 2.2.2. Overview of engine booming noise ANC system

Figure 8 shows the composition of the system. This system is made up of an ECU, microphone units positioned on the ceiling at the front and rear of the cabin, an audio unit fitted with an input terminal for the control sound signal from the ECU, and speakers shared with the audio system. The microphone unit positioned on the ceiling at the front of the cabin also has a voice recognition function for hands-free telephone or navigation system use. The system uses a feed-forward control method, in which the engine speed signal is used as a reference signal. In other words, the signals input to the ECU are the engine speed pulse signal (number of engine pulses: NEP) from the ENG-ECU that controls the combustion state of the engine, and the signals from the microphones that detect in-cabin noise. The output signal is a control signal that drives the speakers via the amplifier that is shared with the audio system.

### 2.2.3. Engine booming noise control technology

Because engine booming noise is a single-frequency phenomenon, it can be expressed on a complex plane, as shown in Fig. 9. As this figure indicates, using the reference signals corresponding to the real and imaginary axes (signals  $RX$  and  $RY$ ), the canceling sound signal can be produced using the amplitude of signal  $RX$ ,  $A$ , and the amplitude of signal  $RY$ ,  $B$ . This method expands single-frequency adaptive notch (SAN)<sup>(5)</sup> control, which internally generates orthogonal reference signals and adaptively operates adaptive notch filters  $A$  and  $B$  corresponding to the amplitudes of these signals, to filtered-x SAN, and is able to rapidly trace changes in engine speed as it reduces engine booming noise. In this method, the acoustic characteristic  $\hat{c}$  (impulse response) is frequency decomposed as shown in

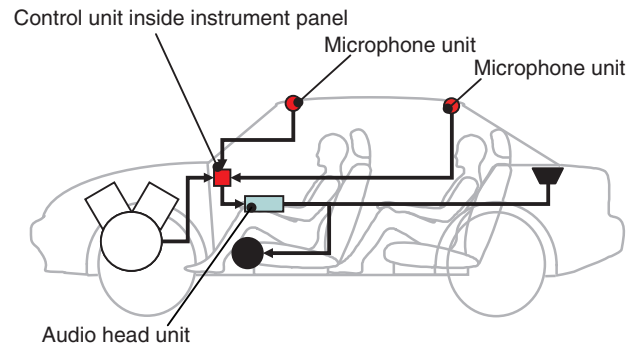


Fig. 8 On-board system configuration of engine booming ANC

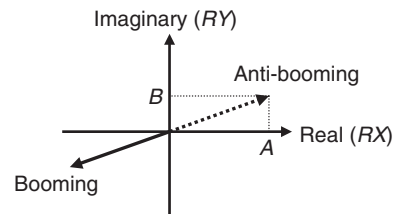


Fig. 9 Polar coordinates of booming noise

Eq. (2), and the frequencies are saved as  $\hat{C}$  on a table for the control frequencies. The values corresponding to the control frequencies are read from the table and used in calculations.  $j$  represents an imaginary unit.

$$\hat{C}(f) = C_R(f) + j \cdot C_I(f) \quad (2)$$

The calculation of the filtered-x segment of the reference signal  $RX$  can be expressed as Eq. (3).

$$\hat{C}(f) \cdot RX = C_R(f) \cdot RX + C_I(f) \cdot RY \quad (3)$$

Because the signal  $RY$  is advanced  $90^\circ$  in phase in relation to  $RX$ ,  $j \cdot RX = RY$  is used in Eq. (3). Similarly, for the reference signal  $RY$ :

$$\hat{C}(f) \cdot RY = C_R(f) \cdot RY - C_I(f) \cdot RX \quad (4)$$

Figure 10 shows a block diagram of the engine noise booming control realized using Eqs. (3) and (4). The benefit of this method is that it can be realized using scalar operations alone, with no need for convolution (vector) operations, which involve a significant volume of computation. Because of this, the method can be realized using today's low-cost microcomputers.

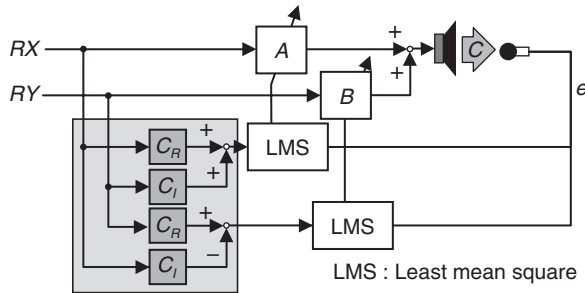


Fig. 10 Block diagram of frequency-filtered reference SAN algorithm

### 3. Integrated Control Technology for Low-frequency Noise

#### 3.1. Issues in Integrating Two Control Technologies

The engine booming noise control technology employs digital data processing using low-cost microcomputers. Given this, it is possible to apply the technology to multiple vehicle types by rewriting the control parameters in the memory, and thus production costs can be kept low by sharing the hardware. By contrast, because the drumming control technology is realized using analogue circuits, the application of the technology to multiple vehicle types necessitates redesign of the circuit configuration for each model, which represents an impediment to the sharing of hardware. The project therefore decided to digitalize the drumming control technology to make it possible to modify

it solely by adjusting parameters, and at the same time to create a new technology by integrating this technology with the engine booming noise control technology.

#### 3.1.1. Issues related to the digitalization of the drumming control technology

As indicated in the overview of existing technologies above, Honda's drumming control ANC employs feedback control. Feedback control can be expressed as a geometric series with the open loop transfer function  $L (=CG)$  a series (Fig. 5). That is, while noise can be reduced in the frequency band in which the open loop function is close to  $0^\circ$ , noise increases in the frequency band in which the open loop is close to  $180^\circ$ . Because the speakers and the microphone cannot be positioned at the same point, the cabin sound field (the target of control) will necessarily display a phase lag, and despite the fact that the open loop function is adjusted to  $0^\circ$  in the drumming frequency band, there will necessarily be a frequency band in which the open loop function becomes  $180^\circ$  in the high-frequency range. This results in the "waterbed effect" shown in Fig. 11, in which bands in which there is a noise reduction effect and bands in which noise increases repeatedly alternate due to the occurrence of frequencies other than the control frequency. The waterbed effect is minimized and control is stabilized in feedback ANC by setting a control filter, using a band pass filter to control increases in noise, and a low pass filter to reduce the high-frequency range. The following describes issues associated with realizing this control filter using digital filters.

##### (1) Issues when using a finite impulse response (FIR) filter

In order for the engine booming noise ANC to rapidly trace changes in engine speed, the system samples frequencies that are very high in relation to the engine booming noise frequencies to be controlled. When an FIR filter is used to control low-frequency waves with long cycles while a high sampling frequency is maintained, the length of the filter increases (the number of elements in the filter coefficient increases). Because the inner products of the filter coefficient vector and the input signal vector are taken for each sample during filtering using an FIR filter, if the filter becomes long, the amount of computation

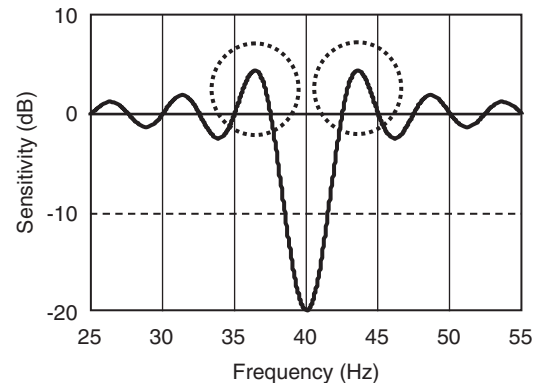


Fig. 11 Waterbed effect

increases. Given this, realization of this method using low-cost microcomputers represents a challenge.

### (2) Issues when using an infinite impulse response (IIR) filter

When an IIR filter is used to adjust low-frequency waves while a high sampling frequency is maintained, the filter coefficient becomes low. Because of this, issues of calculation accuracy make it challenging to realize this method using low-cost microcomputers, which rely on fixed-point computation.

These digital filter-related issues made it challenging to simply digitalize the standard control filter used in drumming ANC.

### 3.1.2. Issue of interference between two control devices

#### (1) Waterbed effect on engine booming noise

The frequency of engine booming noise changes synchronously with changes in engine speed. Because of this, engine booming noise increases when the engine booming noise frequency overlaps with the frequency band in which noise increases due to the waterbed effect produced by drumming ANC.

#### (2) Changes in sound field characteristics from the perspective of the engine booming noise control device

Both drumming ANC and engine booming noise ANC store the transfer characteristic from the speakers to the microphone (to be precise, the input provided by the microcomputer output) and use it for control purposes. Given this, it is essential to consider changes in the transfer characteristic from the perspective of both control devices when integrating them.

These issues of interference between the control devices made it challenging to simply integrate the two existing ANC technologies.

### 3.2. New Drumming Control Technology

This section will discuss the digitalization of the drumming control technology. As indicated in Sub-section 3.1.1, when using feedback control, it is normal to employ a control filter created by combining a low pass filter and a band pass filter in order to help ensure stability in consideration of the water bed effect. This method presents issues from the perspective of integration with engine booming noise control. The project therefore did not choose to digitalize fixed filters, but instead proposed a new drumming control method, as shown in Fig. 12, in which drumming was extracted from the microphone signal, and the resulting signal was adjusted to fixed gain and phase characteristics and output. SAN control, which is also used in engine booming noise control, is used to extract drumming from the microphone signal. Drumming and engine booming noise differ in the width of their frequency bands. Specifically, while engine booming noise can be treated as a single-frequency signal, drumming is a phenomenon that occurs within a narrow frequency band

of around 8 to 10 Hz. The project therefore considered a method of expanding the SAN control frequency band. Equation (5) expresses the output signal for the SAN control block diagram shown in Fig. 13. Here,  $\phi$  expresses the phase characteristic determined by the magnitude of the amplitudes at  $A$  and  $B$  in Fig. 13.

$$Out = \sqrt{A^2 + B^2} \cos(\omega \cdot t + \phi) \quad (5)$$

Equations (6) and (7) are the renewal equations for adaptive filters  $A$  and  $B$ .

$$A_{n+1} = A_n - \mu \cdot \cos(\omega t) \cdot e \quad (6)$$

$$B_{n+1} = B_n - \mu \cdot \sin(\omega t) \cdot e \quad (7)$$

Here,  $\mu$  expresses the step size parameter. In Eq. (5), when  $A$  and  $B$  have fixed values, the output signal,  $Out$ , is a signal depending on the reference frequency,  $\omega$ . However, when  $A$  and  $B$  are fluctuating at frequency  $\omega_a$ , the output signal can be expressed by Eq. (10).

$$A = Q \cdot \cos(\omega_a t) \quad (8)$$

$$B = Q \cdot \sin(\omega_a t) \quad (9)$$

$$Out = Q \cdot \cos\{(\omega - \omega_a) \cdot t\} \quad (10)$$

From Eq. (10), the output signal,  $Out$ , is a signal of frequency  $\omega - \omega_a$ . That is, if  $A$  and  $B$  are fluctuating at 1 Hz, a signal that is shifted by 1 Hz in relation to the reference signal,  $\omega$ , which causes the SAN to operate, can be output. When  $A$  and  $B$  rotate counterclockwise around the polar coordinates shown in Fig. 14,  $\omega_a$  takes on a positive value, and when they rotate clockwise, it takes on a negative value.  $\mu$  in Eqs. (6) and (7) is adjusted to make adaptive filters  $A$  and  $B$  fluctuate. Figure 16 shows the frequency characteristic of the output signal obtained from the signal

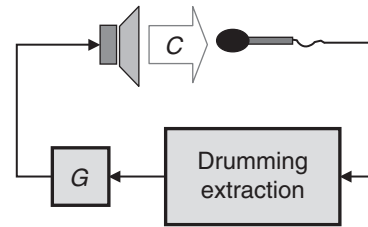


Fig. 12 Block diagram of drumming ANC

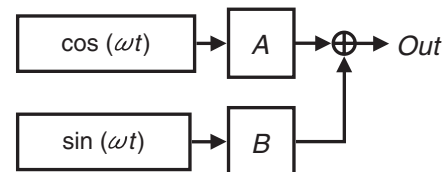


Fig. 13 Block diagram of SAN

processing simulation shown in Fig. 15. As Fig. 16 shows, the frequency width becomes narrower in proportion to the decrease in  $\mu$ , and wider in proportion to the increase in  $\mu$ . Figure 17 shows a block diagram of drumming ANC employing SAN control.  $\omega$  in the figure corresponds to the central frequency of drumming.  $\phi_d$  is the phase for control of drumming, and in the transfer characteristic from the speaker to the microphone, this corresponds to the phase that becomes  $180^\circ$  when the signal reaches the microphone.  $G_d$  is the gain for drumming control, and this is a parameter that is adjusted in response to the target degree of noise reduction. For example, following Eq. (1), a noise reduction of 6 dB can be expected if a  $G_d$  is selected to be 0 dB when multiplied by the transfer characteristic from the speakers to the microphone.

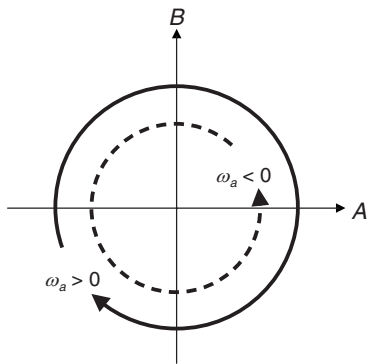


Fig. 14 Polar coordinates of gain coefficient

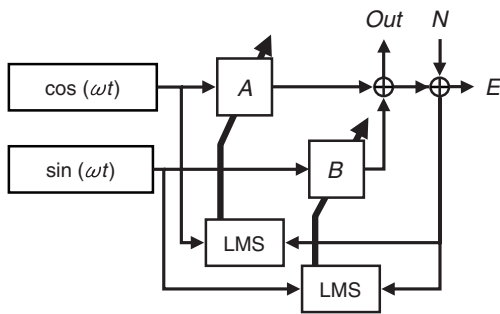


Fig. 15 Block diagram of signal extraction

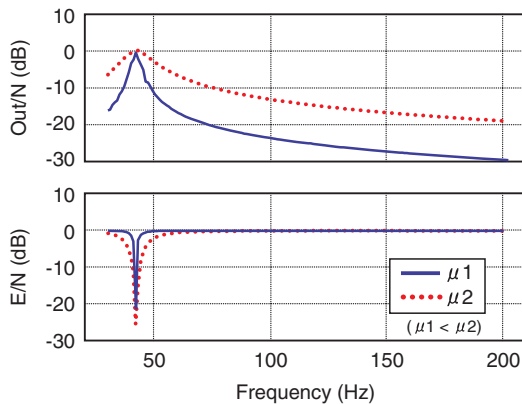


Fig. 16 Effect of signal extraction

### 3.3. Integration of Engine Booming Noise Control Technologies

#### 3.3.1. Attempting to eliminate the increase in engine booming noise

The new drumming ANC employed a narrow frequency band pass filter for which a SAN filter was used. The phase difference was  $0^\circ$  at the central frequency,  $\omega$ , but the phase varied significantly at frequencies above and below this. Because of this, the filter was affected by the waterbed effect. Engine booming noise increased when the frequency at which it occurred overlapped with the frequency band in which noise increased due to the waterbed effect. To attempt to prevent the waterbed effect, an SAN was employed to help eliminate the engine booming noise component when the signal from the microphone was processed. The effect of this measure can be expressed as  $E$  against noise  $N$  in Fig. 15, and corresponds to the graph shown at the bottom of Fig. 16.

#### 3.3.2. Setting changes in $\hat{C}$ due to drumming ANC

The engine booming noise ANC and drumming ANC systems were integrated as shown in Fig. 18. Because the transfer function between output and input from the perspective of the control algorithm is stored as a control parameter ( $\hat{C}$ ) in engine booming noise ANC, it was necessary to make settings with consideration of the effect of drumming ANC on  $\hat{C}$ .

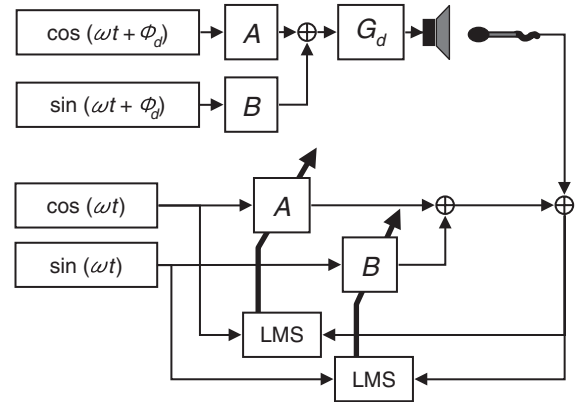


Fig. 17 Block diagram of drumming ANC using SAN

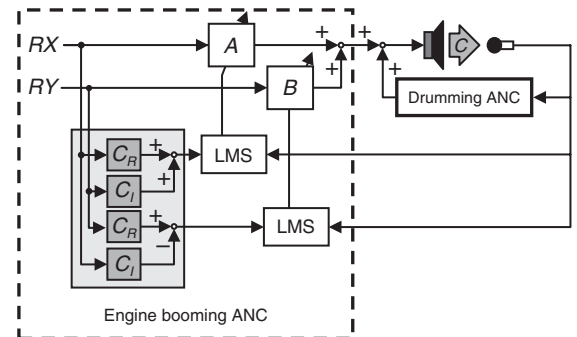


Fig. 18 Block diagram of integrated ANC

#### 4. Integrated Control System for Low-frequency Noise

Figure 19 shows the configuration of the system. The engine booming noise ANC switches between operation and non-operation in response to engine speed. In addition, the system is configured to cease operation when a door is open, because the change in the distance between the speakers and the microphone changes the value of  $\hat{C}$ . Similarly, because the drumming ANC uses feedback control, the system is configured to cease operation when the vehicle is stopped in order to increase stability from the perspective of the market. It was further necessary to link operation of the system to the state of opening of the trunk in addition to the doors. The reason for this was that the opening of the trunk would change the central frequency of drumming due to the effect of a change in the sound field characteristic. Increasing the number of control signals in order to switch operating states in this way would make it necessary to increase the number of harnesses and change the size of the ECU couplers if system communications employed the analogue signals used to date (Fig. 8). This would make it challenging to share hardware between vehicle models. It was therefore decided that in the new system, vehicle data would be transmitted to the ECU from the control area network (CAN) via digital communications. Figure 20 shows the control block diagram for the newly developed integrated ANC system. In addition

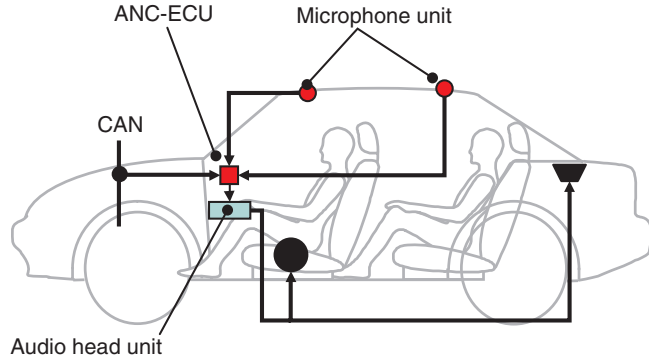


Fig. 19 On-board system configuration of integrated ANC

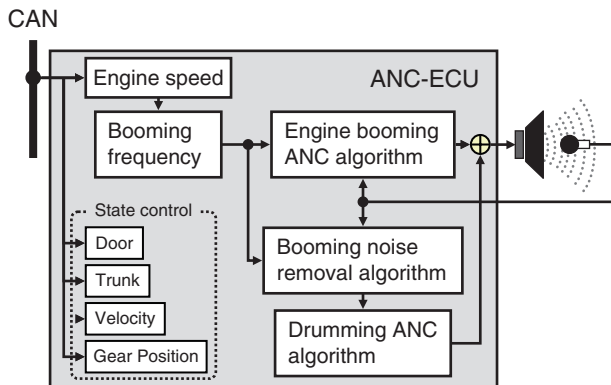


Fig. 20 Block diagram of on-board system for integrated ANC

to the control system shown here, the system also enhances the in-cabin acceleration noise control function<sup>(6)</sup> that operates when the vehicle is accelerating, using vehicle state data including powertrain state and vehicle speed transmitted by the CAN, by adjusting the function for each gear stage, and incorporates a new function that reduces, as propeller shaft rotation order noise, the engine booming noise originating in imbalances in the rear differential in AWD and FR vehicles.

#### 5. Effects of Application

##### 5.1. Control Effect of New Drumming ANC

Figures 21 and 22 show the control effect of the new drumming ANC when the vehicle is driving on rough road surfaces. Figure 21 shows the time-average frequency spectrum for several seconds. The graph indicates a reduction of approximately 6 dB. Figure 22 shows the control effect on a time series signal for in-cabin noise produced by impulse input from the road surface when the vehicle crosses seams in the road such as might be encountered, for example, on a freeway. The graph indicates that the new drumming control technology is even able to reduce the drumming component in in-cabin noise produced by impulse input, indicating the realization of adequate performance in tracing changes in engine speed.

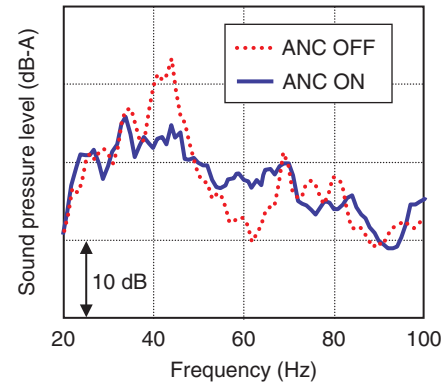


Fig. 21 Drumming reduction effect of developed drumming ANC

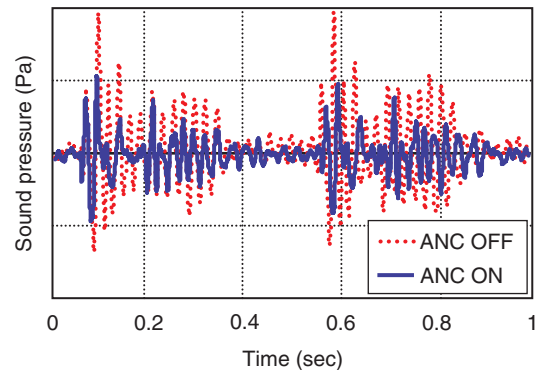


Fig. 22 Noise reduction effect of developed drumming ANC when receiving impulse input from road surface

## 5.2. Control Effect of Integrated ANC System

Figure 23 shows the control effect in relation to engine booming noise. As in the conventional system, the target order component can be reduced by around 10 dB. Figure 24 shows the control effect when the vehicle is driving on an undulating road surface at a vehicle speed of 50 km/h and an engine speed of 2300 rpm. The graph shows that the newly developed integrated ANC technology is able to simultaneously reduce drumming and engine booming noise, indicating the achievement of the desired functioning.

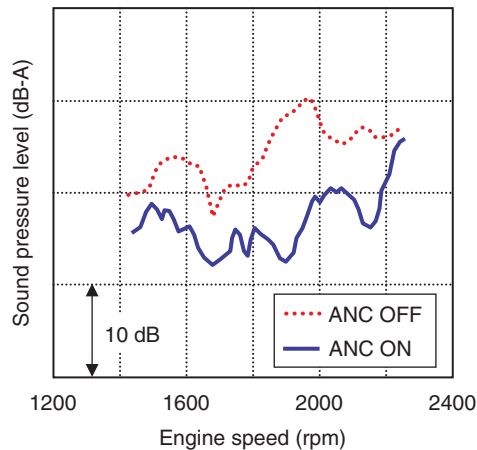


Fig. 23 Engine booming noise reduction effect of developed ANC

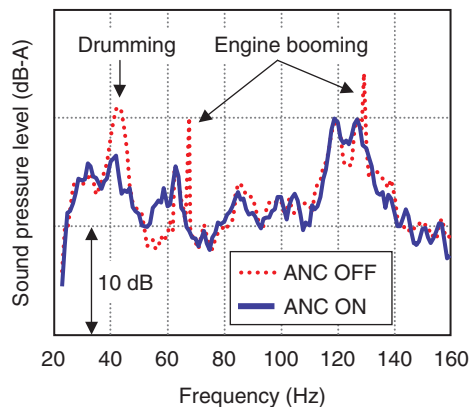


Fig. 24 Low-frequency noise reduction effect of developed integrated ANC

## 6. Conclusion

Using SAN control, which necessitates minimal calculation operations, a new drumming ANC technology employing digital feedback was developed. The use of a new technology that resolved the issue of interference between control devices in this new drumming ANC technology made it possible to integrate it with the existing engine booming noise control technology employing feed-forward control. By this means, it was possible to create an

ANC system able to simultaneously reduce drumming and engine booming noise. The new integrated ANC system discussed in this paper has been employed in new vehicle models. Because the system reduces low-frequency noise by emitting a canceling sound into the vehicle cabin rather than employing existing passive measures, it significantly boosts noise performance without increasing vehicle weight, and balances fuel efficiency with noise and vibration performance.

## References

- (1) Sano, H., Inoue, T., Takahashi, A., Yamashita, T.: Development of Active Control System for Low-frequency Road Noise Combined with Audio System, Honda R&D Technical Review, Vol. 12, No. 2, p. 71-80
- (2) Sano, H., Inoue, T., Takahashi, A., Terai, K., Nakamura, Y.: Active control system for low-frequency road noise combined with an audio system, IEEE Trans. on Speech and Audio Processing, 9 (7), 755-763 (2001)
- (3) Inoue, T., Takahashi, A., Minowa, S., Sano, H.: Development of Active Noise Control System for Engine Booming Noise, Honda R&D Technical Review, Vol. 15, No. 2, p. 201-208
- (4) Inoue, T., Takahashi, A., Sano, H.: NV countermeasure technology for a cylinder-on-demand engine – Development of active booming noise control system applying adaptive notch filter, Soc. of Automotive Engineers Technical Paper, 04ANNUAL-976 (2004)
- (5) Widrow, B., Stearns, S. D.: Adaptive Signal Processing, Prentice-Hall, Englewood Cliffs, New Jersey (1985)
- (6) Kobayashi, Y., Inoue, T., Takahashi, A., Active sound control in automobiles, Proc. INTER-NOISE 2008, in08\_0791 (2008)

## ■ Author ■



Toshio INOUE



Akira TAKAHASHI



Kosuke SAKAMOTO



Yasunori KOBAYASHI

# Development of SRS Module Damping Structure for Vibration Enhancement and Weight Reduction

Takashi KONDO\*

Masashi TERADA\*

Yukihiro KUNITAKE\*

Kunitomo MIYAHARA\*

---

## ABSTRACT

Resonance phenomena represent an issue in terms of marketability, and dynamic dampers are employed as one means of responding to this issue. However, because the use of dynamic dampers increases vehicle weight, methods that entail less weight increase are demanded in the area of the steering system. A new SRS module structure able to be employed as a dynamic damper was therefore developed. This new structure reduces vibration without the use of a conventional dynamic damper. The distinctive feature of the new structure is the addition of a floating bush to the SRS module in order to obtain a frequency characteristic suited to the resonance characteristic of the steering system. The use of this structure has balanced increased marketability with reduced weight, and helped to enable the realization of a method of ameliorating steering vibration with a high level of general applicability.

---

## 1. Introduction

Honda's desire to reduce fuel consumption and CO<sub>2</sub> emissions generates a demand for the realization of further automotive weight-savings. At the same time, the demand for vibration performance as an aspect of marketability continues to increase.

During the development of steering systems, studies are conducted from the initial stages in an attempt to reconcile weight-savings and performance. However, the realization of vibration performance for vehicle marketability by means of design specs that focus exclusively on stiffness represents a challenge, and in many cases dynamic dampers are ultimately fitted. Methods that modify the vibration characteristic of the response side or reduce vibration input are also available, to make it unnecessary to rely on methods involving dynamic dampers. Methods of modifying the response characteristic include modification of the body configuration and modification of the components of the steering system. In the main, structural optimization studies are conducted using CAE, and response characteristics are modified while attempting to minimize weight increases<sup>(1),(2)</sup>. However, the reduction of response at the steering resonance frequency without increasing weight presents a high level of technological challenge. While reduction of the spring constant of the engine mount and the suspension bush is

effective as a means of reducing vibration input<sup>(3)</sup>, in many cases this method is antagonistic to ride comfort and steering stability, and little beneficial effect can therefore be expected from it in terms of vibration reduction. Because there is a high level of vibration input when cylinders are deactivated in vehicles that employ cylinder idling systems, active engine mounts are employed as a method of reducing the vibration input transmitted to the body<sup>(4)</sup>. This is, however, a method that involves some cost.

The body, steering hanger, steering column, steering wheel, and Supplemental Restraint System (SRS) module determine the steering vibration characteristic. The relationship between the respective resonance characteristics of these parts determines the vibration characteristic in the completed vehicle. Based on the results of analyses of the resonance characteristics of these parts, the research discussed in this paper investigated a structure that would actively employ the resonance characteristic of the SRS module in order to increase marketability. In concrete terms, by modifying the support structure of the SRS module and using it as a dynamic damper, the research was able to realize a reduction in vibration with extremely minimal increases in weight and cost. This new structure can be employed in multiple vehicle types, and is able to contribute to increased vibration performance for marketability and reduced weight.

---

\* Automobile R&D Center

## 2. Mechanism of Vibration Reduction

### 2.1. Characteristics of Steering Vibration Phenomena

Three steering vibration phenomena can be indicated as resulting in a decline in marketability. These are idle steering vibration and launch vibration originating in the transmission of engine vibration, and driving vibration originating in the transmission of suspension vibration. These vibration phenomena represent an issue in the area in which the 1st-order resonance frequency of the steering system and the frequency of the input vibration in each case coincide. Today, given the tendency to use an expanded transmission lock-up range in order to increase fuel efficiency, the frequency with which engines operate at low engine speeds is becoming higher than was previously the case. In addition, the generation of an engine vibration component when cylinders are deactivated with the use of cylinder-idling systems is making steering vibration issues more complicated. This demands the realization of increased marketability in a wider range of frequencies than was previously the case.

Figure 1 shows the steering resonance frequency range and the frequency band for each vibration phenomenon. As a result of efforts to balance performance and weight-savings, the 1st-order resonance frequency of the steering system in the complete vehicle exists in the restricted range between 30 and 45 Hz in automobiles ranging from mini-sized to luxury vehicles. Steering vibration becomes an issue at frequencies at which this range and the input vibration for each phenomenon coincide. As the colored section of the figure shows, there is a shared frequency range in which this is the case of around 15 Hz for multiple vehicle types.

### 2.2. Conception of Using SRS Module as Dynamic Damper

The dynamic dampers used as a method of ameliorating steering vibration are positioned in the steering wheel. The steering wheel generates the greatest amplitude of vibration in the 1st-order resonance mode of the steering system, and the positioning of even a small mass close to

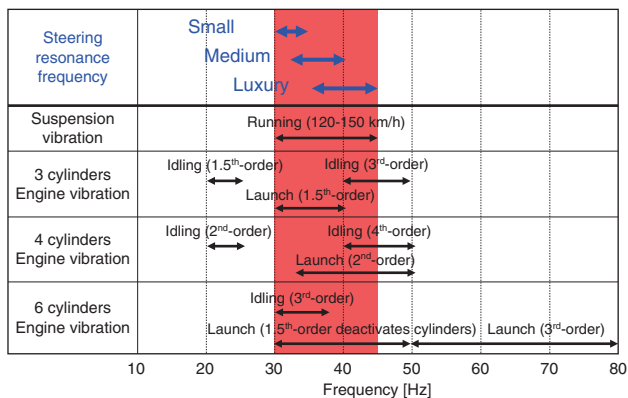


Fig. 1 1st-order resonance frequency of steering system and vibration phenomena

this component is able to produce a vibration reduction effect. However, frequency characteristics must be set for each vehicle model, and the dampers therefore lack general applicability.

The idea of using the mass of an SRS module as a dynamic damper was therefore conceived. This would make it possible to use an increased mass as a dynamic damper in approximately the same position as a conventional damper. SRS modules are safety devices employed as standard, and the main type in use is a floating type, in which the inflator floats by means of a horn spring. This configuration produces a resonance dependent on the mass of the module and the characteristics of the spring used to float the inflator. The resonance characteristic of this floating configuration was analyzed, and the potential for its use as a dynamic damper was verified.

### 2.3. Evaluation using Standardized Components

The resonance characteristic of the steering system of a complete vehicle was analyzed, and the effect of component parts was compared. In order to simplify the structural components, standardized components with sufficiently increased stiffness were employed (Fig. 2). Figure 3 shows these components when fitted and a comparison of their resonance characteristics.

Configuration D shows the body side of the steering hanger fixed to a rigid plate, and the other components made up of the components used in a complete vehicle, while Configuration A shows this configuration with the steering wheel replaced by a standardized component. In this configuration, steering wheel resonance and the effects of SRS module resonance are excluded. Measurements were conducted using Configuration C, in which the steering wheel was fixed to a rigid plate, and Configuration B, in which the SRS module only was fixed to a rigid plate.

A comparison of Configurations A and D shows that the response level is higher for Configuration A. Based on the results of a comparison of the resonance characteristics of the SRS module and the steering wheel, it was conjectured that the 84 Hz resonance frequency of the steering wheel was sufficiently higher than the 1st-order resonance frequency of the steering system to have no effect in Configuration D. The resonance frequency of the SRS module was 56 Hz, and it was considered that the effect of vibration reduction produced by adding a dynamic damper

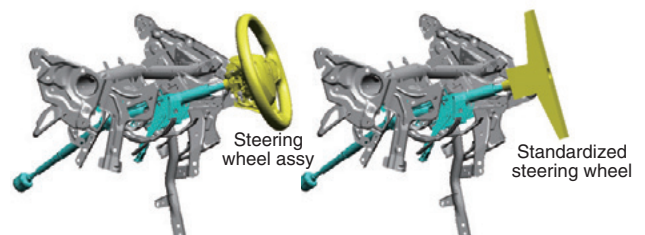


Fig. 2 Standardized components for measurements

to Configuration A was incorporated in Configuration D. It would be possible to expand this vibration reduction effect by optimally setting the resonance frequency of the SRS module.

#### 2.4. Range of Effect of Mass of Dynamic Damper

The necessary mass for the dynamic damper in order to realize a vibration reduction method with a high level of general applicability was determined. In order to help enable consideration of the relationship between the mass of the dynamic damper and its vibration reduction effect, the difference between two resonance frequencies obtained from the equation of motion of a two-degree-of-freedom vibration model was derived.

As shown in Fig. 4, the external force  $F_1$  acts on the main system determined by the mass  $m_1$  and the spring  $k_1$ .

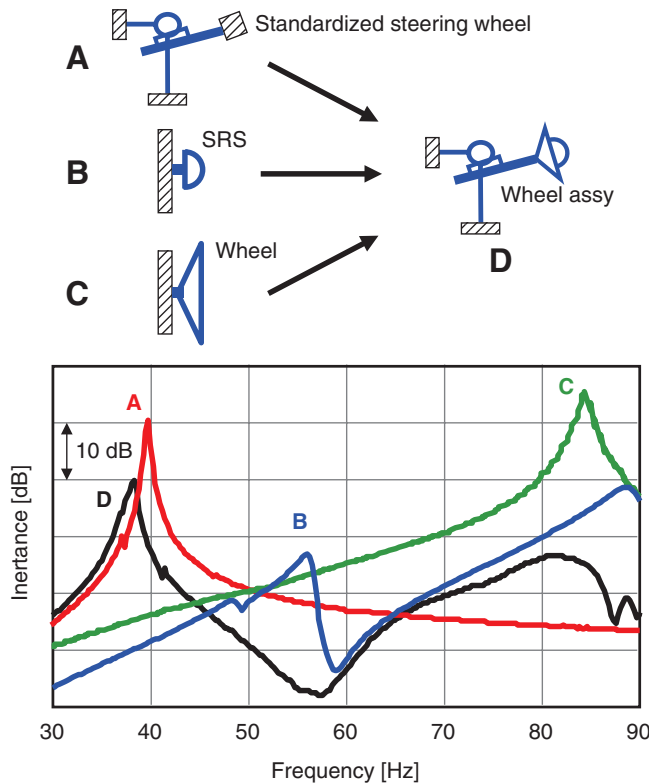


Fig. 3 Comparison using standardized components

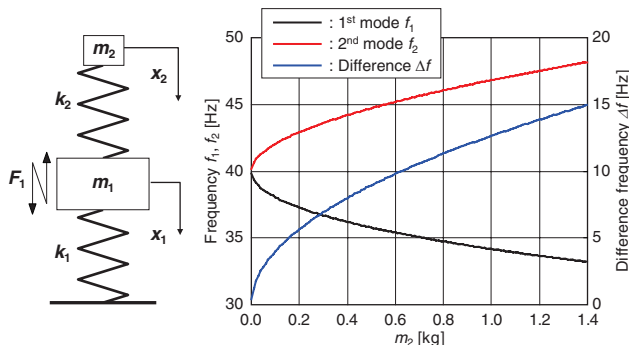


Fig. 4 Effect of dynamic damper and weight ratio

Above the main system is a dynamic damper determined by the mass  $m_2$  and the spring  $k_2$ . Equation (1) represents the equation of motion in this case.

$$\begin{aligned} m_1 \ddot{x}_1 + k_1 x_1 + k_2 (x_1 - x_2) &= F_1 \\ m_2 \ddot{x}_2 + k_2 (x_2 - x_1) &= 0 \end{aligned} \quad (1)$$

Equation (2) expresses the resonance frequencies of the general solution,  $f$ , which satisfy this condition.

$$f = \frac{1}{2\pi} \sqrt{\frac{1}{2} \left( \frac{k_1 + k_2}{m_1} + \frac{k_2}{m_2} \pm \sqrt{\left( \frac{k_1 + k_2}{m_1} + \frac{k_2}{m_2} \right)^2 - \frac{4k_1 k_2}{m_1 m_2}} \right)} \quad (2)$$

The respective resonance frequencies are further termed  $f_1$  and  $f_2$  from lower to higher. Here, with the resonance frequency of the main system and the dynamic damper identical at  $f_0$ , and the mass ratio  $\mu$  termed  $m_2/m_1$ , the difference in width between the two resonance frequencies,  $\Delta f$ , is given by Eq. (3).

$$\begin{aligned} \Delta f &= f_2 - f_1 \\ &= \sqrt{\frac{1}{2} \left( (2+\mu)f_0^2 + \sqrt{\mu(4+\mu)}f_0^2 \right)} - \sqrt{\frac{1}{2} \left( (2+\mu)f_0^2 - \sqrt{\mu(4+\mu)}f_0^2 \right)} \end{aligned} \quad (3)$$

The resonance frequency of the steering system and the frequency of the dynamic damper,  $f_0$ , was set at 40 Hz, and the mass  $m_1$ , the mass of the dynamic damper subtracted from the total mass, was set at 10 kg. Figure 4 shows results calculated with the mass of the dynamic damper added to the steering wheel considered as  $m_2$ , and the range of effect of the dynamic damper as the difference between the two resonance frequencies in the two-degree-of-freedom vibration model,  $\Delta f$ . In order to reduce steering vibration in a frequency width of 15 Hz, the mass of the dynamic damper would ideally be 1.4 kg. Given that the mass of currently used SRS modules is around 1.2 kg, it was predicted that the use of SRS modules as dynamic dampers would represent an effective vibration reduction method with general applicability.

#### 2.5. Resonance Characteristic of SRS Module and Target Characteristic

The results of unit excitation tests conducted on floating SRS modules indicated the existence of the vertical and horizontal resonance modes shown in Fig. 5.

The resonance characteristics of SRS modules differ significantly due to factors including their design, the position of their center of gravity, and their spring characteristics. Figure 6 shows the vertical and horizontal resonance frequency modes for eight different SRS modules used in different vehicle models. When the resonance frequency of the SRS module is close to the range of the 1st-order steering resonance frequencies in complete vehicles, a vibration reduction effect can be observed, but this effect is small when the frequencies are further separated. In order

for the SRS module to display an optimal steering vibration reduction effect, its resonance frequency should be set in the middle of the range of the 1st-order steering resonance frequencies in complete vehicles.

Given this, it would be necessary to make the resonance frequency lower than that of conventional modules, and a new structure able to stably realize this demand was therefore investigated.

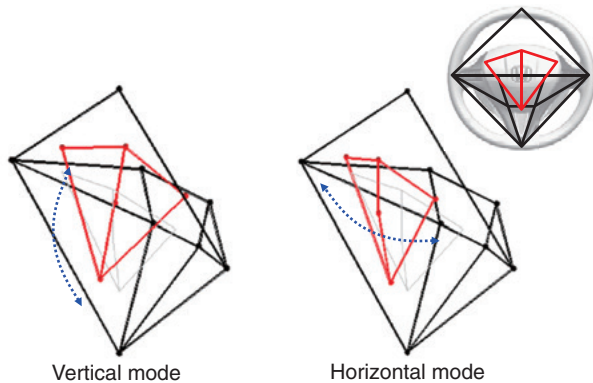


Fig. 5 Shapes of SRS module resonance frequency modes

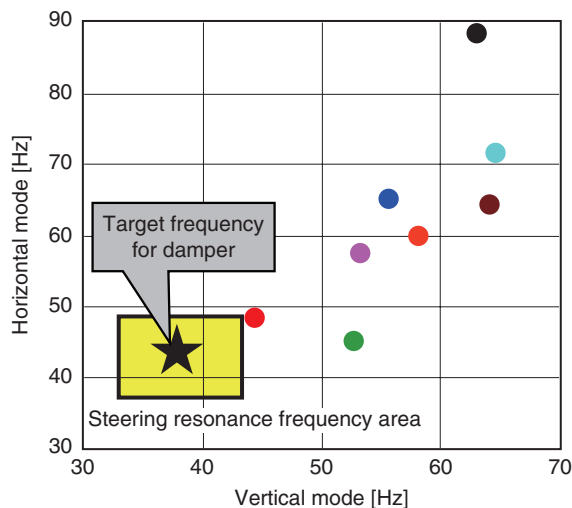


Fig. 6 SRS module resonance frequency

### 3. Development of SRS Module Damper Structure

#### 3.1. Overview of New Structure

This paper proposes an extension of the conventional concept of the dynamic damper, leading to a method of using an SRS module as a dynamic damper. Figure 7 shows a conceptual diagram of the proposed method. The target resonance characteristic was realized by changing the support structure at the point of connection to the horn that had been used in the existing floating structure. Figure 8 shows the component parts of the developed structure. The

distinctive feature of the new structure is the positioning of a floating bush between the insulators A and B.

#### 3.2. Proposal of Independent Spring Structure

Figure 9 shows the points of change in the developed structure and the concept of the associated spring model. In the conventional structure, the coil spring and the stopper bush are connected, and the resonance characteristic of the structure is determined by the shear spring characteristic ( $K_r$ ).

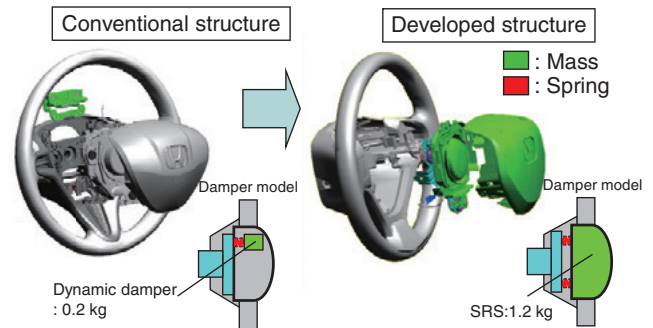


Fig. 7 Models of conventional and developed dynamic dampers

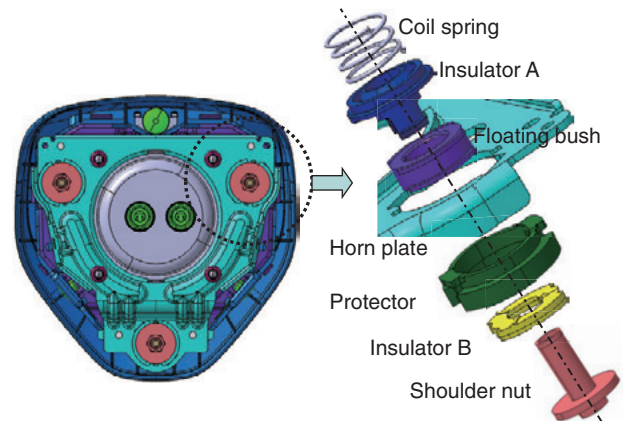


Fig. 8 Structure of SRS module damper

|              | Structure | Stiffness model |
|--------------|-----------|-----------------|
| Conventional |           |                 |
| Developed    |           |                 |

Fig. 9 Comparison of conventional and developed structures

of the rubber bush that is pushed by the coil spring. The structure's resonance frequency was higher than the 1st-order steering resonance frequency, and it was necessary to reduce it in order to realize the desired vibration reduction effect. It would have been possible to reduce the resonance frequency by lowering the spring characteristic of the coil spring ( $K_a$ ), but this would have resulted in the possibility of the horn sounding when the steering tilt and telescopic mechanisms adjusted the position of the steering system.

In order to resolve this issue, a method of using a separate spring that was not dependent on the spring characteristic of the coil spring was considered. A structure in which a spring characteristic,  $K_r$ , independent from the coil spring, could be set by positioning a floating bush between the insulators A and B was therefore proposed. Because the reaction force of the coil spring would be received by insulator A and the shoulder nut, it would be possible to reduce the resonance frequency of the SRS module with no effect on the spring characteristic perpendicular to the axis, as in the case of the conventional structure.

Figure 10 shows the relationship between the target for horn function and the target for SRS resonance frequency in relation to the coil spring.

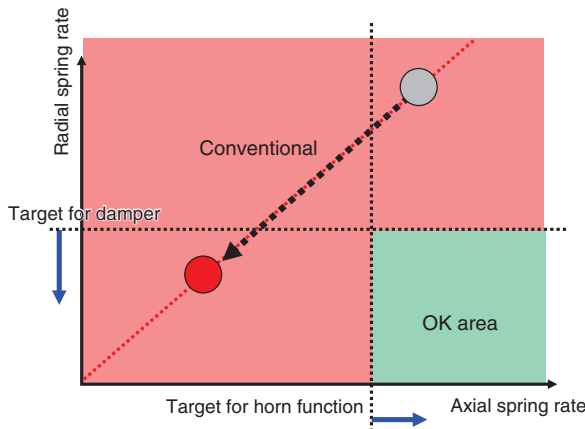


Fig. 10 Compatibility with horn function

## 4. Test Results

### 4.1. Results of Unit Verification Tests

The unit resonance characteristic of the developed SRS module was tested, and the results indicated that it realized the target characteristic shown in Fig. 6. Figure 11 shows the results of excitation tests. These results indicate that the peaks of inertance are the vertical and horizontal resonance modes shown in Fig. 5. The developed structure uses a circular floating bush, but vertical and lateral resonance could each be changed by altering the shape of the bush.

Unit tests were conducted in which the spring characteristic of the coil spring was varied in order to verify the viability of the proposed structure. The results of these

tests are shown in Fig. 12. While lowering the spring characteristic of the coil spring in the conventional structure resulted in a similar reduction in the resonance frequency of the SRS module, the frequency did not change in the new structure.

With regard to functioning as an airbag system, it was also demonstrated that the new structure achieved the same level of performance in terms of expansion strength and durability as the conventional structure.

### 4.2. Results of Vehicle Verification Tests

The results of a study of the steering vibration characteristic in a complete vehicle using the SRS module damper showed that both the vibration reduction effect and the range of frequencies in which the effect occurred were significantly increased with the use of the module against the use of conventional dynamic dampers. Figure 13 shows the vertical steering resonance characteristic. The vibration reduction effect of the new structure was 12 dB against 3 dB for a conventional dynamic damper, and the frequency range in which the effect occurred was approximately four times greater for the new structure at 18 Hz.

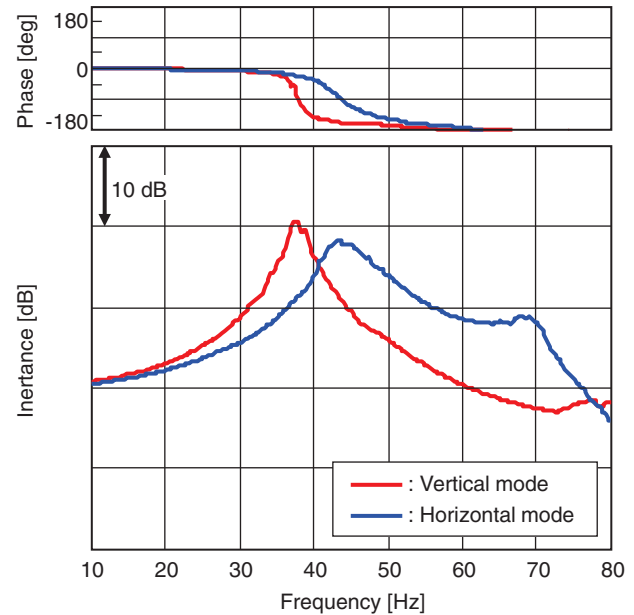


Fig. 11 Resonance frequency of SRS module damper

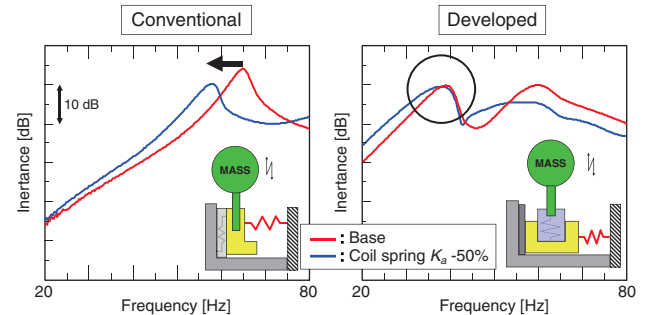


Fig. 12 Verification of stiffness model mechanism

The effect on the three vibration phenomena in completed vehicles was then examined. This study used the same SRS module dampers in different vehicle models.

Figure 14 shows the effect of the developed structure in ameliorating driving vibration and idle steering vibration. Driving vibration occurs when the 2nd-order component of tire rotation and the steering resonance frequency coincide. A vibration reduction effect of 6 dB or more was achieved at vehicle speeds from around 120 to 160 km/h. Strong idle steering vibration is generated when engine vibration components and the steering resonance frequency coincide. In the case of the tested vehicle using a four-cylinder engine, a high level of steering vibration was generated by the 4th-order engine vibration. The use of the SRS module damper resulted in a 10 dB reduction in vibration in the engine speed range set for idling. Similarly, in the case of six-cylinder engines, 3rd-order engine vibration produces significant steering vibration. The use of the developed SRS module resulted in a 12 dB reduction in vibration in the test vehicle.

Figure 15 shows the effect of the developed SRS module in ameliorating launch vibration. In many cases, launch vibration becomes an issue when the vehicle is operating in the low-engine-speed range under high loads for the purpose of increasing fuel efficiency. The use of

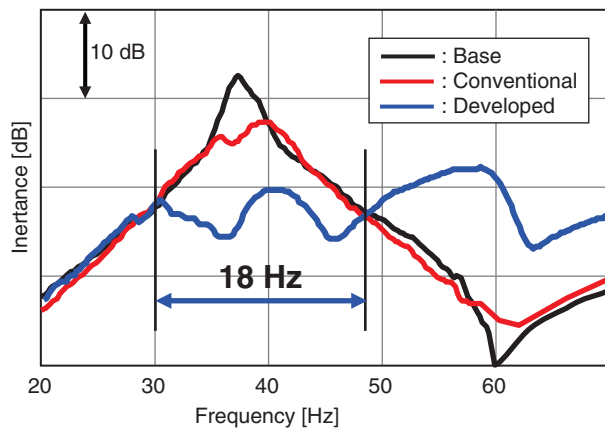


Fig. 13 Effect of SRS module damper in actual car

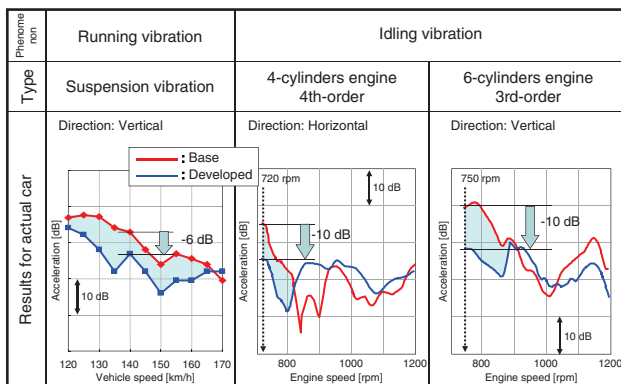


Fig. 14 Measurement results of running vibration and idling vibration

the developed SRS module reduced vibration by 10 dB or more in the vehicle using a four-cylinder engine, and realized a vibration reduction effect from 1200 rpm to 1800 rpm. Some six-cylinder engines are provided with cylinder-idling systems that deactivate the cylinders on one side of the engine. When this is the case, the engine speed and frequency conditions that excite launch vibration are different, and the achievement of vibration reduction for marketability can represent a further challenge. The use of the SRS module realized a reduction of 7 dB in launch vibration in the 1.5th-order range with deactivated cylinders. In addition, an 8 dB vibration reduction was also achieved in the case of 3rd-order vibration with all cylinders activated.

In designing a dynamic damper, it is necessary to consider the range in which the reduction of vibration is desired and the range in which vibration will increase due to an increase in the degree of freedom. In the case of the SRS module damper, by comparison with the vibration characteristic of conventional units, there is an increase in the vibration in the range close to 60 Hz (Fig. 12). However, while a tendency for launch vibration to increase in this range was observed in the cases of the four- and six-cylinder engines, the vibration level was low and its effect minimal in terms of marketability.

The results of the measurements in complete vehicles discussed above indicate the achievement of significant reductions in steering vibration and the realization of the goal of achieving an effect in multiple vehicle models using a single structure. Because the use of this structure makes it possible to realize the target characteristic for steering vibration from the initial stage of development, specifications can be decided on at an early stage.

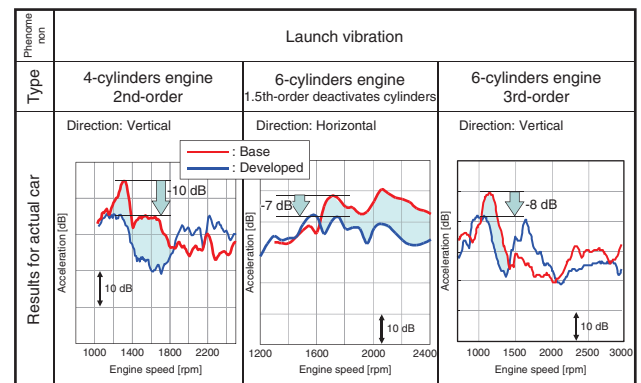


Fig. 15 Measurement results of launch vibration

## 5. Conclusion

A new method of ameliorating vibration phenomena originating in 1st-order resonance of the steering system was proposed, and a new SRS module structure was developed that realized the following.

(1) SRS module damper structure

A method in which an SRS module was employed as a dynamic damper was proposed, and the SRS module damper structure was realized.

(2) Balance between increased marketability and weight savings

This technology has achieved a balance between increased commercial appeal, cost and vehicle weight reduction.

(3) Early decision on specifications and reduction of development man-hours

The use of a vibration-reducing structure with a high degree of general applicability from the initial stages of the development process makes a conventional dynamic damper unnecessary, and also contributes to a reduction in development man-hours.

This new structure was employed in the new model CIVIC which went on sale in the spring of this year. It is also scheduled to be employed in subsequent models.

## References

- (1) Yokoi, T., Suzuki, K.: Steering Vibration Prediction with Finite Element Method, Proceedings of Society of Automotive Engineers of Japan, No. 25-04, P. 5-8 (2004)
- (2) Yamamoto, T., Maruyama, S., Nishiwaki, S., Yamada, H.: Feasibility study of a shape optimization technique for the car body in the initial design stage, Proceedings of Society of Automotive Engineers of Japan, No. 55-06, P. 17-20 (2006)
- (3) Yamazaki, K., Tsujiuchi, N., Koizumi, T., Takenaka, T., Iwagase, Y.: Improvement of Road Noise and Riding Comfort by Stiffness Change of Rubber Mounts Installed in the Suspension System, Proceedings of Society of Automotive Engineers of Japan, No. 52-01, P. 1-4 (2001)
- (4) Matsuoka, H., Mikasa, T., Nemoto, H.: Development of Active Control Engine Mount, Proceedings of Society of Automotive Engineers of Japan, No. 84-03, P. 5-8 (2003)

## ■ Author ■



Takashi KONDO



Masashi TERADA



Yukihiro KUNITAKE



Kunitomo MIYAHARA



---

Honda R&D Co., Ltd.  
Honda R&D Technical Paper Committee

Motorcycle R&D Center  
Automobile R&D Center  
Power Products R&D Center  
Aircraft Engine R&D Center  
Fundamental Technology Research Center  
Honda R&D Americas

---

Editorial Afterword

When members of the Editorial Committee meet to talk about the editorial policy of the Technical Review, each member has an opinion of what a technical paper is supposed to be, what an introduction of new technology is supposed to be, and how that is different from a product introduction. The committee similarly discusses science and technology. What is science, and what is technology? Ordinarily the boundary between them is not so clear, so we simply use the expression “science and technology.”

Generally speaking, “science” refers to an attempt to discover universal principles from observed phenomena, through the use of fixed and formulaic techniques, while “technology” is the application of science and practical arts to serve some purpose in people’s lives. Honda engineers are primarily engaged in the latter, because their job is to develop and market useful technologies for customers.

However, when engineers develop technologies and products, they encounter a variety of previously unknown phenomena. These phenomena range from the subtle (such as local stress) to the general (such as system optimization) and cover a wide array of areas, including performance, vibration, noise, combustion, measurement, control and materials.

Although engineers are supposed to develop technology, they often notice new phenomena that they must explain scientifically. The attempt to explain such a phenomenon begins when he realizes, “That wasn’t supposed to happen.” The process occasionally leads to new discoveries that become hints for future technical development. New technologies that lead to innovative products in this manner revitalize the developer, the company and the customer. It is the hope of the Editorial Committee that its work on the Technical Review will continue to inspire discussions of technology that lead to science, which will eventually lead to further technology for the development of innovative products.

—Kazuyuki Shiomi—

---

©2011 Honda Motor Co., Ltd. (All rights reserved)

ISSN 1881-6223

Published by:  
Honda Motor Co., Ltd.  
2-1-1, Minami-Aoyama, Minato-ku,  
Tokyo 107-8556, Japan

Edited by:  
Honda R&D Co., Ltd.  
Honda R&D Technical Paper Committee

Printed:  
Sep. 1, 2011



