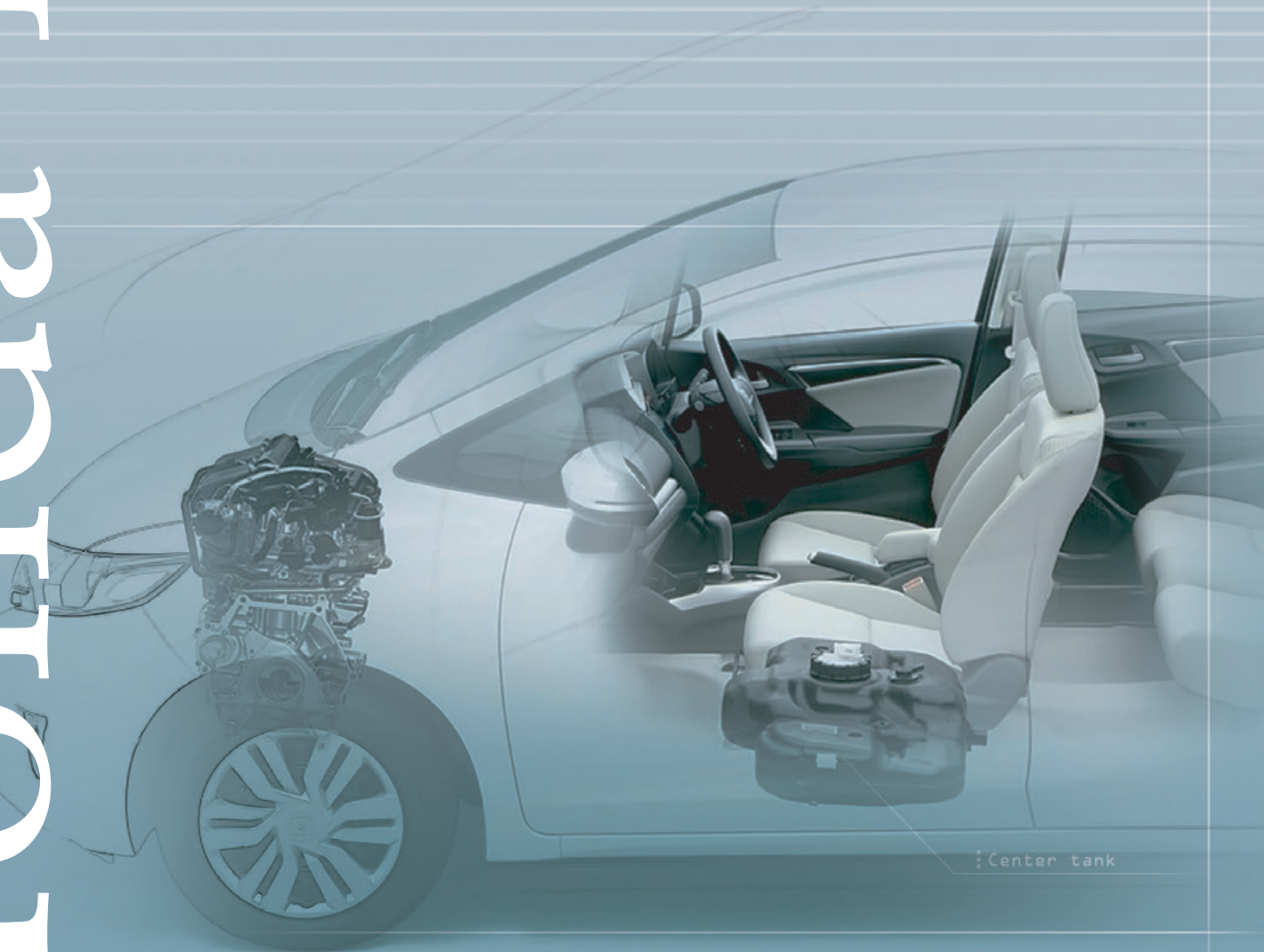
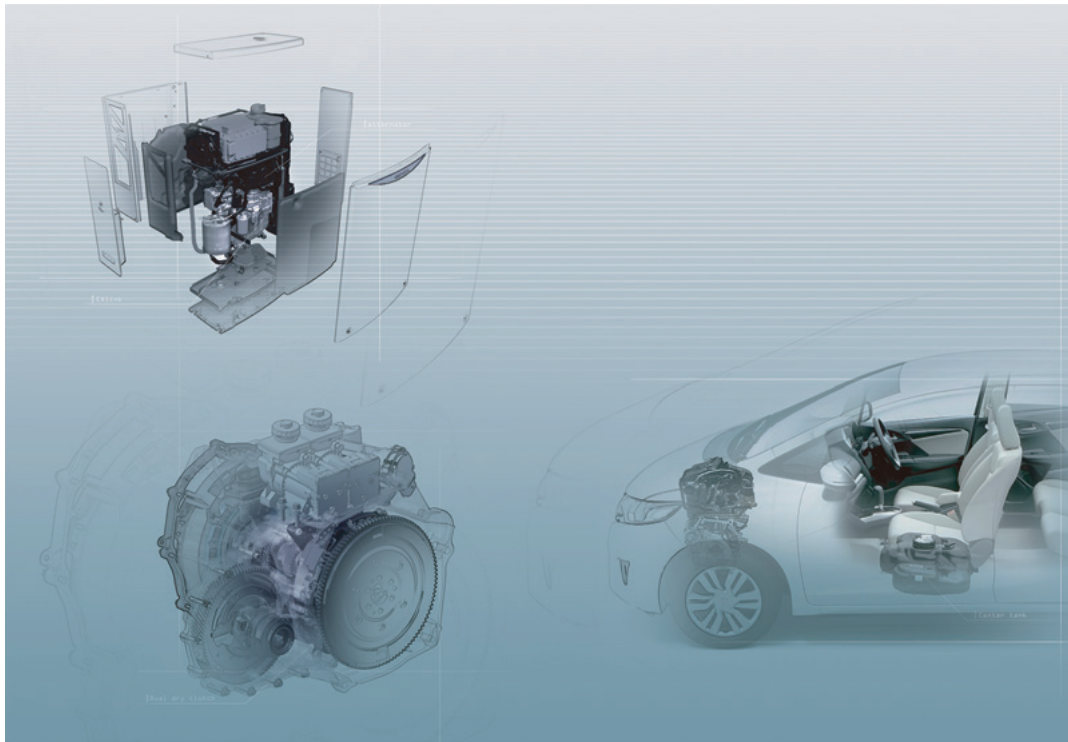


Honda R&D

Technical Review



April 2014
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Comment on the Cover

Honda's products are repositories of the dreams of the company's employees.

"It would be great if this product existed! I want people to use it in this way!"

"I want to make products that are better for the environment."

"I want to provide safe, high-quality products."

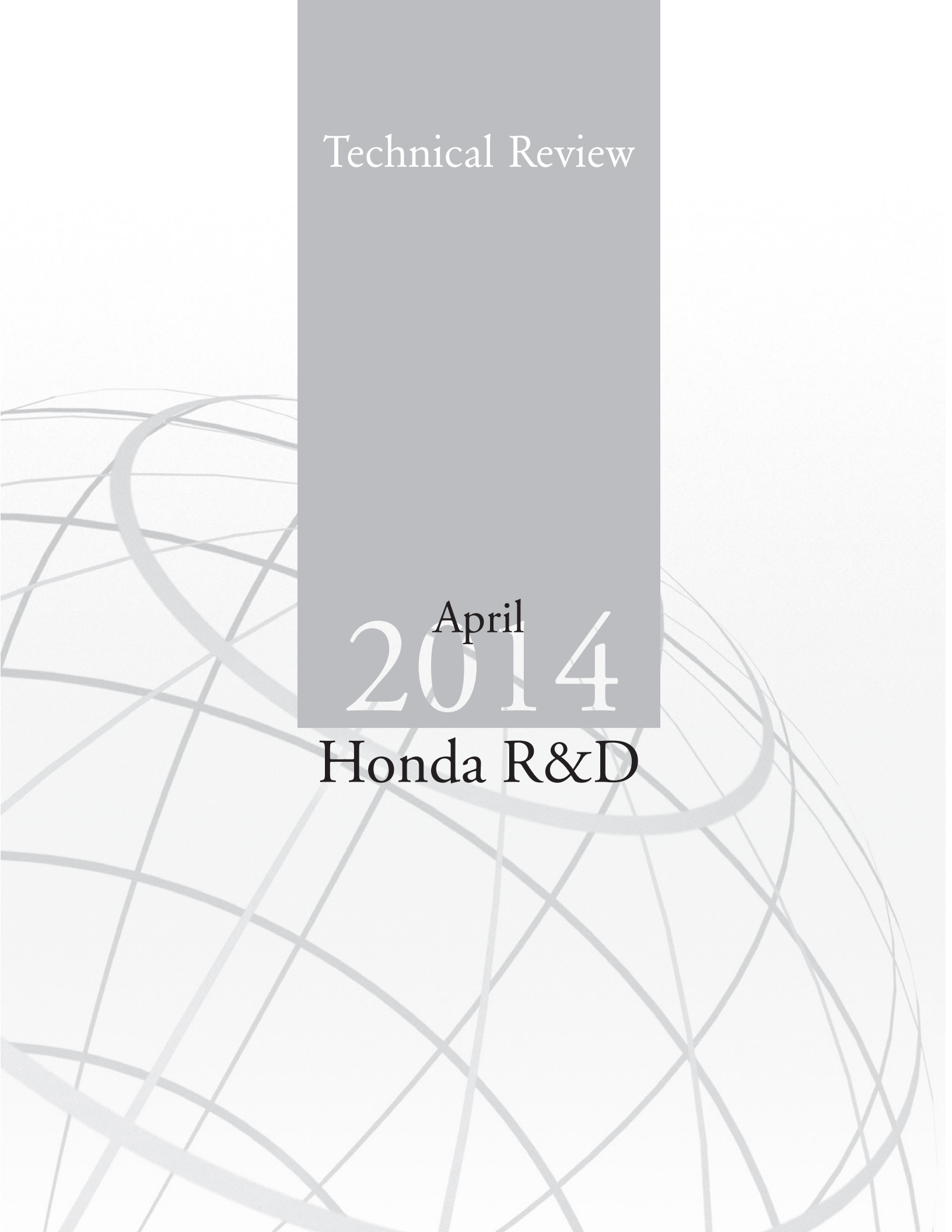
These dreams of our team members are incorporated in our products, ensuring that we send them to the market only after rigorous examination of even the smallest part.

We would be delighted if the products born from our dreams play a part in helping other people to realize their own dreams.

Our cover design offers close-ups of products and components discussed in the papers, and gives the impression of advancement and technological sophistication. Please enjoy this collection of our dreams.



Automobile R&D Center
Masahiro Sagawa

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

Technical Review

April
2014

Honda R&D

CONTENTS

Introduction of new technologies

2014 Model Year FIT Design Based on New Honda Design Concept	Yoshiya KITAJIMA•Atsushi MURAKAWA.....	1
Development of Honda Smart Home System	Koichi TSUNO•Takafumi KOTANI•Manabu DOBASHI Fumitaka KOIDE•Takahiro FUKUSHIGE•Tokitomo ARIYOSHI.....	11
Development of Highly Functional ECU for Next Generation of Inline-four Engines	Makoto MURAKAMI•Tatsuo YAMANAKA•Yasumasa KAITANI.....	18
Development of IPU for SPORT HYBRID i-DCD	Norihisa KOBAYASHI•Iori KANAMORI.....	23
Development of 7-speed Dual Clutch Transmission for SPORT HYBRID i-DCD	Norikazu KAWAI•Koji OHTA•Yasuyuki MASAOKA Hiromichi KONISHI•Heisuke KOBAYASHI.....	28
Study of High-efficiency CVT Shifting Mechanism Using Metal Pushing V-belt	Shuji ICHIJO•Tooru YAGASAKI•Taiki ANDO Kyohei SAKAGAMI•Soichiro SUMIDA.....	34
Development of New CVT for Global Compact Car	Tomoyuki KANDA•Hirohiko TOTSUKA•Masashi OONUMA Norimasa SHUDO•Tomohiro HIRATA•Ukyo OGATA.....	42
Development of Agile Handling Assist	Takeshi KOJIMA•Tomoyuki FUTAMURA•Kazutaka OHMURA.....	49
Noise Reduction Design in Printed Circuit Boards	Takashi NAMBO.....	58
Development of Material for Lightweight Catalytic Converter	Osamu SUZUKI.....	64
Development of Low-Nickel Heat Resistance Alloy for Exhaust Valves	Takashi TSUYUMU•Shigeki UETA•Mototsugu OSAKI Akihiro TOJI•Katsuhiko OHISHI.....	71
Aerodynamic Development of 2014 Model Year FIT	Kentaro MACHIDA•Munetsugu KANEKO•Terumasa KOTADA Atsushi MURAKAWA•Atsushi OGAWA.....	78
Aerodynamic Development of 2013 Model Year Accord and Accord Hybrid	Daisuke NAKAMURA•Yasuyuki ONISHI•Yoshiyasu TAKEHARA.....	86
Development of Side Curtain Airbag Deployment Simulation Technology Using CPM	Hisaki SUGAYA•Kazuo IMURA•Hiroyuki MAE.....	94
Investigation of Demand for Electric Vehicles in China	Junya TSUNEISHI•Yoshitaka TAKASUKA•Yoshiko TAKAHASHI.....	101

Technical papers

Research on Engine Control to Make Effective Use of Bioethanol-Blend Fuels Tadashi KUROTANI•Kohei KUZUOKA•Hirotsugu KUDO.....	110
Optimization of Driving Force Distribution Control in All-Wheel Drive Based on Wheel Rotation Speed Difference Between Front and Rear..... Shoichi OHYAGI•Takayuki SEKI.....	118
Development of Carburized High-strength Gear Steel with Less Rare Metals Added Akihito NINOMIYA•Yoshinari OKADA Masayuki HORIMOTO•Shusaku MAEDA.....	125
Development of Two-sensor Crash Detection Method for Motorcycle Airbag System Yuki KOBAYASHI•Takumi MAKABE.....	131
Development of High-performance PUF and Organoleptic Evaluation of Polyurethane Foam for Car Seat Naoto ISHII•Tomoki MATSUMOTO•Tutomu AO.....	139
Life Design and Evaluation Method of Type 3 High-Pressure Hydrogen Tank for Fuel Cell Electric Vehicle Toshihiko KANEZAKI•Satomi MANO•Kazuo MIYAGAWA Noboru HAYASHI•Takeshi OGAWA.....	150
Robust Optimization for Windmill Airfoil Design under Variable Wind Conditions Hiroki TACHIKAWA•Daniele Schiavazziz Toshiyuki ARIMA•Gianluca Iaccarino.....	160
Development of Inverse Problem Analysis System in Electromagnetics Masatoshi HARUMATSU•Katsuya YASHIRO•Kensuke SEKIHARA.....	166

2014 Model Year FIT Design Based on New Honda Design Concept

Yoshiya KITAJIMA* Atsushi MURAKAWA*

ABSTRACT

The 2014 model year FIT is the third generation of the model having undergone a full model change. Under the grand concept of “The World Best Functional Compact”, it was developed with the aim of making it the world’s most convenient compact car to use by offering exceptional space, fuel economy, and coolness (design and cruising).

The design of the 2014 model year FIT is the first to incorporate the new Honda design concept, “Exciting H Design.”

“Exciting H Design” consists of three facets: “High Tech” (people-friendly advanced technology built on the human-centered “Man Maximum, Machine Minimum” concept), “High Tension” (dynamic-looking framework), and “High Touch” (texture that appeals to the senses).

By incorporating the “Exciting H Design” concept, Honda has achieved a 2014 model year FIT design capable of asserting stronger individuality and presence in the world.

1. Introduction

The 2001 model year Fit (the first generation) used innovative packaging technology, called a center tank layout, to cause a “revolution” in small cars. The 2008 model year FIT (the second generation) retained the advantages of the first generation while advancing them in every way. It continued the success of the first generation as the downsizing trend spread worldwide. The FIT continues

to be popular with many customers having sold 4.87 million units in 123 countries to date.

The new 2014 model year FIT (the third generation) was built on the grand concept “The World Best Functional Compact” (Fig. 1). With this generation, the intention was to build a basic world car that would feel familiar to people, support their lifestyles, and make every day pleasant for customers in the same manner as Honda’s global long-seller SUPER CUB.



Fig. 1 Grand concept

* Automobile R&D Center

After deciding that the universal values of the small car should be space, fuel economy, and coolness, the designers developed the 2014 model year FIT with the aim of elevating each of these characteristics to a surprising new level.

2. Design Concept

The world small car market is growing but also increasingly competitive, so design has to be highly distinctive to avoid getting buried in the crowd. The FIT incorporated the new Honda design concept “Exciting H Design” (Fig. 2) consisting of the three facets “High Tech.” (people-friendly advanced technology), “High Tension” (dynamic-looking framework), and “High Touch” (texture that appeals to the senses). The exterior design was made to follow the “CROSSFADE MONOFORM” concept (Fig. 3) and the interior design follows the “Sophisticated Futuristic Cockpit” concept (Fig. 4).

In order to express its strong distinctiveness globally the 2014 model year FIT uses the Solid Wing Face, the new unified face design of Honda cars. It achieves “High Tech”



Fig. 3 CROSSFADE MONOFORM



Fig. 4 Sophisticated Futuristic Cockpit

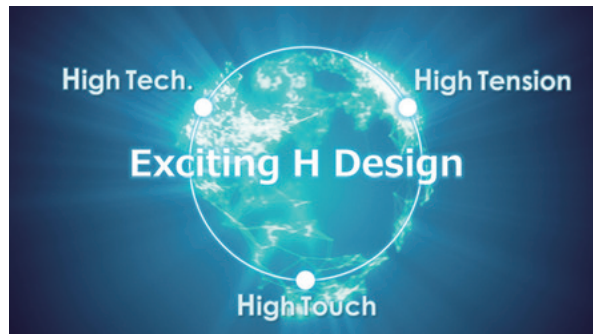


Fig. 2 Exciting H Design

with an advanced interface, “High Tension” with its dynamic mass look and personal cockpit framework, and “High Touch” with high texture transcending this small class.

3. Package Layout

The package concept for the new FIT was defined as “SMALL MAX PACKAGE” (Fig. 5). Following Honda’s traditional development concept of “Man Maximum, Machine Minimum” (MM), it aimed to further increase space efficiency and offer a more spacious interior than the first and second generations.

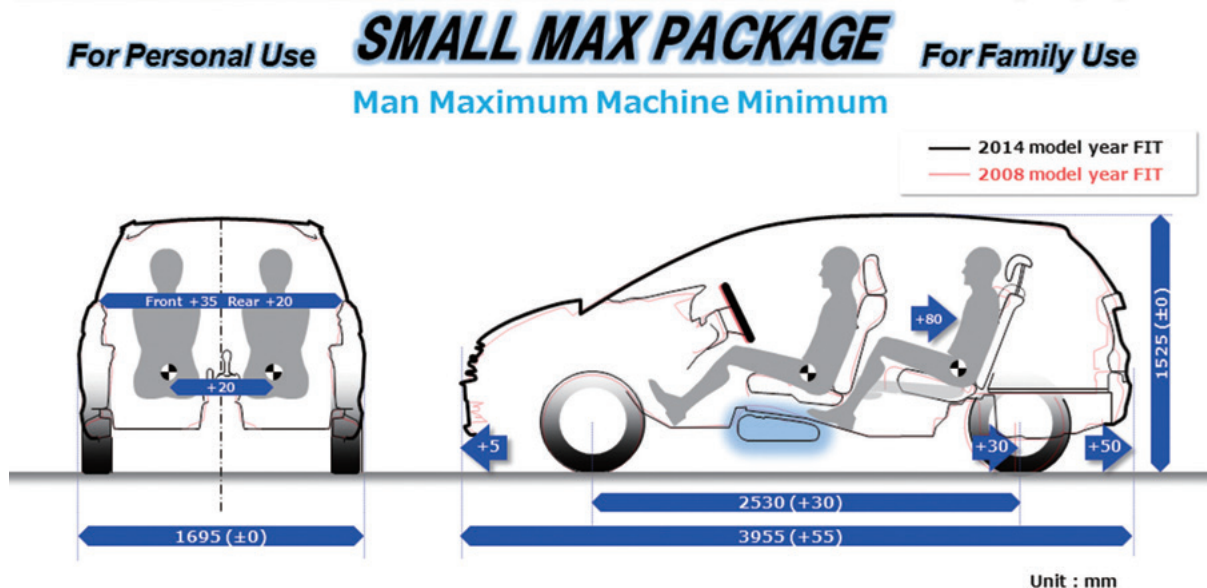


Fig. 5 Package concept

The vehicle is 55 mm longer and the wheel base 30 mm longer than the second-generation FIT, and the tandem distance (the distance between front and rear passengers) has been extended by 80 mm, a distance greater than the extension of the wheel base. By shortening the front-to-back length of the rear suspension, the cross member supporting the rear seat was moved toward the back. Shortening the rear wheel arch flange and enlarging the rear door opening allowed the designers to increase the tandem distance without sacrificing ease of rear seat access.

While the interior space has the same width as the second-generation FIT, the form of the door lining and the placement of the side windows were altered to expand the passenger shoulder space. The couple distance (the lateral distance between passengers) is 20 mm longer, and the shoulder space 35 mm greater in the front seat and 20 mm greater in the rear seat, combining for more spaciousness than in the second-generation FIT.

4. Exterior Design

To express the new FIT's grand concept of "The World Best Functional Compact," the exterior design retains the first and second generations' "one-motion silhouette", the single-stroke design that is symbolic of the vehicle's spaciousness and convenience.

The designers applied the concept "CROSSFADE MONOFORM" which suggests taking the upper mass consisting of a short engine hood (symbolic of advanced technology) and a cabin that continues smoothly from

it, and the solid-looking lower mass that emphasizes the tires, and knocking the top and bottom together to fuse them. Adding to this the designers pursued the look of an automobile that ranks above its own class and the dynamism of a moving object that would be suitable anywhere in the world. Furthermore, following the "Exciting H Design" that Honda is pioneering with the new FIT, they aimed for a design that would incorporate the three elements "High Tech," "High Tension," and "High Touch" and provide a strong presence (Fig. 6).

To achieve "High Tech," the designers aimed to embody a Solid Wing Face as a new design identity (Fig. 7). The role this plays is to express the innovative power train which uses the new FIT's Earth Dreams Technology to achieve best-in-class fuel economy.

The front face of the new FIT was designed with an aim to create an expression based on the Solid Wing Face concept that would be suitable globally, not just in Japan. Both the first-generation Fit and second-generation FIT had an expression of a small, sweet animal. The new FIT aimed to move away from this to the expression of a small but fearless and hardy lion cub (Fig. 8). Another aim was to give the design a sense of the distinctiveness and strength to survive in the world's highly competitive societies. To emphasize such distinctiveness, the front design features headlights that are long and narrow and give a sharp impression. The front grill is positioned to be continuous with the headlights and matches the texture of the lenses. This front design creates a dynamic image and advanced-looking expression.



Fig. 6 Final exterior rendering

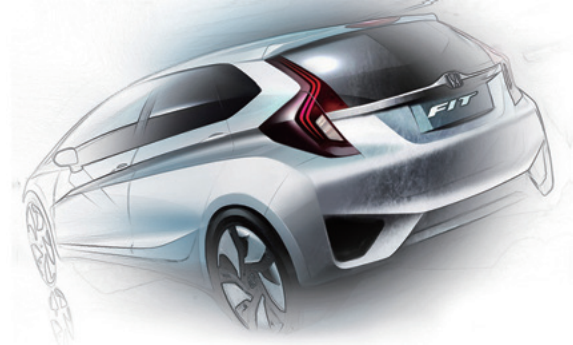


Fig. 7 Front face identity



Fig. 8 Change in facial expression

The Solid Wing Face evokes an image of two spreading wings with the “H” mark emblem in the center. This embeds the hope that Honda’s advanced technology will fly like a bird flapping its wings throughout the world. Expressing the image of flapping wings required a design that connects everything from the front grill to the headlights. An idea was needed that would make the texture of the front grill more like that of the lens on the headlight surface. The designers could have simply given the two the same texture, with a front grill consisting of a lens and housing plated by vapor deposition in the same manner as the headlights. However, while this would give a uniform feel, there was concern that the weight and cost would be more than double those of an ordinary front grill and adversely impact the fuel economy and cost performance that people expect of the new FIT. The designers then asked if it were not possible to achieve the desired feel at lower weight and cost. The result was to use a transparent smoke acrylic plate 3 mm thick for the front grill surface and apply mirror coating from behind so that the grill would give the appearance of being plated by vapor deposition. This made it possible to create an appearance similar to the construction of the headlight such that a plated layer can be seen through the lens. The back side of the acrylic plate, moreover, was given a vaulted form. The effect is that the front grill, as seen from the front, has high texture and depth such that it does not appear to be a 3-mm plate.

The designers similarly tried to give an advanced expression with the front and rear bumper designs by using a layered wing motif, like one wing overlapping another. The purpose here was to give a new sporty expression. Whereas the model sought to convey power in the past, the image that the designers were aiming for here was propulsive power using natural wind. The front bumper’s

center opening and the area below the center opening all the way to the lower end of the bumper are black. This further accentuates the three-dimensional, deep-looking wing form of the bumper (Fig. 9).

The rear likewise projects an advanced feel with the rear combination lamps and tailgate lamps located alongside the tailgate. The lamps use two LED light tubes which create illuminated lines and serve as a tail lamp. Even when it is dark the vehicle is recognizable right away as the new FIT. The result is an advanced-looking design with presence (Fig. 10).

In the pursuit of “High Tension”, the designers achieved physically beautiful proportions and an athletic form by dynamically showing off the one-motion silhouette retained from the first and second generation FIT.

The silhouette in side view needed to provide a sense of strength and confidence so that customers could drive the new FIT with a sense of assurance that this is a car driven around the world. The headlights were placed in a low position and the leading edge of the engine hood was set high and made thicker to emphasize the new FIT’s characteristic short engine hood. The expression is one of strength with a look of confidence as the vehicle cruises by.

The designers also incorporated techniques to give a dynamic look, from the engine hood’s high leading edge to the roofline that connects to it in a single flowing stroke. The roofline has its highest point above the heads of the persons in the front seats, and from there as it moves to the back it gradually slopes downward to the point above the heads of persons in the rear seats and on to the end of the roof. This roofline is flowing but projects a forward sense of motion as a moving object (Fig. 11). This silhouette modeling actually helps enhance aerodynamic performance. Techniques were also used to impart a sense of motion to the rear-end silhouette as seen from the side. The upper edge of the rear window on the tailgate was brought forward to give the window a forward tilt. The resulting side-



Fig. 9 Layered wing bumper



Fig. 10 Rear design

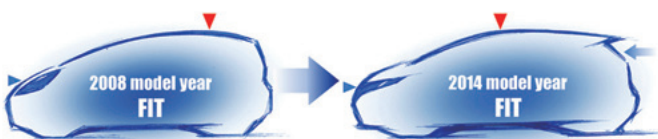


Fig. 11 Side view silhouette



view silhouette makes it look as if the mass, together with the roofline, was pushed forward. To further emphasize dynamism, the design incorporates the expression of a moving object in the front as seen from diagonally in front of the vehicle, and in the back as seen from diagonally behind it.

The aim was to give the front a dynamic expression using the long horizontal Solid Wing Face, the wide and rich-looking engine hood surface, and the three-dimensional front fender. The second-generation FIT had large headlights positioned so that they encroached deep into the upper face of the front fender. As a result, the fender design could only be expressed by the body side face, which limited the designers' modeling freedom. The new FIT, on the other hand, uses the Solid Wing Face to create a three-dimensional front fender with an upper face and side face without the headlights encroaching on the upper face (Fig. 12). The result is a dynamism and wide expression surpassing anything else in the class.

The back consists of an upper mass and lower mass to express an uplifting feel. The upper mass, comprising of the cabin, aimed to lessen the spacious wagon image by compressing the cabin. The lower mass, comprising of the body, projects much farther out than the upper part. The aim of this was to express cruising stability. The tires are also located farther out than the cabin. With this framework, the tires appear to be planting themselves into the ground which was intended to elicit a sense of stability. The voluminous rear bumper area has deeply sculpted modeling. Along with the planted tires, this creates an image not only of strength but also of the nimble cruising characteristics of a small car (Fig. 13). This two-mass (upper and lower) construction gives the new FIT rich-looking modeling beyond its class from the engine hood to the tailgate. It achieves a framework and structure that impart a sense of dynamism



Fig. 12 Front fender



Fig. 13 Rear design

and performance, the intrinsic attractions of a vehicle.

The designers aimed to achieve "High Touch" with dramatic surfaces – in other words, the deeply sculpted, modulating surfaces that draw in the human heart – in the limited 5-number size (measuring just 1695 mm side to side). Dramatic surfaces seek to express a surface quality rich in modulation and change by combining sharp edges and rich surfaces.

The front is expressed with sharp edge lines where the front pillars cut vigorously into the engine hood surface, and with surface quality changes with three-dimensional modulation at the roots of the front pillars where they merge with those edges. The designers considered the modeling countless times on clay models. They achieved modeling that offers strong support for the large cabin while presenting a surface that evokes a dynamic vigor and sultriness (Fig. 14).

The package layout and mechanical layout clashed at the body side. Because the aim of the package layout was to increase space, the new FIT, with its limited width, has a thin wall between the interior and exterior. A thin wall means a smaller design space for the exterior design and a smaller mechanical layout space. Multiple elements are involved with the mechanical layout, including the door-hinge areas, beams that respond well in collisions, the tracks of the side windows as they go up and down, and the position of the door handle. As design work proceeds, the designers constantly check the layout. The new FIT's designers used fine line tape to draw design lines on a full-size clay model to help them avoid interference with the mechanical layout. Then, using these lines as a guide, they performed modeling by adding or taking away clay. When modeling was finished, they applied silver film to the clay model, then checked the model's appearance and points of improvement outdoors. After that, they used measuring equipment to measure the model and check to what degree the mechanical layout and package layout interfered with each other. This process was repeated again and again.

The side panel's modulating design and the formation of the door hinge's mechanical layout were particularly challenging aspects of modeling, but the designers were persistent since these were exactly the body sections that would express the dramatic surface of the side. The body



Fig. 14 Engine food design

section is highly modulated: it starts with the groove of the highly sculpted character line, folds at an acute angle, comes back out, swelling as it moves downward then turning sharply in, then coming out at a gentle curve before folding again and moving inward (Fig. 15). The lower side of the door hinges, located above and below each other on the center pillar, were always interfering or sticking out. Achieving this highly modulated section was an important point affecting the new FIT's uplifting feel. In order to study the optimal positions for the design's fold line and door hinges, the designers rebalanced all lines and surfaces on the body side, and as necessary rebalanced even the side-view silhouette. As a result, they achieved a surface quality rich in modulation and change that would seem impossible in a width of only 1,695 mm. In this manner, they completed the new FIT's exterior design with a dynamism and high quality far beyond those of this small-car class (Fig. 16).



Fig. 15 Side panel design

5. Interior Design

The design team incorporated the three facets of “Exciting H Design” (“High Tech,” “High Tension,” and “High Touch”) into the concept “Sophisticated Futuristic Cockpit” expressing a high-texture advanced interface that lets the driver concentrate on driving and the modeling that envelops it. As a result, they expressed a more advanced character and higher quality beyond anything else in the small-car class (Fig. 17).

To provide “High Tech”, they incorporated leading-edge technologies in pursuit of an Advanced Interface that is easy to see and use (Fig. 18). The speedometer scale and numerals were made bigger, and the large, independent information display was set to the right of the meters, enhancing the advanced character and making information instantly recognizable.

The design has a more advanced feel owing to the virtual blue ring floating above the center of the speedometer (Fig. 19). To create the virtual display, a blue light source is located at a spot closer to the passenger side from the center of the meter indicator, and the light is radiated toward the speedometer dial face. The blue ring is displayed to look farther away than the actual dimensions allow creating an advanced appearance with depth.

The navigation system and fully automatic air conditioner located in the center panel use an electrostatic touch panel which is intuitively operated with taps, flicks, and other light touches like a smartphone (Fig. 20).

At the bottom of the fully automatic air conditioner is a silver-decorated horizontal line with a subtly irregular surface positioned to align with the display. This is designed



Fig. 16 Exterior design



Fig. 17 Final interior rendering

so that when the driver's finger touches it the driver will be able to feel around for the switch position.

For "High Tension", the designers aimed to create a sporty and emotional personal cockpit. The configuration concentrates the handling and display systems towards the driver's seat.

The meter panel directly in front of the driver's seat is located in the center, and outlets with metallic decorations

stretch out from it to the right and left. This modeling emphasizes the driver-centric instrument panel design.

The center panel is tilted toward the driver expressing a personal cockpit space that allows the driver to operate the controls easily even while driving. The upper part of the instrument panel is modeled with a clean surface, and the windshield wipers' resting point is below the front hood. These ensure a clear field of vision so the driver can



Fig. 18 Advanced interface



Fig. 19 Meter panel



Fig. 20 Center panel

concentrate on driving. The door mirror is attached to the body which eliminates any unevenness in the front quarter window frame.

In pursuit of “High Touch”, the designers’ careful work focused on the feel of the materials. The result is a diligent “craftsmanship” that reaches into every corner. To increase the value of the decorative parts, the design team pursued modeling consistent with the feel of the materials. The metallic decorative portions of the two sides of the meter panel, the steering wheel, shift panel, and door lining were modeled sharply to give a hard expression (Fig. 21). A soft pad panel was used for the instrument panel in the middle on the front passenger seat side. This created a rich, tension-filled surface. Adding a stitched look conveys a soft and high-texture feel. There is also a metallic decoration on the underside of the soft pad panel. With this contrast of softness and hardness, the two textures make each other stand out (Fig. 22).

Thus, by carefully crafting each element, the designers achieved high texture (Fig. 23).



Fig. 21 Silver decoration panel



Fig. 22 Soft pad panel

6. Color Design

6.1. Exterior

Exciting colors fitting the individuality of the customer were chosen for the exterior. The keywords were fresh and energetic colors, and sophisticated and personal colors, suited to customers’ various lifestyles. The lineup of 11 colors includes five new ones.

For fresh and energetic colors, the designers insisted on unique color gamuts and new textures, choosing colors with impact and a clear character. The hybrid theme color Vivid Sky Blue Pearl uses fine, solid-like pearls and high-chroma blue to express the exhilaration of a blue, cloudless sky. The color evokes both the eco-friendlier advanced character and strong cruising performance of the FIT.

Attractive Yellow Pearl is meant as a bright, vivid color that makes everyone turn their heads when they see it on the street. A high-chroma greenish yellow was chosen to fully emphasize the lightness and youthfulness of the new FIT.



Fig. 23 Interior design

It uses three coats of pearl to present a high-quality three-dimensional look emphasizing the dynamic form.

Four sophisticated and personal colors were chosen. Three new ones were developed with the sophistication to appeal to adult sensitivities taking into account the different tastes of customers of a wide range of ages. To these three was added one existing color (Fig. 24).

6.2. Interior

The basic concept for the interior colors was to provide high-class, high-texture coordination fitting the individuality of each customer. The designers pursued truly high texture

by insisting on materials that are pleasant to the touch and feel authentic.

The interior is a black monotone which creates a highly modulating space since the bright silver decorations showing the texture of aluminum jump out visually.

A soft leather pad is used for the middle of the instrument panel bringing out high texture beyond its class. The seats consist mainly of woven fabrics that are soft to the touch and designed to withstand dirt. The colors and patterns for each grade were chosen in consideration of the tastes of a wide range of customers (Fig. 25).



Fig. 24 Exterior colors



Fig. 25 Interior colors

7. Conclusion

The aim of the new FIT was to set a new standard for small cars. As such, it has remarkably advanced functionality and all aspects of design. The design team's hope is that the new FIT will be recognized as the first example of a new Honda design identity asserting a strong individuality and presence worldwide, and that this vehicle, along with the "Exciting H Design," will fit the lifestyles of our many customers and continue to be loved by them for a long time.

■ Author ■



Yoshiya KITAJIMA



Atsushi MURAKAWA

Development of Honda Smart Home System

Koichi TSUNO*

Takafumi KOTANI*

Manabu DOBASHI*

Fumitaka KOIDE*

Takahiro FUKUSHIGE**

Tokitomo ARIYOSHI**

ABSTRACT

The Honda Smart Home System was developed with the aim of reducing CO₂ emissions from household living down to one-half the 2000 level. The Honda Smart Home System conducts integrated energy management of heat and electricity by using a solar power generation panel and a gas cogeneration system as an energy creation mechanism and using the battery of a next-generation electric vehicle for energy storage.

The Honda Smart Home System uses vehicle battery information and navigation data from Honda's proprietary Internavi communication system. A cloud server assembles the Internavi information with household heat and electric power information to calculate household demand prediction. The Smart e Mix Manager functions according to operation planning based on the demand prediction results, and it was configured to allow highly efficient operation in modes that give priority to either CO₂ emissions reduction or utility cost reduction. This method is expected to yield a reduction of CO₂ emissions from a household using a car by one-half relative to the year 2000.

1. Introduction

As the world economy advances, total emissions of greenhouse gases are continually increasing, and energy problems and climate change are occurring on a global scale. According to the latest statistical data from Japan's National Institute for Environmental Studies, total fiscal year 2011 emissions of CO₂, which account for 90% or more of greenhouse gases in Japan, increased 8.4% relative to 1990 (the Kyoto Protocol reference year). The household sector accounted for 15.2% of that (an increase of 48.1%) and the transportation sector accounted for 18.6% (an increase of 5.9%)⁽¹⁾. At the 19th Conference of the Parties to the United Nations Framework Convention on Climate Change (COP19) held in November 2013, the Japanese government announced its policy for 2020 of making a 3.8% reduction relative to 2005. (This will be submitted to the United Nations by the end of March 2015.) In order to achieve the stated objective, the reduction of CO₂ emissions in the household and transportation sectors will be a key issue.

The housing industry has declared that it would make the net zero energy house the standard for newly built housing by 2020⁽²⁾. Net zero means that increased energy conservation performance, renewable energy utilization, and other such enhancement of the framework and

facilities of a house reduce its net annual primary energy consumption to zero or practically zero. Meanwhile, the shutdown of nuclear power plants due to the Great East Japan Earthquake of 2011 was followed by further growing uncertainties over power supply that have led to heightened calls for safety and security in housing. Corporations are working on smart house technology to meet these demands, and widespread adoption of the technologies has recently begun.

In 2012, Honda began selling a household cogeneration unit⁽³⁾ capable of operating independently during power outages. This unit is a component element of Eco Will Plus, which achieves a balance between optimization of the heat energy generated in households and a sense of security during power outages (Eco Will Plus is a registered trade mark of Osaka Gas Co., Ltd.). Honda is also selling electric vehicles (EV), hybrid electric vehicles (HEV), plug-in hybrid electric vehicles (PHEV), and other such next-generation electric vehicles equipped with batteries. Honda has built a system that combines these products with photovoltaic (PV) panels that use natural energy. That is the Honda Smart Home System (HSHS), which is endowed with safety and convenience, and which has the aim of reducing CO₂ emission from domestic life in households that use automobiles to one-half the 2000 level. This is to report on Honda's development of this system.

* Power Products R&D Center

** Fundamental Technology Research Center

2. Development Goals

The purpose in developing HSHS was to build an energy management system for the household that includes mobility, the conceptual sense of “Realizing the joy and freedom of mobility” and “a sustainable society where people can enjoy life.” Figure 1 shows a breakdown of household CO₂ emissions in fiscal year 2011⁽⁴⁾. Achieving a one-half reduction in CO₂ calls for cuts not just in heat and electricity, but also in mobility and all other elements, as well.

The present development proposes an efficient fusion of the house with the electric vehicle and the energy creation and storage mechanism, with the following two points as the goals.

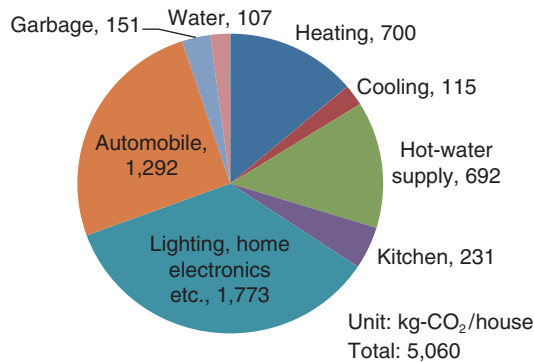


Fig. 1 CO₂ emissions from households (fiscal 2011)

- (1) Build an integrated power conditioner that links the electric vehicle and energy machines with the house. Build technology to enable the application of an EV battery as an energy storage machine without forfeiting the function of mobility.
- (2) Establish the technology to perform household demand prediction using Internavi information and household demand for electric power and heat. Determine operation plans for the energy machines based on that prediction, building a system that achieves a one-half reduction in CO₂ by producing the needed energy in the home (home production and home consumption of energy).

3. System Overview of the HSHS

Figure 2 shows the system configuration of HSHS. The energy creation and storage mechanisms, Smart e Mix Manager (SeMM), and cloud server that make up the HSHS, perform the following roles, respectively.

- (1) Micro combined heat and power system (MCHP)

The system creates electricity using a gas engine generator, recovers the waste heat from the engine, and has the capability to store the heat in a hot water heater tank. It enables household use of 1 kW of electric power output and 2.5 kW of heat output with combined efficiency of 92%. It serves as an energy creation machine that effectively generates heat and electricity in the household sector, where the level of CO₂ emissions from heating is high.

- (2) Photovoltaic (PV) panel

This serves as an energy creation machine that generates

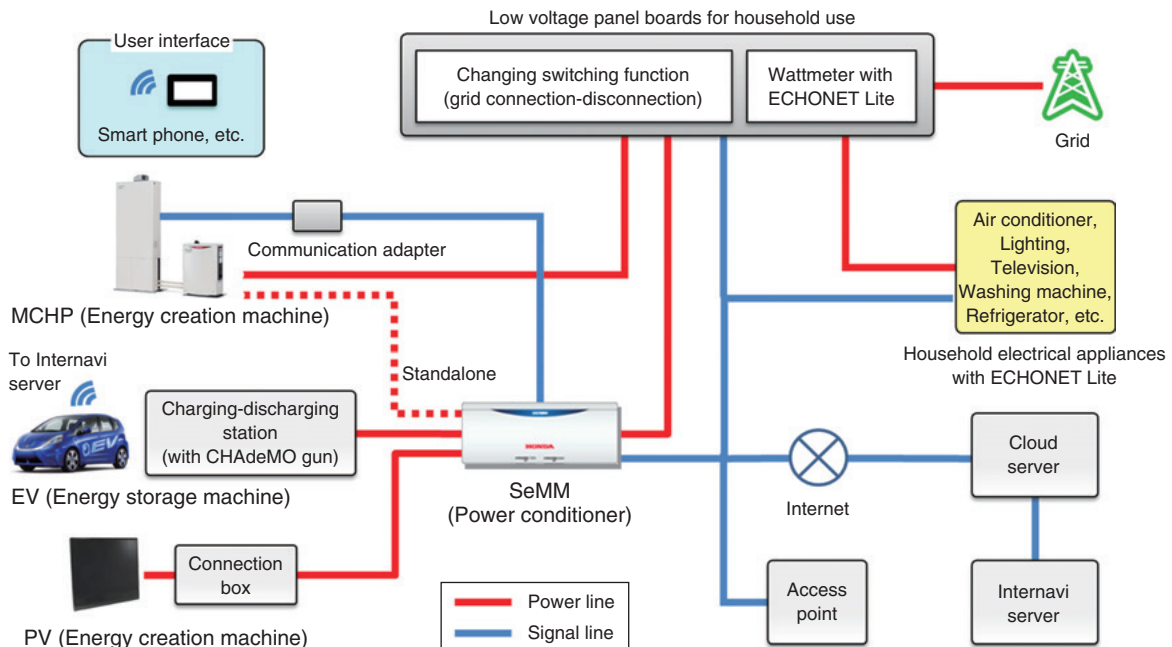


Fig. 2 System configuration of HSHS

electricity from solar energy, which is a natural energy. Its CO₂ reduction effect is large because the source of its energy is natural. The electricity can also be sold, making the system economically advantageous.

(3) Electric vehicle (EV)

In addition to its use for mobility, the utilization of SeMM when an EV is at home enables effective additional use as an energy charging or discharging and storage mechanism for electricity. The surplus power from the MCHP can be stored during midnight power service or when there is a low power load in the house, and stable power can also be supplied in the house when there is a shortfall in household or grid electric power, or when there are power outages. Since the vehicle is at home at irregular times, its application as an energy charging or discharging and storage mechanism calls for accurate prediction of periods at home together with vehicle battery information.

(4) SeMM

Providing integrated control of heat and electricity, SeMM implements the best mix of energy to maximize the separate features of the above three energy creation and storage machines when they are used in combination. SeMM controls grid interconnection, power generation, charging and discharging, and power sale in order to perform household energy management. Operation plans for the machines control the MCHP operating times and the amount of EV power charging and discharging by means of operation plans calculated on the cloud server.

When power outages occur, the combined operation of the SeMM with low-voltage panel boards for household use coordinate the MCHP, PV panel, and EV, and these three energy creation machines thus provide electricity within the household. The SeMM constantly monitors the state of the grid, and when a power outage occurs, it automatically switches to stand-alone power generation. The SeMM activates the grid interconnection/stand-alone switch built into the low-voltage panel board for household use so that power outlets can continue in use without first unplugging and replugging. The stand-alone power is supplied as single-phase three-wire power at 100/200 V so that household appliances can be used as usual even during power outages.

(5) Cloud server

Demand prediction for household heat and electricity, prediction of the amount of PV power generation, and prediction of EV at-home times are carried out on the cloud server in order to optimize the operation of energy creation and storage machines. MCHP operating times and EV battery charging and discharging amounts are controlled according to the prediction results so as to achieve increased system efficiency. The HSHS acquires information on electric power in the household from the low-voltage panel board for household use, heat information from the hot water supply system, and weather and vehicle information as well as navigation information from the Internavi server. It stores all of the information on the cloud server.

The SeMM has a constant connection to the Internet, and it coordinates home appliances by means of the Energy Conservation and Homecare Network (ECHONET) Lite⁽⁵⁾ while making its operation visible through the user interface.

This paper will introduce the technology that coordinates with the EV by means of the SeMM, which is a feature of the HSHS, and the technology that increases system efficiency by means of demand prediction and optimal control.

4. Introduction to the Developed Technology

4.1. EV Coordination Technology in the HSHS

It is a distinctive feature of the HSHS that its use of the SeMM allows the EV battery to be utilized as an energy storage machine without forfeiting the mobility function. Since it serves as an alternative to stationary storage batteries, it can reduce the user's initial investment. An overview of the EV coordination system and a description of the technology for coordinating between the house and the EV by means of Internavi communication are given in the sub-sections below.

4.1.1. Overview of the SeMM system for EV coordination

Figure 3 shows a simple function block diagram of the SeMM. A DC link system was adopted so that the EV could be charged at high efficiency using clean electric power generated by the PV panel. Higher efficiency can be anticipated because this approach enables the elimination of two power converters, an inverter and a rectifier, from an AC link with a stand-alone PV system. This also offers the advantage that maximum use can be made of electric power generated by the PV panel during grid power outages. MCHP stand-alone power can be fed in so that, if a power outage occurs, the MCHP can be operated to enable EV charging even during times of day when the PV panel is not generating power. The system can store electricity as a means for mobility during emergencies.

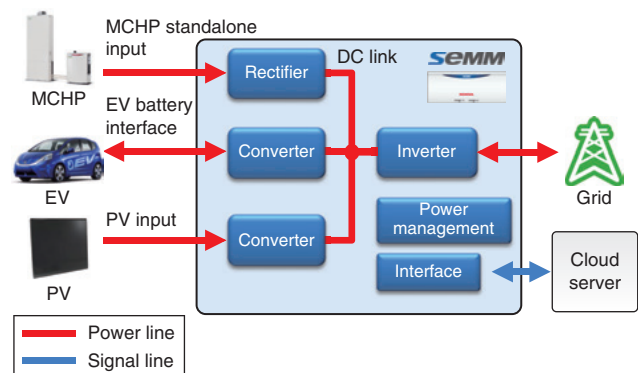


Fig. 3 Simple function block diagram of SeMM

The interface with the EV was made compliant with the DC version of the vehicle-to-home (V2H) guidelines published by the Electric Vehicle Power Supply System Association (EVPOSSA) in May 2013, as well as with CHAdeMO⁽⁶⁾, as a measure to enhance versatility and compatibility with products from other companies. When a vehicle that supports V2H is connected, the system goes to V2H mode and engages in charging or discharging operation. An isolated-type converter unit is used to connect to the EV in order to comply with Article 199, Paragraph 2⁽⁷⁾ of the Interpretation of Technical Standards for Electrical Equipment as amended in June 2012.

4.1.2. Technology for combining house-EV information using Internavi communication

Figure 4 shows a simple network configuration of SeMM. An EV battery is unlike a stationary storage battery in that it has a mobility function. This system uses remaining state of charge (SOC) information about the EV battery together with navigation information so that it can employ the EV battery as an energy creation and storage machine without obstructing that function. When the EV is linked to the house, SOC information is downloaded through communication between the SeMM and the EV, and when the EV is away, the information is downloaded from the EV by means of the Internavi server. This arrangement allows the cloud server to monitor the SOC at all times. The EV navigation information is uploaded through the Internavi server to the cloud server. The configuration was arranged so that household and EV information is centrally managed on the cloud server, where optimal energy machine operation plans are calculated.

Automatic control is the basic mode of HSHS operation. However, the system is also configured with a charge and discharge mode that the user can choose to set to give priority to the means of mobility so as not to obstruct “the joy and freedom of mobility” when the user, for example, suddenly decides to go out.

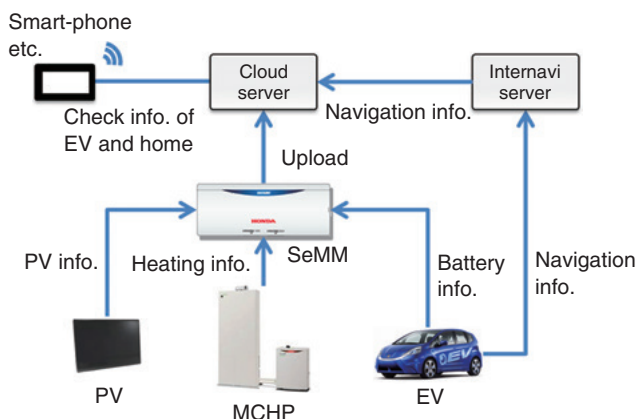


Fig. 4 Simple network configuration of SeMM

4.2. Energy Demand Prediction and Operation Planning Technology

HSHS has two strategies for reduction of CO₂ emissions. One is improvement of efficiency in each component (e.g., MCHP, PV) and the other is an optimal operation plan in each component using energy demand prediction. When the prediction is in good agreement with actual energy demand, the SeMM controls each component on the basis of the operation plan. If the prediction is not in good agreement with actual energy demand, the SeMM uses instantaneous control in consideration of the contracted electric power. Under present laws and regulations, reverse power flow to the grid is not allowed when an EV is discharging. The system is therefore provided with a reverse flow prevention function for when sharp load fluctuations occur. This system mainly consists of two parts: one is energy demand prediction and the other is the operation plan. Therefore, in the following sub-sections, an overview of the system, energy demand prediction methods and their accuracy, and operation planning based on energy demand prediction are shown.

4.2.1. System Overview

Figure 5 shows a conceptual diagram of the energy demand prediction and optimal control system. The present energy demand status of the house is sent from the SeMM over the Internet to the cloud server. On the cloud server, the future energy demand is predicted and operation plans for the components are generated. The generated operation plans function by means of the Internet and the SeMM to control the operation of each component. It is a distinctive feature of this system that it always keeps energy demand data stored in a database (DB), and by using that data to make predictions, it keeps up with changes in family structure and electric appliances in the household as it continues to yield a CO₂ reduction effect. The system detects the differences between demand predictions and actual demand, and when the difference is large, it makes a further prediction. As described above, the system continues to reduce CO₂ emissions by means of improved prediction accuracy.

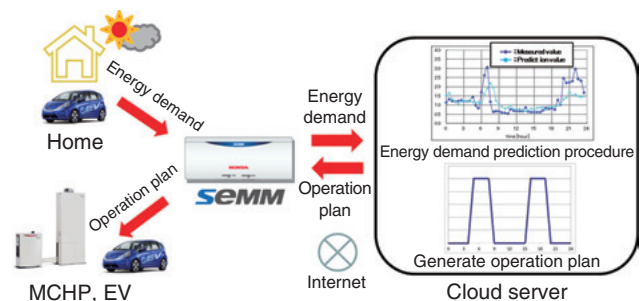


Fig. 5 Conceptual figure of prediction energy demand and operation plan

4.2.2. Energy demand prediction

Figure 6 shows a conceptual diagram of energy demand prediction. The system predicts future energy demand by an approximation model that is generated by machine learning of past energy demand. The HSHS improves prediction accuracy by a method of performing automatic categorization of energy demand and generating an approximation model⁽⁸⁾ of each domain that has been categorized.

More detailed explanations of the approximation model, categorization, and performance evaluation are described below.

(1) Approximation model

The approximation model of energy demand prediction is a function that takes past energy demand as input values and outputs future energy demand. That approximation model includes a number of parameters, the values for which are estimated by machine learning using DB. Support Vector Regression (SVR)⁽⁹⁾, which possesses a high degree of generalization performance in approximation models, is used for energy demand prediction.

(2) Categorization

Categorization of energy demand data uses the K-means method⁽¹⁰⁾. The K-means method is a form of unsupervised learning that uses an algorithm to automatically perform categorization just by inputting data.

(3) Performance evaluation

The prediction performance evaluation data consisted of 230 days' worth of actual demand data from the demonstration house⁽¹¹⁾ that Honda has been operating since April 2012. The system used data from the past 72 hours to predict the coming 48 hours. Prediction accuracy was evaluated by taking one day's worth of data from the 230 days of demand data as evaluation data, with the remainder used as learning data. Following the evaluation, the testing was repeated using the leave-one-out method so that all the days would be taken as evaluation days one time.

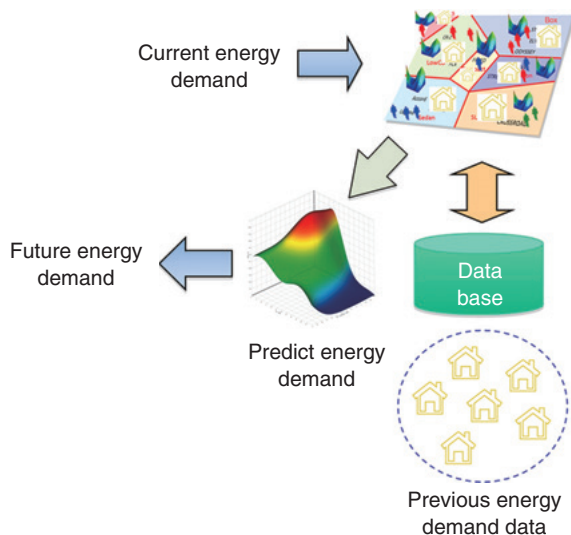


Fig. 6 Energy demand prediction procedure

Table 1 Prediction accuracy results

Item	Average 3 days	w/o clustering	w/ clustering
RMSE	0.267	0.225	0.215
Correlation	0.069	0.591	0.647

Root mean square error (RMSE) and correlation coefficient were used as the two prediction performance evaluation indices. A larger root mean square error value indicates a larger prediction error, and a correlation coefficient value that approaches one more closely indicates a stronger correlation, meaning that the model has a high degree of reproducibility. The above evaluation indices were used to evaluate prediction performance. The following two methods were used for the purpose of comparison.

- Taking the average of the preceding three-day period as the prediction data
- Prediction by the approximation model alone without categorization

Table 1 shows evaluation results. Both the root mean square error and the correlation coefficient indicate a high level of performance by the proposed method, verifying the validity of this method.

4.2.3. Operation planning technology using demand prediction

Utilizing the prediction data described above, calculations are carried out on the cloud server to minimize CO₂ emissions or to minimize fuel and lighting expenses. The MCHP operation plans and battery charging and discharging patterns are decided accordingly. The SeMM controls the energy creation and storage machines to yield benefit for the user as well as to meet the CO₂ reduction target. Figure 7 shows an example of an operation plan that includes the vehicle. The EV is in use for mobility

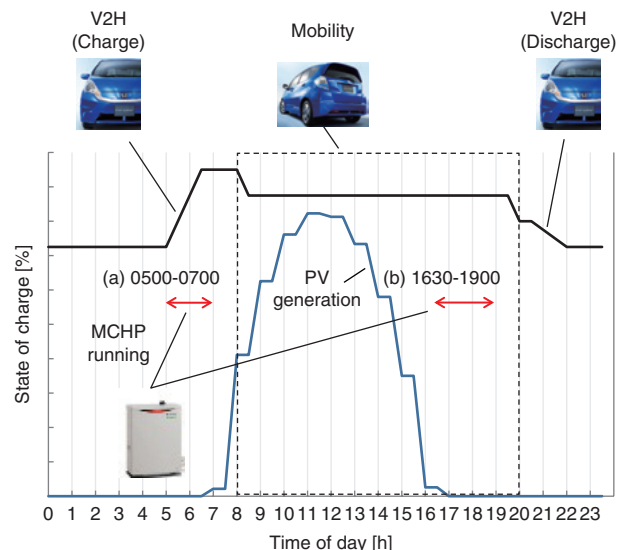


Fig. 7 Example of vehicle and MCHP operation plan

Table 2 Machine specifications for simulation

MCHP		PV panel	EV battery
Heat	Electricity		
2.5 kW	1 kW	4.6 kW	20 kW-h*

*EV battery can be used in V2H mode from 2000 to 0800 hours.

Table 3 Assumptions regarding annual demand

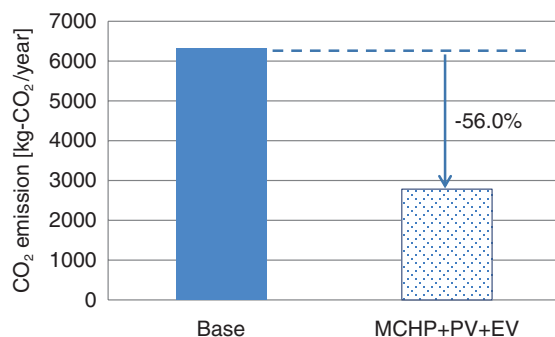
Number of people	Heat demand	Electrical demand	Mobility mileage
4	7028 kW-h	5250 kW-h	8000 km

- Calculation conditions -

Electric power consumption rate 0.69 kg-CO₂/(kW-h), Gas consumption rate 2.31 kg-CO₂/m³, Gasoline consumption rate 2.32 kg-CO₂/L, EV efficiency (JC08 mode)106 W-h/km, Fuel efficiency 16.6 km/L

away from the household from 0800 to 2000 hours. At other times the EV is at the house, and those are designated as possible V2H times. The MCHP ordinarily operates at peak demand times in the morning and evening. Charging operation is carried out to match MCHP operation (a) in the morning, thus limiting the amount of surplus power from the MCHP. During MCHP operation (b) in the evening, surplus power is unlikely to be generated because household power consumption is high, and operation is matched to heat demand even when the EV is not present. After the EV returns home, it is made to discharge at times when the household load is large during the period before midnight power service goes into effect. System efficiency can be expected to increase due to demand prediction and operation planning technology.

The annual amount of the CO₂ reduction achieved when control of the energy machines is implemented so as to minimize CO₂ emissions was calculated. Figure 8 shows the results from simulated optimization in terms of CO₂. Table 2 shows the specifications of the energy creation and storage machines used in the simulation. Table 3 shows the assumptions that apply to annual demand for electric power and heat as well as to vehicle operating distance. These tables indicate that the operation of this system and

Fig. 8 Simulation result of annual CO₂ reduction rate

the application of the control technologies described above, including mobility operation, can be anticipated to achieve an annual reduction by one-half in the base value for CO₂ emissions under conditions of gasoline-fueled vehicle use without operation of the energy creation and storage machines, confirming the usefulness of this approach.

5. Conclusion

The HSHS was developed as a total energy management system that combines mobility and energy machines (MCHP and PV panel) with the house. Realization of the main technologies yielded the following conclusions.

- (1) Coordination of the SeMM with Internavi realized utilization of the EV as an energy charging or discharging and storage mechanism that serves as an alternative to a stationary storage battery. Value-added use of the electric vehicle can thus be anticipated to reduce the user's initial investment.
- (2) Technology that calculates household demand prediction based on Internavi information and household power and heat information, and that implements optimal control for high efficiency based on the demand prediction results, was established. Future application of this technology and operation of the SeMM according to operation plans based on predictions can be anticipated to achieve a one-half reduction in CO₂ emissions relative to conventional machines.

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■ Author ■



Koichi TSUNO



Takafumi KOTANI



Manabu DOBASHI



Fumitaka KOIDE



Takahiro FUKUSHIGE



Tokitomo ARIYOSHI

Development of Highly Functional ECU for Next Generation of Inline-four Engines

Makoto MURAKAMI*

Tatsuo YAMANAKA*

Yasumasa KAITANI*

ABSTRACT

We developed an engine control unit having a new functional composition, with the goal of applying it to the next generation of direct-injection engines, port-injection engines, and hybrid electric vehicle engines. To achieve these functions, the heat generated by the components and the number of parts were reduced. As a result, we decreased generated heat and reduced the number of parts by 30% compared to the conventional engine control unit.

Also, in order to reduce the minimum injection quantity in direct-injection engines, feedback control was performed by detecting the close timing of the injector, and we reduced the minimum value of the injector drive signal time by 46% compared to the conventional system.

1. Introduction

When considering the variations of power train systems, the conventional combinations of the engine and transmission were limited, as Fig. 1 shows. As a result, the combinations of engine control unit (ECU) and transmission control unit (TCU), as well as the functions they required were also limited. To address this, there are past examples of ECUs and TCUs being integrated and developed as a single unit. However, due to environmental regulations that have continued to become more stringent in recent years, the functions required for the engine and transmission have also increased as shown in Fig. 2. As these functions increase, it has become difficult to integrate the ECU and TCU, so they must now be developed separately from one another. In terms of engines, the use of direct-injection engines has expanded, as these offer

superior fuel consumption. Also, the size of the engine is being further reduced through the use of supercharged mechanisms⁽¹⁾. As Fig. 3 shows, in order to achieve these new functions, there are more types of devices associated with the engine, which leads to a higher number of parts composing the ECU and an increase in the heat that is generated from parts. Also, a new algorithm to control devices has become necessary⁽²⁾. Meanwhile, it is necessary to minimize cost and size, while dealing with the increase in functions. Therefore, we developed a highly functional ECU capable of being used in the next generation of direct-injection engines, port-injection engines, and hybrid electric vehicle (HEV) engines.

This paper introduces example applications of function

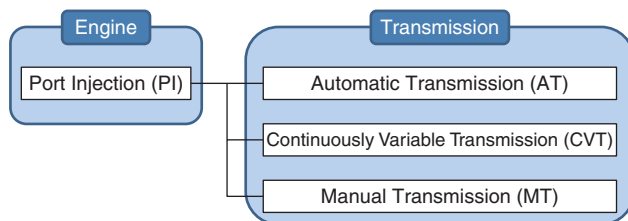


Fig. 1 Power train variations

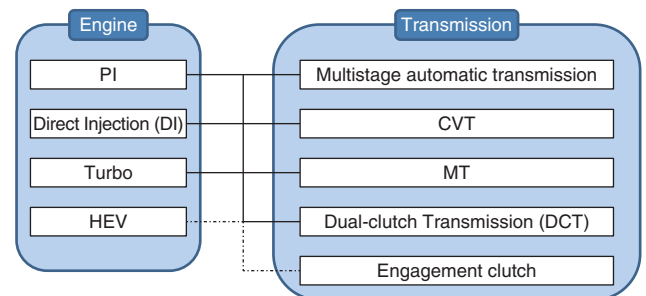


Fig. 2 Diversification of power train variations

* Automobile R&D Center

integration technology for establishing a highly functional ECU and technology for decreasing the ECU circuit-generated heat that increases as the number of devices increase.

Also, CO₂ emissions regulations and fuel consumption regulations for vehicles are becoming more stringent, and regulations on gas emissions are expected to become stricter throughout the world in the future⁽³⁾. In Europe, the new law EURO6 is scheduled to go into effect beginning in 2014. For direct-injection engines in particular, multi-stage injection will be required in order to decrease particulate matter (PM)⁽⁴⁾. Multi-stage injection is known to help reduce PM by enhancing the homogeneity of the air-fuel mixture and suppressing the adhesion of fuel to the surface of the internal wall of the cylinder. For multi-stage injection, it is necessary to reduce the minimum injection quantity by decreasing the minimum usable value (MinTi) of the injector drive signal time (Ti). As an effective means of decreasing this MinTi, we introduce technology that detects and corrects the close timing of the injector.

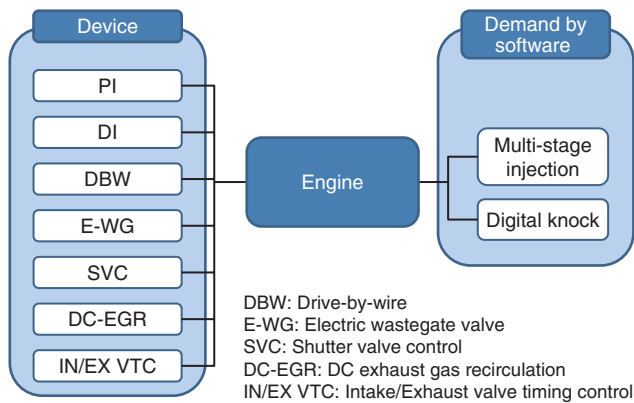


Fig. 3 Engine device and demand by software

2. Development Goals

As mentioned earlier, devices and software are increasing as functions for enabling technology required to address environmental regulations increase. Table 1 shows the relationship between this technology and devices/software. Also, when engines for HEVs are assumed in addition to direct-injection engines and supercharged mechanisms, the number of circuit components for the ECU grows, leading to the concern that heat generated by the circuitry will also increase.

As far as the rising number of part indicates, the circuit components are reduced by using integrated driver ICs and a buck-boost power supply to revise circuits that operate at low voltage and by using microcomputer functions to simplify peripheral circuits.

Table 1 Engine technology

Technology	Device and software
Lean-burn engine technology Engine with supercharger technology Downsizing engine technology	E-WG DC-EGR
Atkinson cycle technology	IN/EX VTC
PM reduction technology	Multi-stage injection Dual injection
Knock detection technology	Digital knock filter

Also, the heat generated by circuits in direct-injection ECUs is higher than in port-injection ECUs. Therefore, heat-radiating structures are becoming more complex and the structural weight is increasing. In order to decrease this structural weight, the actual heat generated by the injector drive circuit was reduced, and heat-radiating structures were simplified.

In terms of software development, we placed a focus on multi-stage injection, which is effective at decreasing the PM of direct-injection engines. Technology that detects and corrects the injector close timing is effective at decreasing the MinTi per injection, and we confirmed the feasibility of that technology.

Our goals are summarized below:

- (1) Part integration to achieve functions that assume applications in various engine types.
- (2) In terms of direct-injection injector control: the reduction of heat generated by drive circuits and creating a means of reducing MinTi.

3. Part Integration to Achieve Functions

The ECU that we developed can be applied to the engine types shown in Fig. 4, and this ECU has approximately 200 functions. We introduce technology that was utilized to achieve these functions.

We performed the required analysis for approximately 200 functions of each type of engine to be controlled by the ECU, and we clarified the control functions that were

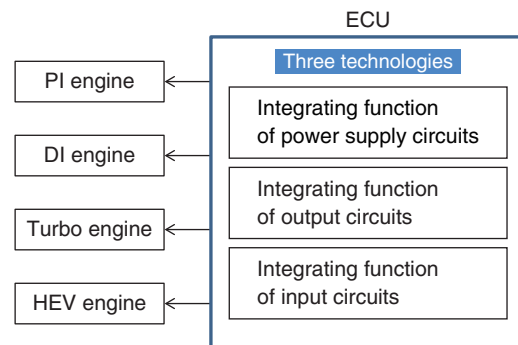


Fig. 4 Application of highly functional ECU

common to each type of engine as well as those that were separate. Based on this, we predicted the processing load and required substrate surface area, selected the size of the central processing unit (CPU), and established target values for substrate size and case size.

We introduce the following three types of technologies shown in Fig. 4 that are used to install the approximate 200 functions: technology that integrates the power supply components, technology that integrates output function parts, and technology that integrates input function parts.

3.1. Integration of Power Supply Components

We utilized a system power supply IC in generating the power supply for the ECU that we developed. In addition to offering the functions of the conventional power supply IC, the system power supply IC also provides CAN, LIN and other communication functions, external hardware input functions, etc. Also, the conventional ECU required a circuit for stabilizing the supplied voltage when the battery voltage varied. However, the new system power supply IC that we developed has a buck-boost function that did not exist in previous power supply ICs, and this function can stabilize the supplied voltage during battery voltage variations. Therefore, we were able to reduce the amount of the conventional circuitry. Figure 5 shows the composition of the system power supply IC that we utilized.

3.2. Integration of Output Function Parts

Most of the circuits for achieving the output functions of the ECU that we developed have been integrated into a semiconductor. In the past there had been the idea of integrating circuits into a semiconductor, and function integration began to move forward. However, the functions that could be integrated were limited, because doing so concentrated generated heat. Due to advances in semiconductor processes in recent years, however, the level of integration was increased and internal resistance was improved, enabling a reduction in generated heat loss and

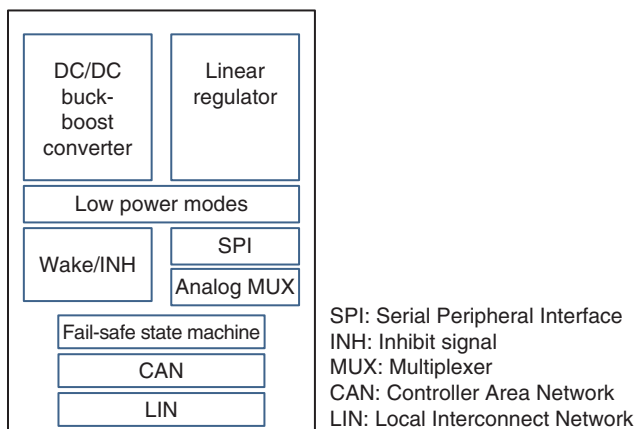


Fig. 5 System power supply IC

more function integration than was previously possible. Figure 6 shows an example of how the output driver IC was integrated. This integration made it possible to simplify the circuitry and reduce the number of parts.

3.3. Integration of Input Function Parts

An example of the reduction of input function parts is the application of the digital knock filter. First, Fig. 7 shows the relationship between the frequency band of the signal required for detecting knocking and the analog digital converter (ADC) sampling frequency. In the conventional CPU, ADC sampling frequency was approximately 100 kHz, close to the frequency band required for detecting knocking, so the use of a high-order analog filter was required. This made the circuit composition complicated and increased the number of parts.

However, as CPUs advanced, ADCs also became capable of high-speed sampling. In the CPU used in the ECU that we developed, sampling was possible at a frequency of approximately 1000 kHz, which was separated at a distance from the frequency band required for knocking, and it became possible to use low-order analog filters. This was how the analog filter composition was simplified and the number of parts reduced.

As described above, integrating the power supply components, output function parts, and input function parts made it possible to reduce the number of parts by 30% over that of the conventional ECU.

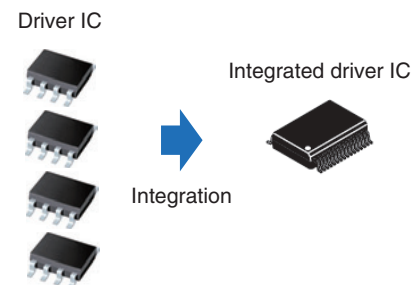


Fig. 6 Integrated driver IC

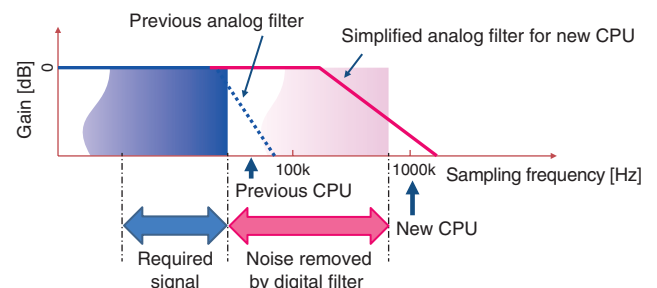


Fig. 7 Analog filter characteristics

4. Direct-injection Injector Drive Circuit

4.1. Reducing Generated Heat

To comply with EURO6, it was necessary to divide the fuel injection quantity that had conventionally been injected only once into multiple stages of injections. Delivering multi-stage injection increases the number of times that the direct-injection injector drives, the quantity of power that is output, and the quantity of heat that is generated. To prevent this from occurring, we introduced a means of decreasing the heat generated by the boost power supply of the direct-injection injector drive circuit.

Every time a direct-injection injector drives, it requires a supply of opening energy from the DC/DC boost converter (boost converter) for the drive voltage. Therefore, in order to deliver multiple injections, it is necessary to decrease the heat that is generated by the boost converter, and the quantity of heat increases in proportion to that energy. In order to calculate the quantity of heat generated by the boost converter, we utilized circuit simulation and optimized the drive circuit composition and components and the switching frequency.

Figure 8 shows an overall diagram of the boost converter utilized in the ECU. The direct-injection injector drive voltage is generated by boosting the input battery voltage. Also, Fig. 9 shows the relationship between the switching frequency and the quantity of heat generated by each element of the boost converter, by providing simulation results and true measurement results for each. Figure 9 shows that when the switching frequency increases, the quantity of heat generated by the inductor decreases, while the heat generated by the semiconductor parts (diodes, FETs) increases. Therefore, a frequency at which the total quantity of heat generated by the entire boost converter is at a minimum can be detected. By utilizing this frequency, the quantity of heat generated by the boost converter was able to be decreased.

4.2. Direct-injection Injector Close Timing Detection

The region of short-time injection required for multi-stage injection was not able to be used until now, because the injection quantity in relation to the drive signal disperses greatly in this region depending on the individual injectors. Therefore, we attempted to accomplish injection in this

previously unusable region by using the ECU to correct the dispersion.

4.2.1. Injector injection quantity and individual dispersion

The relationship between injection quantity Q and T_i , and the dispersion of Q in individual injectors is shown in Fig. 10. Figure 10 clearly shows that the dispersion increases in the region where T_i is short.

4.2.2. Close-timing detection algorithm

The fundamental concept for detecting close timing is as follows: changes occur in the inductance of the injector coil due to movement of the injector core at the close timing, and the resulting inflection point that occurs in the voltage is detected. We therefore added a circuit to the ECU to detect the voltage of the injector.

By processing the voltage signal obtained by the ECU, the voltage inflection point that occurs as a result of closing is detected and is used as the close timing.

4.2.3. Detection results

In order to check detectability, we compared the actual measured values and detected values of the close timing (T_{off}) of 10 injectors with respect to T_i .

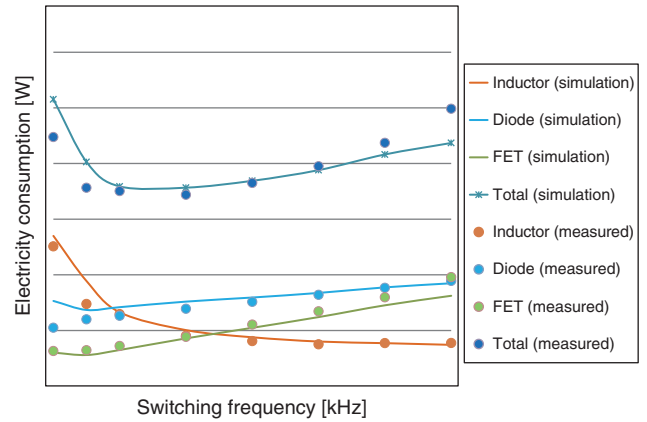


Fig. 9 Relationship between switching frequency and loss of DC/DC boost converter

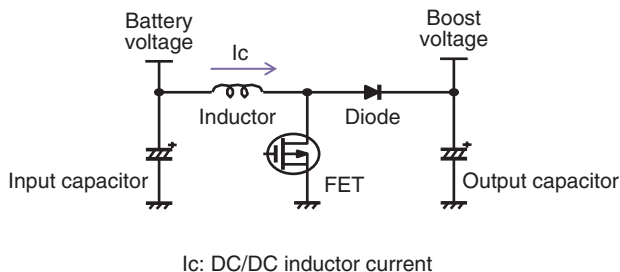


Fig. 8 DC/DC boost converter

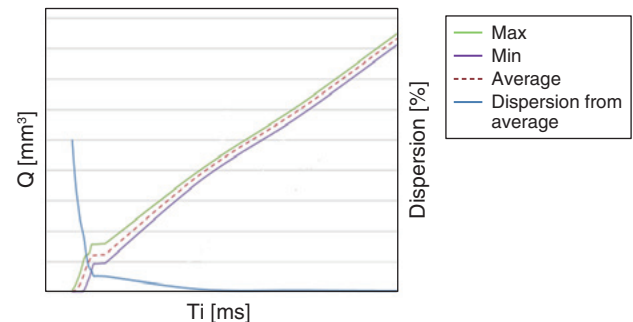


Fig. 10 T_i - Q characteristics

Figure 11 shows a high correlation between the actual measured value and the detected value. As a result, we were able to verify the utility of the close timing detected by the ECU.

4.2.4. Feedback control

Figure 12 shows Ti-Q characteristics when Ti feedback control is performed using the results of the close-timing detection.

Figure 12 also shows a reduction in the dispersion of Q in the region where Ti is short as compared to before control was performed. This control operation was able to decrease MinTi by 46%.

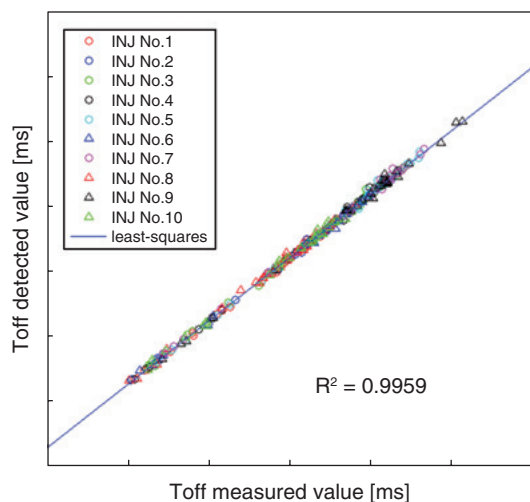


Fig. 11 Comparison of Toff measured value and Toff detected value

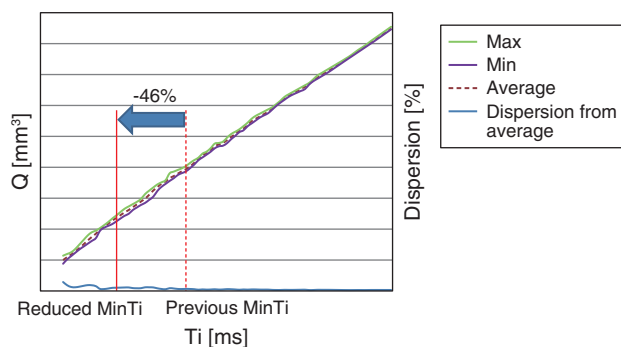


Fig. 12 Ti-Q characteristics after feedback control

5. Conclusion

Amidst the increase in functions for enabling technology required for environmental regulations, we decreased the number of circuit components by utilizing an integrated driver IC, revising low-voltage operation circuits

through the use of a buck-boost power supply, and used microcomputer functions to simplify peripheral circuitry. We also reduced the actual heat generated by the direct-injection injector drive circuit and simplified heat-radiating structures to decrease weight. In addition, we achieved injector control technology that reduces the MinTi in order to enable multi-stage injection of the direct-injection injector that is effective at reducing PM. By developing a highly functional ECU that applies these technologies, we achieved the following results:

- (1) We reduced the number of parts by 30% while assuming applications in future engine types.
- (2) With regard to direct-injection injector control, we reduced the heat generated by the drive circuit, and created a means of reducing the minimum injector drive time value by 46%.

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Author



Makoto MURAKAMI



Tatsuo YAMANAKA



Yasumasa KAITANI

Development of IPU for SPORT HYBRID i-DCD

Norihisa KOBAYASHI* Iori KANAMORI*

ABSTRACT

An Intelligent Power Unit for the 2014 model year FIT SPORT HYBRID i-DCD was developed.

The total height of the Intelligent Power Unit was reduced by the flat installation of its component battery and Power Control Unit, while achieving miniaturization and reduced weight by modifying the frame and cooling duct structure.

By adopting a high-power, large-capacity, lithium-ion battery developed for the SPORT HYBRID, an energy performance 1.5 times greater than that of the conventional model was achieved. In addition, by reducing the volume of the Intelligent Power Unit 23% compared to that of the conventional model, energy density was increased by a factor of two.

Reducing the total height of the Intelligent Power Unit allowed it to be installed in both 2WD and 4WD, which have limited spare tire well volume, minimizing cargo room height and ensuring a sufficient cargo volume for practical use.

1. Introduction

Amid rising crude oil prices and an increasing awareness of global warming and other environmental issues, the sales ratio of fuel-efficient vehicles has increased, and automobile corporations are competing fiercely to develop hybrid technologies.

Previously Honda developed a lightweight, compact, single-motor parallel hybrid system, the Integrated Motor Assist (IMA), which has primarily been installed in compact vehicles. In order to further improve the efficiency of hybrid vehicles, the SPORT HYBRID intelligent Dual-Clutch Drive (SPORT HYBRID i-DCD) was developed and installed in the 2014 model year FIT (new FIT). Because the engine and the motor are separated by the clutch, this system experiences less friction during EV operation than the conventional IMA system. In order to enable EV operation over a broad range, the power and capacity of the battery were increased. This report introduces a high-power, compact Intelligent Power Unit (IPU) suitable for the SPORT HYBRID i-DCD.

2. Development Goals

The conventional IMA system uses the engine as the primary source of power and has superior mountability because it is compact and the motor is directly connected to the engine crank axis⁽¹⁾. In the SPORT HYBRID i-DCD, a single motor is used to ensure compactness, and the engine and motor were separated by integrating the clutch within the transmission. In this way, it became possible to provide the following three operation modes: EV operation in which

the engine is separated and the motor is the only power supply; hybrid operation, which uses the motor to assist the engine; and engine operation, which uses only the engine as a power supply. In order to enable EV operation over a wide range, it was necessary to further improve battery output and capacity. Also, in order to install the IPU in a compact vehicle, it was necessary to decrease its size and weight.

Meanwhile, for the SPORT HYBRID i-DCD, it was also necessary to decrease the total height of the IPU to make the i-DCD suitable for the vehicle body combining hybrid system and 4WD, which is Honda's first, and to ensure cargo room capacity. By developing an IPU capable of being used in both 2WD and 4WD, the number of parts for 4WD was decreased and costs were reduced.

3. IPU Structure

3.1. Appearance

Figure 1 shows the appearance and dimensions of the IPU as well as changes in dimensions compared to the 2011 model year FIT (previous FIT). The height of the IPU was reduced by 20% compared to the previous FIT. Figure 2 shows the changes in the volume and weight of the IPU compared to the previous FIT. The IPU for the new FIT has been reduced in both size and weight.

3.2. Configuration

Figure 3 shows the configuration of the IPU. The IPU comprises a battery module, Power Control Unit (PCU), Junction Board (J/B), Electronic Control Unit (ECU) and cooling fan. There are four battery modules installed in parallel, and the PCU is installed on the side of the battery

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module. Also, by using a flat-installation package in which the J/B is installed on the front of the battery module and a cooling fan is installed on the rear of the PCU, it was possible to minimize the height of the IPU to achieve a

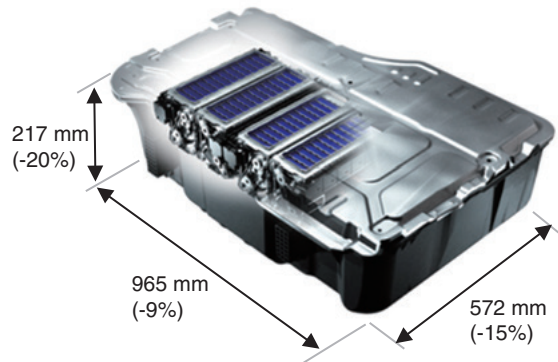


Fig. 1 Dimensions of IPU

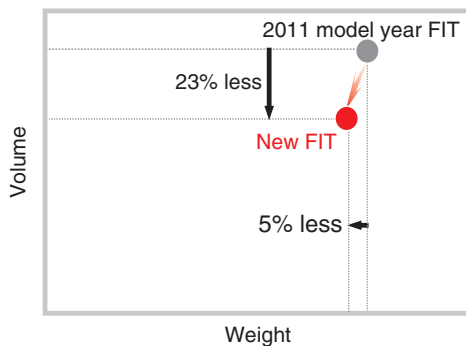


Fig. 2 Volume and weight of IPU

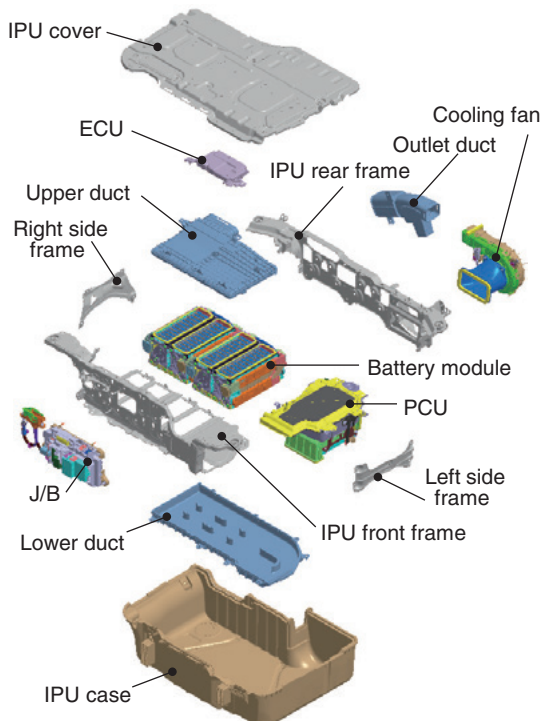


Fig. 3 Configuration of IPU

package with a low total height. Each component is held by two IPU frames that pass horizontally over the front and rear of the module.

3.3. Battery Module

Figure 4 shows the configuration of the battery module. A single module comprises 12 cells stacked in series. This module was also used in the 2014 model year Accord SPORT HYBRID i-MMD⁽²⁾.

3.4. PCU

Figure 5 shows the configuration of the PCU. The PCU comprises an inverter, DC/DC converter, and electric current sensor. Installing the inverter and DC/DC converter vertically allowed the PCU to be installed in the space beside the battery module.

3.5. Frame Structure

Figure 6 shows the frame structure of the previous FIT. The battery module and PCU are attached using the battery frame and PCU frame, and a structure that uses two IPU frames to suspend the battery module and PCU was utilized.

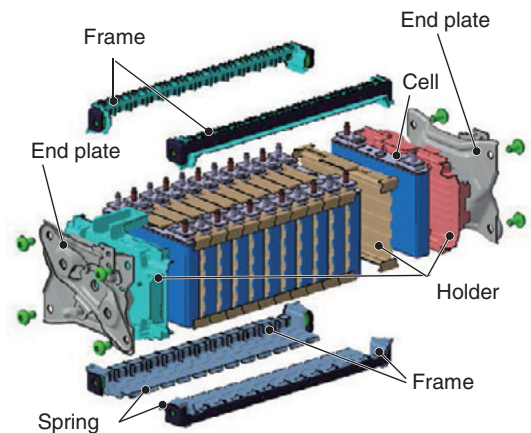


Fig. 4 Configuration of battery module

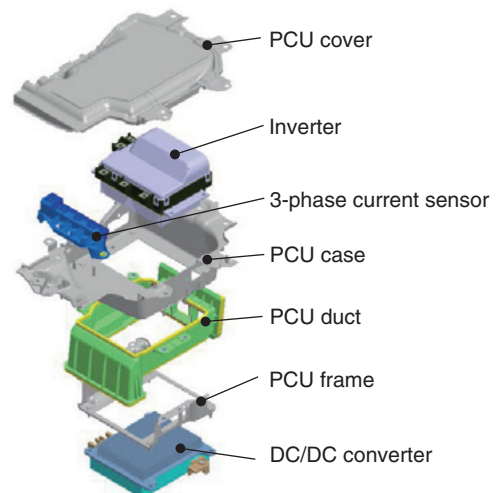


Fig. 5 Configuration of PCU

In contrast to this, the frame structure of the new FIT SPORT HYBRID i-DCD holds the component units with only two frames as shown in Fig. 3: the IPU front frame installed in the front of the module and the IPU rear frame installed in the rear of the module. It was therefore possible to not only reduce the number of parts and the cost but also to simplify assembly. Figure 7 shows the shape of the IPU front frame. The main frame is a press-formed product without joints. Also, a lower patch and an upper patch were welded to both ends of the main frame, the part that attaches to the vehicle, forming a closed section and increasing stiffness. The IPU rear frame has the same structure as that of the front frame.

In addition, a side frame that joins and attaches the front and rear frames was installed on each side of the IPU frame. This side frame minimized vibration by increasing the stiffness of the IPU in the longitudinal direction, thereby improving vehicle steering stability.

3.6. Cooling Structure

The battery module, inverter, and DC/DC converter must be efficiently cooled and controlled to within the optimal operating temperature range.

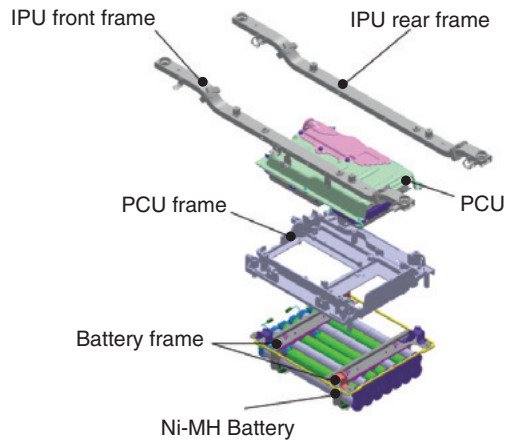


Fig. 6 Frame structure of 2011 model year FIT

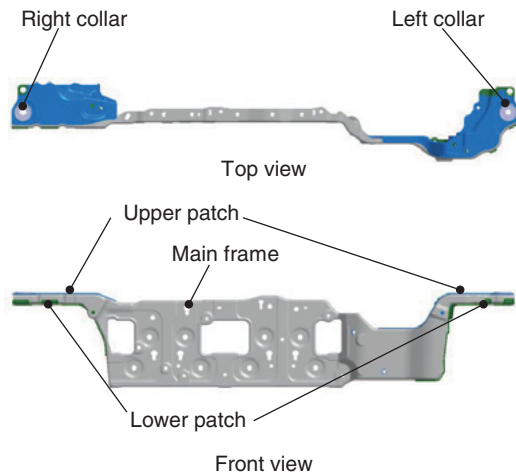


Fig. 7 IPU front frame of new FIT

Figure 8 shows the flow of cooling air. The cooling air taken up from the air-intake grill installed on the right side of the rear seat passes through the air-intake duct placed on the trunk side and flows into the IPU from the rear of the vehicle. Cooling air is evenly distributed to the four battery modules from the upper duct installed on top of the battery. After passing through the battery, the cooling air flows into the lower duct, cooling the inverter and DC/DC converter. Then, it passes through the cooling fan and is expelled into the spare tire well via the outlet duct.

3.7. 4WD Installation

Figure 9 shows the appearance of the IPU for 4WD. In 4WD, the rear differential is installed below the spare tire well, so the tire well is shallower than that for the 2WD. Therefore, by attaching the IPU stay during 4WD installation, the IPU is elevated 65 mm above the installation position of the 2WD. As far as the points in the IPU that were changed in the 4WD compared to the 2WD,

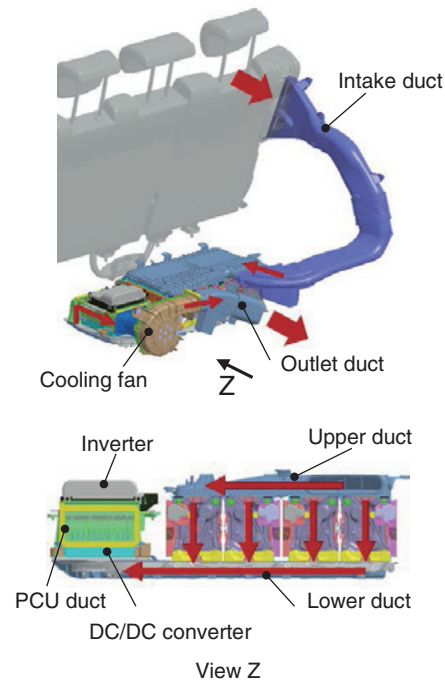


Fig. 8 Cooling air flow

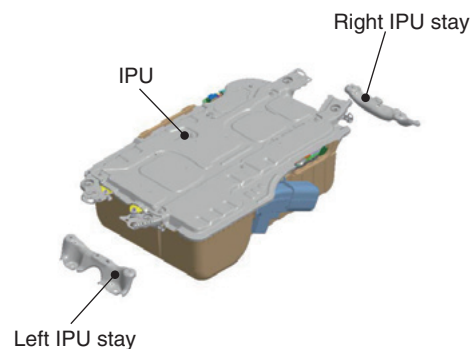


Fig. 9 Configuration of IPU for 4WD

only the position of the collar on both ends of the IPU frame was changed in order to attach the IPU stay, while the other parts are the same as in the 2WD.

3.8. IPU Vehicle Installation

Figure 10 shows the IPU mounted in the vehicle. Adopting an IPU package with a low total height prevents increasing the height of the cargo room in the 4WD. As a result, the height of the trunk is below the lower edge of the trunk rear gate opening, ensuring convenient use of the cargo room.

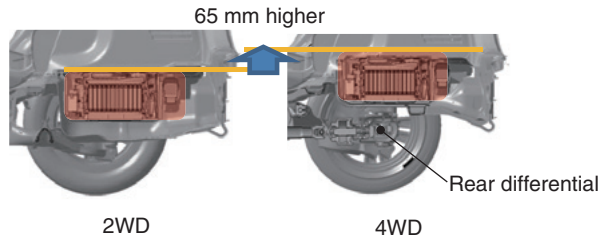


Fig. 10 Mounting of IPU

4. IPU Performance

4.1. Battery Performance

Table 1 shows a comparison of the battery specifications of the previous FIT and the new FIT, and Fig. 11 shows the results of an energy comparison. In the new FIT, the separation of the engine and the motor by the clutch allows EV travelling over a broad range as well as increased regeneration energy during deceleration. As a result, the new FIT requires more battery output and energy than the conventional FIT. While a Ni-MH battery was used in the previous FIT, a Li-ion battery is used in the new FIT, increasing rated voltage and system output to increase energy performance by approximately 1.5 times.

4.2. Cooling Performance

Figure 12 shows the results of a comparison of the previous and new FIT models in terms of battery cooling capacity in relation to battery-generated heat in the LA4 and US06 representative operation modes. Note that the maximum quantity of cooling air is established by considering NV during cooling FAN operation.

In the new FIT, the cell cooling capacity improvement due to the adoption of a square cell and the pressure loss reduction due to optimization of intake and outlet duct shape and cooling air flow path inside the IPU improved the cooling capacity performance by approximately 1.5 times over that of the previous FIT. As a result, it became possible to decrease the output limitation frequency by increasing the temperature of the battery in US06 mode, which is the high-heat-generating battery mode. Also, it is predicted that the cooling capacity will be capable of adequately supporting derivative models in which this IPU is planned to be installed.

Table 1 Specifications of battery

	2011 model year FIT	New FIT
Battery type	Ni-MH	Li-ion
Voltage	100 V	173 V
Output	10 kW	22 kW

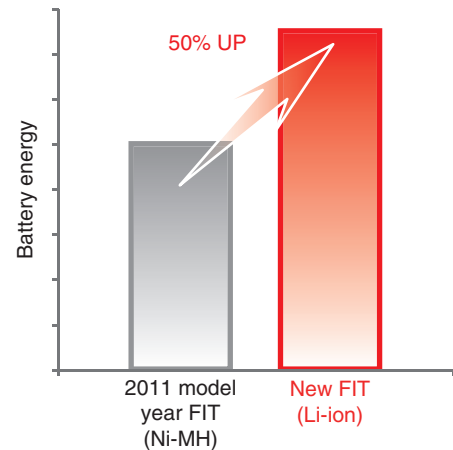


Fig. 11 Comparison of battery energy

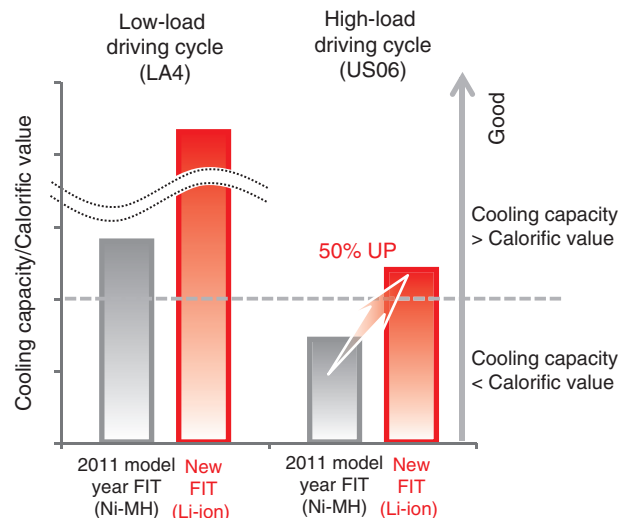


Fig. 12 Cooling capacity

5. Conclusion

An IPU for the 2014 model year FIT SPORT HYBRID i-DCD was developed. By utilizing an IPU package with a low total height, the IPU was made capable of being used in both 2WD and 4WD, while ensuring sufficient cargo room space even in 4WD.

In addition, using a lithium-ion battery improved energy performance by 1.5 times, and reducing the volume of the IPU by 23% allowed a two-fold increase in energy density.

The newly developed IPU will be installed not only in the new FIT, but also in the 2014 model year VEZEL.

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■ Author ■



Norihisa KOBAYASHI



Iori KANAMORI

Development of 7-speed Dual Clutch Transmission for SPORT HYBRID i-DCD

Norikazu KAWAI*
Hiromichi KONISHI*

Koji OHTA*
Heisuke KOBAYASHI*

Yasuyuki MASAOKA*

ABSTRACT

A 7-speed dual-clutch transmission was developed for SPORT HYBRID i-DCD to achieve the aims of compactness and efficiency. For compactness, the motor was built into the transmission, and the planetary gear that makes up first gear was located on the interior side of the motor. This created a lightweight and compact transmission structure with one motor, which was optimized for small-sized vehicles like the FIT. Maximum input torque of up to 190 Nm is supported. For efficiency, the structure of a manual transmission capable of high transmission efficiency is used as the base and its efficiency enhanced by using an electric actuator and dry clutch. In terms of driving, the transmission has two clutches, one for odd-numbered gears and one for even-numbered ones, and the next gear stage up from the current one is put in standby in advance. Alternating between the two clutches while shifting through the gears reduces the time lag when shifting so that power transmission is seamless, and at the same time it achieves acceleration that feels direct, as with a manual transmission.

1. Introduction

Globally, initiatives in recent years have sought to protect the environment and lower burdens on it.

The automobile industry is under pressure to improve fuel economy, which has led to the introduction of a variety of technologies. These include technologies to boost engine efficiency, lower drive system friction, reduce vehicle cruising resistance, and use an idling-stop function, as well as hybrid, electric, and fuel cell vehicles.

Honda developed the Integrated Motor Assist (IMA) system and applied it to the INSIGHT in 1999. Since then, it has expanded the system to the CIVIC, Accord, and FIT⁽¹⁾. The CIVIC Hybrid sought to increase fuel economy with the Variable Valve Timing and Lift Electronic Control (VTEC) valvetrain system that increases electric energy with deceleration regeneration⁽²⁾.

To achieve greater fuel economy increases than the existing IMA system, Honda developed the SPORT HYBRID Intelligent Dual-Clutch Drive (SPORT HYBRID i-DCD). An inexpensive system that could be used on small-sized vehicles was needed to promote hybrid vehicles and thus lower environmental burden. To achieve this, Honda developed this system as a hybrid system with one motor and used it on the third-generation FIT. By

combining a 7-speed dual-clutch transmission with built-in high-output motor and a lithium-ion battery with the newly developed in-line four-cylinder 1.5 liter engine, Honda achieved an efficiency boost of more than 30% compared to a conventional one-motor hybrid system⁽³⁾.

Increasing fuel economy requires improving operations wherein energy is converted to waste heat, and it is important to increase electric energy during deceleration regeneration and reduce energy loss during takeoff.

Hybrid vehicles use regenerative brakes during deceleration, turning kinetic energy into regenerated electric power that is recovered and can be used to increase fuel economy. In conventional IMA systems, the engine and motor cannot be separated from each other. As a result, the electric power regenerated during deceleration is wasted to the extent that there is engine friction. Moreover, takeoff with a conventional clutch is accomplished by sliding the clutch, so there is loss that occurs as thermal energy. The newly developed system uses the motor for takeoff, which reduces energy loss as compared to takeoff with conventional clutches and torque converters. It also separates the engine and motor, which reduces loss during EV driving and during deceleration regeneration.

This paper explains the transmission of the SPORT HYBRID i-DCD system.

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2. Aim of Development

The purpose of developing a 7-speed dual-clutch transmission (DCT) for SPORT HYBRID i-DCD was to build a one-motor hybrid electric vehicle (HEV) transmission small enough to go on a model with an in-line four-cylinder engine. Specific development aims were as follows.

- (1) Enable EV driving and deceleration regeneration with the engine separated.
- (2) Make the transmission lightweight and compact enough to go on the FIT class.
- (3) High transmission efficiency

3. Basic Layout and Major Specifications

The main sectional view and perspective view of the 7-speed DCT for SPORT HYBRID i-DCD are shown in Figs. 1 and 2, respectively, and the specifications are given in Table 1.

The transmission uses a dual dry clutch (DDC), which is operated by an electric actuator. First gear consists of a planetary gear located on the interior side of the motor. Odd-numbered stages are located on the main shaft and even-numbered stages on the secondary shaft, and these share the driven gear on the counter shaft. The actions of the two clutches cause the gears to shift, and also cause the engine and motor to engage or disengage. EV driving and deceleration regeneration are enabled with the engine separated.

Table 1 Specifications of FIT

Engine	Type	In-line 4 1.5 L Port injection
	Max. power	81 kW (6000 rpm)
	Max. torque	134 Nm (5000 rpm)
Motor	Max. power	22 kW
	Max. torque	160 Nm
Transmission	Type	7-speed DCT
	Size	420.8 mm
	Weight	70.6 kg (Without motor parts)
	Gear ratio	1st 4.148
		2nd 2.007
		3rd 1.481
		4th 1.098
		5th 0.810
		6th 0.605
		7th 0.446
		Reverse 3.221
		Final 4.842

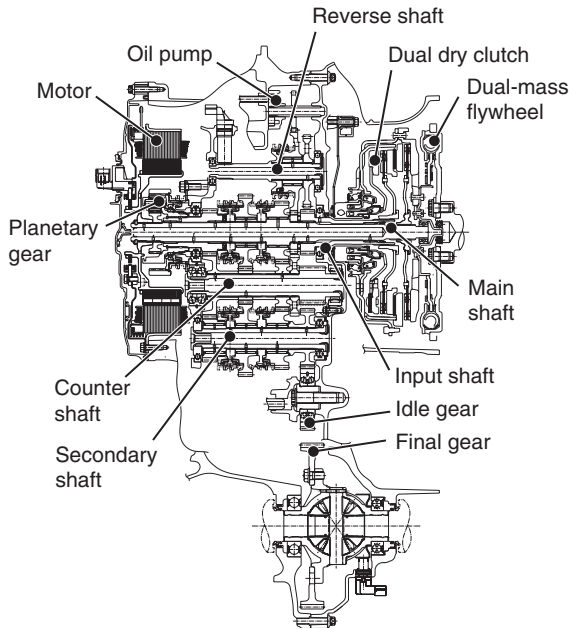


Fig. 1 Main sectional view of developed DCT

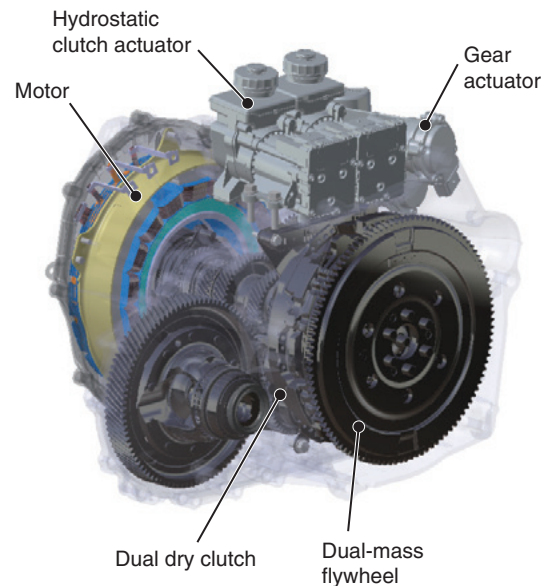


Fig. 2 Perspective view of developed DCT

4. Major Technologies

4.1. EV Driving and Deceleration Regeneration with the Engine Separated

The motor is installed on the main shaft, and the engine and motor are separated by disconnecting the dual clutch. Examples of the action of the DCT are given in Fig. 3.

When the motor connected to the main shaft and the sun gear of the planetary gear cause the transmission to enter first gear, the ring gear becomes fixed. EV driving in first gear is enabled through third gear, which is connected to the carrier. The ring gear is released and the transmission goes into third gear, enabling EV driving in third gear. EV driving and deceleration regeneration can only be accomplished through odd-numbered-stage gears (EV drive mode and EV regeneration mode).

From EV driving, it is possible to start the engine and go into motor-assisted driving by engaging the clutch for odd-numbered stages or the clutch for even-numbered stages through any of the even-numbered stages. Even if driving in an even-numbered stage without a connection to the motor, it is possible to get assistance from the odd-numbered stage side, which is connected to the motor (hybrid mode).

It is also possible to drive without even using motor assist (engine drive mode).

Additionally, by shifting to neutral and connecting the odd-numbered-stage clutch, thus separating the drive to the tires, it is possible to connect the engine and motor and recharge with the car parked.

It is possible not only to shift the DCT clutch, but also

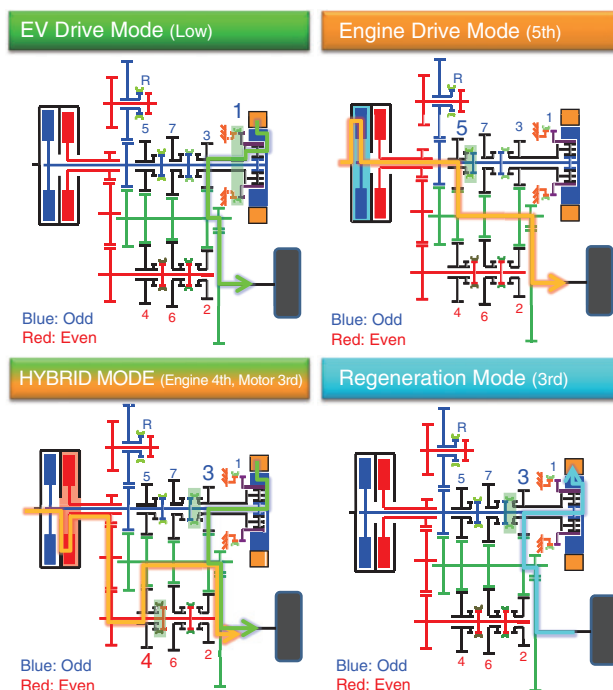


Fig. 3 DCT operation modes

to use engaging or disengaging of the engine and motor. This enables EV driving and deceleration regeneration with the engine separated, even in a lightweight and compact structure. Combining a gear-shift stage and clutch enables motor-assisted driving in each shifting stage, helping to increase the efficiency of the SPORT HYBRID i-DCD system. Additionally, the layout of the odd-numbered gears that perform motor driving and regeneration allows highly efficient motor driving and deceleration regeneration with minimum numbers of gear engagement.

4.2. Lightweight and Compact Design

A transmission has to be lightweight and have a short overall length to be mounted on an in-line four-cylinder engine. The following correspondence was made to make this possible. The gear structure is shown in Fig. 4.

- (1) The IMA motor was built into the transmission and the motor housing was eliminated. In addition, the motor is cooled with automatic transmission fluid (ATF) in order to lower the amount of heat radiated. These steps allow the motor to be more compact.
- (2) First gear consists of a planetary gear, the sun gear is connected to the main shaft, and the ring gear is fixed,

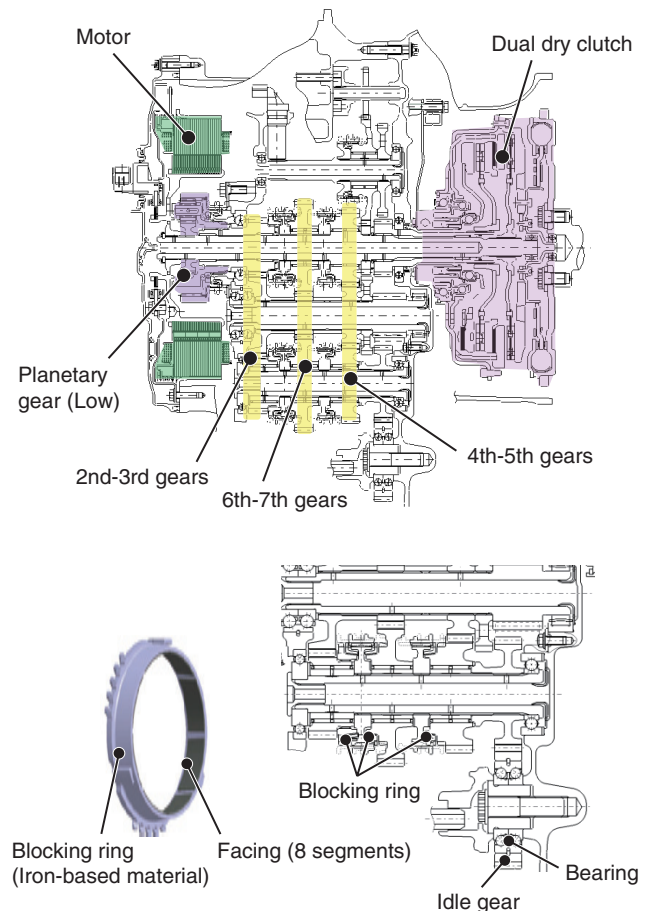


Fig. 4 Structure of gear train

thus forming first gear. The carrier's output is shared with third gear. Placing the motor on the outer periphery of the planetary gear shortens the transmission's overall length. The carrier is made of aluminum, which reduces weight.

- (3) Odd-numbered-stage gears and the motor are located on the main shaft. Even-numbered-stage gears are located on the secondary shaft. Power to the secondary shaft is transmitted from the input shaft via the idle shaft. The trains for second and third gear, fourth and fifth gear, and sixth and seventh gear share the driven gear located on the counter shaft. This reduces the overall length.
- (4) A high-capacity synchronizer was developed that is compatible with ATF. The high-capacity design makes the synchronizer more compact and lets it go under the gears, reducing overall length. The base material was switched from hot-forged brass to press-formed steel, and the friction material switched from powder molded to sheet molded. This ensured capacity and durability when exposed to ATF and helped to cut costs and reduce waste during production.
- (5) Small-diameter, high-capacity four-point contact bearings were used, making it possible to reduce the space between shafts. Making the bearings smaller also helps reduce friction torque, contributing to efficiency.
- (6) The idle gear has a rigid cantilever support structure, and the bearings are located below the gears, making it possible to increase the diameter of the bearings. This ensures rigidity while reducing the space between shafts, enabling a smaller and lighter transmission. Additionally, aligning the final gear with the direction of the shaft reduced the overall length. Similarly, aligning the reverse gear and parking gear with each other helped to make the transmission smaller and lighter.
- (7) Aluminum bolts are used to fasten the clutch case (C case) and transmission case (M case) to each other and to fasten the M case and side cover to each other. Specific gravity of the aluminum bolts is only about one-third that of conventional steel bolts, but aluminum bolts are also weaker, having only about one-half the tensile strength. The design process considered the following aspects to make up for the loss of tensile strength. The coefficient of thermal expansion of aluminum bolts is equal to that of the cases of the members being fastened, so the increase in axial force during heat treatment is an advantage over steel bolts. In addition, using an angle tightening method controls axial force variance when fastening, while surface treatment stabilizes the torque coefficient and minimizes changes in it when liquid packing or oil adhere to the components. As an overall result, the aluminum bolts achieved the same level of residual axial force as conventional steel bolts, taking variance into account. It was therefore possible to satisfy the fastening conditions with the same number of aluminum bolts as steel bolts. The locations where members were fastened with aluminum

bolts are shown in Fig. 5. The total weight of bolts used for case fastening was reduced by about 300 g by using aluminum for the 17 M8 bolts fastening the M case and C case to each other and the nine M6 bolts fastening the M case and side cover to each other.

- (8) For the motor's rotor, the M case was given a cantilever support structure. This eliminated the need for bearing support, so resin was used for the side cover. The structure of the side cover is shown in Fig. 6.

The seal structure of the connector area of the resolver, which senses motor rotation, consists of an annular spring (providing tightening force), a rubber sealing member, and a resin ring.

The resin side cover undergoes radial and thrust displacement owing to heat and absorbed water. The sealing member is shaped to allow such displacement. Additionally, using a ring made of resin made it possible to weld it to the resin cover, reducing both cost and the number of parts.

The oil guide, carrying oil from the oil pump, was welded in the same fashion as the seal area to create a single piece. The oil put out by the oil pump lubricates the main shaft interior, the planetary gear, and the

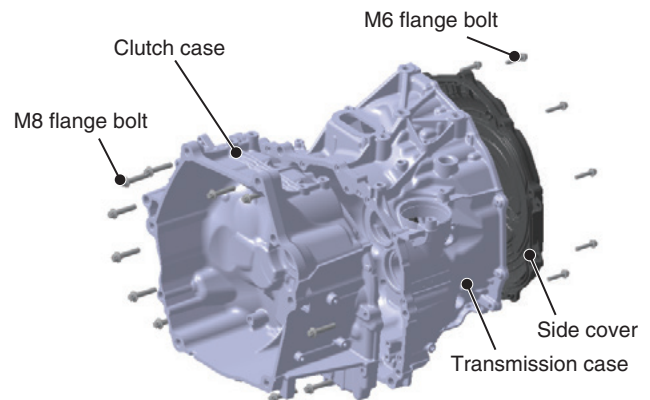


Fig. 5 Locations of aluminum bolts

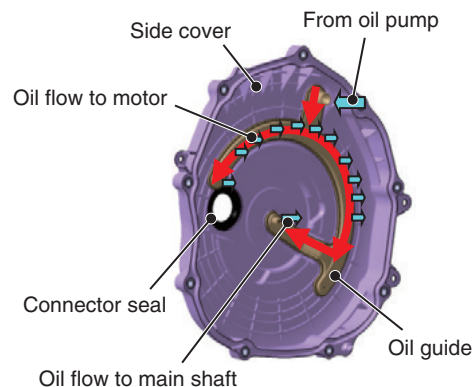


Fig. 6 Structure of side cover

pinions, while cooling the motor winding through the resin cover's oil guide. Using resin as the material made it unnecessary to provide an insulating distance from the motor. This fact, along with the layout freedom allowed by resin molding, enabled an oil guide to be constructed in a place where the motor could be cooled efficiently. This contributed to making the motor smaller and enabled lightening and shortening of the overall transmission length.

As a result of the above, the transmission's overall length is equal to that of the existing IMA system (motor and CVT), and is lightweight and compact enough to go on the FIT series.

4.3. High Transmission Efficiency

The newly developed transmission is based on the more efficient manual transmission (MT) technology. The clutches and gears are divided into two systems (odd-numbered and even-numbered stages), and the transmission switches between the two systems as it shifts through the gears.

For the clutch, which engages and disengages with the engine, a dry clutch is used to reduce drag. The clutch consists of LuK's DDC and a hydrostatic clutch actuator (HCA) that generates and controls clutch hydraulic oil pressure. The HCA is driven by an electric motor. The gear is chosen when the gear change actuator (GA) causes the shift fork to shift and select a gear. The configuration of the GA is shown in Fig. 7.

When the GA acts in the direction of the shaft, a shift rail is selected, then depending on the direction of rotation, the shift fork's rail is moved and the shift action is accomplished. Parking action is also accomplished by the action of the GA. The HCA and GA both drive an electric motor, which eliminates the need for an oil pump to generate hydraulic pressure and increases efficiency. This

also enables the use of shift-by-wire, eliminating the need for shift to the auto body side and for a select cable. This increases the freedom to locate the shifter part, helping to expand design potential.

Low-viscosity ATF is used as the transmission's lubricating oil, which reduces friction. Because conventional MTs use manual transmission fluid, a high-capacity synchronizer and shift fork were developed to work with ATF in this transmission.

A polyamide-based resin was used for the tips of the shift fork to ensure abrasion resistance. The structure of the shift fork is shown in Fig. 8. Molding resin tips on a steel shift fork as a single unit achieved abrasion resistance while also reducing the number of processes and lowering the cost as compared to conventional hard chrome plating.

The lubrication structure is shown in Fig. 9. To reduce friction, baffle plates were put on the final gear and secondary shaft to reduce agitation. The gutter plate allows efficient lubrication even with a low fluid surface to ensure that the bearings of the differential are efficiently lubricated by the final gear as it scoops up oil.

The main shaft, motor, and so on are lubricated by the

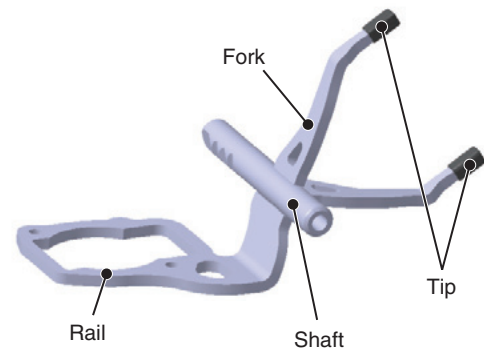


Fig. 8 Shift fork

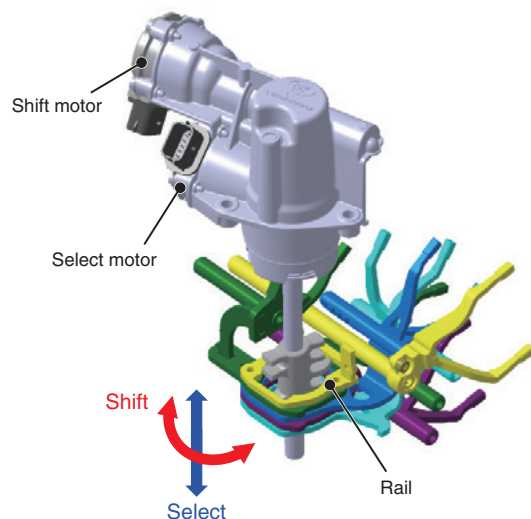


Fig. 7 Configuration of GA

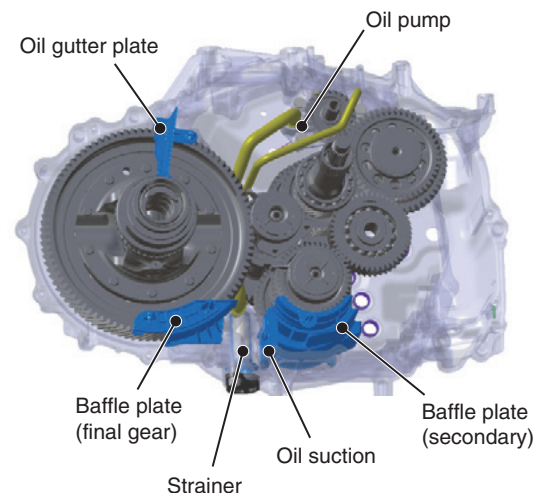


Fig. 9 Lubricating system

oil pump. However, if the amount of oil is reduced, air could be sucked in from the oil inlet if the fluid surface tilts when turning or on a hill. The location of the strainer is limited by the layout, so it cannot be placed in an appropriate inlet position when the inlet area is on the case. If the amount of oil is increased to prevent air from being sucked in through the oil inlet, agitation resistance and weight are increased as well. Therefore, this inlet was made a single piece with the secondary baffle, optimizing the oil inlet location without adding any parts. This made it possible to reduce the amount of oil and agitation resistance.

These measures increased transmission efficiency by about 5% compared to a conventional AT.

5. Conclusion

A 7-speed DCT with the following characteristics was developed as a SPORT HYBRID i-DCD transmission.

- (1) By using a dual clutch and locating the motor on the main shaft, EV driving and deceleration regeneration are enabled with the engine separated.
- (2) By locating the planetary gear that comprises first gear on the interior side of the motor, by using a shared gear train structure, etc., and by using alternative materials, a smaller and lighter transmission was achieved.
- (3) By using a dry clutch and electric actuator and reducing oil agitation, high transmission efficiency was achieved.

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■ Author ■



Norikazu KAWAI



Koji OHTA



Yasuyuki MASAOKA



Hiromichi KONISHI



Heisuke KOBAYASHI

Study of High-efficiency CVT Shifting Mechanism Using Metal Pushing V-belt

Shuji ICHIJO*

Tooru YAGASAKI*

Taiki ANDO*

Kyohei SAKAGAMI*

Soichiro SUMIDA*

ABSTRACT

Pulley thrust control, changes in the trajectory of the belt as it winds around the pulleys, and the amount of friction transmission were focused on in order to reduce transmission loss and increase the transmission efficiency of CVT. In the case of pulley thrust control, making use of the linear relationship between the rotary speed fluctuation transfer characteristic and the torque transmission capacity between the pulleys and the belt, it was possible to reduce the excess safety factor of the torque transmission volume. Due to pulley tilt, the trajectory of the belt displays deviations with the theoretical geometrical winding radius. The structure of the pulleys was modified in order to reduce this deviation and increase transmission efficiency. Optimization of the additives in the CVT fluid increased the coefficient of friction, decreasing pulley thrust and increasing transmission efficiency. At the same time, the application of a carbonitride heat treatment and a zinc electric discharge coating to help prevent wear in the parts of the belt elements and pulleys making contact balanced a high coefficient of friction with wear resistance.

1. Introduction

The use of hybrid systems and CVT using metal pushing V-belts in automotive powertrains is becoming more widespread today from considerations of increasing fuel efficiency and reducing exhaust emissions. However, it has been indicated that increasing transmission efficiency during high-speed cruising is an issue for CVT using metal pushing V-belts⁽¹⁾. The main factor in transmission loss in metal pushing V-belts is thought to be friction loss due to slipping. If this friction loss could be reduced, it would be possible to achieve further increases in fuel efficiency. In addition, reports indicate that it is possible to increase the efficiency of the belts by optimizing pulley thrust⁽²⁾, and that deviation of the winding diameter of the belt from the theoretical winding diameter is a factor in reducing belt efficiency⁽³⁾, suggesting that there is potential for increasing the efficiency of metal pushing V-belts. The research discussed in this paper focused on friction loss due to slipping in metal pushing V-belts, and in an attempt to increase transmission efficiency, studied a configuration that would reduce pulley thrust and help to control pulley tilt. This process produced findings regarding modifications that would help enable metal pushing V-belts to display sufficient levels of performance. This paper presents an overview of the research.

2. Development Goals

Metal pushing V-belts are made up of two sets of multi-layered rings and several hundred elements (Figs. 1 and 2). Running between a fixed pulley and a movable pulley, they transmit drive force by means of friction of the V-surfaces. While transmitting drive force, the belts are affected by the clearance between the elements, the elastic deformation of the elements, and the inclination of the pulley V-surfaces, resulting in micro slip between the pulleys and the elements in the direction of rotation and the radial direction. In addition, because the element pitch radius and the ring pitch radius are different, a

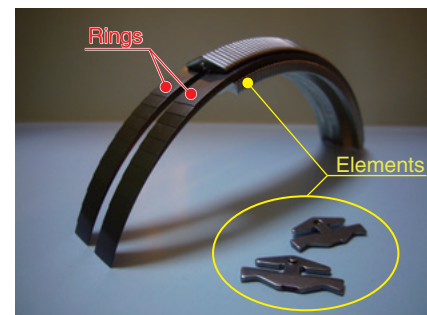


Fig. 1 Metal pushing V-belt

* Automobile R&D Center

difference in path is generated between the elements and the rings, and with the rotation of the pulleys, this results in slippage. Similarly, because the pitch radius of the rings differs due to their thickness, differences in path result in slippage between the layered rings.

Reducing ring tension, which generates a contact load, is an effective way of reducing friction loss due to slippage between the rings, and ring tension can be reduced by reducing pulley thrust. In addition, because slippage between the elements and the pulleys in the radial direction results from misalignment of the belt trajectory due to the tilt of the pulley V-surfaces, the slippage of the elements in the radial direction could be reduced by controlling the tilt of the pulley V-surfaces. The research discussed in this paper sought to increase the unit efficiency of the belt in the OD ratio, which is frequently used during cruising in actual vehicles. Seeking to realize a 3% increase in the unit efficiency of the belt, the project conducted research on a new pulley thrust control method, increasing the coefficient of friction, and controlling the tilt of the pulley V-surfaces. Here, the rate of increase in transmission efficiency will be defined as the difference between the efficiency of the conventional technology and the developed technology with the unit efficiency of the belt when transmission loss is zero as 100%.

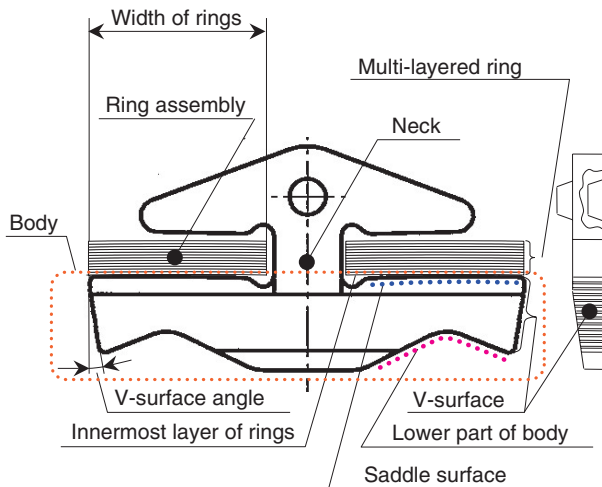


Fig. 2 Configuration of elements and ring assembly

3. Methods of Increasing Transmission Efficiency

Low pulley thrust determines the transmission volume and high pulley thrust determines the shift ratio, and depending on driving conditions, they can be applied to either the input or the output pulley. Low pulley thrust consists of the thrust that is necessary for torque transmission, as determined by the angle of the V-surfaces,

the winding radius, and the coefficient of friction, in addition to a level of thrust that considers the safety factor in view of belt slip. High-efficiency operation can be realized by reducing this safety factor to the minimum value. Conventional control considers variations in control hydraulic pressure against estimated input torque and calculates pulley thrust with consideration of the safety factor using a correction coefficient, in order to impart a pulley thrust that is more securely on the safe side against belt slip during operation. The new control method detects the margin against belt slip and controls pulley thrust with a desired margin as a target value, making it possible to operate the belt with the minimum necessary pulley thrust. In order to increase the coefficient of friction between the pulley and the belt, additives were employed to modify the characteristics of the oil. At the same time, a zinc electric discharge coating treatment was applied to the pulley V-surfaces in order to help control the increased wear that would result from an increase in the coefficient of friction. The trajectory of the belt when winding around the pulleys deviates from the theoretical winding radius due to the inclination of the V-surfaces. It was considered that because the configuration of the conventional movable pulley was such that the majority of pulley thrust was concentrated on the inner periphery, when the belt-winding radius was tending towards larger diameters, the inclination of the V-surfaces would increase. The configuration of the pistons was modified and the balance of pulley thrust on the inner and outer peripheries of the pulleys was optimized, helping to control the inclination of the pulleys and reduce the amount of deviation of the belt trajectory.

3.1. New Pulley Thrust Control

With the torque ratio, T_r , the ratio between input torque, T , and the maximum amount of torque able to be transmitted, T_{max} , as an indicator, a coefficient of correction was determined from the difference between the torque ratio set as a control target value and the actual torque ratio measured in the vehicle, and a new control method that determines pulley thrust based on this coefficient was developed. The transmission efficiency of metal pushing V-belts displays a linear relationship with the torque ratio, and an increase in the torque ratio results in an increase in transmission efficiency, with an associated increase in micro slip. Efficiency is at its maximum when the torque ratio is close to 1; when the ratio exceeds 1, significant slippage that damages the belt occurs, and efficiency declines (Fig. 3). With an increase in the torque ratio, fluctuations in engine speed produce an increase in micro slip, resulting in damping of vibration energy and phase lag, which are transmitted by the belt. Using this relationship between the torque ratio and the transmission characteristic, it is possible to detect the torque ratio during operation and control it to the desired level.

3.1.1. Detection of torque ratio

Making use of the fact that engine speed fluctuates during the ignition cycle, torque ratio detection determines the power spectrum ratio or the phase difference from the speed fluctuation of the input shaft and the output shaft, and uses the correlation between the transmission characteristic and the torque ratio to determine the torque ratio. The present research used existing engine speed sensors, which powertrains are provided with for monitoring of engine speed in actual vehicles, to detect engine speed. The speed of the input and output shafts was measured using the output of the engine crankshaft speed sensor for the detection of the speed of the input shaft, and using the output of the driven pulley shaft speed sensor for the detection of the speed of the output shaft (Fig. 4). The basic frequency of engine speed changes was calculated from the ignition cycle, and the input shaft speed signal and the output shaft speed signal were used, respectively, to determine the power spectrum ratio and the phase difference using a two-phase lock-in amplifier (Fig. 5). The relationships between the torque ratio and the power spectrum ratio and between the torque ratio and the phase difference were studied experimentally, and it was found that phase difference increases linearly with increase in the torque ratio across the entire input frequency range. By contrast, the power spectrum ratio does not increase linearly with increase in

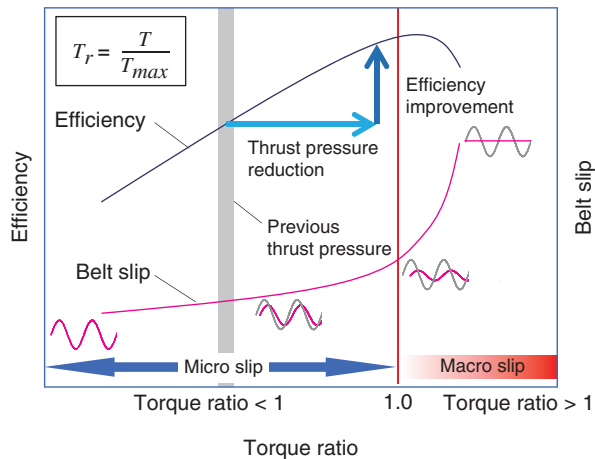


Fig. 3 Relationship between torque ratio and efficiency

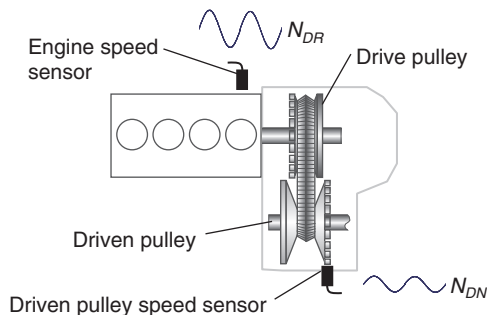


Fig. 4 Rotary speed detection system

the torque ratio at input frequencies of 55 Hz and below. The results discussed above indicated that phase difference was the appropriate parameter for the detection of the torque ratio (Fig. 6). A three-dimensional map of phase difference, frequency, and torque ratio was formulated for use in vehicle control (Fig. 7).

3.1.2. Application of new pulley thrust control in vehicle

The system switches from the conventional control specifications to new pulley thrust control under the

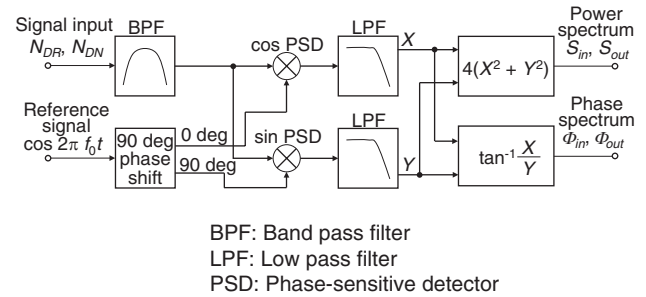


Fig. 5 Two-phase lock-in amplifier curve

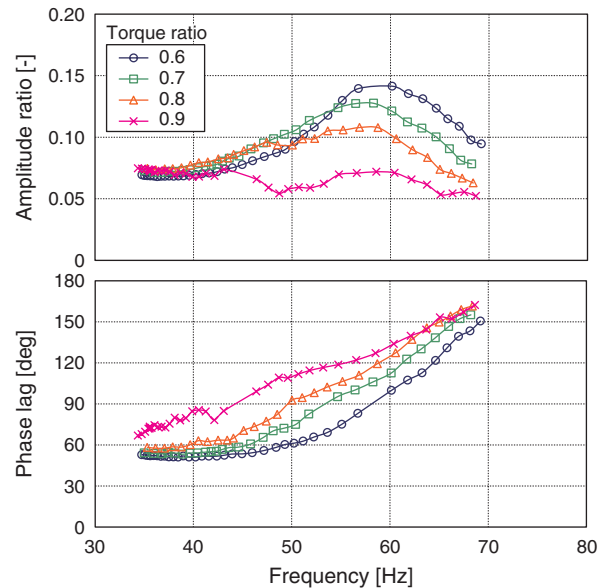


Fig. 6 Amplitude ratio and phase delay

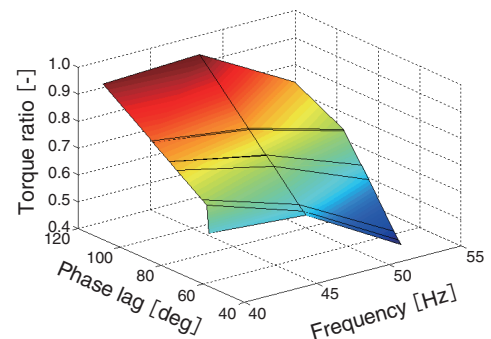


Fig. 7 3D torque ratio map

condition that has been preset for switching to the new control; under other conditions the system returns to conventional control. The condition for application of the new control is when the belt shift ratio is the OD ratio, which is frequently used when a vehicle is cruising. The calculation of pulley thrust in the conventional control method considers belt slip and adds a certain amount of pulley thrust to the pulley thrust necessary for torque transmission as a safety margin. The new pulley thrust control method determines the actual torque ratio from the running state of the vehicle and the three-dimensional map, and calculates a coefficient of correction for the margin from the discrepancy between the target torque ratio and the actual torque ratio. When it has switched from conventional control to new pulley thrust control, the system substitutes the corrected margin for the conventional control margin and calculates pulley thrust. The application of new pulley thrust control has increased the torque ratio from 0.45 to 0.7 and reduced pulley thrust by approximately 40% against the conventional control method when cruising in OD ratio (Fig. 8).

3.2. Method of Increasing Coefficient of Friction

The transmission of drive force between the pulleys and the elements occurs as a result of friction force generated by shear of the boundary film, which is a reaction film produced by additives in the CVT fluid (Fig. 9). In order to increase the coefficient of friction, it is necessary to increase the shear strength of this boundary film. The research experimented with additives with the aim of increasing

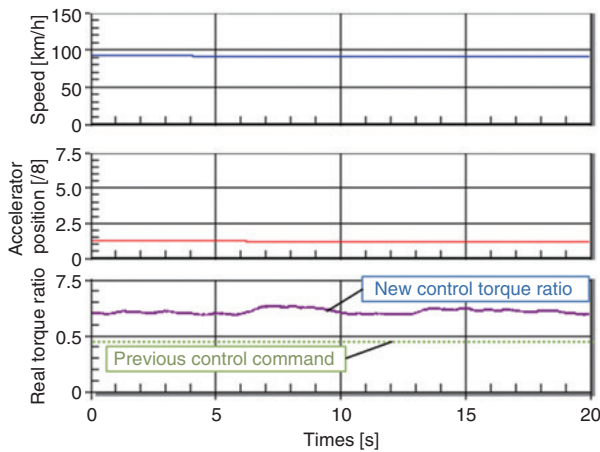


Fig. 8 Test results for new pulley thrust control

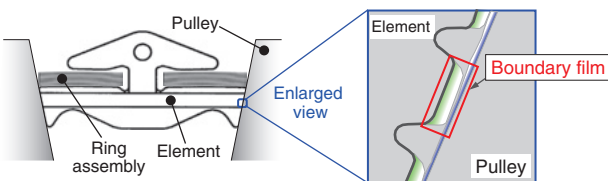


Fig. 9 Principle of frictional force

the coefficient of friction in order to realize a reduction in pulley thrust of 10% against the conventional value.

Sliding tests were conducted using CVT fluids of differing coefficients of friction, and the metallic structures of the sliding parts were examined. It was found that the depth of the micro-crystallizing metal structure close to the surface of the sliding parts increased with an increased coefficient of friction (Fig. 10). In addition, cracks appeared at the boundary between the normal metal structure and the micro-crystallizing metal structure, and peeling occurred (Fig. 11). When increasing the coefficient of friction, it would therefore be necessary to control the generation of a micro-crystallizing metal structure and attempt to prevent peeling wear.

3.2.1. Formulation of additive

The coefficient of friction between the pulleys and the elements is affected by the belt shift ratio and temperature. With regard to the temperature characteristic, the temperature at which the friction-modifying agent (FM agent) in the CVT fluid is activated differs depending on the type of FM agent used. The research therefore selected the FM agent and adjusted ratios in order to minimize the reduction in the coefficient of friction at low temperatures. With a phosphorous-based extreme-pressure agent as the main ingredient, a cleaning agent and a dispersant were added in order to produce an FM agent created with consideration of belt noise and the clutch characteristic. The coefficient of friction at the temperature used as the

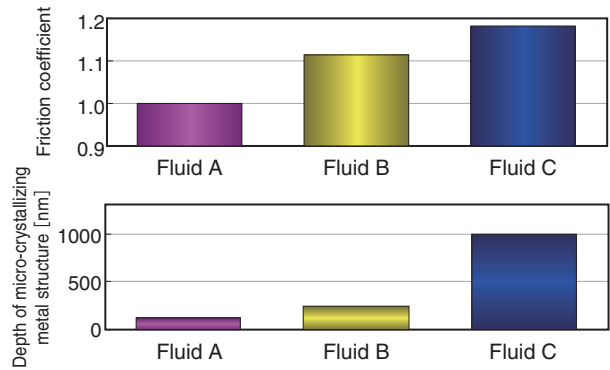


Fig. 10 Friction coefficient and depth of micro-crystallizing metal structure

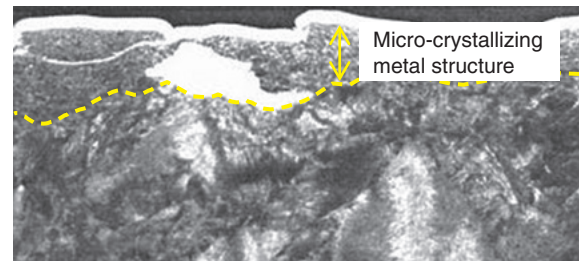


Fig. 11 Micro-crystallizing metal structure resulting from sliding friction

benchmark for evaluation was 18.1% higher than for the conventional CVT fluid in LOW ratio, and 10.8% higher in OD ratio (Fig. 12).

3.2.2. Attempting to reduce pulley V-surface wear

It was considered that because a correlation exists between the value of the coefficient of friction and the depth of the micro-crystallizing metal structure, the micro-crystallizing metal structure is formed in response to loads on the V-surfaces. Durability tests were conducted in LOW ratio, in which face pressure on the parts in contact is high, and in OD ratio, in which the face pressure is low in comparison to LOW ratio, and the state of the micro-crystallizing metal structures was compared using transmission electron microscopy (TEM). Martensite structures, characteristic of carburized heat treatments, were observed in the outer periphery of the drive pulley and the inner periphery of the driven pulley, which are used in the OD ratio, and in the outer periphery of the driven pulley, which is used in the LOW ratio, but the formation of a micro-crystallizing metal structure was not observed. However, in the inner periphery of the drive pulley, which is used in the LOW ratio, a micro-crystallizing metal structure of approximately $3\mu\text{m}$ in depth was formed, and cracking originating at the surface was observed close to the boundary between the martensite structure and the micro-crystallizing metal structure (Fig. 13). In addition, observation using scanning electron microscopy (SEM) showed that cracking also originated internally, at a depth of approximately $15\mu\text{m}$ from the surface (Fig. 14). Based on this evidence, it was considered that wear of the

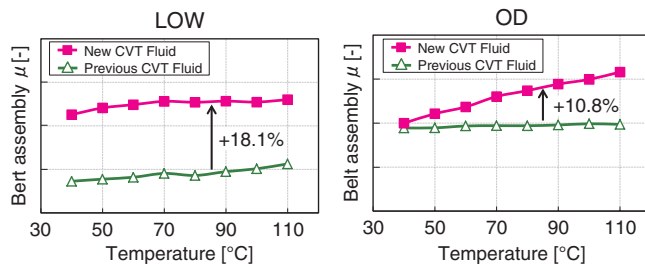


Fig. 12 Belt assembly friction coefficient

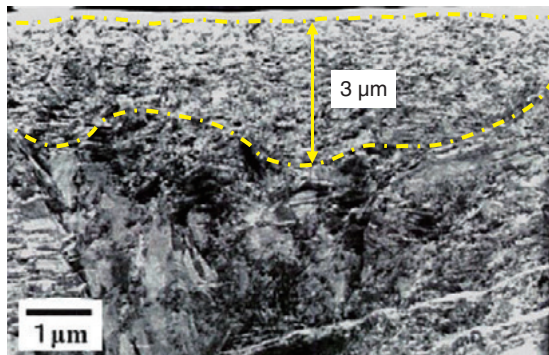


Fig. 13 Micro-crystallizing metal structure in drive pulley

V-surfaces occurs due to these cracks progressing, resulting in peeling of the surface. It was therefore necessary to attempt to prevent cracking originating at the surface and in the interior of the inner periphery of the drive pulley.

As methods of preventing wear, a zinc electric discharge coating was applied to the surface of the pulley in response to cracks originating on the surface, and carbonitride heat treatment was applied in response to cracks originating internally. A cross-section of the pulley close to the surface was observed using SEM following the durability test (Fig. 15). No cracks were observed either on the surface or in the interior of the cross-section.

3.3. Configuration to Help Control Movable Pulley Tilt

The trajectory of a metal pushing V-belt around the pulleys deviates slightly from the theoretical winding radius due to factors including the pulley tilt (Fig. 16). This is due

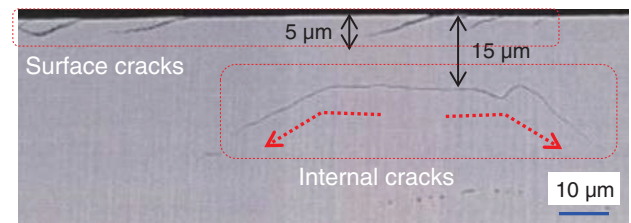


Fig. 14 Pulley cracks in durability test

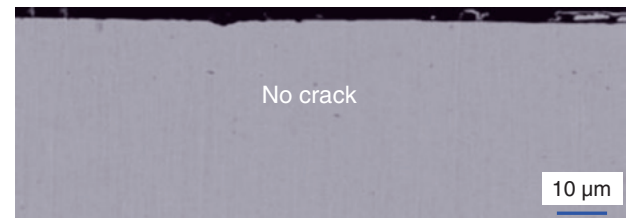


Fig. 15 Sectional view of pulley surface

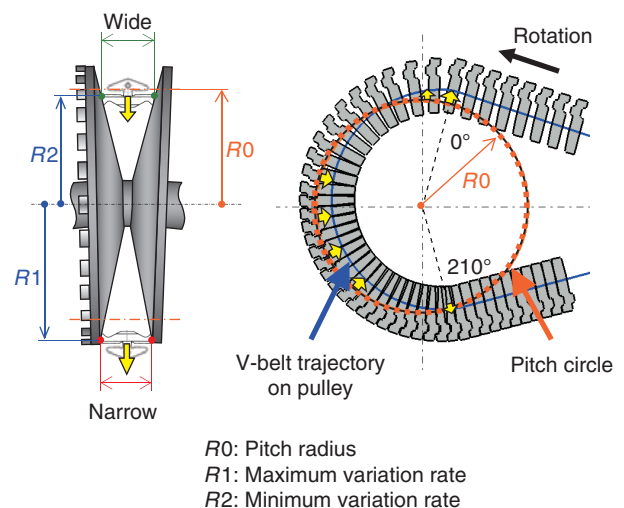


Fig. 16 Mimetic diagram of metal pushing V-belt trajectory on pulley

to the fact that when the movable pulley is pressed against the belt by thrust force, the distance between the V-surfaces of the section of the pulley that the belt is not wrapped around becomes smaller, while the distance between the V-surfaces becomes greater in the section that the belt is wound around, reducing the winding diameter in the latter section. This movement of the elements in the radial direction causes wear loss. Seeking to achieve its goal of increasing efficiency in the OD ratio during cruising, the research discussed here modified the configuration of the drive pulley pistons, which it was considered had a significant effect on the winding diameter and on pulley tilt, and succeeded in controlling pulley tilt and reducing deviation of the trajectory.

3.3.1. Pulley piston configuration

The generation of pulley thrust by the conventional configuration of the drive pulley is due to the No. 1 hydraulic chambers, located on the rear faces of the movable pulley V-surfaces, and the No. 2 hydraulic chambers, containing the secondary pistons that push the inner periphery of the pulley. Because the secondary pistons push on the inner periphery, approximately 50% of thrust acts on the inner periphery, and when the belt runs through the pulleys, the outer circumference of the movable pulley opens and the pulley becomes inclined. Because of this, the belt's winding trajectory deviates slightly from the theoretical winding radius. In order to help control this inclination, a No. 3 hydraulic chamber, which pushes the outer periphery of the movable pulley, was added to the outside of the No. 2 hydraulic chamber. Figure 17 shows a comparison of the previous and the new configurations. The transmission efficiency of the belt was studied by varying the thrust on the inner and outer peripheries of the movable pulley, and it was found that transmission efficiency was highest when pulley thrust was higher on the outer periphery side, and was at its highest when the ratio between pulley thrust on the inner and outer peripheries was 1:1 (Fig. 18).

3.3.2. Belt trajectory

The trajectory of the belt as it wraps around the pulleys was compared for the conventional and the

new configurations in order to verify the effects of the modification of the configuration. The belt trajectory from the beginning to the end of winding around the pulleys was measured by eddy current displacement sensors positioned at 30° intervals (Fig. 19). 0° represented the beginning of the belt winding around the pulleys, and 210° represented the end. In terms of the trajectory of the section of the belt winding around the pulleys, the belt displays its highest radius at the commencement of winding around the pulleys, and the minimum winding radius occurs in the vicinity of the center of the section; the radius increases slightly at the end of winding around the pulley. Based on the difference between the trajectories of the belts in the conventional and the new configurations, it is considered that the friction loss due to movement in the radial direction displayed by metal pushing V-belts occurs because of the difference between the maximum radius at the commencement of winding around the pulley and the minimum radius at the central

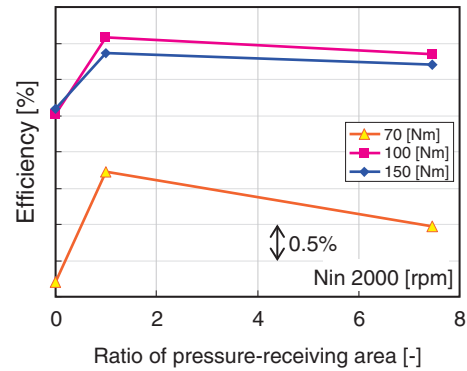


Fig. 18 Relationship between pressure-receiving area and efficiency

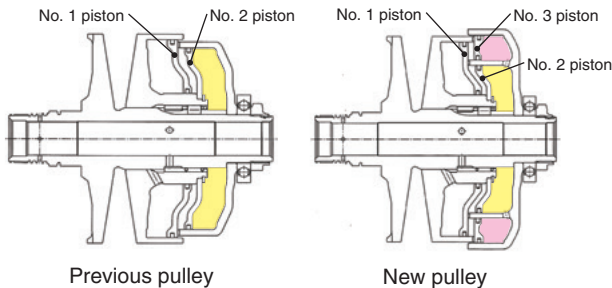


Fig. 17 Comparison of pulley configurations

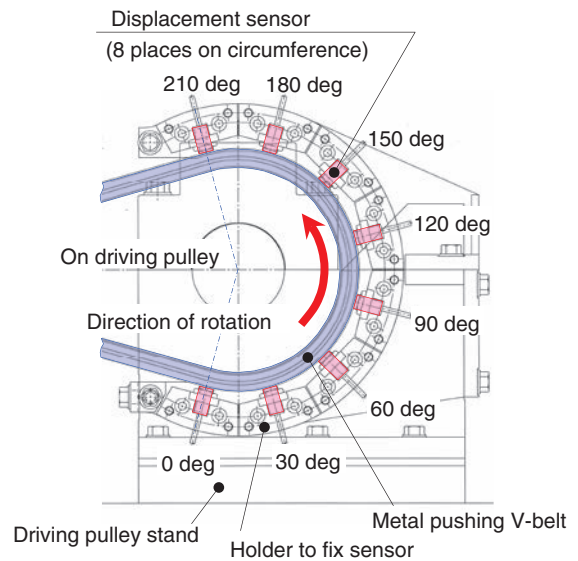


Fig. 19 Equipment for measurement of trajectory of metal pushing V-belt on pulley

part of the belt on the pulley (Fig. 20). The use of the new configuration has reduced the radius of the belt at the commencement of winding by 0.1mm against the conventional structure.

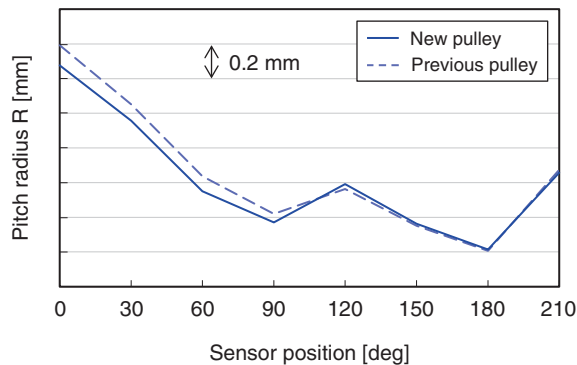


Fig. 20 Results of measurements of trajectory of metal pushing V-belt

4. Increase in Belt Transmission Efficiency

The effects of the three methods of increasing transmission efficiency studied in this project were verified in tests. The new control method employed reduced side pressure by approximately 40% and increased transmission efficiency by 2.6%. The use of the CVT fluid with an increased coefficient of friction increased the coefficient of friction by 10%, and setting side pressure in accord with the increased coefficient of friction increased transmission efficiency by 0.2%. The structure for control of tilt of the moving pulley reduced deviation of the belt-winding trajectory by 0.1mm, and increased transmission efficiency by 0.4%. The application of the three technologies to increase the transmission efficiency of metal pushing V-belts has increased the unit efficiency of the belt by 3.2% (Fig. 21).

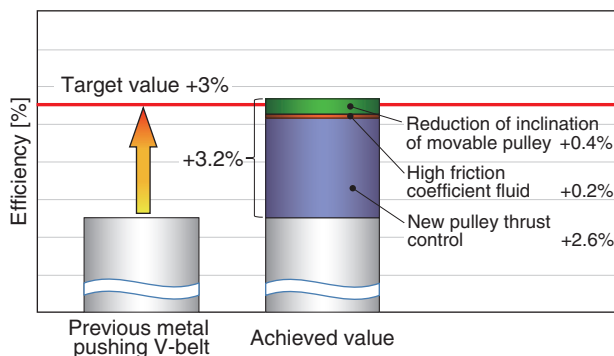


Fig. 21 Increase in efficiency

5. Conclusion

Three methods that were successful in increasing the unit efficiency of the belt have been developed.

- (1) Focusing on the correlation between the torque ratio, transmission efficiency, and the phase difference between the input and output shafts when engine speed fluctuates, the project developed a system making it possible to detect the torque ratio during vehicle operation and apply hydraulic control in order to operate the vehicle at the desired torque ratio. The optimization of pulley thrust increased transmission efficiency by 2.6% in OD ratio during cruising.
- (2) In order to increase the coefficient of friction between the pulleys and the belt, the mixture of additives to the CVT fluid was modified, increasing the coefficient of friction at the benchmark temperature for measurement of the coefficient of friction by 10%, and transmission efficiency by 0.2%. It was also discovered that carbonitride heat treatment and zinc electric discharge coating are effective in helping to prevent wear of the pulley V-surfaces.
- (3) In order to control deviation of the winding trajectory of the belt due to tilt of the movable pulley, a third piston was added to the outside of the second piston; the application of pulley thrust to the inner and outer peripheries of the movable pulley in a ratio of 1:1 reduced the amount of deviation of the winding trajectory of the belt by 0.1mm and increased transmission efficiency by 0.4%.
- (4) The simultaneous application of all three methods of increasing transmission efficiency resulted in a unit increase of 3.2% in transmission efficiency in a metal pushing V-belt in OD ratio during cruising.

6. Afterword

The development project discussed in this paper commenced with the question “Do the issues involved in increasing the transmission efficiency of CVT using metal pushing V-belts really only involve the belts themselves?” As discussed in this paper, the project determined that the belts were being used in an environment in which they could not adequately display their level of performance. The authors intend to push ahead with further performance increases in future, seeking to realize even higher figures.

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■ Author ■



Shuji ICHIO



Tooru YAGASAKI



Taiki ANDO



Kyohei SAKAGAMI



Soichiro SUMIDA

Development of New CVT for Global Compact Car

Tomoyuki KANDA*

Hirohiko TOTSUKA*

Masashi OONUMA*

Norimasa SHUDO*

Tomohiro HIRATA*

Ukyo OGATA*

ABSTRACT

A new CVT that is lighter in weight and more highly efficient than the previous CVT for use in compact vehicles has been developed and used in the 2014 model year FIT. The allowable torque capacity was expanded to that of the 1.8-L engine class, making this CVT usable in a greater number of vehicle models. The ratio coverage was also expanded and the transfer efficiency was increased to enhance fuel economy and drivability.

Integration of hydraulic control system functional parts and reduction in the number of case component parts were carried out as structural modifications. Pulley side pressures were also reduced by the use of new CVT fluid so that the pulley could be made more compact and lighter in weight. These measures achieved a weight reduction of approximately 10%.

Enhancements were made in cooperative control of the power train and in CVT shift control, providing a more linear acceleration feeling than in previous models.

1. Introduction

There has been growing concern about environmental problems on a global scale, and exhaust gases from motor vehicles are said to have a major impact on CO₂ emissions. Measures to heighten the environmental performance of motor vehicles are being pursued in many directions, and enhancement of the environmental performance of transmissions is an important part of this effort, together with the evolution of internal combustion engine technology.

CVT that enable efficient operation of an internal combustion engine are being widely adopted in Japan for use as equipment to enhance environmental performance in vehicles ranging from mini-compact cars to midsize cars. Meanwhile, step automatic transmissions and Dual Clutch Transmissions (DCT) are being adopted overseas as drive systems optimized to their various regional environments of use, but there is also a movement to adopt the CVT as part of recent initiatives to reduce CO₂ emissions. For expanded application, the CVT will need increased drivability as well as lighter weight and greater compactness to increase installability in vehicles, in addition to increased environmental performance.

Honda has used the CVT since 1995⁽¹⁾, expanding its application from hybrid vehicles⁽²⁾ to compact⁽³⁾ and midsize vehicles⁽⁴⁾. It has also been applied to mini-compact cars in

Japan⁽⁵⁾ and the range of CVT-equipped vehicles in other countries has also been expanding, starting with use in midsize vehicles⁽⁶⁾ for the North American market in 2013. Currently, this CVT is being applied in global models of compact vehicles to bring about increased environmental performance worldwide. This paper will describe the system development goals, its basic construction, and its main technology.

2. Development Goals

The construction of this CVT was changed from that of the CVT previously used in compact vehicles. In order to supply a framework provided with performance for use in global models, technology to increase environmental performance was applied, it was made more compact and lighter in weight, and its driving performance was enhanced. The development goals are as follows:

- (1) Expand the ratio coverage and increase the transmission efficiency in order to achieve a balance between driving performance and fuel economy adapted to the environment of use
- (2) Reduce the weight and increase the torque capacity to enable installation in multiple vehicle models for global deployment
- (3) Enhance the drivability of CVT vehicles by evolution of cooperative power train control and CVT shift control

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3. Basic Specifications and Performance

Figure 1 shows the external appearance of the new CVT, Fig. 2 shows the main section view, and Table 1 shows the main specifications.

In order to reduce the weight, the framework construction has been modified from that of the CVT previously used in compact vehicles and instead adopts the same framework as the CVT used in midsize vehicles. The pulley bearing layout has been modified and the hydraulic control system and hydraulic supply unit have been integrated.

The basic specifications and dimensions are the same as the previous CVT, with the ratio coverage expanded and the final gear changed to a lower ratio. The basic framework parts were made installable in vehicles with a wide range

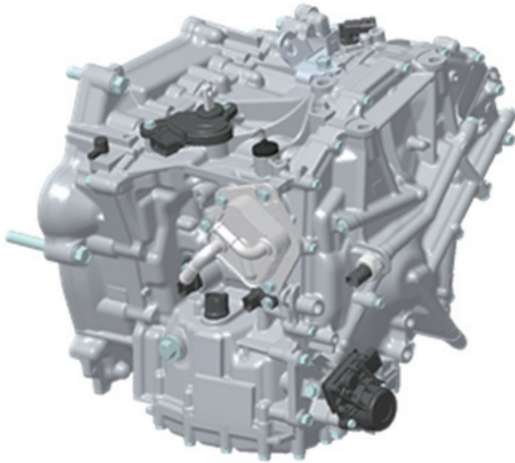


Fig. 1 Appearance of new CVT

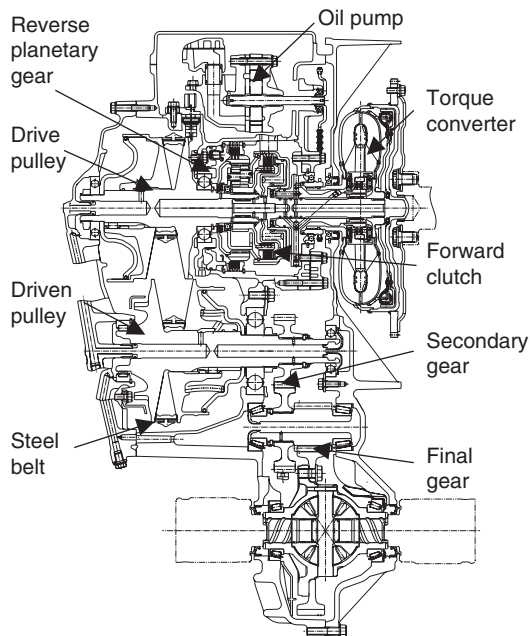


Fig. 2 Main section

Table 1 Main specifications

		2014 model year FIT	2008 model year FIT
Engine	Type	1.3 L/1.5 L	1.3 L/1.5L
	Max. power [kW/rpm]	97/6600 (1.5 L)	88/6600 (1.5 L)
	Max. torque [Nm/rpm]	155/4600 (1.5 L)	146/4800 (1.5 L)
Ratio	Low OD	2.526 0.408	2.419 0.421
	Secondary	1.266	1.266
	Final	3.941	3.875
	Reverse	1.119	1.024
	Overall	12.60 2.036	11.86 2.065
Dimension	Overall length [mm]	368.5	368.5
	Pulley axis distance [mm]	156	156
	Weight [kg]	69.4 (1.3 L)	77.6 (1.3 L)

of engine displacements by designing the system to support engines up to the 1.8-L class. A weight reduction of approximately 10% compared to the previous CVT was achieved.

4. Main Technologies

4.1. Expansion of Ratio Coverage

In order to achieve a higher-level balance between performance in acceleration from a standing start and performance in fuel economy in accordance with the distinctive characteristics of vehicles with this CVT, the ratio coverage was set at 6.2, which is approximately 8% broader than Honda's previous CVT. Figure 3 shows a comparison of the ratio coverage settings. The ratio was set by keeping the ratio on the overdrive (OD) side the same as previously and expanding the ratio on the low side. In other words, standing start performance was enhanced and acceleration performance that in combination with shift control technology yields responsiveness was achieved. For the purpose of reduction in size and weight, the center distance between the drive pulley and driven pulley was made the same 156 mm as previously, the shaft diameter was reduced, and the pulley outer diameter was enlarged. The new CVT fluid developed for the 2013 model⁽⁶⁾ was adopted. The effect of that fluid's high metallic coefficient

of friction enabled reduction of the load needed to hold the belt, allowing a reduction in shaft rigidity. Reducing the diameter of the shaft allowed the winding diameter of the belt to be reduced so that the ratio could be enlarged.

4.2. Increase in Efficiency

Figure 4 shows a comparison of transmission efficiency in cruising vehicles. An increase of approximately 1.0% to 1.5% relative to the previous CVT was achieved, enhancing performance in the high-vehicle-speed range. The following measures were taken to achieve higher efficiency.

4.2.1. Reduction in oil pump capacity

The higher coefficient of friction in the new CVT fluid has enabled a reduction in the thrust. The area of the pulley

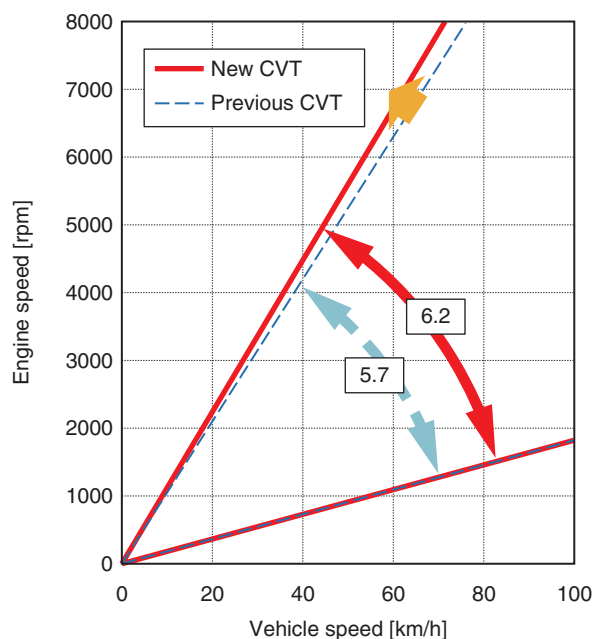


Fig. 3 Comparison of ratio coverage

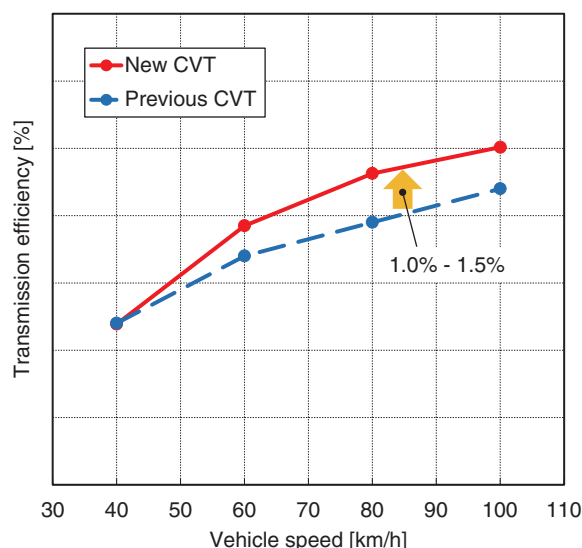


Fig. 4 Comparison of transmission efficiency

piston was therefore reduced and the piston shape was enhanced, reducing the dead volume in the piston chamber. This enabled a reduction in the amount of oil needed to fill the pulley piston chamber during sudden speed changes, for example, and the oil pump capacity was therefore reduced by approximately 10%.

4.2.2. Reduction in oil stirring resistance

The amount of lubrication for each part and the oil flow routes were laid out with the aim of reducing the friction that occurs when the oil is stirred by the internal rotor. The effect that stirring resistance has on the final gear is particularly strong because that gear is located in the bottom portion of the transmission. A baffle plate was therefore installed to limit the inflow of oil to the final gear unit. Figure 5 shows a diagram of the oil flow in the final gear unit. The inflow of oil to the final gear unit is regulated by the area of the opening at the bottom. The oil that is scooped up is channeled onto the rear surface of the baffle plate and is used for lubrication. The slope of the rear surface was further given a sloping shape so that lubricating oil coming from the upper portion will be smoothly channeled back into the oil pan chamber.

4.2.3. Reduction in friction

The friction was also reduced and low-friction bearings were adopted in such areas as the sealing ring that serves as a rotating sliding part, the oil seal, and so on. The reverse brake disk unit drag torque was reduced.

4.3. Reduction in Weight

In order to increase the vehicle driving performance and fuel economy performance while seeking increased torque capacity in the basic framework, the CVT was made approximately 10% lighter in weight than the previous CVT.

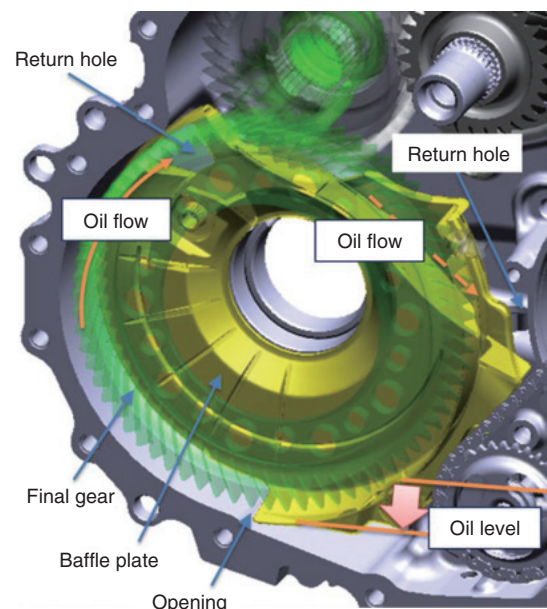


Fig. 5 Oil flow image around final gear

4.3.1. Reduction in pulley size and weight

The effect of pulley side pressure reduction resulting from the new CVT fluid's effect in increasing the coefficient of friction between belt and pulley⁽⁶⁾ enabled a reduction in the applied load on the shaft and the V-surface portion. This enabled a reduction in weight while maintaining strength and rigidity equal to previous levels. Figure 6 shows a comparison of stress generated in the driven pulley at low ratio, and Fig. 7 shows the displacement that occurs in the V-surface portion at that time. When the V-surface portion was made thinner and the shaft portion smaller in diameter, the high stress generated at the base of the V-surface and the displacement of the V-surface portion were made equal to the previous CVT.

4.3.2. Function-integrating construction

The hydraulic control system parts were previously scattered around the case. In the new CVT, however, the hydraulic supply unit was integrated functionally as a valve body assembly and the supply passages were integrated into the oil passage body, thus simplifying the parts. This achieved an approximately 1.2-kg weight reduction.

Figure 8 shows the configuration of the hydraulic control system parts.

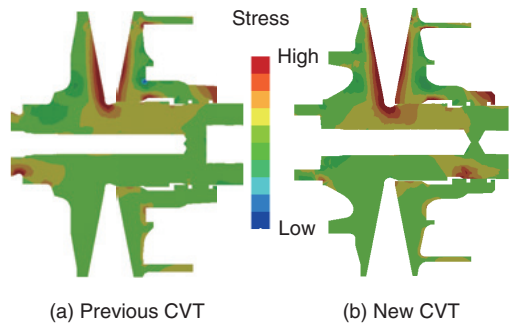


Fig. 6 Comparison of generating stress map

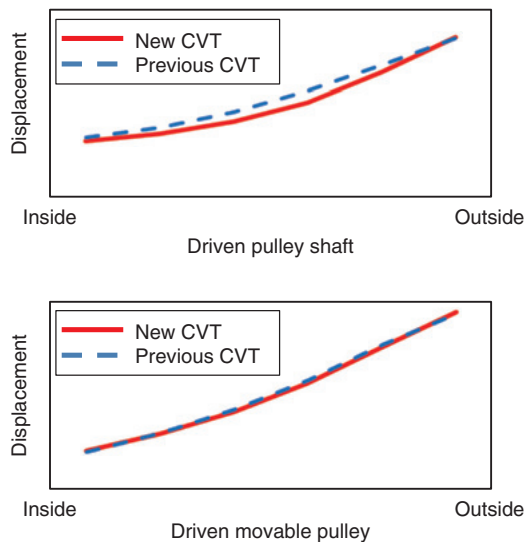


Fig. 7 Comparison of generated displacement

4.3.3. Reduction in case size and weight

The transmission case and the left side cover were used as support structures for both ends of the drive and driven pulley shafts. The elimination of the intermediate plate found in the previous structure realized an approximately 3.1-kg reduction in the total weight of the case parts. Figure 9 shows the configuration of the case parts.

A formed-in-place gasket was also adopted in place of the previous seal made by paper packing. This enabled the gasket to follow the relative displacement between the matching surfaces of the case. As a result, the tightening stiffness could be optimized and the height of the tightening boss as well as the bolt length could be reduced 10 mm, achieving a reduction in the case weight.

Figure 10 shows a diagram of areas where formed-in-place gaskets are used. Figure 11 shows details of bolt tightening.

4.3.4. Reduction in size of reverse mechanism

New nitrided material was used to achieve greater strength in the reverse planetary ring gear. The increased

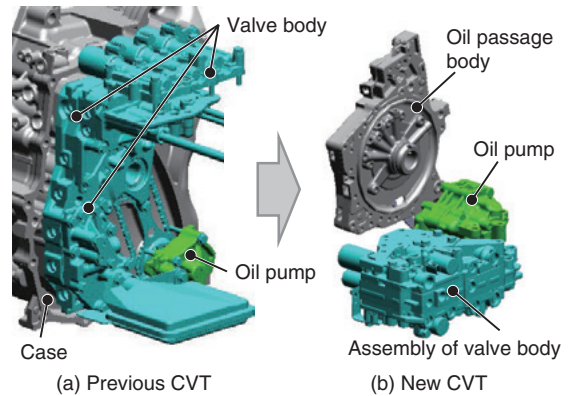


Fig. 8 Configuration of hydraulic control unit

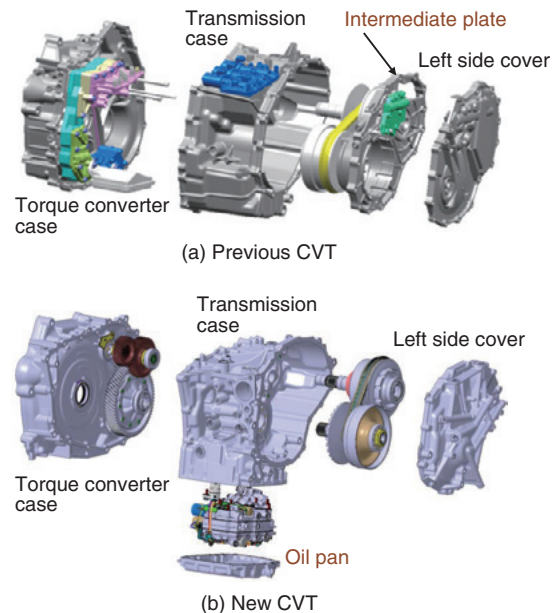


Fig. 9 Case parts composition

coefficient of friction of the new CVT fluid enabled the surface contact pressure on the reverse brake disk facing to be maintained while making the reverse brake disk more compact. This reduced the size of the reverse mechanism by approximately 12%. Figure 12 shows comparative section views of the reverse system.

4.3.5. Reduction in weight of bearings

As a result of the reduction in thrust achieved by use of the new CVT fluid, the load on the pulley supports was reduced and the pulley support bearings were reduced in size and weight. Taper bearings were used for the secondary and final shafts to maintain the needed capacity

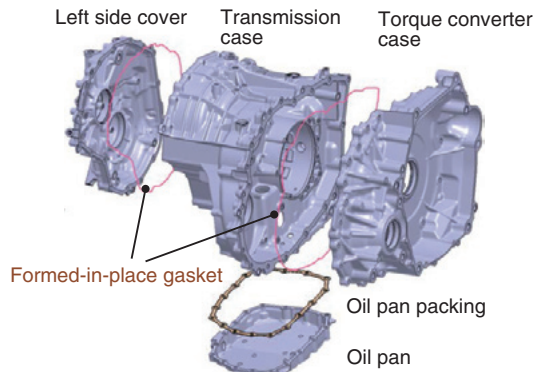


Fig. 10 Formed-in-place gasket

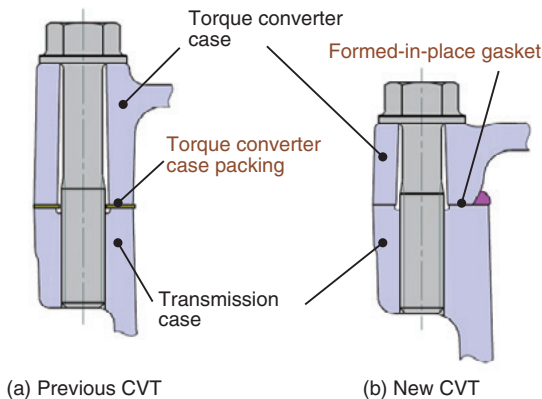


Fig. 11 Detail of bolt tightening

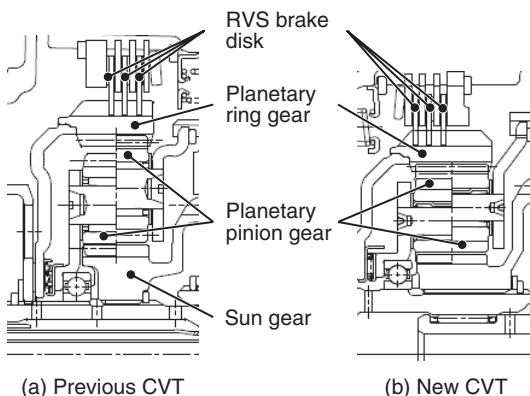


Fig. 12 Comparison of RVS system

while reducing the size and weight. As a result, a weight reduction of approximately 23% was achieved in the bearings compared to the previous CVT. Figure 13 shows comparative section views of the secondary shaft and final shaft. Low friction specifications were adopted for the taper bearings, giving consideration to the effect on efficiency, as well.

4.4. Technology to Enhance Driving Performance

Cooperative control of the power train, with drive force as the target value, was adopted as a technology for enhancing drivability of the CVT vehicle. In CVT vehicles, the CVT responds to driver demand for acceleration by stepless downshifting, and this can tend to delay the acceleration response. Previous drive-by-wire (DBW) cooperative control would set the throttle position according to the accelerator pedal position, so that when engine speed increased due to the ratio change, the engine load decreased and torque was reduced. In this CVT, implementation of power train cooperative control of the kind shown in Fig. 14 that has drive force as a target value determines engine torque output according to the ratio change in the CVT.

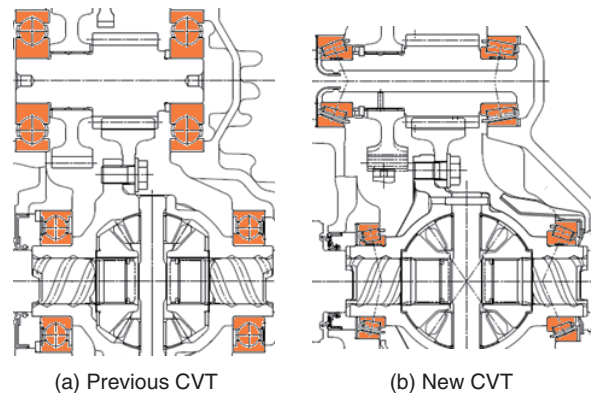


Fig. 13 BRG shape at secondary and final shaft

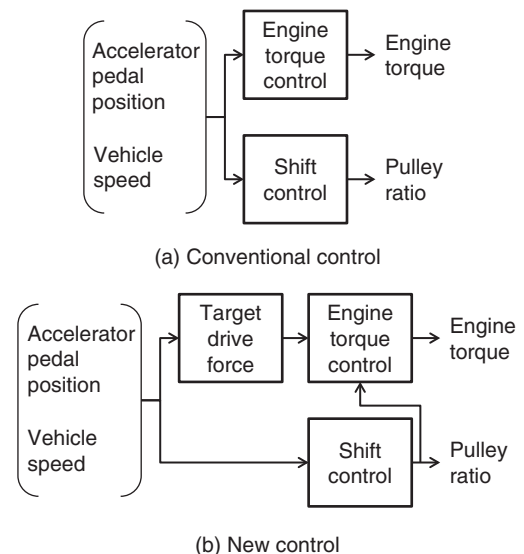


Fig. 14 Schematic diagram of cooperative control

Therefore the reduction in engine torque due to rise in engine speed could be limited, achieving a stable increase in acceleration response. Figure 15 shows time-series changes in engine torque during acceleration under conventional control and the new control.

Enhancement of shift control during kick-down resulted in enhanced acceleration feeling in response to sudden driver demand for acceleration. Figure 16 shows

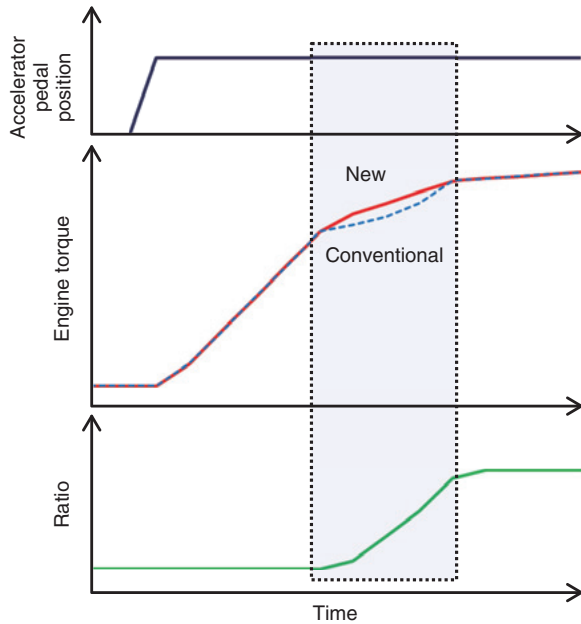


Fig. 15 Schematic diagram of torque response

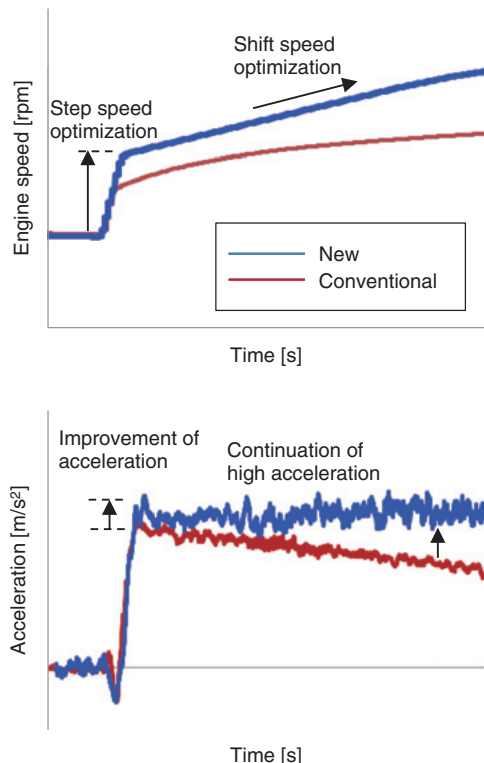


Fig. 16 Comparison of kick-down characteristics

time-series waveforms for acceleration and engine speed during kick-down. Acceleration during initial kick-down was increased by increasing the amount of the step shift. Continued acceleration after kick-down was also achieved by maintaining the higher engine speed after the step shift.

4.5. Control Technology for Increased Fuel Economy

The accuracy of input torque information provided to the CVT control system was increased as a technique for increasing fuel economy.

Previously, engine torque was estimated from engine speed, intake manifold vacuum, ignition timing, and other such information, and CVT pulley pressure was decided accordingly. This CVT uses highly accurate engine torque information according to the engine control state calculated in cooperative control of the power train, as described above, and CVT pulley pressure is thus optimized. Setting the ratio by the target drive force also assures controllability of the drive force while limiting the rate of fuel consumption.

Increased transmission efficiency and the new cooperative control have yielded an approximately 7% increase in fuel economy in JC08 mode.

5. Conclusion

A new CVT was developed for use in a global compact car and the following technologies were established:

- (1) A balance between driving and fuel economy performance was achieved by expanding the ratio coverage by approximately 8% and increasing the transmission efficiency by approximately 1.0% to 1.5%.
- (2) A reduction in weight of approximately 10% relative to the previous CVT was achieved by modification of the framework structure and use of a new CVT fluid.
- (3) Cooperative control of the power train and enhancement of shift characteristics yielded increased drivability and enhanced acceleration feeling.

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■ Author ■



Tomoyuki KANDA



Hirohiko TOTSUKA



Masashi OONUMA



Norimasa SHUDO



Tomohiro HIRATA



Ukyo OGATA

Development of Agile Handling Assist

Takeshi KOJIMA*

Tomoyuki FUTAMURA*

Kazutaka OHMURA*

ABSTRACT

Agile Handling Assist has been developed in order to increase vehicle responsiveness and stability, offering users handling that is both exhilarating and secure. To realize the new system, a unique Honda Agile Handling Assist logic was incorporated in high-function Vehicle Stability Assist. By contrast with Vehicle Stability Assist, which goes into operation in the limit range, Agile Handling Assist operates from the daily use range, applying light braking force, unnoticeable to the driver, to the inner wheels when the steering wheel is turned, and to the outer wheels when the steering wheel is returned, in order to increase responsiveness and stability. The system has been put into mass production in the 2014 model year ACURA RLX.

1. Introduction

The automobile is the very symbol of personal mobility, and a level of handling performance that offers the driver the joy of vehicle operation is therefore demanded. Today, the results of performance evaluation tests by actors including public agencies and motoring magazines are available to consumers, and are being used as standards of judgment when purchasing an automobile. Honda has developed and mass produced numerous handling technologies to realize comfortable and immediate responsiveness to the driver's steering operations. The Active Torque Transfer System (ATTS)⁽¹⁾ increased ease of handling by controlling the allocation of drive force to the left and right wheels of FF vehicles, and was employed in the 1997 model Prelude. Following on from this, the company developed Super Handling-All Wheel Drive (SH-AWD)⁽²⁾, a system that controls drive force to all four wheels of AWD vehicles, and employed it first in the 2005 model LEGEND. More recently, the company has developed Precision-All Wheel Steer (P-AWS), a system that controls the rear toe angle, and which has been applied in the 2013 model ACURA RLX. Honda has also expanded the range of operation of its Vehicle Stability Assist (VSA), a safety function that formerly went into operation at the turning limit in order to control vehicle instability, to the entire daily use range, and has worked to boost handling performance by means of brake control. Here, however, the issues have been the development of new control logics and achieving the quietness and durability of standard VSA. A high-function VSA has now been developed that, in addition to danger avoidance functions, also offers functions that increase user convenience, for example a hill-hold function and a low-speed following function, and

also boasts increased quietness and durability. Using this high-function VSA, a control logic based on a new concept has been developed for Agile Handling Assist (AHA), a handling function that applies brake control across the daily use range and increases automotive responsiveness and stability performance.

2. Purpose of AHA and Development Goals

The purpose of AHA is to increase responsiveness and stability performance in relation to steering operations by the driver.

AHA reduces corrective steering, and offers the driver agile, exhilarating, and secure handling performance.

Concretely, the following three aims were set for the development of AHA:

- (1) Increased steering responsiveness
- (2) Market appeal, as exemplified by a smooth control feel
- (3) Increased at-the-limit handling performance

3. Operating Scenarios and Effects of AHA

The operating scenarios for AHA will be explained with reference to Fig. 1. The horizontal axis shows the degree of turning of the vehicle, and zero represents the commencement of straight-line driving: the further to the right, the closer to the turning limit. The vertical axis shows the degree of operation of the steering wheel, with zero representing fixed steering: the further upward, the more rapid the steering operation.

In the upper left area of the AHA operating range, shown as (1), control is applied to agilely turn the vehicle and achieve an early convergence of roll when a sudden steering operation is engaged from a state close to straight-

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line progression, for example a lane change to avoid an obstacle or passing through a roundabout.

In the lower right area of the AHA operating range, shown as (2), steering operation is applied when turning, with the vehicle going from the medium to the high range of lateral acceleration. This encompasses such scenarios as steering around S-shaped curves on a winding road or turning the steering wheel while cornering. Here as well, AHA offers agile turning, making it easy for the driver to trace the desired line.

In the upper right area of the AHA operating range, shown as (3), rapid steering operation is applied when turning, with the vehicle going from the medium range of lateral acceleration to the high range of lateral acceleration. This includes such scenarios as a rapid maneuver to avoid an obstacle or a rapid avoidance maneuver during a turn. By realizing smooth and agile handling close to the limit, AHA offers the driver a sense of security that is unaffected by conditions on the road.

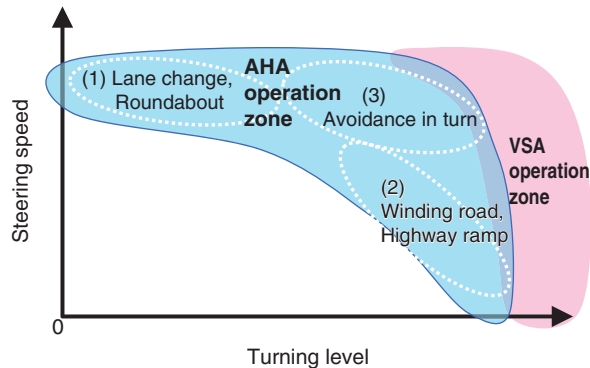


Fig. 1 AHA activation scenario

4. System Composition

The AHA functions were realized by adding an original control logic to high-function VSA, which offers the merits of high levels of responsiveness, quietness and durability.

4.1. Hardware

High-function VSA realizes increased market appeal, responsiveness to brake pressure, and durability compared to standard VSA.

In concrete terms, the number of cylinders in the pump has been increased, (high-function VSA uses six pistons against the two used in standard VSA), reducing pumping pulsation, and thus reducing operating noise and vibration. At the same time, the discharge volume has been increased, increasing responsiveness to brake pressure. In addition, a high-capacity motor has been employed, resulting in greater durability. Motor vibration and operating noise have been reduced by increasing the control frequency of motor control. This increase in control frequency also increased heat generation, leading to the use of a heat-dissipating configuration in which the sides of the Electronic Control Unit (ECU) case are manufactured from aluminum, and the motor control elements are connected using a heat-conductive adhesive.

4.2. Software

A unique AHA control logic was developed and incorporated in the high-function VSA ECU software.

4.2.1. AHA control logic

The composition of the AHA control logic will be discussed with reference to Fig. 2.

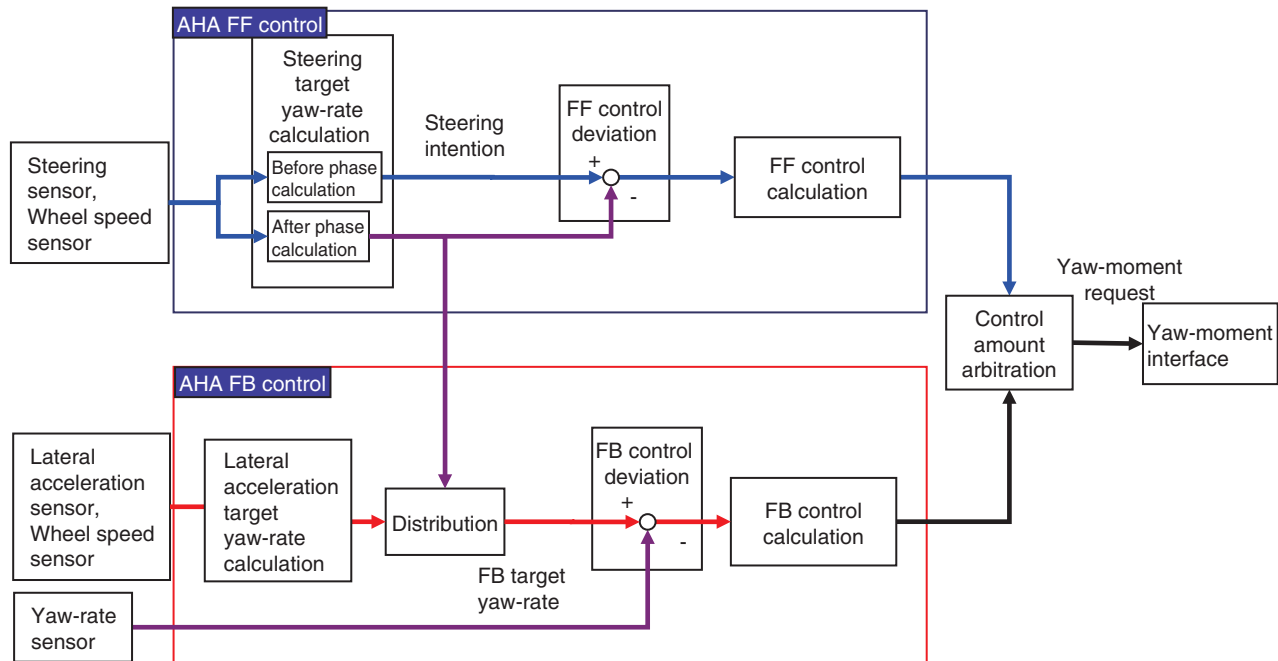


Fig. 2 AHA control logic

The control logic is made up of feed-forward (FF) control and feedback (FB) control.

- (1) FF control applies a light braking force to the inner wheels when the steering wheel is turned, and to the outer wheels when the steering wheel is returned, in order to reduce phase lag (Fig. 3). Using as a parameter the difference between a steering target yaw rate at which there is no phase lag, as determined from the steering angle, and a steering target yaw rate that considers phase lag, the control judges when to initiate and conclude control by comparing this parameter with the threshold value for FF control. In addition, the control calculates the feed-forward yaw-moment demand. In terms of operating scenarios, FF control is effective when rapid steering operation is initiated from a state of straight-line progression (Area (1) in Fig. 1 in Chapter 3) and when the steering wheel is operated while the vehicle is turning in the medium to high range of lateral acceleration (Area (2) in Fig. 1 in Chapter 3).
- (2) FB control is applied to make the actual yaw rate conform to the target yaw rate. However, the target yaw rate can be modified in line with the driver's operation of the steering wheel, and the driver's intended steering operation is thus reflected. This control realizes smooth and agile vehicle motion at close to the turning limit. Like VSA, the logic calculates a steering target yaw rate that considers phase lag and reflects the driver's intention, and a target yaw rate for lateral acceleration that estimates the turning limit. When the driver turns the wheel in excess of the turning limit or the target yaw rate is greater than the target yaw rate for lateral acceleration, VSA calculates the final target yaw rate, to realize stable control of the vehicle, as approaching the target yaw rate for lateral acceleration. By contrast,

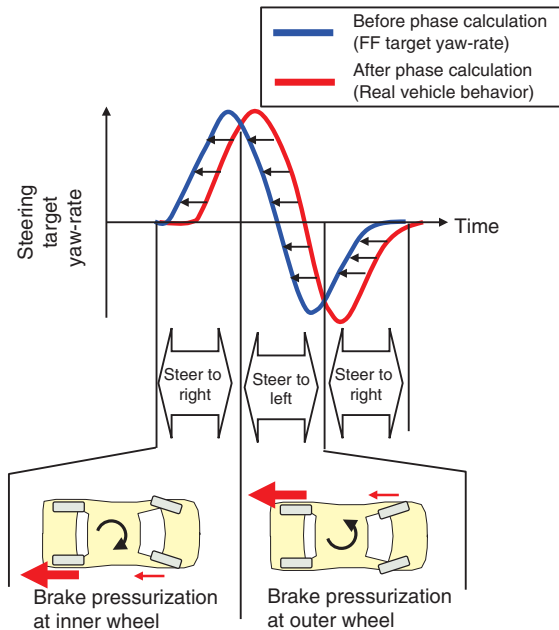


Fig. 3 AHA FF control

AHA FB control calculates the FB target yaw rate as approaching the steering target yaw rate, thus offering a form of control that reflects the driver's intention to turn the steering wheel. If the vehicle then becomes unstable when the driver returns the wheel or due to a sudden change in the actual yaw rate, the control calculates the target FB yaw rate as approaching the lateral acceleration target yaw rate, realizing control that both reflects the driver's intention and helps to ensure stability (Fig. 4). Using the difference between the target FB yaw rate and the actual yaw rate as a parameter, FB control judges when to commence and when to conclude operation based on comparison of the parameter with the threshold value for FB control. In addition, the logic calculates the FB yaw-moment request. FB control is effective in situations in which rapid steering operations are initiated while turning in the medium to high ranges of lateral acceleration (Area (3) in Fig. 1 in Chapter 3).

- (3) The necessary yaw moment for FF control and the necessary yaw moment for FB control are added, and the resulting figure is the necessary yaw moment for AHA control.
- (4) Figure 5 shows the composition of the ECU software for high-function VSA.

If the AHA control logic used an external ECU, and sent commands to VSA via CAN communications, delays in CAN communication would result in delays in response. For this reason, the AHA control logic was incorporated in the ECU software of high-function VSA.

Based on calculation results, the necessary yaw moment for AHA control is output to a yaw moment interface between AHA and VSA. The yaw moment interface reconciles the necessary yaw moments for AHA and VSA control, producing a final yaw moment for control. This is then converted into brake pressure, and brake pressure control is applied. The yaw moment interface is also provided with a limiter that controls the upper limit of the yaw moment necessary for AHA control, helping to maintain safety if the sensors malfunction.

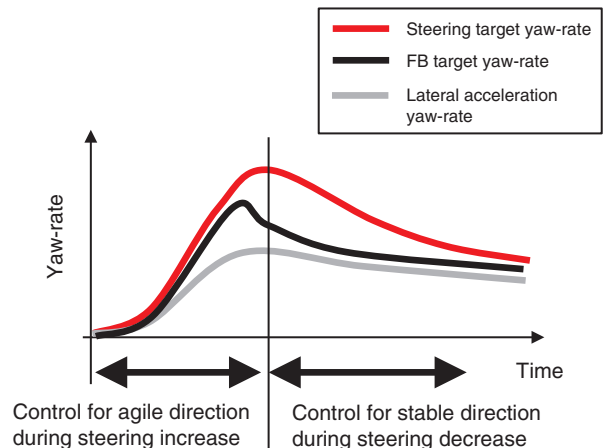


Fig. 4 AHA FB control

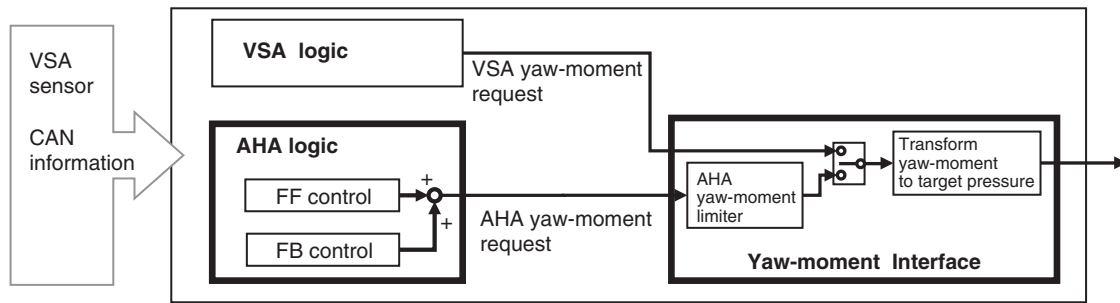


Fig. 5 VSA software structure

5. Effect in Boosting Performance

5.1. Increased Steering Responsiveness

In order to verify the effect of AHA in increasing steering responsiveness, the response time of yaw rate to steering wheel input was measured during a J-turn. Commencing at an initial vehicle speed of 60 km/h, the steering wheel was turned at a speed such that it would reach 90 deg in 0.5 s, and held at an angle of 90 deg. Response time was defined as the time from the steering angle reaching 45 deg to the time at which the yaw rate produced reached 90% of its maximum value. The results are shown in Fig. 6. The use of AHA reduces response time by 55 ms, from 330 ms to 275 ms. This demonstrates that the system realizes the development aim of achieving increased steering responsiveness.

5.2. Market Appeal

In order to verify the smooth control feel offered by AHA, steering angle and yaw rate were measured

during slalom driving on a dry road surface. The Lissajou waveform for yaw rate against steering angle is shown in Fig. 7. From an initial vehicle speed of 60 km/h, a sine wave input was applied at a cycle of 0.5 Hz, with a maximum steering angle of 90 deg. As the Lissajou waveform shows, when AHA was not in operation, the rate of increase or decrease in yaw rate against the steering angle changes, there is no linearity, and hysteresis also tends to be high.

By contrast, when AHA is in operation, the rate of increase or decrease in yaw rate is constant, linearity increases, and hysteresis is reduced. Realizing control linearity represented an issue in the application of brake pressure control from the daily use range, and this result indicates that the issue has been addressed and that the system contributes to increased linearity.

The results also indicate that the system has the merit of increasing hysteresis. These results demonstrate that the system realizes the development aim of achieving a smooth control feel.

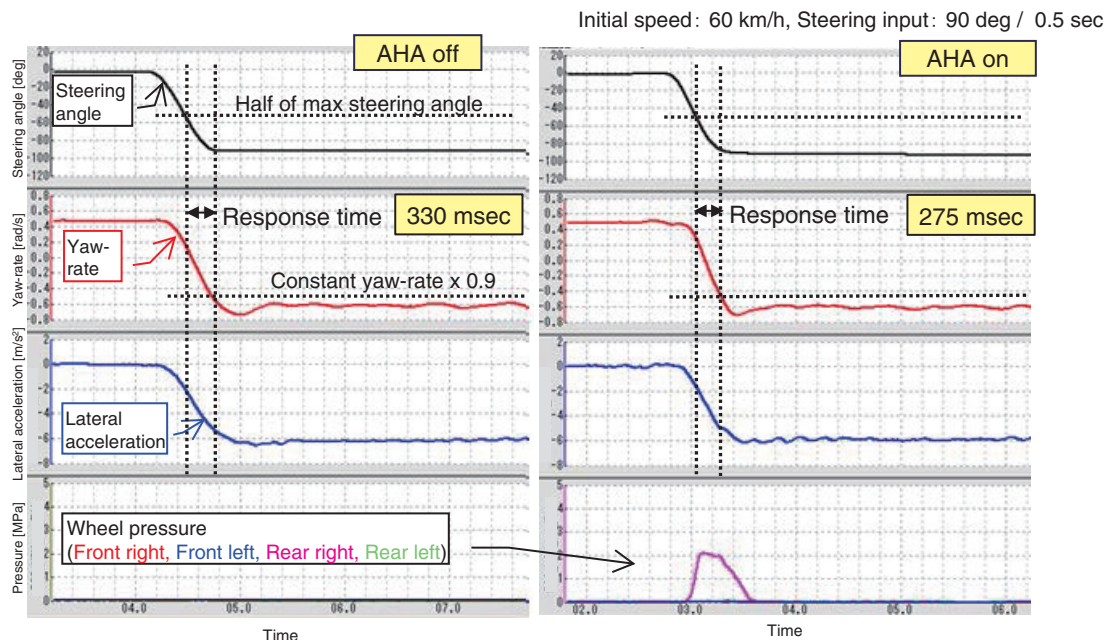


Fig. 6 J turn test result

5.3. Increased At-The-Limit Handling Performance

Test modes used by motoring magazines and public agencies were used on a Honda test course in order to test this performance factor.

5.3.1. Low-speed slalom driving on dry road

This is a test mode used by European motoring magazines. Twenty pylons are set up at intervals of 18 m, and the test vehicle passes between them at the highest speed possible, commencing from an initial vehicle speed of 50 km/h. Tests using this mode were conducted on a Honda test course with the results shown in Fig. 8. When

AHA was functioning, the time for the vehicle to pass between the pylons was reduced by 1.3 s. In addition, the Lissajou waveform of the yaw rate against steering angle demonstrates a reduction in yaw rate hysteresis, indicating an increase in yaw rate tracking performance and a reduction in the amount of necessary steering operation. This shows that the increase in responsiveness and ability to trace steering operation in the low-speed range and close to the limit offered chiefly by AHA FF control helps to eliminate excess steering operation, making it possible for the vehicle to pass more quickly between the pylons.

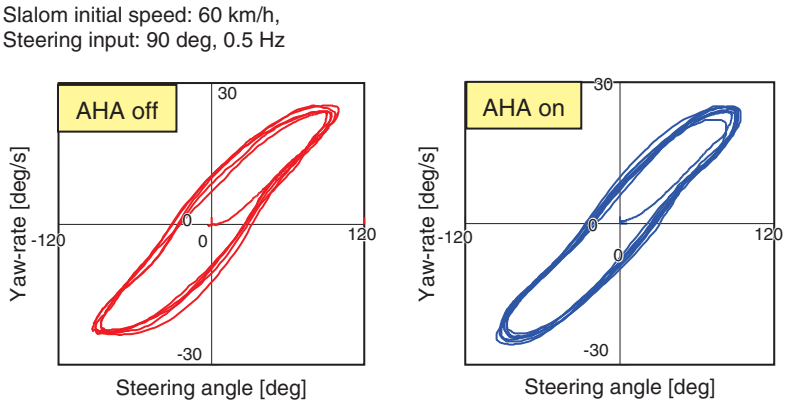


Fig. 7 Linearity control test result

Test mode of dry slalom at low speed

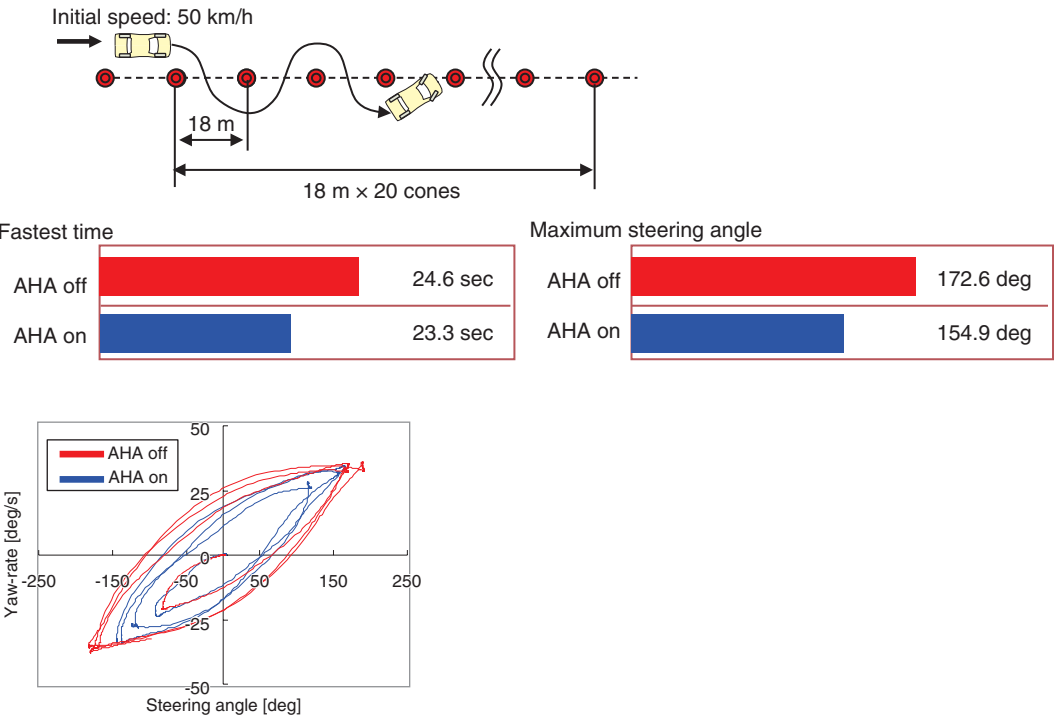


Fig. 8 Dry slalom test result at low speed

5.3.2. High-speed slalom driving on dry road

This is a test mode used by U.S. motoring magazines. Seven pylons are set up at intervals of 100 ft, and the test vehicle passes between them at the highest speed possible. Tests using this mode were conducted on a Honda test course. The results are shown in Fig. 9. When AHA was functioning, the time taken to pass between the pylons was reduced by 0.29 s. In addition, control of yaw rate diffusion made it possible to continue slalom driving with stable steering operation, and a reduction in the amount of steering operation was also observed. This indicates that the increased responsiveness, ability to track steering operation, and stability in the high-speed range and close to the limit offered by AHA FF and FB control offers the same high level of handling performance as was the case at low speed.

5.3.3. Slalom driving on low- μ road

This is a Honda test mode. Six pylons are set up at intervals of 30 m on a low- μ basalt road, and the test vehicle drives between them at the highest speed possible. These tests were conducted on a Honda test course with the results

shown in Fig. 10. When AHA was in operation, the time to complete the course was reduced by 1.1 s. In addition, control of yaw rate diffusion made it possible to continue slalom driving with stable steering operation, and there was also a reduction in the amount of steering operation. This demonstrates that a high level of handling performance even at close to the limit on a slippery road surface is realized chiefly by AHA FB control.

5.3.4. V lane change on dry road

This is a test mode used by U.S. government agencies. During the test, the vehicle passes into the specified lanes at maximum initial speed. The results of tests conducted using this mode on a Honda test course are shown in Fig. 11. When AHA was in operation, maximum vehicle speed increased by 3.9 km/h. In addition, the control of yaw rate diffusion has reduced the amount of steering operation. This indicates that the increased agility, ability to track steering operation and stability close to the limit offered chiefly by AHA FB control increases danger avoidance performance.

Test mode of dry slalom at high speed

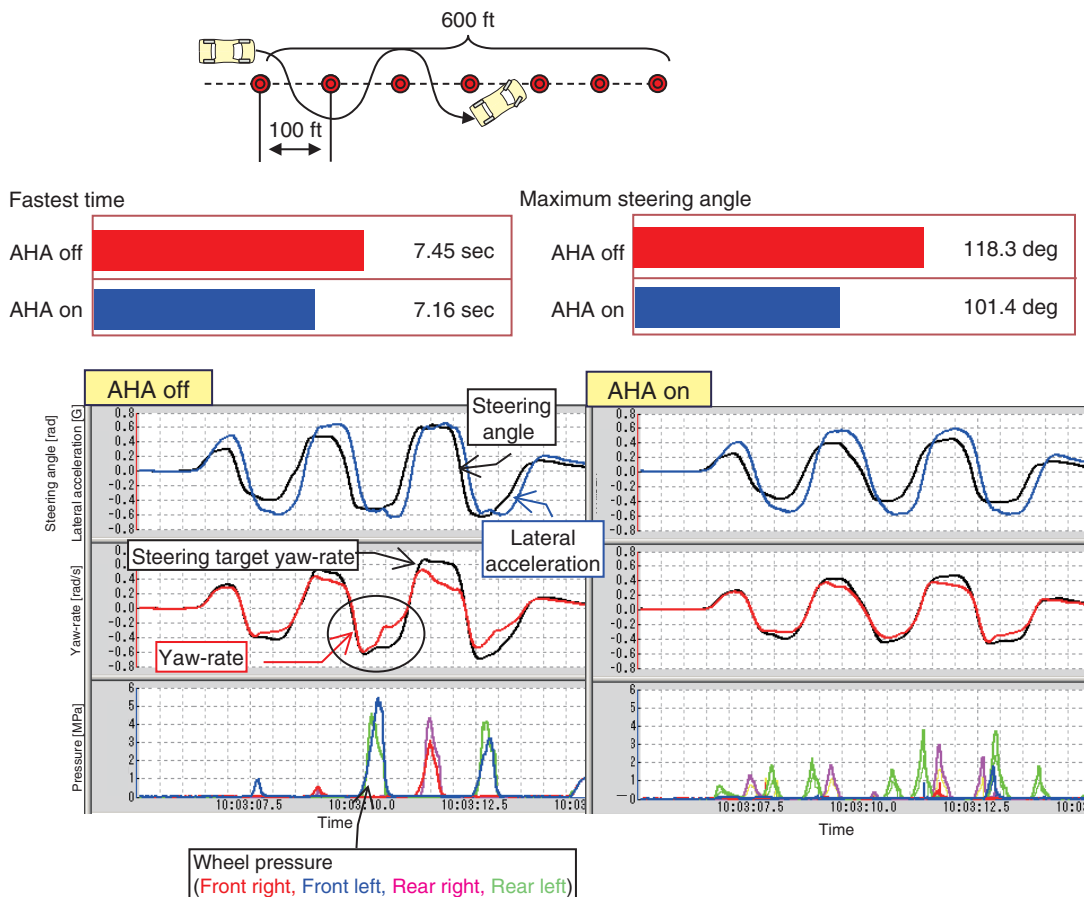
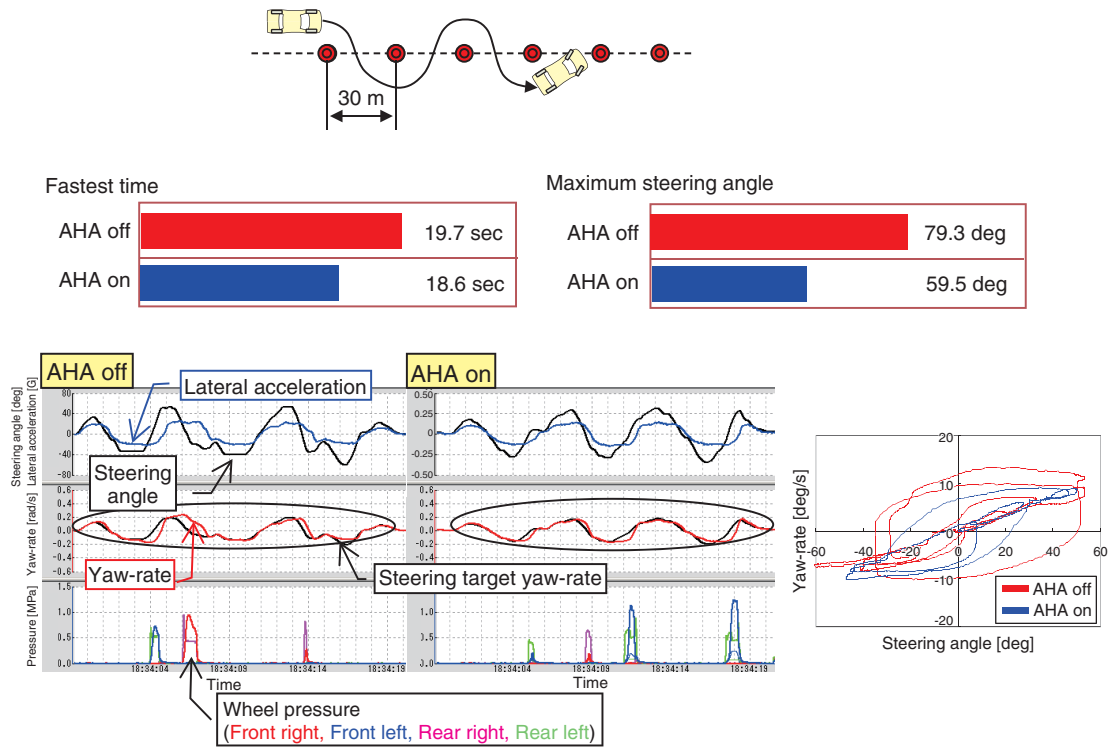


Fig. 9 Dry slalom test result at high speed

Test mode of low μ slalom

 Fig. 10 Low μ slalom test result

Test mode of dry V-lane change

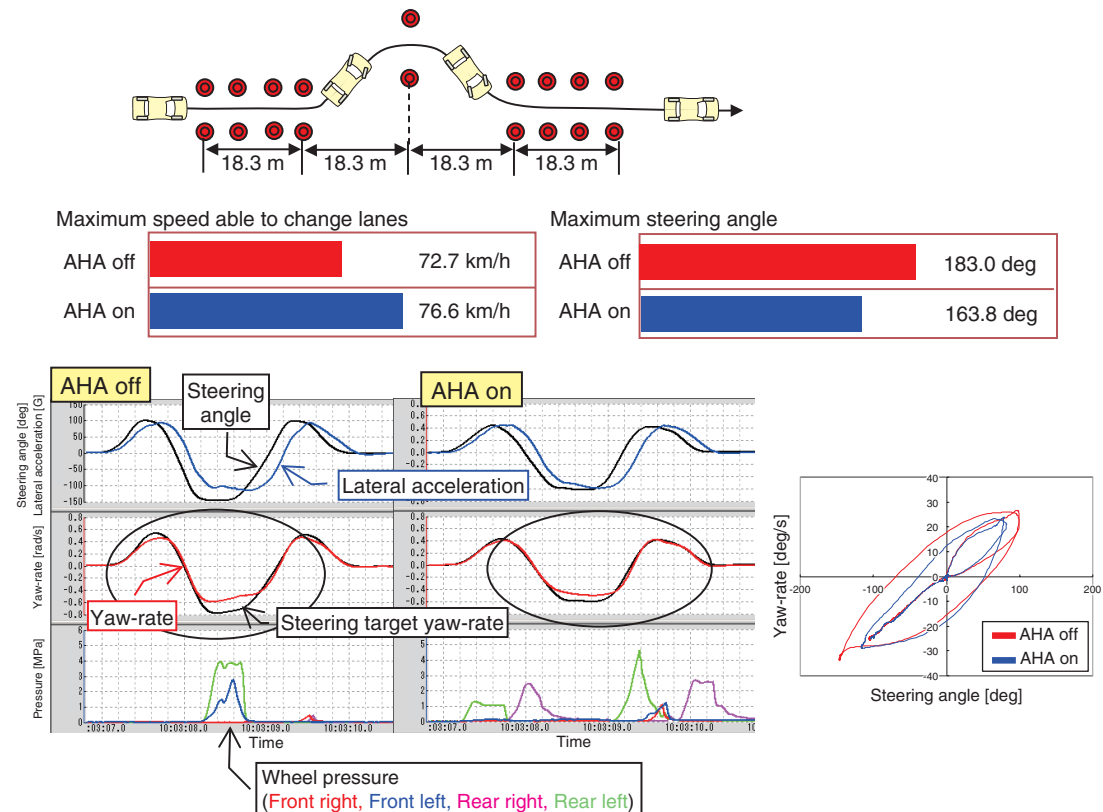


Fig. 11 Dry V-lane change test result

6. Effect of AHA in Combination with P-AWS

AHA and P-AWS have been used in combination in the 2014 model ACURA RLX. In order to help ensure stability at high speed, P-AWS applies control to make the phase of the steering angle of the rear wheels the same as the phase of the steering angle of the front wheels. At low speed, it applies control to make the phase of the steering angle of the rear wheels opposite to the phase of the steering angle of the front wheels in order to increase turn-in ability and create an inward-facing posture. A synergistic effect on handling was observed from the combination of AHA and P-AWS.

6.1. Responsiveness of Lateral Acceleration during Initial Stage of Steering when Inputting Rapid Steering Operation

A test mode in which lanes are changed while turning was used to study the effect of the system in this respect with the results shown in Fig. 12. When P-AWS was used in isolation, the application of rear-wheel reverse

phase control to generate a yaw rate resulted in good turn-in ability. The use of AHA in combination with P-AWS increased the lag in the response of lateral acceleration during high-frequency steering input, and the synergistic effect made it possible to realize ideal turning performance.

6.2. Increased Convergence of Yaw Rate in Relation to Steering Input under Low- μ and Other Limit Range Conditions

The effect of the system was studied using a test mode involving a single lane change on a low- μ road with the results shown in Fig. 13. When P-AWS was used in isolation, the application of rear-wheel reverse phase control to generate a yaw rate resulted in good turn-in ability even on a low- μ road. When AHA was then used in combination with P-AWS, AHA FB control increased the yaw rate convergence in the limit range that is characteristic of rear-wheel reverse phase steering, and the synergistic effect made it possible to realize ideal turning performance.

Test mode of lane change in turning

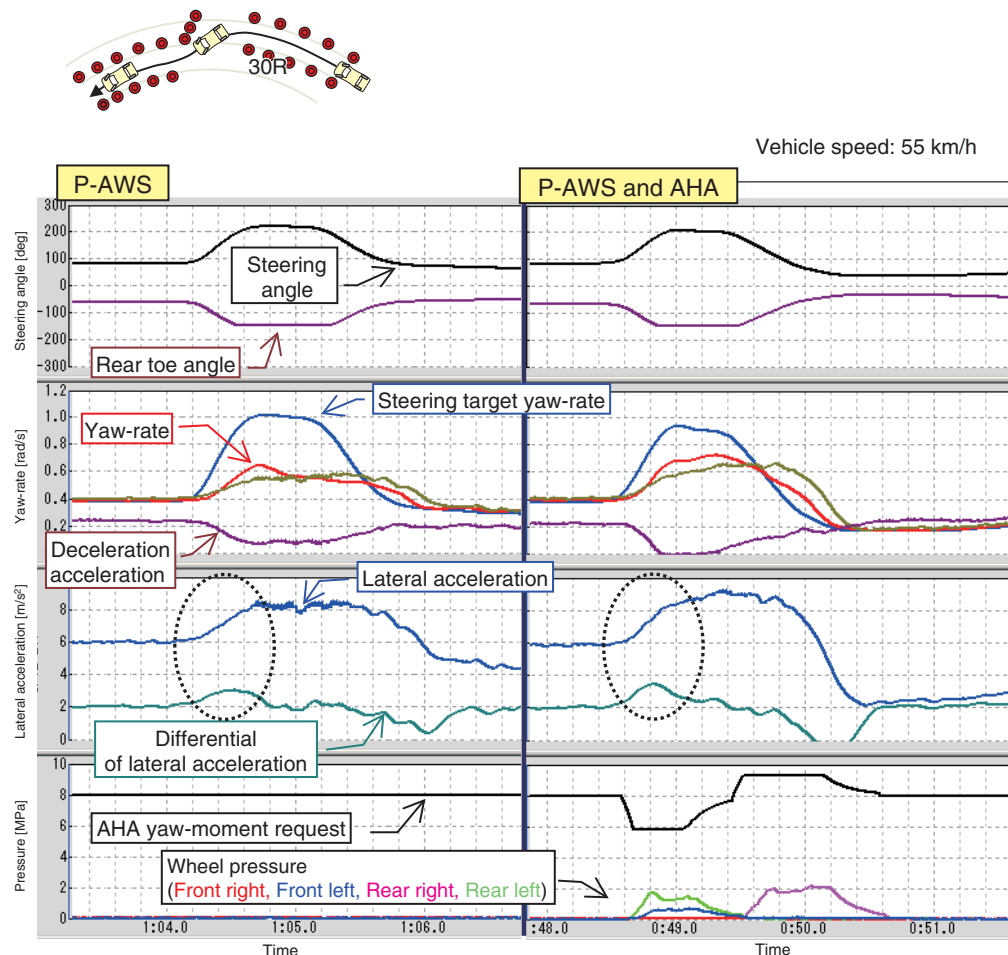


Fig. 12 AHA effect with P-AWS (part 1)

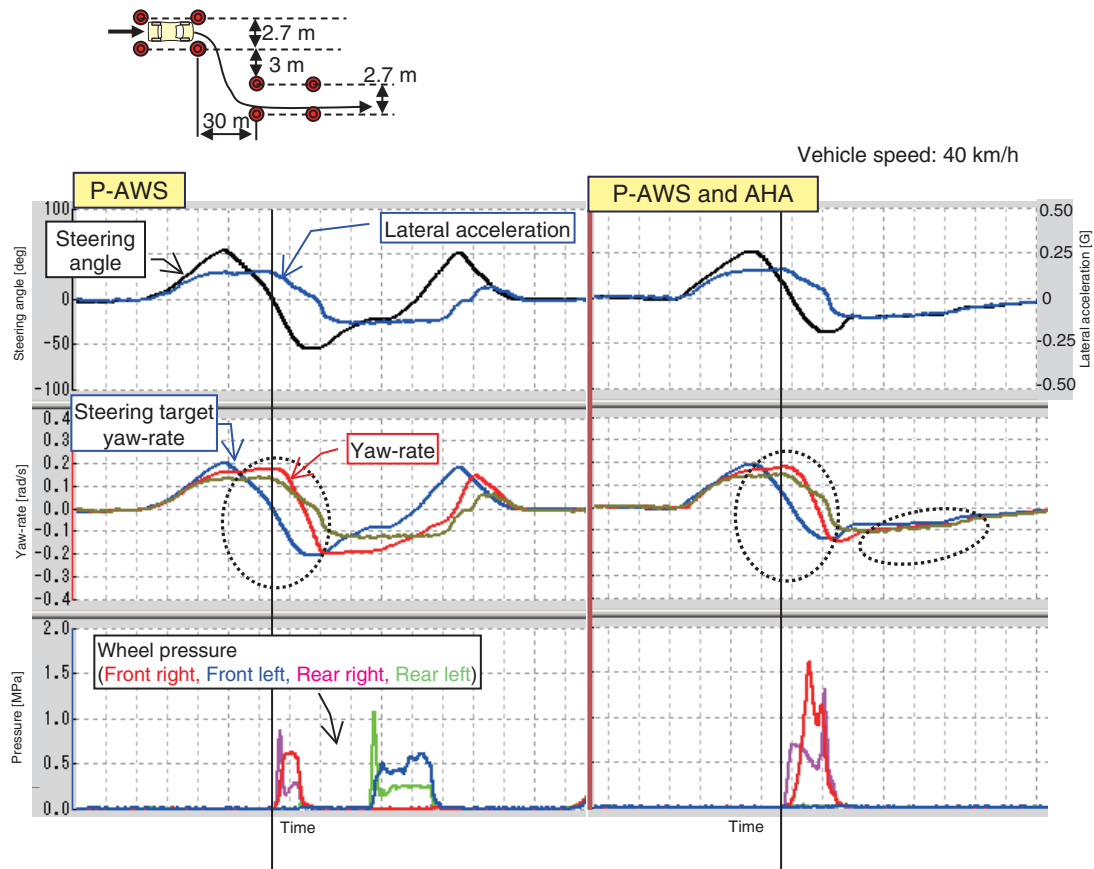
Test mode of low μ lane change

Fig. 13 AHA effect with P-AWS (part 2)

7. Conclusion

The following three aims for the development of AHA were realized, and the system is able to increase responsiveness and stability in relation to the driver's steering operation.

- (1) The system demonstrated an effect in increasing the responsiveness of yaw rate during a J-turn, indicating increased steering responsiveness.
- (2) During slalom driving, the steering-angle yaw-rate characteristic was linear and smooth, and the system thus displays sufficient market appeal.
- (3) Performance was increased in media test modes, indicating increased at-the-limit handling performance. AHA displayed a synergistic effect when used in combination with P-AWS, making it possible to realize ideal turning performance.

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■ Author ■



Takeshi KOJIMA



Tomoyuki FUTAMURA



Kazutaka OHMURA

Noise Reduction Design in Printed Circuit Boards

Takashi NAMBO*

ABSTRACT

To reduce the amount of high-frequency noise generated in automotive electronics, we have compiled circuit board design techniques that are based on the physical properties of high-frequency signals into a set of circuit board design guidelines.

These guidelines were created according to a fundamental approach toward noise reduction, which dictates that high-frequency noise generated as a result of electronic circuit operation must not be allowed to propagate widely through the printed circuit board, and noise radiation sources that resonate with the noise frequency and act as antennas must not be created.

To check the noise reduction effect of the circuit board design guidelines, we used them to redesign the printed circuit board of an information-processing electronic control unit that normally has a large number of FM bandwidth noise suppression parts mounted therein. Using this test ECU, we examined the noise reduction effect resulting from using the redesigned printed circuit board.

As a result, we were able to confirm that an information-processing electronic control unit using the printed circuit board that was designed in accordance with the design guidelines exhibited a noise reduction effect of up to approximately 10 dB μ V/m in the high-frequency bandwidths at and above the FM bandwidth.

1. Foreword

Modern automobiles are equipped with a variety of electronics, and these electronics fulfill a variety of functions. Meanwhile, as greater performance is demanded of automotive electronics, digital circuit operations are becoming faster. As a result, it is becoming important to take measures to suppress the high-frequency noise generated by automotive electronics. There are cases in which developers must struggle to find ways to deal with noise issues that become apparent in the later stages of development. Because trying to deal with an issue after it becomes apparent is time consuming and labor intensive, it would be preferable to take measures to deal with noise in the initial stages of design. One way to do this is to design printed circuit boards in accordance with design guidelines whose noise reduction effects have been verified⁽¹⁾.

We focused on the physical properties of high-frequency signals in printed circuit boards and compiled design principles for reducing high-frequency noise into a set of circuit board design guidelines.

If electromagnetic noise leakage from automotive electronics interferes with an automobile's FM radio, the noise will create audible static and reduce the marketability of the automobile. That is why for the sake of marketability, it is necessary to strictly suppress the electromagnetic noise leaked by automotive electronics in the FM bandwidth (76 MHz to 108 MHz). Thus, automotive electronics designers are keenly

interested in whether circuit board design guidelines are effective for FM bandwidth noise reduction.

For this reason, we examined whether circuit board design guidelines are effective for FM bandwidth noise reduction.

We used the circuit board design guidelines to redesign the printed circuit board of an information-processing electronic control unit (ECU) that normally has a large number of FM bandwidth noise suppression parts mounted therein. Using this test ECU, we examined the noise reduction effect resulting from using the redesigned printed circuit board. The details of the above process are reported below.

2. The Physical Properties of High-Frequency Signals in Printed Circuit Boards

All conductors have inductance. The effects of this inductance can be ignored for low-frequency signals but cannot be ignored for high-frequency signals. Therefore, when thinking of how to reduce high-frequency noise, it is necessary to take into consideration the fact that all the conductor circuit patterns that form circuits on printed circuit boards have inductance. Figure 1 shows the ground plane formed on a printed circuit board, the integrated circuits (IC1 and IC2) mounted thereon, and the circuit pattern that corresponds to the signal transmission path from IC1 to IC2. However, it is assumed that the ground plane is isolated from the positive power supplies of IC1 and IC2, the signal terminals, and the circuit pattern.

* Automobile R&D Center

When a signal is transmitted along the circuit pattern from IC1 to IC2, the feedback current of the signal travels through the ground plane from IC2 to IC1. If the signal is a low-frequency signal, most of the current flows across the shortest distance between the integrated circuits, which is indicated by the purple line. This is because the dominant factor of the impedance of a low-frequency signal is the DC resistance component of the ground plane.

On the other hand, if IC1 is a digital circuit that operates at a high speed and the high-frequency noise generated by the high-speed operation travels through the circuit pattern, most of the feedback current will flow along the path indicated by the blue line in the figure that follows the circuit pattern. This is because the dominant factor of the impedance of a high-frequency signal is the inductance component of the ground plane.

Figure 2 shows how the mutual inductance M between the two inductances L_o and L_r of a two-line signal

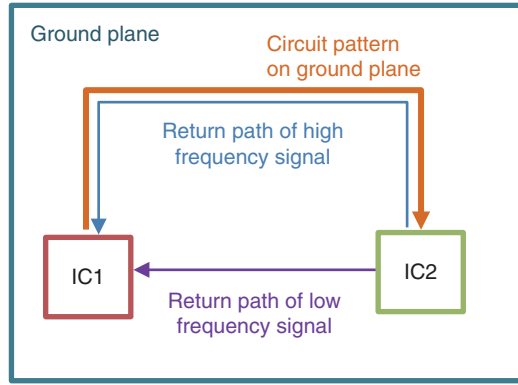


Fig. 1 Return path of signal on printed circuit board

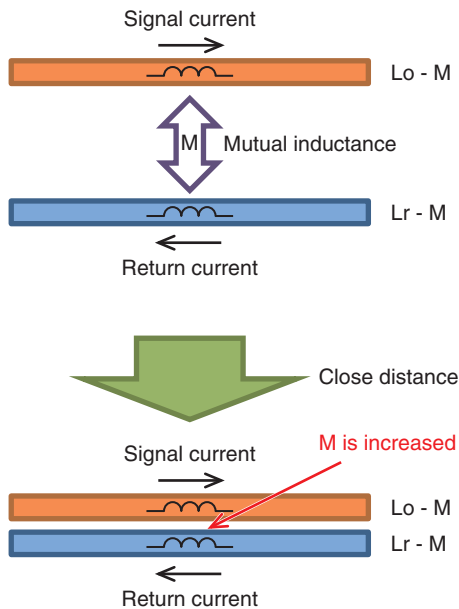


Fig. 2 Inductance of printed circuit with high frequency signal

transmission path increases as the lines get closer. As a result, the inductance on the return path becomes smaller along with $L_r - M$, and the impedance to the high-frequency signal decreases.

In the same manner, the high-frequency impedance in the ground plane is lower closer to the signal line, so most of the high-frequency noise feedback current flows along a path in the ground plane that follows the signal line.

3. Circuit Board Design Guidelines

The circuit board design guidelines that we developed are based on the following objectives: reducing the return-path impedance of high-frequency noise on the basis of the physical properties of high-frequency signals, suppressing common mode current, and preventing the shapes of printed circuit board patterns from leading to antenna formation.

3.1. Conduction Modes and Radiated Noise Strength

A conduction mode is the way that a signal is transmitted. There are two conduction modes: differential mode and common mode. The amount of the noise radiated by a conductor varies depending on the conduction mode of the high-frequency noise traveling through the conductor.

3.1.1. Differential mode radiation

In differential mode, the signal pattern is the forward path, the ground pattern is the return path, and the current orientations in the forward path and return path are reversed (Fig. 3).

The disturbance field strength E_d [V/m] caused by differential mode current I_d can be approximated using Eq. (1).

$$E_d = 1.316 \times 10^{-14} \times \frac{I_d \times f^2 \times S}{d} \quad (1)$$

I_d : Differential mode current [A]

f : Signal frequency [Hz]

S : Area of loop formed by forward path and return path [m^2]

d : Distance to noise measurement point [m]

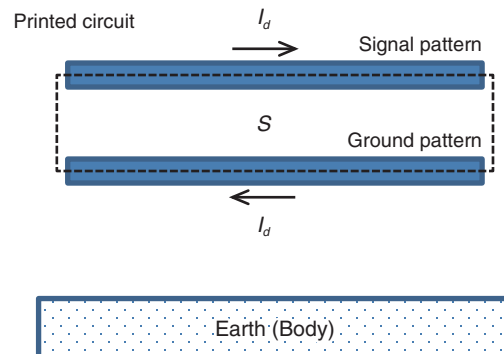


Fig. 3 Simple model of differential mode current

The strength of the disturbance field generated by the differential mode current under the conditions listed below is as follows.

$$\begin{aligned} I_d &= 1 \text{ mA} \\ f &= 100 \text{ MHz} \\ S &= 0.3 \text{ cm}^2 \\ d &= 10 \text{ m} \end{aligned}$$

$$E_d = 0.3948 \text{ } \mu\text{V/m}$$

3.1.2. Common mode radiation

In common mode, the current orientations in the forward path and return path are the same (Fig. 4).

The disturbance field strength E_c [V/m] caused by common mode current I_c can be approximated using Eq. (2).

$$E_c = 1.257 \times 10^{-6} \times \frac{I_c \times f \times l}{d} \quad (2)$$

$$\begin{aligned} I_c &: \text{Common mode current [A]} \\ f &: \text{Signal frequency [Hz]} \\ l &: \text{Wiring length [m]} \\ d &: \text{Distance to noise measurement point [m]} \end{aligned}$$

The strength of the disturbance field generated by the common mode current under the conditions listed below is as follows.

$$\begin{aligned} I_c &= 1 \text{ mA} \\ f &= 100 \text{ MHz} \\ l &= 10 \text{ cm} \\ d &= 10 \text{ m} \end{aligned}$$

$$E_c = 1257 \text{ } \mu\text{V/m}$$

In this example, when the intensity of the noise current is the same, the disturbance field strength of the noise radiation generated by the common mode current is approximately 3000 times greater than that of the noise radiation generated by the differential mode current.

Thus, suppressing the generation of common mode current is an important part of noise reduction.

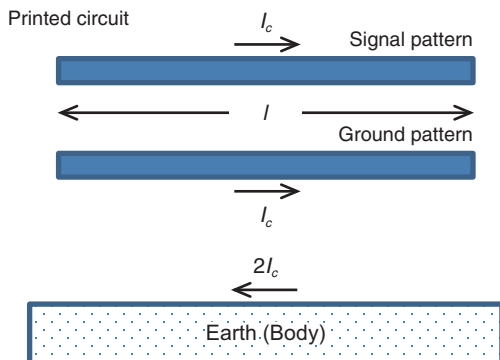


Fig. 4 Simple model of common mode current

Table 1 Conductive pattern length for each frequency

Signal frequency [MHz]	Conductive pattern length [cm]					
	$\lambda/4$	$3\lambda/4$	$5\lambda/4$	$\lambda/2$	$3\lambda/2$	$5\lambda/2$
25	136.8	410.5	684.2	273.7	821.0	1368.4
50	68.4	205.3	342.1	136.8	410.5	684.2
100	34.2	102.6	171.0	68.4	205.3	342.1
200	17.1	51.3	85.5	34.2	102.6	171.0
500	6.8	20.5	34.2	13.7	41.1	68.4
1000	3.4	10.3	17.1	6.8	20.5	34.2

3.2. Antennas Formed by Printed Circuit Board Patterns

When a pattern of a length that resonates with a certain frequency (wavelength) is formed, that pattern may function as an efficient antenna and generate strong noise radiation.

Equation (3) shows how the wavelength on the circuit board corresponds to the frequency.

$$\lambda = \frac{C}{f \times \sqrt{\mu \times \epsilon_r}} \quad (3)$$

$$\begin{aligned} \lambda &: \text{Wavelength [m]} \\ C &: \text{Speed of light [m/s]} \\ f &: \text{Signal frequency [Hz]} \\ \mu &: \text{Relative permeability [-]} \\ \epsilon_r &: \text{Dielectric constant [-]} \end{aligned}$$

Here, on a glass epoxy printed board whose dielectric constant ϵ_r is 4.8, Table 1 shows examples of pattern lengths that produce efficient antennas for different noise frequencies.

As is evident from the table, even on a relatively small printed board, it is highly likely that patterns will be formed that can become antennas for high-frequency noise.

Also, patterns whose lengths are close to those listed in Table 1 can also serve as radiation sources, albeit less efficient ones. Thus, for all bandwidths, including the FM bandwidth, it is necessary to be careful to avoid unintentionally forming patterns that can act as antennas.

3.3. Printed Circuit Board Design Guidelines

On the basis of the two points discussed in sections 3.1. and 3.2., we came up with design principles for suppressing common mode current and preventing circuit board patterns from becoming antennas.

Section 3.3.1. explains the principle for suppressing common mode current on a printed circuit board. Section 3.3.2. provides an example of a guideline that has been determined on the basis of this principle.

3.3.1. Suppressing common mode current

In Fig. 5, IC1 and IC2 are integrated circuits that have

been mounted on a printed circuit board. An electrical signal is transmitted from IC1 to IC2 through a wiring pattern on the printed circuit board. Z_o is the forward path wiring impedance to the electrical signal. Z_r is the return path wiring impedance. Z_c is the earth (vehicle body, etc.) impedance. I_d is the differential mode current, and I_c is the common mode current. As shown in Fig. 5, when the sizes of the forward path current ($I_d + I_c$) and return path current ($I_d - I_c$) are different, a common mode current equivalent to the difference ($2I_c$) is generated.

This means that making Z_r as small as possible makes I_d larger and thus makes I_c smaller.

The circuit board design guidelines provide concrete design principles for decreasing the return-path impedance Z_r to the circuit board with high-frequency signals in the FM bandwidth and above.

3.3.2. Guideline example

In a circuit pattern such as that shown in Fig. 6 in which there is a slit in the ground plane directly beneath the signal line, the high-frequency feedback current tries to proceed along the signal line but is separated and diverted by the slit. As a result, the return path impedance increases, and

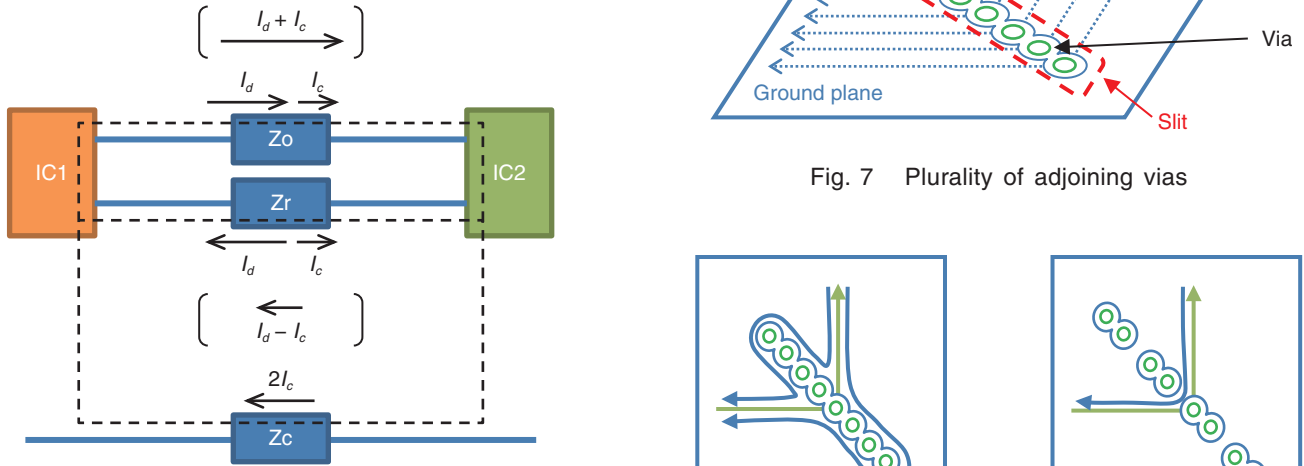


Fig. 5 Common mode current on printed circuit board

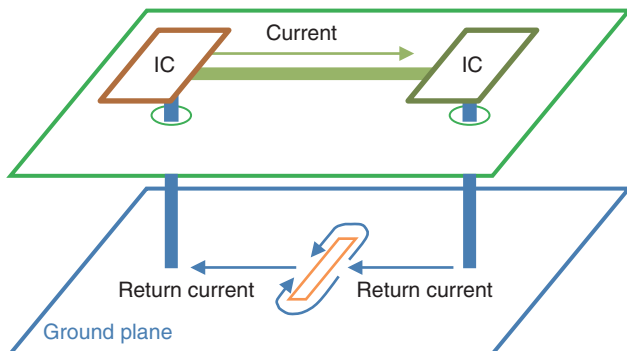


Fig. 6 Effect of slit in ground plane

this results in the generation of common mode current. Also, because the slit makes the electromagnetic field discontinuous, the electromagnetic field widens, and the slit may become a noise radiation source. This is the same phenomenon that occurs in a slot antenna.

Figure 7 provides a concrete example of a wiring design in which this type of large slit may be formed on a printed circuit board from a plurality of adjoining vias.

Because of this phenomenon, the design guidelines stipulate that when there is a plurality of adjoining vias, intervals should be provided between the vias in the manner shown in Fig. 8 so that a return path for the feedback current can be secured.

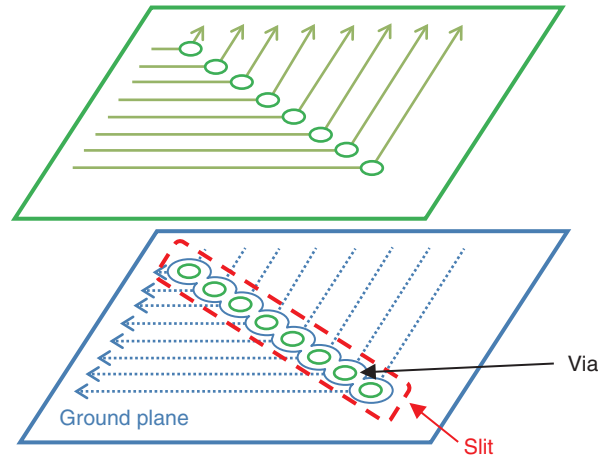


Fig. 7 Plurality of adjoining vias

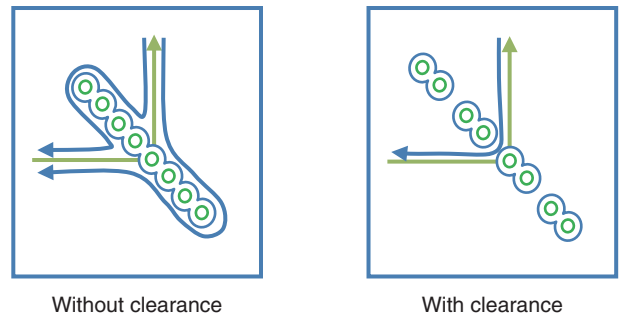


Fig. 8 Secure return path

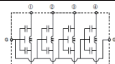
4. Printed Circuit Board for Testing

4.1. Circuit Configuration of the Circuit Board Used in the Noise Comparison Test

To examine whether the circuit board design guidelines are effective for reducing noise, we created the test ECU described below.

We created a test ECU whose circuit board was redesigned according to the circuit board design guidelines

Table 2 Removed noise suppression parts

Type	Equivalent circuit	Quantity
Three-terminal filter		4
Capacitor		4
CR		2
Ferrite bead		6
Ferrite bead array		13
Three-terminal filter array		5

from a conventional ECU, which would normally use a variety of noise suppression parts to reduce FM bandwidth noise. Neither the original ECU nor that with the redesigned circuit board had noise suppression parts mounted thereon. We chose not to use noise suppression parts so that we could compare the noise resulting from the different circuit board designs.

Figure 9 shows the ECU circuit configuration. The circuit comprised a microprocessing unit (MPU), gate array, random access memory (RAM), and power source. The circuit was connected to a wire harness via a connector. The printed circuit board had high-speed signal wiring that included signal buses that could easily serve as noise propagation paths. Also, the circuit board was housed in an aluminum chassis.

4.2. Noise Suppression Parts

Table 2 shows the noise suppression parts that we removed for the purpose of examining the noise reduction effect resulting only from differences in printed circuit board design.

We performed a test to determine whether the test ECU with a printed circuit board that was redesigned according to the circuit board design guidelines (ECU2) exhibited a noise reduction effect compared to the original ECU (ECU1).

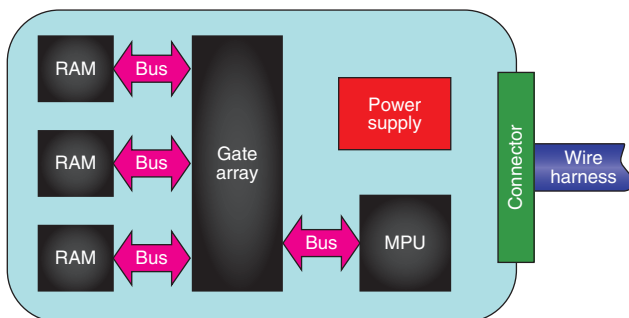


Fig. 9 ECU circuit configuration

5. Results of Noise Reduction Effect Examination

We measured the noise of ECU1 and ECU2 in a 3 m method anechoic chamber that conforms to the CISPR 25 ALSE standard⁽²⁾. The noise measurement results for each ECU are shown in Figs. 10 and 11.

Comparing ECU1 and ECU2, we can see that ECU2 exhibits a noise reduction effect starting from approximately 70 MHz. We were able to confirm that the FM bandwidth noise in Fig. 11 is up to approximately 10 dB μ V/m lower than that in Fig. 10. It is also clear that a noise reduction

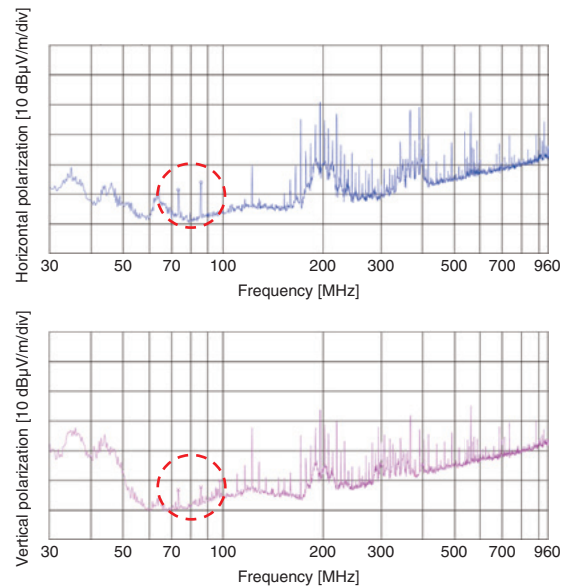


Fig. 10 Noise measurement result (ECU1)

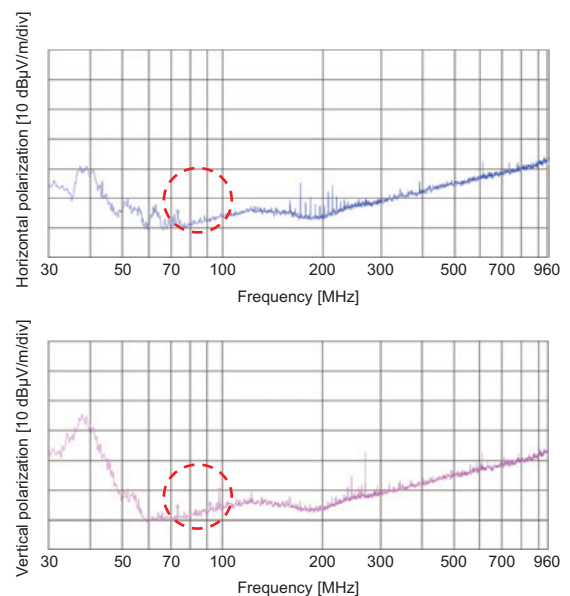


Fig. 11 Noise measurement result (ECU2)

effect is exhibited in bandwidths above the FM bandwidth.

The above results verify that circuit board design techniques that follow the circuit board design guidelines are effective for reducing high-frequency noise in the FM bandwidth and above.

6. Design Process

Appropriately following the circuit board design guidelines requires a significant amount of time for checking and other work. As a result, the necessary design time when the circuit board design guidelines are used is approximately 1.5 times greater than when they are not used. However, we believe that using the guidelines is beneficial from a broader perspective because doing so reduces the number of circuit board revisions and test circuit boards that need to be created in later stages to address noise issues through trial and error.

7. Conclusion

We compiled design principles for printed circuit board design that can be expected to exhibit high-frequency noise reduction effects into a set of circuit board design guidelines.

We were able to confirm that an ECU using a printed circuit board designed according to the circuit board design guidelines exhibited a noise reduction effect of up to 10 dB μ V/m in the FM bandwidth.

There was also a noise reduction effect in high-frequency bandwidths above the FM bandwidth, and we were able to confirm that designing circuit boards in accordance with the circuit board design guidelines is an effective way of dealing with the high-frequency noise that accompanies the high processing speed of ECU circuits.

Circuit board design following the design guidelines is a front-loading design method in which noise countermeasures normally taken in later stages are addressed at the circuit board design stage. The goal of this design is to spend time in the design stage and thereby reduce the time and effort required for countermeasures in later stages. As a result, it is important to understand that the design period will be extended and to create a system for effectively applying the design guidelines.

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■ Author ■



Takashi NAMBO

Development of Material for Lightweight Catalytic Converter

Osamu SUZUKI*

ABSTRACT

High-strength ferritic stainless steel HSS430CU has been developed as a material for use in catalytic converters. This material has superior strength at elevated temperatures and excellent press-forming properties.

HSS430CU does not contain the expensive element Mo that is normally added to the conventional high-strength ferritic stainless steel that is represented in Type444. Instead, Cu is added, which achieves equal or better high-strength properties. This material combines both the workability and strength properties required for catalytic converters. These properties are achieved as follows: superior workability is ensured by incorporating Cu, which is required for high strength, into solid solution when the materials are still raw materials, prior to press-forming; and high strength is ensured once the part is mounted in the vehicle, when heat energy is received as a result of exhaust gas, causing the Cu that was incorporated into solid solution to precipitate as fine ϵ -Cu crystals.

When this material was used in the catalytic converter for the 2014 model year FIT, the high-strength-providing properties of the material were utilized to reduce the plate thickness by 20% or greater. In addition, that superior workability enabled parts that had previously been formed as two pieces to be integrated into one piece, and it reduced weight while simultaneously cutting costs as well.

1. Introduction

Environmental awareness has increased in recent years, and as a result, various industries are pursuing efforts to cut CO₂ emissions. The automobile industry is also investigating the improvement of fuel efficiency through efforts such as increasing power-train efficiency, electrification, and weight reduction. Measures are also being implemented for the exhaust system, which has a great need for weight reduction. These include exchanging iron products with press components that use stainless steel. Presently, the main type of stainless steel plate that is used in catalytic converters is ferritic stainless steel, such as Type409L (11Cr-0.2Ti) and Type429M (14Cr-1Si-0.4Nb). When weight is further reduced by making the plates thinner, endurance reliability must be enhanced, requiring the use of high-strength ferritic stainless steel, represented by Type444 (18Cr-2Mo-0.4Nb). However, Type444 contains Mo, an expensive, rare metal element, inevitably increasing the part cost.

This paper introduces the ferritic stainless steel HSS430CU that was developed, which combines both high strength and high workability and does not contain Mo. It

also introduces the catalytic converter for the 2014 model year FIT that achieved reduction of weight and cost by using HSS430CU.

2. Development Goals

2.1. Catalytic Converter Weight Reduction

The overall structure of the catalytic converter for the 2008 model year FIT is shown in Fig. 1, and the weight percentages of its materials are shown in Fig. 2. Except for some of its components (the catalyst, the catalyst carrier holding mat, the flange, and the nuts), a catalytic converter uses stainless steel, which makes up approximately 70% of the total weight. Therefore, in order to decrease the weight of the catalytic converter, it is important for the weight of the stainless steel parts to be reduced. In order to decrease the weight of the stainless steel parts, it is necessary to make the plates thinner by increasing the strength of the stainless steel itself, and to decrease joint structures by performing integral press-forming of the parts that had been welded as lap joint structures. To accomplish this, the strength of the stainless steel needs to be increased, and workability also needs to be improved to perform integral press-forming of

* Automobile R&D Center

components that are difficult to form. In this development, integral press-forming was investigated with focus on Part A shown in Fig. 1.

2.2. High-Strength Stainless Steel Development

There are two types of stainless steel used in exhaust system parts: ferritic and austenitic. The coefficient of thermal expansion of austenitic stainless steel is approximately 1.5 times that of ferritic stainless steel. Therefore, when the austenitic type is used as the material for a catalytic converter, which undergoes cooling and heating cycles from high to low temperatures during vehicle operation, it is repeatedly exposed to high thermal stress, making durability an issue. Also, austenitic stainless steel contains Ni, which makes it expensive. As a result, a high-strength ferritic stainless steel similar to Type429M, which is the ferritic stainless steel currently used in mass-produced vehicles, has been developed.

First, four targets below in relation to the properties required for the material that is used in a catalytic converter were established. Because the stress that is generated increases as a result of decreasing the thickness of the

plates, strength at elevated temperatures was sought that was equivalent to or better than that of the pre-existing high-strength ferritic stainless steel Type444. We aimed for a material with oxidation resistance and corrosion resistance that were equal to or better than those of the conventional Type429M, since the environment in which the material was to be used was unchanged in terms of the demand for these resistance properties. Also, to reduce the weight and cost of the material, we aimed for press-forming properties that enabled the number of parts in the converter to be reduced as workability targets.

- (1) Strength at elevated temperatures: equivalent to or better than Type444
- (2) Oxidation resistance: equivalent to or better than Type429M
- (3) Corrosion resistance: equivalent to or better than Type429M
- (4) Workability: press-forming properties that enabled the number of parts used in the converter to be reduced.

3. Alloy Design Concept

As stated in chapter 1, the high-strength ferritic stainless steel, represented by Type444, contains the expensive chemical element Mo. When Mo is incorporated into solid solution as a metal, it serves as a chemical element that reinforces the solid solution and helps ensure strength at elevated temperatures. In addition, Mo is known to be effective at increasing oxidation resistance and corrosion resistance^{(1), (2)}. However, in order to reduce costs, Mo was not added to the material. Instead of using Mo as the means of increasing strength at elevated temperatures, focus was placed on Cu for the following reasons:

- (1) The cost of Cu is relatively stable;
- (2) Workability is good when the stainless steel material contains Cu in the form of a solid solution.; and
- (3) After the catalytic converter containing the material is installed in the vehicle, the Cu that had been incorporated into solid solution precipitates out as fine ϵ -Cu crystals as a result of heat energy from exhaust gas.

In order to achieve increased workability, which was the target of this development, the two property values, the elongation value and r -value, must be increased.

Figure 3 shows the relationship between the elongation and tensile strength of ferritic stainless steel (plate thickness 1.5 mm). Materials with high tensile strength have low elongation. Conversely, materials with high elongation have low tensile strength. Therefore, it is known to be difficult to simultaneously achieve both strength and elongation.

Meanwhile, ferritic stainless steel containing Cu has relatively good elongation when the Cu is incorporated into solid solution, but once the high-temperature region (from approximately 500°C to 700°C) is reached, a fine precipitant (ϵ -Cu) is formed, decreasing elongation and increasing strength⁽³⁾. Therefore, when press forming, which requires high elongation, good elongation is ensured by Cu being

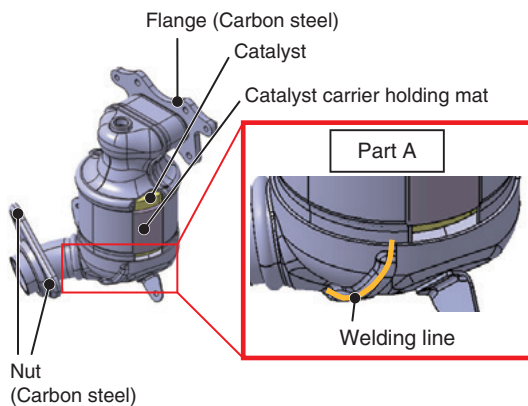


Fig. 1 2008 model year FIT catalytic converter

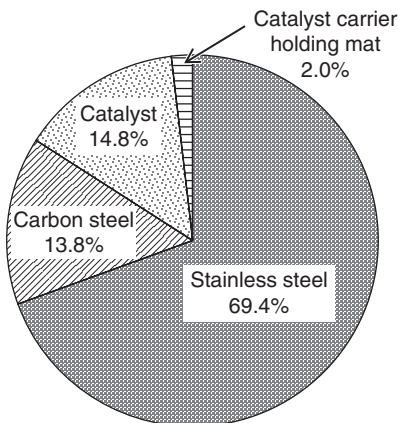


Fig. 2 Weight percentages of materials of 2008 model year FIT catalytic converter

incorporated into solid solution; and after the converter is installed in the vehicle, which is when high strength is required, strength can be increased by the ϵ -Cu precipitation effect caused by the heat energy from the exhaust gas.

Similar to elongation, the r -value is an index for evaluating workability. When tensile deformation of a material occurs, the r -value expresses whether it tends to contract in the thickness direction or the width direction. The r -value is expressed by Eq. (1).

$$r = \ln(W_1/W_0)/\ln(t_1/t_0) \quad (1)$$

r : r -value

W_0 , W_1 : The width of a test strip before and after tensile deformation

t_0 , t_1 : The thickness of a test strip before and after tensile deformation

Figure 4 shows the effect of the r -value on deep-drawability. The larger the r -value of a material, the more a deeply drawn shape can be made^{(4),(5)}. Therefore, the r -value is particularly important for a catalytic converter having a drawn shape. Figure 5 shows the effect of plate thickness on the r -value of Type429M. As the figure shows, since the r -value increases as the plate thickness decreases, it was expected that decreasing the plate thickness by using high-strength materials would increase r -value as a result. In

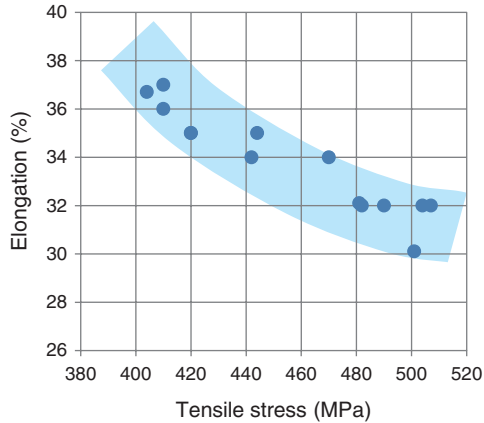


Fig. 3 Relationship between elongation and tensile stress

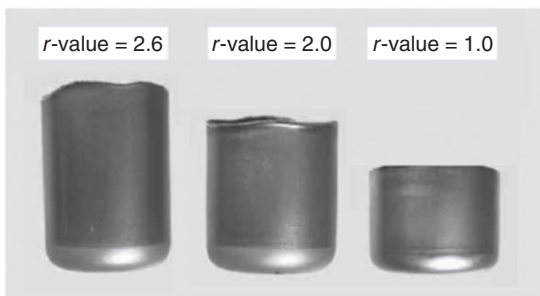


Fig. 4 Effect of r -value on deep-drawability

Table 1 Chemical composition (mass%)

	C	Si	Mn	Cr	Mo	Cu	Nb	N
Developed steel	0.007	0.8	0.2	17	-	1.3	0.5	0.007
Type429M	0.008	1.0	0.3	14	-	-	0.4	0.009
Type444	0.009	0.3	0.2	18	2	-	0.4	0.010

addition, we aimed to improve the r -value further by using a means to decrease C and N⁽⁶⁾, which prevent the r -value from increasing.

The addition of Cu is known to interfere with oxidation resistance. To improve oxidation resistance, it is essential to reinforce the oxide layer that exists on the surface of the stainless steel. Therefore, in this development effort, we aimed to help ensure oxidation resistance by increasing the quantity of Cr, which is effective at providing oxidation resistance.

It is known that the Pitting resistance equivalent (PRE) expressed by Eq. (2) can be used to estimate corrosion resistance⁽⁷⁾, and by increasing the quantity of Cr, which is the measure taken to increase oxidation resistance, it is also possible to help ensure corrosion resistance.

$$PRE = Cr + 3.3Mo \text{ (wt\%)} \quad (2)$$

The representative values of chemical composition established in the abovementioned investigation are shown in Table 1. The characteristics of the developed material composition are as follows.

- (1) In order to provide improved strength over the pre-existing material Type429M, Cu was added, and in order to ensure oxidation resistance and corrosion resistance, the quantity of Cr was increased.
- (2) The expensive chemical element Mo that is normally added to high-strength ferritic stainless steel was not added.
- (3) In order to improve workability, C and N were decreased to the extent possible.

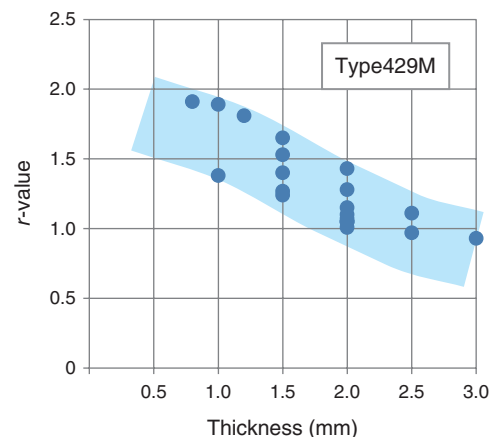


Fig. 5 Effect of plate thickness on r -value

4. Material Characteristics

4.1. Strength at Elevated Temperatures

Strength at elevated temperatures is a characteristic that is important when decreasing the thickness of stainless steel plates. In addition to the engine and other components conveying vibration to the exhaust system parts, in the high-temperature environment, thermal stress is produced as a result of engine heating and cooling cycles. Therefore, the material to be used at high temperatures must have both the properties of high-cycle fatigue strength and thermal fatigue strength.

Figure 6 shows the changes in hardness when thermal processing was performed at 600°C. Type429M, which does not contain Cu, does not have any changes in hardness and exhibits a constant value. In contrast, in the developed material, ϵ -Cu precipitates during thermal processing, and an increase in hardness is observed as a result.

Figure 7 shows the results of the high-temperature tensile test. Due to ϵ -Cu precipitation, the developed material has higher tensile strength than Type429M in all temperature regions, and it retains equal or better strength

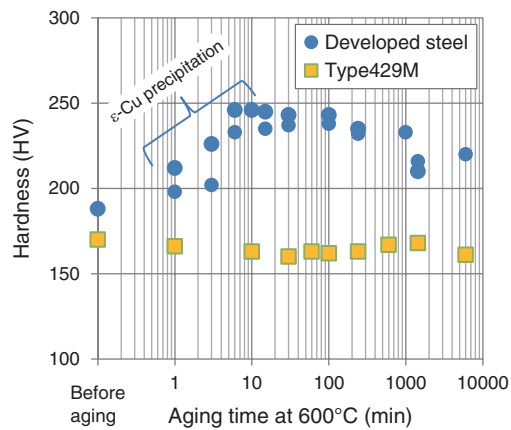


Fig. 6 Change in hardness at 600°C

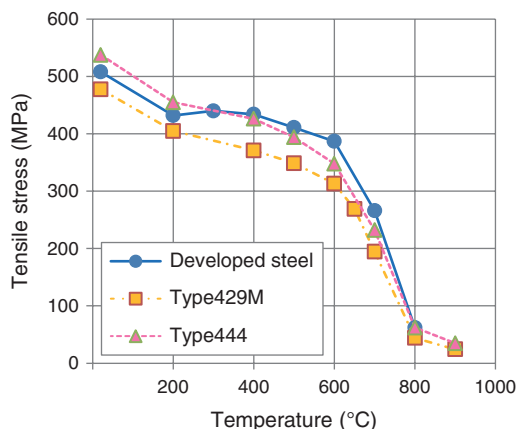


Fig. 7 Result of high-temperature tensile test

compared to the high-strength ferritic stainless steel Type444.

Figure 8 shows the results of the completely reversed bending high cycle (10^7 times) fatigue test at 800°C. The fatigue limit of the developed material improved by 20% or more over that of Type429M, and the material retained tensile strength that was equal to or better than that of Type444.

Figure 9 shows the results of the thermal fatigue test conducted under conditions in which cooling and heating cycles from 100°C to 700°C were added. The thermal fatigue life of the developed material was 40% longer than that of Type429M, and it retained a lifetime equal to or better than that of Type444.

4.2. Oxidation Resistance

Because stainless steel suddenly oxidizes when a limit temperature is exceeded⁽⁸⁾, that temperature is called the oxidation-resistant marginal temperature, and it determines the limits for use of a material. Figure 10 shows the oxidation-resistant marginal temperature of various materials in the atmosphere. Because the oxidation-resistant marginal temperature of the developed material is 950°C, it is equal to or higher than that of Type429M, and the material exhibits sufficient oxidation resistance for use in a catalytic converter.

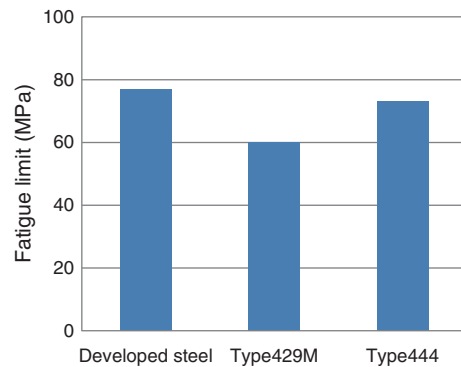


Fig. 8 Fatigue limit at 800°C

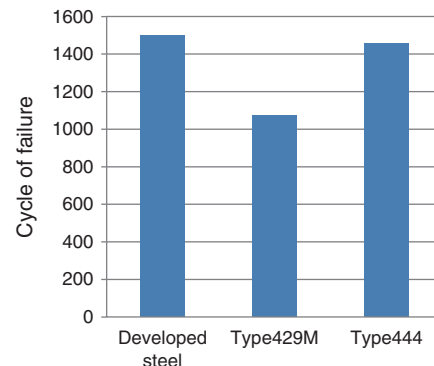


Fig. 9 Result of thermal fatigue test

Table 2 Pitting resistance equivalent (*PRE*)

	Cr	Mo	<i>PRE</i>
Developed steel	17	0	17
Type429M	14	0	14
Type409L	11	0	11
Type444	18	2	25

PRE = Cr + 3.3Mo (mass%)

4.3. Corrosion Resistance

As stated in chapter 3, the corrosion resistance of stainless steel is known to be correlated with *PRE*. As *PRE* increases, corrosion resistance improves. Table 2 shows the *PRE* of various materials for exhaust systems. The *PRE* of the developed material is higher than it is for Type429M, so superior corrosion resistance is expected. An evaluation of corrosion resistance was performed using the pitting potential method specified in JIS G 0577, and the pitting potential measurement results for various materials are shown in Fig. 11. Corrosion resistance increases as pitting potential increases. The pitting potential of the developed material is equal to or greater than that of Type429M, and the material demonstrated sufficient corrosion resistance for use in a catalytic converter.

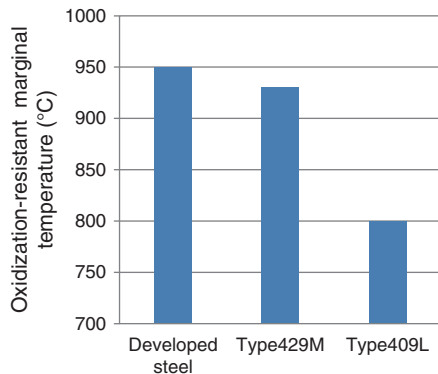


Fig. 10 Oxidation-resistant marginal temperature

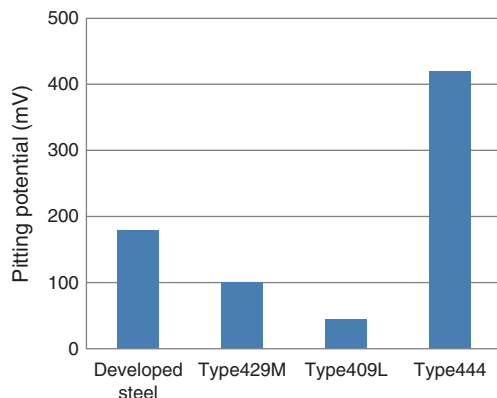


Fig. 11 Result of pitting potential measurement

Table 3 Test result of *r*-value

	Thickness (mm)	<i>r</i> -value			
		<i>r</i> ₀	<i>r</i> ₄₅	<i>r</i> ₉₀	<i>r</i> _m
Developed steel	2.0	1.38	1.23	1.98	1.46
	1.5	1.74	1.26	2.36	1.65
Type429M	2.0	1.15	0.88	1.49	1.10
	1.5	1.30	0.96	1.85	1.27
Type444	2.0	1.01	0.96	1.29	1.06
	1.5	1.30	0.87	1.69	1.18

$$r_m = (r_0 + r_{90} + 2r_{45})/4$$

4.4. Workability

r-value measurement results are shown in Table 3. *r*₀, *r*₄₅, and *r*₉₀ are the angles 0°, 45°, and 90°, respectively, with respect to the rolling direction. *r*_m is the mean *r*-value. The results of the test are expressed in Eq. (3).

$$r_m = (r_0 + r_{90} + 2r_{45})/4 \quad (3)$$

When C and N are reduced and comparisons are made at the same plate thickness, the *r*-value of the developed material is higher than that of Type429M. Also, because we were able to decrease the plate thickness from 2.0 mm to 1.5 mm as a result of increasing the strength of the material, we achieved an even higher *r*-value. Utilizing this high *r*-value, integral press-forming of one part of the converter (Fig. 12, Part A) was investigated. Figure 13 shows the results of the press-forming test of Part A. Although the

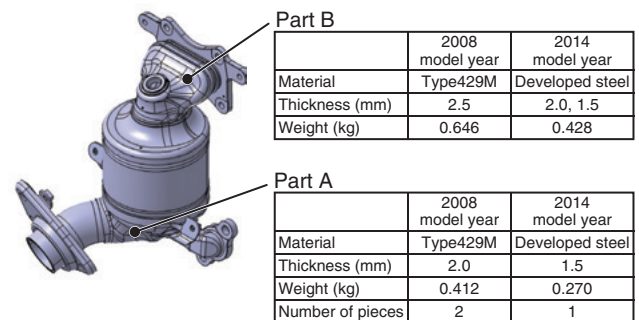


Fig. 12 2014 model year FIT catalytic converter

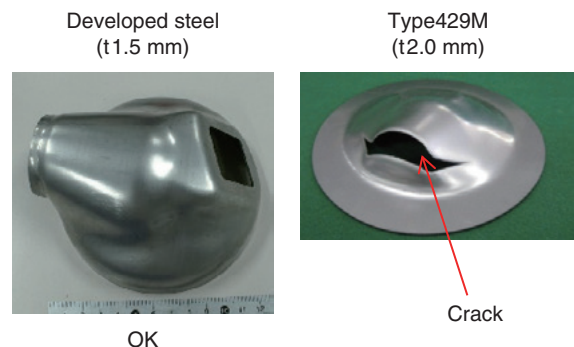


Fig. 13 Press-forming result of part A

part breaks during press-forming with Type429M, we were able to press-form the part when using the developed material. Because Type429M cannot be used to perform integral press-forming of this part, conventionally it had been produced by welding together a 2-piece press part. However, using the developed material, it was possible to perform integral press-forming of the part to replace it with a one-piece press part.

During complicated press-forming, materials are locally extended, decreasing the plate thickness. However, with a higher rate of decrease, the plate thickness becomes thin locally, which is not desirable from the perspective of endurance reliability and so on. Therefore, it is preferable to minimize the rate of decrease of the plate thickness as much as possible. Figure 14 shows the measurement results of the plate thickness decrease rate after the press-forming of Part B. The value of the plate thickness decrease rate is obtained using Eq. (4).

$$D = (t_{p0} - t_{p1}) / t_{p0} \times 100 (\%) \quad (4)$$

D : Plate thickness decrease rate

t_{p0} , t_{p1} : Plate thickness before and after press-forming

Both the developed material and Type429M are capable of forming Part B. However, the developed material had the lowest plate thickness decrease rate (plate thickness 2.0 mm). The second-lowest was Type429M (plate thickness 2.0 mm), and Type429M (plate thickness 2.5 mm) was the highest. The plate thickness decrease rate following press-forming is correlated to the r -value of a material. We observed that the high r -value of the developed material minimized the plate thickness decrease rate during press-forming.

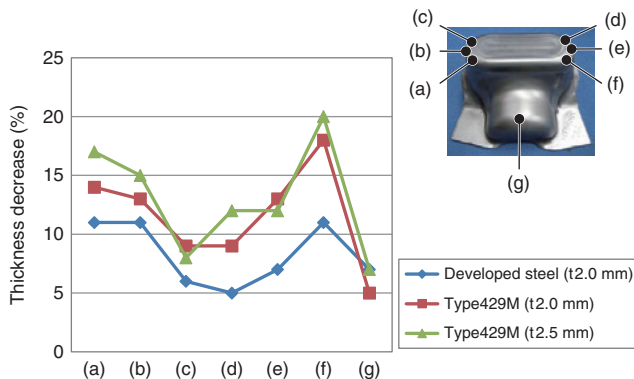


Fig. 14 Thickness decrease after part B press-forming

5. Weight Reduction Effect

Plate thickness was decreased by 20% or more by using the developed material in two parts of the catalytic converter (part A and part B) for the 2014 model year FIT shown in Fig. 12. Integrated press-forming was also

performed for Part A. As a result, the weight of the parts where the material was used was reduced by a total of 0.36 kg.

Also, the weight reduction of material while simultaneously cutting costs was achieved by doing the following: developing a high-strength ferritic stainless steel that did not contain Mo, decreasing the quantity of material used by decreasing plate thickness, and reducing the press-molding cost and decreasing welded parts by integrating two pieces into one.

6. Conclusion

HSS430CU, which has superior strength at elevated temperatures and excellent press-forming properties, thereby decreasing the weight and cost of the catalytic converter for the 2014 model year FIT, has been developed.

- (1) The high-strength ferritic stainless steel HSS430CU demonstrated good workability, as a result of Cu being incorporated into solid solution during forming. This material also exhibited strength at elevated temperatures following vehicle installation, as a result of fine ϵ -Cu precipitation caused by thermal energy from exhaust gas.
- (2) The weight of the catalytic converter was reduced by decreased plate thickness with application of high-strength material and integrated press-forming with increased r -value.
- (3) The cost was reduced by not adding Mo and by performing integral press-forming of parts to reduce welded parts.

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■ Author ■



Osamu SUZUKI

Development of Low-Nickel Heat Resistance Alloy for Exhaust Valves

Takashi TSUYUMU*
Akihiro TOJI***

Shigeki UETA**
Katsuhiko OHISHI***

Mototsugu OSAKI**

ABSTRACT

The DY2 heat-resistant alloy was developed with lower nickel content than in the conventional heat-resistant nickel-based super alloy for use in exhaust valves. In order to compensate for the diminished strength that accompanies reduction in the nickel content, steps were taken with this alloy to optimize the additive elements (carbon, nitrogen, phosphorus, molybdenum) that contribute to solid solution hardening, increasing the quantities of these elements within the range that would not impair this material's hot workability. DY2 has reliable high-temperature strength and wear resistance, can be used to manufacture valves by hot forging, and achieved a 20% reduction in materials costs.

1. Introduction

Engine exhaust valves have the role of maintaining the airtight seal of the combustion chamber while being exposed to combustion gases at high temperatures and high pressures. They are important components, since deterioration of the airtight seal can lead to loss of power or loss of engine functionality. Given their function, exhaust valve materials require strength and wear resistance at high temperatures as well as corrosion resistance and oxidation resistance in the presence of combustion products.

In general, high-manganese heat-resistant steel SUH35 (21-4N steel) has been widely used in exhaust valves for automobiles. Since this material does not have adequate strength and hardness at high temperatures, alloy thermal spray (hard-facing) is applied to the valve seat face as necessary for high performance engines, or a highly heat-resistant material such as heat-resistant nickel-based super alloy is used.

Honda has consistently used heat-resistant nickel-based super alloy for automobile exhaust valve material since 1979. The company has developed material for exhaust valves to meet the particular needs of the times, and has been using heat-resistant nickel-based super alloy for automobile exhaust valves for the following reasons:

(1) It eliminates the need for hard-facing the valve seat face.

(2) It has reliable high-temperature strength and enables the valve stem to be made thinner.

(3) It has reliable oxidation resistance.

In order to reduce the costs of exhaust valves for automobiles, NCF3015 (30Ni-15Cr) has been used in mass-production vehicles and high-nickel heat-resistant alloy NCF440 (70Ni-19Cr) has been used in high-performance engines. Exhaust valves have been facing rising exhaust gas temperatures with the adoption of increasingly stringent exhaust emission regulations and heightened fuel economy measures. Greater strength at high temperatures is being required even in mass-production vehicles, while at the same time there are also calls for resource conservation and cost reductions. Rare metals have become essential materials for society today as motor vehicles and electronics take on increasing functionality, and the importance of these materials is steadily growing. At the same time, however, technology for resource conservation and recycling has become an important issue from the perspective of measures to curb the depletion of resources.

In order to meet these demands, the heat-resistant alloy DY2 was developed. Its purpose is to achieve a lower nickel content than in the present heat-resistant nickel-based super alloy NCF3015 and to reduce costs while also providing equal high-temperature strength and wear resistance.

* Automobile R&D Center

** Daido Steel Co., Ltd.

*** Hitachi Metals, Ltd.

2. Development Goals

In taking steps to reduce the nickel content, the characteristics sought for the valve material were defined as equal to the heat-resistant nickel-based super alloy currently used as material in mass production (NCF3015). Development of low-nickel heat-resistant alloy content was pursued with the following four points as the main aims.

- (1) High-temperature fatigue strength and wear resistance at 800°C.
- (2) Oxidation resistance
- (3) Corrosion resistance
- (4) Hot workability

3. Alloy Design

Conventional high-nickel heat-resistant alloy is material made for greater strength and wear resistance at high temperatures by the precipitation of gamma-prime phase $\text{Ni}_3(\text{Al}, \text{Ti})$. Since nickel is the basic element composing the gamma-prime precipitation phase, which is the hardening phase, increased strength becomes less readily achieved through the use of gamma-prime phase when the nickel content is reduced, and the strength is diminished. By contrast, the developed material DY2 has reduced nickel content while simultaneously assuring high-temperature strength, wear resistance, and hot workability. The alloy design was therefore carried out using the following two types of strengthening technique.

DY2-D: Solid solution by the addition of molybdenum and carbide precipitation-hardening through an increase in the amount of carbon and the addition of niobium.

DY2-H: Grain boundary precipitation-hardening through the addition of phosphorus

3.1. DY2-D Material

Making a solid solution with molybdenum or tungsten and precipitation-hardening through the addition of carbides are effective ways⁽¹⁾ of making high-strength material from the austenite heat-resistant steel generally used as a heat resistance alloy.

The addition of molybdenum to the DY2-D alloy increases high-temperature strength by solid solution-strengthening, but larger amounts will cause embrittlement phase to precipitate and diminish hot workability⁽²⁾. Based on Fig. 1, concern over the precipitation of a Laves phase, which is an embrittlement phase, arises when the molybdenum content exceeds 3.5% or when the tungsten content exceeds 3.2%. Furthermore, the strengthening amount of elements for strengthening a solid solution is generally related to the atomic concentration, and as the dotted line in the figure indicates, tungsten, which has approximately twice the atomic weight of

molybdenum, has approximately one-half the solid-solution effect even at the same mass concentration. For DY2-D, therefore, the choice was made to add molybdenum alone, for its greater effectiveness by comparison with the addition of tungsten alone or in combination with molybdenum, and the amount of 3.2% molybdenum was decided on.

Increasing the amount of carbon in order to utilize carbide precipitation-hardening is effective in promoting the carbide precipitation of M_{23}C_6 -type material (chromium carbide) and increasing its high-temperature strength and wear resistance. However, the amount of carbon needs to be set within the range that will not degrade toughness. Since adding a small amount of niobium is effective in enhancing toughness⁽³⁾, 0.1% of niobium was added to DY2 and a portion of the carbon was replaced by the carbide precipitation of MC-type material (niobium carbide). When the amount of carbon exceeds 0.6%, hot workability is degraded. In consideration of productivity, therefore, the amount of carbon was set at 0.52%. MC-type carbide is effective in heightening the wear resistance of the valve seat face area, and one feature of this alloy, as shown in Fig. 2, is that in DY2-D, the total amount of carbide precipitation is increased and the composite reinforcement by the solid solution is further increased, yielding greater heat resistance.

Since molybdenum and niobium are ferrite stabilizing elements, reducing the amount of nickel raises concerns regarding the precipitation of delta ferrite phase when manufacturing the material. When delta ferrite phase is precipitated, the nitrogen that is a component will cause

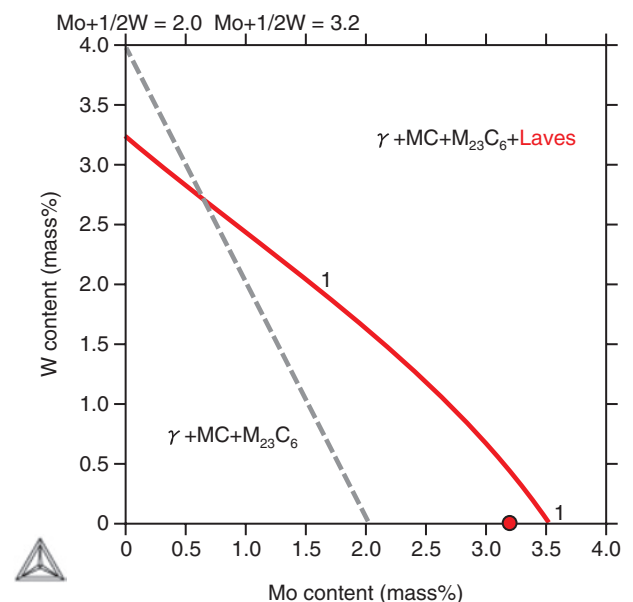


Fig. 1 Phase diagrams of Fe-0.52C-0.1Si-9Mn-6Ni-0.1Nb at 800°C

nitrogen-blow when melting, which may result in the formation of defects within the material. Therefore, there is a need to prevent reduction in microstructural stability. Figure 3 shows the nickel content on the horizontal axis and the temperature on the vertical axis. Since delta ferrite is precipitated during manufacturing of the material when the nickel content is 4% or lower, casting doubt on microstructural stability, the amount added in DY2-D was set at 6% to suppress the precipitation of delta ferrite.

3.2. DY2-H Material

A solid solution with molybdenum is effective in increasing high-temperature strength, but molybdenum is a rare metal and is also high in cost.

In DY2-H, grain boundary precipitation (phosphorus)-hardening, which is effective in increasing high-temperature fatigue strength, was used and the solid solution with molybdenum was reduced. The setting of

the phosphorus content will be discussed next. As shown in Fig. 4, increasing the added amount of phosphorus causes M_3P to form and high-temperature strength is enhanced, but when the amount becomes excessive, the grain boundary becomes brittle and hot workability is diminished. In consideration of manufacturability, therefore, the amount of phosphorus was set at 0.22%, where microstructural stability is reliable. Figure 5 shows the total precipitation amounts of carbides and phosphides.

The chemical compositions of the DY2-D and DY2-H produced by the above alloy design as well as the chemical compositions of existing heat-resistant nickel-based super alloys are shown in Table 1. The material DY2 has a nickel content 20 or more points lower than the high-nickel heat-resistant alloy NCF3015, and also possesses greater high-temperature strength.

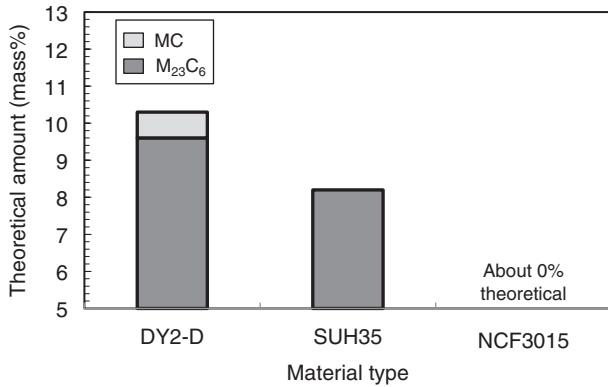


Fig. 2 Theoretical amount of carbides at 800°C

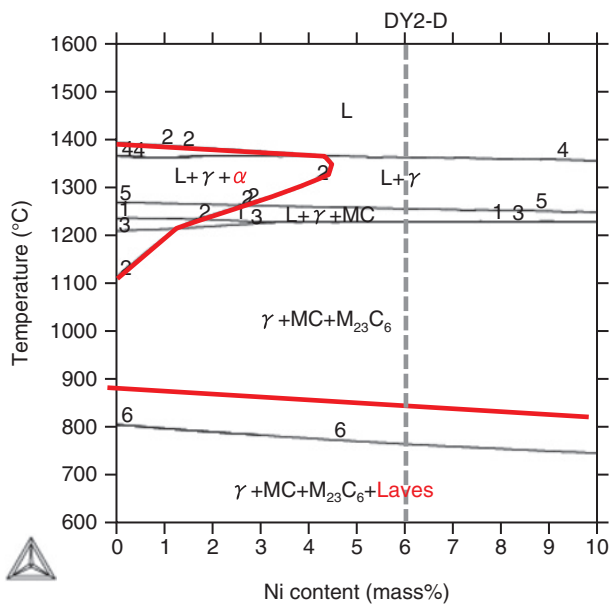


Fig. 3 Phase diagrams of Fe-0.52C-0.1Si-9Mn-3.2Mo-0.1Nb

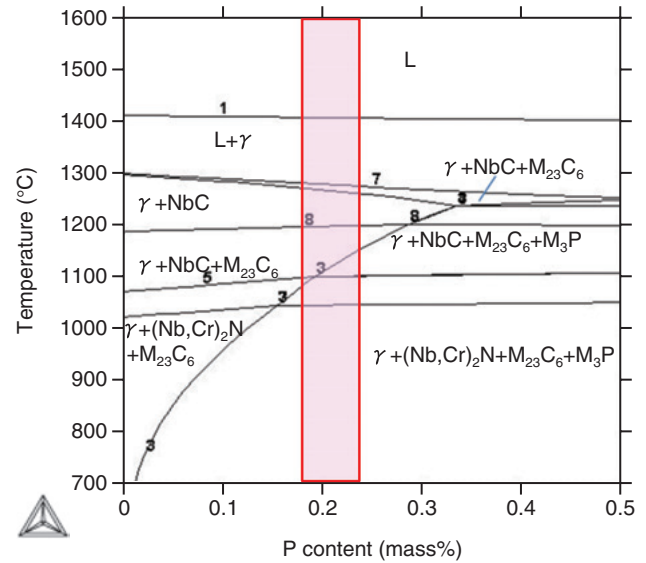


Fig. 4 Phase diagrams of Fe-0.33C-0.30Si-1Mn-0.5Mo-10.5Ni-0.2%Cu-0.12Nb-0.18N

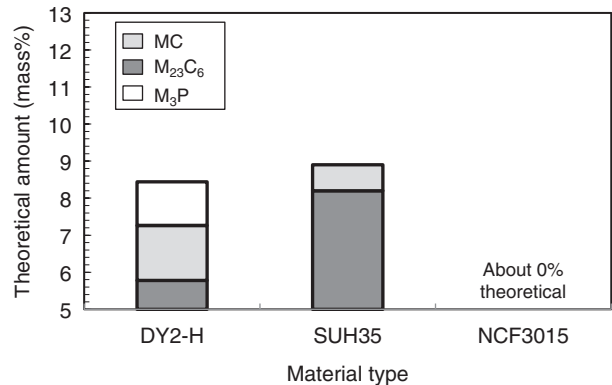


Fig. 5 Theoretical amount of carbides and phosphide at 800°C

Table 1 Nominal chemical composition

Material	Composition (mass%)												
	C	N	Si	Mn	Ni	Cr	Fe	Mo	Ti	Al	Nb	P	Other
DY2-D	0.52	0.40	0.10	9.0	6	21	Bal.	3.2	-	-	0.10	-	
DY2-H	0.33	0.18	0.10	1.0	10.5	20	Bal.	0.5	-	-	0.12	0.22	
SUH35	0.50	0.40	0.10	9.0	4	21	Bal.	-	-	-	-	-	
NCF3015	0.03	-	0.10	0.10	32	16	47	-	2.70	1.20	0.80	-	0.004B

4. Phase Stability

Figure 6 shows a photograph of the structure following solution and aging heat treatment of the developed materials at 800°C for 400 hours. It is apparent from the structure photograph that Laves phase and delta ferrite phase precipitation, which reduce the materials' strength, are not present after aging heat treatment at 800°C for 400 hours.

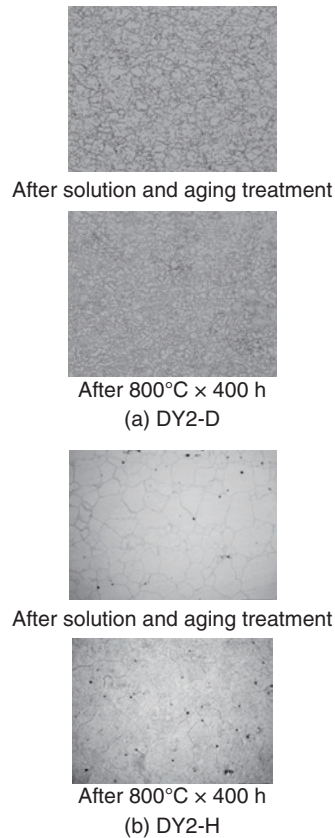


Fig. 6 Microstructure of developed materials

5. Material Characteristics

5.1. High-Temperature Fatigue Strength

High-temperature fatigue strength is an important characteristic indicating the fatigue strength of the valve stem. Figure 7 shows the results of rotary bending fatigue

testing at 800°C and a speed of 3500 rpm. DY2-D and DY2-H demonstrate fatigue strength on a par with that of the target heat-resistant nickel-based super alloy NCF3015 at 800°C.

5.2. Wear Resistance

High-temperature hardness is an important characteristic value that indicates the wear resistance of the valve seat face. Figure 8 shows the hardness of DY2-D and DY2-H from room temperature to high temperature. The developed material has greater hardness than SUH35 from room

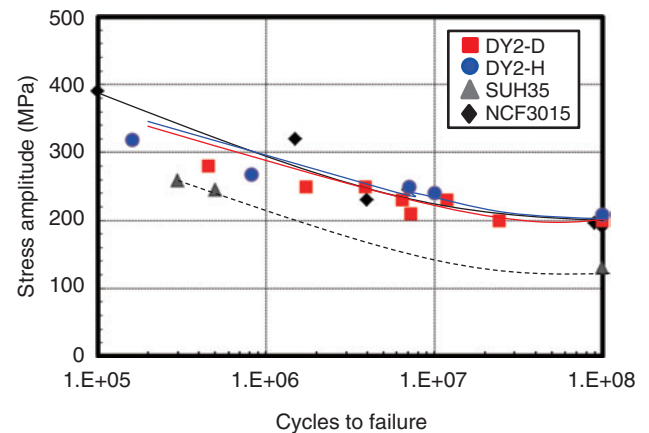


Fig. 7 Fatigue strength at 800°C

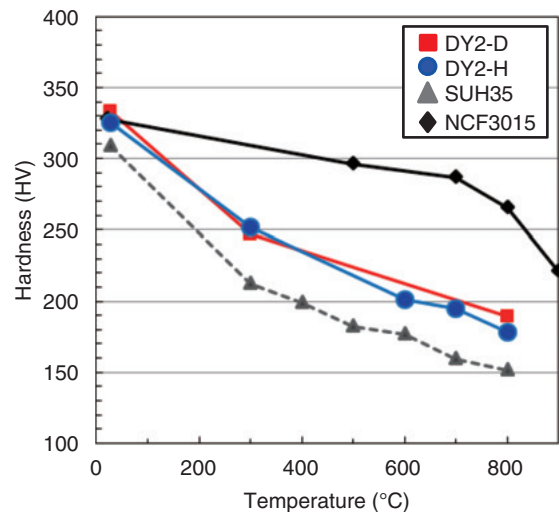


Fig. 8 Hardness at high temperatures

temperature to the high temperature of 800°C, but less than that of NCF3015.

Since these alloys have been given greater wear resistance by increased carbide precipitation, their wear characteristics were verified by endurance testing in an actual engine and by stand-alone slide testing as described below in Chapter 6.

5.3. Oxidation Resistance

Exhaust valves are used under a thermo cycle accompanying a high-temperature oxidation environment and engine acceleration and deceleration load change. Oxidation of the valve generates oxide scale on the valve surface. This increases wear and sometimes causes leakage from the valve face. Figure 9 shows the increase in amount of oxidation after the valve is heated at 850°C for 400 hrs. DY2-H is on a par with NCF3015, and, together with DY2-D, shows greater oxidation resistance than SUH35.

5.4. Corrosion Resistance

Exhaust valves are exposed to sulfides that are a product of combustion. Figure 10 shows the results from sulfur attack testing (heated at 870°C for 80 hours in corrosion

ashes mixed with $\text{CaSO}_4\text{-BaSO}_4\text{-NaSO}_4$). DY2 demonstrates greater corrosion resistance than NCF3015.

5.5. Physical Characteristics

Among the various physical characteristics, the coefficient of thermal expansion indicates the elongation characteristic of the valve material, which affects the tappet clearance. Thermal conductivity affects the heat-reducing ability of the valve. The specific gravity affects the weight of the valve. In general, the coefficient of thermal expansion should be small in order to reduce thermal stress and minimize the effect on the tappet clearance, while thermal conductivity should be large in order to enhance the heat-reducing ability, and the specific gravity should be small from the viewpoint of lightness⁽⁴⁾. Figure 11 shows the coefficient of thermal expansion, Fig. 12 shows thermal conductivity, and Table 2 shows specific gravity. DY2 has a coefficient of thermal expansion and thermal conductivity that are higher than NCF3015. Testing in an actual engine

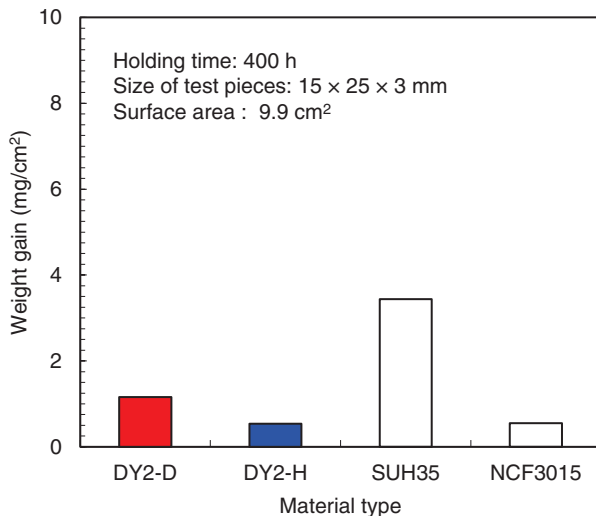


Fig. 9 Oxidation resistance at 850°C

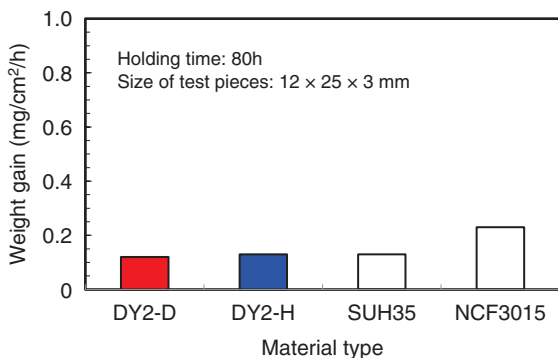


Fig. 10 Sulfur attack resistance at 870°C

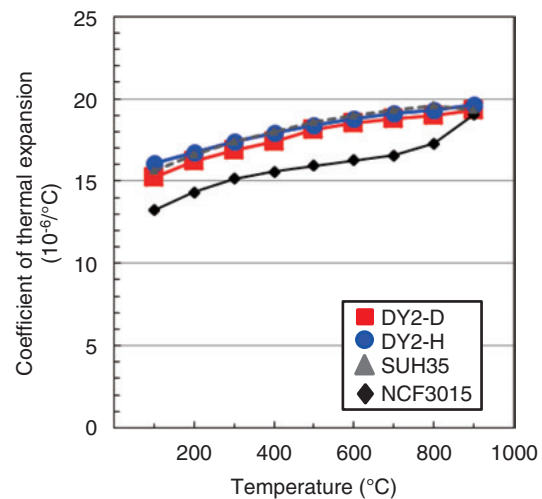


Fig. 11 Coefficient of thermal expansion

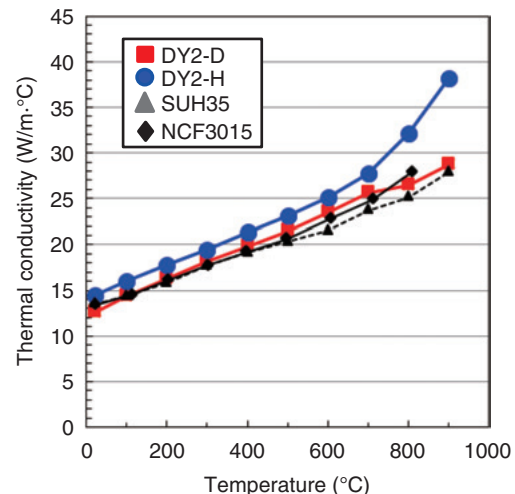


Fig. 12 Thermal conductivity

Table 2 Specific gravity

DY2-D	DY2-H	SUH35	NCF3015
7.83	7.86	7.75	7.89

was performed to verify the characteristic of thermal elongation, and this showed that there were no issues regarding effects on tappet clearance. The specific gravity has been lowered by reducing the nickel content.

5.6. Cost

Reduction of the nickel content has enabled a reduction of approximately 20% in the cost of materials for DY2-D and DY2-H.

6. Characteristics of Parts

6.1. Hot Workability

Hot workability is used to evaluate the formability of the valve. Figure 13 shows the results of performing a hot high-speed tension test using a Greeble testing machine. In general, the temperature region in which a reduction in area of at least 50% can be obtained in the Greeble test is the formable range. The temperature region in which DY2-D and DY2-H show a reduction in area of 50% or more has a range of 250°C or more, and these materials have workability on a par with NCF3015.

6.2. Wear Resistance Characteristic

Figure 14 shows the results from stand-alone sliding tests of valve parts. DY2-D and DY2-H have wear resistance characteristic on a par with NCF3015.

6.3. Durability in an Actual Vehicle

Figure 15 shows the amount of valve seat face wear after 100 hours of operation at 6800 rpm in a 1.5-liter direct-injection engine. DY2-D and DY2-H show the same wear characteristic in an actual engine as NCF3015.

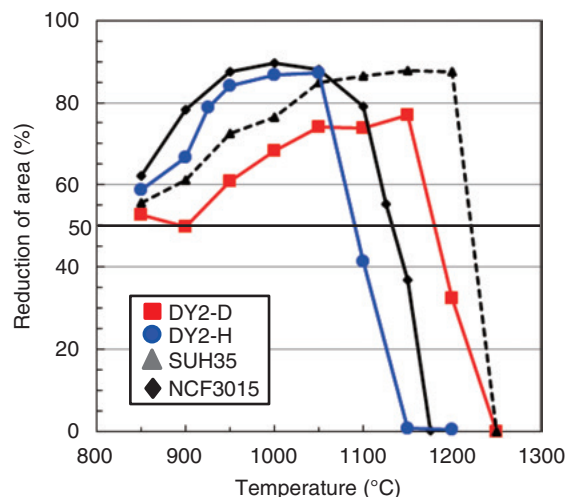


Fig. 13 Hot forgeability

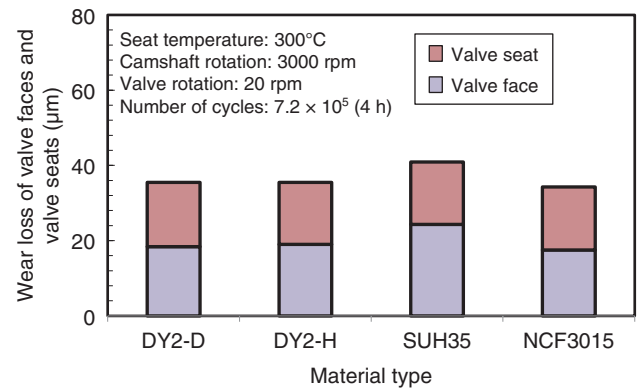


Fig. 14 Valve sliding test

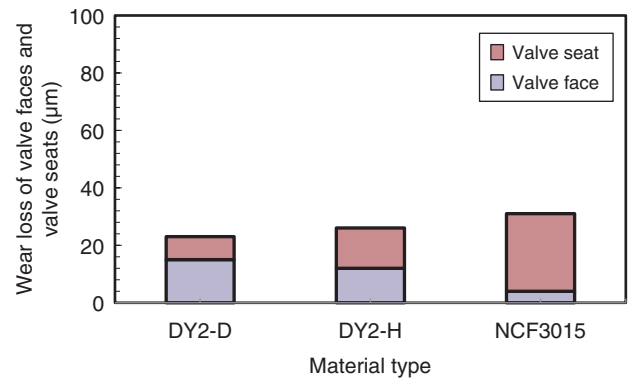


Fig. 15 Wear loss of exhaust valve face and valve seat

7. Conclusion

DY2 materials that save resources and have lower cost were developed in response to the exhaust valve material characteristics that are being called for.

Carbides were increased within the range that would not degrade hot workability while also achieving a reduced nickel content. The addition of optimal amounts of molybdenum, which is an element for strengthening solid solution, and phosphorus, which is a grain boundary strengthening element, is also a feature of these materials, which have achieved high-temperature fatigue strength, wear resistance, oxidation resistance, and corrosion resistance on a par with heat-resistant nickel-based super alloy NCF3015, which is the material previously used. The use of this alloy has enabled a cost reduction of approximately 20% over the conventional heat-resistant nickel-based super alloy.

Use of these developed materials in mass-production vehicles has begun with the 2014 model FIT.

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■ Author ■



Takashi TSUYUMU



Shigeki UETA



Mototsugu OSAKI



Akihiro TOJI



Katsuhiko OHISHI

Aerodynamic Development of 2014 Model Year FIT

Kentaro MACHIDA*
Atsushi MURAKAWA*

Munetsugu KANEKO*
Atsushi OGAWA*

Terumasa KOTADA*

ABSTRACT

Aerodynamic development of the 2014 model year FIT achieved an approximately 7% reduction in aerodynamic drag compared to the 2008 model, giving it among the world's lowest drag coefficients in the hatchback category. Based on wind tunnel testing and simulation results, a vehicle body with high aerodynamic performance was realized. The Honda-created CFD system was used for the exterior design development, helping to optimize aerodynamic shape while supplementing scale-model testing. For the platform development, CFD analysis of the flow field established a direction for enhancing aerodynamic performance and helped optimize the shape in detail. From this process, it was confirmed that to inhibit total pressure loss in the vehicle wake, it is important to ensure uniform flow velocity and pressure distribution in the vehicle front (including the engine room) and rear end.

1. Introduction

Reducing aerodynamic drag of on-road vehicles and thereby improving fuel economy and reducing emissions is important to mitigate the effects that vehicles have on the environment. Therefore, reduction of drag ($CD \cdot A$), that is, the coefficient of drag (CD) multiplied by frontal projected area [$A(m^2)$], is an important objective of aerodynamic development. Development of hatchback vehicles, however, has continued to focus on other marketability (safety, interior comfort, cargo capacity, and so on), which severely limits how far the vehicle can be downsized. At the same time, changes in vehicle shape to reduce drag are also naturally limited because they need to be balanced against styling design concerns. Therefore reducing drag poses the challenge of efficiently extracting features of improved aerodynamic performance and deciding how to balance drag reduction with other functions, such as crash safety, field of vision, and interior comfort^{(1), (2)}. Under these limitations, it is effective to analyze the flow around the vehicle by CFD simulation to establish a drag reduction strategy. Achieving the drag reduction target while quantitatively confirming aerodynamic performance in collaboration with the exterior styling and parts design sections would also appear to be an efficient approach. In addition to conventional wind tunnel testing and full-vehicle CFD, development of the 2014 model year FIT (new FIT) took advantage of Flow Analysis Simulation Tool (FAST)⁽³⁾, the Honda-created CFD system, to strike a balance with styling design and other functions. This report discusses the characteristic flow

field of a hatchback as determined from the aerodynamic development process, and introduces the technique that reduced aerodynamic drag in the new FIT.

2. Goal and Targets of Development

Changes in the drag of B-segment hatchbacks, a category that includes the 2008 model year FIT (previous model), are shown in Fig. 1. The figure shows drag distribution of various models as the ratio relative to the previous model ($CD/CD_{\text{previous model}}$). All plots shown in this figure were the test results obtained in Honda's full-scale wind tunnel. The goal at the start of development of the new FIT was to provide customers with reduced aerodynamic drag for better fuel economy, and aerodynamic performance

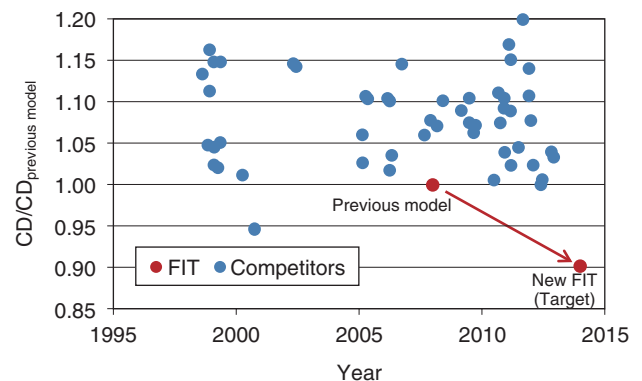


Fig. 1 CD history of B-segment hatchback

* Automobile R&D Center

that is among the best known in the segment. As preconditions, the vehicle had to retain the interior comfort (one of the FIT's strengths), and was therefore required to balance aerodynamic performance with the one-motion form and forward cabin form styling motifs.

Under these conditions, the new FIT took course to reduce drag without lowering A; that is, by decreasing CD alone.

Trade-offs between improvement of fuel economy with reduced drag and the other influencing factors were made, thus narrowing down the ideas to be addressed for improving aerodynamics. Aerodynamics items to be addressed in the new FIT, as listed at the start of development, are shown in Fig. 2. Here, items related to vehicle styling are classified as "Exterior design," and those related to the engine compartment including bumper and front grill, and underfloor are "Platform." The total drag target was set based on all of these aspects, and subsequently the development proceeded by evaluating the achievement of the targets simultaneously.

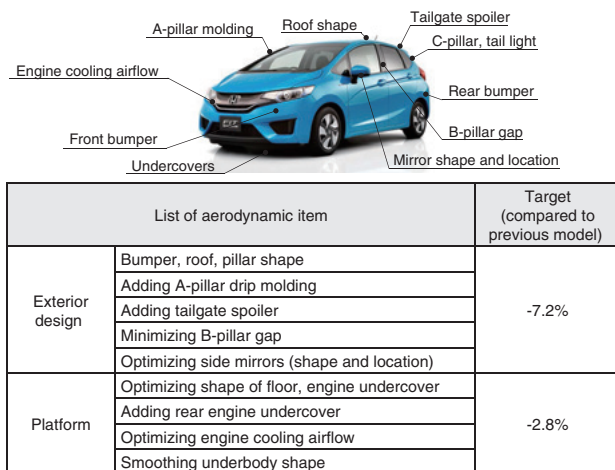


Fig. 2 List of aerodynamic items of new FIT

3. Aerodynamic Analysis of the New FIT

As Fig. 1 shows, the aerodynamic performance of the previous model was already among the best in the B-segment hatchback category. To further reduce drag in the new FIT, CFD analysis of aerodynamic phenomena started immediately after the development of the previous model was completed. Results were shared with styling design and engineering sections and a strategy was established for optimizing aerodynamics.

The following section introduces the aerodynamic characteristics of the new FIT in the styling design and platform development. Note the coordinate system used in this paper, wherein the X axis runs in the longitudinal direction of the vehicle, the Y axis in the transverse, and the Z axis out of the XY plane, as shown in Fig. 3.

3.1. Styling Design Development

The new FIT was the first model that took full advantage of the FAST system in the development. The FAST system performs aerodynamic simulation by automatically linking exterior surface design with predefined platform layout. This allowed engineers to run calculations efficiently, independent of their level of experience in CFD, whether they are aerodynamicists or stylists, and the results can be shared among them.

In the early phase of styling design development, a large number of styling proposals were analyzed and compared to the previous model. Figure 4 is an example of aerodynamics study results for various styling proposals. Here, analysis was done using models provided by the styling design section as well as models applying aerodynamic improvements. In this phase, the objective is not to narrow down the styling proposals but to extract features of aerodynamic improvement.

Among several features that were common in low-drag models as compared to the previous model, characteristics of the body parts that were especially sensitive to drag, specifically the front bumper, roof, and C-pillar side surface, were extracted from the analysis results. Figure 5 compares the surface pressure distribution of the previous model with that of the new FIT incorporating the aerodynamic features of each body part. To help prevent severe changes in pressure at the corner of the front bumper, the side surface length in the flow direction was increased, as was the radius of curvature [Fig. 5(a)]. In side view, the peak of the roof was moved forward as compared to the previous model so that pressure gradually increases toward the rear [Fig. 5(b)]. This coincided with the direction of styling that intended to give the form a sense of lively motion⁽⁴⁾.

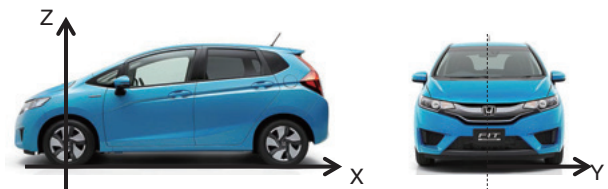


Fig. 3 Definition of coordinates

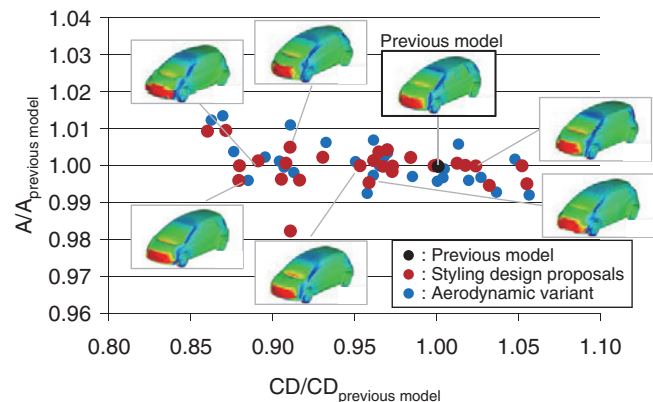


Fig. 4 Simulation results on styling development phase

In the previous model, the flow from the side tended to swirl around the C-pillar, behind the back of the vehicle, because of the shape of the pillar, hence the strong peak of negative pressure [Fig. 5(c)]. The new FIT therefore adapted the shape of the tail end so that it points outward in the down view, which weakened the negative pressure peak. For the front bumper, roof, and C-pillar, the objective was to reduce local pressure gradients, therefore causing the pressure at the tail end on all three surfaces to be equal. Simultaneously, the shapes of these parts were modified to rectify the air-flow magnitude in the X direction. As a result, the pressure on the back surface of the vehicle increased [Fig. 5(d)].

The stylists worked to balance aerodynamic performance and styling design with these features of aerodynamic improvement.

Figure 6 shows the pressure distribution along the X axis of the new FIT and previous model comparing surface pressure distribution quantitatively. The pressure distribution on the roof and rear gate along the section through the center of the vehicle is shown in Fig. 6(a). The new FIT shows less change in pressure distribution in the X direction and higher pressure on the roof. This is because moving the roof peak forward greatly increases the distance over which pressure can increase as the air flows to the rear. This increases pressure on the back of the vehicle as well. The pressure distribution on the section at mid-height of the C-pillar is shown in Fig. 6(b). Pressure on the forward side window is about the same as in the previous model, but at the C-pillar towards the rear, the negative pressure has a lower peak and smaller gradient. The results also show that

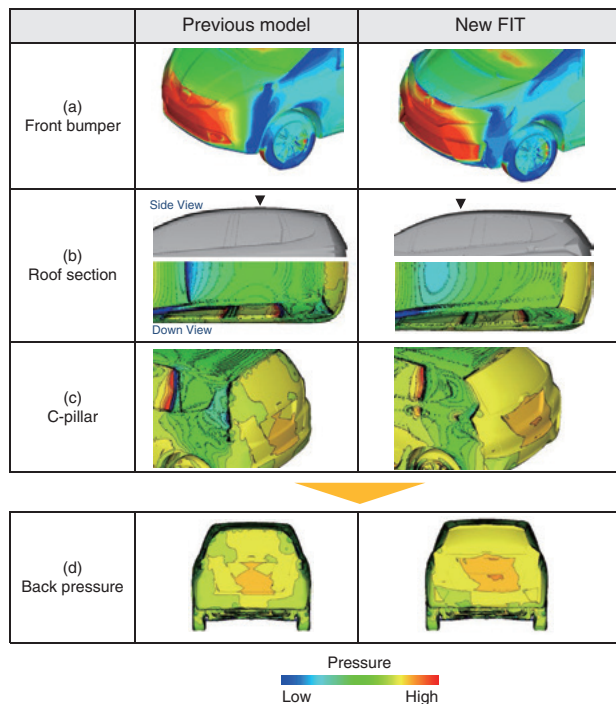


Fig. 5 Surface pressure comparison of previous model and aerodynamically optimized styling

in the new FIT, the pressure on the roof end is close to that on the C-pillar. These indicate that the new FIT achieved its goal of an exterior design shape with uniform pressure distribution on the vehicle surfaces extracted during the advance study.

Figure 7 shows the relationship between drag and total number of measurements taken during the entire period of development in wind tunnel tests using 25% scale models of the new FIT. Each dot in the figure represents one variation on the shape. In total, more than 2000 tests were run during development; however, the target drag level had already been achieved with the optimized aerodynamic shape found in the early phase, where FAST was used intensively. In most of the subsequent tests, the objective was to study the likelihood of achieving aerodynamics together with the desired exterior design, engineering, and production technology. However, the shape that achieved the target level of drag was efficiently proposed by applying the aerodynamic improvements found with FAST.

3.2. Platform Development

For the platform development, Fig. 8 compares the underfloor devices applied to the new FIT with those of the previous model. A cover was added at the rear end of the front sub-frame, which rectified the flow from the engine compartment. In addition, the cover surface area was increased to reduce total pressure loss at the rear of the underfloor, particularly around the rear suspension beam, to

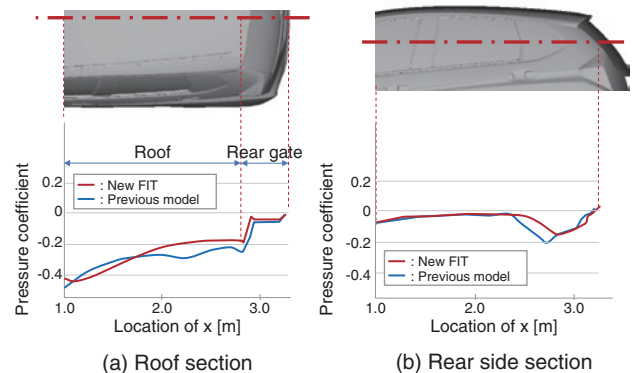


Fig. 6 Pressure coefficient on roof and rear side section

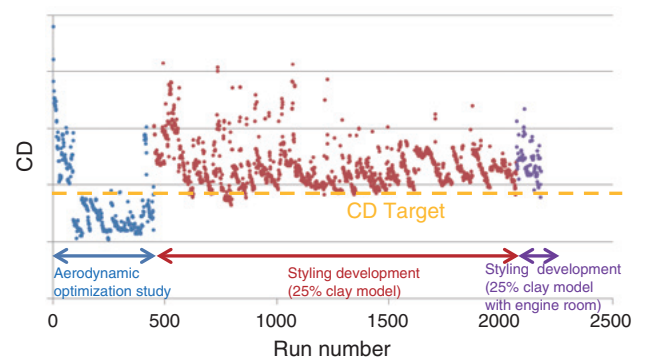


Fig. 7 CD history on wind tunnel tests

further improve aerodynamics.

The necessity of these aerodynamic devices was examined in terms of the target fuel economy and cost for each destination market, and the devices were selected accordingly.

For platform development, locations were extracted where great influences on aerodynamics are expected based on simulation results with the previous model. This section discusses the flow field around the front floor, which has a particularly significant effect, as well as the technique used to improve aerodynamics.

Figure 9 shows the pressure (left) and velocity (right) distribution on the YZ cross-section at the front wheel center position in the previous model. Paying particular attention to the pressure on the body side and underfloor, it shows that the pressure near the wheelhouse [Fig. 9(A)] is lower than that on the underfloor [Fig. 9(B)]. Due to this pressure difference, the airflow with less energy [Fig. 9(C)] is drawn from the underfloor to the front wheelhouse and

out to the body side. This flow results in an increase in total pressure loss at the back of the vehicle, which increases drag. This is the effect caused by the pressure difference between the bumper corner and underfloor. In order to restrain the total pressure loss by the flow from the front wheelhouse, pressure difference in this region needs to be reduced.

The surface pressure distribution of the previous model is shown in Fig. 10. Since its first generation, the FIT has been characterized by a shape with a short front overhang, based on the MM ideology, which creates strong negative pressure close to the front wheel edge [Fig. 10(A)]. This creates a large pressure difference with the underfloor, hence the greater impact of the outflow from the wheelhouse. Effective ways to control this in hatchbacks like the FIT are to reduce the negative pressure of the front bumper by optimizing the shape of the exterior design and to lower the pressure on the front underfloor. Since its first generation, the FIT has had a chin spoiler (air dam) to try to achieve this effect.

Based on these analysis results, using the previous model, a new shape was introduced for the new FIT in an effort to further increase the negative pressure on the front underfloor, which is to a bump at the leading edge of the engine undercover (Fig. 11). The airflow from the lower end of the front bumper accelerates as it goes around the forward edge with its small radius of curvature, producing

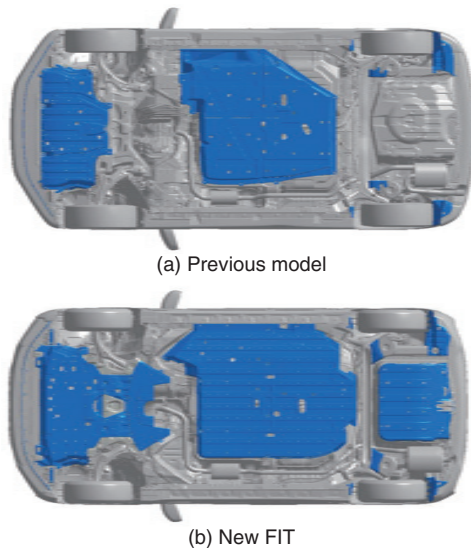


Fig. 8 Underbody covers

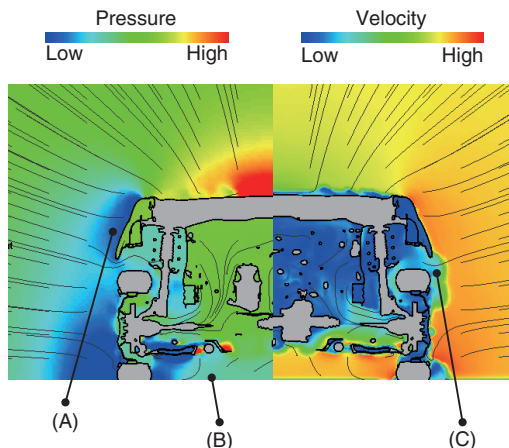


Fig. 9 Pressure (left) and velocity (right) contours at front wheel center section

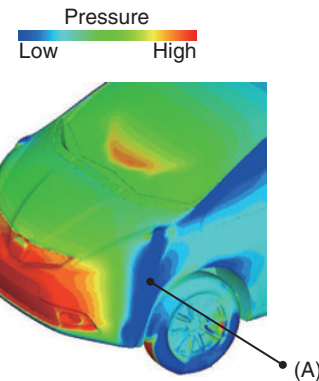


Fig. 10 Low surface pressure region at front bumper corner (previous model)

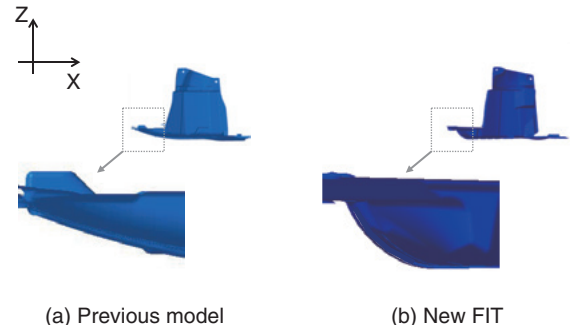


Fig. 11 Shape of engine undercover

a negative pressure peak in this area. Figure 12 compares the underfloor pressure of the new FIT and previous model. It shows that the new FIT has greater negative pressure under its undercover. To compare the pressure difference between the underfloor and front bumper, Fig. 13 shows the pressure distribution along the X axis for both the previous model and the new FIT. In the new FIT, negative pressure is increased in the area with the engine undercover, as indicated in Fig. 12. Optimization of the exterior design gives the new FIT a moderate pressure peak at the front bumper corner as compared to the previous model, resulting in a smaller pressure difference between the side and underfloor in the front section.

The flow velocity distribution at the underfloor along the section through the center of the vehicle is shown in Fig. 14 for both the new FIT and the previous model. The effect of reducing the pressure difference at the front decreased the total pressure loss of the underfloor. The new FIT shows higher flow velocity in most areas.

A cross-section of total pressure loss on the body side is shown in Fig. 15. Areas where the flow is losing energy are shown in red in this figure. Compared to the previous

model, outflow from the front wheelhouse in particular is being restrained on the new FIT [Fig. 15(A)] and has a smaller area of pressure loss towards the back of the vehicle. This is the result of creating uniform pressure at the front section, as shown in Fig. 13.

To improve the aerodynamics of the front, we not only investigated these factors but also worked to reduce engine compartment airflow resistance (cooling drag). To reduce cooling drag, it is important to efficiently draw incoming airflow from the front grill to the radiator and other cooling devices and reduce the flow volume to the minimum necessary. To do this, air guides were installed to restrict excess air from flowing out from the engine compartment to the underfloor and body side and optimized the cooling efficiency (Fig. 16). As a result, cooling drag was reduced 19% compared to the previous model when the grill opening was optimized.

Using these study results, a new platform was developed to reduce drag. The results were also used for other Honda models that share the common platform.

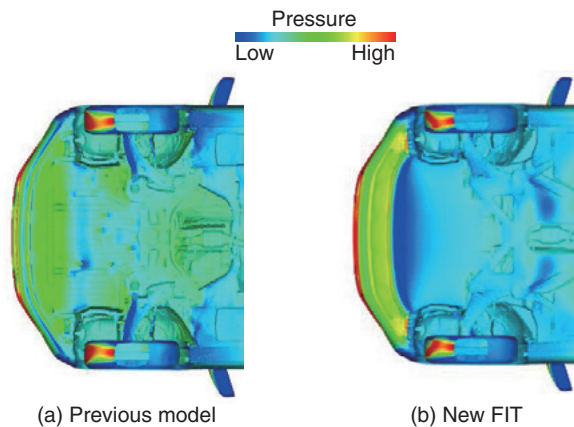


Fig. 12 Surface pressure distribution on engine undercover

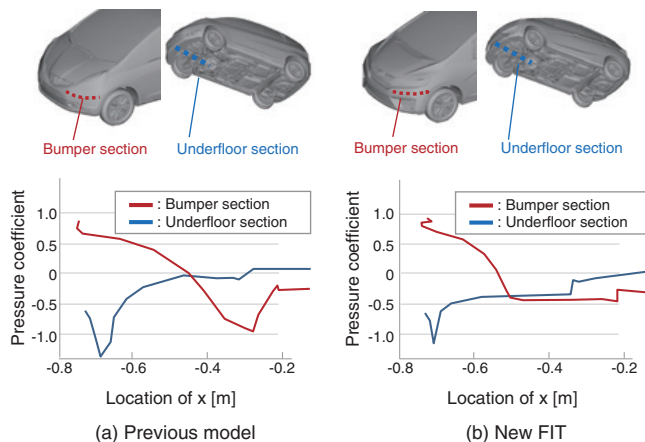


Fig. 13 Pressure coefficient on front bumper and underfloor

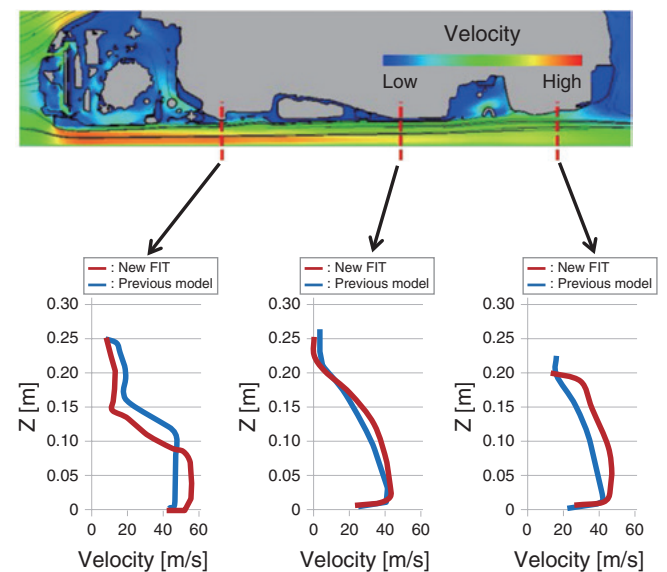


Fig. 14 Flow velocity on underfloor section

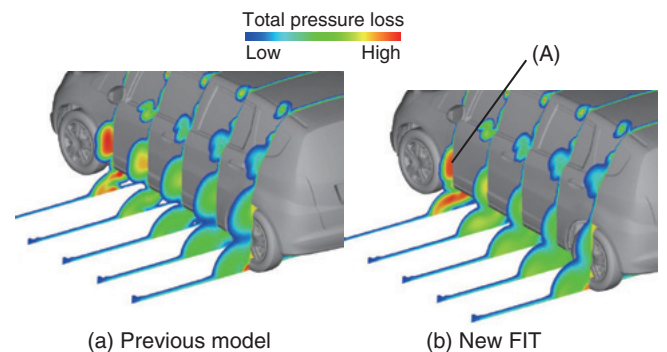


Fig. 15 Total pressure loss comparison of new FIT and previous model

3.3. Comparison of Wake

Drag factors acting on vehicle bodies are broadly classified into those that increase total pressure loss in the vehicle wake due to flow separation, and those that induce velocity components in the Y and Z directions due to pressure difference on the vehicle surface, so that the strength of the longitudinal vortex (a vortex with its axis in the X direction) increases. The following section compared the new FIT's vehicle wake with that of the previous model, with a particular focus on these factors.

3.3.1. Distribution of total pressure loss

The distribution of total pressure loss on the YZ section of the wake is shown in Fig. 17. In this figure, there is less total pressure loss where the red area behind the vehicle is smaller and is at a lower level.

In the wake of the new FIT, total pressure loss resulting from the A-pillar [Fig. 17(A), (A')] has been reduced. This is due to the pillar garnish on the new FIT. Furthermore, the area of the total pressure loss is smaller on the body side due to the newly designed platform, which reduces outflow from the front wheelhouse [Fig. 17(B), (B')]. On the other hand, the new FIT has greater loss near the center of the vehicle [Fig. 17(C), (C')]. This is because the downward flow (downwash) from the back of the roof is controlled at the vehicle center on the new FIT.

To make the difference between the new FIT and previous model clearer, Fig. 18 shows the difference in total pressure loss in the YZ section five meters behind the vehicle. Here, areas where the new FIT has smaller

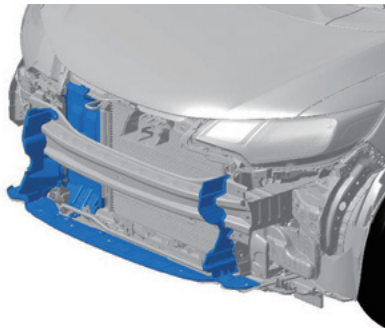


Fig. 16 Air guide devices in engine room

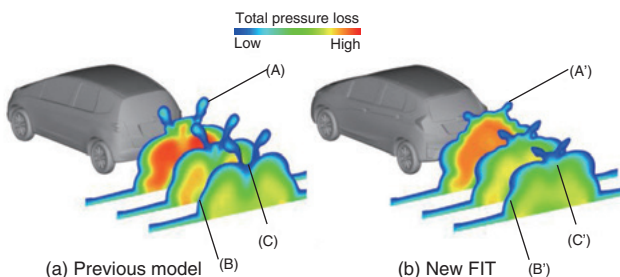


Fig. 17 Total pressure loss behind vehicle

total pressure loss, hence contributing to less drag than the previous model, are indicated in blue. The figure shows that the new FIT has less total pressure loss at the A-pillar [Fig. 18(A)] and side [Fig. 18(B)], and reduced loss in the entire cross-section. This confirms that it is mainly the A-pillar and body side that contribute to aerodynamic improvement by reducing total pressure loss.

3.3.2. Distribution of longitudinal vortex

Figure 19 shows the distribution of X-axis vorticity in the YZ section at the back of the vehicle for the new FIT and the previous model. Red indicates clockwise and blue counterclockwise strong vorticity. Vortices from the A-pillar [Fig. 19(A)] and a pair of longitudinal vortices behind the vehicle [Fig. 19(B)] can be observed, however these are smaller with the new FIT. This is because, in the exterior design, the roof peak was moved forward, and the C-pillar position and the shape of the lid light were optimized. That created a uniform pressure distribution on the vehicle surface and prevented the airflow from swirling around towards the back of the vehicle.

Longitudinal vortex drag in the YZ section behind

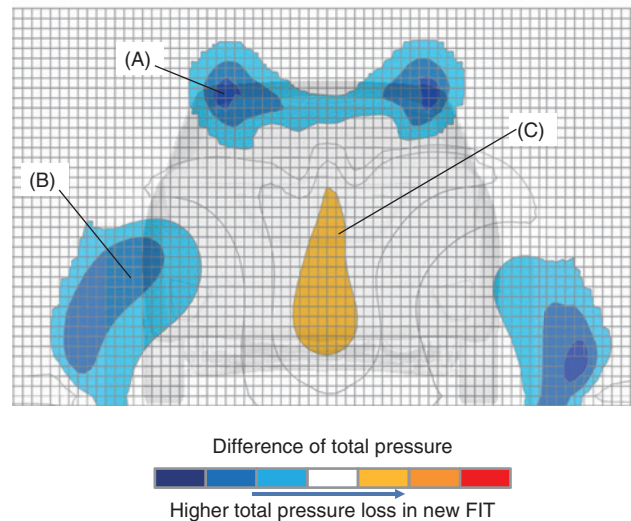


Fig. 18 Total pressure difference of New FIT and previous model

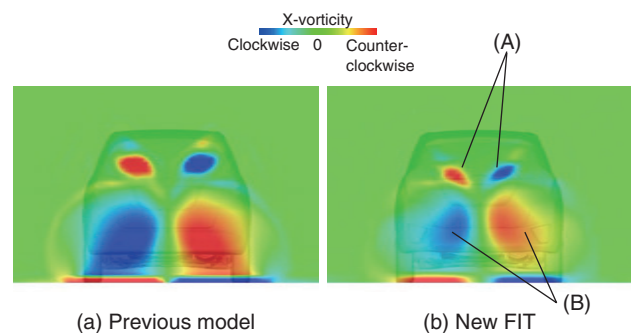


Fig. 19 Vorticity comparison of New FIT and previous model

the vehicle is shown in Fig. 20. This has a dimension of momentum per unit surface area, found by multiplying the density, ρ by the sum of squares of velocity in the YZ direction: $\rho(v_y^2 + v_z^2)/2$. It represents magnitude of air resistance resulting from airflow swirling behind the vehicle⁽⁵⁾. In the new FIT, longitudinal vortex drag is smaller, as shown in Fig. 20(A) and (A'). The difference is shown in Fig. 21. Here, blue indicates those areas where the new FIT has a smaller longitudinal vortex drag than the previous model. These are the areas that contribute to drag reduction. Longitudinal vortex drag is reduced in all areas with the new FIT, and there is a particularly great difference at the vehicle center. This is due to pressure on the vehicle surface being uniform with respect to the previous model, resulting in reduction of downwash from the roof and longitudinal vortex caused by the downwash [Fig. 21(A)]. In this area, there is a trade-off between increasing total pressure loss and controlling longitudinal vortex; however, Figs. 18 and 21 show that, compared to the previous model, the new FIT has lower total pressure loss, better control of longitudinal vortices, and less drag in total.

3.4. Aerodynamic Achievements of the New FIT

The efforts to improve the aerodynamic performance of the new FIT as described above were examined for commercial viability in close collaboration with the

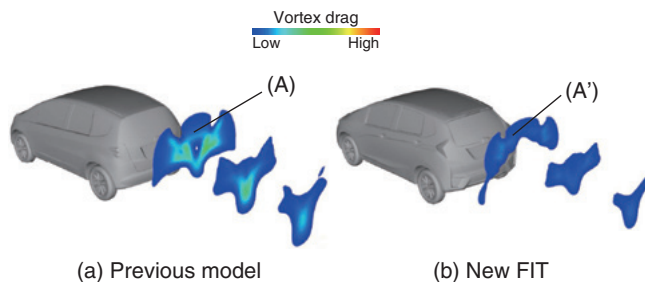


Fig. 20 Longitudinal vortex drag behind vehicle

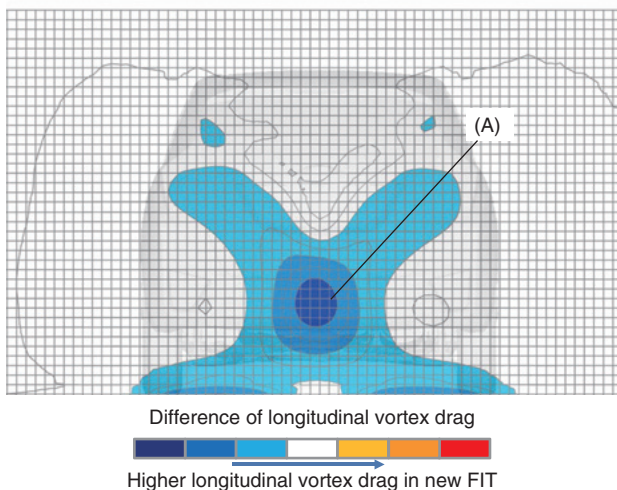


Fig. 21 Longitudinal vortex drag difference of new FIT and previous model

engineering section. Figure 22 shows the position of the new FIT's drag relative to other B-segment hatchbacks. The new FIT achieved one of the lowest drags known, as intended. This contributed to its excellent fuel economy of 36.4 km/l in JC08 mode.

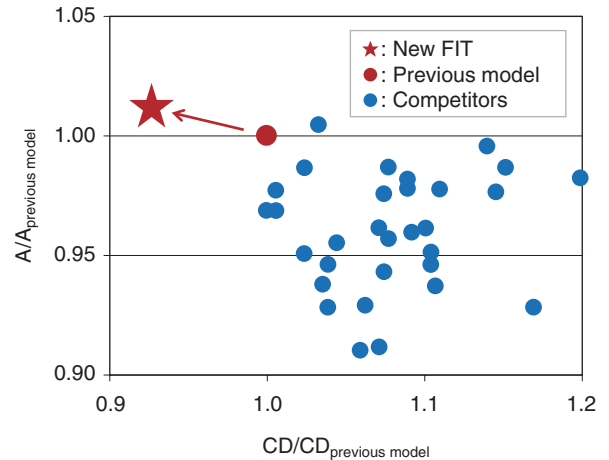


Fig. 22 Schematic of CDA

4. Conclusion

Based on wind tunnel testing and simulation results from the advance research in both the styling design and platform, a vehicle body with high aerodynamic performance was realized in the aerodynamic development of the new FIT. The results are summarized below.

- (1) In the development of the styling design, effective use of the CFD system and wind tunnel testing enabled both uniformity of flow velocity and pressure distribution around the vehicle body and a shape with the desired design characteristics.
- (2) In the development of the platform, the use of ducting in the engine compartment and optimization of the undercover shape optimized pressure around the front bumper. Thus, a vehicle body that minimized interference with the airflow of the exterior design was developed.
- (3) The aerodynamic vehicle body development technology described above reduced drag by 7% over the previous model and contributed to the excellent fuel economy of 36.4 km/l in JC08 mode.

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■ Author ■



Kentaro MACHIDA



Munetsugu KANEKO



Terumasa KOTADA



Atsushi MURAKAWA



Atsushi OGAWA

Aerodynamic Development of 2013 Model Year Accord and Accord Hybrid

Daisuke NAKAMURA*

Yasuyuki ONISHI**

Yoshiyasu TAKEHARA*

ABSTRACT

Improvement of the $C_D A$, which is an index of aerodynamic performance, is one factor in improving fuel economy. Honda has achieved a 1.3% enhancement in the $C_D A$ of the LX grade of the 2013 model Accord relative to the previous model, and an 8.3% enhancement in the $C_D A$ of the Accord Hybrid relative to the EX grade of the previous model Accord.

Based on an analysis of the phenomenon of the flow field around the car body, parameter studies were conducted using the previous model Accord before beginning aerodynamic development. Aerodynamic features contributing to the reduction of air resistance were applied to the package design before development was begun.

During aerodynamic development, aerodynamic performance of the exterior styling was enhanced through wind tunnel testing using 25%-scale detailed models, conducted jointly with the Styling Design Office.

The aerodynamic performance of the under-floor aerodynamic devices was optimized using CFD, a wind tunnel with rolling road, and other such tools in addition to the detailed model.

1. Introduction

Reducing a vehicle's running resistance can contribute to improvement of fuel economy and reduction of gas emissions. This is important in terms of having less negative impact on the global environment.

In the Accord, even though the reduction in running resistance in a single vehicle may be extremely small, with approximately two million vehicles sold in one model cycle, it plays a much greater role in reducing the impact on the global environment.

Running resistance includes air resistance, rolling resistance, grade resistance, and acceleration resistance. Since air resistance increases in proportion to the square of the speed, it accounts for approximately 75 to 80% of the running resistance of a 2013 model year Accord (the new Accord) at 100 km/h. There is, therefore, a need to reduce running resistance through aerodynamic performance, by paying attention to the details of geometry and reducing the $C_D A$, which is the product of the vehicle body's coefficient of drag (CD value) and the frontal projected area A (in m^2).

On the other hand, there are the areas and the volume taken up by passenger space, cargo space, collision safety performance, and other such factors of necessity in terms of the vehicle's product appeal. There is therefore a limit to how much the area A can be reduced in connection with making the vehicle more compact, since the styling of the exterior is

given priority in terms of product appeal^{(1),(2)}. There are also limited areas where aerodynamic features for lower C_D can be applied to the vehicle exterior that encloses those spaces.

It can be effective, therefore, to use Computational Fluid Dynamics (CFD) to analyze the flow field around the vehicle during full model changes and isolate the features of improved aerodynamic performance. Based on the results from this approach, Honda is carrying out repeated wind tunnel tests jointly with exterior styling sections and applying aerodynamic features in styling design.

This paper will introduce elements that have significant impact on aerodynamic performance according to the results from analysis of the flow field phenomenon in prior studies. The paper will also describe methods used to improve the aerodynamic performance of the new Accord.

2. Aerodynamic Development Goals

Figure 1 shows a $C_D A$ distribution graph that expresses aerodynamic performance in the three-box sedan category. These distribution values were all obtained from CD values measured in Honda's full-scale wind tunnel.

Prior studies were conducted to find a method to achieve improvement of the $C_D A$ by 7% over the previous model Accord, which is already at the top level of aerodynamic performance.

At the development stage, the styling concept of

* Automobile R&D Center

** HONDA R&D AMERICAS, INC.

“exhilarating, advanced, and high quality” provided a basis for pursuing a balance between the horizontally oriented three-box proportions and aerodynamic performance. On that basis, achieving fuel consumption performance was set as the objective through developing a highly efficient aerodynamic package in terms of cost, weight, and various other such factors.

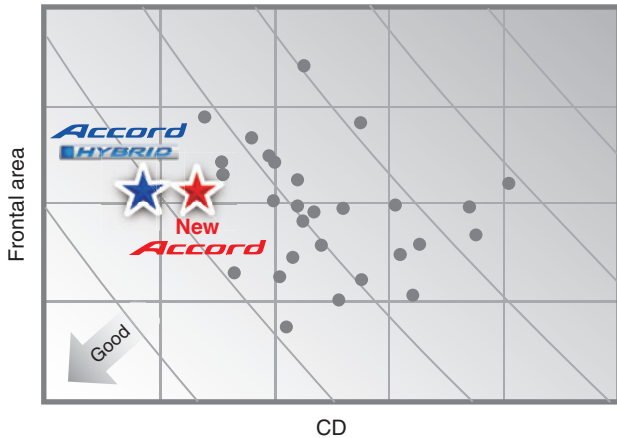


Fig. 1 CD-A distribution of 3box sedan category

3. Exterior Aerodynamic Analysis

Phenomenon analyses and parameter studies were conducted to explore directions for geometry that would achieve the aerodynamic objectives of the new Accord by using a 25% scale clay model of the previous model. Vehicle development proceeded on the basis of a package design together with the vehicle concept that determines the fundamental frame, including the total length, total width, and total height, as well as the placement of passengers, the volume of the cargo compartment, and other such factors. We therefore began with and analyzed those areas that have a significant impact on the aerodynamics, that cannot be readily changed after the package design has been decided, and that affect the overall form of the vehicle. The parameter study of 16 items included the area around the A pillar, narrowing of the sides (tapering in toward the rear), the trunk height, and mirror placement, for which a total of 500 or more variation measurements were carried out.

As a result, a proposal was made to the vehicle package designers before exterior styling development began regarding quantitative changes with regard to the area around the A pillar, the narrowing of the sides (tapering in from the cabin toward the rear), and the relationship between the roof and the trunk end that could achieve the aerodynamic objectives while balancing each of them with other functions.

3.1. Area Around A Pillar

At the A pillar, a vortex with its axis in the direction of flow (longitudinal vortex) ordinarily generates resistance when the flow wraps around from the front windshield to

the cabin side surfaces. That longitudinal vortex, like the vortex A shown in Fig. 2, continues as far as the wake of the vehicle, and it promotes the decrease in pressure at the trunk end. This causes an increase in the wrap-around flow from the side surfaces of the vehicle and an increase in longitudinal vortex drag caused by Vortex B. In order to improve aerodynamic performance, therefore, it is important to weaken the longitudinal vortex as much as possible that is generated at the A pillar.

Basically, the longitudinal vortex generated at the A pillar can be weakened by smoothing the flow from the front windshield to the side glass (Fig. 3). Methods that are effective for this purpose include increasing the angle between the front windshield and the side glass at the A pillar in order to reduce the amount of deflection from wrap-around flow, increasing the pillar width in order to reduce the curvature change of the wrap-around flow, and reducing the height of the drip molding in order to lessen the difference in level in the path of the flow.

However, these elements will affect various functionalities, such as visibility, drip treatment, wind noise,

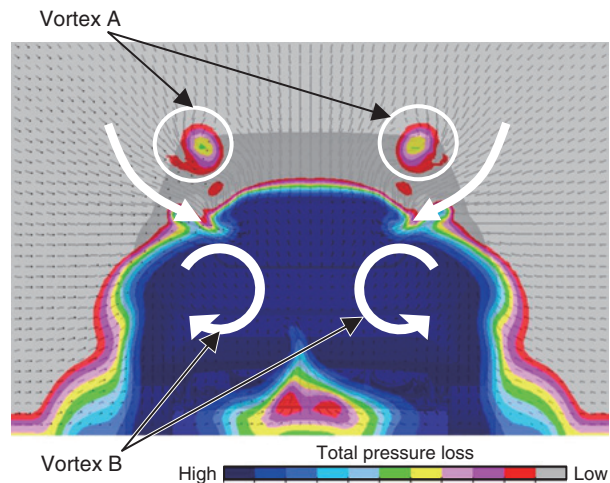


Fig. 2 A-pillar vortex on total pressure loss distribution

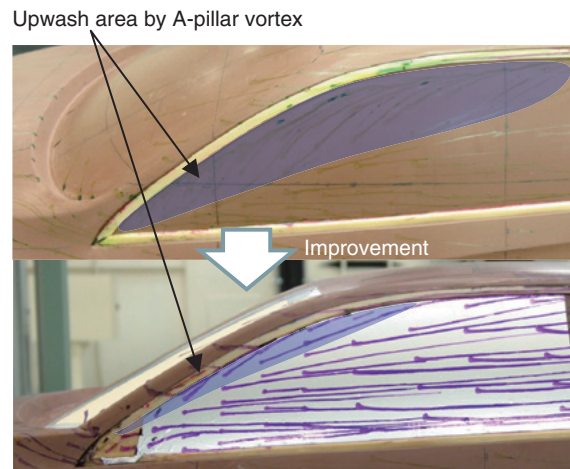


Fig. 3 Vortex improvement around A-pillar

and collision safety. At the stage of the prior studies, a 1.6% increase in aerodynamic performance was proposed by changes in geometry within a range that would effectuate those functionalities. These changes included increasing the angle between the front windshield and the side glass by moving the A pillar inward, and also virtually reducing the height of the drip molding by increasing the curvature of the leading edge of the A pillar molding.

3.2. Narrowing of the Side Panels (Tapering in from the Cabin Toward the Rear)

In order to keep the flow from separating from the vehicle until it reaches the tail of the vehicle, the rear side surfaces of the vehicle can be drawn inward and the total pressure loss that occurs behind the vehicle can be reduced. This can improve aerodynamic performance. This narrowing can also reduce the separation vortex that occurs at the C pillar as well as the longitudinal vortex drag that is generated when the flow from the upper surface of the trunk joins that from the side surfaces of the vehicle.

In the case of the side panels, however, statutory requirements limit exposure of the tires to 30 degrees on the front sides and 50 degrees on the rear sides, both measured from the tire center⁽³⁾, as shown in Fig. 4. Consequently, the wheel arches are the outermost side of the vehicle in almost all vehicle types, and narrowing toward the rear portion of the side panels is limited to the relatively short distance rearward of the wheel arches.

At the stage of the prior studies, the outermost sides were located in the vicinity of the front door, near the middle of the vehicle, and the side panels were narrowed smoothly toward the rear. A geometry with improved aerodynamic performance was also proposed drawing the rear 50-degree portion of the wheel arch inward by making the rear tread narrower.

With regard to the cabin, the B pillar was to be kept in the same position for interior space while the curvature of the side glass would be increased to make it continuous with the A pillar, which was to be moved inward, as described above. The C pillar was also to be moved inward to form a barrel shape, which is advantageous for aerodynamic performance. With these proposals, the geometry would yield a 1.9% increase in aerodynamic performance.

3.3. Relationship Between the Roof and the Trunk End

When the pressure drops caused by downwash to the rearward of the trunk that is deflected downward from the

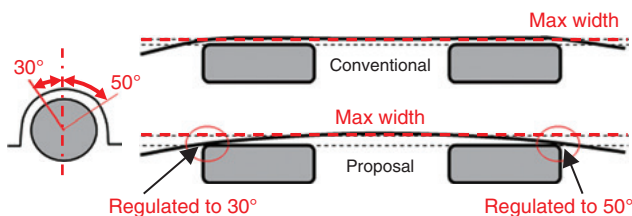


Fig. 4 Proposal of barrel shape to hide wheels

rear edge of the roof, the drag is increased by strengthened longitudinal vortex in the wake of the vehicle (Fig. 5). Effective methods for pressure recovery and suppressing the longitudinal vortex include extending the upper trunk surface toward the rear and raising it higher (Fig. 6). However, when the pressure on the upper surface of the trunk is extremely high relative to the side surfaces of the vehicle, reverse-phase vortices are generated at the trailing edges of the trunk sides. Since this can contribute to increased longitudinal vortex drag, there are threshold values for the magnitude of the geometry changes.

Meanwhile, the height of the trunk affects rear visibility, and extending the trunk to the rear affects the rear bumper stroke in the event of a collision. There are limits, therefore, to the amount of change. Since the height of the rear edge of the roof affects the angle of the flow toward the trunk, the height and location of the trailing edge of the trunk were optimized for reduction of drag by the height and shape of the rear edge of the roof.

With the trunk at a height that does not impact rear visibility, aerodynamic performance would be increased by extending the trailing edge of the trunk to the limit for the rear bumper stroke. This can be combined with optimization of the lowered height of the rear edge of the roof. With these proposals, the geometry would yield a 2.4% increase in aerodynamic performance.

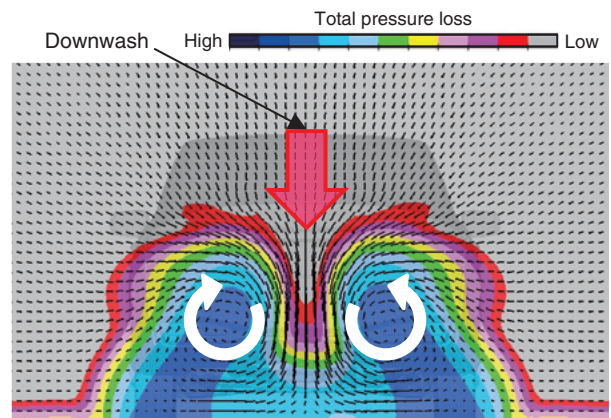


Fig. 5 Example of vortex caused by strong downwash at trunk end

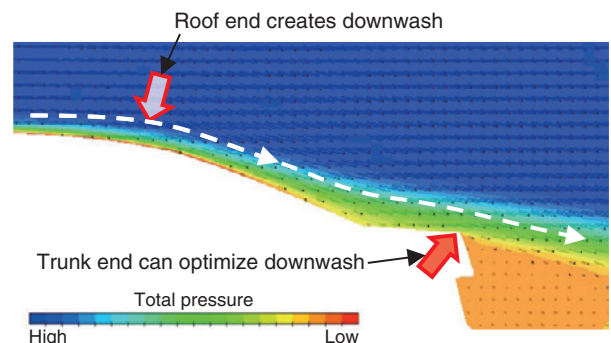


Fig. 6 Relationship between roof end and trunk end

4. Exterior Aerodynamic Design

During development that began with the fundamental frame proposed in prior studies, wind tunnel testing using 25% scale clay models was conducted jointly with the Styling Design Office to achieve a balance between styling and aerodynamic performance.

In Wind tunnel testing, a clay model applying the new Accord styling was used. A thorough, detailed review was conducted again on the basis of results obtained in prior studies, and the styling was enhanced while also increasing aerodynamic performance (Fig. 7).

Where the A pillar had been moved inward, that change was maintained while the pillar was moved forward to enhance the styling. The resulting slight reduction in visibility was recovered by making the A pillar thinner. Although making the A thinner pillar reduces aerodynamic performance slightly, and a thoroughgoing review of section curvature profile and a reduction in the drip molding level difference weakened the A pillar vortex (Fig. 8).

With regard to the cabin sides, the B pillar was moved outward to enhance passenger space in the cabin. When this was done, the curvature of the glass was also increased,

achieving a more barrel-shaped body without moving the A pillar outward.

With regard to narrowing toward the rear of the side panels, the area around the rear wheel arch was expanded outward without making the rear tread narrower to achieve a balance between styling and handling stability performance. However, in order to maintain the narrowing from the middle of the vehicle, the area around the rear edge of the front door was made the outermost point on the sides. A continuous narrowing from the cabin side surfaces was also extended to the trunk sides, making the vehicle more barrel-shaped than the previous model (Fig. 9).

The rear edge of the roof was configured to be even lower than on the previous model, with the trailing edge of the trunk at the same height as the previous model. Considering that the wheelbase has become 25 mm shorter, this has yielded the effect of extending the trailing edge position of the trunk rearward relative to the rear edge of the roof. Since the height of the trailing edge of the trunk is close to the limit in terms of visibility, the top surface of the trunk was given a ducktail shape to adjust the departure angle to optimize pressure recovery. The trailing edge of the trunk was also configured higher for the entire width, and this has been useful not only in reducing drag but also in enhancing rear lift (Fig. 10).

The above refinements yielded better flow from the cabin to the trunk and along the side panels. Figure 11 shows the results from visualization of these effects.

After the overall geometry and dimensions were decided, the wind tunnel model was shifted from clay models to a detailed model (Fig. 12) with an engine compartment. Further optimization was then explored.

The detailed model has an open front grille for cooling, and the exhaust from the engine compartment is simulated. This, when combined with the effect of the under-floor

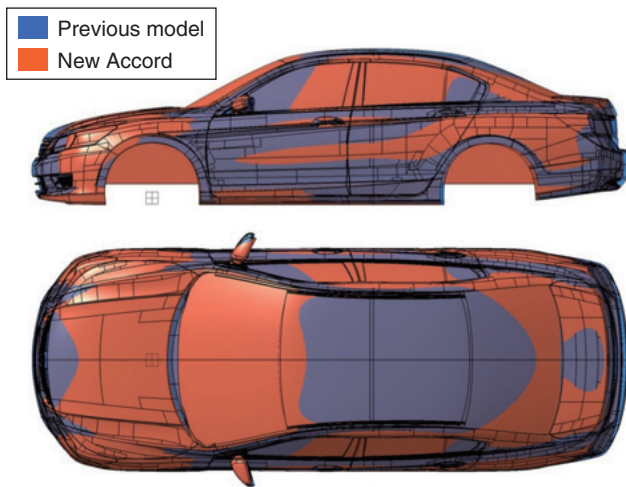


Fig. 7 Side and top view comparison between new Accord and previous model

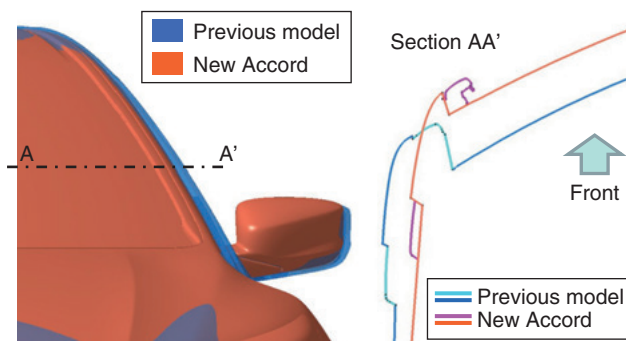


Fig. 8 A-pillar moved inboard

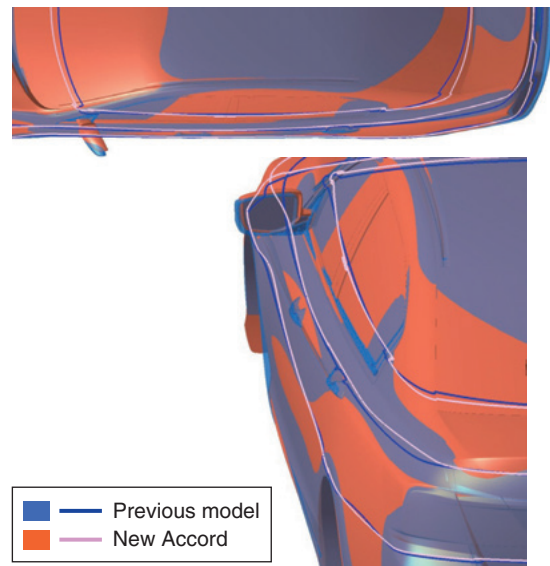


Fig. 9 Barrel-shaped cabin and body side panel

devices, enables aerodynamic optimization of the exterior. It was particularly useful that the shape of the lower edge of the front bumper, which has an important relationship with the chin spoiler, the trunk height, which is affected by the flow from under the floor, and other such areas could be optimized under circumstances closer to an actual vehicle.

With regard to the exterior, as well, the front bumper corner geometry was optimized so that flow would wrap around to the side panels without separation. With the

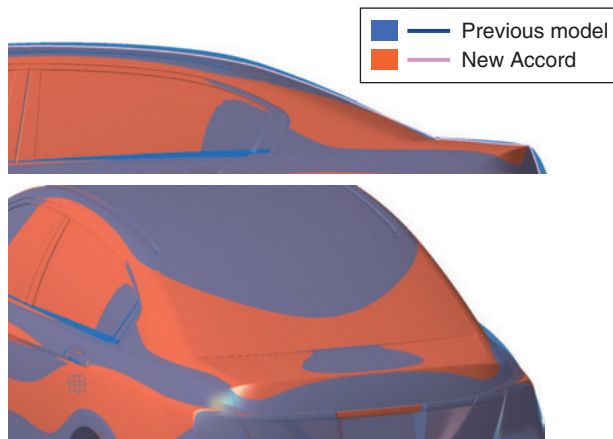


Fig. 10 Roof end and trunk height comparison

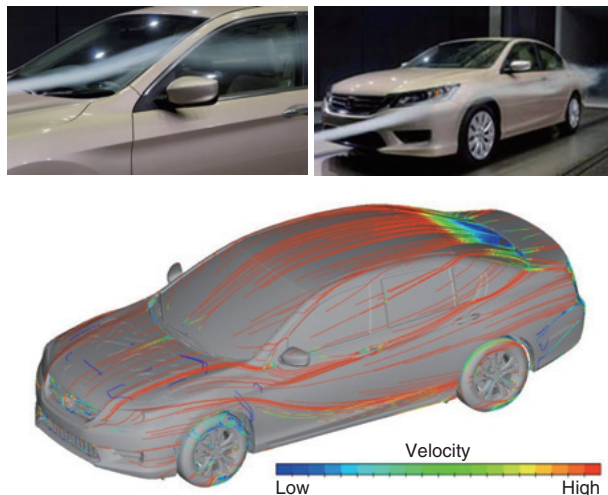


Fig. 11 Flow visualization of barrel-shaped body

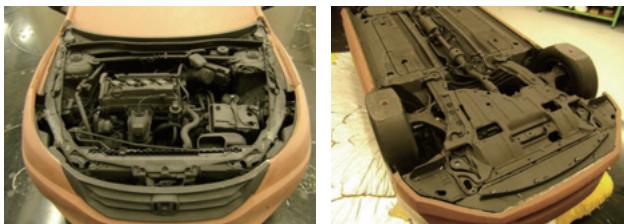


Fig. 12 25% scale model with engine room and under covers for wind tunnel test

vehicle rear made narrower as an extension of the barrel shape, the rear bumper corners were also optimized by such means as adding an edge shape that stabilizes the point of separation so that the flow would not be turned back onto the rear surface of the vehicle.

Optimization of the exterior aerodynamic design extended to fine details and involved measurements in wind tunnel made a total of more than 1,000 times.

5. Analysis of Under-Floor Devices

The reduction in drag by means of under-floor devices naturally involves smoothing the under-floor flow. It is also achieved, however, by reducing longitudinal vortex drag.

The flow that has passed through the under-floor could change the structure of longitudinal vortex by reducing downwash behind the vehicle.

In the prior studies, the task of confirming the CD level that could be achieved by current design methods began with installation of maximum size under-covers together with floor devices using CFD to examine the result.

The flow field could then be analyzed on the basis of those results. Attention was focused on areas that could contribute to enhancement of the under-floor flow, and the range and shape of an effective under cover were considered.

Given the results from analysis of the effect of each device, it was found that reducing the pressure in the area between the rear tires could reduce the total pressure loss region and reduce the longitudinal vortex drag behind the vehicle.

Lowering the pressure in the area between the rear tires improves the flow field inside the rear tires by means of the influence from inward deflection. As a result, total pressure loss area behind the rear tires is reduced (Fig. 13).

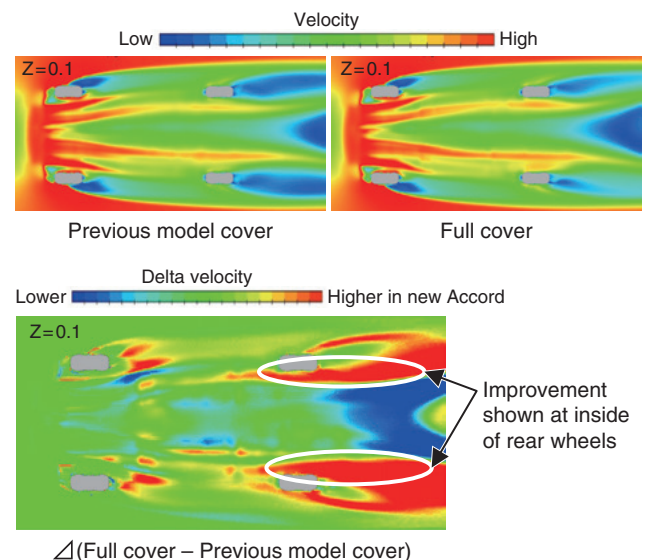


Fig. 13 Velocity distributions and delta velocity between full cover and previous model cover

There is a large vortex (Vortex A in Fig. 14) behind the vehicle caused by downwash at the middle of the vehicle. Reducing the total pressure loss region to the rear of the rear tires strengthens Vortex B, which is in the same phase with Vortex A and located behind the rear tires. This Vortex B weakens Vortex A in the area behind the rear bumper corners, and so causes the rear surface pressure on the vehicle to be recovered. Weakening the downwash from Vortex A also increases the angle of the upwash from the under-floor at the middle behind the vehicle. This reduces the longitudinal vortex behind the vehicle.

For the design policy of the under-floor devices, it was decided to achieve that flow field in the new Accord to lower the pressure in the area in front of the rear tires.

Increasing the velocity in the area between the rear tires was considered an effective means of lowering the pressure in that area. The tank located in the middle and to the front of the rear tires has basically a flat bottom surface. However, the tank cannot be located so that its bottom surface is at the minimum height above ground because of the difference in the above-ground height configuration of the bottom surface from covers made of resin.

A bridge is therefore added between the tank bands. This increased the clip bases so the under-cover could be extended. This measure produced a flat surface at the minimum height above ground between the rear tires. This can increase the velocity under the flat surface to achieve lower pressure (Fig. 15).

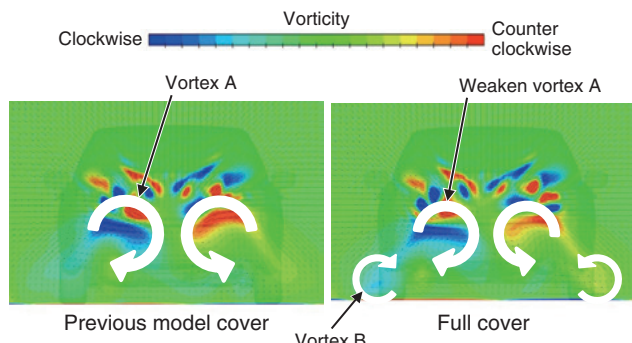


Fig. 14 Vorticity distributions between full cover and previous model cover

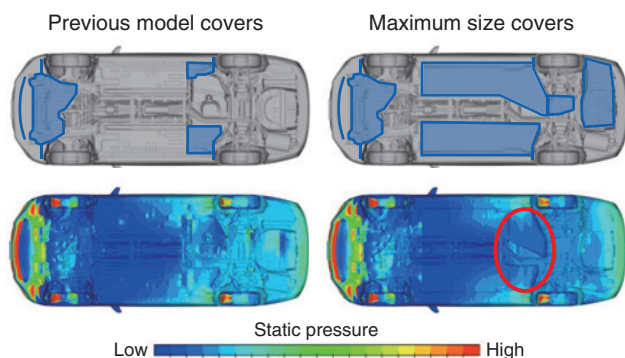


Fig. 15 Static pressure distributions

6. Design of Aerodynamic Devices

6.1. Under-Floor Devices

The geometry of the under-floor devices was optimized using the detailed model, with the prior studies conducted with CFD as a basis.

Upholding the design policy of reducing the pressure in the area in front of the rear tires, and taking a device studied using the largest area under current design methods as the basis, a geometry that takes attachment points and strength into account was applied. Making the cover size more efficient, optimizing the shape of the leading edges of the floor under-cover and rear under-cover, and other such tasks were carried out under conditions more closely resembling an actual vehicle by using a detailed model that simulates the engine compartment exhaust. The geometry based on those results was applied to the prototype vehicle, and the prototype vehicle was used in turn to make detailed improvement with regard to the enlargement of area and the synergistic effects from the cover.

In development of the new Accord, there were improvements in fuel economy performance due to enhancements of engine performance compared to the initial plan at the start of development. The judgment on whether or not to adopt under-floor devices was made taking cost and weight into consideration along with the fuel economy objective.

In the LX grade vehicle, the floor under-cover incorporating design policy was enlarged and it very efficiently improved the aerodynamic performance in comparison with the previous model. The improvement achieved in this way allowed the engine under-cover, which needed to be aluminum for its heat resistance, to be made smaller than in the previous model. This also allowed engine compartment cooling and heat resistance performance to be enhanced while avoiding the adoption of aluminum.

In order to achieve still greater fuel economy performance in the Accord Hybrid, the devices from prior studies were largely adopted. Aerodynamic performance was increased 2.5% (Fig. 16).

6.2. Ducting in Cooling Opening

Air needs to be drawn in through the front grille in order to cool the engine compartment. However, engine compartment airflow resistance (cooling drag) increases approximately in proportion to that flow rate.

In order to improve aerodynamic performance, therefore, it is important to reduce the flow volume taken in from the front grille to the necessary minimum. One effective means of reducing the necessary flow rate to the minimum is to direct the incoming flow where it will be used efficiently for cooling, without flowing to areas other than the radiator and other such places where cooling is needed. In the case of the new Accord, this involves installing air guide plates all around the radiator for ducting (Fig. 17). By optimizing the

ducting and the grille opening, cooling drag was reduced 13% relative to the previous model.

6.3. Front-edge Shape of Exhaust System

The exhaust system is exposed under the floor so that there is a vertical wall at the front edge of every enlarged portion. Aerodynamic features were applied to reduce the resulting drag by even the slightest amount (Fig. 18).

There is a protective barrier to prevent contact in front of the pre-chamber. This was therefore given a semispherical shape that flares out toward the rear.

Laser welding was used on the forward edge of the

muffler in order to eliminate the crimped portion. At the same time, the front surface was made to bulge out and the muffler was shaped so that the flow could be deflected smoothly under it. This increased aerodynamic performance 0.6%.

7. Development Using Wind Tunnel with Rolling Road

Toward the end of development of the new Accord, a new wind tunnel with a single-belt rolling road was put into operation⁽⁴⁾ with the capability to simulate flow under conditions closer to actual vehicle operation.

For the aluminum wheels to be used in the Accord Hybrid, development of aerodynamic wheels was carried out using the results of research being pursued during that same period. This took into consideration tire rotation on the rolling road, and the result was improved aerodynamic performance under vehicle operating conditions (Fig. 19).

The final stage of prototype testing in development of the new Accord was also tested in the wind tunnel with rolling road. The size and placement of the front strakes was optimized, and the effectiveness of each under-floor device was confirmed in the wind tunnel with rolling road. The selection of under-floor devices was made under operating conditions.

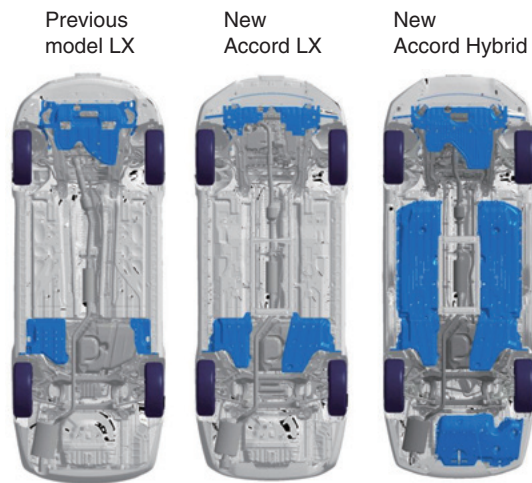


Fig. 16 Under cover variations

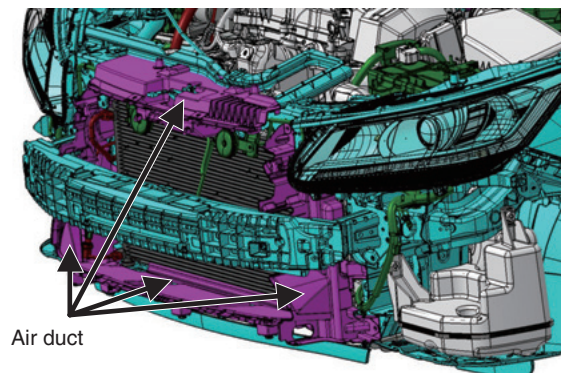


Fig. 17 Ducting around radiator

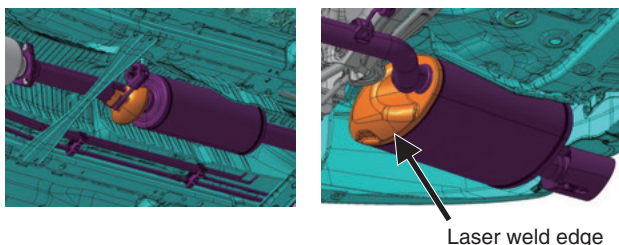


Fig. 18 Aerodynamically modified exhaust

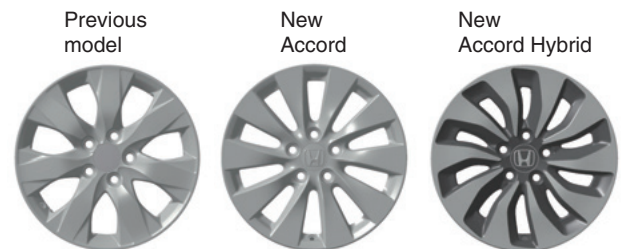


Fig. 19 17-inch wheel variations

8. Conclusion

Results from detailed prior studies were used to analyze airflow phenomena in the new Accord, and aerodynamic improvements were proposed accordingly. Incorporating those proposals at the package design stage of initial styling design enabled improvement of the aerodynamic performance of the exterior styling.

The under-floor flow design policy proposed in the prior studies were applied also to the under-floor devices. This yielded a balance between cost, weight, and fuel economy in the LX grade vehicle, which was taken as the base grade, so that a highly cost- and weight-efficient under-cover could be incorporated. In the Accord Hybrid, all the under-floor devices were adopted and aerodynamic performance was pursued.

In the LX grade vehicle, the increase in aerodynamic performance over the previous Accord did not exceed 1.3%. The decline in aerodynamic performance resulting from the change from steel to aluminum wheels and other measures to increase product appeal was absorbed by the improvement in the exterior. As a result, the vehicle achieved its goals for fuel economy of 24 mpg in city mode, 36 mpg in highway mode, and 30 mpg in combined mode according to U.S. Environmental Protection Agency standards.

The pursuit of aerodynamic performance in the Accord Hybrid led to an increase in aerodynamic performance of 8.3% over the EX grade of the previous model Accord, which had the same tire size. This contributed to achievement of 30 km/L fuel economy in JC08 mode.

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■ Author ■



Daisuke NAKAMURA



Yasuyuki ONISHI



Yoshiyasu TAKEHARA

Development of Side Curtain Airbag Deployment Simulation Technology Using CPM

Hisaki SUGAYA*

Kazuo IMURA*

Hiroyuki MAE*

ABSTRACT

A technology for the simulation of the deployment of side curtain airbags that considers gas flow has been developed using the corpuscular particle method available in the crash analysis software LS-DYNA. The research developed an appropriate modeling method by elucidating the mechanism governing the behavior of the airbag during deployment, and increased the predictive accuracy of deployment simulations. Selecting four modeling parameters, the project analyzed their degree of effect on the behavior of the bag during deployment and identified input values for the parameters using simulations of airbag deployment. As a result, mesh size and time step were identified as the modeling parameters with the major effect on bag behavior. The determined computation input values for these parameters were employed in a side curtain airbag impactor simulation, and results for bag behavior during deployment and the impactor load displacement characteristic corresponded well with test results.

1. Introduction

Today, demand to make the transition to the use of simulations in automotive development is increasing, driven by the need to shorten the development period and reduce the number of prototypes created. In the case of simulations of occupant injuries in a collision, this necessitates modeling technologies for the vehicle, occupant dummies, seats, and safety equipment such as airbags. Simulation technologies for the vehicle framework and for plastic parts such as internal and external fittings have already been proposed by Takada, et al.⁽¹⁾ and Mae, et al.⁽²⁾, respectively. Among safety technologies, while examples of analyses of driver and passenger airbags are available, there have been few analyses of side curtain airbags (SCAB).

SCAB are components designed to protect vehicle occupants in the event of a side collision or a rollover. The period for which occupants' heads need to be protected during a rollover is longer than that for a side collision, and it is necessary to maintain gas pressure for several seconds. For this reason, SCAB employ a structure in which gas is not discharged to the exterior, but instead flows into a separate expanding section known as a sub-chamber in order to maintain pressure, as shown in Fig. 1. SCAB are normally stored folded in the roof; during a collision, gas flows from the inflator, and the airbag is deployed from a folded state (termed "static deployment"). These characteristics make consideration of gas flow essential when modeling SCAB for a simulation.

The Arbitrary Lagrangian Eulerian method (ALE), a

method which uses a computational grid, the Corpuscular Particle Method (CPM) and Smooth Particle Hydrodynamics (SPH) are known as gas flow computation methods employing LS-DYNA. The computation cost of the CPM and SPH is lower than that of ALE, and it is therefore suited to automotive collision computations, which necessitate a high volume of computation. SPH features an influence radius around the particles, which is a range in which the particles mutually interact. This makes it unsuitable for modeling bodies with small gaps, such as a folded airbag. CPM was therefore focused on in the research. According to previous studies of CPM conducted by Lee, et al.⁽³⁾ and Lien, et al.⁽⁴⁾, in the case of front airbags, the method displays a low level of accuracy in simulating the shape and internal pressure of the airbag during deployment. The shape and internal pressure of the airbag during deployment are important elements in the load displacement characteristic of the impactor, which restrains vehicle occupants, and an increase in simulation accuracy is therefore demanded. The fact that the particles used in the model are larger than gas molecules is generally indicated as the reason for the decline in accuracy, but the

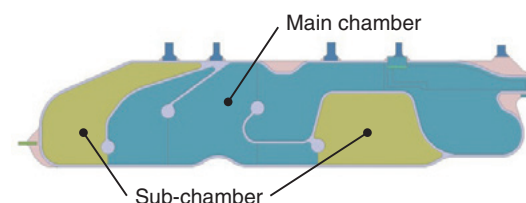


Fig. 1 SCAB structure

* Automobile R&D Center

mechanism of this decline has not yet been clarified.

This paper will therefore elucidate the mechanism governing the behavior of the airbag during deployment, discuss the development of an appropriate modeling method and the associated research process, and consider the results of a comparison of tests and simulations.

2. Development Goals

The following two main aims were set for the development of a SCAB simulation technology that considered gas flow:

- (1) Isolation and determination of the important modeling parameters that affect the static deployment of SCAB; and
- (2) The realization of a technology to increase the accuracy of prediction of the impactor load displacement characteristic.

3. Theory of Computation of Gas Flow and Self-Contact Force

This chapter will discuss two computation methods used in the simulation of the static deployment of SCAB: CPM, a computation method based on the motion of particles, a parameter that affects the deployment of SCAB; and a method of computing self-contact force, which is a reaction force generated when the fabric composing the SCAB is in contact with itself.

CPM is a computation method based on factors including the motion, pressure and temperature of gas. The general concept is grounded in molecular dynamics. Based on the ideal gas law, Eq. (1), and the pressure equation Eq. (2), which employs the average dynamic energy of the molecules and is in general use in the field of molecular dynamics, the velocity of the particles, v_{rms} , can be calculated using Eq. (4).

$$PV = nRT \quad (1)$$

$$P = \frac{1}{V} \sum_{i=1}^N m_i v_i = \frac{1}{3V} m \overline{v^2} = \frac{nMv_{rms}^2}{3V} \quad (2)$$

$$T = \frac{Mv_{rms}^2}{3R} \quad (3)$$

$$v_{rms} = \sqrt{\frac{3RT}{M}} \quad (4)$$

Here, P is pressure, V is the volume occupied by the gas, R is the gas constant, T is the thermodynamic temperature of the gas, M is molar mass, m is the mass of the gas molecules, N is the number of gas molecules in the volume V , and n is the mole number of the gas. The particles lose internal energy due to factors including adiabatic expansion. The velocity of the particles is recalculated based on these

changes in energy. The repetition of these calculations makes it possible to reproduce the expansion of the gas.

The self-contact force, F_{CF} , is expressed by means of Eq. (5), using a penalty method with ΔL as the degree of penetration in relation to the contact thickness and k as the penalty stiffness. In the research discussed here, the penalty stiffness was calculated using the following equations⁽⁵⁾:

$$F_{CF} = k_{CF}(t) \Delta L \quad (5)$$

$$k_{CF}(t) = 0.5 \cdot SLSFAC \cdot SFS \cdot \left(\frac{m_1 m_2}{m_1 + m_2} \right) \left[\frac{1}{\Delta t_c(t)} \right]^2 \quad (6)$$

Here, $SLSFAC$ and SFS are penalty coefficients, m_1 and m_2 are the mass of the elements, and Δt_c is the calculation time step.

4. SCAB Simulation using CPM

The SCAB simulation technology using CPM was constructed in the following three steps. First, the simulation computation input values for each component of the SCAB were determined. Next, using these characteristics, a SCAB static deployment simulation computation was performed, and the important modeling parameters affecting the deployment behavior of the SCAB were identified. Using the determined parameter values, test and simulation results for the deployment time and internal pressure of the SCAB were compared. Finally, using these computation input values, an impactor simulation was conducted, and the load displacement characteristic was compared in simulation and test results.

4.1. Determination of Simulation Computation Input Values for Components of SCAB

Before conducting a simulation of the static deployment of a SCAB, it is necessary to determine the mechanical and leakage characteristics of the SCAB fabric and the characteristics of the inflator. Material and leak tests of the fabric and inflator tank tests were therefore conducted in order to obtain test results and identify simulation parameters.

Tensile and shear characteristics were identified as the mechanical characteristics of the fabric. As Fig. 2 shows, the shape of the SCAB during deployment is a three-dimensional shape made up of a non-inflated area, A , and inflated areas,

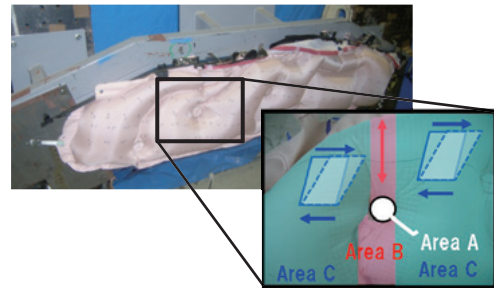


Fig. 2 Deformation of SCAB

B and C. Area B undergoes tensile deformation, and Area C undergoes shear deformation. Because the fabric is subjected to tension in the biaxial direction, a biaxial tensile test was conducted. The shear characteristic was calculated based on the picture frame test shown in Fig. 3⁽⁶⁾. The characteristics obtained from the respective tests were reflected in materials computation values for the SCAB model.

The fabric leakage characteristic was obtained using a blast tube test (Fig. 4). The blast tube was composed of two tanks, A and B, of differing diameters. Tanks A and B were separated by a film that functioned as a diaphragm. SCAB fabric was positioned at the end of Tank B. Air was introduced to Tank A, and when its internal pressure reached 280 kPa, the diaphragm burst and the air moved into Tank B, where it passed through the fabric and was discharged to the exterior. The leakage characteristic of the fabric was calculated based on the pressure history obtained from this test.

$$n = \frac{p_t V_t}{RT} \quad (7)$$

$$D = \frac{d}{V_t m_t} \quad (8)$$

Here, p_t is the pressure obtained from measurements, and V_t is the volume of the tube. The mole number, n , at this time is expressed by Eq. (7). Based on Eq. (7), if the leakage rate per unit of surface area is termed d , the leakage rate for the entire volume, D , can be calculated using Eq. (8)⁽⁶⁾. This leakage rate was set as the computation input value for the fabric leakage characteristic. A blast tube test simulation was conducted in order to verify the validity of the computation input values, and the results were found to display a good correlation with test results for pressure.

The inflator characteristic was obtained by calculating the rate of change in the mass of the inflator gas over time (the

mass flow rate) and the inflator outlet temperature in inflator tank tests. Using a tank with a single enclosed space of 28.3 L, the pressure of the inflator tank at ignition was measured. The mass flow rate of the inflator gas and the inflator outlet temperature were calculated based on the measured value⁽⁷⁾. A simulation of an inflator tank test was conducted using this mass flow rate and temperature as computation input values for the SCAB inflator, and the results were found to display a good correlation with test results for pressure.

4.2. Analysis of Static Deployment of SCAB

4.2.1. Static deployment conditions

In order to determine the modeling parameters that would have an effect on the behavior of the SCAB during static deployment, Model A (Fig. 5) was formulated to help elucidate the mechanism governing the behavior of the SCAB during deployment. Model A was rolled up and constrained at six points. Because the inflator was positioned at Point A in Fig. 5, the gas flowed in the direction shown by the arrows.

4.2.2. Identification of dominant modeling parameters

Model A was used to perform a static deployment simulation, making it possible to clarify the effect of the modeling parameters on the behavior of the SCAB during deployment. To make it possible to quantitatively analyze the relationship between the modeling parameters and the behavior of the SCAB during deployment, the time taken for the gas to pass Line A in Fig. 5 was defined as the deployment time.

The physical elements originating in the behavior of the SCAB during deployment are the self-contact force of the fabric, gas flow, and roll shape. There are four modeling parameters affecting these elements that a user is able to set: Time step, self-contact setting, particle number, and mesh size.

SCAB deployment time was therefore compared under the survey conditions for these four parameters shown in Table 1 (Fig. 6). In order to determine the degree of effect of the parameters, the minimum value for deployment time divided by the maximum value was regarded as the rate of change of deployment time. Calculated rates of change were 32% for mesh size, 32% for time step, 7% for the self-contact setting, and 10% for particle number. Mesh size and time step were therefore judged to be modeling parameters with a major effect on deployment time.

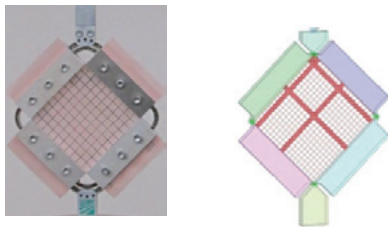


Fig. 3 Picture frame test

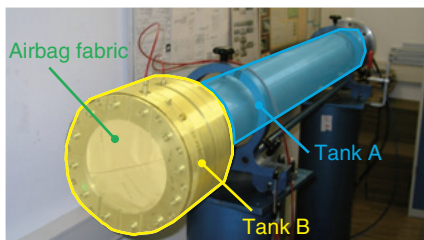


Fig. 4 Blast tube test

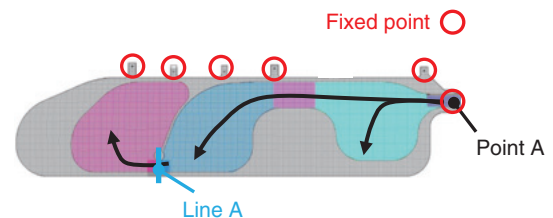
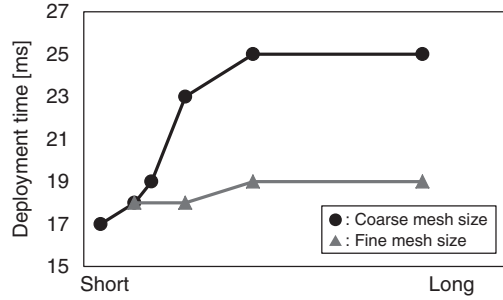


Fig. 5 Model A

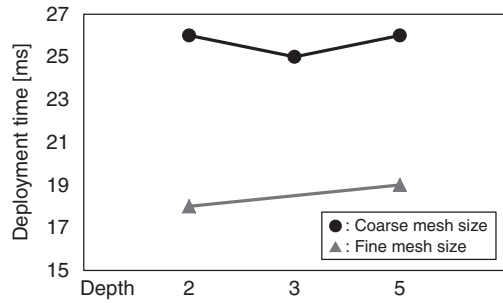
Table 1 Parameter survey conditions

Parameter	Survey conditions
Time step	-100%, -70%, -60%, 200%
Self contact setting	3 patterns
Number of particles	-10%, 200%
Mesh size	-50%, -75%

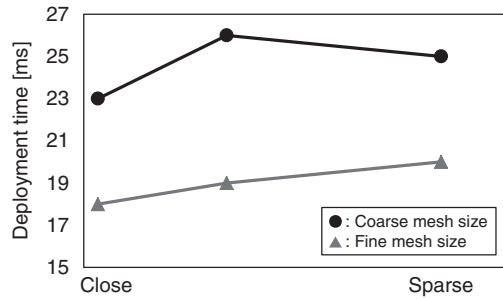
Next, in order to clarify the reason for the change in deployment behavior due to mesh size, a rolled model of an airbag with a single inflating chamber was formulated using



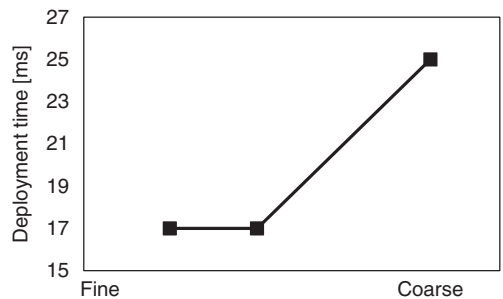
(a) Time step



(b) Self contact setting



(c) Number of particles



(d) Mesh size

Fig. 6 Effects of each parameter on deployment time

different mesh sizes. Figure 7 shows this airbag model prior to deployment and during deployment. When this model was statically deployed using the CPM, deployment behavior changed with the mesh size, as in the case of Model A. Two factors can be considered to be involved in this phenomenon.

First, the degree of freedom of nodal points increases. Figure 8 shows the status of the particles on the lower surface of the airbag fabric (hidden in the model) for two cases, Models α and β , which use the coarsest and the finest mesh sizes respectively, and that were created in order to identify the gas flow state. Compared to the coarse model, there are large numbers of particles flowing into the gaps in the fabric in the model using the fine mesh. Figure 9 shows sections from the centers of the models to provide a time history. In the case of the fine mesh, the roll diameter is compressed into concentric circles, and the gaps between the fabric increase. Because of this, a greater number of particles flow into those gaps. It is considered that this is because the degree of freedom of mesh nodal points increases, and the fabric on the interior of the roll is able to deform freely within the roll (Fig. 10).

Another factor is the increase in the degree of penetration in relation to the contact thickness of the fabric in the case of the coarse mesh. As Fig. 11 shows, with the

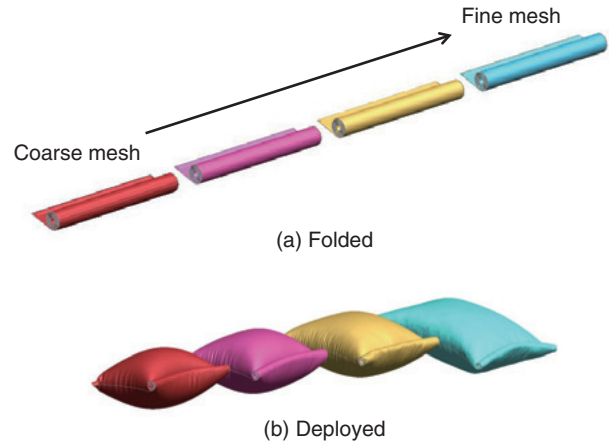


Fig. 7 Single-chamber model

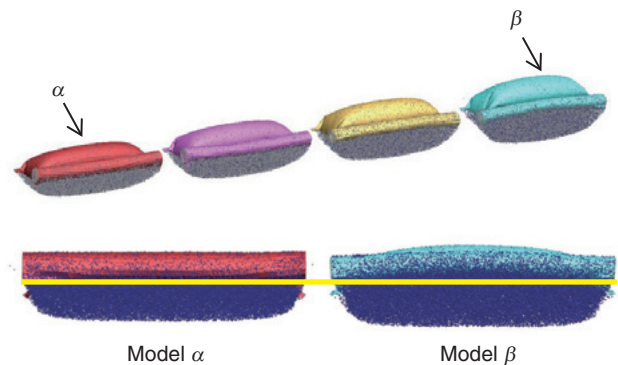


Fig. 8 Front view of single-chamber model

increase in penetration of the contact thickness of the fabric when the coarse mesh is used, it is possible that the contact reaction force will act to resist the unfolding of the SCAB.

In the case of time step, Eq. (5) showed that this parameter has an effect on the self-contact force of the airbag. It was predicted that an increase in time step would reduce the contact force, thus delaying deployment. The results shown in Fig. 6(a) correspond to this prediction, indicating the necessity of setting an appropriate value for this parameter. Because there is a high rate of penetration between the fabric sections when the coarse mesh is used, the effect of the time step is significant, with a rate of change in deployment time of 32%. In the case of the fine mesh, the rate of change in the behavior of the bag during

deployment dropped to 5%, indicating that the use of a fine mesh counteracted the effect of this parameter.

4.2.3. Verification of parameter values

With the mesh size and the time step set at fixed values, the fluctuations in behavior during deployment disappeared, and deployment time converged on a fixed value. A mesh size and time step that would commence the convergence with minimal computation time were selected, and the simulation results were compared with test results. Figure 12 shows the behavior of the SCAB at the time when the gas has passed Line A in Fig. 5. In both sets of results, the gas passes Line A at 18 ms, and results for deployment time match. Figure 13 shows a comparison of the time

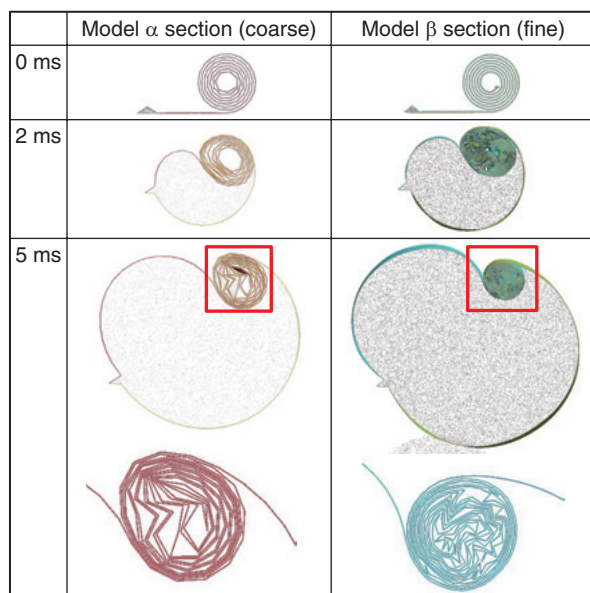


Fig. 9 Sections of centers of single-chamber models

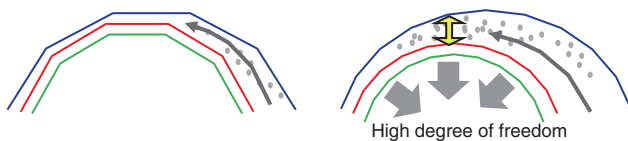


Fig. 10 Effects of mesh size (Degree of freedom)

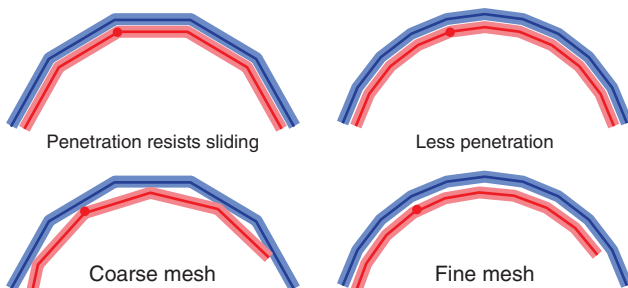


Fig. 11 Effects of mesh size (Penetration)

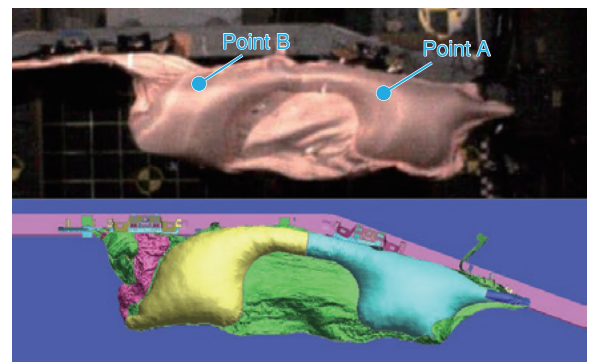
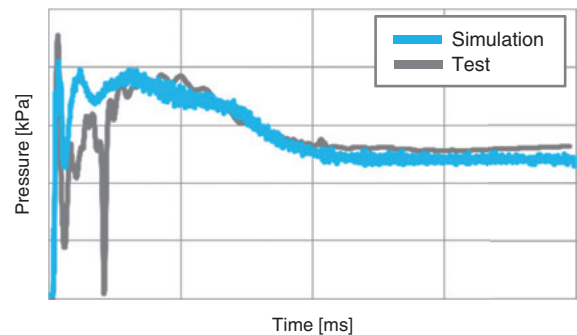
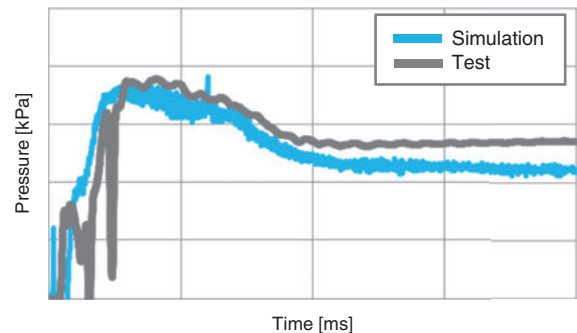


Fig. 12 Model A deployment at 18 ms



(a) Pressure at Point A



(b) Pressure at Point B

Fig. 13 Pressure during Model A deployment

history of internal pressure in the airbag. The simulation and the tests produced similar results for internal pressure.

4.3. Comparison of Impactor Performance

Using two SCAB that are fitted in different vehicles (Model B, fitted in sedans, and Model C, fitted in SUVs), impactor simulations were conducted based on the modeling computation input values discussed above, and results were compared with test results with a focus on the load displacement characteristic. The conditions for the impactor test were an impactor mass of 4.5 kg and an initial velocity of 32 km/h.

Model B SCAB, which is employed in sedans, is stored folded in a roll. The diameter of the folded roll is 30 mm, and the inflator is positioned at the rear end of the vehicle. Using this SCAB, two simulations were conducted: Case A, using standard setting values for mesh size and time step; and Case B, using the setting values obtained through this research, and the results were studied. Figure 14 shows a time history of images of static deployment of the SCAB during the impactor test, and compares simulation results with test results. In Case A, deployment time was slower in the simulation results than in test results, and the volume of the SCAB was lower at the points at which the impactor contacted it.

Because of this, the impactor load for Case A was considered too high in the impactor load displacement characteristic shown in Fig. 15. For Case B, simulation results matched well with test results for SCAB deployment time and the impactor load displacement characteristic, and the load displacement characteristic of the impactor

received from the airbag also matched test results well.

Like Model B, the SUV-type SCAB, Model C, is stored rolled, and when rolled its diameter is 33 mm. The inflator is positioned close to the center of the vehicle, and the gas flow differs from the sedan-type SCAB, with gas flowing from the inflator to the front and rear of the vehicle.

Results for the load displacement characteristic from an impactor simulation conducted using the modeling settings identified in this research were compared with test results. Even in the case of a gas flow differing from the sedan-type SCAB, simulation results for airbag behavior during deployment (Fig. 16) and impactor load displacement characteristic (Fig. 17) displayed a good correlation with test results.

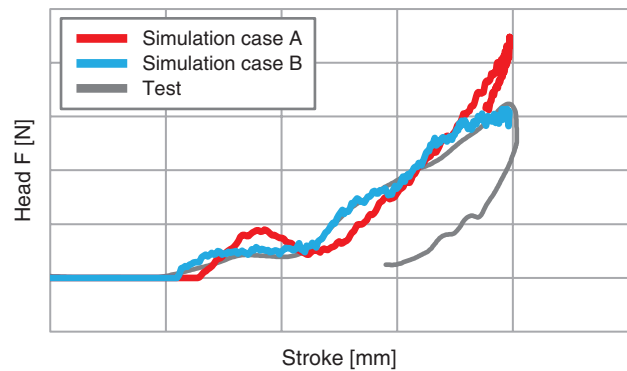


Fig. 15 Load-displacement history of impactor (Model B)

	CAE		Test
	Case A: Coarse mesh	Case B: Fine mesh	
15 ms			
17 ms			
20 ms			
25 ms			

Fig. 14 Deployment behavior of sedan-type SCAB (Model B)

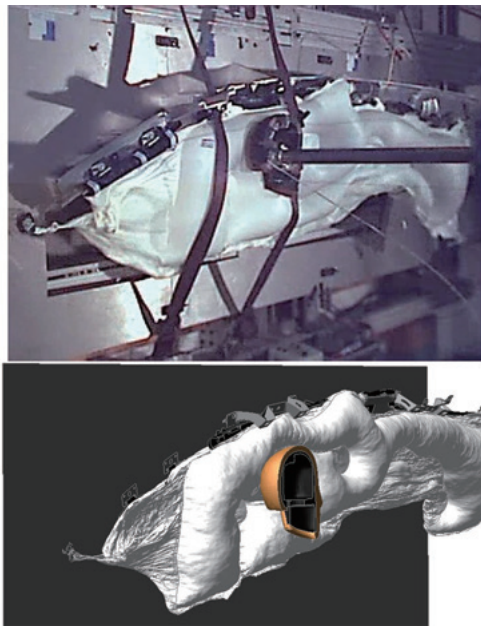


Fig. 16 Deployment behavior of SCAB SUV-type (Model C)

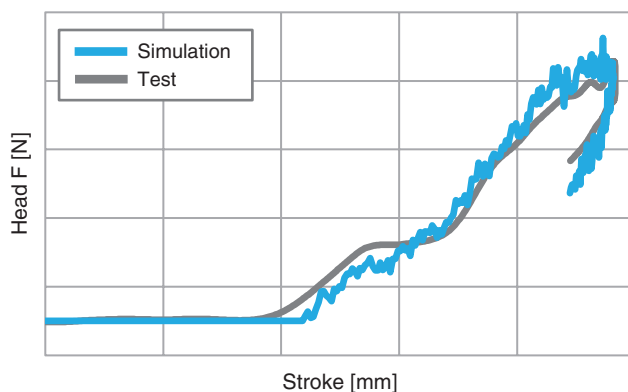


Fig. 17 Load-displacement history of impactor (Model C)

5. Conclusion

A technology for the simulation of the deployment of SCAB, which considers gas flow, has been developed using CPM. This project produced the following outcomes.

- (1) Mesh size and time step were identified as the modeling parameters with the greatest effect on the static deployment time for SCAB, and the mechanism governing the behavior of the SCAB during deployment was elucidated, making it possible to develop a method of setting appropriate modeling parameters.
- (2) It was possible to use simulations to predict the behavior of SCAB during static deployment and the impactor load displacement characteristic.

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■ Author ■



Hisaki SUGAYA



Kazuo IMURA



Hiroyuki MAE

Investigation of Demand for Electric Vehicles in China

Junya TSUNEISHI*

Yoshitaka TAKASUKA*

Yoshiko TAKAHASHI*

ABSTRACT

Electric vehicles (EV) are expected to play an important role as next-generation mobilities. Honda, in cooperation with the People's Government of Guangzhou Municipality in the People's Republic of China and the Guangzhou Automobile Group Co., Ltd., has conducted a social field test using EV with the aim of popularizing new energy vehicles. The results confirmed by this field test and the results of user surveys showed that although EV have lower running costs than gasoline vehicles and are also eligible for preferential treatment as new energy vehicles, users feel some unease and hesitate to purchase EV due to short driving ranges and a lack of charging locations in urban areas, so demand remains latent. In addition, it was also found that users do not fully understand the technical safety aspects of the lithium-ion rechargeable batteries used in EV. In order to elicit demand for EV in China, it is necessary to enhance performance such as driving ranges, construct charging infrastructure, and address unease regarding the safety of lithium-ion rechargeable batteries.

1. Introduction

The Chinese government promoted the installation of charging facilities and other infrastructure in parallel with trial operation of new energy vehicles such as electric vehicles (EV), plug-in hybrid vehicles (PHEV) and fuel cell electric vehicles (FCV) in major metropolises from 2008 to 2012. In addition, the Plan for Development of the Energy Efficient (HEV) and New Energy Automobile Industry (2012-2020)⁽¹⁾ announced in June 2012 stated the goals of popularizing new energy vehicles (cumulative sales of five million units) and expanding the production scale (two million units/year) by 2020.

On the other hand, sales of new energy vehicles in China⁽²⁾ increased to 8,638 units (including 5,655 EV) in 2011 and 12,791 units (including 11,375 EV) in 2012. Sales for the most recent six months from January through June 2013 totaled 5,889 units (including 5,114 EV), which is an increase of 67 points (48 points for EV) over the 3,525 units (including 3,535 EV) sold from January through June 2012. Thus, it appears that new energy vehicles are starting to become popular.

Against this background, Honda conducted a social field test (field test) using EV in Guangzhou City in China from November 2011 to January 2013. This field test confirmed EV driving ranges and driving cost reduction effects in actual traffic environments, and the charging infrastructure situation and power supply quality in Guangzhou City. In addition, surveys were conducted to clarify issues such as when general automobile users (users) use EV and purchase demand (demand). This paper describes the background

of EV popularization in China and reports the results confirmed by the field test and the results of investigating conditions that will lead to increased EV demand in China based on user surveys.

2. Background and Purpose of Investigation

2.1. EV Technical Targets in Official Policies to Popularize New Energy Vehicles

The Plan for Development of the Energy Efficient and New Energy Automobile Industry (2012-2020)⁽¹⁾ announced in June 2012 stated the EV technical targets to be achieved by 2015 as follows.

- (1) Vehicle performance: maximum speed of 100 km/h or more and a driving range of 150 km or more
- (2) Rechargeable battery module performance: energy density of 150 Wh/kg or more and a cycle life of 2,000 times or more or 10 years or more

These technical targets are not yet designation criteria for new energy vehicles as of September 2013, but equivalent specifications are generally acknowledged with regards to EV driving range and rechargeable battery module life.

2.2. Subsidies and Preferential Treatment Systems for New Energy Vehicles

According to the official publication of the Central People's Government of the People's Republic of China released in September 2013⁽³⁾, EV approved to receive subsidies as new energy vehicles are eligible for preferential measures such as the following.

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The subsidies (upper limit) provided to automobile manufacturers and reflected to sales prices for fiscal year 2013 are CNY 35,000 per vehicle for EV with a driving range of 80 km or more to less than 150 km, CNY 50,000 per vehicle for EV with a driving range of 150 km or more to less than 250 km, and CNY 60,000 per vehicle for EV with a driving range of 250 km or more. Subsidies provided to automobile manufacturers for fiscal years 2014 and 2015 are calculated by multiplying the fiscal year 2013 subsidies by the coefficients 0.9 and 0.8, respectively. In addition, subsidies are also provided to purchasers from local governments. For example, Guangzhou City provided CNY 10,000 per vehicle from 2012 to 2013. Furthermore, the cities of Beijing, Shanghai, Guiyang and Guangzhou restrict the number of vehicles registered due to concerns over chronic traffic congestion in urban areas and environmental issues. However, new energy vehicles are eligible for preferential measures during vehicle registration, such as the establishment of a separate category. Guangzhou City has restricted the number of vehicles registered since 2012, and persons planning to purchase a vehicle need to either win a free lottery or apply with a minimum bid of CNY 10,000 in order to register the vehicle. However, the 2012 and 2013 limits for the number of new energy vehicles registered were set separately from the limits for gasoline vehicles and there were also few applications, so new energy vehicles could be registered free of charge without a lottery.

2.3. Purpose of Investigation of EV Demand

In April 2011, Honda signed a memorandum concerning the joint promotion of the new energy vehicle industry with the People's Government of Guangzhou Municipality and the Guangzhou Automobile Group Co., Ltd. A social field test and investigations of charging infrastructure aimed at popularizing new energy vehicles were then promoted jointly based on the provisions of that memorandum. Actual driving using EV started in Guangzhou City from November 2011, and the following items were investigated and verified.

- (1) Data was collected and verified regarding driving ranges and year-round power consumption for air conditioning in actual traffic environments.
- (2) Information was gathered regarding the planned pole-type charging infrastructure in Guangzhou City, and the current installation situation and general power supply quality were investigated.
- (3) User attitudes regarding EV usage were surveyed and EV demand was investigated.

The EV used in the field test were prototype vehicles consisting of an EV system installed in a 2008 model year FIT body. The driving range absolute values and variance were evaluated and compared with Japanese field test data, and Chinese market trends were confirmed. The charging infrastructure was investigated by on-site verification and also by exchanging information from June 2011 with each division involved with the new energy vehicle business of

the Guangzhou Power Supply Bureau of the China Southern Power Grid (Southern Power Grid), which is installing the charging infrastructure. User surveys were conducted by questionnaires and interview surveys of people visiting prototype vehicle exhibitions and test drive events in Guangzhou City, and by quantitative and qualitative surveys to confirm issues and demand for EV in major cities.

This investigation report examines whether EV that satisfy the conditions described in section 2.1 can elicit user demand based on the results confirmed by the field test and user concerns confirmed by user surveys, and confirms the conditions for popularizing EV.

3. Field Test in Guangzhou City

3.1. Overview of Field Test and Results

Urban areas in China have high traffic concentrations and experience chronic traffic congestion. Many people living in urban areas commute by car to factories and other workplaces located in designated development districts in the suburbs. In addition, rental cars are not popular, so visitors on business trips and foreigners are generally driven to and from airports. In consideration of this vehicle usage environment, driving courses were set as shown in Table 1 and a field test was conducted using three EV. The following results were confirmed.

The EV driving range per full charge averaged 169 km as shown in Fig. 1. The center values and upper and lower limit values of the driving range for each course confirmed that the variation was greater for the lower limit side from the center value. The driving range was shortest for the traffic jam route, but the chance of experiencing unexpected traffic congestion due to a traffic accident or a disabled vehicle was 1 percentile or less, so it is thought that there is almost no risk of running out of electricity as long as the driving distance is 100 km or less.

Figure 2 shows the year-round power consumption for air conditioning confirmed by field tests in Guangzhou City and in Kumamoto City in Japan. There was no noticeable difference when using the cooler during the summer with outside temperatures in excess of 30°C, and the power consumption was 0.5 kW in both cases. However, the temperature drops to 10°C or less during the winter in Japan, so the heater power consumption was approximately 0.5 kW to 1 kW higher than that in Guangzhou City. This result indicates that the drop in driving range during the winter due to air conditioning is less in warm regions where heater power consumption is lower compared to cold districts, which is advantageous for EV usage.

Figure 3 shows the vehicle stopping frequency on the vertical axis and the average vehicle speed for the corresponding sections on the horizontal axis when driving in urban areas as confirmed by field tests in Saitama City and Kumamoto City in Japan and Guangzhou City in China. Under urban traffic conditions with an average vehicle speed between 20 km/h and 30 km/h, the stopping

Table 1 Driving route and distance in Guangzhou area

(1) Traffic jam route in Guangzhou city

Route name	Driving route	Distance	Via points
Traffic jam route 1	Inside Guangzhou city area	20 km	Pearl River
Traffic jam route 2	Inside Guangzhou city area	14 km	Garden hotel
Traffic jam route 3	Inside Guangzhou city area	31 km	LiuHuaHu Gong Yuan, Pearl River
Traffic jam route 4	Inside Guangzhou city area	32 km	LiuHuaHu Gong Yuan, Garden hotel

(2) Commuting route to suburban work place

Route name	Driving route	Distance	Via points
Commuting route	Guangzhou city to Guangzhou Honda factory	58 km	Guangyuan Expressway

(3) Baiyun airport route via highway

Route name	Driving route	Distance	Via points
Airport route 1	Guangzhou city to Baiyun airport via highway	76 km	Garden hotel
Airport route 2	Guangzhou city to Baiyun airport via highway	130 km	Garden hotel, Guangzhou Honda factory

frequency in the Saitama urban area when travelling at an average vehicle speed of 20 km/h exceeded approximately 30%. In contrast, the stopping frequency in the Guangzhou urban area when travelling at the same average vehicle

speed was approximately 20%. The fact that the average vehicle speed was the same even though the stopping frequency was less is thought to indicate that the speed range used when driving in the Guangzhou urban area is lower than that when driving in urban areas in Japan. In addition, Figure 4 shows the power consumption minus

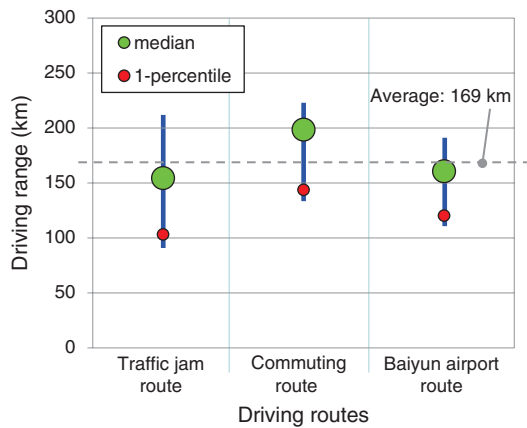


Fig. 1 EV driving range

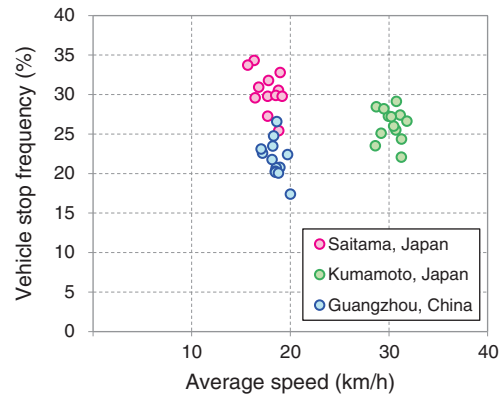


Fig. 3 Vehicle stop frequency in city

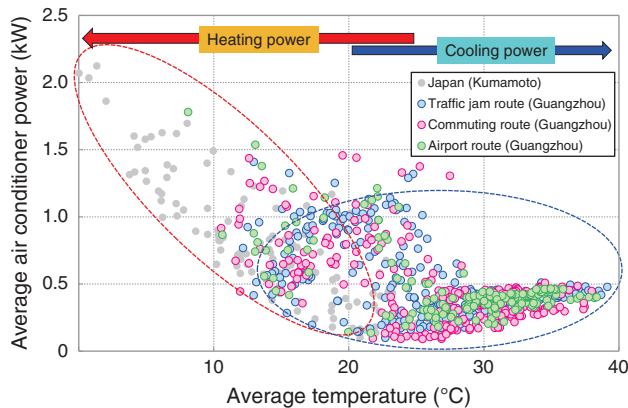


Fig. 2 Annual power consumption from air conditioning

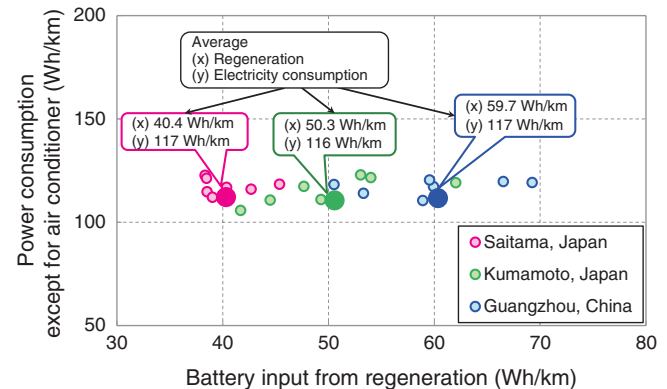


Fig. 4 Regeneration from braking in city

Table 2 Nominal electricity rate and gasoline price

City	Nominal electricity rate CNY/Wh	Regular gasoline price CNY/L	Ratio of Gasoline price to electricity rate
Guangzhou, China	0.610	7.69	12.6
Shanghai, China	0.617	7.75	12.6
Beijing, China	0.483	7.81	16.2
Yokohama, Japan	(25.19 JPY/Wh)	(149.2 JPY/L)	5.9

the power consumed by air conditioning on the vertical axis and the power recovered by regenerative braking on the horizontal axis. Regenerative braking in the Guangzhou urban area averaged 59.7 Wh/km and the driving power consumption was 117 Wh/km, so adding these together yields an energy consumption of 176.7 Wh/km for driving. Similar calculations show that energy of 157.4 Wh/km is consumed for driving in the Saitama urban area and 166.3 Wh/km in the Kumamoto urban area. That is to say, the power required for acceleration, or the power loss, is thought to be greater when driving in the Guangzhou urban area than when driving in urban areas in Japan. Therefore, technology that suppresses power consumption from launch to approximately 30 km/h should be considered to support the user environment in China.

Figure 5 shows that the annual electricity cost assuming a commuting distance of 30 km each way and 240 days of use per year (annual driving distance of 14,400 km) is approximately CNY 1,300. This is a driving cost reduction effect of approximately CNY 5,600 compared to a 2008 model year FIT gasoline vehicle of the same class, and is more than the average monthly wage for a nonmanufacturing sector general worker⁽⁴⁾. In addition, Table 2 shows the difference in electricity rates and gasoline prices between China and Japan, and shows that the price of one liter of gasoline is equivalent to electric energy of approximately 5.9 kWh in Japan and approximately 12.6 kWh in Guangzhou City. The average wage in China⁽⁴⁾ is approximately 1/4 to 1/3 that in Japan, so it is thought that the driving cost when replacing a gasoline vehicle with an EV that operates by electricity with stable supply and rates

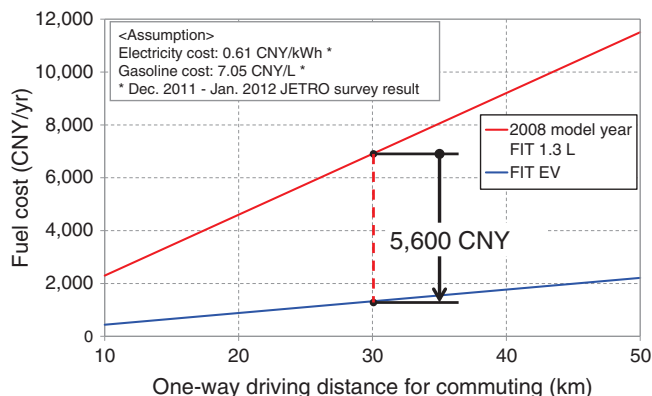


Fig. 5 Fuel cost reduction from EV in China

will seem much less expensive from a Chinese perspective compared to a Japanese perspective.

These results are thought to indicate that EV are suited to users with an estimated daily driving distance of 100 km or less who emphasize driving cost.

3.2. Technical Issues to be Addressed

Trouble occurred when charging the running battery during the field test in Guangzhou City, so the power supply quality was investigated. This investigation confirmed the following situation that should be taken into account as conditions when designing the charging functions.

Power receiving facilities equipped with a voltage converter from high voltage to low voltage such as shown in Fig. 6(a) have a power factor regulator such as shown in Fig. 6(b) on the low voltage side to reduce the contract electricity rate. When the power load becomes high, this power factor regulator automatically connects to and disconnects from the power distribution source circuit in accordance with the power factor. At this time the power supply voltage used for charging fluctuates as shown in Fig. 6(c) due to the effects of the large-capacitance capacitor of the power factor regulator.

The general power supply (AC220 V) of collective housing is distributed using a three-phase four-wire system. The TN-C earthing system is widely used, and the earth wire (E wire) of each housing unit is connected to the protective earth conductor (PE conductor) for the entire building. In the field test, cases of wiring work were confirmed where the neutral wire (N wire) is connected to ground on the side of low-power devices such as fluorescent lights. In these cases, noise from other devices is superimposed onto the E wire of devices via the PE conductor as shown in Fig. 7, even when the room power supply is wired correctly.

The on-board charger specifications were reviewed in consideration of these power supply quality investigation results. In addition, it was confirmed that a stabilized power supply that can convert from AC to DC and then to AC again needs to be added when the distortion in the power supply voltage waveform increases. Furthermore, adding commercially available reactors and ferrite cores to the signal lines was confirmed to be an effective measure against noise entering via the PE conductor.

When developing EV for the Chinese market in the future, it is thought that the power supply quality should be investigated as in this field test, and compatibility with the

charging infrastructure to be installed should be verified as appropriate.

3.3. Charging Infrastructure Layout Plan

In parallel with on-site investigation of the general power supply quality and confirmation of countermeasure effects, information was also exchanged with the Southern Power Grid from June 2011 to obtain information regarding the installation plans and layout situation of pole-type charging infrastructure. The charging system, standards and approach to power distribution of the Southern Power Grid are as follows.

(1) The Southern Power Grid is promoting the construction of battery replacement-type charging stations together

with quick charging facilities within Guangzhou City in line with the government policy aiming for popularization of battery replacement-type EV.

(2) The Southern Power Grid is also promoting the installation of pole-type charging infrastructure equipped with normal chargers and quick chargers. However, priority is being given to areas used for field tests and other requests by automobile manufacturers that are currently producing or have plans to produce EV rather than locations that take into account general use. As a precedent example, this infrastructure is already installed in the Huadu district and Shenzhen City.

(3) All charging-related standards except for test measures requested by automobile manufacturers use the national

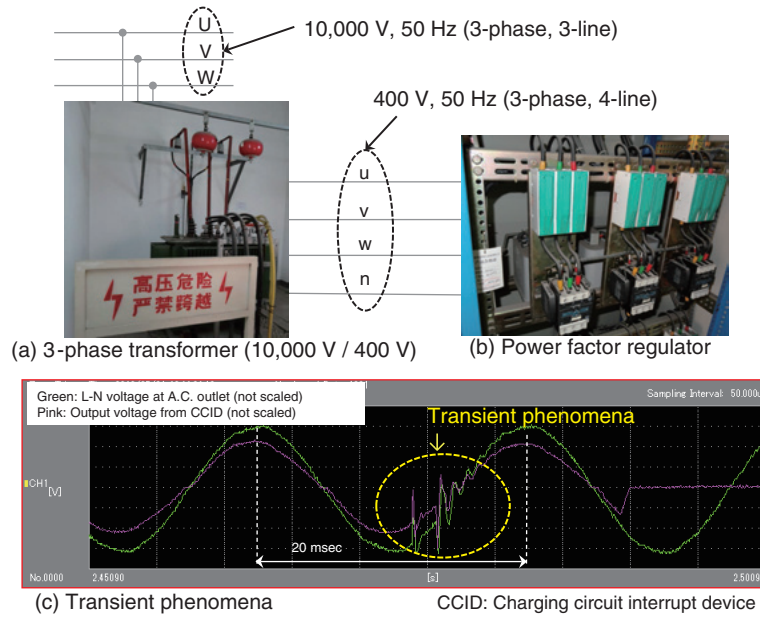


Fig. 6 Transient phenomena of voltage from power factor conditioner switching

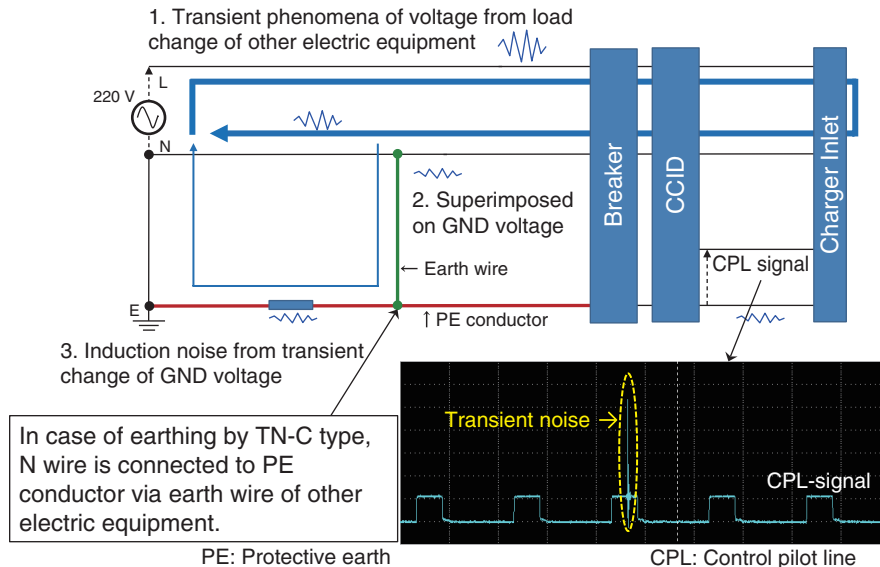


Fig. 7 Induction noise in CPL-signal from GND voltage's transient change

Table 3 Issues of EV dissemination found in preliminary survey

(1)	People will not be satisfied with EV in terms of driving distance and charging time.
(2)	People will have uneasiness from seeing almost no available charging facilities.
(3)	People will have uneasiness about technical methods for battery safety.
(4)	The battery is expensive, and thus EV are expensive.
(5)	People will think that the EV acceleration and maximum speed are insufficient.

standard (GB/T)⁽⁵⁾.

- (4) There are concerns over drops in power supply quality such as power supply voltage fluctuation within the same power distribution area when large numbers of EV are charged at the same time from general outlets (AC 220 V, 16 A). Therefore, installation of a power distribution network for charging is planned.

Pole-type charging infrastructure was not installed along the driving courses set for the field test, so there was no chance to use this infrastructure. However, the installation of charging infrastructure is essential for the popularization of EV, and requires attention in the future.

4. Investigation of EV Purchases and Usage

4.1. Results of Preliminary Survey

According to an investigation by the China Association of Automobile Manufacturers⁽²⁾, 5,655 EV were sold in China in 2011, which accounted for 0.03% of the total vehicle sales of 18,505,100 units. Given this percentage it was predicted that a reliable number of samples (n) could not be gathered for a general statistical survey concerning EV, so prospects for popularization and an image of users were obtained as an advance survey in 2012 by interviewing editors of automotive trade journals and informed authorities who had written about environmental investigations and EV. As of 2012, these authorities held generally positive attitudes toward the use of EV as a direction for replacing gasoline vehicles. However, they also pointed out the issues from the user's viewpoint shown in Table 3, and said that they felt current conditions were not conducive to expanding popularization. User surveys were conducted as described below to confirm whether there is actually latent demand for EV and investigate the actual situation regarding the indicated issues.

4.2. User Survey Results

Figure 8 shows the attributes of the subjects of the survey conducted from late April through early June 2013. The three major metropolises of Guangzhou, Shanghai and Beijing were selected as the subject cities from among the cities announced in the official publication of the Central People's Government of the People's Republic of China released in June 2012⁽¹⁾ concerning the popularization of new energy vehicles for promotion of charging infrastructure installation and provision of subsidies to purchasers. n was set at 500 for each city, for a total survey n of 1,500.

Table 4 Frequency of driving (n=1,500)

(1) Frequency of driving (%)

Almost every day	72.4
4-5 times per week	22.3
2-3 times per week	4.6
1 time per week	0.7
2-3 times per month	0.0
1 time per month	0.0

(2) Monthly driving distance (%)

Less than 100 km	1.6
100-499 km	15.1
500-999 km	22.5
1000-1999 km	33.4
2000-2999 km	14.9
Over 3000 km	12.5

Table 4(1) shows the car usage frequency. Almost every day accounted for 72.4% and four or five days a week accounted for 22.3%. In addition, Table 4(2) shows the monthly driving distance. Here, 500 km to 999 km accounted for 22.5% and 1,000 km to 1,999 km accounted for 33.4%. Calculating the average daily driving distance shows that (2,000 km ÷ 20 days) 100 km or less covers nearly all cases. Based on these results, it is thought that even EV with a driving range of approximately 100 km would enable movement without hindrance in most cases of car usage for daily living and commuting.

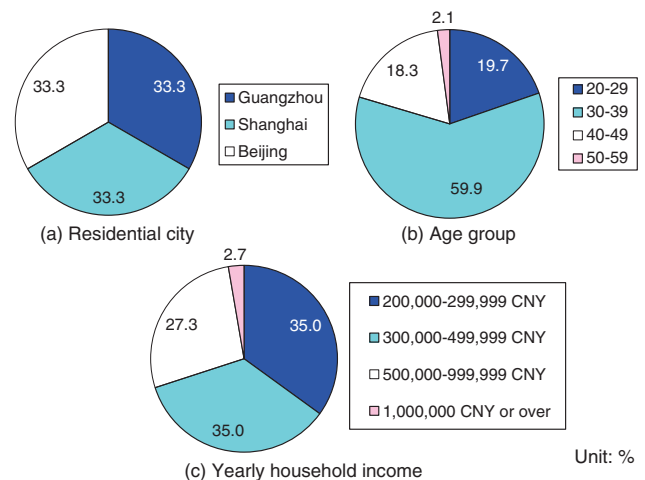


Fig. 8 Basic profiles of persons for survey (n=1,500)

Figure 9 shows that the most common purchase price range for a first vehicle is CNY 200,000 to CNY 290,000 at 29.1%. The most common budget range for the next car considered for purchase is also CNY 200,000 to CNY 290,000 at 29.9%, but the less than CNY 200,000 percentage is lower and the CNY 300,000 or more percentage is higher, which is thought to indicate a tendency toward higher grade vehicles. Figure 10 shows that the most common purchasing budget when assuming an EV is CNY 200,000 to CNY 290,000 at 33.7%, and the overall percentages are approximately the same as for currently owned vehicles. High price was noted as an issue for EV popularization in Table 3, but this result could be said to indicate that as long as the price is approximately the same as their current gasoline car, users would not feel that EV were expensive and would consider purchasing.

Figure 11 shows the desire to use EV, and a high percentage of 82.6% of users responded that they want to use very much or want to use EV. However, as shown in Fig. 12, 62.6% responded that the motive power type of the next car they were considering purchasing was gasoline, 32.5% responded HEV, and only 1.7% responded EV, so there is a large gap compared to the desire to use percentages. That is to say, EV demand is thought to be latent.

Table 5 shows the priority items (multiple answers possible) when considering EV purchase. In order from the most common, driving range per charge accounted for

Table 5 Significant points for EV purchase (n=1,500)

Significant points	%
Driving range per one charge	72.6
Battery safety	63.5
Availability of on-street charging infrastructure	62.0
Short charging time	60.9
Maintenance cost	50.1
Availability of service in case of battery trouble	46.1
Low charging cost	40.7
Can charge at home	39.1
Low purchase price	33.1
Design	27.3
Subsidy and support from government	20.9

72.6%, battery safety for 63.5%, availability of on-street charging infrastructure for 62.0%, and short charging time for 60.9%. These are actual reasons that kept users from purchasing EV, so it is thought that demand will materialize if these issues can be addressed. Note that these issues also match the issues for EV popularization indicated by the advance investigation as noted in Table 3.

Desire to use EV is high, but issues perceived by users have not been addressed, so the current situation is that the only people purchasing and using EV are early adopters, and popularization is lagging.

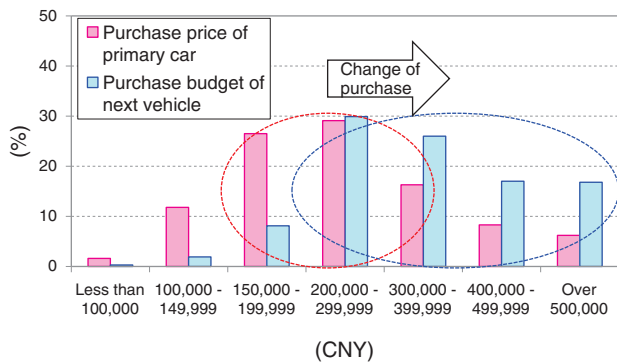


Fig. 9 Purchase price and budget of vehicle (n=1,500)

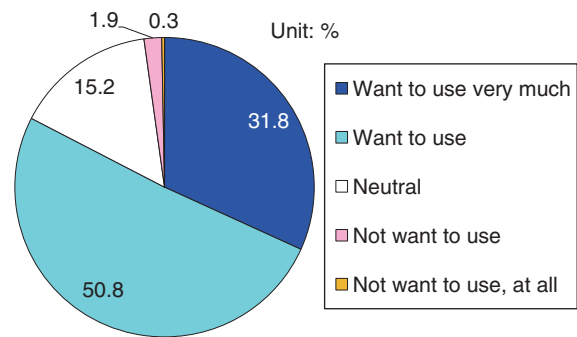


Fig. 11 Willingness to use EV (n=1,500)

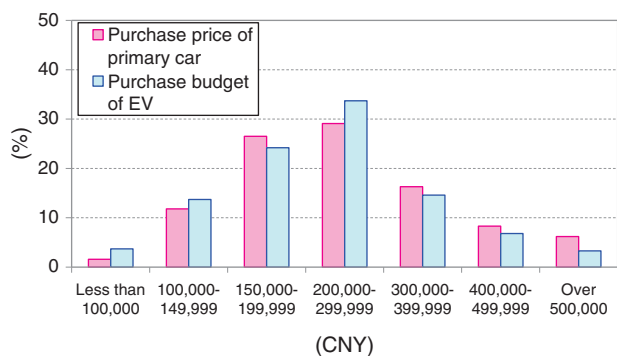


Fig. 10 Purchase budget of EV (n=1,500)

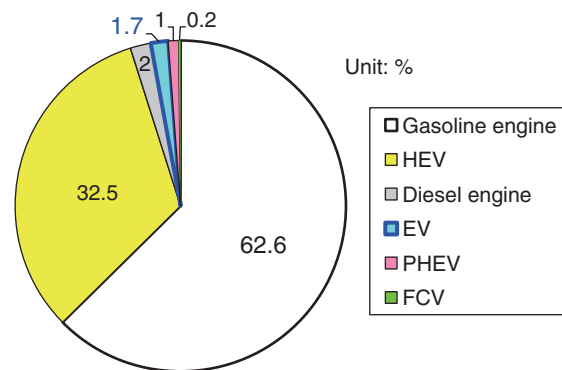


Fig. 12 Power type of next vehicle (n=1,500)

Table 6 Perception gap between Honda and prospective users

Concerns	Honda	Users
Driving range per one charge	· 100 km per day is sufficient.	· At least 80 km per day is necessary. · 200-300 km for 2 or 3 days is ideal.
On-street charging infrastructure	· Number of on-street chargers is insufficient as main charger. · Charging from electrical outlet at home is recommended.	· Number of on-street charger is insufficient. · People don't know where available chargers are.
Shortening of charging time	· Short charging time may be necessary for emergency. · In case of emergency, fully charging every night is recommended.	· Shortening of charging time is necessary. · People often forget when last charging was.
Safety technology of rechargeable battery	· Technologies to use Li-ion battery are established. · Manufacturer's warranty is provided.	· Most of people don't know safety technologies and information about Li-ion battery. · Sometimes people hear news of battery explosion.

4.3. Issues for EV Popularization

Items (1) to (3) in Table 3 that were indicated by the advance investigation were confirmed quantitatively by the prospective user survey, and qualitative research was also performed to confirm the differences from the field test assumptions and perceptions. Table 6 organizes the differences between the field test assumptions and prospective user attitudes regarding EV usage. These results confirmed that while it was assumed in the field test that EV would be charged every night at prospective users' homes, prospective users thought differently. This is thought to be influenced by the characteristic Chinese housing style whereby nearly all urban residents live in condominiums. In addition, it was also confirmed that prospective users do not fully understand the technical safety aspects of the lithium-ion rechargeable batteries used in EV.

Regarding the lack of power performance that prospective users feel is an issue, cheap and slow EV that use lead batteries are already used in China, so it was predicted that this has created preconceptions, and prospective user opinions were confirmed after having them test drive the EV used in the field test. As shown in Fig. 13, prospective user images of EV acceleration performance were fairly poor to inferior before the test drive but changed for the better after the test drive, which confirmed that there had been negative preconceptions.

The field test results described in section 3.1 and the survey results described in section 4.2 are thought to indicate that there are potential prospective users considering EV purchase and use. In order to elicit this demand, manufacturers and parties promoting EV popularization need to understand the background for prospective user assumptions and perceptions, develop technology to extend driving ranges, install charging infrastructure, explain the technical safety aspects of lithium-ion rechargeable batteries, and clear up mistaken

images concerning power performance.

Given the current situation of latent EV demand, possible approaches are as follows.

- (1) Manufacturers should work to extend EV driving ranges and also address the issue of power supply quality to enable charging from typical household outlets. If charging outlets are available at homes and workplaces where prospective users park for long periods, it is thought that the charging task will become a habit for prospective users, which will help to reduce dependence on pole-type charging infrastructure in urban areas and also reduce concerns over short driving ranges, charging locations and charging times.
- (2) Manufacturers should investigate prospective user concerns about the safety of lithium-ion rechargeable batteries and explain that proper technical measures are taken.
- (3) Manufacturers and parties promoting EV popularization should offer short test drives and rental opportunities that enable prospective users that want to use EV to confirm driving ranges, the charging task and power performance in actual usage environments.

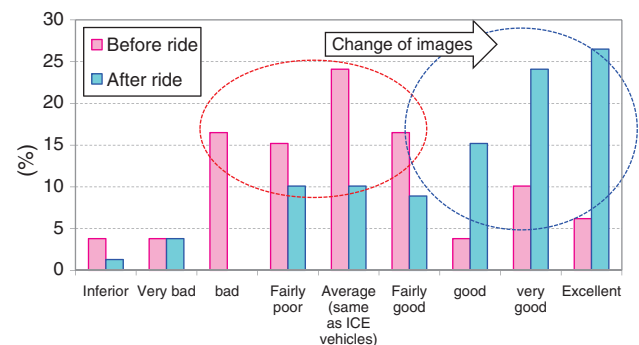


Fig. 13 Images of EV acceleration (December 2012, Guangzhou, China, n=79)

5. Conclusion

A field test conducted in Guangzhou City in China confirmed that given a driving distance of 100 km or less, there is almost no risk of running out of electricity, and a reduction in driving cost can be expected. In addition, it was confirmed that technology to suppress power consumption from launch to approximately 30 km/h and measures to support the quality of the typical power supply used for charging are needed to suit the Chinese market.

User surveys confirmed that approximately 70% of users have an average daily driving distance of 100 km or less, and approximately 80% responded that they would like to use EV. However, users do not see much of a charging infrastructure, so they feel that an EV driving range of approximately 100 km is short. Furthermore, users are also skeptical regarding the safety of lithium-ion rechargeable batteries. As a result, few people are considering EV as their next vehicle purchase, and demand remains latent.

In order to elicit EV demand in China, it is thought necessary to enhance performance such as driving ranges, construct charging infrastructure, and address user concerns regarding the safety of lithium-ion rechargeable batteries.

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■ Author ■



Junya TSUNEISHI



Yoshitaka TAKASUKA



Yoshiko TAKAHASHI

Research on Engine Control to Make Effective Use of Bioethanol-Blend Fuels

Tadashi KUROTANI*

Kohei KUZUOKA*

Hirotsugu KUDO*

ABSTRACT

Research was carried out on engine systems and control that increase both thermal efficiency and maximum torque by making effective use of the limited ethanol content in E10 gasoline, which contains ethanol. Control that separates E10 gasoline into high octane number fuel (high-concentration ethanol) and low octane number fuel (low-concentration ethanol), using the ethanol effectively under conditions that tend to cause knocking during high-load operation and using mainly gasoline during low-load operation, was applied to a high-compression-ratio engine. A fuel management control logic was created so that, under this system, the basic characteristics of ignition timing and octane number (ethanol fraction) inside the cylinder are determined in accordance with the residual percentage of the fuel, and the gear ratio is cooperatively adjusted to yield an optimal engine speed and load matched to the engine's thermal efficiency at that time. Confirmation of this result in vehicle simulation indicated that in US06 mode, known to be an operating mode that generally is frequently under high load, the system is capable of operating without running short of ethanol. It also indicated that in a low-load operating mode such as the US combined mode, the system was capable of a maximum 26.8% increase in fuel economy through the effective use of separated ethanol.

1. Introduction

The use of renewable energy as a solution for environmental issues has been growing increasingly widespread. One such solution has been the manufacture of bio-ethanol from corn, sugar cane, or other such raw materials, for mixture in small quantities with gasoline to use as fuel for motor vehicles. This method is known to yield approximately the same performance as conventional gasoline engines^{(1), (2)}. The use of E10 gasoline comprising a 10% mixture of bio-ethanol in gasoline is in widespread use in the United States. Flexible Fuel Vehicles (FFV) that support the use of fuels containing high concentrations of bio-ethanol have also been developed and are in widespread use⁽³⁾.

Ethanol is a fuel with greater anti-knocking properties than gasoline, and it is known that this characteristic can be used to obtain greater thermal efficiency by raising the compression ratio⁽⁴⁾. It has also been demonstrated that gasoline containing bio-ethanol can be used effectively and has potential for high efficiency and high torque in an engine that can separate E10 gasoline into high octane number fuel (high-concentration ethanol) and low octane number fuel (low-concentration ethanol)⁽⁵⁾. It has further been reported that fuel that has actually been separated in

a fuel separation system has properties making it usable as motor vehicle fuel, and that separation can also yield sufficient quantities of the high octane number fuel that is needed in US06 mode, which calls for frequent high-load operation⁽⁶⁾.

One issue in the application of vehicles equipped with this system for actual users is that various operating conditions need to be considered. Operating conditions differ with the way drivers use the system, and driving conditions are also variable, so there may be unevenness in the amounts of high-concentration ethanol or low-concentration ethanol consumed. When the residual amount of high-concentration ethanol becomes low, for example, the system cannot ensure that a sufficient percentage of ethanol is injected, and it may become necessary to limit torque due to the occurrence of knocking. It is therefore necessary to configure the system so that no shortage of ethanol will occur even in driving modes such as US06 mode, in which high loads are frequent. With such a configuration alone, however, it may happen instead that the reduced use of high-concentration ethanol in low-load driving modes results in a surplus of ethanol. The surplus ethanol will not be used effectively to limit knocking, and the advantages of fuel separation will not come into play. In order to make effective use of the separated high-

* Automobile R&D Center

concentration ethanol, therefore, what is needed is a control method that limits the unevenness in the residual amounts of the two kinds of fuel and that will not degrade fuel economy even when unevenness occurs.

In addition to engine control, therefore, the present research has considered a system of fuel management control that incorporates cooperative control of the transmission. The effectiveness of this approach was confirmed with simulation of vehicle operation.

2. Concept and System Overview

2.1. In-Cylinder Octane Number Control

With ordinary high-compression-ratio engines, the octane number envisioned as necessary to set ignition timing at minimum spark advance for best torque (MBT) will have the engine speed and engine torque (engine load) in the relationship shown in Fig. 1. A high octane number is required only during operation at high load, and ignition at MBT timing can be achieved during low-load operation even with a low octane number. In accordance with this characteristic, ethanol with a high octane number is effectively supplied only when demanded during high-load operation, and in this way the limited amount of ethanol in E10 gasoline can be used effectively to limit knocking. In-cylinder octane number control with the capability to vary the octane number of the fuel that is supplied can, in this way, achieve a balance between the increase in torque obtained by limiting knocking and the enhancement of efficiency obtained by using a high-compression-ratio engine.

Cooperative control of the transmission is also used with respect to the demanded vehicle driving force, enabling variation of the engine speed and engine torque along the engine's constant output curve. In other words, the engine's demand octane number is controlled by the transmission, and the octane number for supply in the cylinder is controlled in accordance with that demand octane number. This enables control of the amount of ethanol consumed.

2.2. System Overview and Fuel Injection Configuration

Figure 2 shows the system configuration for achieving

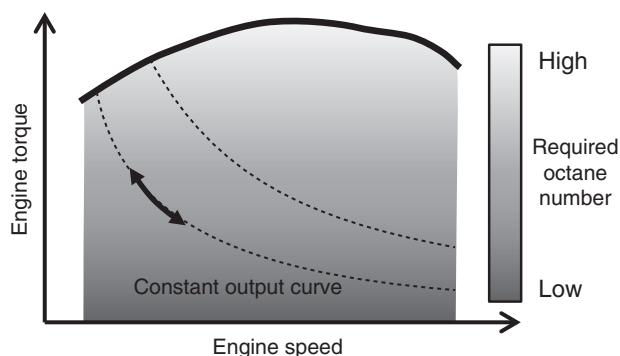


Fig. 1 Image of required octane number at MBT timing

in-cylinder octane number control. A fuel separation system was created to separate gasoline containing high-concentration ethanol from gasoline containing ethanol by means of a membrane using the principle of pervaporation (PV)⁽⁶⁾. The separated high-concentration ethanol (Fuel 1) and low-concentration ethanol (Fuel 2) are variously held in two tanks and then supplied to each engine cylinder in the desired injection ratio. In order to freely control the octane number in the cylinder in accordance with the required driving conditions, fuel injectors of two systems are used for each cylinder. This configuration allows the ethanol fraction in the cylinder to be changed instantaneously. The automatic transmission is interlocked with this system so as to control engine speed and engine torque to produce the demanded vehicle driving force.

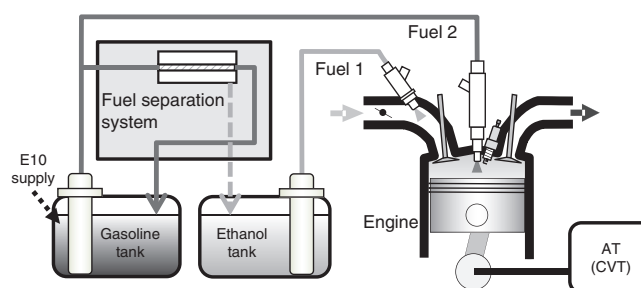


Fig. 2 System configuration

3. Experimental Device

A four-cylinder supercharged engine was used for the experiments in order to provide engine performance and to estimate the ethanol fraction required in the cylinder to obtain that performance. Table 1 shows the specifications for the two types of fuel used in the experiment, Table 2 shows the engine specifications, and Fig. 3 shows the set-up of the engine system. For the fuel, it is envisioned that ethanol and gasoline are separated from E10 gasoline. The system specification calls for the regular gasoline and ethanol to be held separately in independent tanks from which they are then supplied to each engine cylinder in the injection ratio desired. Each cylinder is configured with two fuel injection systems so that the ratio of the two types of fuel can be freely controlled to match driving conditions and the ratio of the total amount of fuel to the amount of intake air will be stoichiometric. Two injector patterns were verified. One was a basic configuration in which ethanol is supplied to the port injector and gasoline to the direct injector. The other, for comparison, was a configuration in which gasoline is supplied to the port injector and ethanol to the direct injector. The engine was equipped with a turbocharger, and the timing for opening and closing the intake valves could be controlled by variable valve timing and lift electronic control (VTEC) and continuous variable valve timing control (VTC). A specification was

Table 1 Fuel specifications

Fuels	Gasoline	Ethanol
Density [g/cm ³]	0.72	0.79
Boiling point [deg C]	-	78
Calorific value [J/g]	43.3	26.8
Octane number	91	115

Table 2 Engine specifications

Number of cylinders	4
Displacement [cm ³]	1755
Compression ratio	12.5
Combustion chamber	Pent roof
Valve control systems	VTEC, VTC
Fuel systems of each cylinder	Port injector Direct injector

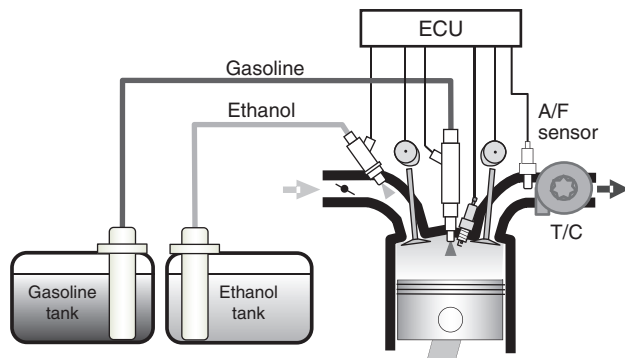


Fig. 3 Experimental engine system set-up

adopted that enables the timing for opening and closing the intake valves to be changed according to driving conditions, changing the intake air volume and the effective compression ratio and enabling supercharged Atkinson cycle operation.

4. Experimental Results

4.1. Comparison of Fuel Injection Configurations

Figure 4 shows the results from comparison of two injection system patterns, Injection System 1, in which ethanol is supplied to the port injector and gasoline to the direct injector, and Injection System 2, in which gasoline is supplied to the port injector and ethanol to the direct injector. The ethanol fraction required to obtain the same engine load (indicated mean effective pressure: IMEP) and the indicated thermal efficiency with Injection System 1 could be reduced to approximately 10 points lower than with Injection System 2. This is thought to be mainly because the influence of vaporization heat from gasoline injected directly into the cylinder was able to limit knocking even with small amounts of ethanol. Given the limited amount of usable ethanol made available in this system by separating

ethanol from E10 gasoline, it was considered advantageous to use smaller amounts of ethanol, and Injection System 1 was therefore adopted as the basic configuration.

4.2. Influence of the Ethanol Fraction

Figure 5 shows the influence on the basic configuration when the ethanol fraction in the cylinder is varied. The horizontal axis shows the IMEP, in relation to which are shown the ethanol fraction, the point at which the mass fraction burned (MFB) is 50% as an indicator of combustion timing, the indicated thermal efficiency, and the indicated specific fuel consumption (ISFC). In supercharged Atkinson cycle operation fueled only by gasoline, retarded ignition timing becomes necessary due to knocking when the IMEP exceeds 900 kPa, and the indicated thermal efficiency diminishes. The IMEP then reached the maximum load of 1600 kPa, and the engine could not operate at that high a load or higher. Under Case 1 conditions, by contrast, the use of ethanol in a high concentration enabled ignition at MBT timing up to high load (IMEP of 1800 kPa), and the indicated thermal efficiency increased. Furthermore, taking the influence of ethanol's lower calorific value into consideration, trial operation was conducted using an ethanol fraction adjusted to a lower level so that the engine would have about the same ISFC as during MBT operation under Case 2 conditions. The result was that the required ethanol fraction could be kept down to a level approximately 17 points lower. It was confirmed in this way that stoichiometric operation could be achieved by limiting the amount of ethanol used while also operating under high load (IMEP of 2100 kPa).

Comparison of Case 1 and Case 2 shows that when the driving mode is assumed to be the same because the ISFC is the same, the fuel consumption per mass is equal. Looking now at the amount of ethanol consumed from driving

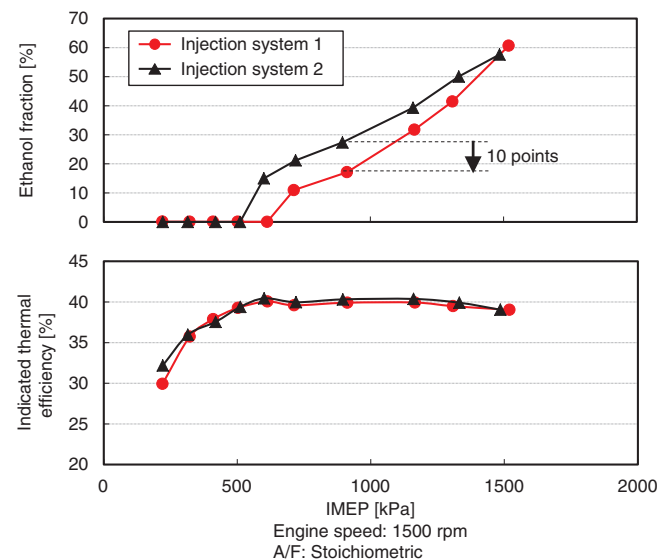


Fig. 4 Comparison of injection system

one time, Case 2 enables driving with a smaller amount of ethanol. By applying optimal ethanol injection control in this way so that thermal efficiency is reduced slightly but fuel economy is not reduced, the effects of ethanol (increased fuel economy and acceleration performance) can be provided with the limited amount of usable ethanol in E10 gasoline even to users who engage in relatively greater degrees of high-load operation. Taking Case 2 as the basic configuration, the amount of ethanol consumed can be adjusted within the margins of engine control by switching over to Case 1 when there is a surplus of ethanol or by other such means.

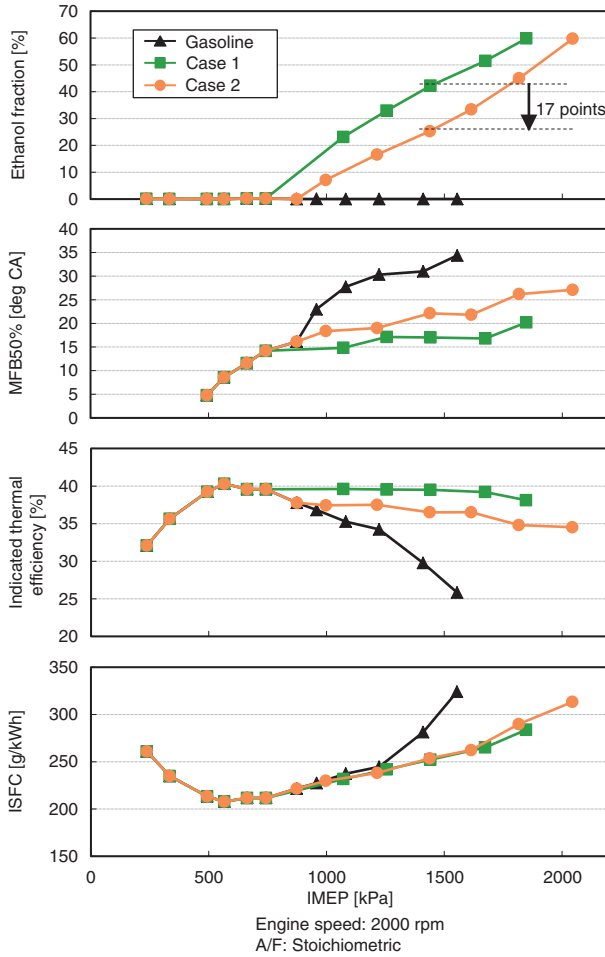


Fig. 5 Effect of altering ethanol fraction

5. Fuel Management Control Verification

5.1. Engine Speed and Load Control According to Residual Amount of Fuel

In light of the properties obtained through the experiments in chapter 4, a fuel management control logic that achieves effective use of ethanol in accordance with the residual amount of fuel was created. Figure 6 shows the concept of control by varying the engine operating points

(engine speed and load) according to the ethanol retention rate. The configuration that yields MBT operation when the engine speed is 1500 rpm is shown as Case 1. Similarly, the configuration that yields ISFC at about the same level as in MBT operation is shown as Case 2. Envisioning that the residual quantity of ethanol is small, the configuration that has the engine speed at 2000 rpm to conserve on ethanol use is shown as Case 3. For the purpose of comparison, the configuration that keeps the engine speed at 1500 rpm to conserve on ethanol use is shown as Case 4. Taking, for example, the case of an engine output of 30 kW, when the engine speed is 1500 rpm, the IMEP is approximately 1400 kPa, and when the engine speed is 2000 rpm, the IMEP is approximately 1100 kPa (envisioning the engine specifications in Table 2). When ethanol is in surplus, the engine operates at Point A of Case 1 where a high indicated thermal efficiency can be obtained. When the residual quantity of ethanol begins to diminish, operation at Point B of Case 2 can reduce the ethanol fraction to a level approximately 13 points lower than at Point A. The indicated thermal efficiency is lowered approximately two points, but the ISFC remains virtually unchanged. When the residual quantity of ethanol is further diminished, the configuration that limits the ethanol fraction is combined with cooperative control with the transmission for operation at Point C of Case 3. At this time, the indicated thermal efficiency drops approximately three points lower than

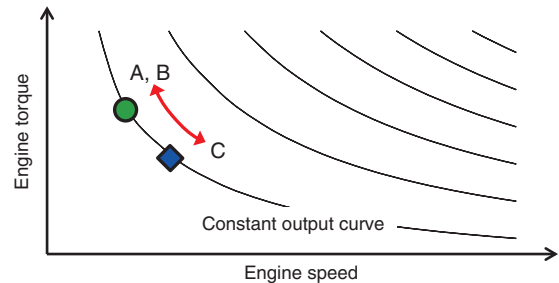
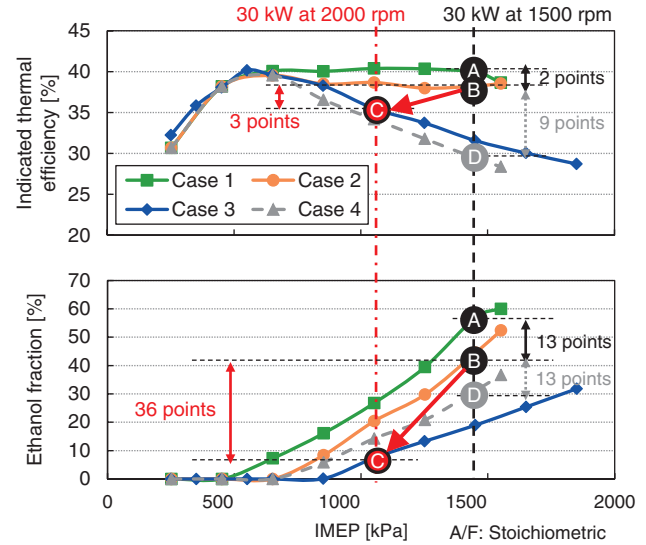


Fig. 6 Fuel management concept

at Point B, but the ethanol fraction can be held down by approximately 36 points. Operation at Point C can limit the amount of decrease in the indicated thermal efficiency by comparison with operation at Point D of Case 4, and can hold down the ethanol fraction by approximately 23 points. This combined use not only of engine control according to the residual amount of fuel, but also of transmission control, can limit the reduction in fuel economy while limiting the amount of ethanol consumed. In the case of a continuously variable transmission (CVT) that demands a constant vehicle driving force, the engine speed and engine torque trace a virtually constant output curve from Points A and B to Point C.

5.2. Fuel Management Control Logic

Figure 7 shows an overview of fuel management control and the control flow. First, the system calculates the vehicle driving force required due to driver operation. Next, the fuel concentration estimator calculates the ethanol concentration in the two types of fuel. The retained fuel amount estimator then uses equation (1) with those concentration values and the amounts of the two kinds of fuel remaining in the vehicle's tanks to calculate the percentage of ethanol present in the vehicle's fuel.

$$ER = (EC_1 \times RQ_1 + EC_2 \times RQ_2) / (RQ_1 + RQ_2) \quad (1)$$

ER: Total ethanol residual rate [%]

EC₁: Ethanol concentration of fuel 1 [%]

EC₂: Ethanol concentration of fuel 2 [%]

RQ₁: Ethanol residual quantity of fuel 1 [L]

RQ₂: Ethanol residual quantity of fuel 2 [L]

In the distributor, the optimal case is selected from the instances given, including those shown in Fig. 6.

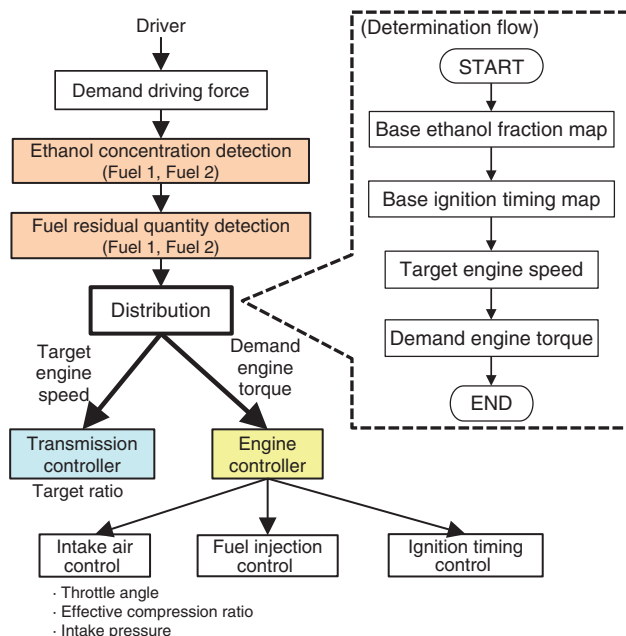


Fig. 7 Overview of fuel management control logic

Specifically, it determines the in-cylinder octane number demanded according to the ethanol retention rate, or in other words, the target value for the in-cylinder ethanol fraction (base map), and it decides the base ignition timing characteristic to match that value. The system then decides the transmission map (shifting schedule) so that the engine speed and load will produce optimal fuel consumption matched to the changing thermal efficiency characteristic. The driving force demanded from the driver is distributed as a target engine speed to the transmission and demand torque to the engine. In the transmission, the gear ratio is controlled so that the engine speed matches the target engine speed (target ratio, target gear). In order to make the engine generate the demand torque, the system controls the intake air volume as well as the fuel injection volume and ignition timing.

Figure 8 shows an overview of engine and transmission control. Following the base map previously determined from the present engine speed, intake air volume, and ethanol retention rate, the system decides the ethanol fraction and base fuel flow rate. The air-fuel ratio feedback coefficient from the exhaust gas together with knocking detection value, ethanol concentration sensor value, and other such information are used as the basis for calculating the estimated ethanol concentration, the fuel adhesion quantity, and other environmental correction values to factor in to determination of the injection volume and injection timing for the port injector and the direct injector. The base ignition timing is similarly calculated from the base map, corrections are factored in to the estimated octane number value estimated from knocking and other characteristics, and the ignition timing is decided. The transmission controller uses the demand driving force, the target engine speed, and the vehicle speed to decide the target ratio (target gear).

5.3. Fuel Economy and Ethanol Consumption Rate Verification Results

The fuel economy and ethanol consumption rate were verified by vehicle simulation. The engine performance obtained above was used, taking a mass-production vehicle equipped with a naturally aspirated engine (NA) as the base, and envisioning it equipped with a fuel separation system and supercharged engine. The vehicle conditions used in the simulation are shown in Table 3. In order to make effective use of this technology, which yields high efficiency characteristics up to high-load conditions, the engine was envisioned as a downsized supercharged engine with the engine displacement reduced. Case 1 was defined as using the ethanol fraction of Condition A, operating at MBT when there is a surplus of ethanol. Case 2 was defined as using the ethanol fraction of Condition B, operating with about the same ISFC as at MBT. Case 3 was defined as Condition C, using the injection ratio map configured to conserve ethanol when the ethanol retention rate has become low, and adding cooperative control with the transmission.

Envisioning that the available ethanol is separated from

E10 gasoline, it becomes necessary to limit the quantity of ethanol used to no more than 10% of the total fuel quantity. Here it is assumed that all the ethanol in E10 gasoline has been extracted by separation, and when all of this has been used, the ethanol consumption rate is defined as 100%. The calculations for this simulation use US06 mode as the driving mode with a high frequency of high-load operation, and US combined mode as the low-load fuel consumption reference mode. Figure 9 shows the characteristic of net

Table 3 Vehicle simulation conditions

	Ethanol fraction	Displacement [cm ³]	Weight [kg]	Compression ratio	Fuels
Mass-production vehicle 5AT (NA, Base)	—	1799	1361	10.5	E10
Case 1	A	1496	1423	12.5	Separated
Case 2	B				
Case 3	C				

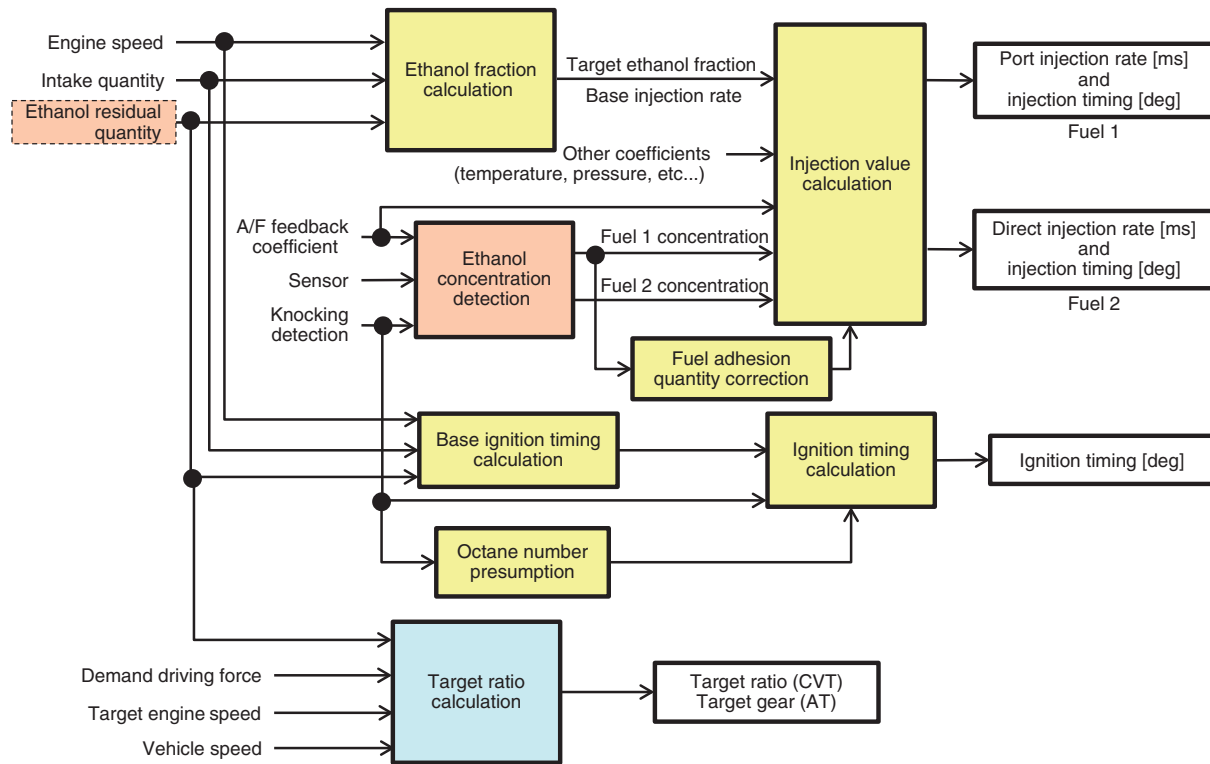


Fig. 8 Overview of engine and transmission control

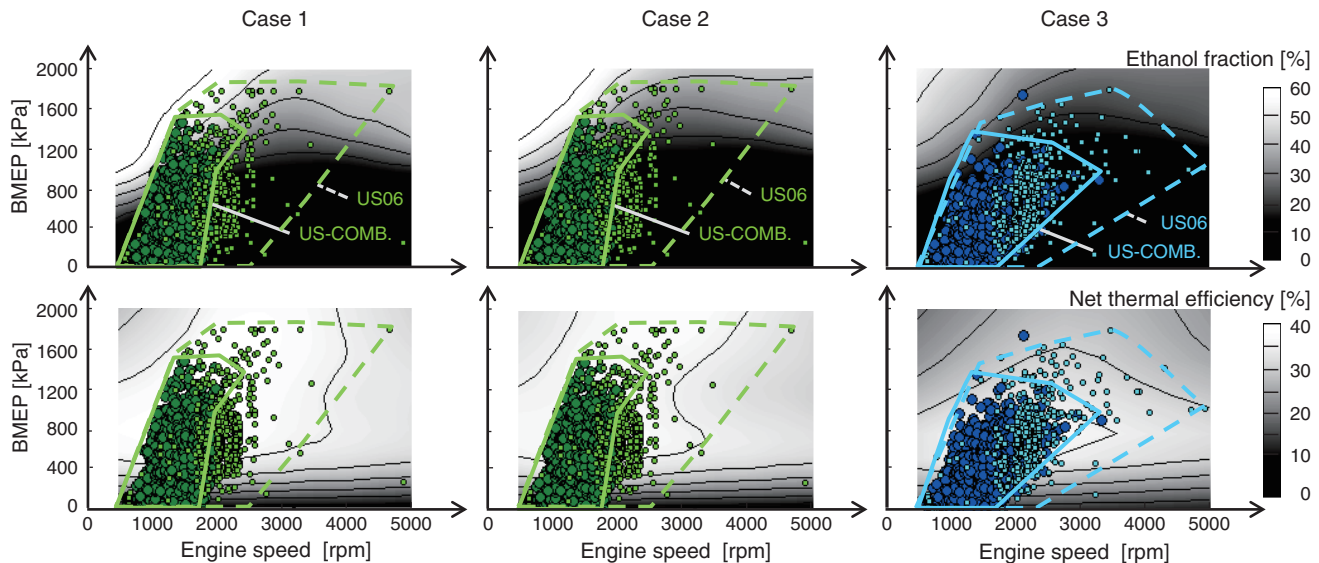


Fig. 9 Operating range with ethanol fraction and thermal efficiency

thermal efficiency when the ethanol fraction is taken as the variable, together with the driving plots (engine speed and brake mean effective pressure: BMEP) for the various driving modes in the simulation. Case 1 and Case 2 have the ethanol fraction set at a high level under high load, and it is apparent that the engine is operating at low speed and high load due to the shifting schedule matched to the high thermal efficiency found up to the high-load level. In Case 3, it is apparent how the operating range has shifted toward high speed and low load to match the change in the thermal efficiency characteristic.

Figure 10 shows the rate of increase in fuel economy relative to the base performance obtained in simulation together with calculated results for the ethanol consumption rate. In the US combined mode of Case 1, the rate of increase in fuel economy is the greatest, at approximately 26.8%, and the ethanol consumption rate comes out to approximately 98%. In the US06 mode, however, the ethanol consumption rate comes out to approximately 196%. This indicates that although there was no difficulty when the ethanol was in surplus, when driving conditions that involve repeatedly driving under frequent heavy load are envisioned, the ethanol retention rate begins to decline. In the configuration of Case 2, the increase in fuel economy in US combined mode is approximately 25.3% and that of the ethanol consumption rate is approximately 53%, showing that the quantity of ethanol consumed has successfully been limited by comparison with Case 1. Although the ethanol consumption rate was successfully limited to approximately 131% in US06 mode, however, when driving conditions that involve repeatedly driving under frequent heavy load are envisioned, the ethanol retention rate begins to decline. In the Case 3 configuration, by contrast, the ethanol consumption rate is approximately 65% in US06 mode, showing even greater limitation than in Case 2.

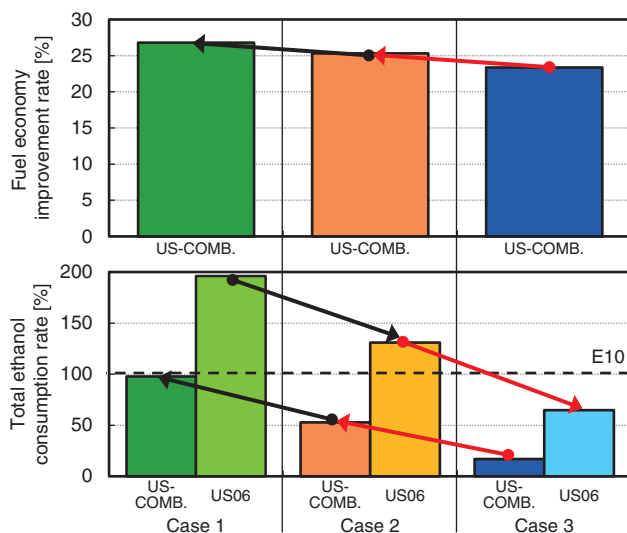


Fig. 10 Fuel economy improvement rate and ethanol consumption rate

The increase in fuel economy in US combined mode is approximately 23.4%, indicating that the decline in fuel economy has been successfully limited. The ethanol consumption rate then comes out to approximately 17%. When low-load operation continues, a surplus of ethanol occurs, at which point shifting to Case 2 and then to Case 1 will enable use of the surplus ethanol to increase fuel economy. Actual drivers will be able to respond to varying driving conditions, such as driving situation, residual amount of fuel and other such factors, by switching control configurations continuously from Case 1 to Case 3, in accordance with driver demand.

Given the above results, this system equipped with the fuel separation system and supercharged Atkinson cycle engine can be considered a technology capable of increasing fuel economy by making effective use of the ethanol separated from E10 gasoline in scenarios envisioning actual users. The implementation of fuel management control according to the ethanol retention rate enables driving without shortage of ethanol even under conditions when frequent high-load operation is required. The system is also able to increase fuel economy by making effective use of surplus ethanol under ordinary driving conditions.

6. Conclusion

The following findings were obtained from the results of verification of a supercharged Atkinson cycle engine combined with transmission control in order to make effective use of the separated fuels in engine systems that separate E10 gasoline containing ethanol into high-concentration ethanol and low-concentration ethanol:

- (1) Configuring the fuel injectors so that the port injector supplies ethanol and the direct injector supplies gasoline enables limitation of the quantity of ethanol used so that the limited quantity of ethanol contained in E10 gasoline can be used effectively.
- (2) Envisioning the separation of high concentration ethanol from E10 gasoline, the use of this fuel to effectively limit knocking enables high-load operation. The point is not just to inject large quantities of ethanol under every driving condition to increase the thermal efficiency, but rather to inject the optimal amount considering the indicated specific fuel consumption so that the reduction in fuel economy can be limited while limiting the quantity of ethanol used.
- (3) The fuel economy enhancement effect of this system, which makes optimal use of the two fuels obtained by fuel separation of E10 gasoline, was verified by vehicle simulation. The results indicate that, when applied in a downsized supercharged gasoline engine, this technology can increase the vehicle's fuel economy by up to a maximum of 26.8%.
- (4) When combined with fuel management control logic that implements cooperative control of the transmission according to the ethanol retention rate, this technology

enables driving without running short of ethanol even in the frequent high-load operation of US06 mode. Under low-load driving conditions such as found in US combined mode, the technology is also able to make effective use of surplus ethanol to increase fuel economy.

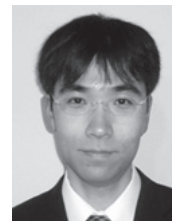
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■ Author ■



Tadashi KUROTANI



Kohei KUZUOKA



Hirotsugu KUDO

Optimization of Driving Force Distribution Control in All-Wheel Drive Based on Wheel Rotation Speed Difference Between Front and Rear

Shoichi OHYAGI*

Takayuki SEKI*

ABSTRACT

Electronically controlled all-wheel drive (AWD) systems that are built on a front-wheel drive base can control spinning of the front wheels by increasing the amount of driving force distributed to the rear wheels according to the speed difference generated between the front and rear wheels. On the other hand, the turning characteristics of the wheels change with the amount of driving force distributed. When the magnitude of this change is large, vehicle controllability may be degraded. Driving force distribution therefore needs to take into account the effects during turns that involve a greater wheel rotation speed difference between front and rear.

A spin suppression control system was therefore created that performs theoretical calculations of front and rear wheel rotation speed differences that occur due to turns and tire radius variations, and that takes those values as the target values for the system. The factors that cause front and rear wheel rotation speed differences are divided between front wheel spin and all others, and it was confirmed that this would enable convergence only on front and rear wheel rotation speed differences that are due to front wheel spin.

1. Introduction

The magnitude of the cornering force relative to the slip angle of the tire (cornering power) is known to vary according to the driving force on the tire in the longitudinal direction⁽¹⁾. AWD systems with electronic control and a clutch for driving force distribution can determine the amount of driving force distributed to sub-driving wheels by implementing electronic control of the clutch fastening force. Adjustment of the amount of driving force distributed in order to control cornering power and control the vehicle's turning state is a widely used practice⁽²⁾⁻⁽⁵⁾.

This kind of AWD system may also implement control to suppress the rotation speed difference⁽⁶⁾⁻⁽⁸⁾ by increasing the clutch fastening force according to the magnitude of the rotation speed difference between the front and rear wheels. This is control to reduce the slip ratio of the main driving wheel when the driving force applied to the main driving wheel exceeds the friction force limit of the tire, causing the wheel to spin, by using the sub-driving wheel to take up the surplus driving force. Such control can be expected to have the effects of mitigating the reduction in driving force due to spinning, the energy loss, and the load on the machine.

A rotation speed difference between front and rear wheels may also occur for reasons other than main driving wheel spin, such as differences in wheel direction or path. If spin suppression control does not take the influence of these factors into account, the system may estimate the amount of wheel spin to be greater, and a greater amount of driving force will be distributed than is needed to suppress spin. This distribution of an extra amount of driving force will change the turning characteristics, and the change may in some cases interfere with the turning state control by means of driving force distribution mentioned above. For this reason, spin suppression control needs to correct for turns. This used to be done by applying a correction value to the driving force according to the ratio of lateral acceleration⁽⁶⁾ or the speed difference between left and right wheels. In neither case, however, was correction applied on the basis of the calculated value for the rotation speed difference between front and rear wheels that occurs when turning.

This paper will discuss the causal factors in wheel rotation speed differences between front and rear wheels and will present a method of spin suppression control that does not interfere with turning state control by means of driving force distribution.

* Automobile R&D Center

2. Basic Theory of Spin Suppression

This paper will describe an AWD system of the kind shown in Fig. 1, in which the main driving wheel is a front wheel, the sub-driving wheel is a rear wheel, and there are differential gears that distribute the driving force equally between the left and right wheels, both front and rear, while also absorbing the rotation speed difference between left and right. When the rear wheels start turning before the front wheels in this kind of system, the torque transmission goes from the rear wheels to the front wheels. The rear wheels do not have their own source of driving force, so the torque transmission causes convergence of the rotation speed difference. Consequently, spin suppression will be considered only for the case of spin in the front wheels.

Spin suppression is implemented by increasing the amount of driving force distributed to the rear wheels in accordance with the amount of front wheel spin. The amount of spin can be determined by the ratio of the wheel rotation speed to the rotating directional component of the wheel center velocity vector. While the vehicle is turning, however, the wheel direction and the direction and magnitude of the wheel center velocity vector differ for each wheel. Therefore the amount of spin cannot be determined from the wheel rotation speed alone, and so, in order to determine the amount of front wheel spin from the rotation speed difference between front and rear wheels, the influence of the change in rotation speed that occurs while the vehicle is turning must be eliminated.

A rotation speed difference also occurs when the tire dynamic loaded radius differs between the front and rear wheels. This rotation speed difference is also a value that is unrelated to the amount of spin, and so, with the exception of cases of extreme rotation speed difference that result in excessive mechanical load, the influence of the tire dynamic loaded radius error must also be eliminated from the rotation speed difference between front and rear wheels.

This chapter will describe the conceptual approach

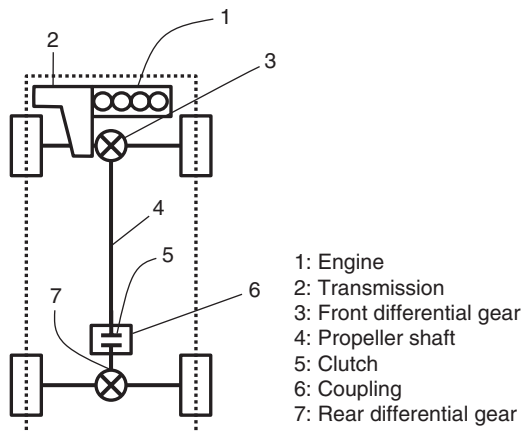


Fig. 1 Structure of AWD system

to correction for tire radius variation and the influence of vehicle turns on spin suppression control based on rotation speed difference between front and rear wheels.

V : Vehicle body speed

V_{ij} : Wheel center velocity

The character i indicates the front wheel (f) or the rear wheel (r).

The character j indicates the left wheel (l) or the right wheel (r).

R_{ij} : Tire dynamic loaded radius

ω_{ij} : Wheel rotation speed

s_{ij} : Longitudinal wheel slip ratio

α_x : Vehicle body longitudinal acceleration

δ : Steering angle

γ : Yaw rate

β : Vehicle body slip angle

l : Wheelbase length

l_f : Length of a perpendicular line from the vehicle center of gravity to the front wheel axis on a horizontal plane

l_r : Length of a perpendicular line from the vehicle center of gravity to the rear wheel axis on a horizontal plane

d_f : Front wheel tread

d_r : Rear wheel tread

h : Vehicle center of gravity height above ground

ω_f : Front wheel average rotation speed

ω_{in} : Input rotation speed of longitudinal driving force distribution clutch

ω_{out} : Output rotation speed of longitudinal driving force distribution clutch

A : Ratio of clutch input rotation speed to front wheel average rotation speed

B : Ratio of clutch output rotation speed to rear wheel average rotation speed

I_f : Inertia moment of front wheel rotating system

T_{att} : Driving torque generated by source of driving force (drive shaft equivalent)

T_r : Rear wheel driving torque

F_f : Tire reaction force in longitudinal direction on front wheel

g : Acceleration of gravity

2.1. Theoretical Value of the Rotation Speed Difference Between Front and Rear Wheels

This section will use the two-dimensional vehicle model shown in Fig. 2 for the sake of simplicity. The rotating directional components of the wheel center velocity for each wheel while the vehicle is turning are shown by Eqs. (1), (2), (3), and (4).

$$V_{fl} = V \cos(\delta - \beta) + \gamma(l_f \sin \delta + \frac{d_f}{2} \cos \delta) \quad (1)$$

$$V_{fr} = V \cos(\delta - \beta) + \gamma(l_f \sin \delta - \frac{d_f}{2} \cos \delta) \quad (2)$$

$$V_{rl} = V \cos \beta + \gamma \frac{d_r}{2} \quad (3)$$

$$V_{rr} = V \cos \beta - \gamma \frac{d_r}{2} \quad (4)$$

When driving force is being generated by the wheels, slipping occurs in a longitudinal direction. If the slip ratio under driving force is defined by Eq. (5), then the wheel rotation speed is shown by Eq. (6).

$$s_{ij} = \frac{V_{ij} - R_{ij} \omega_{ij}}{R_{ij} \omega_{ij}} \quad (5)$$

$$\omega_{ij} = \frac{V_{ij}}{(1 + s_{ij}) R_{ij}} \quad (6)$$

When converted into the rotation speeds (ω_{in} and ω_{out}) of input and output at the driving force distribution clutch, the rotation speeds of the left and right wheels are averaged out and shown by Eqs. (7) and (8), respectively.

$$\omega_{in} = A \frac{\omega_{rl} + \omega_{rr}}{2} \quad (7)$$

$$\omega_{out} = B \frac{\omega_{rl} + \omega_{rr}}{2} \quad (8)$$

The clutch slippage (hereafter represented as ΔN) is shown by Eq. (9).

$$\begin{aligned} \Delta N &= \omega_{in} - \omega_{out} \\ &= A \{ x_{fa} [V \cos (\delta - \beta) + \gamma l_f \sin \delta] + x_{fd} \gamma \frac{d_f}{2} \cos \delta \} \\ &\quad - B (x_{ra} V \cos \beta + x_{rd} \gamma \frac{d_r}{2}) \end{aligned} \quad (9)$$

$$x_{fa} = \frac{1}{2} \left[\frac{1}{(1 + s_{fl}) R_{fl}} + \frac{1}{(1 + s_{fr}) R_{fr}} \right] \quad (10)$$

$$x_{fd} = \frac{1}{2} \left[\frac{1}{(1 + s_{fl}) R_{fl}} - \frac{1}{(1 + s_{fr}) R_{fr}} \right] \quad (11)$$

$$x_{ra} = \frac{1}{2} \left[\frac{1}{(1 + s_{rl}) R_{rl}} + \frac{1}{(1 + s_{rr}) R_{rr}} \right] \quad (12)$$

$$x_{rd} = \frac{1}{2} \left[\frac{1}{(1 + s_{rl}) R_{rl}} - \frac{1}{(1 + s_{rr}) R_{rr}} \right] \quad (13)$$

2.2. Rotation Speed Difference Between Front and Rear Wheels that Occurs During Turns

If $s_{ij} = 0$ and $R_{fl} = R_{fr} = R_{rl} = R_{rr} = R$ in Eq. (9), the result will be Eq. (14).

$$\Delta N_{id} = \frac{A [V \cos (\delta - \beta) + \gamma l_f \sin \delta] - BV \cos \beta}{R} \quad (14)$$

Equation (14) does not include variation in the tire dynamic loaded radius, and it represents ΔN that does not involve slip in a longitudinal direction, so this value is a reference value.

2.3. Variation in Tire Dynamic Loaded Radius

When the tire dynamic loaded radius is different for each wheel, ΔN will always occur even if the vehicle is driving with every tire gripping. The following will describe the ΔN (ΔN_a) that is generated by tire dynamic loaded radius variation in a state of no slipping in a longitudinal direction.

Simplify Eq. (9) so that the slip ratio is zero for each wheel. If correction of the tire dynamic loaded radius variation is kept within the range of $\pm 5\%$, X_{fd} and X_{rd} are sufficiently small by comparison with X_{fa} and X_{ra} . Ignoring the terms X_{fd} and X_{rd} in Eq. (9), this can be shown by Eq. (15).

$$\Delta N_a = \frac{AX_{fa} [V \cos (\delta - \beta) + \gamma l_f \sin \delta] - BX_{ra} V \cos \beta}{R} \quad (15)$$

$$X_{fa} = \frac{1}{2} \left(\frac{R}{R_{fl}} + \frac{R}{R_{fr}} \right) \quad (16)$$

$$X_{ra} = \frac{1}{2} \left(\frac{R}{R_{rl}} + \frac{R}{R_{rr}} \right) \quad (17)$$

Equations (16) and (17) represent the average of the ratio of the reference tire radius to the radius of the wheels for the left and right front wheels and the left and right rear wheels, respectively. Here, consider the maximum value of ΔN_a .

$$\max (R_{fl}, R_{fr}, R_{rl}, R_{rr}) = R \quad (18)$$

In this case, it is clearly when $R_{rl} = R_{rr} = R$ that ΔN_a reaches its maximum (ΔN_{amx}).

$$R_{rl} = R_{rr} = R \quad (19)$$

In other words, this is under Eq. (20).

$$\Delta N_{amx} = \frac{AX_{fa} [V \cos (\delta - \beta) + \gamma l_f \sin \delta] - BV \cos \beta}{R} \quad (20)$$

Comparing Eq. (20) and Eq. (14), it is apparent that X_{fa} appears as a ratio of the coefficient A term (the rotation speed of the front wheel).

If the difference here between the actual ΔN (ΔN_{re}) and ΔN_{amx} is taken as ΔN (ΔN_{sl}), which is generated by the front and rear wheel slip ratio difference, then it follows that:

$$\Delta N_{re} = \Delta N_{amx} + \Delta N_{sl} \quad (21)$$

Considering the deviations of ΔN_{re} and ΔN_{id} (if there is no longitudinal slip and no variation in the tire dynamic loaded radius, then these will be zero), the result will be Eq. (22).

$$\begin{aligned} \Delta N_{re} - \Delta N_{id} &= (\Delta N_{amx} - \Delta N_{id}) + \Delta N_{sl} \\ &= \frac{A}{R} (X_{fa} - 1) [V \cos (\delta - \beta) + \gamma l_f \sin \delta] + \Delta N_{sl} \end{aligned} \quad (22)$$

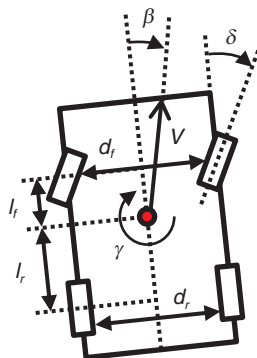


Fig. 2 Vehicle model

Transforming Eq. (22), the result will be Eq. (23).

$$\frac{R(\Delta N_{re} - \Delta N_{id})}{A[V \cos(\delta - \beta) + \gamma l_f \sin \delta]} + 1 = X_{fa} + X_{st} \quad (23)$$

$$X_{st} = \frac{R\Delta N_{st}}{A[V \cos(\delta - \beta) + \gamma l_f \sin \delta]} \quad (24)$$

If the time change in the dynamic loaded radius of each wheel is ignored in Eq. (16), then X_{fa} is a constant. In this case, the left side of Eq. (23) can be divided into the constant value X_{fa} and the variable term X_{st} .

3. AWD Driving Force Distribution Control that Includes Spin Suppression

This chapter will describe specific methods for implementing spin suppression control. Figure 3 shows a summary block diagram of this control. The amount of driving force distributed to the rear wheels is the sum determined by the amounts of driving force distributed by the base distribution controller and the rotation speed difference controller, which implements spin suppression. The amount of driving force distributed is determined predominantly by the base distribution controller, and the system is configured so that the rotation speed difference controller operates as a compensator when the driving force distribution to the rear wheels is insufficient, such as when the front wheels are tending to slip.

3.1. Base Distribution Controller

The maximum friction force of a tire is determined by the ground load and the coefficient of friction between the road surface and the tire. When the ratio of distribution of driving force between the front and rear wheels is decided by the ground load ratio, tire use can be well balanced, which makes this a useful method in terms of driving force performance and turn stability. When load shift due to longitudinal acceleration is taken into account, the ground load ratios u_f and u_r of the front wheels and rear wheels to the vehicle weight are expressed by Eqs. (25) and (26).

$$u_f = \frac{l_r}{l} - \frac{\alpha_x}{g} \cdot \frac{h}{l} \quad (25)$$

$$u_r = \frac{l_f}{l} + \frac{\alpha_x}{g} \cdot \frac{h}{l} \quad (26)$$

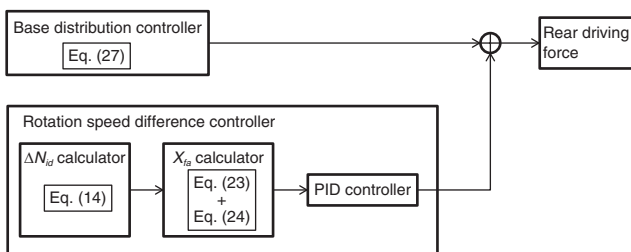


Fig. 3 Block diagram of control

The base distribution (T_b) is the value resulting from multiplication of the rear wheel ground load ratio by the estimated value of driving force generated by the source of driving force, or in other words, this is:

$$T_b = u_r T_{all} \quad (27)$$

3.2. Rotation Speed Difference Controller

The rotation speed difference control suppresses front wheel spin by increasing the distribution to the rear wheels when ΔN exceeds a certain target value. Providing an appropriate target value prevents spin suppression from disturbing the base distribution ratio in scenarios where it is unnecessary. By setting the target value to the maximum value envisioned when there is no spin, the system is configured to not do anything when ΔN_{re} is lower than the target value, and this configuration limits the amount of control implemented by the rotation speed difference control.

The control is configured with (1) a turn correction ΔN calculator that calculates ΔN when there is no longitudinal slip, (2) a ΔN variation correction unit that increases system robustness with regard to tire dynamic loaded radius variation, and (3) a ΔN servo-controller that determines the amount of increase in the distribution amount due to deviation in the actual ΔN and the derived reference ΔN .

3.2.1. Turn correction ΔN calculator

This unit calculates ω_{in} , ω_{out} and ΔN_{id} in cases where, given the driving state of the vehicle, no slip is envisioned to be occurring in a longitudinal direction. These figures are used for calculations in the ΔN variation correction unit, to be discussed later.

Note here that Eq. (14) includes the vehicle body slip angle. Cost-performance considerations, however, make it impractical to install an actual vehicle slip angle sensor to measure this quantity. For this system, therefore, the slip angle is estimated using a vehicle dynamics model⁽⁹⁾.

3.2.2. ΔN variation correction unit

This unit determines the correction factor X_{fa} for the ΔN that occurs due to the tire dynamic loaded radius variation, and calculates the reference ΔN . It suffices to choose a maximum value for the envisioned target value, and Eq. (20) can be used for the purpose. A steady-state value is derived by using a low-pass filter on the values calculated on the left side of Eq. (23).

The smaller the time constant in this low-pass filter, the shorter the time that the correction factor X_{fa} will remain a constant value. If it is too small, however, the calculated value also follows the time variation, and therefore diverges from the true correction factor X_{fa} .

There are some cases, however, in which robustness can be further heightened by causing this time variation to be followed to some extent. The left side of Eq. (23) may generate calculation errors relative to true values from sensor errors, errors in estimation from vehicle models, or

disturbances caused by uneven road surfaces. These can be considered values that vary over time. Since an increase in ΔN due to front wheel spin has a relatively fast response, it can be considered acceptable to follow the calculated value for time variation in a time constant that is sufficiently slower than an increase in ΔN that is assumed to be due to spin. However, continued spinning of the front wheels will end up generating a constant ΔN , so that the correction factor X_{fa} will need to be kept within the envisioned range of error.

The correction factor X_{fa} obtained is substituted in Eq. (20). This value is ΔN when there is no slip in a longitudinal direction, but when driven, the front-rear wheel slip ratio difference results in ΔN (ΔN_{sl}). This is handled by use of the allowable slip ratio difference ΔN (ΔN_{os}) as an offset. In other words, reference ΔN (ΔN_t) is expressed by Eq. (28).

$$\Delta N_t = \Delta N_{amx} + \Delta N_{os} \quad (28)$$

3.2.3. ΔN servo-controller

Proportional integral differential (PID) control is implemented on the basis of the deviations between ΔN_t obtained by the ΔN variation correction unit and ΔN_{re} . As described above, as a rule no control is implemented when ΔN_{re} is low relative to the target value, but the system is configured so that control will not become unstable.

PID was adopted for this ΔN servo-control in order to enhance convergence with regard to the steady-state deviation. In a state in which the deviation occurs continuously, however, the amount of control implemented will end up increasing without being limited by the integral term. When the vehicle is making turns with a short turning radius, constraints on the front and rear wheels may result in generation of braking force (the tight corner braking phenomenon).

In ΔN servo-control, therefore, the total rear wheel driving force distribution is as expressed in Eq. (29). Control is applied as a value that does not exceed the driving force generated by the source of driving force.

$$T_r \leq T_{all} \quad (29)$$

The following describes the principle of tight corner braking.

The amount of change in the front wheel rotation speed ω_f is expressed by Eq. (30).

$$I_f \frac{d\omega_f}{dt} = T_{all} - T_r - RF_f \quad (30)$$

Looking then at a steady state in which there is no change in the front wheel rotation speed, the left side of Eq. (30) becomes zero, and this can be transformed to Eq. (31).

$$T_r = T_{all} - RF_f \quad (31)$$

Braking force is generated when $F_f < 0$, but if $T_r \leq T_{all}$, then F_f will necessarily be positive. In other words, steady-state braking from constraints on the front and rear wheels can be prevented.

4. Confirmation of the Effectiveness of Spin Suppression Control

The controls described in chapter 3 were installed in a vehicle with electronically controlled AWD on a front-wheel drive base. The controls were then evaluated in terms of the validity of the reference rotation speed difference, the effectiveness of reference value correction, and the effectiveness of ΔN servo-control spin suppression.

4.1. Confirmation of the Validity of Reference Rotation Speed Difference

Figure 4 shows a comparison of the actual rotation speed difference ΔN_{re} when no driving force is applied (in other words, when the slip ratio of each wheel is virtually zero) with ΔN_{id} .

When the transient change component of ΔN_{re} is excluded, its waveform largely matches the ΔN_{id} waveform. This indicates that the calculated results for reference rotation speed difference are valid.

4.2. Confirmation of the Effectiveness of Reference Rotation Speed Difference Correction

By constantly monitoring the deviation between the reference rotation speed difference ΔN_t and the actual rotation speed difference ΔN_{re} , reference rotation speed difference correction promptly corrects for the ΔN that occurs due to such factors as tire dynamic loaded radius variation. Confirmation of the operation of reference value correction was carried out with regard to variations that occurred.

First, the value for front wheel rotation speed while the vehicle is turning in a constant circle was replaced by that value multiplied by a factor of X to artificially generate a tire dynamic loaded radius variation. It was then confirmed how ΔN_t would change with respect to ΔN_{re} , and how the correction factor X_{fa} would change. Figure 5 shows the results.

The point where ΔN_{re} increases instantaneously is the time when the front wheel rotation speed value is replaced. It is apparent that at the instant when ΔN_{re} changes, ΔN_t is

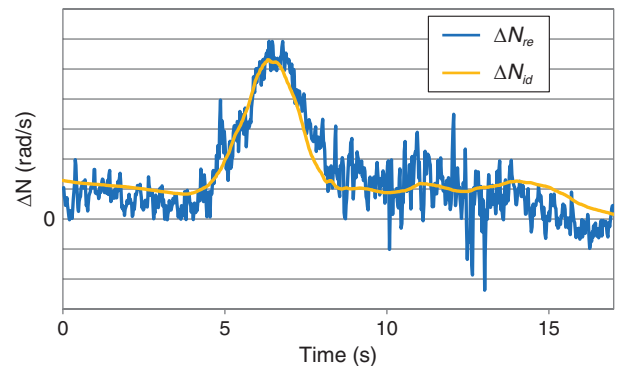


Fig. 4 Comparison of actual and calculated rotation speed difference

smaller, but the increase in X_{fa} causes ΔN_t to be corrected so that it again exceeds ΔN_{re} . As stated in section 3.2, rotation speed difference control is not implemented when ΔN_{re} is lower than ΔN_t . In other words, the tire dynamic loaded radius variation does not exert a continuous influence on the amount of driving force distributed, which is determined by the rotation speed difference control.

Next, the vehicle was driven with the front wheel air pressure lowered to create an actual front and rear wheel radius difference. Figure 6 shows the variations in ΔN_{re} , ΔN_t and X_{fa} that occurred when this was done. It is apparent that the existence of a radius difference between front and rear wheels means that there is less time when X_{fa} is one, and correction of ΔN_t is being implemented at all times. Moreover, the way that X_{fa} changes gently over time enables ΔN_t to follow the lower frequency component of ΔN_{re} .

4.3. Confirmation of the Effectiveness of Spin Suppression

Up to section 4.2, this paper has described methods for correcting the reference value without causing rotation speed difference control to operate when there is no spin. However, by correcting the reference value, correction factor X_{fa} reduces the steady-state deviation. This means that the amount of driving force distributed, which is determined according to the deviation, may be insufficient. Here the suppression effect of ΔN servo-control when there is front wheel spin was confirmed.

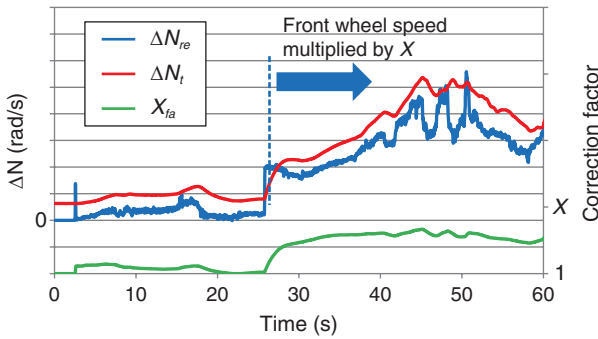


Fig. 5 Variation of correction factor for tire radius error

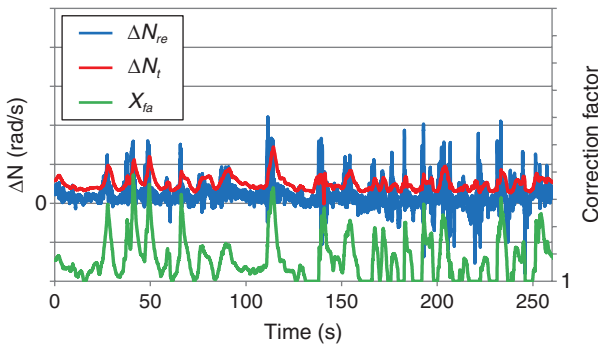


Fig. 6 Low front tire pressure driving data

Figure 7 shows the change component in rear wheel driving torque, which is determined by ΔN_{re} , ΔN_t and ΔN servo-control, when driving on a snowy road. ΔN_t does not follow the rapid change in ΔN_{re} when spin occurs, and ΔN servo-control therefore increases the rear wheel driving torque. It is apparent that this keeps ΔN_{re} in check. Moreover, the amount of ΔN servo-control implemented is largely zero, showing that the operation of spin suppression is kept in check when it is not needed.

Next to be considered is a scenario when front wheel slip can be expected, such as an uphill start from a complete stop on a snowy road. In such a case, the front wheels continue to slip for a relatively long time, causing a steady-state component in ΔN_{re} to occur. When correction is applied to this steady-state component, the amount of control implemented may be less than that needed to suppress spin.

As described in sub-section 3.2.2, therefore, a limiting value is applied within the range of error in which correction of X_{fa} is intended. Figure 8 shows the amount of ΔN_{re} , X_{fa} and ΔN servo-control implemented when starting from a complete stop on a snowy uphill road with the limit $1 \leq X_{fa} \leq X$.

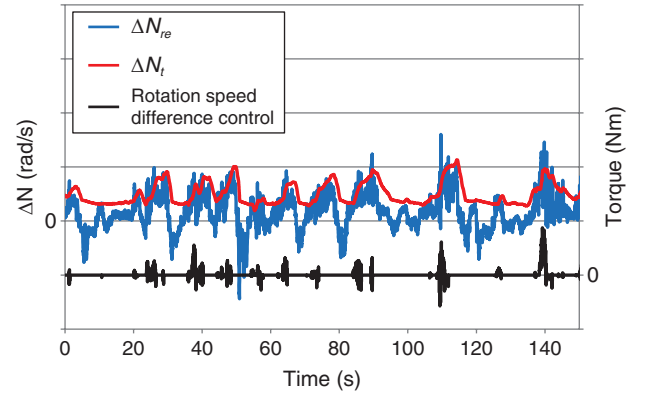


Fig. 7 Snowy road driving data

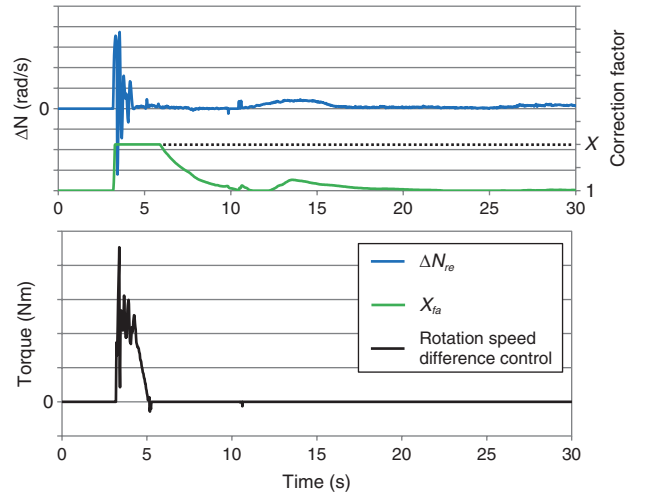


Fig. 8 Uphill snowy road driving data

It is apparent that X_{fa} rapidly increases with respect to ΔN_{re} , which is large when starting, and reaches the upper limit value X . This eliminates the decrease in deviation due to correction, so that PID is activated in ΔN servo-control, and the rotation speed difference converges.

5. Conclusion

In the implementation of spin suppression control that increases the amount of driving force distributed to the rear wheels in conjunction with the occurrence of spin in the front wheels, the rotation speed difference between front and rear wheels that occurs when the vehicle is turning was sought by geometrical means and applied to correct the target value. A correction factor determined from the steady-state deviation between the actual value and target value for rotation speed difference between the front and rear wheels was also used successfully to correct for rotation speed differences that occur due to such factors as variations in the tire dynamic loaded radius.

The use of these methods showed that spin suppression control would not operate even in vehicle states with a rotation speed difference between front and rear wheels that occurs when the front wheels are not spinning. This also showed that defining the range for the correction factor appropriately would allow limiting the reduction in the amount of spin suppression control implemented that occurs due to correction, and could rapidly produce convergence in the rotation speed difference between front and rear wheels.

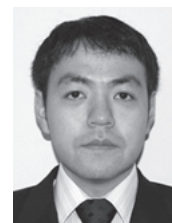
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■ Author ■



Shoichi OHYAGI



Takayuki SEKI

Development of Carburized High-strength Gear Steel with Less Rare Metals Added

Akihito NINOMIYA*

Yoshinari OKADA*

Masayuki HORIMOTO**

Shusaku MAEDA***

ABSTRACT

In conventional high-strength gear steel, the addition of Ni and Mo reduces the non-martensitic structure at the process of quenching the gear surface after carburizing. This makes the gear surface martensitic and thereby guarantees high fatigue strength. However, Ni and Mo are rare metals, and it has become impossible to ignore the increase in material costs caused by the addition of large quantities of these metals. Based on the mechanism by which non-martensitic structures form, we have developed a method for producing carburized high-strength steel in which the application of cold-oil to the quenching process improves the cooling speed and the chemical composition of the steel is optimized. Using no Ni and reducing Mo to less than half, we have developed a new gear steel with a greater level of fatigue strength than conventional steel. In other words, it was made possible to reduce the amount of rare metals used while simultaneously achieving high fatigue strength.

1. Introduction

The gears used to transmit power in a transmission use either general steel or high-strength steel depending on the performance required of them. In general, because low-speed gears, which have high-torque and frequent input, and the final gear, which is always subject to torque input, need to have high fatigue strength, there is a tendency for high-strength steel to be used for these gears. Additionally, in recent years, the need for automobiles to have reduced environmental impact and increased market appeal has led to more lightweight, compact multi-stage transmissions, and this in turn has resulted in the gears being subjected to greater loads. Typically, after a transmission gear has been forged and machined, to improve fatigue strength, carburizing and quenching are used to increase hardness, and shot peening is used to increase compressive residual stress.

Honda started using high-strength steel⁽¹⁾ in mass-production vehicles in the 1990s and is still using high-strength steel in many models today. In addition to the primary components of carbon steel, namely, C, Si, and Mn, the case-hardening steel that is used in gears that will undergo carburizing and quenching and other forms of

surface hardening treatment has Cr, Ni, Mo, V, and other alloy elements added to it in accordance with the purpose that it is being used for. While the components of the general steel known as SCr420 are C, Si, Mn, and Cr, high-strength steel has Ni and Mo added to its composition. This results in an increased rise in hardness in the outermost layer during carburizing and quenching and improved fatigue strength.

The reason carburization hardening is used to strengthen most transmission gears is that carburizing and quenching yields martensite, which is the hardest and strongest type of metal structure. Martensite can be formed by the rapid cooling of austenite. The hardness of martensite depends on the carbon concentration of the structure. The greatest hardness can be achieved at a carbon concentration between 0.7% and 0.8%⁽²⁾. The ideal state would be for the entirety of the gear from the inside to the surface to have a martensitic structure. However, this ideal state may be unachievable depending on the chemical composition of a steel and heat treatment conditions. Figure 1 shows the cross-sectional surface microstructure of gas carburized and hot-oil-quenched (a) SCr420 and (b) conventional high-strength steel. The black structure visible in the outermost layer of the SCr420 is an incompletely hardened,

* Automobile R&D Center

** Nippon Steel & Sumitomo Metal Corporation

*** NIPPON GREASE Co., Ltd.

non-martensitic structure (fine pearlite structure). This structure appears because intergranular oxidation reduces the hardenability of the steel surface. Figure 2 shows the mechanism of said reduction. The phenomenon of intergranular oxidation is unique to gas carburizing. When the residual furnace oxygen or the oxygen generated by decomposition of the carburizing gas diffuses into the steel, especially at the prior austenite grain boundary near the surface, the oxygen combines with the Si, Mn, and Cr in the steel to form oxides. The austenite near the intergranular oxides has locally reduced concentration of Si, Mn, and Cr dissolved therein. This leads to a reduced hardenability⁽³⁾⁻⁽⁶⁾ and makes it difficult for martensite to be formed during quenching. Because the oxygen diffusion rate in the steel is lower than the carbon diffusion rate, the depth of intergranular oxidation is shallower than that of carburizing layer. As a result, while the gas carburized steel is high in carbon, there is a possibility that localized

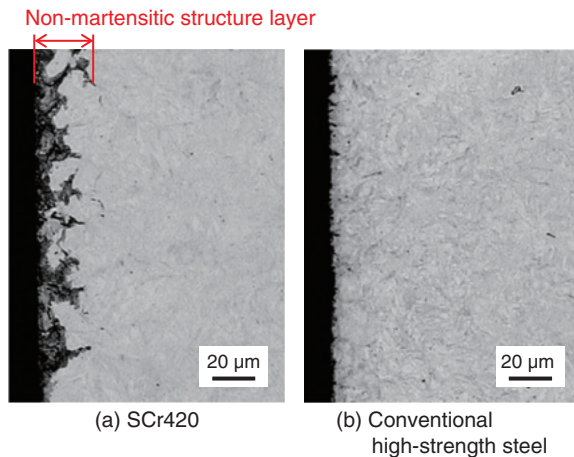


Fig. 1 Microstructure of gas-carburized steel surface

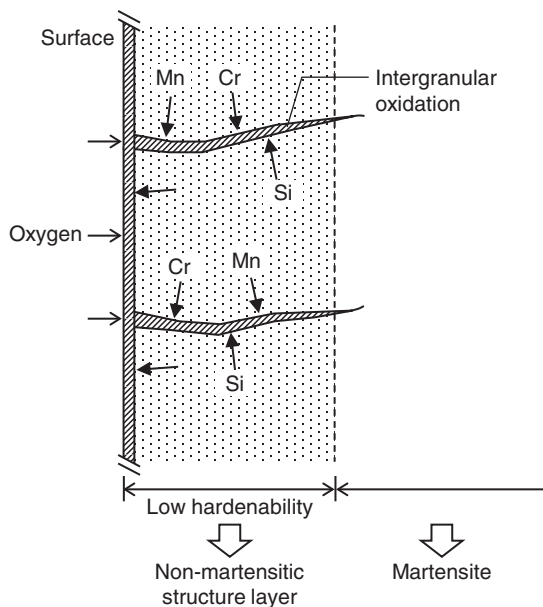


Fig. 2 Mechanism of non-martensitic structure layer generation

reduction in the hardenability in the outermost layer results in a non-martensitic structure with low hardness forming solely in the outermost layer. In conventional high-strength steel, Ni and Mo, which do not readily form oxides, are added. The addition of approximately 0.5%Ni and 0.8%Mo compensates for the reduction in the hardenability caused by the formation of intergranular oxides such as Si, Mn, and Cr. As shown in Fig. 1 (b), this results in reduced formation of non-martensitic structures and improves fatigue strength. Figure 3 shows the relationship between the depth of non-martensitic structure layer and the bending fatigue strength. The figure confirms that reducing the non-martensitic structures is an effective way of improving the fatigue strength. However, because supplies of Ni and Mo, which are rare metals, are unstable, and because these elements are subject to drastic increases in price, rising material costs have been an issue.

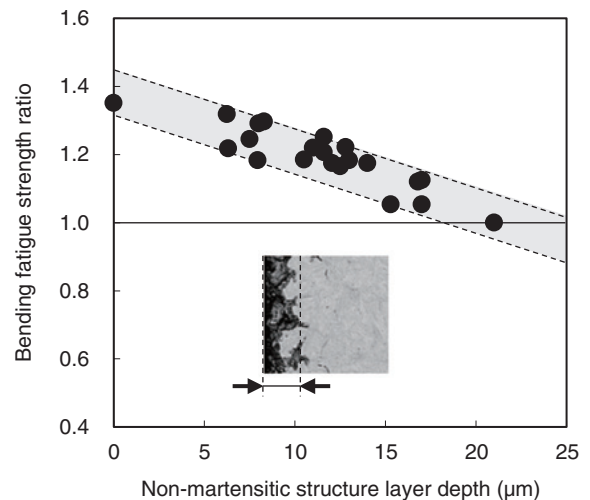


Fig. 3 Relationship between depth of non-martensitic structure layer and bending fatigue strength (SCr420 gas-carburized = 1.0)

2. Development Goals

Our goal was to reduce the cost of high-strength gear steel by a material composition design and heat treatment specifications that would make it possible to simultaneously reduce the addition of Ni and Mo and reduce the formation of non-martensitic structure (thereby increasing fatigue strength).

3. Reducing Non-martensitic Structure by Using Cold-oil Quenching

Two methods of reducing non-martensitic structure without high alloying (adding Ni and Mo) are vacuum carburizing and tooth-surface grinding after carburizing. In vacuum carburizing, there is no oxygen in the furnace, and intergranular oxidation does not occur. This means that there

is no reduction in the hardenability in the steel surface, and there is generally no non-martensitic structure. However, gas carburizing is currently dominant at Honda and other related manufacturers, and given existing facilities, the widespread application of vacuum carburization would be difficult. In tooth-surface grinding after carburizing, even if non-martensitic structure is formed, it can be ground away. However, in normal tooth-surface grinding, only the tooth flank surfaces are ground. This means that non-martensitic structures remain at the root of the teeth, and the tooth bending fatigue strength cannot be improved. Additionally, presupposing the use of tooth-surface grinding will lead to increased part costs. For the above reasons, it was decided to focus on developing a method using cold-oil quenching, which can be implemented using existing facilities and can reduce the formation of non-martensitic structure over the entire gear surface.

Typically, a gear is carburized at approximately 930°C and then cooled to 850°C. This temperature is held, and then the gear is rapidly cooled through oil quenching. At this time, whether non-martensitic structure forms on the steel surface depends on the hardenability of the material and the cooling rate during quenching. Figure 4 is a schematic diagram of the continuous cooling transformation (CCT) curve. When the cooling curve passes the pearlite transformation start line (line Ps), fine pearlite structures form. Figure 4 corresponds to the state of the surface of the gas-carburized SCr420 discussed in chapter 1. On the other hand, if the cooling curve does not pass line Ps, non-martensitic structure will not form. In other words, to reduce the formation of non-martensitic structure, it is necessary to help prevent the cooling curve from passing line Ps by shifting line Ps to the right along the time axis, shifting the cooling curve to the left along the time axis, or doing both. Mn, Cr, Ni, Mo, etc. are elements that have the effect of shifting line Ps to the right along the time axis. Among those elements, Ni and Mo, which have low affinities for oxygen and low susceptibility to intergranular oxidation, have been used for a hardening method for conventional high-strength steel.

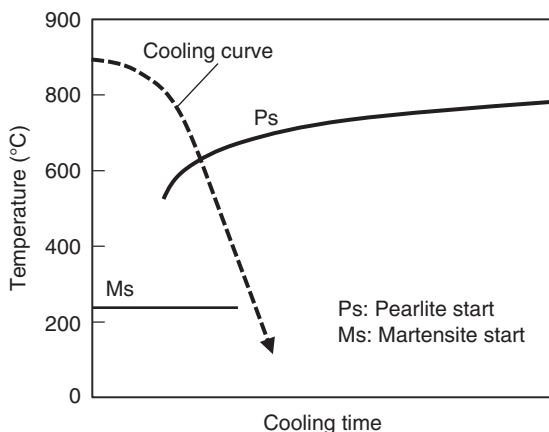


Fig. 4 Schematic diagram of CCT curve

In the developed method, to reduce the addition of Ni and Mo, cold-oil quenching was used as a means of moving the cooling curve to the left along the time axis; that is, increasing the cooling rate. Typically, cold-quenching oil is used to ensure the internal hardness of the large gears used in construction equipment and other parts with high heat masses. However, in the developed method, we cooperated with an oil manufacturer to select a cold-quenching oil that excels in cooling the outermost surface of steel. Figure 5 shows the cooling curves of some of the cold-quenching oils that were considered, which are labeled as cold-quenching oils (A), (B), and (C)⁽⁷⁾. The figure also shows the cooling curve of one of the hot-quenching oils used in mass production. Figure 6 shows the measured Barkhausen noise (BN)⁽⁸⁾ values of the outermost root surfaces of the gear carburized and quenched by cold oils (A), (B), and (C). The BN value correlates with the microstructure and hardness of the steel, and a smaller BN value corresponds to a higher hardness. Therefore, cold-quenching oil (C) was selected as the cold-quenching oil that most reliably increases the hardness of the outermost gear root surfaces.

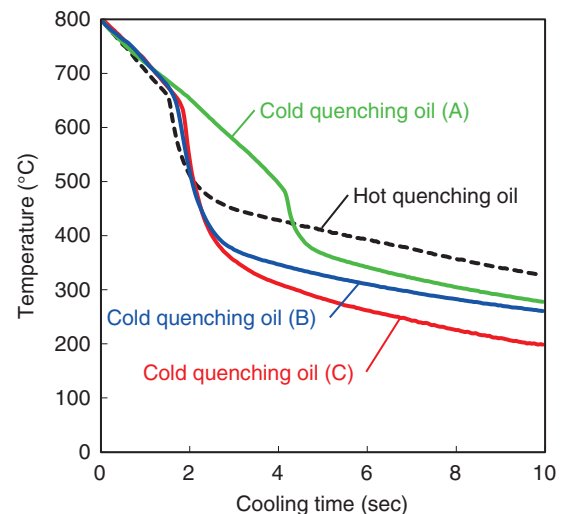


Fig. 5 Cooling characteristics of quenching oil (JIS-K-2242 method, silver probe test)

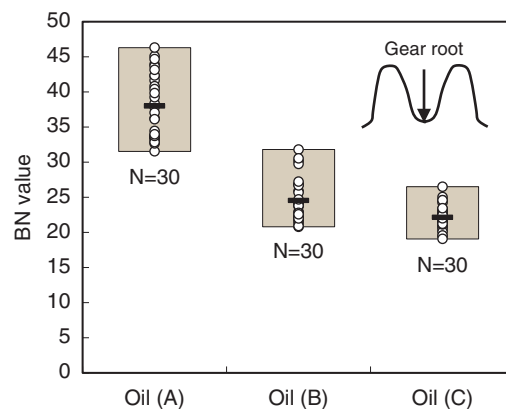


Fig. 6 Barkhausen noise (BN) value of gear root quenched by cold oil (A), (B), (C)

4. Material Composition Design

Presupposing the gas carburizing and quenching using the cold-quenching oil selected in chapter 3, a component balance that would be effective for reducing the formation of non-martensitic structure was searched. First, we calculated the perlite transformation start line of the SCr420 carburized surface by using Sente software Ltd.'s JMatPro simulation software. In this calculation, we set the carbon concentration on the surface of SCr420 as 0.75% to simulate a carburized surface. In Fig. 7, both Ps_1 and Ps_2 were the calculated results. Ps_1 stands for the perlite transformation start line with no intergranular oxidation, and Ps_2 stands for the same line in consideration of concentration reduction of dissolved Si, Mn, and Cr accompanying intergranular oxidation. In the figure, temperature hysteresis when gear surface temperatures were measured in the cold-oil quenching from 850°C is indicated as a cooling curve. Compared to Ps_1 , Ps_2 is shifted to the left on the time axis and intersects with the cooling curve. This means that even if the selected cold-quenching oil is used, in compositions, such as SCr420, in which components with low affinity for oxygen, such as Ni and Mo, are not added, intergranular oxidation will lead to localized insufficiencies in the hardenability, and non-martensitic structure may form. Thus, in our steel development, we focused on Mo because it has higher hardenability than Ni though used in smaller amounts. Optimal amount of added Mo was sought. In Fig. 7, Ps_3 shows the calculated result of the developed steel (with 0.3% Mo added). As in Ps_2 , it is simulated that the steel surface during carburizing with a 0.75% carbon concentration and intergranular oxidation is also taken into account. Ps_3 does not intersect with the cooling curve, and this indicates that it is possible to suppress the formation of non-martensitic structure through adding no Ni and reducing Mo to less than half compared to conventional high-strength steel.

Ways to reduce the depth of intergranular oxidation were sought. Figure 8 shows the effects of Si, Mn, and Cr on intergranular oxidation depth. It was found that to reduce

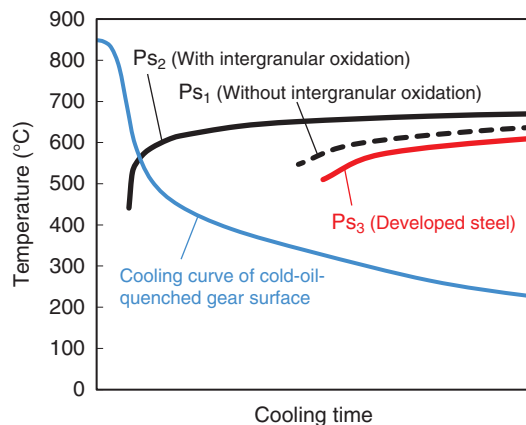


Fig. 7 CCT curve of SCr420 (0% Ni, 0% Mo) and developed steel (0% Ni, 0.3% Mo)

Table 1 Chemical composition

	(mass%)					
	C	Si	Mn	Ni	Cr	Mo
Developed Steel	0.20	0.08	0.65	-	1.05	0.35
Conventional high-strength steel	0.20	0.05	0.40	0.50	0.60	0.80

the intergranular oxidation depth, in addition to reducing the amount of Si added, increasing the ratio of Cr to Mn is also effective. The depth of the intergranular oxidation of conventional high-strength steel is approximately 5 μm . To achieve a similar grain boundary oxidation depth, the amount of Si added was reduced and a composition that satisfies the condition $\text{Cr/Mn} \geq 1.3$ was used.

On the basis of the above results, composition of the developed steel was determined. Table 1 shows a representative composition of the developed steel. Figure 9 shows the cross-sectional surface microstructures of SCr420 and developed steel that are gas carburized and cold-oil quenched. In the SCr420, non-martensitic structure

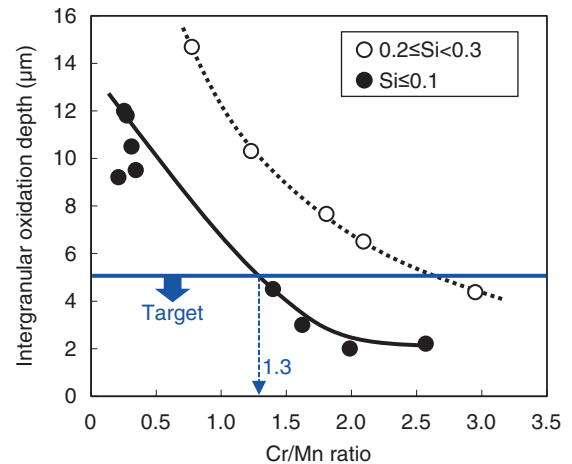


Fig. 8 Relationship between Cr/Mn ratio and intergranular oxidation depth

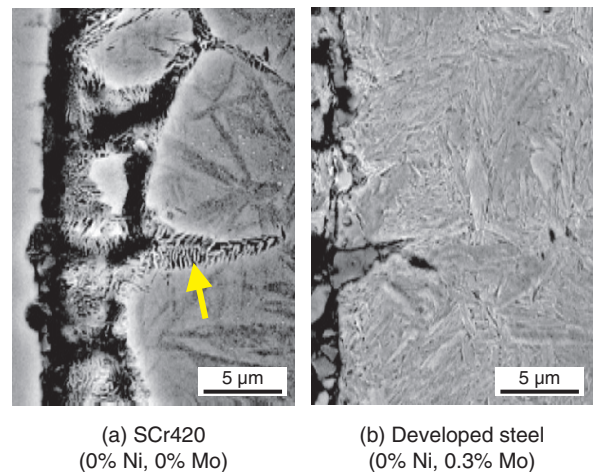


Fig. 9 Scanning electron micrograph of steel surface that was gas-carburized and cold-oil quenched (Arrow mark indicates non-martensitic structure)

can be identified near the intergranular oxidation. However, suppression of these structures was confirmed in the developed steel.

5. Characteristics of Developed Steel

5.1. Heat Treatment Quality

Figure 10 provides an example of the microstructure and hardness distribution of the developed steel after gas-carburizing and quenching and tempering. Just as in the conventional high-strength steel shown in Fig. 1 (b), the developed steel has a structure in which the formation of non-martensitic structure is reduced and also has a high surface hardness (of 720 HV or greater).

5.2. Gear Fatigue Strength

To compare the gear fatigue strengths of the developed steel and conventional high-strength steel, the tooth bending fatigue strength and the pitting strength were evaluated by using a gear fatigue testing machine. The gears used for the evaluation of both types of steel had the same shape, and no tooth-surface grinding was performed after carburization. Figure 11 shows the results of the gear fatigue testing. In

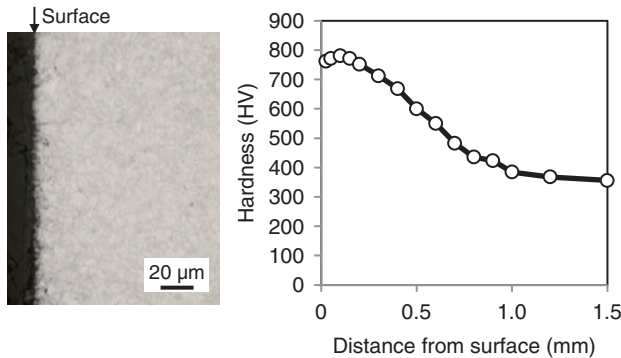


Fig. 10 Microstructure and hardness distribution of gas-carburized developed steel

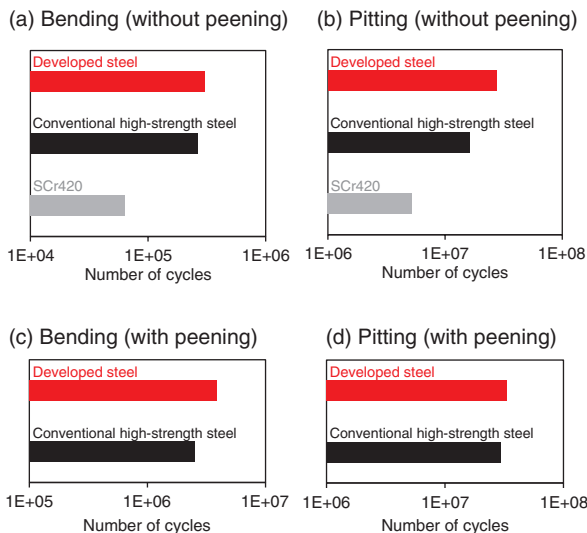


Fig. 11 Gear bending and pitting fatigue strength

the tooth bending, a test torque of 320 Nm was used, and in the pitting, a test torque of 200 Nm was used. Results (a) and (b) are for when there is no peening after carburization. Results (c) and (d) are for when there is peening after carburization. In all the results, the developed steel exhibits a fatigue strength that is equivalent to or greater than that of conventional high-strength steel. Figure 12 shows the strength position of the developed steel.

5.3. Tooth Precision

To compare the heat-treatment distortion characteristics of the developed steel and conventional high-strength steel, we evaluated the dimensional variation after heat treatment by using a final gear that is a representative example of high-strength steel parts. The final gear made of the developed steel was produced with the same processes as mass production except for carburizing and quenching processes. As one of the evaluation results, Fig. 13 shows lead slope deviation. Both the dimensional variation and its deviation of the developed steel were confirmed to be equivalent to or less than those of conventional high-strength steel.

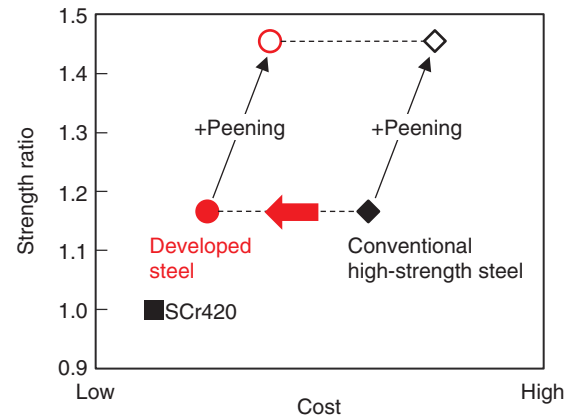


Fig. 12 Position of developed steel

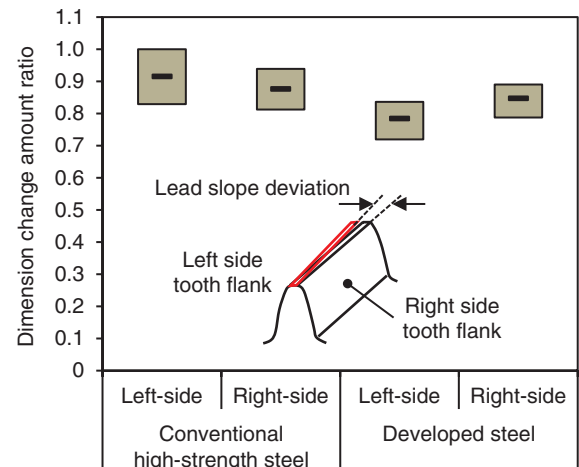


Fig. 13 Dimension (Lead slope deviation) change amount (Maximum change amount = 1.0)

6. Conclusion

By designing the optimal steel composition for carburizing and cold-oil quenching, a high-strength gear steel of equivalent or greater strength than conventional high-strength steel was developed, with lower amounts added of the rare metals Ni and Mo. Thus, cost reduction was achieved by using the developed steel instead of conventional high-strength steel. The following facts have been made clear in the course of development.

- (1) It is possible to reduce the formation of non-martensitic structure while mitigating increases in material costs by appropriately combining increased cooling rate during quenching and material composition design.
- (2) Cold-oil quenching eliminates the need for Ni and reduces the amount of Mo to less than half, which were necessary for conventional, hot-oil quenched high-strength steel.
- (3) To decrease the intergranular oxidation depth, which is one cause of non-martensitic structure formation in gas carburizing and quenching, in addition to reducing the amount of Si added, increasing the ratio of Cr to Mn is also effective.

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■ Author ■



Akihito NINOMIYA



Yoshinari OKADA



Masayuki HORIMOTO



Shusaku MAEDA

Development of Two-sensor Crash Detection Method for Motorcycle Airbag System

Yuki KOBAYASHI*

Takumi MAKABE*

ABSTRACT

An airbag system for a motorcycle was developed by establishing a crash detection method using two accelerometers, which reduced the number of sensors compared to the previous system. In the previous system, the Safing Judgment method was employed based on signals from four accelerometers. In the newly developed system, the judgment method was established with only two sensors by utilizing the difference in characteristics of the waveforms of sensor outputs. Installation freedom for the sensors was enhanced due to the space for installation gained by the removal of two sensors.

1. Introduction

Exploratory studies of motorcycle airbags began in the 1970's. In those studies, airbag concepts were examined. The crash detection method was also studied as a vital element of an airbag system. In the studies previously conducted, preliminary crash detection methods were proposed based on impact measurements in a front wheel-suspension system⁽¹⁾⁻⁽³⁾.

Kuroe et al. proposed a system with accelerometers installed on each suspension fork leg, and preliminary studies of an impact sensing system for a large scooter type and a large touring type motorcycle were conducted^{(4), (5)}. This system used the averaged value of the signals from the two accelerometers on each of the two front suspension fork legs to remove the influences of the rotational movement of the steering system, which occurs during a collision. Based on these detection methods, Honda developed a crash detection method that used the signals from a total of four accelerometers, two on each front suspension fork leg⁽⁶⁾. This system was applied to the 2006 Honda Gold Wing, which was released as the first airbag system for a production motorcycle. Figure 1 shows the configuration of the airbag system applied to the Gold Wing.

Four sensor signals were utilized in the system to satisfy several requirements, such as a reduction of the influence of the steering motion in a crash event and inclusion of a fail-safe function. However, this system was only applicable to particular models because the space availability for the installation of accelerometers depended on the structure of the motorcycle. In many cases, the space availability was limited because of the suspension arrangement, such as inverted telescopic, trailing links, etc., the suspension length, and pre-installed devices, including brake calipers.

This study was intended to overcome those restrictions and we tried to establish a crash detection method using the signals from just two accelerometers, one each on the right and the left front suspension fork legs.

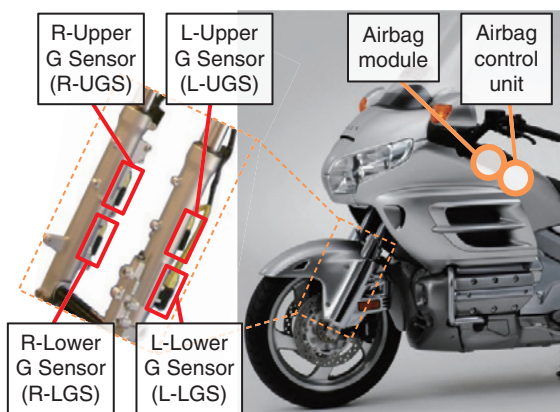


Fig. 1 Configuration of first airbag system for Gold Wing

2. Previous System

Table 1 shows the major specifications of the motorcycle shown in Fig. 1.

In this section, the data was measured using this motorcycle.

2.1. Crash Detection Method

Figure 2 shows a block diagram of the crash detection method for the previous system.

This system makes a crash judgment by means of four

* Motorcycle R&D Center

Table 1 Specifications of test motorcycle

Length	(mm)	2,630
Width	(mm)	945
Height	(mm)	1,525
Wheelbase	(mm)	1,690
Curb mass	(kg)	425
Engine displacement	(cm ³)	1,832

accelerometers: R-LGS (Right-Lower G Sensor), L-LGS (Left-Lower G Sensor), R-UGS (Right-Upper G Sensor) and L-UGS (Left-Upper G Sensor).

The Main Judgment was used to determine whether a crash had occurred that was severe enough to require deployment of the airbag.

The Safing Judgment acted as a fail-safe function to prevent the unwanted deployment of the airbag, due to a malfunction of, or an erroneous signal from, the accelerometers used for the Main Judgment.

2.2. Averaging Process

This method adopted the judgment processes based on the averaged values of the signals from the right and the left accelerometers. The averaging processes effectively compensated for the effect caused by the rotation of the front wheel-suspension system being forcefully steered by the impact in the early stages of a crash.

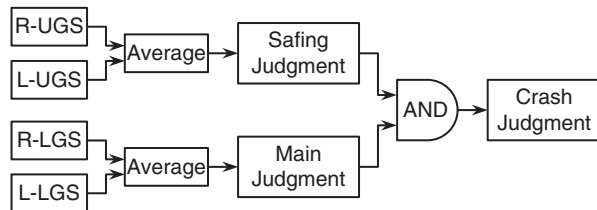
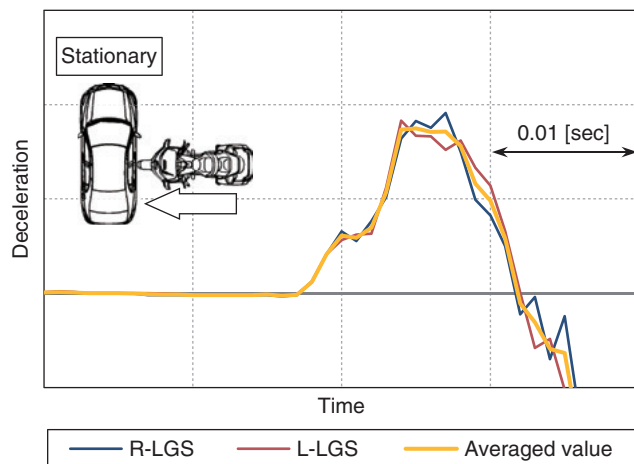
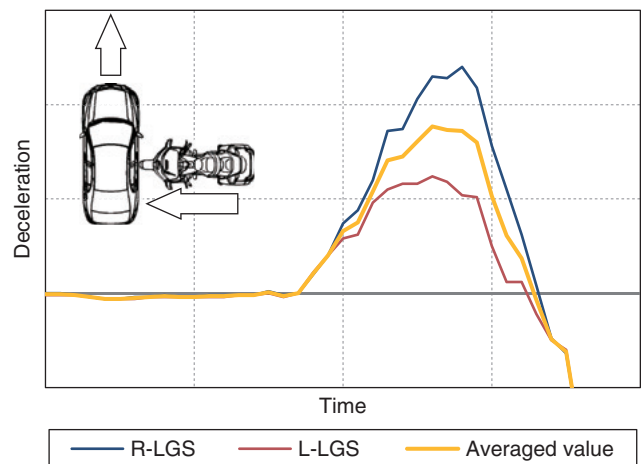


Fig. 2 Block diagram of previous crash detection method



(a) Crash on stationary vehicle



(b) Crash on moving vehicle

Fig. 3 Comparison of LGS deceleration

As an example of this effect, Fig. 3(a) and (b) show the deceleration signals measured at the front suspension fork legs in the two configurations of impact in line with ISO13232⁽⁷⁾. These impact configurations differ in the speed of the opposing vehicle; (a) is stationary and (b) is moving. Other conditions (e.g. motorcycle impact speed, contact point, relative heading angle) are the same between the two configurations.

While the deceleration value of R-LGS was equivalent to that of the L-LGS in the configuration of the stationary car impact, shown in Fig. 3(a), the deceleration value of the L-LGS was lower than that of the R-LGS in the configuration of the moving target impact, shown in Fig. 3(b). This is because the front wheel-suspension system was forcibly steered to the right by the opposing vehicle, which was moving in a lateral direction to the motorcycle. However, the averaged value of the deceleration values from the R-LGS and the L-LGS was equivalent in both impact configurations. Consequently, by using the averaging process, the effect of the steered front wheel was nearly eliminated.

2.3. Safing Judgment

The Main Judgment, which was based on the averaged signals of the R-LGS and the L-LGS, detected a crash erroneously when the R-LGS or the L-LGS output a high deceleration value due to a malfunction of the accelerometer. The Safing Judgment was provided in order to prevent the unwanted deployment of the airbag during normal operation.

In the previous airbag system, in addition to the accelerometers for the Main Judgment, two other accelerometers (the R-UGS and the L-UGS) were installed on the right and the left front suspension fork legs for the Safing Judgment, which operated also based on the averaged value. As crash detection was made by the logical conjunction of the Main Judgment and the Safing Judgment, even if either judgment improperly detected a crash due to the failure, or the like, of one accelerometer, the other judgment prevented an improper deployment signal to the airbag inflator.

The functions required for the Safing Judgment mentioned above are summarized as follows;

(1) Fail-safe function

The Safing Judgment is required to be always OFF in normal riding conditions. This is because the prevention of an airbag deployment is required when the Main Judgment detects a crash incorrectly.

(2) Inhibition of unnecessary prevention of a proper deployment of the airbag

The Safing Judgment is required to be ON under conditions that require proper deployment of the airbag. At the same time, there should not be any prevention of deployment based on crash detection by the Main Judgment.

In order to meet these requirements, the Safing Judgment necessitates the ability to discriminate between collisions and events during normal riding conditions.

3. Two-sensor Crash Detection System

The data used in this section was measured using the same vehicle as introduced in section 2.

3.1. Crash Detection Method

Figure 4 shows a block diagram of the two-sensor crash detection method. Two upper accelerometers were removed from the previous system. Crash detection was performed by the logical conjunction of the Main Judgment, which was calculated using the averaged value of the R-LGS and the L-LGS, and the R-Safing Judgment and the L-Safing Judgment, which were calculated using the output of the R-LGS and the L-LGS. With this configuration, for example, if an output abnormality occurs in the R-LGS, the Main Judgment and the R-Safing Judgment may incorrectly detect the occurrence of a crash. However, because the L-Safing Judgment is OFF, an undesired deployment can be prevented.

In the two-sensor crash detection system, the same judgment process as in the previous system was applied to the Main Judgment. Therefore, the Main Judgment had comparable discrimination performance. However, the averaging process was not applied to the signals for the Safing Judgments. The individual signals from the R-LGS and the L-LGS were inevitably affected by the steering motion of the front wheel as described earlier. This is why it was necessary to develop a different method for this system.

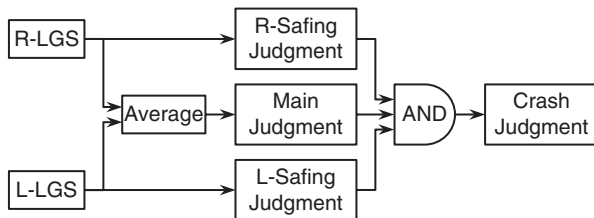


Fig. 4 Block diagram of two-sensor crash detection method

3.2. Safing Judgments Based on Unaveraged Signals

The same target performance as achieved by the previous method was set for the Safing Judgments. Discrimination between events during normal riding conditions and collisions that accompany the effect of the steered motion was required for the Safing Judgments.

By the way, normal riding conditions include running on rough roads. Figure 5 shows the waveforms measured when running on a specific rough road. The rough road included some potholes, and a deceleration peak of G_0 was recorded when the vehicle ran over the pothole.

To compare with those rough road running conditions, the collision where a passenger vehicle gave the impact to a motorcycle with the handle fully steered to the left was considered as an example of a situation where the steering motion occurred. The waveforms measured in such a collision are shown in Fig. 6. The scales of the vertical axes of Figs. 5 and 6 are the same.

There was a difference between the measured decelerations of the R-LGS and the L-LGS. The deceleration peak of the L-LGS was 2.2 times as big as G_1 , which was the deceleration peak of the R-LGS caused by the collision accompanying the left-steered state. The magnitude of G_1 was the same level as G_0 (shown in Fig. 5). Therefore it was

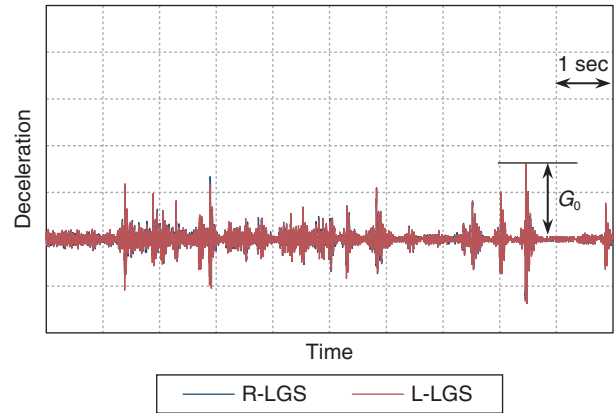


Fig. 5 Deceleration time history in rough road running

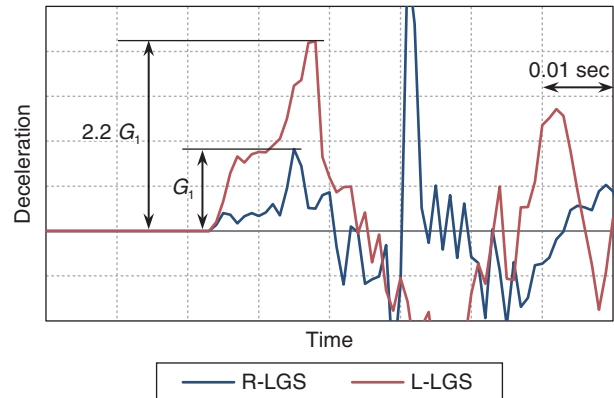


Fig. 6 Deceleration time history in steered collision

concluded that the Safing Judgment could not be conducted by just comparing the peak deceleration values.

3.3. Frequency Characteristics of Signals

Figure 7 shows the waveform in the vicinity of peak value of the waveform measured during rough road running, as shown in Fig. 5. The signals measured during rough road running show damped oscillation waveforms.

The spectrograms of measured deceleration values are shown in Fig. 8. The vertical axis of the figure indicates

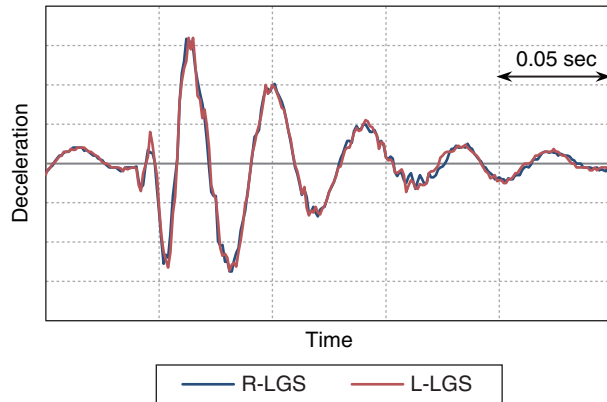


Fig. 7 Oscillation waveform of front suspensions

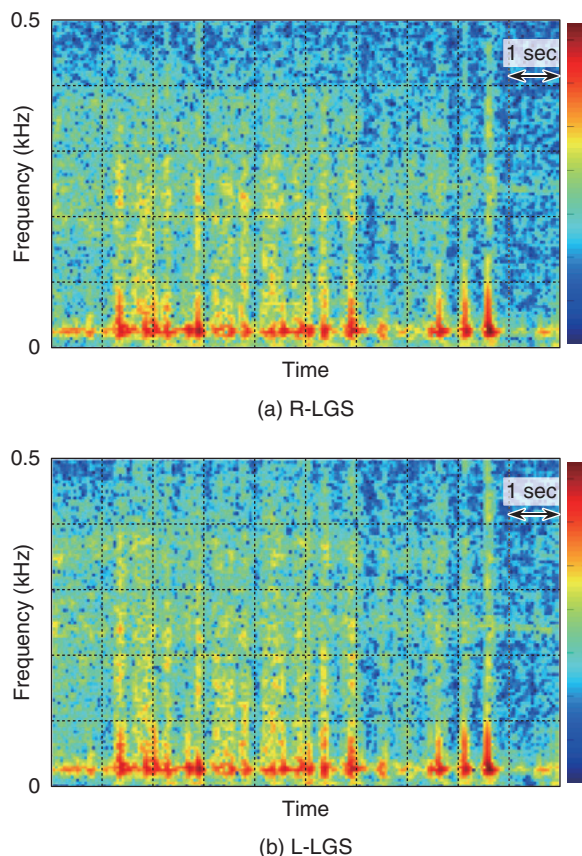


Fig. 8 Spectrogram of waveform of deceleration measured during rough road running

the frequency, the horizontal axis indicates the time of the measurement, and the colors of the pixels indicate the intensity of the signals.

During rough road running, random-seeming excitation forces are applied to the front wheel-suspension system with the uneven road surfaces. However, the spectrograms show high spectral density at around a fixed low frequency of 0.02 kHz. This frequency is almost equal to that of the natural frequency of the front wheel-suspension system. As the direction of the sensitivity of the accelerometer was perpendicular to the direction of the front suspension stroke, the acceleration along the axis of the suspension, which was caused by the suspension strokes absorbing the road unevenness, was not detected by the accelerometers. On the other hand, the front wheel-suspension system, which had a cantilever structure, was vibrated back and forth by the road unevenness in an orthogonal direction to the stroke of the front suspension. The oscillation cycle period during rough road running was determined by the inherent characteristics of the front wheel-suspension system, regardless of the attributes of the road unevenness.

We made the same frequency analysis on the waveforms of the decelerations measured in the steered collision (Fig. 9).

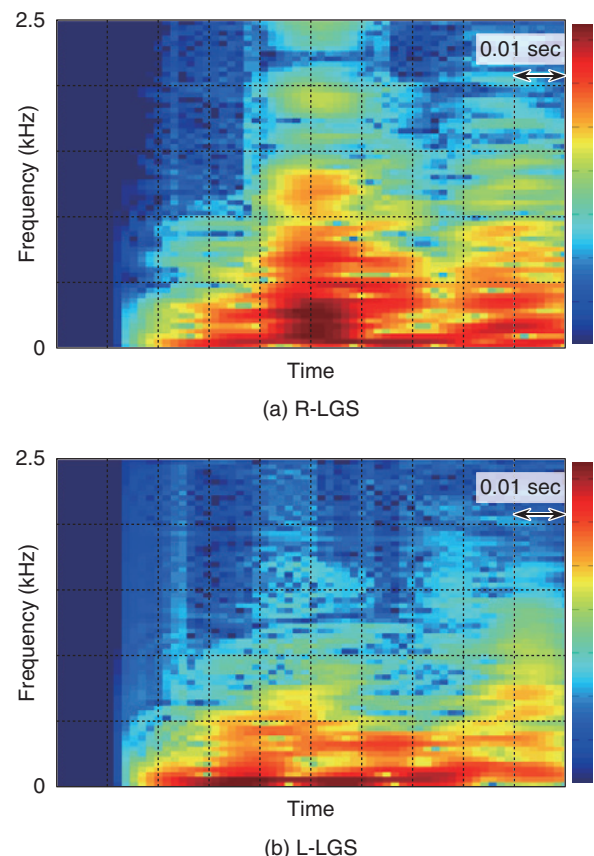


Fig. 9 Spectrogram of waveform of deceleration measured in steered collision

As the front fork was deformed beyond the extent of the elastic deformation during the steered collision, the detected waveform had no effect of the natural oscillation. As a result, the spectrum was dispersed over a wide area below 1.0 kHz.

3.4. Characteristics of Displacement-Deceleration Curve

Based on the spectral characteristics of the waveforms mentioned in section 3.3, we tried to establish a method to discriminate between a collision and rough road running. For this purpose, a theoretical analysis was conducted using approximated waveforms of the data.

To conduct the approximation of the waveform to present rough road running as part of normal riding conditions, all of the data should have been considered as the target data for the approximation. This was because the Safing Judgment process should stay OFF during any normal riding conditions. On the other hand, for the approximation of the waveform to present the actual impact in a steered collision, the deceleration that appeared in the initial part of the waveform was the only part to be approximated. This was because the ON-judgment was executed based on only the deceleration that was generated in the early stage of the impact.

The deceleration waveform measured when running on rough roads is approximated by Eq. (1) as a damped oscillation.

$$m\ddot{x} = -kx - c\dot{x} \quad (1)$$

Where

- m : Mass
- k : Spring constant
- c : Damping coefficient

Assuming initial position $x(0)$ as 0 and initial velocity $\dot{x}(0)$ as v_0 , the solutions are as follows:

$$x(t) = \frac{v_0}{\sqrt{1-\zeta^2}\omega_0} e^{-\zeta\omega_0 t} \sin(\sqrt{1-\zeta^2}\omega_0 t) \quad (2)$$

$$\dot{x}(t) = \frac{v_0}{\sqrt{1-\zeta^2}} e^{-\zeta\omega_0 t} \cos(\sqrt{1-\zeta^2}\omega_0 t + \phi) \quad (3)$$

$$\ddot{x}(t) = -\frac{v_0\omega_0}{\sqrt{1-\zeta^2}} e^{-\zeta\omega_0 t} \sin(\sqrt{1-\zeta^2}\omega_0 t + 2\phi) \quad (4)$$

Where

$$\zeta = \frac{c}{2\sqrt{mk}}$$

$$\omega_0 = \sqrt{\frac{k}{m}}$$

$$\phi = \tan^{-1}\left(\frac{\zeta}{\sqrt{1-\zeta^2}}\right)$$

We determined each parameter from the results of measurements in actual running tests, and obtained the approximate waveform shown in Fig. 10. The damping coefficient ζ was 0.08. Assuming $\phi \approx 0$ which is considered feasible as $\zeta \ll 1$, Eq. (5) can be obtained from Eq. (2) and Eq. (4).

$$\ddot{x}(t) \approx -\omega_0^2 x(t) \quad (5)$$

Thus, the waveform of the acceleration represents an oscillation almost reverse in phase to that of the displacement.

On the other hand, the initial part of the waveform in a steered collision is approximated by half-sine wave as described in Eq. (6).

$$\ddot{x}(t) = \begin{cases} 0 & (t < 0) \\ a_0 \sin \omega t & (0 \leq t \leq \pi/\omega) \end{cases} \quad (6)$$

The displacement is expressed by Eq. (7).

$$x(t) = \begin{cases} 0 & (t < 0) \\ a_0 (\omega t - \sin \omega t)/\omega^2 & (0 \leq t \leq \pi/\omega) \end{cases} \quad (7)$$

Figure 11 shows the approximated waveform of the R-LGS shown in Fig. 6, where the constants of a_0 and ω are assumed to be the same as those in the damped oscillation in Eq. (4).

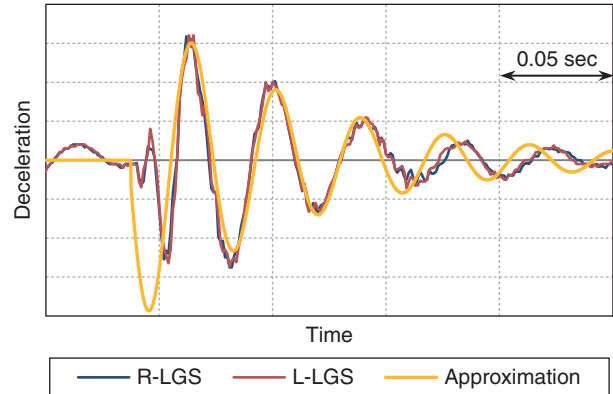


Fig. 10 Approximation of oscillation of front suspensions

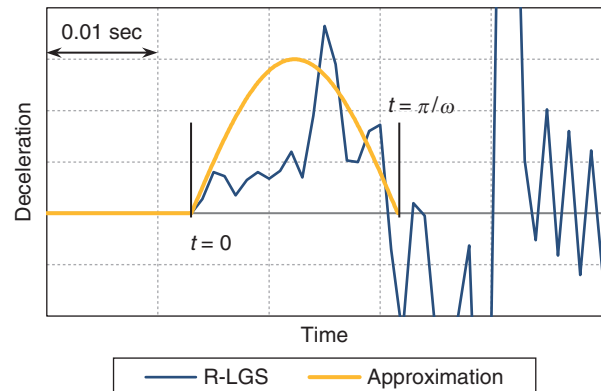


Fig. 11 Approximation of deceleration of steered collision

These approximated waveforms were transformed into the Displacement-Deceleration curves shown in Fig. 12. The vertical axis is defined for the deceleration and the horizontal axis is for the displacement. Curves of the damped oscillations are situated over the second and fourth quadrants as the displacement and the deceleration have opposite phases to each other. On the other hand, the half-sine wave is situated in the first quadrant as the displacement increases monotonically.

By utilizing these characteristics, it was possible to distinguish between rough road running and a steered collision with an appropriate judgment threshold line in the first quadrant of the Displacement-Deceleration plane.

3.5. Application to Crash Detection

Figure 13 shows a block diagram of the Safing Judgment applied to the two-sensor crash detection system.

The displacement (ΔS) was derived by short-term integration. The displacement (ΔS) of the damped oscillation, given by Eq. (4), is derived as follows;

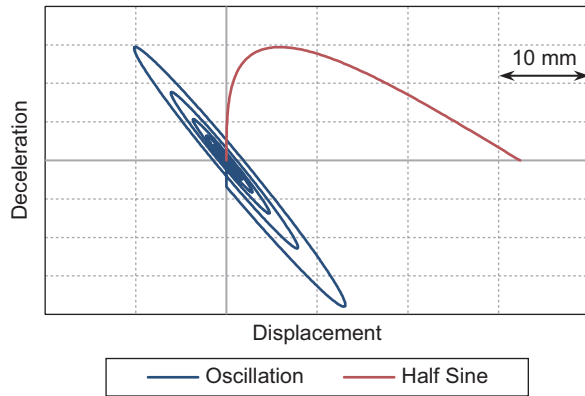


Fig. 12 Displacement-Deceleration curves of approximations

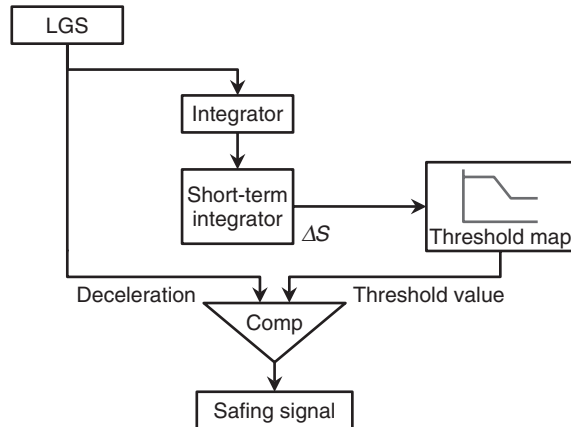


Fig. 13 Block diagram of Safing Judgment for two-sensor crash detection system

$$\Delta S(t) = \int_{t-T_0}^t \dot{x}(t) dt$$

$$= \frac{v_0}{\sqrt{1-\zeta^2}} e^{-\zeta\omega_0 t} \sin \{ \sqrt{1-\zeta^2} \omega_0 t - \alpha(T_0) \} \quad (8)$$

Where

ΔS : Displacement

T_0 : Interval of short-term integration

$$\sin \{ \alpha(T_0) \} = - \left| \cos \left(\frac{\omega}{2} T_0 \right) \right|$$

$$\cos \{ \alpha(T_0) \} = \left| \sin \left(\frac{\omega}{2} T_0 \right) \right|$$

The interval of the integration (T_0) was determined from the natural frequency and was defined so that the displacement was to be opposite in phase to the acceleration.

$$T_0 = \frac{\pi}{\sqrt{1-\zeta^2} \omega_0} \quad (9)$$

Figure 14 shows the Displacement-Deceleration curves of the actual data, which were measured during rough road running and a steered collision.

Plots on the first quadrant on the Displacement-Deceleration plane show that the status of the front wheel-suspension system is shifted to an area behind the normally located position and is decelerating as well. The ON area was set in this area and the threshold line was specified as shown in Fig. 14. Because it was translated that the impacts were applied to the extent at which the deformation of the front wheel-suspension system was lead to the plastic deformation range.

The waveforms when running on rough roads and when experiencing a steered collision are distinguishable on the Displacement-Deceleration plane even though their deceleration peaks reach the same levels.

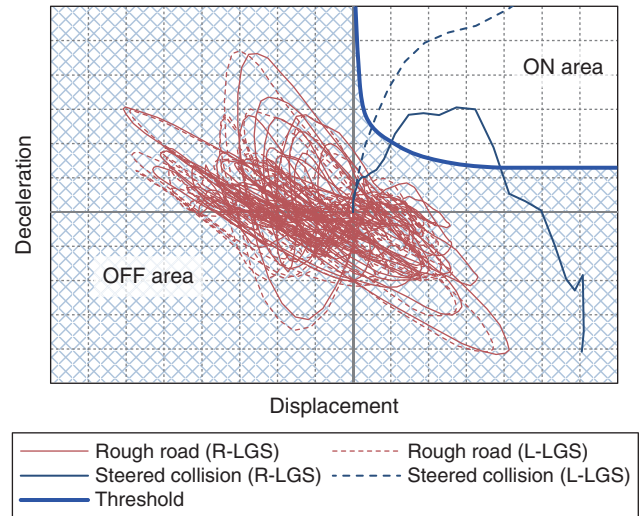


Fig. 14 Displacement-Deceleration curves and threshold of two-sensor Safing Judgment

Table 2 Evaluation result of two-sensor Safing Judgment

Configurations	Result				
	Main Judgment		Safing Judgment		
	Previous method	Two-sensor	Previous method	Two-sensor	
				R	L
Rough road	OFF	OFF	OFF	OFF	OFF
413-0.0/13.9*	ON	ON	ON	ON	ON
413-6.9/13.9*	ON	ON	ON	ON	ON
Steered collision	ON	ON	ON	ON	ON

*Impact configuration codes defined in ISO 13232

Table 3 Specifications of large scooter

Length	(mm)	2,185
Width	(mm)	750
Height	(mm)	1,180
Wheelbase	(mm)	1,545
Curb mass	(kg)	204
Engine displacement	(cm ³)	248

4. Verification

We compared the ON/OFF judgment results between the previous system and the two-sensor crash detection system by using the acceleration data measured in the same motorcycle discussed in section 3. As shown in Table 2, we obtained results equivalent to those obtained using the previous method in all configurations.

Furthermore, verification of the method was conducted using a large scooter as well. The major specifications of the scooter are shown in Table 3.

Unlike the large touring motorcycle, the large scooter did not have an upper triple clamp on its front fork and had a smaller front wheel. However, the waveforms measured during the rough road running tests showed damped oscillations (Fig. 15).

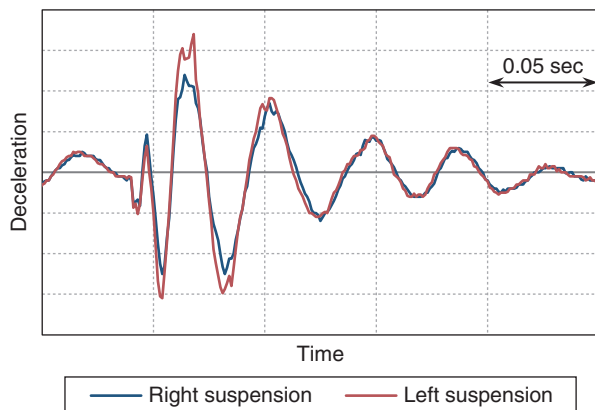


Fig. 15 Oscillation waveform of front suspension of large scooter during rough road running

The Displacement-Deceleration curves of the large scooter are shown in Fig. 16. They showed features similar to those in the large touring motorcycle, even though the parameters of natural oscillations in the large scooter were different from those in the large touring motorcycle because of the differences in the structures of the front wheel-suspension system.

These results verified that the new method was applicable to the large scooter type motorcycle as well.

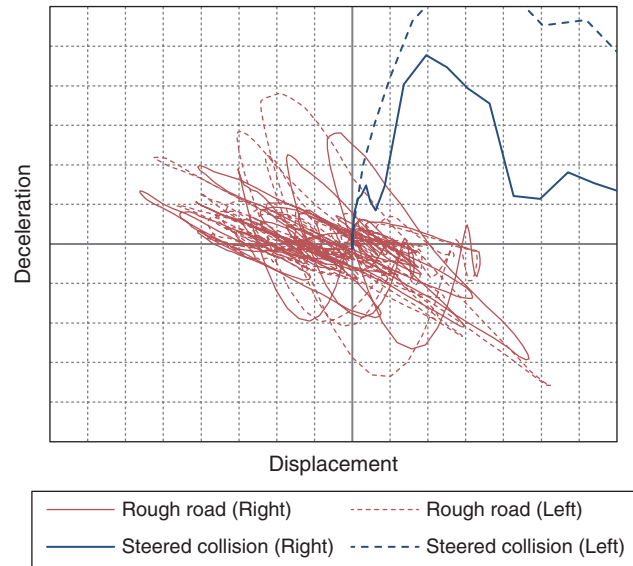


Fig. 16 Displacement-Deceleration curves measured in large scooter

5. Conclusions

A two-sensor crash detection method was successfully developed. The method utilized the characterized oscillation behavior that appeared in the motion of the front wheel-suspension system experienced when running on rough roads.

Following the development of the method, the validity of the method was then confirmed and a judgment performance equivalent to the previous method was verified.

In addition, based on data measured using large scooters it was confirmed that this method was also applicable to motorcycles with different front wheel-suspension system configurations.

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■ Author ■



Yuki KOBAYASHI



Takumi MAKABE

Development of High-performance PUF and Organoleptic Evaluation of Polyurethane Foam for Car Seat

Naoto ISHII* Tomoki MATSUMOTO* Tsutomu AO*

ABSTRACT

A polyurethane foam for a vehicle seat cushion was developed in order to improve seat comfort and drivability, as well as to reduce ride vibration and weight. An organoleptic evaluation of seat comfort, which is a static property, was performed by examining hardness, wobbling, bouncing sensations and the bottoming feel, and the correlations between each evaluation marker and the polyurethane foam property values were clarified. For ride comfort, which is a dynamic property, emphasis was placed on the vibrational properties of the human body, and a clear determination was made that adjusting polyurethane properties improves performance. A method for designing the polyurethane foam was established using these results. Using this development method, it was possible to achieve a lightweight seat with superior seat comfort and ride comfort that produced little sensation of vibration or wobbling.

1. Introduction

A desirable vehicle seat cushion is one that gently supports the passenger, is comfortable to sit on, blocks uncomfortable vibrations, and does not wobble when driving. Also, due to the rise in environmental consciousness in recent years, the demand for a lightweight vehicle with improved fuel efficiency has increased greatly.

Polyurethane foam (PUF) is often used these days as the material in vehicle seat cushions. The technology for forming and manufacturing PUF molds was established around 1960, and Honda applied it to the first-generation CIVIC sold in 1974. Presently, cold-cured (HR) PUF has become mainstream, because its high rebound properties allow it to be made thin and it requires little energy to manufacture.

The general method of decreasing the weight of the seat cushion is to decrease the density of PUF. Although PUF is used in the seat cushions of some light vehicles, etc., it has had difficulty providing seat comfort, which is an aspect of static performance. Meanwhile, high-density PUF is being used to improve ride comfort, which is an aspect of dynamic performance. As a result, low-density, high-performance PUF is required.

In order to achieve such a PUF, first the relationship between PUF properties and a human organoleptic

evaluation of static seat comfort was determined, and then a plan for developing the properties to impart to PUF was established, including a control method for dynamic properties. Next, in achieving PUF properties, various PUF property values, such as the polyurethane formulation and F-S properties were revealed, and a development method for controlling properties according to the cell structure of PUF was determined.

Using this development method, it was possible to develop a lightweight, high-performance PUF for a seat cushion.

2. Organoleptic Evaluation of Static Performance

The indicators of seat comfort, an aspect of static performance, were considered to be the feeling of hardness, wobbling sensation, the bottoming feel, and bouncing sensation. In order to determine the relationship between these organoleptic evaluation indicators and PUF properties, various formulations of urethane were used to create test pieces (TPs) with a broad range of property values, and these TPs were used to conduct a static organoleptic evaluation test. The basic concept is shown in Fig. 1.

Here, Indentation Load Deflection (ILD) hardness is the load necessary to push a disk-shaped TP of $\phi 200$ to

* Automobile R&D Center

25% and 65% of its thickness, respectively. 25% ILD hardness represents the first touch, and 65% ILD hardness represents hardness when the passenger is stably seated. As Fig. 1 shows, when 25% and 65% ILD hardnesses are both low, the cushion is too soft, and when both parameters are high, the cushion is too hard. In addition, when 25% ILD hardness is low and 65% ILD hardness is high, the first touch is soft, and the cushion tends not to feel wobbly due to the rapid rise in load against stroke. By contrast, when 25% ILD hardness is high and 65% ILD hardness is low, the first touch is hard, but the cushion tends to feel wobbly due to the retardation of the rise of load against stroke.

It was considered that in terms of seat comfort, an ideal balance existed between 25% ILD hardness and 65% ILD hardness, and organoleptic evaluations were carried out on that basis.

2.1. Test Sample

The test samples (TPs) that were used were made by Mitsui Chemical, Inc., and their dimensions were 400 × 400 × 100 mm. Polyurethane foaming with a handmixer was used as the method of producing the TPs, and in order to control the 25% ILD hardness to between 180 N and 300 N using an aluminum polyurethane foam mold, the isocyanate index and density were varied in four types of formulation with the mean molecular weight (Mw) of polypropylene glycol (PPG) in a range of 4,000 to 7,000 (Fig. 2).

2.2. Organoleptic Evaluation Test Method

The panelists were 20 males and females weighing between 45 and 95 kg. The study participants were publicly recruited and participated according to their own free will. After these participants were provided with a sufficient written explanation of the study, organoleptic evaluations were performed according to the study method shown in Fig. 3 and Table 1. Also, this test was

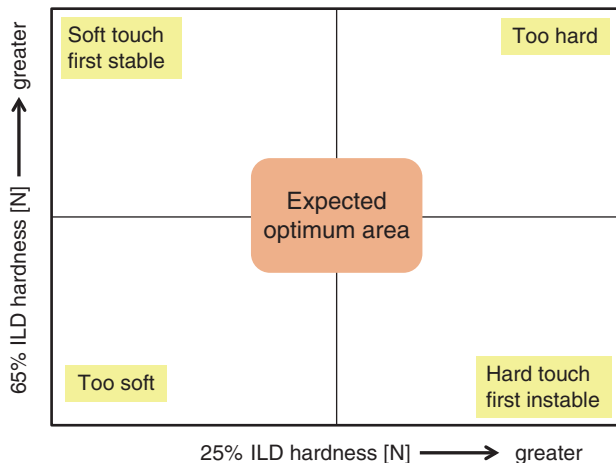


Fig. 1 Material characteristics of polyurethane

Table 1 Procedures of organoleptic evaluation method

Feeling	Test method
Feeling of hardness	Sit still on a seat and make a sensory evaluation on a 6-point scale.*1
Wobbling sensation	Based on urethane test piece equivalent to the reference one, swing upper body back and forth and around to make a sensory evaluation on a 6-point scale.
Bottoming feel	Stand up from urethane test piece and sit down to make a sensory evaluation on a 5-point scale.*1
Bouncing sensation	

*1 (To correct feeling change with the passage of time, compare seating comfort with reference samples and other test piece.)

performed after obtaining the approval of the Honda Bioethics Committee.

2.3. Results of Organoleptic Evaluation Study

2.3.1. Feeling of hardness and ILD hardness

Generally, 25% ILD hardness is used as the indicator for PUF hardness in seat cushion pads for vehicles. In this instance, organoleptic evaluations of PUF were performed by varying PUF properties in various PUF formulations, and a correlation was found between ILD hardness and the feeling of hardness. As Fig. 4 shows, the coefficient of correlation was higher for 65% ILD hardness.

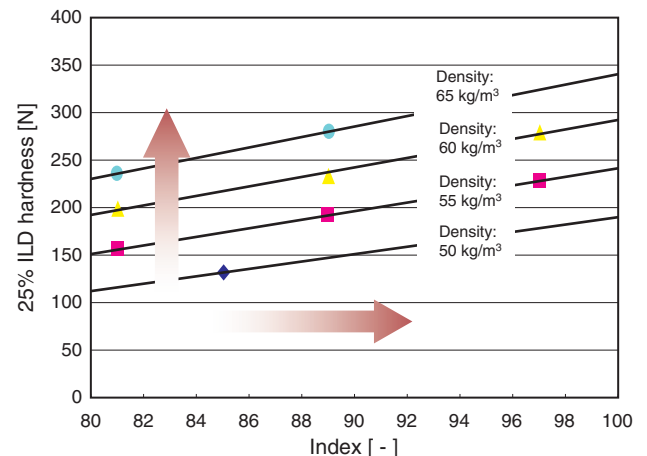


Fig. 2 Hardness controlled under different density and indexes

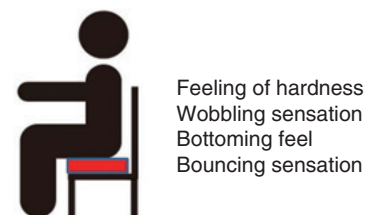


Fig. 3 Organoleptic evaluation method

2.3.2. Sensation of wobbling and sag factor

Figure 5 shows the results of using the TPs to test the correlation of the sag factor, which is the ratio of 25% ILD hardness and 65% ILD hardness. An extremely high correlation was found between the sensation of wobbling and the sag factor. These results show a higher correlation than in Fig. 6, which shows the 25% ILD hardness, 65% ILD hardness, and the sensation of wobbling.

2.3.3. Bottoming feel and sag factor

The correlation between the bottoming feel and the sag

factor is shown in Fig. 7. It was found that when the sag factor is in a range from 2.6 to 3.0, bottoming is difficult to feel. When the sag factor is in excess of 3.0, the load rise in the sag process is felt as the bottom of the seat. When the sag factor is at or below 2.6, the load with respect to sagging does not rise, so the stroke weight with respect to the load increases, and finally the load suddenly rises, causing the bottoming feel.

2.3.4. Bouncing sensation and hysteresis loss rate

A correlation was observed between bouncing sensation

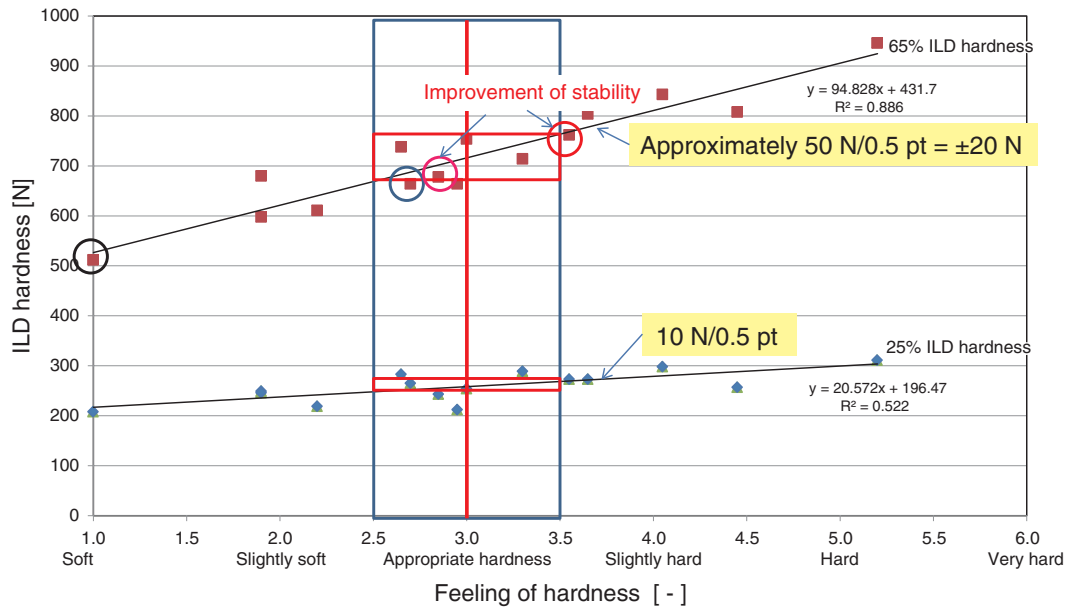


Fig. 4 Feeling of hardness and ILD hardness

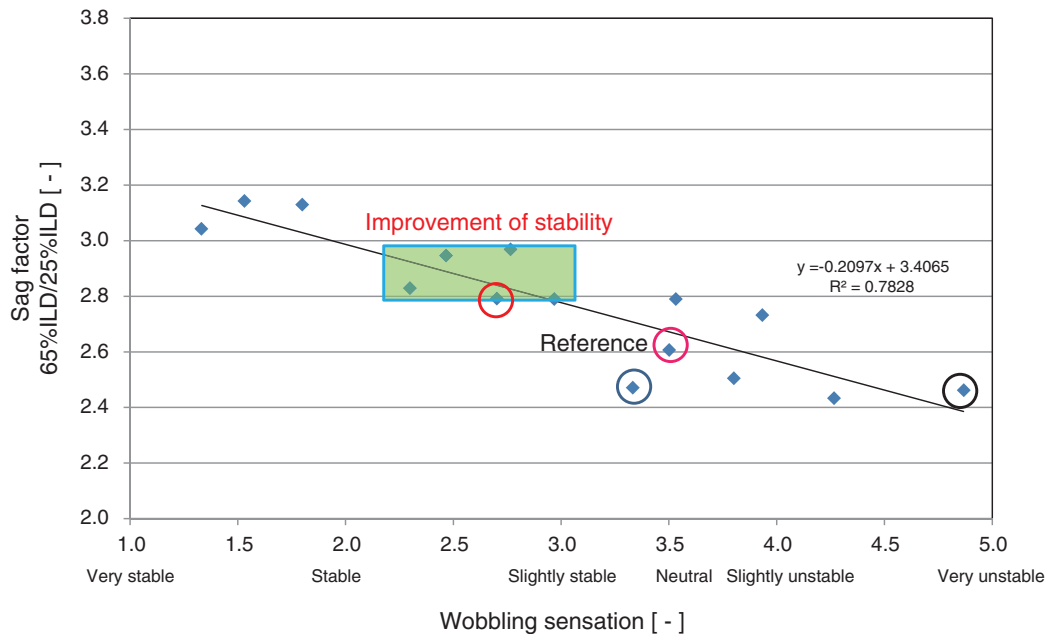


Fig. 5 Relationship between wobbling sensation and sag factor

and the hysteresis loss rate (Fig. 8). The red surface area represents the hysteresis loss when unloading from the compression of F-S properties. It was observed that as the hysteresis loss rate dropped, bouncing was felt more intensely. As for vibration absorption that affects the bouncing sensation, it is known that the heat energy conversion is caused by hysteresis loss of resin as well as the airflow resistance of the PUF cell. With TPs that are produced under the same conditions, when the conditions

for crushing those TPs immediately after foam formation are varied and the TPs are strongly crushed, airflow improves and bouncing sensation heightens.

2.3.5. Comfortable hardness range obtained from hardness feel and wobbling sensation

Comfort level during static sitting at 25% ILD hardness, 65% ILD hardness and static sitting was used as the main indicator for the organoleptic evaluation. As a result of

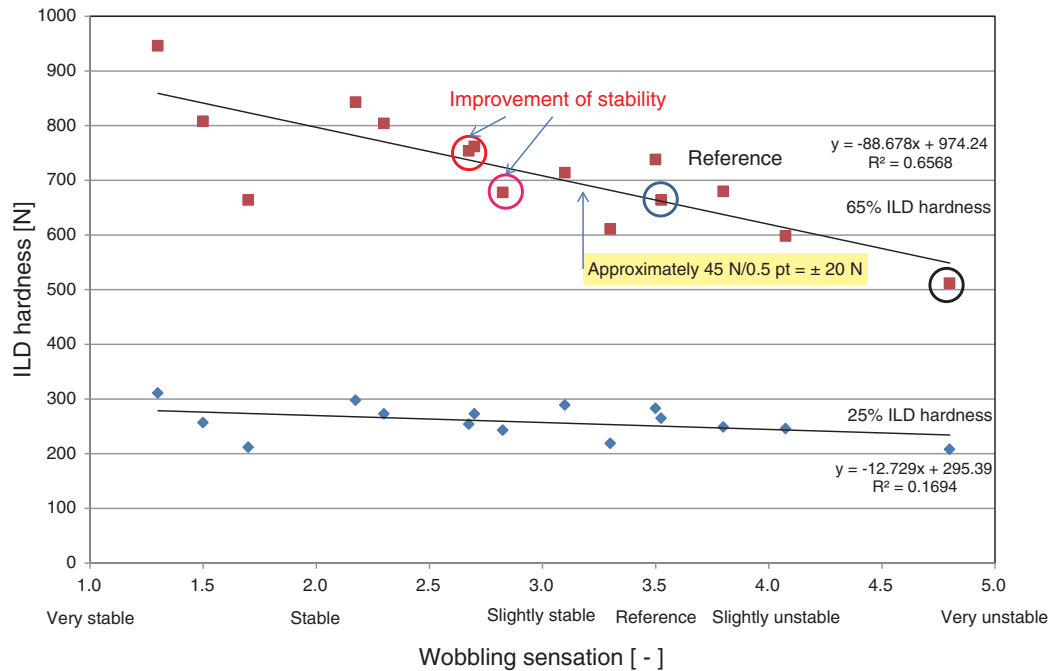


Fig. 6 Wobbling sensation and ILD hardness

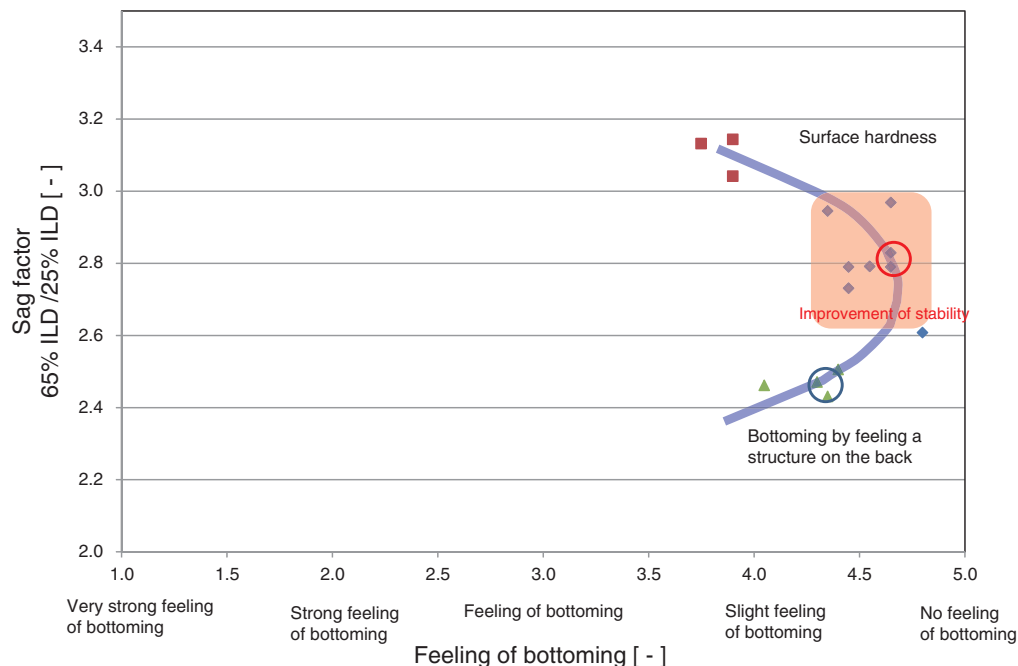


Fig. 7 Relationship between bottoming feel and sag factor

Table 2 Feelings and urethane foam characteristics

	Urethane compression Physical property					Ventilation volume
	25% ILD hardness	65% ILD hardness	Sag factor	Hysteresis loss rate	Impact resilience rate BR%	Ventilation volume
	25% compression Reaction force	65% compression Reaction force	Value obtained by dividing 65% ILD hardness by 25% ILD hardness	Rate of mechanical energy loss during 1 cycle of compression deformation and recovery	Height of an iron ball bouncing back when it is dropped	<Vibration absorption> Approximate energy loss due to ventilation resistance of urethane cell
Feeling of hardness	○	◎				
Wobbling sensation		◎	◎			
Bottoming feel	(○)	(○)	◎			
Bouncing sensation				◎	◎	◎
Comfort level	○	◎	◎			

○ : Good

◎ : So good

interviewing the panelists, it was found that the judgment for comfort level is made based upon the feelings of hardness and wobbling. As shown in Fig. 9, even from the viewpoint of hardness feel and wobbling sensation, perceptions of somewhat comfortable and somewhat uncomfortable often occurred together at the same level of hardness at 25% ILD hardness. However, at 65% ILD hardness, comfort level and hardness could be easily differentiated. For this and other reasons, it was determined that 65% ILD was an effective hardness indicator, in addition to the conventional hardness indicator of 25% ILD.

2.4. Summary of Static Performance and PUF Properties

The correlation between static seat comfort and PUF properties is summarized in Table 2. It is necessary

to increase the efficiency of seat development and the stability of mass-production quality by clearly showing the properties required for improving reproducibility and the optimal ranges within which to control those properties.

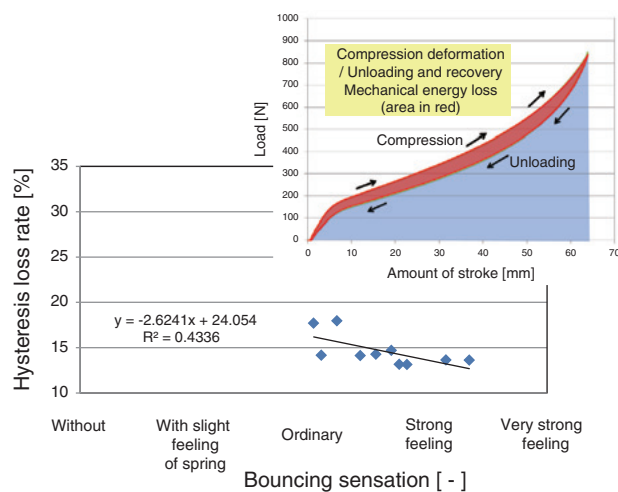


Fig. 8 Relationship between bouncing sensation and hysteresis loss rate

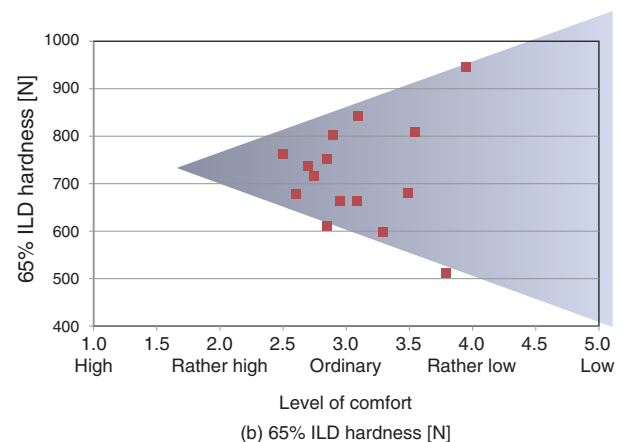
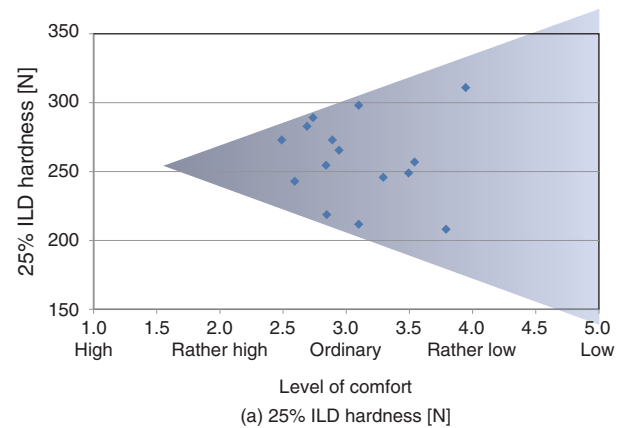


Fig. 9 Hardness of polyurethane test piece and level of comfort

3. Dynamic Performance Evaluation

3.1. Dynamic Performance and Resonance Frequency of Human Body

The measurement results of the input acceleration from the vehicle body during actual vehicle traveling are shown in Fig. 10. Peaks exist from 1 Hz to 2 Hz, 4 Hz to 8 Hz, and 10 Hz to 20 Hz. Vibrations from the vehicle body are transmitted from the seat frame through the urethane pad to each part of the human body. These vibrations are transmitted to the human body as the following sensations: bobbing up and down at 2 to 4 Hz; pressure on the abdomen at 4 to 8 Hz; and roughness, trembling, and vibrating sensations to the thighs at 8 Hz and above.

In addition to blocking vibrations from the vehicle body and reducing uncomfortable vibrations at or above 4 Hz, which are easily perceived by the human body, it is also important to reduce resonance in order to minimize car sickness and fatigue.

3.2. Dynamic Performance and PUF Properties

Vibrational input from the vehicle body being received by a human sitting on a urethane pad is shown in a simple model (Fig. 11). The resonance frequency f_0 Hz changes

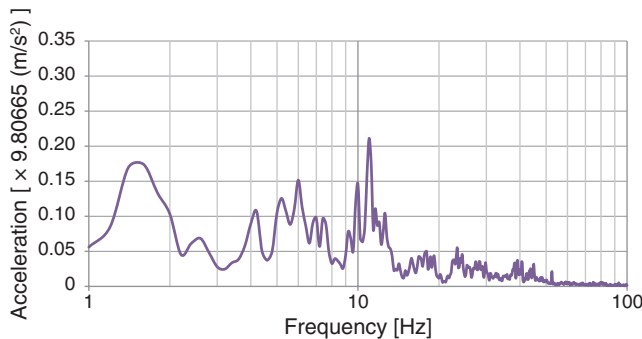


Fig. 10 Input acceleration from body (Attachment point of front seat)

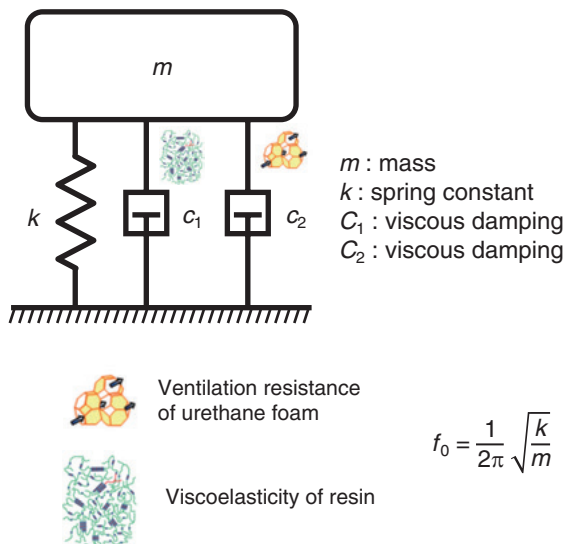


Fig. 11 Viscosity mass damper with spring model of PUF

according to the spring constant, and decreasing the spring constant when a person is sitting in the seat lowers the resonance frequency, as shown in Fig. 12.

From the perspective of ride comfort, which is an aspect of dynamic performance, it is preferable to lower the PUF resonance frequency, shifting to the low-frequency range, and decrease the vibrations transmissibility in all ranges by decreasing the resonance magnification.

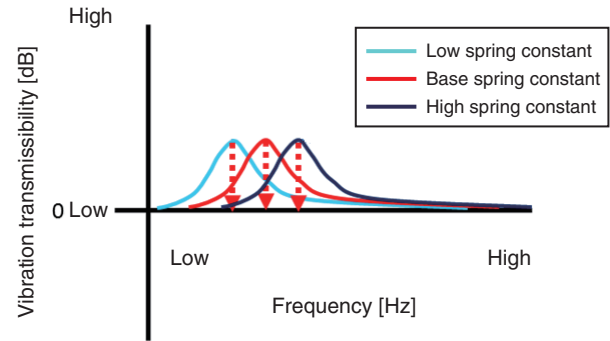


Fig. 12 Movement by spring rate of resonance frequency

4. Design Method

The correlation between the urethane resin design and the PUF cell structure plays an important role in the vehicle seat. This correlation is summarized with respect to the sag factor and vibration blocking, which affects wobbling sensation, which greatly affects body stability.

4.1. ILD Control

ILD is highly correlated with the feeling of hardness. Generally, ILD is adjusted to 25% by adjusting the isocyanate index and density. However, it is difficult to use this method to control the desired values of both 25% ILD and 65% ILD. The PPG molecular weight and polymer polyol (POP) quantity are important in designing hardness in the optimal property region shown in Fig. 1.

4.2. Sag Factor and PPG Molecular Weight

As Fig. 13 shows, the molecular weight of polyols has a significant effect on the sag factor, which in turn significantly affects the wobbling sensation. Figure 14 shows a schematic diagram of polyurethane resin. Phase separation between PPG and amino bases tends to occur more in polyurethane resin that uses high-molecular-weight PPG as opposed to low-molecular-weight PPG. When phase separation occurs, the hard segment formed by isocyanate and a chain-extending agent increase, causing initial elasticity to be higher. Also, the soft segment comprising mainly PPG has a high-molecular-weight chain that easily extends. Viewed from the standpoint of PUF properties, these are F-S properties in which it is difficult for a load to stand even at a high stroke.

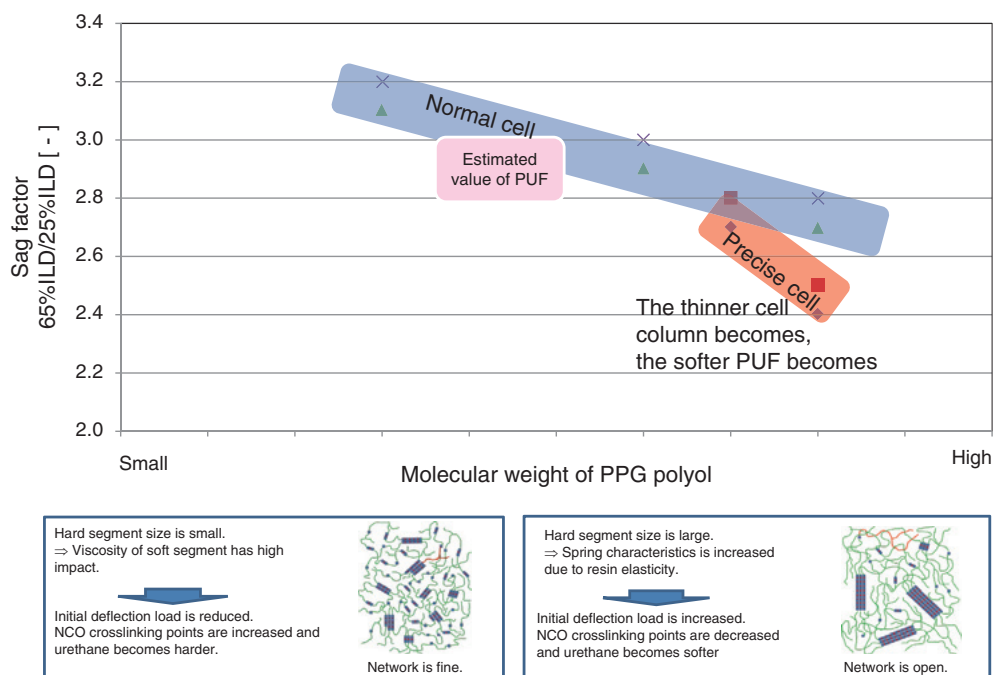


Fig. 13 Sag factor and molecular structure

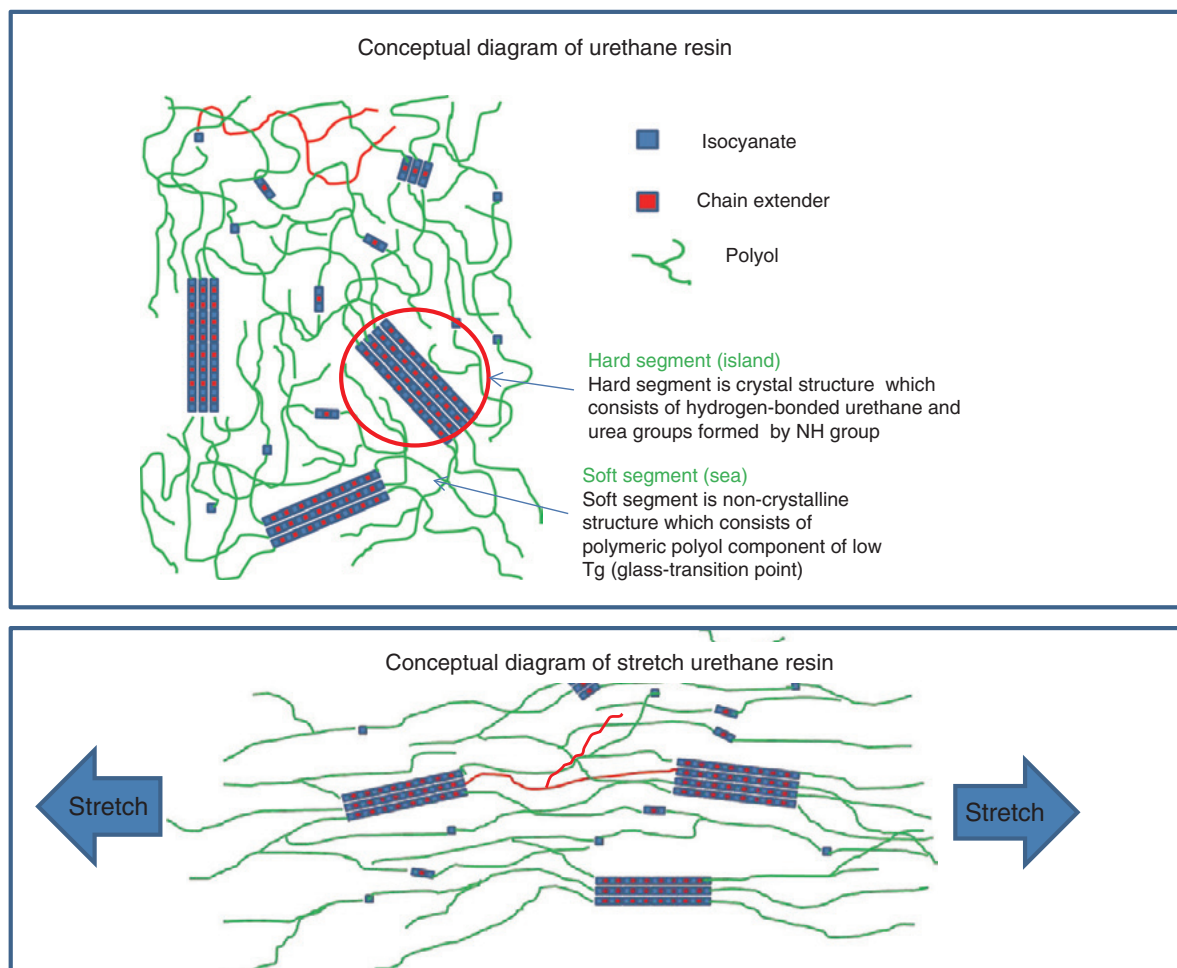


Fig. 14 Schematic diagrams of polyurethane

4.3. Combination of Dynamic Performance and Wobbling Sensation

As shown in Fig. 13, when viewed from the perspective of urethane resin design, there is a large effect caused by the molecular weight of PPG, which is a component of polyol, and it was found to be possible to lower the resonance frequency by using high-molecular-weight PPG. However, if the value of the spring constant k is decreased by simply using high-molecular-weight PPG, the resonance frequency will decrease but the resonance magnification will increase, as shown in Fig. 15. As Fig. 13 shows, high-molecular-weight PPG decreases the sag factor, and when the sag factor decreases, it becomes easy to sense wobbling. Therefore, it is important to find a balance between the sag factor and the sensation of wobbling. When simply changing the PPG to a high-molecular-weight PPG, airflow increases as a result of changing to open cell, and resonance magnification increases. By optimizing the type and weight of the silicon-based foam stabilizer, the airflow

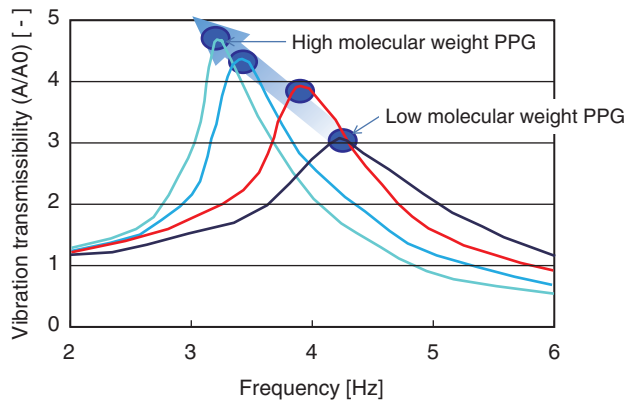


Fig. 15 Molecular weight and vibration transfer characteristics of PPG

resistance is increased in a compact cell structure and the resonance magnification is decreased. Airflow rate and resonance magnification are related as shown in Fig. 16. It was discovered that by setting the cell size and crushing conditions to yield the optimum airflow rate, the resonance magnification is reduced, making it possible to achieve a balance with resonance frequency. The same change in resonance frequency and change in sag factor were observed even when the PPG and POP formulation ratio was changed.

4.4. Summary of PUF Design Method

In terms of static performance, this analysis showed a correlation between the sag factor, which greatly impacts wobbling sensation, and the molecular weight of PPG (Fig. 13) and a correlation between the hysteresis loss rate and the molecular weight of PPG (Fig. 17). In addition to the conventional 25% ILD hardness management based on the density and isocyanate index, it was discovered that it was important to specify a hysteresis loss rate for which it is possible to specify the molecular weight of PPG, which

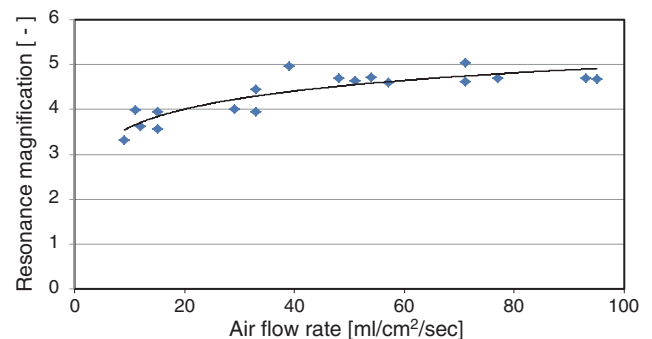


Fig. 16 Air flow rate and resonance magnification

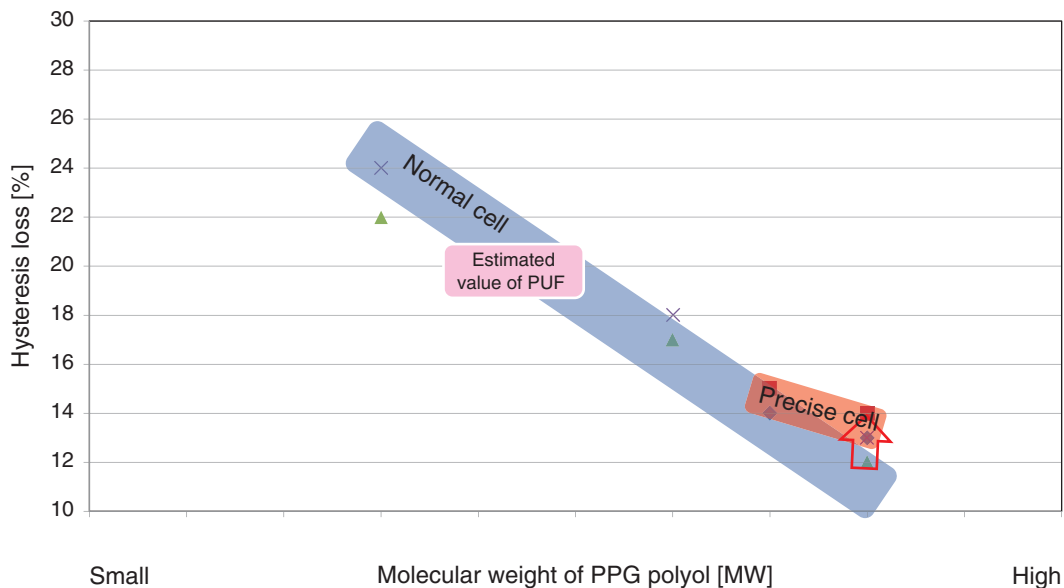


Fig. 17 Hysteresis loss rate and molecular structure of PPG

Table 3 Specifications of flexible PUF (400×400×100 mm)

		Unit	Development	Reference existing mass-produced low repulsion urethane	Reference high density product urethane
Density	Dcore	kg/m ³	63	63	83
Hardness	25%ILD	N	250	245	250
	65%ILD	N	730	700	675
Hysteresis loss rate		%	<15	19	14
Repulsive elastic core		%	>65	65	68
Resonance frequency		Hz	3.3	3.6	3.3
Resonance magnification (A/A0)		-	2.6	2.6	2.8
6 Hz transmissibility (A/A0)		-	0.6	0.8	0.65
10 Hz transmissibility (A/A0)		-	0.2	0.3	0.28

greatly affects the sag factor of the 65% ILD hardness and wobbling sensation indicator that is the comfortable hardness indicator.

Also, in terms of dynamic performance, a high correlation was observed between the resonance frequency and hysteresis loss rate (Fig. 18) and between the hysteresis loss rate and the molecular weight of PPG (Fig. 17). Also, it was discovered that the molecular weight specification of PPG required for obtaining the desired resonance frequency could be substituted with the hysteresis loss rate. This indicated that it would be possible to regulate the limit for the molecular weight of PPG, the main component of polyols, and major aspects of performance such as the resonance frequency of the urethane vibration characteristic by specifying the hysteresis loss rate. Also, the resonance magnification could be estimated by specifying the airflow rate.

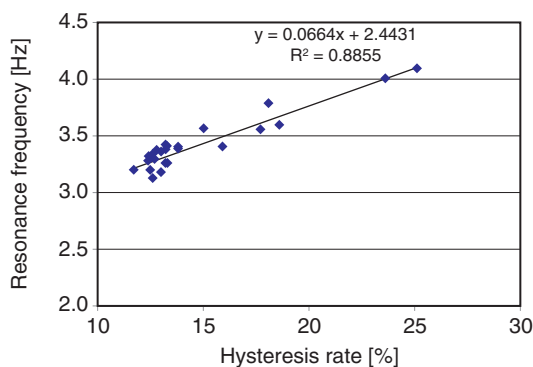


Fig. 18 Hysteresis loss rate and resonance frequency

5. High-performance Seat Development

5.1. Goals

In developing PUF material for vehicle seats, we developed a PUF (“developed PUF”) with new properties

having the following four main goals:

- (1) An appropriate hardness feel
- (2) A superior vibration-damping property
- (3) A wobbling sensation equivalent to that of the conventional PUF
- (4) Reduced weight

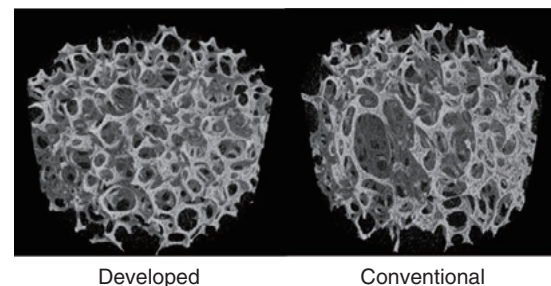
5.2. Developed PUF Characteristics

The main specifications of the developed PUF are shown in Table 3. The developed PUF is soft with a core density of 63 kg/m³, which is the same as mass-produced, low-density urethane. The developed PUF has a low resonance frequency and reduced vibration transmissibility at all frequencies. Also, it has a seat comfort at the same level of the 80 kg/m³ high-density PUF, a low wobbling sensation, an appropriate bouncing sensation, and superior vibration-damping property.

Fig. 19 shows an X-ray microtomography image of the developed urethane and the current mass-produced, low-density urethane. The developed urethane has finer and more uniform cells compared to the mass-produced, low-density urethane, and it has a structure with greater airflow resistance between cells.

5.3. Concept of the Developed PUF Design

The urethane resin of the developed PUF uses high-

Fig. 19 X-ray Microtomography $\phi 3$ mm 3D View

molecular-weight PPG of a low monool and has a low hysteresis loss rate. It achieves a balance between the wobbling sensation and vibration properties. Specifically, the resonance frequency is reduced by using a high-molecular-weight PPG to decrease the bouncing sensation. Also, the type and weight of the silicon-based foam stabilizer are optimized, resulting in a compact urethane cell structure and optimized crushing quantity. This improves the vibrational absorption performance of the air damper, minimizing resonance magnification and improving the 6-Hz and 10-Hz vibration-damping property.

Figure 20 shows the F-S properties of typical urethane. The developed urethane has vehicle cushioning urethane that is neither too soft nor too hard and causes little wobbling sensation. Also, the developed urethane was similar to the MDI urethane used in Europe, and the 65% ILD hardness value was found to be almost the same.

When urethane resin is designed using low-molecular-weight PPG at the identical 25% ILD hardness, the slope of the F-S properties increases and the PUF becomes hard, causing the bottoming feel. This is illustrated by the red-purple line in the diagram. Meanwhile, when high-molecular-weight PPG is used, the F-S properties slope flattens out, as illustrated by the blue line in the diagram. The PUF is soft and tends to allow a wobbling sensation. Although this PUF has extremely excellent vibration-damping property, the wobbling sensation is too great in the actual seat. Therefore, countermeasures are required.

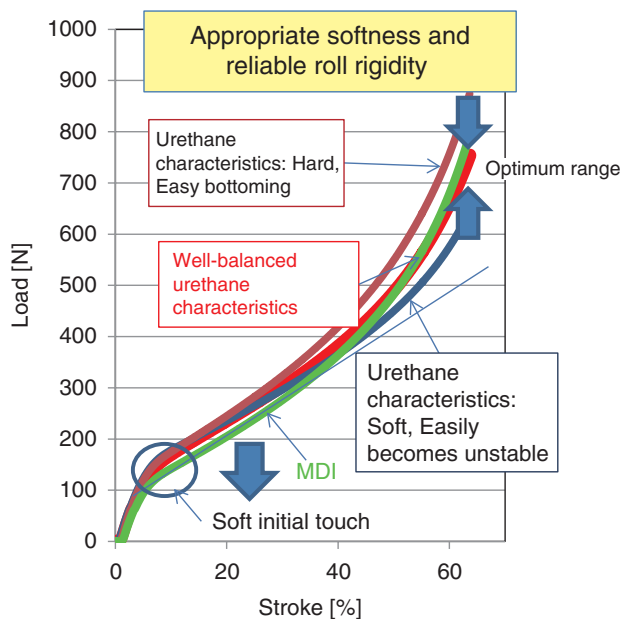


Fig. 20 Compression load characteristics of flexible PUF 400×400×100 mm

Table 4 Specifications of trial seat

Cushion frame	Pan frame
Cushion pad thickness (HP)	105 mm
PUF density (core)	63 kg/m ³
Hardness 25%ILD (Pad)	245 N
Hardness 65%ILD (Pad)	790 N

Figure 21 shows the vibration properties when 100-mm-thick PUF is used. The red line in the diagram represents the developed product. Compared to the mass-produced vibration absorption urethane shown in the black line in the diagram, the developed product exhibits a lower resonance frequency and a lower resonance magnification. Also, the developed product has lower vibration transmissibility in the range of 4 to 8 Hz, which is the range that is easily perceived by the human body. The blue line represents the PUF property before countermeasures were taken for wobbling sensation. Although the vibration-damping property is extremely high, countermeasures were required due to high wobbling sensation in the actual seat.

5.4. Organoleptic Evaluation of Developed PUF Seat

Experts conducted an organoleptic evaluation of the developed PUF in a test course drive by installing a test seat having the specifications shown in Table 4 in a vehicle. The results obtained from that evaluation are shown in Table 5. A well-balanced seat was achieved by improving of softness, vibration attenuation, and rolling stiffness (wobbling sensation). These aspects were improved by changing the molecular weight of the PPG in the PUF in order to increase the bouncing rate and thus decrease the load during low stroke.

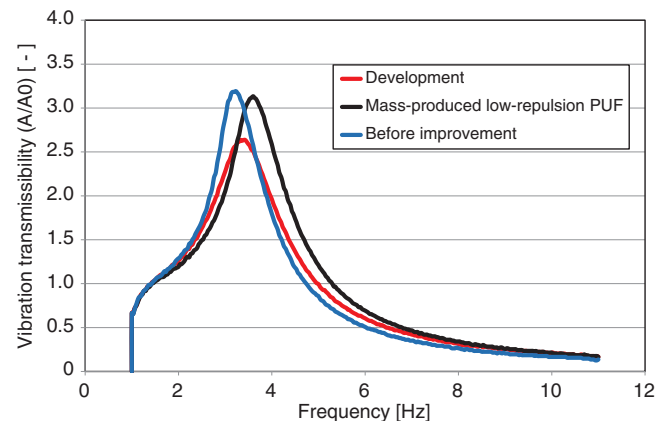


Fig. 21 Vibration transmissibility of PUF 400×400×100 mm

Table 5 Organoleptic evaluation of test track driving

	Structure with spring				Full pan structure			
	Mass-production foam		Foam with formulation to reduce wobbling sensation	High resilience and high-vibration absorption foam after reducing wobbling sensation		High resilience and high-vibration absorption foam after reducing wobbling sensation	High resilience and high-vibration absorption foam before reducing wobbling sensation - baseline	
Soft-touch feeling	- Felt	-0.5 x	- Slight permanent set - Felt	-1 x	- Felt	+1 ○	- Felt	+1 ○
Damping feeling	- Choppy road (US) 80 - 100km/h large vertical motion - Vibration not absorbed when suspension fully stroked.	± 0	- Choppy road less vertical motion - Vibration absorbed when suspension fully stroked - Unsteadiness of spring hardly felt	+1 ○	- Choppy road less vertical motion - Vibration absorbed when suspension fully stroked - Slight stroke observed and sufficient thickness of foam felt	+1 ○	- Choppy road less vertical motion - Vibration absorbed when suspension fully stroked - Slight stroke observed and sufficient thickness of foam felt	+1 ○
Rolling stiffness feeling	- Rolling observed when turning corner	± 0	- Movement of spring felt but rolling stiffness sufficient	+1 ○	- Good firmness	+1.5 ◎	- Good firmness	+1.5 ◎
Shaky feeling	- Hardly felt	± 0	- Not felt other than at belly	-0.5 x	- No	± 0	- No	± 0
Fluffy feeling	- Felt	-0.5 x	- No	± 0	- No	± 0	- No	± 0
Rebound feeling	- Slightly felt	-0.5 x	- No - Felt thinner than actual	-1 x	- Firmness, felt thicker than actual	+0.5 ○	- Firmness, felt thicker than actual	+0.5 ○
Bottoming feeling	- No	+1 ○	- No - Slight vertical acceleration felt at suspension bottoming	+0.5 ○	- No - Right amount of stroke, a sense of fullness	+1.5 ◎	- No - Right amount of stroke, a sense of fullness	+1.5 ◎

◎ : So good ○ : Good x : Poor

6. Summary

A PUF with excellent properties for a vehicle seat cushion was developed. A soft PUF that wobbled little and maintained posture well was achieved by determining the optimal region for the molecular weight of the high-molecular-weight PPG and the POP formulation ratio. Optimizing the type and weight of the high-molecular-weight PPG and silicon-based foam stabilizer and optimizing the crushing conditions allowed vibrations that are uncomfortable to the human body to be blocked.

This developed PUF provides superior seat and ride comfort not only when using a pan-frame seat, but also when using a bouncing-frame seat.

The following findings were simultaneously made during this research:

- (1) The important PUF properties to control that reproduce the organoleptic evaluation indicators are density, 25% ILD hardness, 65% ILD hardness, hysteresis loss rate, rebound elasticity rate, and airflow.
- (2) The molecular weight of PPG and the POP formulation ratio greatly affect PUF properties, and the effect is particularly large on the sag factor, which represents load sagging, and on the hysteresis loss.
- (3) In vibration blocking, the resonance frequency is reduced by switching to a high-molecular-weight PPG. The resonance magnification is lowered by optimizing the silicon-based foam stabilizer type and quantity as well as the crushing conditions. There are also conditions that minimize the transmission rate of vibrations that are uncomfortable to the human body.
- (4) The properties required for a urethane resin design that greatly affect seat and ride comfort were clarified, and PUF reproducibility was improved.

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■ Author ■



Naoto ISHII



Tomoki MATSUMOTO



Tsutomu AO

Life Design and Evaluation Method of Type 3 High-Pressure Hydrogen Tank for Fuel Cell Electric Vehicle

Toshihiko KANEZAKI*

Satomi MANO*

Kazuo MIYAGAWA*

Noboru HAYASHI*

Takeshi OGAWA**

ABSTRACT

Failure in Type 3 high-pressure hydrogen tanks occurs as a result of the growth of cracks in the boss of the liner or the cylinder until the liner is penetrated. The boss of the liner is prone to exposure to corrosive environments, for example by being splashed with water during driving. For this reason, it is essential to evaluate the life of a tank until a leak occurs with consideration of the corrosive environments in an actual vehicle. A life design method based on the mechanism of failure of the boss, in which evaluations are conducted with consideration of stress corrosion cracking and sustained load cracking, was therefore developed. The correlation between the results produced by the method and the results of tests using a Type 3 high-pressure hydrogen tank was examined, demonstrating the validity of the method. This paper also discusses a method of evaluation of Type 3 high-pressure hydrogen tanks that considers their fatigue characteristics in the type of corrosive environments occurring in actual vehicles.

1. Introduction

Development projects and studies are proceeding, centering on the automotive industry, towards the introduction of fuel cell electric vehicles to the market. Hydrogen storage systems are an important element technology of fuel cell electric vehicles. Today, the main focus is on storage systems using 70 MPa hydrogen tanks that make it possible to drive further on a single hydrogen refill, and which make the storage system more compact due to their increased storage density. The main components of high-pressure hydrogen tanks for automotive use are a liner, which functions as a hydrogen gas barrier layer, and a carbon fiber-reinforced plastic (CFRP) layer, which reinforces the outer surface of the liner. When the liner is metal (mainly aluminum alloy), the tank is classified as a Type 3 high-pressure hydrogen tank (Type 3 tank) and when the liner is plastic, the tank is classified as a Type 4 high-pressure hydrogen tank (Type 4 tank) (Fig. 1). Both types of tank are large and high-priced for automotive components, and the reduction of size and cost therefore represent issues.

Weight, price, filling rate, safety and other tank characteristics differ between Type 3 and Type 4 tanks, and tank design methods are also different. This makes it necessary to develop evaluation methods suited to a tank's

type and failure mechanism. Failure in Type 3 tanks occurs as a result of the growth of cracks in the aluminum alloy liner, which eventually penetrate the liner. Type 3 tanks are designed with consideration of crack growth due to fatigue in order to achieve weight reductions (i.e., designed in the finite life range). In addition, due to the way that high-pressure hydrogen tanks are installed in vehicles, there is a strong possibility that a corrosive environment will be created, for example due to exposure of the boss of the tank to water during vehicle operation. Based on the above, it is necessary to evaluate the durability performance of a Type 3 tank with consideration of the corrosive environment in an actual vehicle.

Turning to issues in evaluation methods for the

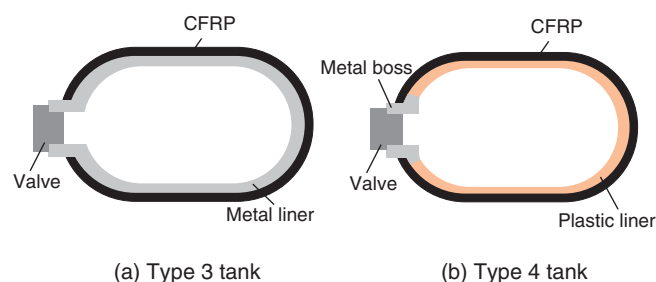


Fig. 1 Structural schematics of Type 3 and Type 4 tanks

* Automobile R&D Center

** Aoyama Gakuin University

corrosion resistance of high-pressure hydrogen tanks as defined by regulations and standards, in the second half of the 1990s, several incidents of failure of scuba cylinders (Type 1) manufactured from A6351-T6 aluminum alloy occurred^{(1), (2)}. In Japan, the cause of these failures was interpreted as Stress Corrosion Cracking (SCC), while overseas the cause was considered to be Sustained Load Cracking (SLC). In an attempt to prevent these incidents, a high-pressure container regulation was formulated that prohibits the use of excess silicon 6XXX series aluminum alloys, which have a yield strength above 250 MPa⁽³⁾. In addition, Annex B of ISO 7866 stipulates SLC tests to help ensure that even when a high-pressure tank remains under load in the use environment for an extended period, failure will not occur due to the crack growth. However, based on reports that in the case of 7XXX series aluminum alloys SLC characteristics are dependent on the test environment (the water content in the test environment)^{(4), (5)}, SLC tests, which are evaluations conducted in the air without specifying humidity, cannot be considered an appropriate evaluation method for high-pressure tanks for automotive use. While the intercrystalline corrosion tests and SCC tests specified in Annex A of ISO 7866 are tests for the evaluation of the occurrence of corrosion and crack growth in aluminum tanks in saltwater environments, they involve passing or failing the material. Because of this, the evaluative tests specified by ISO 7866 are not able to judge the severity of the corrosion environment or the life design of the product. A method of evaluation of stress corrosion cracking in aluminum alloys that considers the corrosion environment and life design is necessary.

A life design method able to adequately evaluate the boss of a Type 3 tank, which could be assumed to be exposed to the exterior environment and damaged by SCC and SLC, was therefore developed. In addition to the development project, this paper will also discuss the results of tests conducted to verify the effectiveness of the method. As part of the process of examination towards reducing the weight and the cost of Type 3 tanks in the future, the paper will also present an analysis of the potential for the high-strength aluminum alloy A6066, which is also the subject of a NEDO project⁽⁶⁾, for use in the liners of Type 3 tanks.

2. Regulations and Standards Related to High-pressure Hydrogen Tanks and Design Concept

2.1. Related Regulations and Standards and Methods of Evaluation

This section will deal with technological standards related to high-pressure hydrogen tanks. Looking at domestic regulations first, the technical standard for containers of compressed hydrogen employed as vehicle fuel devices (KHK S0128)⁽⁷⁾, dealing with devices with a maximum filling pressure of 70 MPa, was issued in 2013. This regulation restricts the metals able to be used in the

tanks to aluminum alloy A6061-T6 and stainless steel SUS316L, the fatigue strength of both of which has been evaluated in high-pressure hydrogen gas⁽⁸⁾. Turning to the situation overseas, Global Technical Regulations (GTR) for fuel cell electric vehicles are presently under examination⁽⁹⁾. The formulation of the GTR has been divided into two phases. In Phase 1, issues including basic evaluation methods for the tanks were deliberated and compiled. Time was required to achieve a consensus between the participating countries regarding the details of materials evaluations, and deliberation of these points is continuing in Phase 2. The details of the GTR are important, given that it can be assumed that if a consensus is achieved, domestic standards will come into line. It is possible that the GTR evaluation standards might relax restrictions on materials and structure for conforming tanks (allowing the use of materials other than the A6061-T6 and SUS316L specified by domestic regulations).

As indicated above, no consensus concerning evaluation methods of the hydrogen compatibility of metals was reached in Phase 1 of the GTR, and discussions are continuing in Phase 2. In addition, accurate evaluation of corrosion fatigue strength and stress corrosion cracking in saltwater environments will also be important in attempting to ensure the soundness of metal materials used in automotive components.

At present, judgment of the compatibility of aluminum alloys for use in high-pressure tanks in terms of corrosion is based on Annexes A and B of ISO 7866⁽¹⁰⁾ and similar regulations. Annex A specifies intercrystalline corrosion tests that determine environmental sensitivity based on the depth of intercrystalline corrosion, and SCC tests, which evaluate stress corrosion cracking via saltwater wetting and drying cycles. These can be considered to be tests for the evaluation of the occurrence of corrosion and crack growth in aluminum alloys when exposed to saltwater environments. Annex B specifies the SLC tests discussed in the previous chapter. However, the evaluation methods specified by ISO 7866 assume aluminum alloy high-pressure tanks that are not reinforced with CFRP (Type 1 tanks). In Type 3 tanks, the outer surface of the aluminum alloy liner is covered by CFRP, and the status of occurrence of corrosion and the areas that are exposed to a corrosion environment differ from Type 1 tanks. Because of this, in addition to the evaluation methods of hydrogen compatibility described above, the evaluation of the aluminum alloys used in Type 3 tanks necessitates evaluation methods of corrosion resistance that adopt a fracture mechanics perspective suited to composite containers.

2.2. Concept of Safety Design

The concept of “leak before burst” (LBB) is part of the thinking regarding safety design for high-pressure equipment and high-pressure tanks^{(11), (12)}. This concept seeks to prevent the bursting of high-pressure equipment

and tanks, which might result in terrible accidents, by promoting leaking of the internal gas prior to bursting when unforeseen events occur. In concrete terms, judgments are first made for each part of the tank regarding whether it will be able to fulfill LBB criteria. For parts that do fulfill the criteria, life design based on an S-N curve expressing the relationship between loaded stress and life is employed. For the parts that do not, life calculations are performed based on crack growth analyses, and design attempts to ensure a life that is sufficiently longer than the parts that fulfill the LBB criteria.

In Type 3 tanks, the aluminum liner normally fails due to fatigue cracks developing and penetrating the liner. Despite this, because the liner is covered by CFRP, the liner leaks rather than bursts. Given this, Type 3 tanks basically fulfill the LBB criteria. Finite life design based on the relationship between the stress occurring in the liner and its life (an S-N curve) is therefore used for these tanks. However, the boss is not completely covered by the CFRP, and does not fulfill the LBB criteria. For example, the existence of a defect at thread root in the boss of a size that has not been considered in the design process may be hypothesized. In this case, if a crack grows from the defect and penetrates the boss, the failure mode of the boss will be bursting. Structurally, therefore, the boss does not fulfill the LBB criteria. In order to help prevent bursting of the boss, the existence of an initial defect of a certain size in the boss is assumed from a design perspective, and the life of crack growth in the boss is designed to be sufficiently longer than the life before leaking of the parts that fulfill the LBB criteria. By this means, the tank as a whole needs to meet the LBB criteria.

3. Life Design and Test Method for the Boss

3.1. Life Design Method

The bosses of hydrogen tanks can be exposed to corrosive environments, for example by becoming wet during vehicle operation. It has been reported with regard to SLC in aluminum alloys that the higher the humidity of the environment to which the alloy is exposed, the lower its resistance to crack growth^{(4), (5)}. Because of this, it is necessary for the life design of the boss to consider SLC in a corresponding corrosion environment. Figure 2 shows the

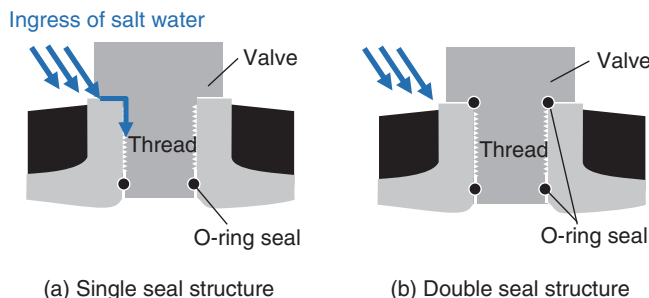


Fig. 2 Schematics of seal structure

structure of a boss seal. Because of the seal structure, the threads are exposed to a different environment. The research discussed in this paper focused on life design of the structure shown in Fig. 2(b), which seals the external surfaces of the threads to help prevent infiltration of saltwater and other corrosive agents. The structure shown in Fig. 2(b) helps to prevent water containing corrosive ions from reaching the thread section of the liner boss from the exterior, making the environment to which the threads are exposed the air. The threshold stress intensity factor at which SLC occurred (the SCC threshold stress intensity factor, K_{ISCC}) was therefore measured in humid air, a more severe environment for SLC, and this measurement was reflected in the life design. The high-pressure container industry treats the phenomenon of crack growth in aluminum alloys under long-term loads as SLC, but this is generally interpreted as SCC. This paper makes no distinction between the detailed mechanisms of SLC and SCC, and treats crack growth under static loads in a corrosive environment as SLC, which it evaluates using K_{ISCC} . Taking SLC into consideration, the life design method for the tank boss is as follows (Fig. 3):

- (1) A limit dimension for a defect that could be detected in a non-destructive inspection in the manufacturing process is projected, and the initial defect dimension for the design is set on this basis. This dimension is treated as the initial crack length.
- (2) The threshold stress intensity factor for SCC in humid air, K_{ISCC} , is measured.
- (3) The fatigue crack growth rate in humid air is measured.
- (4) The life of fatigue crack growth is analyzed.
- (5) The design is adjusted to help ensure that the stress intensity factor of cracks in the boss does not reach K_{ISCC} during the designed life of the tank.

3.2. Initial Crack Length

When setting the initial crack length for design purposes, employing a larger figure than the minimum figure able to be detected in a non-destructive inspection of thread root in the tank boss makes for a safer design. For the analysis of fatigue crack growth conducted in this research, a circumferential defect of 2 mm in depth was assumed. This could be seen with the naked eye, and was assumed to be a

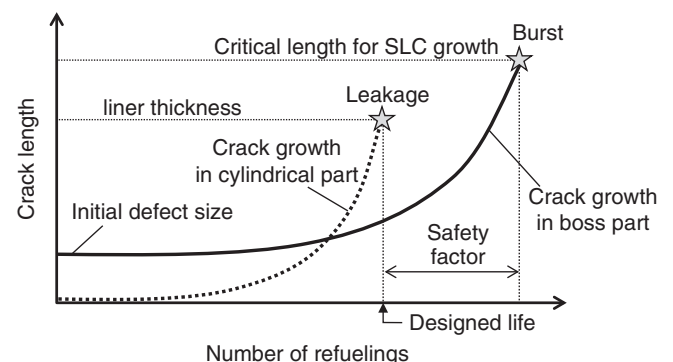


Fig. 3 Schematic of fatigue crack growth life design

defect of a size that could not occur in the actual machining process.

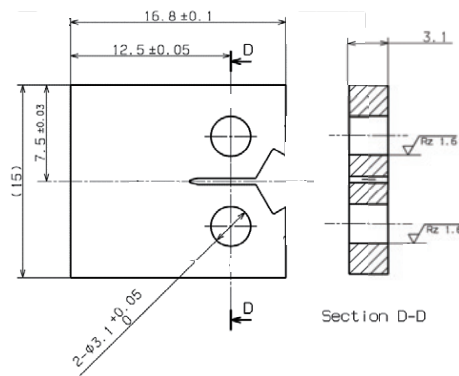
For a crack initiation from a defect, several numbers of cyclic stress load of a certain level are necessary. Normally, when dealing with metal fatigue, the number of cycles to crack initiation is approximately 10% of fatigue life, and the number of cycles for the crack growth to failure represents the greater part of the fatigue life of the product. Given this, for the design life calculations conducted in this research, the number of cycles until the crack initiated from the defect was ignored, and the initial defect was regarded as the crack.

3.3. Measurement Method of Crack Growth Data

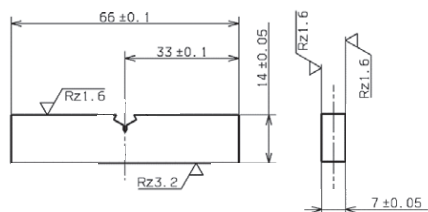
Crack growth tests were conducted in order to measure crack growth rate of material in the tank boss. These tests used a CT specimen taken from the boss of an A6066 aluminum alloy liner. Figure 4(a) shows the shape of the CT specimen. This shape conforms to ASTM E647⁽¹³⁾. The test was conducted in humid air, with a relative humidity of more than 90% at room temperature (25°C). The initial stress intensity factor range for the test was set at $\Delta K = 6 \text{ MPa}\sqrt{\text{m}}$, and the decreasing method and the increasing method were applied. The methods used for procedures such as measurement of crack length and calculation of the stress intensity factor were based on ASTM E647. The stress ratio, R , was 0.1, and the test frequency was 5 Hz to 20 Hz.

3.4. Method of K_{ISCC} Measurement

In order to determine K_{ISCC} , three-point-bend test specimens of the shape shown in Fig. 4(b) (conforming



(a) CT specimen for crack growth test



(b) 3-point bend specimen for measuring K_{ISCC}

Fig. 4 Size and shape of specimens

to ASTM E399⁽¹⁴⁾) were taken from close to the boss of an A6066 aluminum alloy liner and an A6066-T6 plate for liner use. In addition, in order to provide a reference to help determine the difference between the fracture surface of the fatigue pre-crack and the fracture surface of the SLC, test specimens were also created from A6351-T6, which has a similar excess silicon composition to A6066 and is considered to be prone to SLC.

Based on ISO 7359-6⁽¹⁵⁾, K_{ISCC} was determined by observing whether there had been any crack growth in the pre-cracked specimens following exposure to the environment. The load was applied so as to impart a specific displacement. The conditions for exposure to the environment were a temperature of 25°C, a relative humidity of more than 90%, and a period of exposure of 90 days. In order to study the behavior of crack growth under sustained loads, tests were also conducted in which the exposure period was reduced to 10 days and 30 days. Following exposure, the load on the test specimen was measured. Fatigue tests were then conducted in order to propagate cracks for observation of the fracture surface. In the period necessary for the cracks to newly grow by at least 1 mm, care was taken to attempt to prevent the maximum stress intensity factor from exceeding 0.6X the stress intensity factor when the load was initially applied prior to exposure. The crack growth during exposure was examined by observing the fracture surfaces with a scanning electron microscope (SEM). Crack lengths were measured in the direction of thickness of the test specimen, at 25%, 50% and 75% of the thickness of the specimen. The average of these three figures was regarded as the crack length. Based on the loads measured following exposure to the environment and the crack lengths measured by SEM observation, the stress intensity factor when the test specimen was under load was recalculated. Using the results of these measurements of crack length, the maximum value of the stress intensity factor among load conditions in which crack growth was not observed was taken as K_{ISCC} , the threshold stress intensity factor for SLC.

3.5. Relationship between Crack Length and Stress Intensity Factor

The calculation of the stress intensity factor for thread root cracks in the tank boss section was performed using the method specified by KHK S0220 Annex IX⁽¹¹⁾. This method was employed because it makes it simple to calculate the stress intensity factor from the results of stress analysis of the threads of the boss.

The method used for stress analysis of the threads of the boss is described below. This stress analysis used an axisymmetric FEM model (Fig. 5). Normally, the contact surfaces of each thread would be moved relative to each other by the application of the load, but for the sake of simplicity in the present analysis, the contact surfaces were fixed. The load pressure was set at 87.5 MPa, the maximum pressure of a 70 MPa hydrogen tank.

The following method was employed to calculate the stress intensity factor. First, the cubic equation Eq. (1) was used to approximate, σ , the stress distribution on the thread root. Equations (2) to (7) were then run with this figure as a coefficient. These calculations employed the stress distribution in the threads, where the maximum stress occurs in the thread section. For the approximation, the stress distribution was divided into four sections.

$$\sigma = A_0 + A_1x + A_2x^2 + A_3x^3 \quad (1)$$

$$K_I = F\sqrt{\pi a} + \sum_{i=1}^3 \Delta K_{I,i} \quad (2)$$

$$F = A_0F_1 + \frac{2aA_1F_2}{\pi} + \frac{a^2A_2F_3}{2} + \frac{4a^3A_3F_4}{3\pi} \quad (3)$$

$$F_1 = 1.1259 + 0.2344\left(\frac{a}{t}\right) + 2.2018\left(\frac{a}{t}\right)^2 - 0.2083\left(\frac{a}{t}\right)^3 \quad (4)$$

$$F_2 = 1.0732 + 0.2677\left(\frac{a}{t}\right) + 0.6661\left(\frac{a}{t}\right)^2 + 0.6354\left(\frac{a}{t}\right)^3 \quad (5)$$

$$F_3 = 1.0528 + 0.1065\left(\frac{a}{t}\right) + 0.4429\left(\frac{a}{t}\right)^2 + 0.6042\left(\frac{a}{t}\right)^3 \quad (6)$$

$$F_4 = 1.0387 - 0.0939\left(\frac{a}{t}\right) + 0.6018\left(\frac{a}{t}\right)^2 + 0.3750\left(\frac{a}{t}\right)^3 \quad (7)$$

Here, x is distance from the thread root in the radial direction. A_0 , A_1 , A_2 , and A_3 are constants. $\Delta K_{I,i}$ is the difference between the stress intensity factor at the boundaries of a section and the stress intensity factors of the previous section and the following section. a is the crack length from the thread root, and t is the thickness of the boss of the liner.

3.6. Test Method for Design Verification

Tests were conducted using a Type 3 tank with an A6066 aluminum alloy liner in order to verify the validity of the life design of the tank boss. The 2 mm initial defect used in the analysis of crack growth was introduced to the

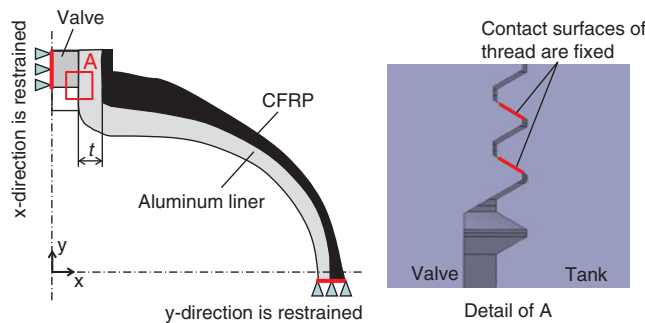


Fig. 5 Simulation model for stress analysis for calculating stress intensity factor

test tank at the position of maximum stress at thread root in the boss, with a circumferential notch being produced by electrical discharge machining (Fig. 6). Using this tank, 87.5 MPa hydraulic pressure cycle tests were conducted until a leak occurred. Following this, cracks from the bottom of the notch were observed, and the correlation with the life design was determined.

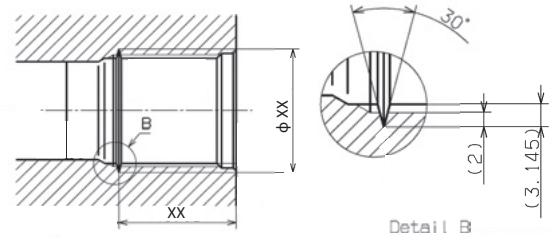


Fig. 6 Configuration of artificial notch machined into thread

4. Results of Crack Growth Analysis and Test Results

4.1. Fatigue Crack Growth in Humid Air

Figure 7 shows the results of a crack growth test in humid air conducted using a test specimen cut from the boss of an A6066 aluminum alloy liner. The approximation curve for the crack growth rate, da/dN , used in crack growth analysis, was set based on these results using the following equations. Here, N is the number of cycles.

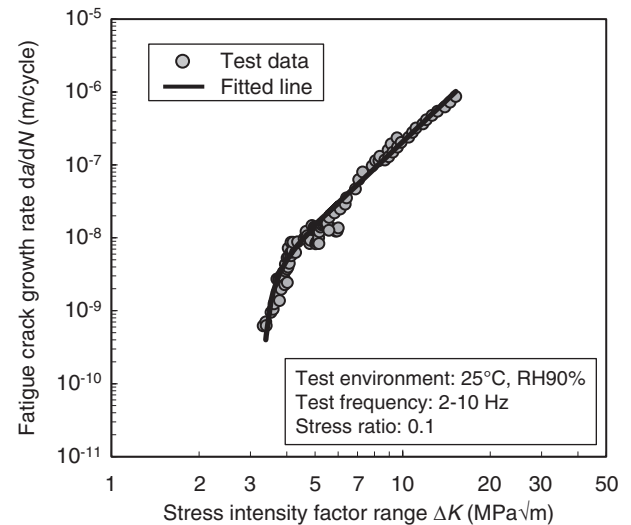


Fig. 7 Fatigue crack growth rate of specimen cut from A6066 aluminum alloy Type 3 tank liner boss

$$\frac{da}{dN} = f(\Delta K) \quad (8)$$

(i) When $3.4 \leq \Delta K \leq 5.0$

$$f(\Delta K) = 1.088 \times 10^{-9} \Delta K^2 - 6.724 \times 10^{-10} \Delta K - 1.020 \times 10^{-8} \quad (9)$$

(ii) When $5.0 < \Delta K$

$$f(\Delta K) = 3.433 \times 10^{-11} \Delta K^{3.781} \quad (10)$$

4.2. K_{ISCC} Test

In the fracture surfaces where cracks had clearly progressed in specimens of A6351, intergranular facets forming dimples were observed (Fig. 8). The fracture surfaces were observed with this characteristic as a reference and the crack lengths were measured.

Figure 9 shows the results of the K_{ISCC} test. Figure 9(a) shows the relationship between crack length and exposure time when the stress intensity factor under the initial load, K_{IAPP} , is the value specified in the ISO 7866 SLC test. For A6351, crack length increases with time under load. By contrast, in the case of A6066, there was almost no growth in the cracks. Figure 9(b) shows the amount of crack growth following 90 days of exposure for differing K_{IAPP} . This figure indicates that for each material, there is a stress intensity factor at which SLC does not occur. In addition, while A6066 is an excess silicon alloy like A6351, it clearly

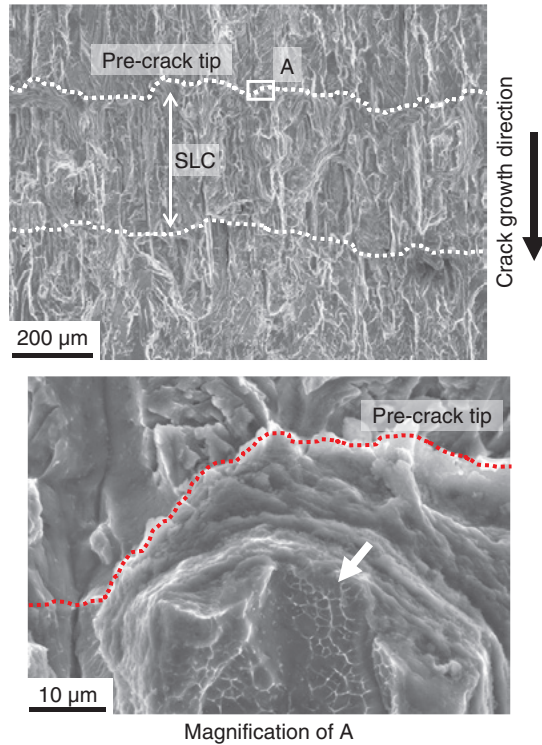


Fig. 8 SEM observation of A6351-T6 fracture surface caused by SLC ($K_{IAPP} = 17.2 \text{ MPa}\sqrt{\text{m}}$)

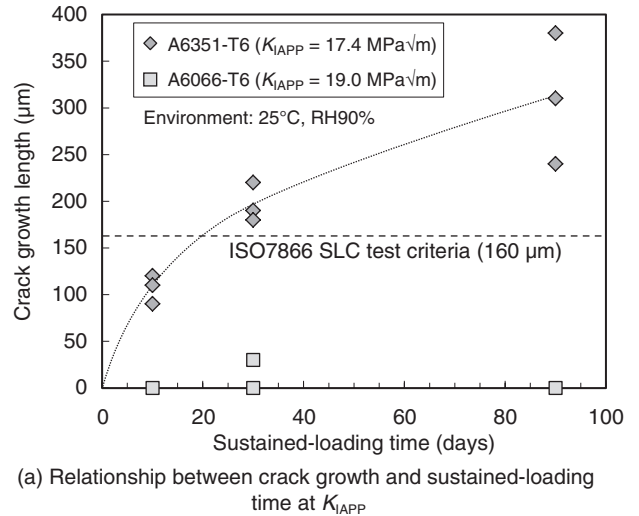
surpasses A6351 in terms of resistance to SLC.

Based on Figure 9(b), the highest stress intensity factor among the stress intensity factors for which SLC does not occur, $18.7 \text{ MPa}\sqrt{\text{m}}$, was taken as K_{ISCC} for A6066.

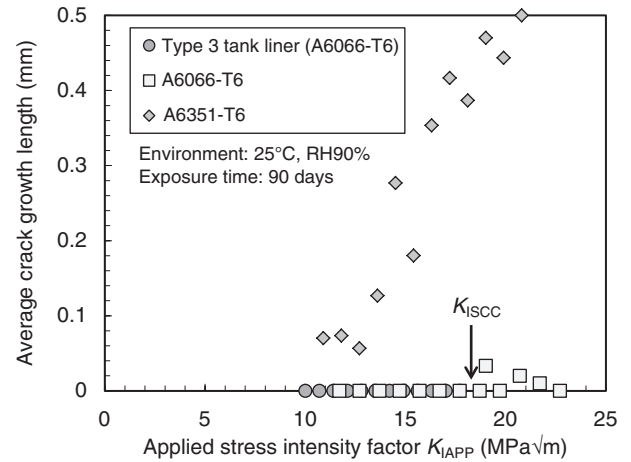
4.3. Calculation of Stress Intensity Factor Range and Crack Growth Life

Figure 10 shows the relationship between the stress intensity factors determined from the results of the stress analysis of the thread section of the liner boss and crack length.

Based on this relationship, crack growth life was calculated by integrating Eq. (8) using the crack length, as shown in Eq. (11). The stress intensity factor range ΔK was found using Eq. (12). This calculation assumed that each time the hydrogen tank was used, it went from the maximum pressure (gauge pressure) of 87.5 MPa to the minimum pressure of 0 MPa .



(a) Relationship between crack growth and sustained-loading time at K_{IAPP}



(b) Relationship between average crack growth length by SLC and applied stress intensity factor

Fig. 9 Results of K_{ISCC} test

$$N = \int_{a_0}^{a_{cr}} \frac{1}{f(\Delta K)} da \quad (11)$$

$$\Delta K = K_{p=8.75 \text{ MPa}} - K_{p=0 \text{ MPa}} \quad (12)$$

Here, a_0 is the initial crack length (the initial defect size assumed for calculations) and a_{cr} is the crack length that represents K_{ISCC} . Figure 11 shows the results of a crack growth analysis. The crack growth life from an initial crack of 2 mm to a crack length ($a_{cr} = 9.7$ mm) at which $K = K_{ISCC}$ (18.7 MPa $\sqrt{\text{m}}$) is 44,500 cycles. This figure is sufficiently long in relation to the maximum of 15,000 cycles in the life of a high-pressure hydrogen tank specified by the regulations⁽¹⁶⁾. The results discussed above indicate that SLC will not occur in the designed Type 3 tank before it reaches the end of its life as specified in regulations.

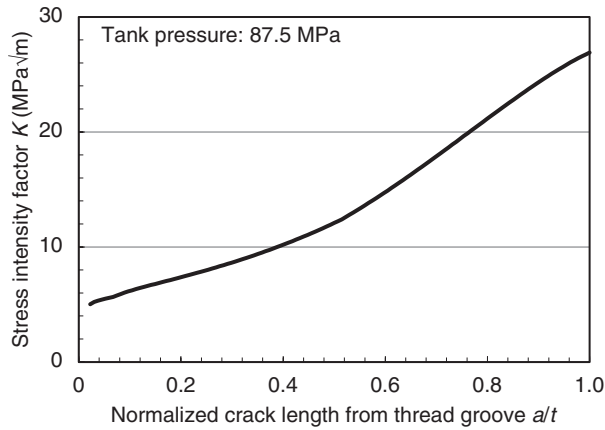


Fig. 10 Relationship between stress intensity factor and crack length from thread root

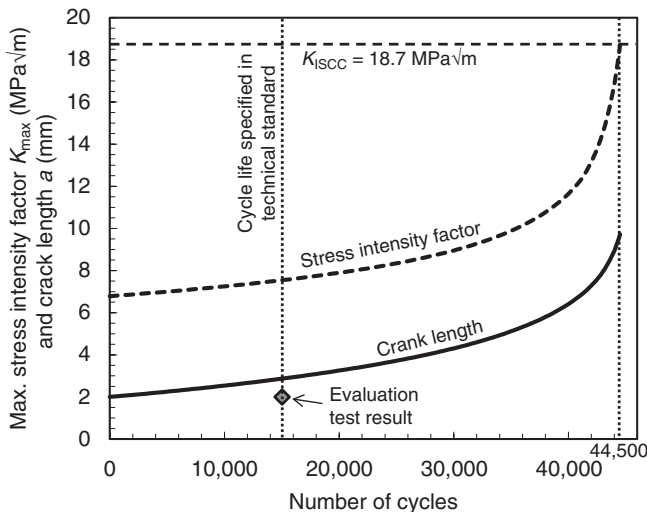


Fig. 11 Result of fatigue crack growth analysis

5. Discussion and Considerations

5.1. Considerations Concerning Results of Crack Growth Analysis

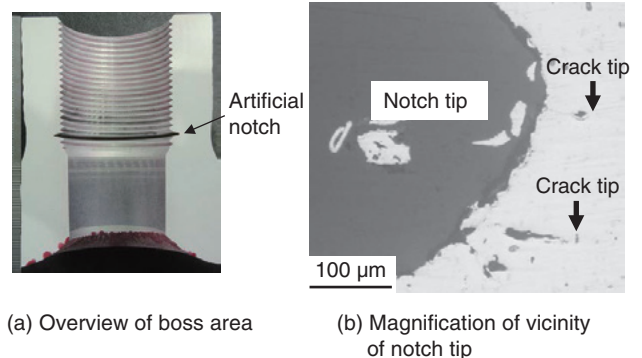
In order to assess the validity of the life design of the tank boss, hydraulic pressure cycle tests were conducted using an A6066 aluminum alloy Type 3 tank on which a notch had been machined in the thread section of the boss. The results showed that a leak occurred in the cylinder section of the tank at a greater number of cycles than 15,000, the maximum number specified in regulations. According to the crack growth analysis, it was predicted that at 15,000 cycles, the 2 mm notch representing the initial crack length would have grown to a maximum of 2.9 mm.

In order to compare these analysis results and the test results, following the completion of the test, a cross-section of the boss cut out from the liner was observed microscopically. Figure 12 shows a photograph of the cross-section. Investigation of the cross-section showed that a crack of approximately 60 μm in length extended from the tip of the notch, and that the length of this crack, including the notch, was 2.06 mm. It was therefore judged that the prediction based on the crack growth analysis was on the safe side. The following two points can be considered as factors in this.

- (1) In this crack growth analysis, because calculations were performed based on the assumption that the machined notch was the crack, the number of cycles to crack initiation from the notch was not considered.
- (2) The calculated value of the stress intensity factor in relation to the crack length was on the safe side.

Because the crack that was initiated in this test was very small at around 60 μm , it was considered that (1) was the main factor. At the same time, understanding the extent to which predictions fall on the safe side is important in discussing the life guarantee, and the following test was performed in order to study factor (2).

First, the depth of the machined notch was increased to greater than 2 mm without changing the shape of the tip or the width. This was to help ensure that cracks would grow from the defect. Next, the tank in which the notch had



(a) Overview of boss area

(b) Magnification of vicinity of notch tip

Fig. 12 Cross-sectional view of boss after cycle test

been machined was subjected to the cycle test described above in order to promote the crack growth from the defect. Following this, the tank boss was cut out, and the fracture surfaces were observed in order to assess the crack growth rate from fatigue striations.

As Fig. 13 shows, a crack has grown from the notch. In addition, this fracture surface shows fatigue striations. The intervals between these striations are known to correspond to the amount of crack growth for a single application of the load. The crack growth rate as obtained from the interval between striations and the crack growth rate as determined by the analysis were compared. Figure 14 shows the relationship between crack length and the crack growth rate. This figure indicates that the crack growth rate obtained

from the striations ranges from approximately 1/3 to 1/2 the rate determined by the analysis. Figures 7 and 10 also indicate that the stress intensity factors in relation to crack length determined by the analysis were 20 to 30% larger, indicating that the calculations were firmly on the safe side. This demonstrates the validity of the life calculation method discussed in this paper, including the simple method used for the calculation of stress intensity factors.

5.2. Evaluation Method with Consideration of Vehicle Use of Tank

The failure in the tank boss occurs as a result of cracks growing from the thread and penetrating the liner. Annexes A and B of ISO 7866, as discussed above, specify intercrystalline corrosion tests and SCC tests using smooth test specimens to evaluate crack initiation and SLC tests to evaluate crack growth. However, none of these can be considered appropriate evaluation methods, because they are not based on the failure mechanism of tanks. An evaluation method that does consider the failure mechanism will therefore be proposed here.

Due to the location where high-pressure hydrogen tanks are installed in vehicles, there is a high possibility. Given this, this section will touch on the reports of crack growth characteristics in various types of aluminum alloys in corrosive environments previously conducted by the authors^{(17), (18)}. Figure 15(a) shows results for A6061-T6, and indicates that the fatigue crack growth rate for this alloy increases even in the presence of only a slight concentration of saltwater (around 0.1%). Figure 15(b) shows the stress intensity factor when the crack growth rate in various test environments reaches 10^{-8} m/cycle. This figure indicates that resistance to crack growth increases in proportion to the stress intensity factor. Like A6061-T6, all of the aluminum alloys tested became prone to the growth of cracks in a low-concentration saltwater environment of around 0.1%. 6XXX series aluminum alloys display approximately the same resistance to fatigue crack growth in saltwater environments of 0.1% and 3.5%. When the shaft seal configuration shown in Fig. 2(a) was used, it was possible for saltwater and other corrosive agents to infiltrate the screw section of the tank boss from the exterior. Because the saltwater infiltrating the screw section would accelerate crack growth, as indicated above, it would be desirable to measure crack growth rate and K_{ISCC} with consideration of the saltwater environment during actual vehicle operation. In addition, sealing the outer surfaces of the screws, for example by using the seal configuration shown in Fig. 2(b), helps to prevent infiltration of saltwater. However, given that the use temperature range for the tanks is -40°C to 85°C , it can be predicted that condensation will occur in the airtight space maintained by the seals. The measurement of crack growth rate and K_{ISCC} for humid environments, as in the analysis discussed in this paper, can therefore be considered the appropriate method.

At the same time, the evaluation of tank performance

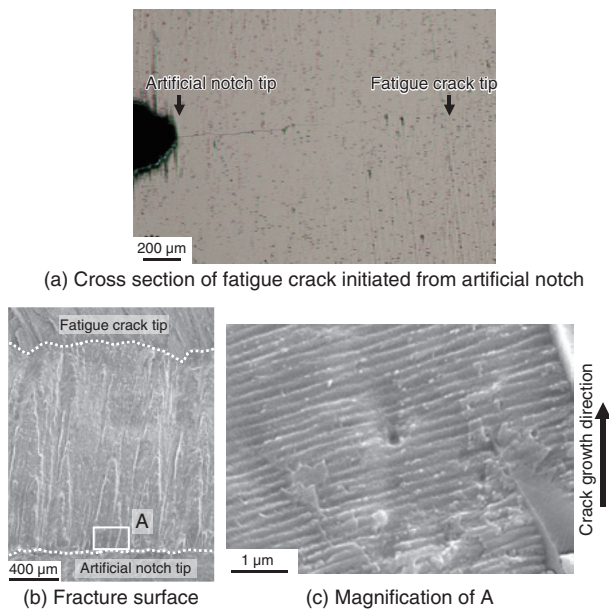


Fig. 13 SEM observation of striations

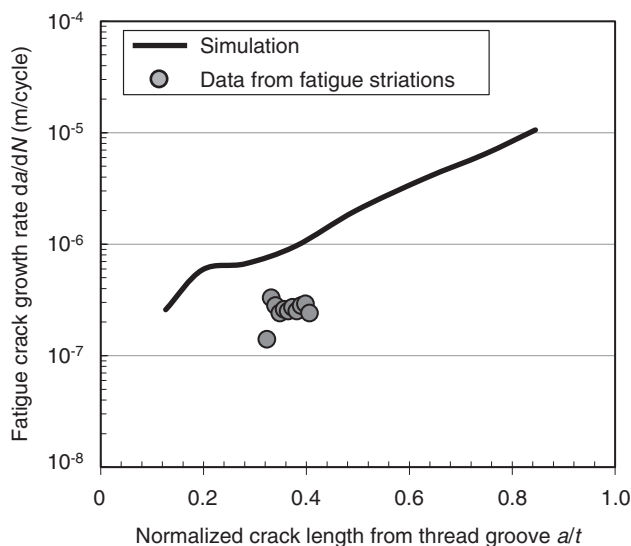


Fig. 14 Comparison between simulation and data obtained by fracture surface observation

in the GTR focuses on methods involving the tank itself rather than the evaluation of materials data for design use. A method of evaluation that takes this into consideration is as follows.

- (1) A limit dimension for a defect that could be detected in a non-destructive inspection in the manufacturing process is projected, and the initial defect dimension for the design is set on this basis.
- (2) A circumferential notch of the dimensions (depth) mentioned above is machined in the position at which maximum stress occurs at thread root in the tank boss.
- (3) The tank is subjected to hydraulic pressure cycle tests while the tank boss is sprayed with saltwater until the

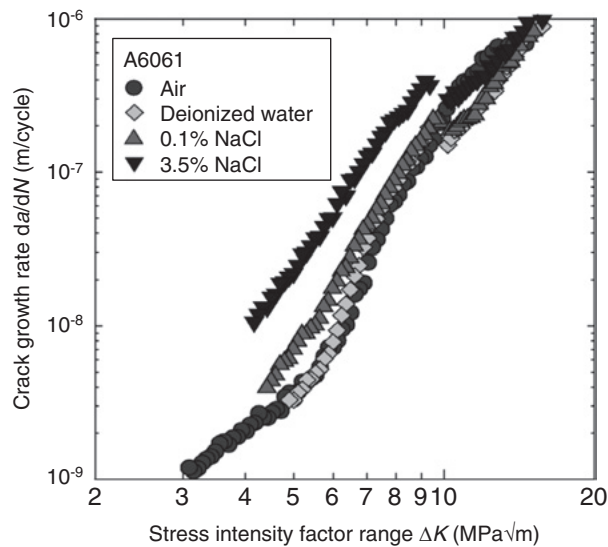
designed life of tank has been exceeded.

- (4) In order to evaluate SLC, the tank is exposed to internal pressure for long periods (for example, 1,000 hours) as a load.

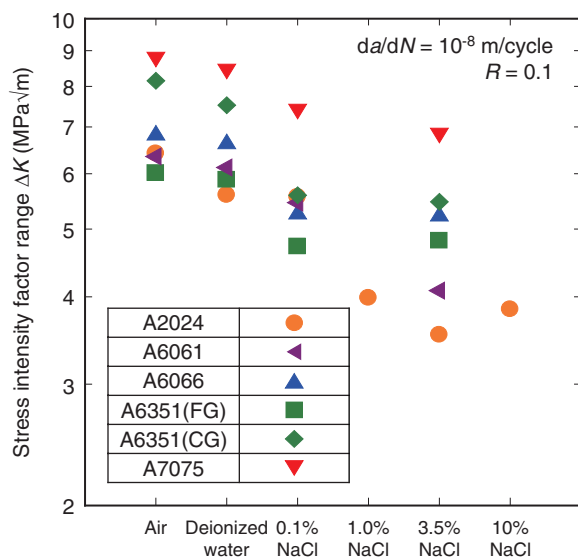
Figure 16 shows a schematic of the method described above. If tank failure does not occur at the boss, the life of the boss is sufficiently longer than the design life of the tank. Based on the following three points, it can be considered essential to use the saltwater spray discussed above in order to demonstrate that the tank will not be damaged even under the most severe conditions that it could be expected to face during its service life.

- (1) Saltwater accelerates the crack growth rate in aluminum alloys.
- (2) The internal and external surfaces of seamless aluminum alloy gas cylinders are inspected at specified intervals. Tanks found to have cracks during these inspections cannot be used⁽¹⁹⁾.
- (3) The hydrogen tanks used in fuel cell electric vehicles are inspected at regular intervals, but the interior surfaces are exempt from inspection⁽²⁰⁾. Because of this, cracks on the inner surfaces of the tanks cannot be detected.

The configuration of the boss for Type 4 tanks is the same as that used in Type 3 tanks, and it will therefore also be necessary to conduct evaluations of the type described in this paper for Type 4 tanks.



(a) Fatigue crack growth rate of A6061-T6 under corrosive environments



(b) Fatigue crack growth resistance under various environments of various aluminum alloys

Fig. 15 Corrosion fatigue crack growth characteristics

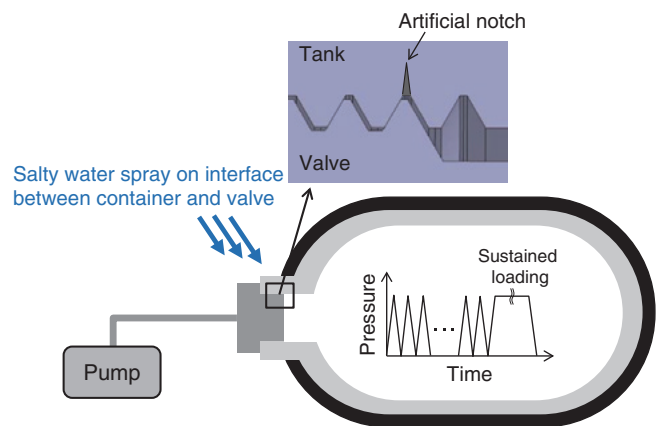


Fig. 16 Schematic of performance-based test for evaluating integrity of boss part against corrosive environments expected in vehicles

6. Conclusion

Suitable design and evaluation methods were developed for Type 3 high-pressure hydrogen tanks based on the failure mechanism of the tank boss, and tests were conducted in order to verify the effectiveness of the methods. The main outcomes of this process were as follows.

- (1) A life design method for the tank boss that considered SLC in humid environments was developed. The

validity of the life determined using this method was verified.

- (2) The environments that hydrogen tanks are exposed to in fuel cell electric vehicles and the fatigue characteristics of aluminum alloys in corrosive environments were taken into consideration, and an evaluation method of tank performance able to be reflected in regulations was proposed.

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Author



Toshihiko KANEZAKI



Satomi MANO



Kazuo MIYAGAWA



Noboru HAYASHI



Takeshi OGAWA

Robust Optimization for Windmill Airfoil Design under Variable Wind Conditions

Hiroki TACHIKAWA*

Daniele Schiavazziz**

Toshiyuki ARIMA***

Gianluca Iaccarino****

ABSTRACT

Robust design of windmill airfoils is carried out in the presence of uncertain environmental conditions. The formulation follows a multi-objective optimization approach, where the mean aerodynamic performance is maximized while sensitivity to changes in wind conditions is kept to a minimum. Due to the complex nature of the physical processes involved, a non-intrusive collocation approach is used to quantify uncertainties, coupled with a RANS solver for the fluid flow. A robust design optimization framework is thus assembled, where Genetic Algorithms are used to iteratively evolve section parameters toward optimal designs. Expectations and variances are computed with a Smolyak sparse grid approach combined with nested Clenshaw-Curtis quadrature, leading to a minimization of the number of deterministic solutions needed. The results show the potential applicability of the proposed methodology to the robust design of industrial devices.

1. Introduction

Wind power generation, the production of electric power by conversion of wind energy into propeller rotation, is increasingly seen as an attractive energy source.

The ability to achieve high conversion efficiency under changing wind conditions is crucial to this technology.

Optimization should guarantee a stable performance while maximizing the rotation thrust or, in other words, should be formulated as a robust process.

However, due to the significant computational cost associated with traditional Monte Carlo-like strategies used in conjunction with CFD simulations, design optimization is often carried out for specific environmental conditions.

The objective of this study is twofold:

- To investigate and show the increase in computational cost involved in the transition from traditional to robust optimization for windmill airfoil section profiles; and
- To explore efficient optimization methods.

2. Problem Formulation

The velocity triangle of a windmill airfoil section illustrated in Fig. 1 shows how the change in wind velocity

V_a results in changes in the angle of attack and inflow velocity relative to the airfoil section.

Two optimization problems are analyzed in the present study. The first approach is using a traditional optimization strategy where no variation of inflow wind conditions is considered. It is formulated as a single objective optimization, maximizing a measure of aerodynamic efficiency, i.e., the lift-to-drag coefficient ratio C_L/C_D .

Robust optimization is successively investigated, where disturbances in the inflow angle and Mach number are associated with uniformly distributed random variables

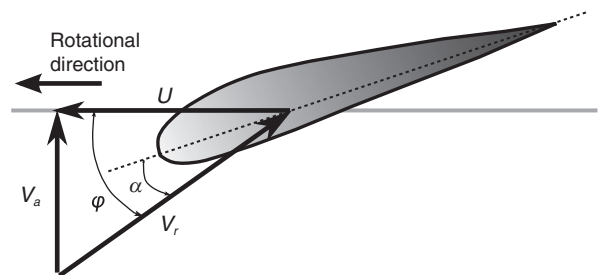


Fig. 1 Schematic representation of velocity triangle for windmill airfoil section

* Fundamental Technology Research Center

** University of California, San Diego

*** Automobile R&D Center

**** Stanford University

α and M , respectively. The average performance is thus maximized over a range of wind conditions, while minimizing the associated variance, i.e., the sensitivity to stochastic environmental changes.

Incompressible flow conditions are assumed throughout; the flow field is computed using a RANS solver at constant Reynolds number, where the airfoil geometry and wind conditions are determined by realizations of the parameters. Assume the airfoil cross-section geometry is determined by a vector $\xi \in D$ of parameters, where $\xi = \{\xi_1, \dots, \xi_{np}\}$, $\xi_i^l \leq \xi_i \leq \xi_i^u$ and D is the compact space of feasible airfoil configurations; ξ_i^l, ξ_i^u are the lower and upper bounds, respectively, for the generic parameter ξ_i . Moreover, consider a probability space (Ω, F, P) in which Ω is the sampling space, F is the σ -algebra of subsets of Ω , and P defines a probability measure on F . A vector of two independent and identically distributed random variables with joint probability density function $\rho(y) = \rho_\alpha(\alpha) \rho_M(M) : \mathbb{R}^2 \rightarrow \mathbb{R}^{\geq 0}$ is indicated by $y = (\alpha, M)$ with $\alpha, M : \Omega \rightarrow \mathbb{R}$. Realizations of the stochastic vector y are denoted by $y^{(i)} = (\alpha^{(i)}, M^{(i)})$, $i = 1, \dots, n_i$, where n_i is the total number of samples. Consider $f : D \times \Omega \rightarrow \mathbb{R}$ a function mapping design and stochastic parameters into lift-to-drag coefficient ratios. We also stress that every evaluation of f requires prior sampling of design parameters ξ and environmental variables $y^{(i)} = (\alpha^{(i)}, M^{(i)})$ followed by a RANS simulation. Design optimization can be formulated using two different approaches:

- (a) Without including the uncertainty in wind conditions (traditional optimization).

Find $\xi^* \in D$ such that

$$\xi^* = \arg \max_{\xi \in D} f(\xi, y^{(i)}) \quad (1)$$

- (b) Including the environmental uncertainty (robust approach).

Find $\xi^* \in D$ such that

$$\mathcal{M}(\mathbb{E}\{f(\xi^*, y)\}, \sigma\{f(\xi^*, y)\}) \geq \mathcal{M}(\mathbb{E}\{f(\xi, y)\}, \sigma\{f(\xi, y)\}) \quad (2)$$

$\forall \xi \in D$

where $\mathbb{E}\{\cdot\}$ and $\sigma\{\cdot\}$ denote the expectation and standard deviation operator, expressed as:

$$\mathbb{E}\{f(\xi, y)\} = \int_{M_i} \int_{\alpha_i}^{\alpha_u} f(\xi, y) \rho_\alpha(\alpha) \rho_M(M) d\alpha dM \quad (3)$$

$$\sigma\{f(\xi, y)\} = \left(\int_{M_i} \int_{\alpha_i}^{\alpha_u} (f^2(\xi, y) - (\mathbb{E}\{f(\xi, y)\})^2) \rho_\alpha(\alpha) \rho_M(M) d\alpha dM \right)^{1/2} \quad (4)$$

and $\mathcal{M} : \mathbb{R}^2 \rightarrow \mathbb{R}$ is a multi-objective decision criterion. In other words, the solution of the robust optimization problem maximizes the mean lift-drag ratio across variable environmental conditions while minimizing the associated standard deviation, i.e., the sensitivity of the airfoil efficiency to alterations in the angle of attack and Mach number. Here $[\alpha_l, \alpha_u]$ and $[M_l, M_u]$ are elementary event intervals associated with the inflow angle α [deg] and Mach number M , respectively. In practice, we adopt

$$[\alpha_l, \alpha_u] = [\alpha_0 - 2, \alpha_0 + 2], \alpha_0 = 10 \quad (5)$$

$$[M_l, M_u] = [M_0 - 0.05, M_0 + 0.05], M_0 = 0.25 \quad (6)$$

As formulated in Equation (2), the problem is essentially a multi-objective optimization whose solution is generated with a trade-off between the two objective functions, i.e., the average performance and sensitivity to environmental conditions. Note that different choices of $\mathcal{M}$ lead to different optimal solutions. For example, given two objectives $o_1, o_2 \in \mathbb{R}$, $\mathcal{M}(o_1, o_2) = o_1$ generates the solution with the maximum average efficiency, $\mathcal{M}(o_1, o_2) = -o_2$ that with minimum sensitivity, while $\mathcal{M}(o_1, o_2) = \sqrt{(o_1/n_1)^2 + (o_2/n_2)^2}$ might be chosen as a compromise. Here, the normalization factors n_1 and n_2 account for cases where objectives are not homogeneous or provided in different units.

3. Airfoil Representation

The PARSEC⁽¹⁾ representation is used to define a parametric airfoil profile.

In this representation, the upper and lower section curves are expressed by polynomials of the following form:

$$z = \sum_{i=1}^6 a_i x^{(i-1/2)} \quad (7)$$

where the section is defined in the (x, z) plane.

The coefficients a_i are determined from the geometric parameters $\xi \in D$ (design variables), illustrated in Fig. 2. Eleven design variables are selected: leading edge radius, upper and lower crest locations and curvatures, trailing edge coordinates (at $x = 1$), thickness, direction angle and wedge angle. Design variables and corresponding ranges are represented in Table 1. As PARSEC tries to minimize the number of parameters needed to generate wing profiles of practical interest in applications, it is useful to consider the possible interaction between some of these parameters. An example is illustrated in Fig. 3, where different values are applied to the parameters z_{TE} and $z_{lo} + z_{up}$, respectively.

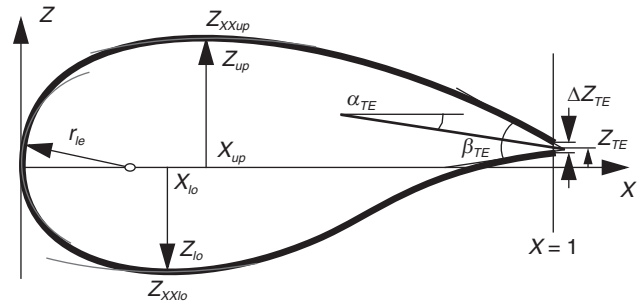
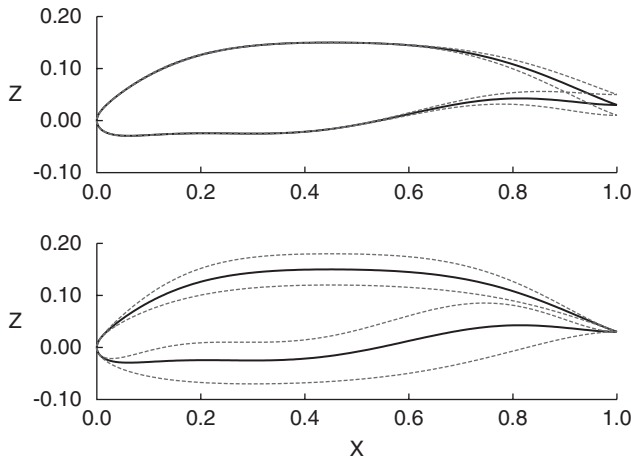


Fig. 2 Graphical representation of input parameters for PARSEC

Table 1 List of parameters used in PARSEC and α , M

ID	Description	ξ_l	ξ_u
r_{le}	Radius of leading edge	0.005	0.02
x_{up}	X at crest of upper surf.	0.3	0.7
z_{up}	Z at crest of upper surf.	0.12	0.18
$z_{xx,up}$	Curv. of upper surf.	-0.4	0.0
x_{lo}	X at crest of lower surf.	0.2	0.6
z_{lo}	Z at crest of lower surf.	-0.07	0.02
$z_{xx,lo}$	Curv. of lower surf.	0.3	0.9
z_{TE}	Z at trailing edge	-0.01	0.05
α_{TE}	Camber gradient at trailing edge	-13.0	-3.0
Δz_{TE}	Thickness at trailing edge	0.0	0.0
β_{TE}	Wedge angle at trailing edge	4.0	8.0
α	Angle of attack	8.0	12.0
M	Mach number	0.2	0.3

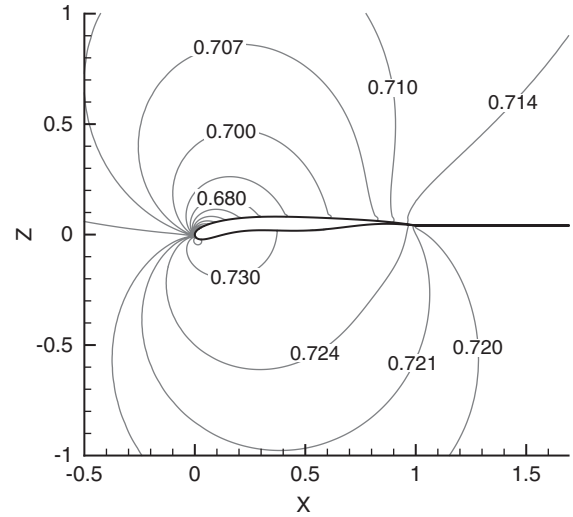
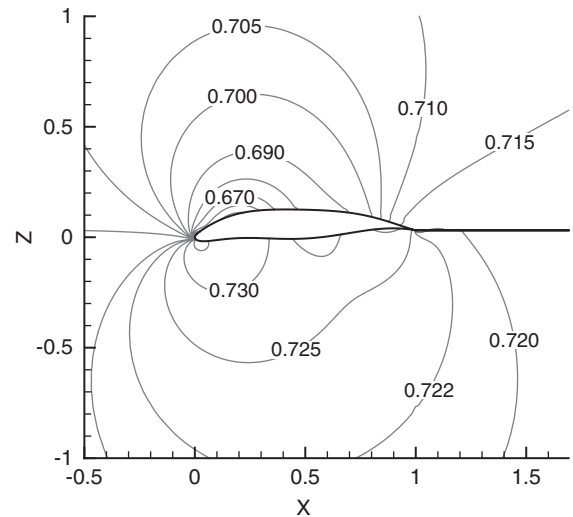
Fig. 3 Change in airfoil configuration as a result of adjusting z_{TE} (top) or $z_{lo} + z_{up}$ (bottom)

4. Computation of Lift and Drag, Preliminary Optimizations

Once the airfoil profile is determined by the parameter set ξ , the solution to the two-dimensional incompressible Navier-Stokes equation is sought using a Reynolds number equal to 10^5 . An automatic meshing procedure is first carried out starting from the profile generated by PARSEC. The CFD solver⁽²⁾ uses a delta form implicit finite difference method⁽³⁾. A 3rd-order Chakravarthy-Osher TVD limiter⁽⁴⁾ is also used for advection together with the Menter k- ω SST fully transitional turbulence model⁽⁵⁾.

As a preliminary step, the windmill profile was optimized under known wind conditions $(\alpha, M) = (10, 0.25)$. The resultant shape is shown in Fig. 4. A very thin airfoil is generated, as expected, exhibiting a pronounced curvature of the leading edge in the direction of the selected angle of attack. This solution, although characterized by very high levels of aerodynamic efficiency, shows an insufficient

section modulus, and thus lacks structural strength. An alteration to the initial design parameter space D is thus required to fulfill both aerodynamic and structural feasibility. The modified “constrained” optimal solution is represented in Fig. 5. In practice, the initial range for z_{up} equal to $[0.08, 0.18]$ has been modified to $[0.12, 0.18]$, resulting in the section thickness range to change from $[0.06, 0.25]$ to $[0.10, 0.25]$, respectively.

Fig. 4 Pressure levels for unconstrained optimal airfoil design under certain wind conditions $(\alpha, M) = (10, 0.25)$ Fig. 5 Pressure levels for constrained optimal airfoil design under certain wind conditions $(\alpha, M) = (10, 0.25)$

5. Robust Optimization Methods

Shape optimization of windmill airfoils is performed by means of Genetic Algorithms (GA).

This family of methodologies seeks optimal solutions by

selectively creating successive generations of individuals. Each individual represents a design configuration with chromosomes ξ , i.e., the 11 section design parameters. Each individual is also associated with a set of environmental conditions, that is, realizations of angle of attack and Mach number $y^{(i)} = (\alpha^{(i)}, M^{(i)})$, $i = 0, \dots, n_i$. These locations are carefully selected to facilitate the computation of statistics via multivariate quadrature. This differentiates a robust approach from traditional optimization, where a single evaluation of f is sufficient for every individual in the population.

Now we assume that $f(\xi^*, y)$ takes the form

$$f(\xi^*, y) \approx \sum_{i=0}^P \alpha_i(\xi^*) \phi_i(y) \quad (8)$$

as a finite linear combination of tensor product orthogonal polynomials of the random vector f (polynomial chaos expansion). The quantities $\mathbb{E}\{f(\xi^*, y)\}$ and $\sigma\{f(\xi^*, y)\}$ can be computed by numerical integration as follows:

$$\mathbb{E}\{f(\xi^*, y)\} \approx \sum_{j=1}^{n_i} \omega_j f(\xi^*, y^{(j)}) \quad (9)$$

$$\sigma\{f(\xi^*, y)\} \approx \left\{ \sum_{j=1}^{n_i} \omega_j [f(\xi^*, y^{(j)}) - \mathbb{E}\{f(\xi^*, y)\}]^2 \right\}^{1/2} \quad (10)$$

where $\sum_{j=1}^{n_i} \omega_j = 1$. Here the n_i quadrature locations are the zeros of the selected tensor product polynomial family $\phi_i(y)^{(6)}$. Optimal convergence to the statistics of sufficiently smooth stochastic responses is obtained by employing polynomial families orthogonal to the input probability measures. Our choice of adopting Clenshaw-Curtis quadrature^{(7), (8)} locations translates into expanding f using Chebyshev polynomials. For applications where f is not known in advance and arbitrary input probability measures can be specified, the good convergence properties of Chebyshev approximants give us a good compromise to be implemented in a general framework. For every computed generation, the steps performed by GA are highlighted in Fig. 6.

The computational cost of evaluating multivariate integrals, like $\mathbb{E}\{f(\xi, y)\}$, with given accuracy leads to a dramatic increase of polynomial terms (or numerical integration points) for a corresponding increase in the number of stochastic input variables. This fact is generally referred to as the “curse of dimensionality”; a typical trend is shown in Table 2 (Top). We also stress that every evaluation of $f(\xi^*, y^{(i)})$ requires a complete RANS simulation. Thus, the number of deterministic realizations must be kept to a minimum if robust optimization is to be performed in a reasonable period of time. In this study, we adopt a Smolyak sparse grid⁽⁹⁾ approach together with nested Clenshaw-Curtis quadrature as a possible mitigation of this phenomenon (Table 2 Bottom). A two-dimensional Smolyak sparse grid with 13 points is used in the present study. Approximation order 0, 1 and 2 are thus recovered with 1, 5 and 13 quadrature points, respectively, allowing rough estimates of convergence and accuracy to be computed. Figure 7 shows the selected quadrature grid in the $\alpha - M$ plane. Finally, we note that this study employs a non-intrusive approach that uses an

Table 2 Increase in multivariate quadrature points with dimensionality for fixed one-dimensional polynomial accuracy

Dimension	1	2	3	4	5	n
Full grid	5	25	125	625	3125	5^n
Smolyak sparse grid	5	13	33	89	253	$3^n + 2n$

unmodified deterministic CFD solver even if stochasticity has been injected into the equations. This approach is justified by the fact that the accuracy of the RANS solution is largely constant for the selected parameter ranges.

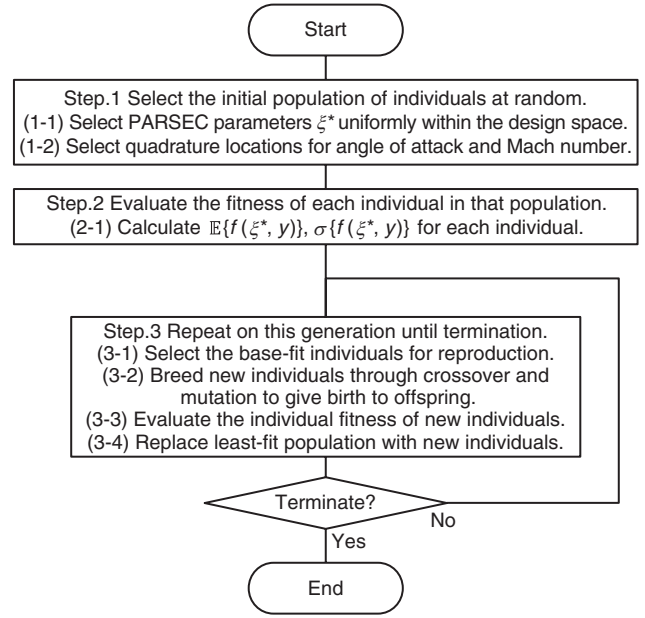


Fig. 6 Genetic Algorithm

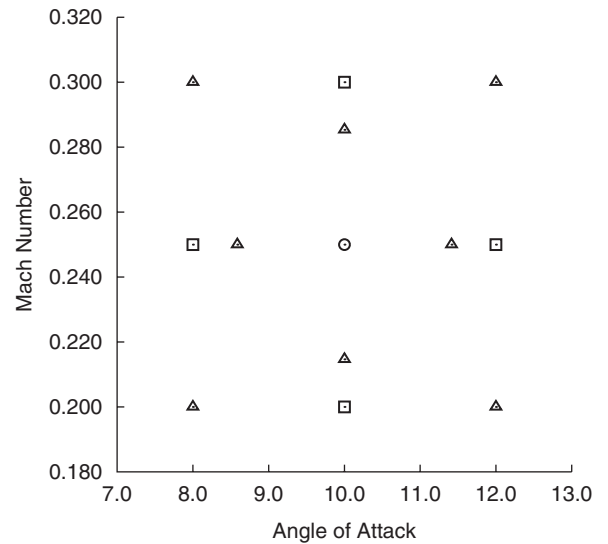


Fig. 7 Two-dimensional Smolyak sparse tensor quadrature grid up to order 2 accuracy. Level 0 ($\circ$), Level 1 ($\square$) and Level 2 ($\triangle$) incremental grids are shown.

Table 3 Sensitivity of optimal design to GA parameters

Population size	20	50	100
Optimal C_L/C_D	38.20	37.70	39.22
Cross probability	0.2	0.4	0.6
Optimal C_L/C_D	37.94	39.23	39.22
Flip probability	0.002	0.004	0.02
Optimal C_L/C_D	39.22	39.58	38.22

6. Optimal Windmill Airfoils

First, a sensitivity analysis is performed to assess how the optimal design is influenced by some of the GA parameters, i.e., population size, cross probability, and flip probability.

This is performed for fixed environmental conditions $(\alpha, M) = (10, 0.25)$; results are illustrated in Table 3. For a sufficient population size, sensitivity to the GA parameters has a limited effect on the optimal solution.

Two optimization tasks denominated “traditional” and “robust” are then carried out, generating optimal designs ξ_T^* and ξ_R^* , respectively.

The location of the traditional optimal design in the $(C_D, 1/C_L)$ plane is highlighted in Fig. 8. After the set of parameters ξ_T^* is found giving the maximum C_L/C_D ratio for fixed environmental conditions $(\alpha, M) = (10, 0.25)$, an uncertainty propagation analysis is carried out leading to $\mathbb{E}\{f(\xi_T^*, y)\} = 30.46$ and $\sigma\{f(\xi_T^*, y)\} = 2.53$ (Table 4, upper).

Some of the generations produced by robust optimization are also shown in Fig. 9. Optimal designs have been reported that correspond to various metrics M , i.e., maximum expected efficiency, minimum standard deviation

Table 4 Comparison of results for traditional and robust optimizations

	Design	Statistic	C_L/C_D
Traditional optimizations	Max C_L/C_D	Mean	30.46
		SD	2.53
		SK	-0.45
Robust optimizations	Maximum average efficiency	Mean	32.68
		SD	1.64
		SK	-0.40
	Minimum combined metric	Mean	15.96
		SD	0.24
		SK	-0.04
	Minimum sensitivity	Mean	2.35
		SD	0.10
		SK	0.12

SD: Standard Deviation, SK: Skewness

and best compromise. Values of $\mathbb{E}\{f(\xi_R^*, y)\}$ and $\sigma\{f(\xi_R^*, y)\}$ for designs selected according to the above metrics are also reported in Table 4 (bottom).

Note that first-, second-, and also third-order statistics have been included in the analysis.

Moreover, a negative skewness is observed for most reported designs, showing an asymmetry of the pdf of f with a longer tail towards values of smaller efficiency. This provides an even stronger motivation for including variance minimization as a further optimization objective.

Robust optimal designs with minimum standard deviation and best compromise metrics are much less sensitive to the environmental conditions than traditional optima.

However, average performance is also significantly

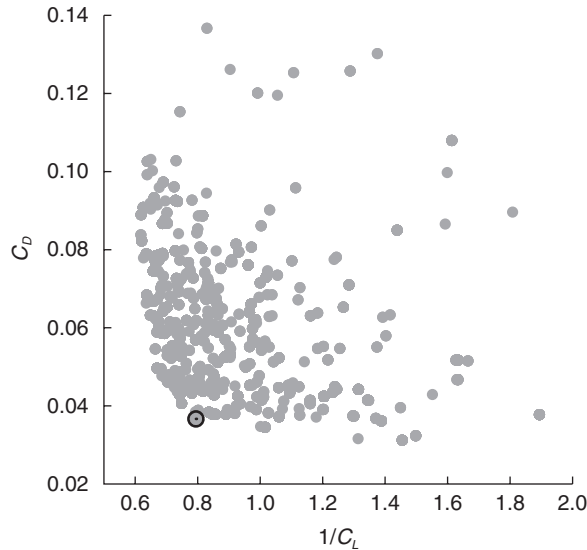


Fig. 8 Individuals generated by GA for single-objective optimization not accounting for variability in wind conditions. The optimal constrained design (○) is highlighted.

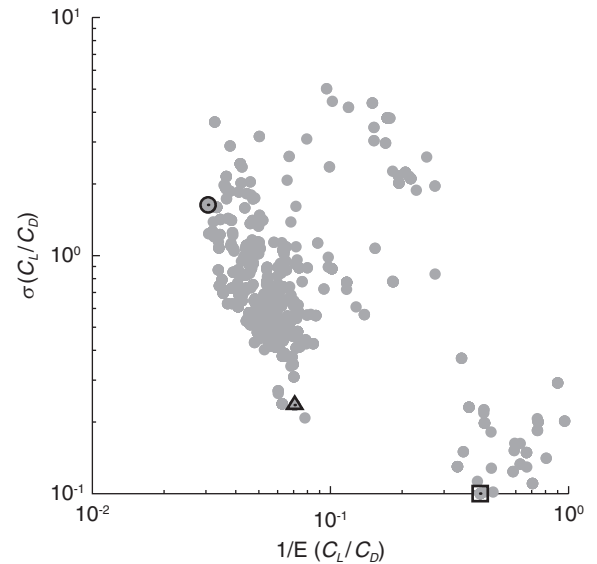


Fig. 9 Efficiency-sensitivity tradeoff resulting from robust optimization. Designs with maximum average efficiency (○), minimum combined metric (△) and minimum sensitivity (□) are highlighted.

lower in this case.

The robust design maximizing the expected efficiency metric results, as expected, in better average performance across environmental conditions than the traditional optimal solution.

As a result of the explicit inclusion of the variance minimization in the optimization task, the robust optimal is also less sensitive to changes of wind conditions.

In conclusion, the proposed robust optimization framework has proven successful in improving the efficiency of a traditional optimum across a spectrum of uncertain wind conditions.

7. Conclusions and Future Work

In the present study, a robust optimization framework has been assembled using Genetic Algorithms with uncertainty propagation techniques and applied to maximize the efficiency of a windmill airfoil over a spectrum of environmental scenarios.

A Smolyak sparse tensor grid of nested Clenshaw-Curtis one-dimensional quadrature rules is used to mitigate the curse of dimensionality, of particular importance here because the airfoil efficiency is evaluated by solving a complete RANS simulation for any realization in the parameter space and given wind conditions.

Robust optimization has proven successful in providing better performing designs than traditional optima over a range of uncertain wind conditions, thus leading to savings in manufacturing resources and increasing the generated power.

This technique is particularly appealing in the development of industrial devices that perform under variable environmental conditions.

If quantities of interest exhibit sufficiently smooth variations in response to parameter changes, then sparse grid approaches can be used to minimize the number of deterministic solutions needed, thus reducing the overall computational cost.

Moreover, robust optimization provides a systematic and theoretically sound way to account for aleatoric uncertainty in engineering design.

This paper is based on the results of research⁽¹⁰⁾ that the authors carried out at the Center for Turbulence Research (Stanford University) sponsored summer program in 2012.

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■ Author ■



Hiroki TACHIKAWA



Daniele Schiavazziz



Toshiyuki ARIMA



Gianluca Iaccarino

Development of Inverse Problem Analysis System in Electromagnetics

Masatoshi HARUMATSU*

Katsuya YASHIRO*

Kensuke SEKIHARA**

ABSTRACT

An inverse problem analysis system for electromagnetic fields, for use in the field of brain-machine interfaces (BMI), has been developed. In the conventional technique, a spherical approximation of the human head is used for the calculation of the electromagnetic field in the brain. The spherical approximation presents the issues that the shape and electromagnetic properties of the human head cannot be considered in the calculations, and that the currents normal to the surface of the sphere cannot be detected. In order to address these issues, we used the boundary element method (BEM) as a numerical method, to calculate the electromagnetic field, in addition to developing the mesh model and mesh generation method that were essential to the calculations. We employed a minimum-variance spatial filter as the method for inverse problems and applied a sub-grid technique to minimize the increase in calculation time of the numerical method. We verified the effectiveness of the method using a numerical simulation problem to reconstruct the hypothetical electric current sources in the brain. The results showed that the system was able to analyze inverse problems with consideration of the shape and electromagnetic properties of the tissues of the human head.

1. Introduction

Research is underway world-wide to develop interface technologies to operate devices by thought alone. Interface technologies of this type are called brain-machine interfaces (BMI)⁽¹⁾. If BMI technologies make it possible for people to operate information devices, home electronic devices and even cars without touching them, it will not simply increase operability, but will also increase the freedom of device design and modes of use. In addition, it would help expand the potential for people with physical disabilities to operate devices around them, and could thus be expected to contribute to the field of welfare. At present, research on basic technologies for BMI is proceeding at a number of research institutes. Because BMI necessitates devices to be operated by specific brain activity, technologies to sense that brain activity are an important element. It is unrealistic to implant sensors in the brain in order to sense brain activity, and it is necessary to develop non-invasive measurement technologies able to detect brain activity and inverse problem analysis technologies to make it possible to calculate the position of the current in the brain on the basis of the measurement data.

Near infrared spectroscopy (NIRS) and functional magnetic resonance imaging (fMRI) are non-invasive

technologies able to measure brain activity. These technologies are able to detect the biological phenomenon of increased blood flow in active areas of the brain. NIRS is a detection method that uses infrared light, while fMRI uses strong magnetic fields. Because there is a several-second time lag between brain activity and increased blood flow, both methods present a delay in detection. Furthermore, NIRS presents the difficulty in detecting changes in blood flow in the deep regions of the brain. Other methods are also available that do not measure secondary biological phenomena such as blood flow, but that directly measure current in the brain as an electromagnetic phenomenon. Methods that detect brain waves (through changes in the electric potential of the skin of the scalp) or brain magnetism (through changes in the magnetic field around the head) are representative, and are termed electroencephalography (EEG) and magnetoencephalography (MEG), respectively. The detection lag for both methods is in the sub-millisecond range, and they are both able to detect activity in deep areas of the brain. They can therefore be expected to find application in BMI.

2. Issues of Conventional Technologies

The position of the current in the brain can be

* Fundamental Technology Research Center

** Tokyo Metropolitan University

determined by solving an inverse problem with the electric signals obtained by EEG or the magnetic signals obtained by MEG as input conditions. A system that determines the position of the current in the brain using MEG has already been developed, and is being used in a variety of medical research. The following steps are used to determine the position of the current.

- (1) The magnetic field produced by brain activity is measured using magnetic field sensors.
- (2) Calculation points are set for the current source (on a grid).
- (3) A coefficient termed the lead field is calculated for each grid.
- (4) An inverse problem is solved based on the sensor signal and the lead field, and the position of the current is determined.
- (5) The determined position of the current is displayed on the grids as an image.

The lead field is a coefficient that expresses the relationship between the current sources and the sensor signals. It expresses the electric field and the magnetic field that a unitary current source creates at the position of the sensor. The lead field is determined by electromagnetic calculations from the shape, electromagnetic properties, the position of the current in human tissues, and from the position and orientation of the sensors. Standard lead field calculation uses a spherical approximation for the brain shape. When the brain is assumed as a perfect sphere with a homogeneous medium and the current density vector $\mathbf{J}$, is localized at position $\mathbf{r}_0$, the magnetic field $\mathbf{H}$ at position $\mathbf{r}$, can be expressed by Eq. (1)⁽²⁾.

$$\mathbf{H}(\mathbf{r}) = \frac{1}{4\pi A^2} (A\mathbf{J} \times \mathbf{r}_0 - \mathbf{J} \times \mathbf{r}_0 \cdot \mathbf{r} \nabla A) \quad (1)$$

$$A = |\mathbf{r} - \mathbf{r}_0| (|\mathbf{r}| |\mathbf{r} - \mathbf{r}_0| + |\mathbf{r}|^2 - \mathbf{r}_0 \cdot \mathbf{r})$$

If spherical approximation is used in the lead field calculations, two issues arise. The first issue is that it is not possible to consider the shape or electromagnetic properties of the tissues of the human head. The broken lines in Fig. 1 show a spherical approximation superimposed on the brain in a human head. It is clear that the brain is not a perfect sphere, and that there exist large gaps between the lower

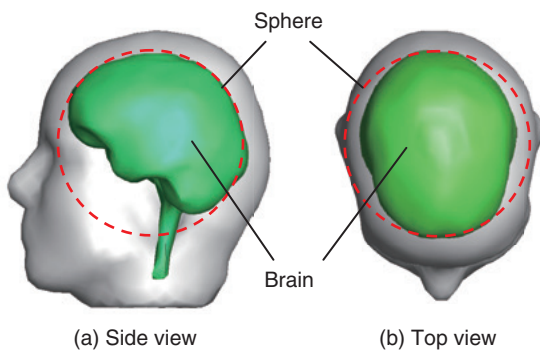


Fig. 1 Human head and sphere

section and sides of the sphere and the brain. The shape of the brain differs for each test subject, and these differences cannot be considered when using a spherical approximation. In addition, there is no term for electromagnetic properties in Eq. (1), meaning that electromagnetic properties cannot be considered.

The second issue is the occurrence of a dead zone. When approximating the brain to a sphere, the calculated value of the magnetic field produced by the electric current normal to the spherical surface is 0. This is caused by the symmetry of the sphere. Because of the various convexities, concavities and asymmetries of the real human subjects, the generated magnetic field is not 0. The currents cannot be calculated with spherical approximation and lead to the calculated current distribution differing from the actual one.

In order to address these issues, a numerical analysis is adopted instead of spherical approximation. The numerical analysis calculates lead fields by solving Maxwell's equations considering shape and electromagnetic properties. As numerical analysis necessitated a mesh model, we developed a human head model and a mesh generation method. We also developed a grid generation method to generate calculation points for the current sources. A minimum-variance spatial filter was applied as the inverse problem analysis method, and, in order to minimize the increase in calculation time due to the numerical method, a sub-grid technique was employed. An inverse problem analysis system for electromagnetic fields was developed using these technologies. Chapter 3 will discuss the technologies making up the system in greater detail. Chapter 4 will consider the testing of the system, and Chapter 5 will provide the conclusions reached.

3. Element Technologies of Inverse Problem Analysis System for Electromagnetic Fields

3.1. Mesh and Mesh Generation Method for Human Model

Because a BEM was employed in the calculation of the lead fields, as will be discussed below, a human surface mesh model was necessary. The head was divided into layers based on the differences in electric conductivity of tissues, and a mesh was generated for the surface of each tissue layer. Figure 2 shows an example of a four-layer model of the human head generated by this system. The tissues of the human head are represented by four mesh surfaces. From the outermost layer, these are the skin, the skull, the cerebrospinal fluid, and the brain. The conductivity of the tissues adjacent to the inside and the outside is imparted to each layer of mesh as data on electromagnetic properties.

The mesh should satisfy the following conditions.

- (1) The size and shape of the elements should be as uniform as possible.
- (2) The model should display good shape reproducibility.
- (3) Mesh generation should be rapid.

We developed a new mesh generation method because there were no appropriate methods and tools that would satisfy the above conditions. Mesh generation has two steps: generation of uniform elements and deformation of mesh into a human head. First, a mesh was generated with a sphere as the base surface in order to verify the uniformity of the elements. The surfaces of regular polyhedral elements were subdivided recursively, and projected onto the surface of a circumsphere in order to generate a spherical mesh with the necessary number of elements. Because of this subdivision of the surfaces of regular polyhedrons and projection onto a spherical surface, the elements are basically uniform with minimal distortion, and the process of mesh generation is rapid. The number of elements is controlled by varying the number and frequency of the subdivisions. The subdivision points for the elements are calculated using Eq. (2). Here, $\mathbf{r}_a$ and $\mathbf{r}_b$ are coordinate vectors of both ends of the edge, and $\mathbf{r}_c$ is the coordinate vector of the point at which the edge is internally divided by p:q and projected onto the circumsphere. In all cases the origin point is the center of the sphere.

$$\mathbf{r}_c = \frac{|\mathbf{r}_a|}{|q\mathbf{r}_a + p\mathbf{r}_b|} (q\mathbf{r}_a + p\mathbf{r}_b) \quad (2)$$

Figure 3 shows an example of subdivision of a mesh in which the edge is divided equally twice. From the left, the figure shows an icosahedron mesh, the mesh after one subdivision, and the mesh after two subdivisions.

When the human head is imaged using MRI, a cross-sectional image of varying shades is obtained in response

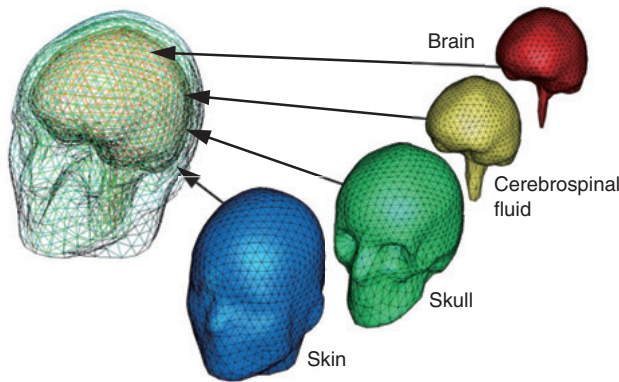


Fig. 2 Mesh model of human head with four shells

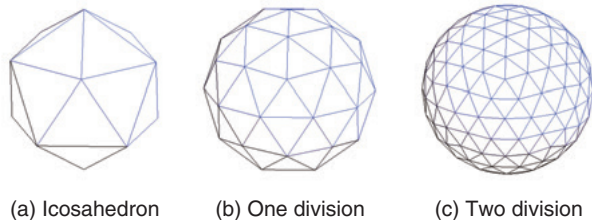


Fig. 3 Mesh refinement

to the physical properties of the tissues. The shape of the head can be reproduced three-dimensionally by layering these cross-sectional images. As shown in Fig. 4(a), a surface mesh can be generated by positioning this three-dimensional MRI image data of the head on a spherical mesh so as to enroll it, and projecting the mesh onto the tissue surface from the MRI data. The tissue for which the mesh is generated can be selected by varying the intensity of the MRI data on which it is projected.

The projection operation is performed by translating the nodes of the spherical mesh onto the points expressed by Eq. (3).

$$\mathbf{r}_p = \mathbf{r}_s + \gamma \frac{\mathbf{v} \cdot (\mathbf{r}_m - \mathbf{r}_s)}{\mathbf{v} \cdot \mathbf{v}} \mathbf{v} \quad (3)$$

$\mathbf{v}$ is the inward normal vector of the spherical mesh, $\mathbf{r}_s$ is the vector of the node of the spherical mesh, $\mathbf{r}_m$ is the closest MRI data point vector to $\mathbf{r}_s$ in the vicinity of the direction of $\mathbf{v}$, γ is a positive real number of 1 or less and is a coefficient expressing the projection ratio, and $\mathbf{r}_p$ is the vector of the projected point. When the concavity and convexity of the tissue surface is comparatively minimal, a mesh deformed into the tissue shape can be obtained using one projection with $\gamma=1$. When there is significant concavity and convexity of the tissue surface, a number smaller than 1 is used for γ , and, following deformation of the spherical mesh to make it slightly resemble the concave and convex shapes of the tissue, smoothing is applied to correct distortion in the mesh. This operation is repeated until the mesh shape sufficiently resembles the concave and convex tissue surface, and is finally projected onto the tissue surface with $\gamma=1$. We used the Taubin method⁽²⁾ for smoothing. Figure 4(b) shows an example of the generation of a mesh model of brain tissue including the spinal cord generated by four stages of deformation and projection.

3.2. Method of Generation of Grid for Inverse Problem

A grid in the brain is generated in order to calculate the distribution of the current source. First, a rectangular

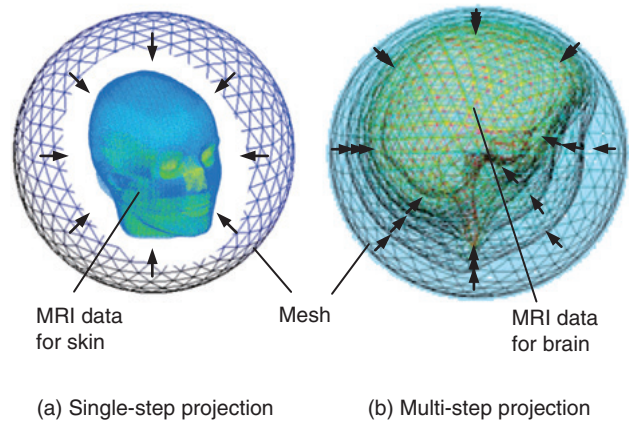


Fig. 4 Projection of spherical mesh

parallelepiped domain that includes the brain is covered by a grid, and the grid outside the brain is deleted. The values of the solid angles are used in judging interior and exterior. The solid angles of the grid points in relation to the brain surface mesh are expressed by Eq. (4). Here, $\mathbf{r}_{i1}$, $\mathbf{r}_{i2}$, and $\mathbf{r}_{i3}$ are position vectors in relation to the grid point of the i^{th} triangular element node, and N is the number of elements in the brain surface mesh.

$$\Omega = 2 \sum_{i=1}^N \tan^{-1} \frac{\det(\mathbf{r}_{i1}, \mathbf{r}_{i2}, \mathbf{r}_{i3})}{|\mathbf{r}_{i1}| |\mathbf{r}_{i2}| |\mathbf{r}_{i3}| + |\mathbf{r}_{i1}| (\mathbf{r}_{i2} \cdot \mathbf{r}_{i3}) + |\mathbf{r}_{i2}| (\mathbf{r}_{i3} \cdot \mathbf{r}_{i1}) + |\mathbf{r}_{i3}| (\mathbf{r}_{i1} \cdot \mathbf{r}_{i2})} \quad (4)$$

When the grid point is in the brain surface mesh, the solid angle is 4π , when the grid point is on the boundary, it is 2π , and when the grid point is outside, it is 0. Figure 5(a) shows a grid covering a rectangular parallelepiped domain containing the brain. Figure 5(b) shows results with the grid outside the brain and the brain surface grid removed.

3.3. Calculation of Lead Field

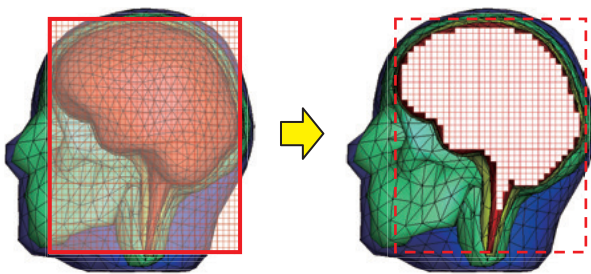
A variety of numerical analyses that are able to consider the shape and electromagnetic properties of the human head are available, including the finite difference method, the finite element method, and the boundary element method (BEM). We adopted a BEM for its calculation accuracy with minimal unknowns.

Equation (5) is Maxwell's equation, which expresses the electromagnetic field.

$$\begin{cases} \nabla \times \mathbf{E} = -\frac{\partial \mathbf{B}}{\partial t}, & \nabla \cdot \mathbf{D} = \rho \\ \nabla \times \mathbf{H} = \frac{\partial \mathbf{D}}{\partial t} + \mathbf{J}, & \nabla \cdot \mathbf{B} = 0 \end{cases} \quad (5)$$

Here, $\mathbf{E}$, $\mathbf{H}$, $\mathbf{B}$, $\mathbf{D}$, $\mathbf{J}$, ρ and t are the electric field, the magnetic field, magnetic flux density, electric flux density, current density, charge density, and time, respectively. Terming the current density in the brain $\mathbf{J}^i$ and conductivity σ in order to consider current in the brain, the current density, $\mathbf{J}$, can be expressed by Eq. (6).

$$\mathbf{J} = \mathbf{J}^i + \sigma \mathbf{E} \quad (6)$$



(a) Full grid

(b) Trimmed grid

Fig. 5 Grid for inverse problem calculation

Taking the fact that current in the brain is a low-frequency signal of around several hundred Hz and applying a quasi-static approximation, Eqs. (5) and (6) can be reorganized to produce Eq. (7), which expresses the electromagnetic field of the current in the brain.

$$\nabla \times \mathbf{H} = \mathbf{J}^i - \sigma \nabla V \quad (7)$$

Here, V is electric potential. Using vector equations and Green's theorem, Eq. (7) is transformed into a surface integration equation. The surface integration equation used to determine electric potential is shown as Eq. (8). The surface integration equations used to determine the electromagnetic field are the simultaneous equations shown as Eqs. (8) and (9)⁽³⁾.

$$\frac{\alpha_k(\mathbf{r}) + \beta_k(\mathbf{r})}{2} V(\mathbf{r}) = \frac{1}{4\pi} \frac{\mathbf{J}^i(\mathbf{r}') \cdot (\mathbf{r} - \mathbf{r}_s)}{|\mathbf{r} - \mathbf{r}_s|^3} - \sum_{j=1}^m \frac{\alpha_j - \beta_j}{4\pi} \int_{S_j} \frac{V(\mathbf{r}') \mathbf{n}(\mathbf{r}') \cdot (\mathbf{r} - \mathbf{r}')}{|\mathbf{r} - \mathbf{r}'|^3} dS_j \quad (8)$$

$$\mathbf{H}(\mathbf{r}) = \frac{1}{4\pi} \frac{\mathbf{J}^i(\mathbf{r}') \times (\mathbf{r} - \mathbf{r}_s)}{|\mathbf{r} - \mathbf{r}_s|^3} - \frac{1}{4\pi} \sum_{j=1}^m (\alpha_j - \beta_j) \int_{S_j} \frac{V(\mathbf{r}') \mathbf{n}(\mathbf{r}') \times (\mathbf{r} - \mathbf{r}')}{|\mathbf{r} - \mathbf{r}'|^3} dS_j \quad (9)$$

Here, $\mathbf{r}_s$ is the position vector of the current source, $\mathbf{n}(\mathbf{r})$ is the surface normal vector at $\mathbf{r}$, α is the conductivity of the internal contiguous tissue, β is the conductivity of the external contiguous tissue, S is the mesh faces, j and k are the numbers applied to the mesh faces, and m is the total number of mesh faces. A comparison of Eq. (1), in which the brain is approximated to a sphere, and Eq. (9) shows that it is the second term on the right-hand side of Eq. (9) that reflects the shape and electromagnetic properties of the tissue in the head.

The lead field can be determined by discretizing Eqs. (8) and (9), applying them to the human mesh and solving them using BEM. If the current at each grid point, $\mathbf{J}^i$, is assumed to be a unitary current, V determined from Eq. (8) becomes the lead field of an EEG. Similarly, $\mathbf{H}$ determined by coupling Eqs. (8) and (9) becomes the lead field of an MEG. Terms unrelated to the position of the current are stored in a coefficient matrix and LU decomposition is applied, and the solution is found through reverse substitution and forward substitution for each current source.

3.4. Method for Inverse Problem

Using the lead field, $\mathbf{L}(\mathbf{r})$, the relationship between the strength of the current source at position $\mathbf{r}$, $s(\mathbf{r}, t)$ and measurement data, $\mathbf{b}(t)$, can be expressed by Eq. (10).

$$\mathbf{b}(t) = \int \mathbf{L}(\mathbf{r}) s(\mathbf{r}, t) d\mathbf{r} \quad (10)$$

Here, measurement data refers to measurement data

for the electric potential, V , in the case of EEG and measurement data for the electromagnetic field, $\mathbf{H}$, in the case of MEG. Using a minimum-variance spatial filter, a technique that is often applied in EEG and MEG inverse problems, the strength of the current source is reconstructed via Eq. (11).

$$\hat{s}(\mathbf{r}, t) = \mathbf{W}^T(\mathbf{r})\mathbf{b}(t) \quad (11)$$

Here, $\hat{\cdot}$ is used as a superscript to distinguish $\hat{s}(\mathbf{r}, t)$ from true $s(\mathbf{r}, t)$. $\mathbf{W}(\mathbf{r})$ is a weight vector for reconstruction. $\mathbf{W}(\mathbf{r})$ is determined so as to minimize the effect on current sources in other positions when the sensitivity to the current source in position $\mathbf{r}$ is 1. It is found by minimizing Eq. (13) based on the constraint conditions given by Eq. (12).

$$\mathbf{W}^T(\mathbf{r})\mathbf{L}(\mathbf{r}) = 1 \quad (12)$$

$$\mathbf{W}^T(\mathbf{r})\langle \mathbf{b}(t)\mathbf{b}^T(t) \rangle \mathbf{W}(\mathbf{r}) \quad (13)$$

Here, $\langle \cdot \rangle$ represents the time average. $\mathbf{W}(\mathbf{r})$ is expressed by Eq. (14).

$$\mathbf{W}(\mathbf{r}) = \frac{\mathbf{R}^{-1}\mathbf{L}(\mathbf{r})}{\mathbf{L}^T(\mathbf{r})\mathbf{R}^{-1}\mathbf{L}(\mathbf{r})}, \quad \mathbf{R} = \langle \mathbf{b}(t)\mathbf{b}^T(t) \rangle \quad (14)$$

Using $\mathbf{W}(\mathbf{r})$, the current source is reconstructed from Eq. (11)⁽⁴⁾.

3.5. Application of Sub-grid Technique to Inverse Problems

The refinement of the grid of inverse problems is useful for the enhancement of the position accuracy of the solution. On the other hand, the calculation time increases due to the increased number of calculation points. In order to address the issue, we applied the sub-grid technique used in forward problems to inverse problems. The sub-grid technique is able to limit the refinement to necessary areas only. First, inverse problems were solved for a coarse grid in order to determine the positions of high current source strength. Following this, the areas of the grid contiguous to these positions were finely subdivided using sub-grids of a pitch of 1/2, and inverse problems were solved in the areas in which the sub-grid had been used in order to determine the positions of high current source strength in greater detail. When this operation is performed a predetermined number of repetitions, the positions of high current source strength can be determined with good accuracy. The grid points in the areas in which the sub-grids were generated at which the first-order differentials of the current source strength in all directions are positive are then determined. Following this, a fixed number of points are determined in order of current source strength. Figure 6 shows a flow chart of the application of the sub-grid technique to inverse problems. Here, N_{max} is the maximum number of iterations of the grid.

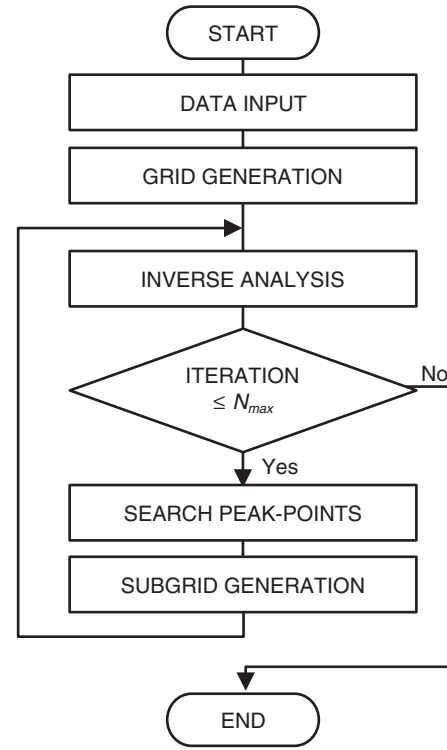


Fig. 6 Flowchart of inverse problem analysis with sub-grids

4. Verification of Effectiveness of System

In order to verify the effectiveness of the system, generating currents freely in an actual human brain is ideal but effectively impossible. We therefore adopted a numerical simulation for the verification⁽⁵⁾. The simulation assumed a 210-channel MEG sensor in the vicinity of the head, and three current sources with differing coordinates, directions and waveforms in the brain. The electromagnetic field at the position of the sensor was calculated in advance as a forward problem. This value was regarded as the sensor signal, and inverse problems were solved in order to verify the reconstruction of the current sources. Figure 7 shows

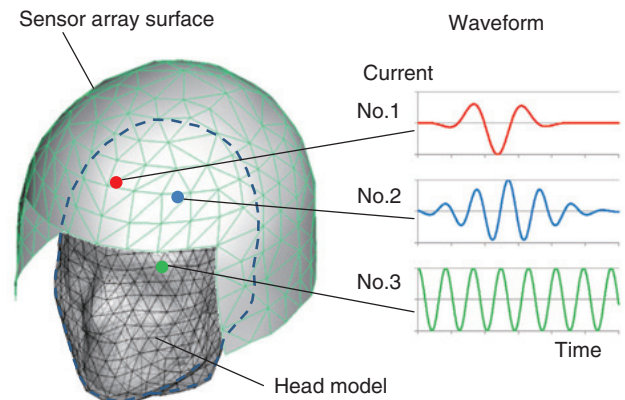


Fig. 7 Model of inverse problem analysis

Table 1 Position and direction of sources

Source No.	Position [mm]	Direction
1	(5, -17.5, 57.5)	(1, 0, 0)
2	(5, 20, 57.5)	(1, 1, 0)
3	(5, 12.5, 20)	(0, 1, 1)

Table 2 Waveform of sources

Source No.	Wave form
1	$\exp\left(-\left(\frac{t-0.47}{1.2/5.9}\right)^2\right) \sin\left(\frac{2\pi \cdot 3.8 \cdot (t-0.556)}{1.2}\right)$
2	$\exp\left(-\left(\frac{t-0.54}{1.2/3.8}\right)^2\right) \sin\left(\frac{2\pi \cdot 6.3 \cdot (t-0.492)}{1.2}\right)$
3	$\sin\left(\frac{2\pi \cdot 7.3 \cdot (t-0.455)}{1.2}\right)$

the head model, the sensor array surface, and the current sources. The electromagnetic sensor was positioned at the nodal points of the sensor array surface mesh. The head model was the four-layer head model discussed in Chapter 3. Table 1 shows the position and direction of the current sources, and Table 2 shows their waveforms. The front, left and top of the head from the perspective of the test subject were the positive directions of the x, y and z axes, respectively, and the center of gravity of the head was the origin point of the coordinates.

Calculations for analysis of inverse problems were compared using the following two types of grids. In each case, the minimum grid pitch was 2.5 mm.

- (1) Analysis of inverse problems using a grid with a pitch of 2.5 mm (Without using sub-grids)
- (2) Analysis of inverse problems using a grid with a pitch of 10 mm (Two iterations for the sub-grids)

Figure 8 shows the two types of grid. The grids in the head mesh are shown as cross-sections.

Table 3 shows the calculation results. Both of the grids were able to reconstruct all three current sources in their simulated positions. The application of sub-grids has made it possible to reconstitute current sources using a coarse grid with a pitch of 10 mm with the same level of accuracy as a grid with a pitch of 2.5 mm. In addition, the use of the

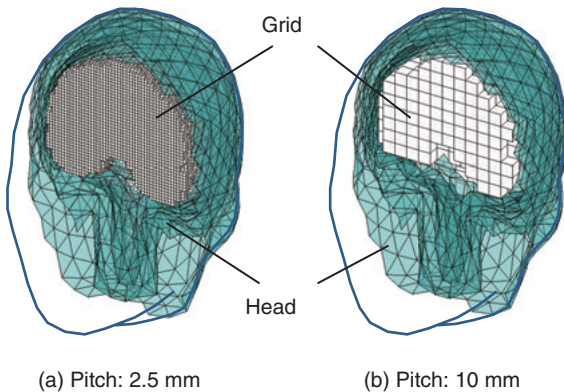


Fig. 8 Grids for inverse problem analysis

Table 3 Results of reconstruction of the sources

Pitch of grid [mm]	2.5	10 (+Sub-grid)
Position [mm]	No.1	(-5, -17.5, 57.5)
	No.2	(-5, 20, 57.5)
	No.3	(5, 12.5, 20)
Calculation time [sec]	4101	190

rough grid reduces calculation time to approximately 1/22, with calculations being completed in 190 seconds, against 4,101 seconds for the 2.5 mm grid.

Figure 9 shows the calculation results. The red points show the points at which current density is high, indicating that the three current sources have been reconstructed. Figure 9(b) shows a 2.5-mm sub-grid being automatically generated around the current source.

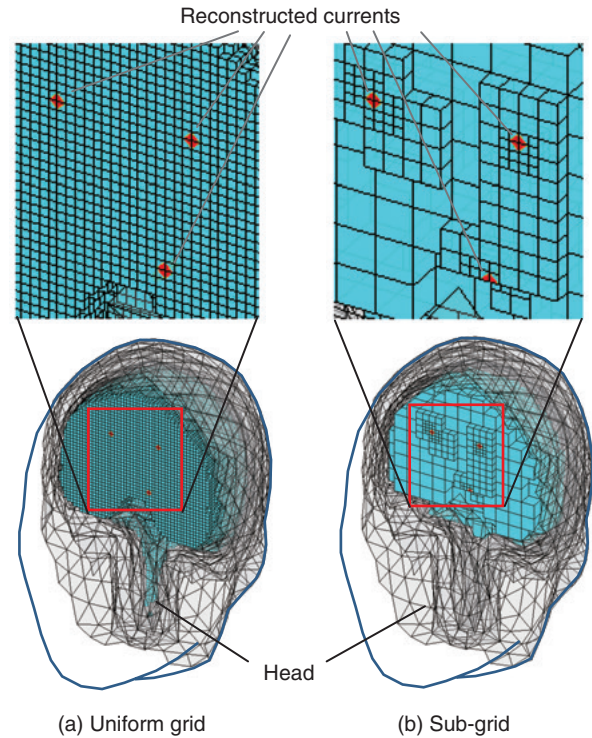


Fig. 9 Reconstructed currents in brain

5. Conclusion

An inverse problem analysis system for electromagnetic fields that reconstructs current sources in the brain has been developed. In order to address the issues of the conventional calculation method, which approximates the head to a sphere, this project applied a method in which the lead field was calculated using BEM, and developed the human head mesh model, mesh generation method, and generation method for the inverse problem calculation grid. In addition, a minimum-variance spatial filter was

employed as the method for inverse problems and a sub-grid technique was applied, increasing the speed of calculations. The effectiveness of the system was verified using a numerical simulation that reconstructed currents in the brain. When the sub-grid technique was applied to a grid with a pitch of 10 mm, the same level of position accuracy in the solution was obtained as a grid with a pitch of 2.5 mm at approximately 1/22 the calculation time. The developed system is able to reconstruct current sources with consideration of the shape and electromagnetic properties of a human head. The system applies the sub-grid technique to the solution of inverse problems for magnetic fields, but it is applicable to inverse problems in a variety of fields, including fluid analysis, structural analysis, and acoustic analysis. The extension of the system to a wide range of fields is scheduled for the future.

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■ Author ■



Masatoshi HARUMATSU



Katsuya YASHIRO



Kensuke SEKIHARA

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— Editorial Afterword —

Five years has flashed by since I first became involved in editing the Honda R&D Technical Review. In that time, the greatest change that I have witnessed is electronic publishing. Papers that would previously not even be seen by everyone at Honda R&D itself are now available free of charge not only to all employees, but also to Internet users throughout the world. Our purpose in this venture is to promote worldwide exchange with engineers through the medium of technical papers.

Commencing publication on the Internet involved discussion of a variety of issues: whether to charge or to publish for free, how to treat confidential matters, what modes of communication to use, and more.

We consulted on the questions that came up with experts on patents and intellectual property, and especially with the Corporate Communications Division of Honda Motor Co., Ltd., which boasts extensive experience in operating a website. Having found appropriate answers to each of these questions, we were able to realize our electronic publishing venture.

After we did start publishing, a wide range of people registered to access our papers, from Honda employees to employees of related parts makers, employees of other companies working in the automotive industry, researchers active in universities and research institutions, and students, providing a good omen for our future development.

On our part, we have enhanced our search function (the single greatest merit of using a non-paper medium), added functions to rank the most popular papers and introduce related papers, and we are working to create a user-friendly page layout.

As I indicated above, our sole purpose in all of this is to work for the mutual benefit of engineers inside and outside Honda by promoting exchange between them. I would be delighted if our readers decide to join the exchange by clicking on the “We would like to get your opinion on this research paper.” buttons to be found on the summary page of each paper.

— Ishibashi —

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