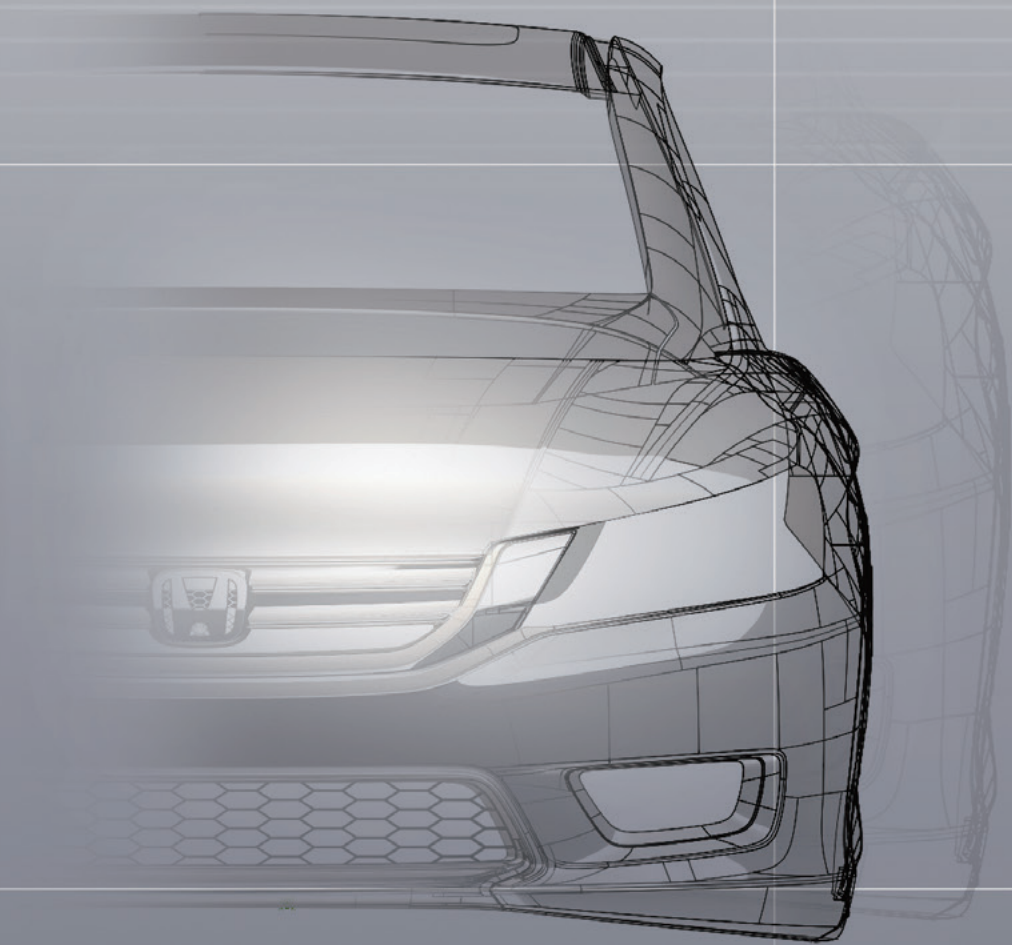
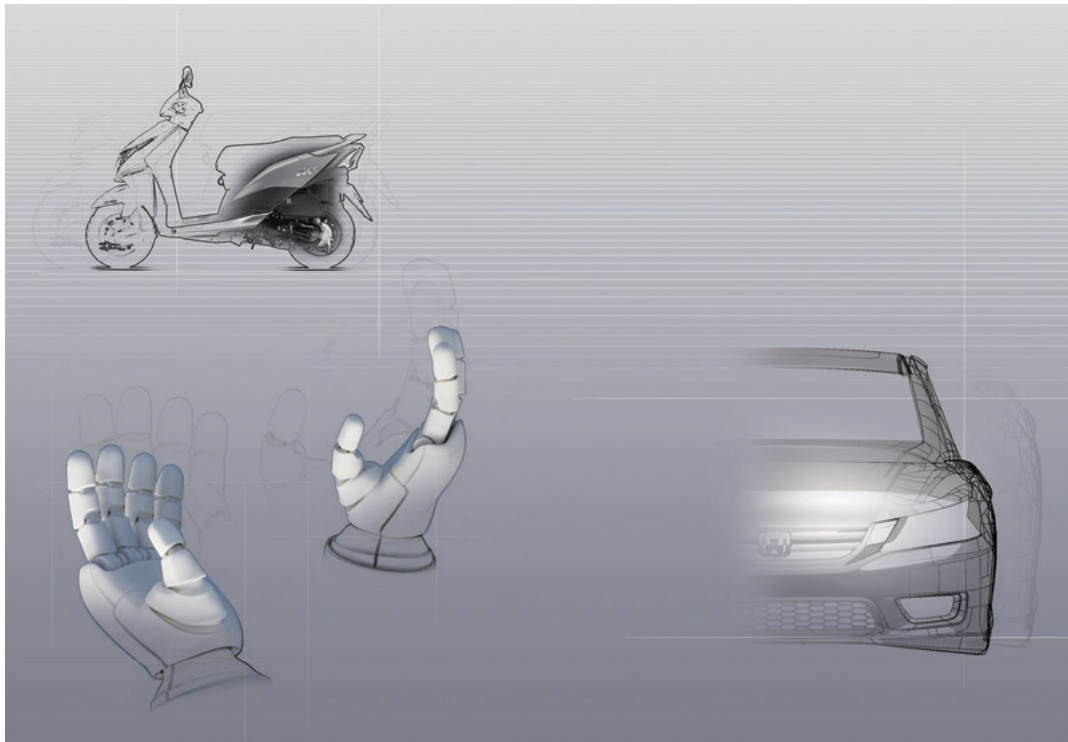


Honda R&D

Technical Review



April 2013
Vol.25 No.1



Comment on the Cover

Human-centered engineering. This concept is deeply rooted at Honda.
For a richer life. For a better world.

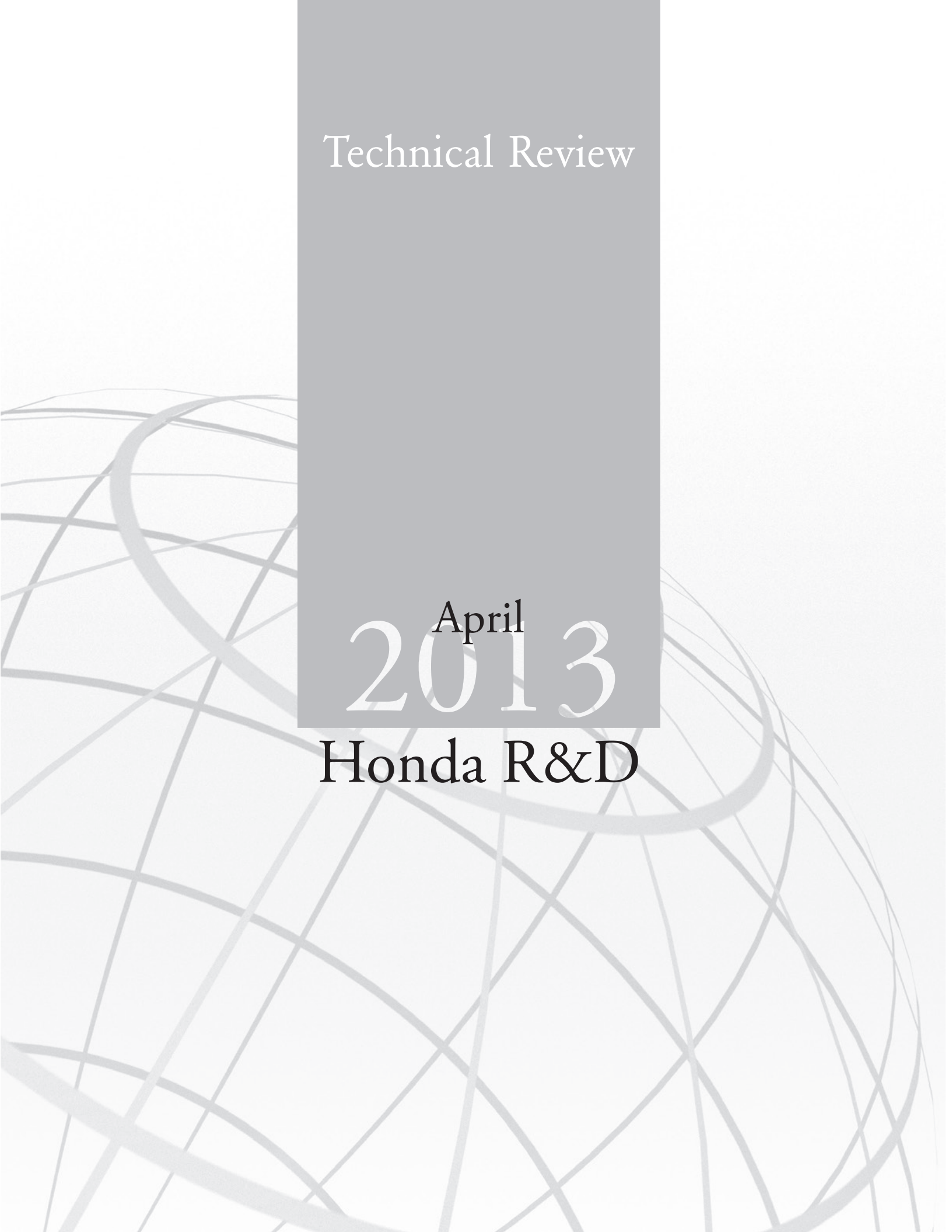
Even amid difficult circumstances, whether acts of nature or economic downturns, we never forget our founding goals, and our aim is engineering of the highest quality.

Using our latest technologies as horizontal and vertical lines, the cover design of this edition provides an image of the technological details provided within its pages.

We hope that you will take this opportunity to learn about the many exciting new technologies born from Honda's commitment to excellence.



Automobile R&D Center
Masahiro Sagawa

The background features a light gray wireframe sphere on the left side. A solid gray vertical bar is positioned in the center, containing the text. The overall color palette is monochromatic, using shades of gray and white.

Technical Review

April
2013

Honda R&D

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Local Styling Design of Dio for Indian Market

Kazuhiko SAITO* Takashi ARAI*

ABSTRACT

Along with high economic growth and modernization, people in India can get easier access to the Internet and other means of communication to instantly acquire information from around the world. Younger people in India in particular are highly conscious of trends in the developed nations and hunt for advanced styling design for motorcycles and scooters as well. This time, the novel design was created to be applied for full model change of the Dio after 10 years since its first release, by consolidating the styling element of sporty motorcycles, which is what young folks in India are yearning for, into the styling form of a scooter. This was also the first model which experienced a full model change under the major involvement in styling design work from Indian local research and development branch of Honda R&D India Pvt. Ltd.

1. Introduction

With a continuing economic growth, the market for motorcycles and scooters more than tripled in a decade, from 3.87 million units in 2000 to 11.82 million in 2010⁽¹⁾. It grew another 17% in 2011 to 13.84 million units. Within the market for that percentage, scooter category was 1.11 million units in 2000, expanding about 70% to 1.92 million in 2010, followed by a further increase of 22% to 2.4 million units in 2011.

The Dio of first-generation (Fig. 1) was launched in September 2000 as the dedicated model for India targeting young people. Scooters in India at the time were regarded as vehicles only characterized by simplicity and fortitude, and were considered as transportation for a practical family use. In the demographical compositions in India⁽²⁾, young people in late teenage and early twenties represent the largest population group. However, there had been almost no models targeting those younger generations.

In such a market situation, the Dio has been keeping the

successful status with its stable sales, earning reputations from students and working women because of its stylish design as well as lightweight and compactness. That resulted in establishing the unchallenged position as the only one scooter in the market which owns sporty styling. However, India has made significant changes in its culture and life style, through experiencing the economic growth, the proliferation of Internet-use and the advances of foreign business entities into India from Europe, the U.S.A. and other Asian countries. In those transitions, there has been a significant change in preferences of our targeting younger people, and the sales of the Dio had been suffering without additional growth despite of the market growth as a total. That was because the styling of Dio had become less attractive with matured familiarity gained through a lapse of 10 years after its release.

The full model change of the Dio, by Indian local initiative, had been planned at the opportunity of the model change of the Activa (Fig. 2), which shared the engine and the frame with the



Fig. 1 Dio 2000 model year



Fig. 2 Activa 2008 model year

* Motorcycle R&D Center

** Honda R&D (India) Pvt. Ltd.

Dio, applying the 109 cm³ fuel efficient engine. Accordingly, Honda R&D India Pvt. Ltd. has got the responsibility to create the styling that fit to young people of India, which are the same target group that the initial model was aiming at, by reflecting new interpretations to the current era.

2. Concept

The Activa, the best-selling scooter in India, has become the benchmark model that anyone can easily ride and purchase with confidence. Young customers, however, have avoided it because of its conservative design, which is intended for family use. Motorcycles and scooters that appeal to young people in India are sports motorcycles like the CBR1000RR and VFR1200F (Fig. 3), a trend that is further evidenced by the arrival in India of several large-motorcycle companies, one after another, from North America and Europe. Those motorcycles are extremely expensive, and as products they are not affordable to anyone but very affluent customers. The Dio developed this time was intended to appeal to young people by offering the image of the sports motorcycles they like but in the form of an affordable scooter. Therefore the model concept chosen was “Youngster’s Honda Sporty Scooter.” A styling sketch of the newly developed Dio is shown in Fig. 4.



Fig. 3 VFR1200F 2010 model year



Fig. 4 Styling sketch showing “Layered design” and “Wedge shape”

The sales price is also important since the model targets young people, but practices like sharing components with mass-produced models give the Dio the ability to offer an affordable price range. Component sharing is a great advantage in terms of service and maintenance in India, where logistics systems can be weak.

In terms of performance, the use of a new 109 cm³ engine has enhanced output, giving the Dio top-class performance as a sporty scooter. Additionally, consumers are particularly interested in fuel efficiency in India, because gasoline is so expensive there. The fuel efficiency of the Dio has been enhanced by 15%, so its major appeal points are performance, fuel efficiency and price.

As for equipment, it uses a combined brake system (Combi-Brake)⁽³⁾, such that the front and rear brakes work in conjunction with each other. As a result, the Dio is not at all inferior to a model with a larger displacement engine, so it appeals to customers with its safety and advanced performance.

3. Styling

3.1. Styling Concept

“Moto-scooter,” a term that signifies the combination of a motorcycle (which appeals to the tastes of young people) and a scooter (which is more accessible), was designated as the styling concept to help embody the model concept, “Youngster’s Honda Sporty Scooter.”

As a method for achieving such styling, keywords like “mass concentration,” “layered form” and “wedge shape” were used in order to incorporate the image of an up-to-date Honda sports motorcycle. In concrete terms, design methods used for the VFR1200F and CBR1000RR⁽⁴⁾ were incorporated, but with a new interpretation for a scooter.

“Mass concentration form” is a method that evokes a look of volume around the engine area while making the rear appear more nimble. On scooters, however, the body volume is divided between the front and rear, with the floor between them, making it difficult to apply this method as is. Thus, the sports motorcycle mass concentration form design motif was incorporated into each of the major components, presenting the image of a “moto-scooter” overall.

3.2. External Appearance of the Body

The challenge in this process was to express the design of a sports motorcycle’s front cowl, its most distinctive part, in the body of a scooter. An actual example of the 2012 model of the Dio is shown in Fig. 5. Scooters are often shaped by design method to feature continuity from front to rear, but the Dio divides the exterior parts into several different elements to project the cowl image of a sports motorcycle. The leg shield is divided into two parts, with a dividing line that is diagonal in relation to the overall leg shield, creating a “layered” design with three-dimensional overlap. This “layered” part provides the function of deflecting the air stream that hits the body to both sides

behind the front wheel. In addition, each of the divided parts features a different color, enabling a distinctive design that draws attention to the layered parts.

The “wedge” shape is a basic element of the design shape, giving a sense of speed and sharpness, but traditionally it has been challenging to present a sharp image on a scooter, which has a large amount of volume in the rear body. On the Dio, the rear body cover, which is of the main body color, was made visually independent as a compact and sharp “wedge” shape. The look of nimbleness and sportiness is also brought out by the shape that boldly sweeps upward from the floor side cover to the rear body cover. Additionally, the portion running from the floor step to the rear body cover is made with black-colored plastic, giving a strong contrast and making the visual compactness and the distinctive shape of the body cover stand out more clearly. The rear grip is made of die-cast aluminum, which is easy to form freely. With a shape that integrates it with the rear body cover, it forms a “wedge”-shaped silhouette that reaches upward from the rear of the body. By assuming a form like that of a spoiler, it serves not only as a part with a practical function, but also as a key styling element for presenting a look of sportiness.

3.3. Exterior Design of Functional Parts

The form of functional parts is likewise not limited by the scooter frame. These parts have design elements found on sports motorcycles and were given a design to match the tastes of Indian users. The headlight was incorporated into the front cover and shaped to integrate with the body surface. The main body of the headlight borrows a reflector designed with a non-standard shape, such as is used on small sports motorcycles, while the lens face is unique to the Dio. The turn signals have clear lenses and a shape that connects them to the headlight, giving a sharp and distinctive front design (Fig. 6).

The form of the handle cover echoes the way the tip of the front cover tapers to a point, a design that evokes a sense of speed whether seen from the front or side. Scooters frequently have slits in the cover to facilitate transmission

of sound from the horn, but the Dio positions the horn to provide sufficient volume even without such openings. The result is a smooth, uninterrupted surface form.

The Dio continues to use the bottom-link front suspension that has such a good reputation for riding comfort, but the front fender has received a new form that matches the overall design. There is an edge running down the center of the front fender, providing a surface configuration that conforms to the front cover form. In addition, part of its form stretches downward to protect the cushion unit, which presents the image of the front fender of a sports motorcycle.

The bottom-link cover, located on the side of the front brake panel, was used as a part that could proactively assert the design, instead of the conventional practice of trying to keep it from standing out. With its wedge-shaped design, the bottom-link cover has a dynamic form that is conscious of its connection to the body.

The design around the tail part of the body also projects sporty and dynamic styling by integrating the turn signals and brake lamp and using a sharply modeled taillight full of edges. For users in India, who are looking for a sense of value greater than the price, a magnificent and opulent look is a very important element, so the reflectors inside the turn signals have a chrome finish. The combination of amber-colored bulbs with the cut shape of the clear lenses was intended to deliver a showy impression similar to that of the headlight (Fig. 7).



Fig. 6 Integrated headlight unit with turn signals



Fig. 5 Dio 2012 model year



Fig. 7 Integrated tail light unit

The twin analog meters, arranged side by side, give a sports motorcycle image based on the “Moto-Scooter” concept. The large circular speedometer on the left and the somewhat smaller circular fuel meter on the right create an aggressive expression with their asymmetrical form. The left and right meter dials are of different colors, reinforcing the asymmetrical form. The handle cover panel surface surrounding the meters has a dimpled texture to leave a sporty and functional impression. Dot graphics that create a dimpled image are used within the meters to unify their design with the panel (Fig. 8).

3.4. Design for Practicality

Scooters are a means of daily transportation essential to lifestyle, and so the enhancement of practicality was incorporated as an important element of the Dio along with presentation of the sporty styling. The floor was narrowed by as much as 20 mm to make it easier to plant the foot on the ground when stopped, enabling even smaller people to ride with confidence. The pillion steps for the passenger are each 10 mm wider on one side than in the first-generation model. This increases the surface area for resting the feet while making it easy to get on and off so passengers can ride with confidence, as well. Although these steps are functional, they were also actively incorporated into the styling. The design method shaped them to a form that integrates them with the body, as if they were wings spreading out from it. The overall slope of the floor was chosen to help riders maintain a more natural, relaxed posture. Whereas the first-generation Dio had a floor with a 13° slope, in the new model it is just 8°. A hook under the seat in the front provides a convenient place to hang a handbag, shopping bag, etc.

For users in India, it is common to have multiple riders on the same scooter, so back seat comfort is an important element. Particularly among women, it is common to wear unique Indian clothing like saris. Because wearers of such garments cannot straddle the seat, they often sit sideways on it (Fig. 9). Thus the seat width was made the same as

the Activa, a scooter for families, while the slope of the seat was set at 3-4°, giving it a form that is easy to mount. The rear grip has a cross-section form that is easy to hold even when seated sideways and that even women with small hands can grasp comfortably and securely.

Under the seat within the body there is 18 L of trunk space, which is larger than the first-generation model. That is large enough to hold small personal belongings and bags when the scooter is being ridden, or one full-face helmet when parked.

Flat tires happen frequently in India because of bad road surface conditions, and it is quite common to see people by the side of the road repairing them. An important point for facilitating flat tire repair is to enable the rear wheel to be easily removed. The Dio has inherited the structure that enables the rear wheel to be taken off without removing the muffler, facilitating repairs. In addition, the adoption of tubeless tires increases safety when flat tires occur.

3.5. Coloring

Five color variations are available, including standard colors and dynamic colors unique to Dio and chosen to appeal to young people (Fig. 10). In the Indian market,



Fig. 8 Concept sketch of asymmetrical meter panel design



Fig. 9 Female passengers in saris



Fig. 10 Color lineup sketch by 3D-CG

achromatic colors like black and dark silver are used as the standard colors of many models. Because the Dio is targeted at young people, matte colors such as those popular for automobiles in Europe were the focus here, which led to the adoption of Mat Axis Grey Metallic. This was the first attempt at using matte colors on a scooter in India, and as such it gives an advanced impression.

For the red version, a combination of Sports Red and Force Silver Metallic was used to make the color image of the VFR1200F more apparent. Using the same image color as used with Honda sports motorcycles emphasizes the sportiness of the Dio.

Since its use on the Activa, white has rapidly grown popular with motorcycle and scooter users in India. The Dio now shares Pearl Sunbeam White with the Activa, but that color, if used alone, creates an overly elegant impression. Therefore, it is used in combination with Mat Axis Grey Metallic in a two-tone color scheme characterized by sportiness and wildness.

For the green version, Candy Palm Green is used. This color is increasingly being used on small sports motorcycles in India and has a strong sense of presence on the street. This high-chroma color is quickly recognizable even from a distance.

Leblon Violet Metallic, a color that is interesting since it changes according to the intensity and angle of light, was chosen as the fifth color. Though close to purple, this is a high-impact color suited for the young people targeted by this model.

The three-dimensional “Dio” logo emblem, which is chrome-finished, and the concept-expressing “Moto Scooter” mark are placed on the body alone without any stripe, which makes the body form itself look distinctive. Figure 11 shows a target user with an actual vehicle.



Fig. 11 Target user and the Dio 2012 model year

4. Conclusion

Honda's competitors have also begun to release models that target young customers in India's growing motorcycle and scooter market, not just practical family models based on the conventional sense of values. The market is now at a stage where trend-setting designs are needed for these models. With the Dio, an advanced design has been created by taking the styling elements of up-to-date Honda sports motorcycles and fusing them in the scooter form. Moreover, the fact that this design was made in India is a major first step towards future full-scale local development.

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■ Author ■



Kazuhiko SAITO



Takashi ARAI

Styling and Design of New ASIMO

– Evolution for Continuing Engagement with People –

Takeshi KOSHIISHI*

ABSTRACT

The new ASIMO was developed as a step toward achieving “a robot that coexists and cooperates with people and performs useful functions in society.” The purpose was to make it capable of sensing changes in the circumstances of real-life environments where people are present, to take action autonomously, and to continue acting. It has new functionality from multi-axis hands capable of independently moving five fingers to enable manipulation of objects, legs and feet made more highly responsive for the purpose of continuing operation in mixed environments with people, and light-emitting displays at its ears and chest to induce interactions with people. With all this, a weight reduction of 6 kg was achieved without changing the robot’s total height.

This design assimilates the basic concept of “harmony between being advanced and being people-friendly” at a higher level. Placing importance on people’s perception of ASIMO, and taking into account its state of movement using 3D software, the design was carried forward with an emphasis on the three perspectives of gentleness, comprehensibility, and distinguishability. The result was achievement of a design that conveys the friendly image cultivated ever since ASIMO was first announced in 2000, and that embodies “evolution for the purpose of continuing engagement with people” by means of new functions and mechanisms.

1. Introduction

Honda has conducted research on humanoid robots as one innovative form of mobility that adds new value by functioning usefully in society through coexistence and cooperation with people. This work yielded the creation of ASIMO at the end of October 2000^{(1),(2)}, and it has enabled occasions for exchange with people in Japan and various other countries around the world. In December 2005, a model change was made for the purpose of enhancing the robot’s ability to perform autonomous work in offices and other such real-life environments as well as its other physical capabilities^{(3),(4)}. Operation tests were conducted of ASIMO as a reception guide and in a delivery service and other such duties, yielding deeper knowledge. One lesson learned was that people tend to take actions that are not envisioned in advance, and automatic machines that act in line with predetermined scenarios are not able to respond adequately to those actions. Therefore the new ASIMO was developed with the aim of achieving an autonomous machine with the capability for autonomous decision-making and behavior in changing circumstances so that it would be able to recognize people and sustain interactions with them in a real-life environment. The new model was

announced in November 2011⁽⁵⁾ (Fig. 1). Continuing pursuit of research and development led to the public unveiling of this ASIMO at the Honda Welcome Plaza Aoyama and other locations in 2012.

The demand made of design is that it achieves new functions and mechanisms fully as needed without losing the knowledge gained through the continued development of ASIMO through three generations over some 10 years or



Fig. 1 Humanoid robot New ASIMO

* Automobile R&D Center

the world view of people and ASIMO as a machine but one they can nevertheless have friendly contact with.

Here the design development of the new ASIMO will be introduced.

2. Basic Concept of ASIMO Design

Humanoid robots have forms that resemble people, but they are not people, they are machines. One approach is to design the robot as an artificial object kept always on the leading edge. In order to coexist with people, however, a design that is too radical gives a sense of incongruity in everyday living spaces, so that people will be unlikely to receive it sympathetically. A design that seeks to give a familiar appearance and aims for cuteness like that of a stuffed toy will be taken as trying too hard to please, and will also tend not to be well received.

The ultimate aim with ASIMO is to achieve “a robot that coexists and cooperates with people and performs useful functions in society.” This requires a design that gives a comfortable feeling of distance that takes the audience into consideration, neither too close to people nor too distant, so it can be approached with a friendly feeling. The optimal design will have a state of harmony between the robot’s advanced position as a machine and its friendliness as a partner to people. This is the basic concept of the “harmony between being advanced and being people-friendly” in ASIMO design (Fig. 2).

The quality of being advanced is something that changes over time, so it is evolved by means of new functionality, while the quality of being people-friendly fosters an affection that deepens over time. It is important to maintain these aspects in harmony with each other while assimilating them in the design at a higher level.

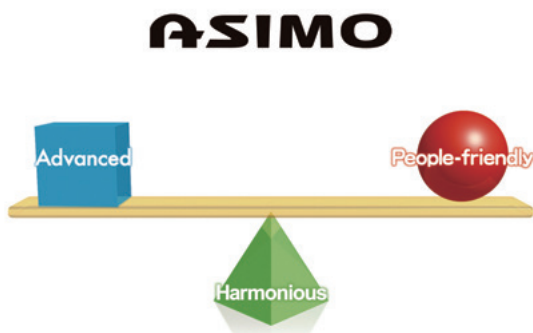


Fig. 2 Basic design concept

3. The Design Process

3.1. Design Techniques

A humanoid robot contains large numbers of mechanism parts (Fig. 3). The feasibility of the mechanisms is naturally a key consideration. Static and dynamic examination of all parts as well as of the whole is essential in order to produce

an appearance that is not incongruous when in motion as well as to give a sense of safety and security.

The new ASIMO has a total of 57 degrees of freedom, with the addition of one axis to the joint axes in the waist portion of the previous ASIMO and 11 axes to each of the two hands. In order for the robot to operate in close proximity to people, the gaps and misalignments between parts that occur in operation have to be drastically reduced, and the robot needs to be given an external appearance that does not expose its internal mechanisms and metal parts. ASIMO design development therefore brought together the styling designers and engineering designers as a combined team that used 3D software as a tool to carry out design work that took full account of the robot’s operational states. The state of movement was confirmed using animation by computer graphics (CG), and the robot’s presence in actual spaces was confirmed using full-size mockups, before completing fabrication of the actual machine. This enabled achievement of a highly precise design (Fig. 4).

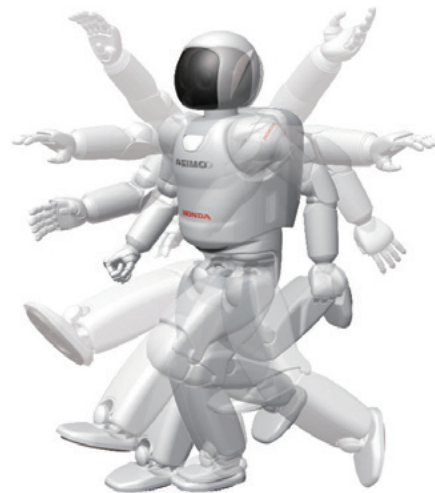


Fig. 3 Many movable parts in humanoid robot

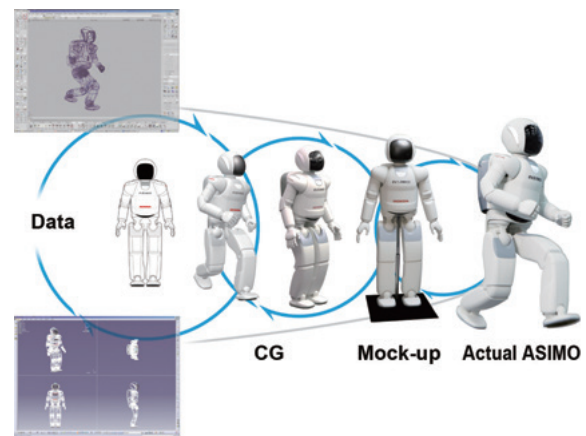


Fig. 4 Digital design process

3.2. Design Studies

Studies were made of the following aspects of ASIMO to determine how people felt about them.

(1) Interviews

Interviews were held with users who had actually operated the previous ASIMO at special events and other such occasions, and the following results were obtained:

Positive points

1. It has been made to evolve while carrying on the image it has acquired so far.
2. The displays that emit light using LEDs are a plus.
3. The overall form has rounded, giving it a soft appearance.

Negative points

1. There is something incongruous in the proportions.
2. It is not as cute, and it has become muscular seeming.
3. The backside looks like it is protruding, making it seem unstable.

(2) Perception of distinguishing characteristics

Drawings by children sent to Honda display tendencies showing that the following are perceived as three distinguishing characteristics of ASIMO:

1. Tinted shield over the face
2. Backpack
3. Geometric physique

(3) Comparative study proposal

Proposed designs include designs emphasizing the

quality of being advanced, emphasizing friendliness, suggesting gender difference, showing facial expressions, and so on. A number of such designs that convey an entirely different external image from the previous ASIMO were developed and interviews about them were held. The results in all cases were responses that the weaker continuity with designs up to this point made these designs seem incongruous for the new ASIMO.

3.3. Design Directions

Given these findings, ASIMO image can be considered to have become fixed to some extent by activities conducted since the robot was announced in 2000. In order to foster further relationships with people, it was determined to be undesirable for model changes to bring a completely different form and appearance. Priority was given to designs that have continuity as ASIMO, rather than being taken up by differences between old and new generations. It was decided to place importance on the friendly image while incorporating functional evolutions into the form. Figure 5 shows ASIMO generations for purposes of comparison.

4. Aims of the Design

The new ASIMO seeks to achieve “a robot that coexists and cooperates with people and performs useful functions in society,” and the design aims to embody “evolution that continues to relate to people.” The three perspectives of gentleness, comprehensibility, and distinguishability explained below were emphasized in developing this design.

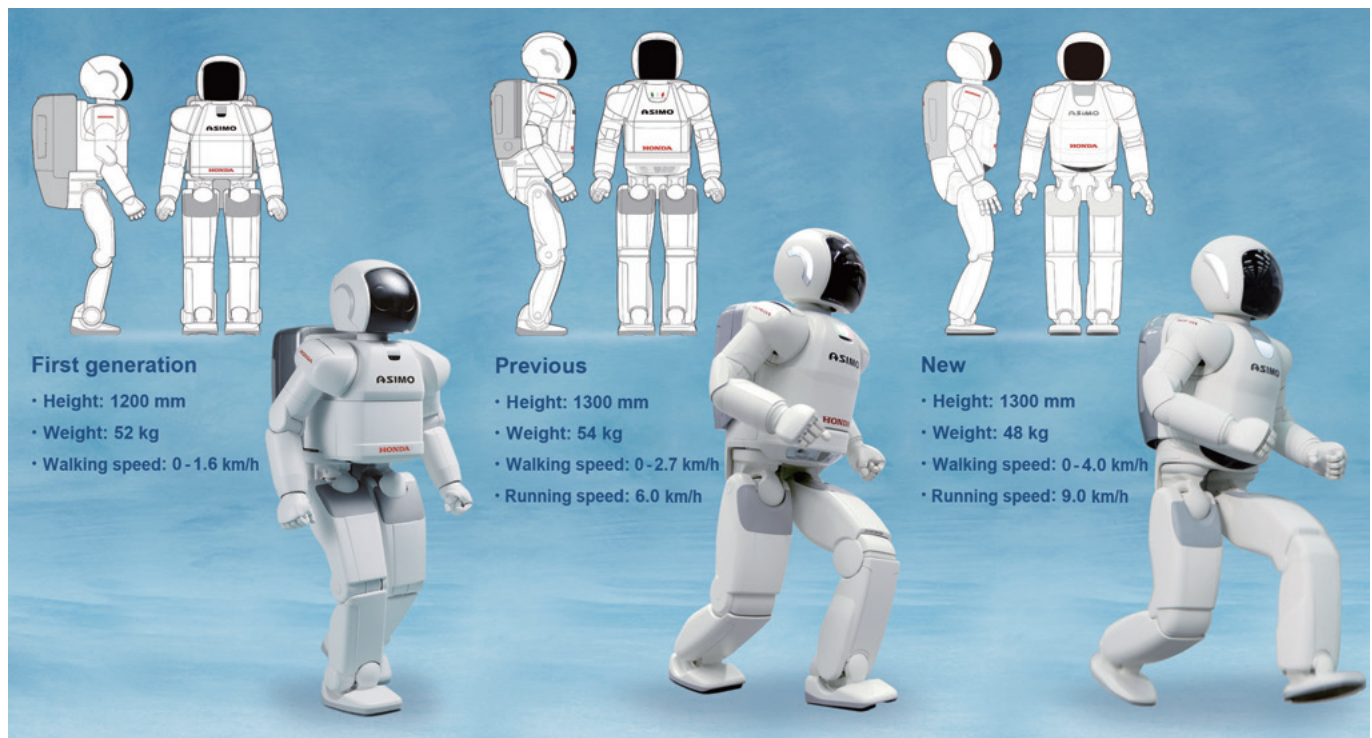


Fig. 5 Evolution of ASIMO

4.1. Gentleness

The roundness of the entire body was heightened relative to the previous ASIMO and the friendliness was increased. Structural lines that appear mechanical and complicated were put in order for a soft, gentle configuration. Parts with functions, such as the LED panels for the ears and chest were designed to embody their functions sharply and clearly. Details of each area are as follows:

(1) Proportions

The offset of head, shoulders, and legs that is created to balance the weight of the upper body was 40 mm in the previous ASIMO. In the new ASIMO this was reduced to 20 mm, enhancing the posture that had felt incongruous [Fig. 6 (a)].

(2) Torso

Efficient placement of the batteries and reduction in size of computers and other internal parts yielded a slim appearance. Compared to the previous ASIMO, there was a reduction of 30 mm on the back and 10 mm in the abdomen, for a total of 40 mm [Fig. 6 (b)].

The waist area was provided with a yaw axis for balance when running, as incorporated from the previous ASIMO, and a pitch axis was also added [Fig. 6(c)]. This also expanded the robot's expressiveness as well as its range of operation when grasping an object. The design was made to minimize the divergence of torso and waist when both these axes are in operation simultaneously (Fig. 7).

The lower part of the backpack on the rear of the robot was equipped with a heat vent and a power port for automatic charging in a unified design (Fig. 8).

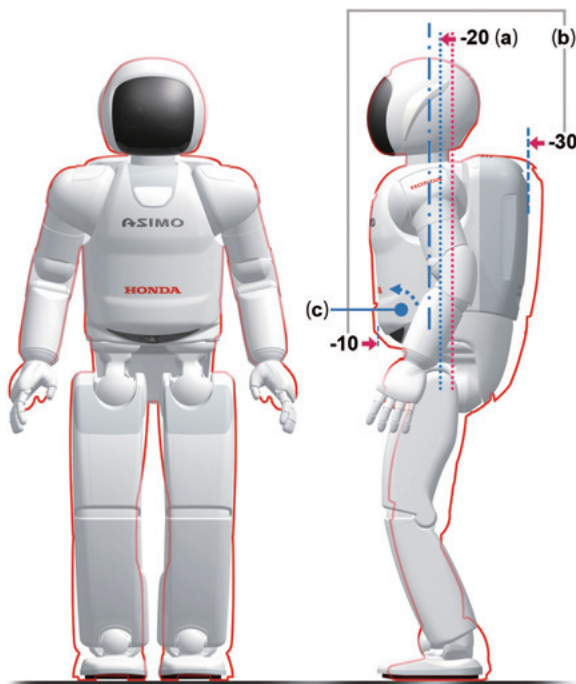


Fig. 6 Proportion (Red line: previous ASIMO)

The available angles of motion in the neck were increased from 22 degrees up and 19 degrees down in the previous ASIMO to 26 degrees up and 32 degrees down. This was an increase by a factor of 1.7. The design was made to avoid interference of the lower head, upper back, and chest, and to have the chest out.

(3) Head

ASIMO's face is covered with a shield that is tinted to a percentage that suggests the presence of eyes. The aim is to not reveal the facial features clearly so that viewers will be able to freely imagine ASIMO's expression.

The new ASIMO design projects greater friendliness by increasing the roundness of the head as a whole. The ratio of the vertical and horizontal components of the shield was brought closer to unity and the four corners were rounded. The shape of the profile was also given greater roundness.

The cameras that are placed in the position of eyes inside the shield have been spaced farther apart and moved down in order to achieve the effect of a mild expression. In order to assure the angular field of view of the cameras, the protrusions and indentations on the face have been toned down. The molded lines that can sometimes be glimpsed as corresponding to eyebrows and a mouth, depending on



Fig. 7 Rotatable waist



Fig. 8 Backpack

the angle of the light, have been made wider and shorter in order to provide greater expression.

The ears have the function of displaying how well the robot is hearing by emitting LED light. The amount the ears protrude from the head has been increased relative to the previous ASIMO so that they are more visible from the front when emitting light. The microphones that are placed to collect sound from all directions for voice recognition are concealed from outside view by arranging them at eight locations around the head inset within slits made at places where parts of the head are joined together. Figure 9 shows a line drawing of ASIMO head.

(4) Arms

The previous ASIMO had hands with a two-axis structure that were only capable of opening and gripping. In the new ASIMO, hydraulics are used to make the five fingers of each hand into independently movable mechanisms. The range of movement of the thumb has also been increased. Tactile sensors are embedded in the fingertips. These enable the robot to remove the lid from a flask and pour a beverage into a cup (Fig. 10), use sign language expressions, and so on. The design is a human fingertip motif with added roundness. The joints were also

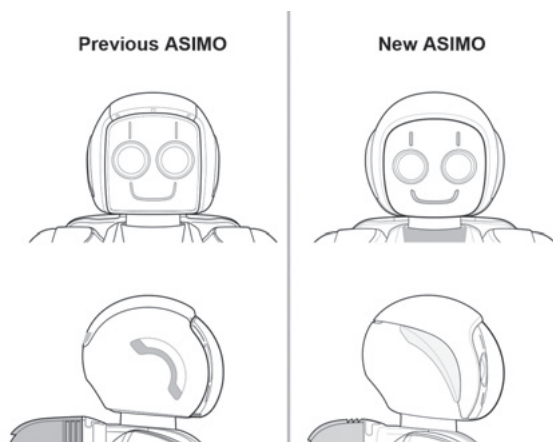


Fig. 9 Line drawing of head



Fig. 10 Action that opens lid of flask

formed so that the internal mechanism is not exposed even when the fingers are bent (Fig. 11).

The wrists, which had an elliptical cross-section in the previous ASIMO, were given a round cross-section in the new robot, and the available angles of motion have been increased.

The arms are structured to hold tubing that hydraulically transmits the driving force of motors built into the chest to move the fingers. They were given a unified design in which no gaps or misalignments occur between parts even when the elbow is bent.

(5) Legs and feet

The new ASIMO was given significantly upgraded physical capabilities, including a walking speed of 9 km/h and the ability to jump with both feet.

The design of the legs and feet as a whole was basically simplified with a smooth surface. The knees are equipped with rotating exterior facing mechanisms that are coupled to the movement of the knees. Sliding exterior facing mechanisms were also developed for the heels, and the available angles of motion for the ankles were expanded from ± 24.5 degrees in the previous ASIMO to ± 32.5 degrees. The feet were rounded at all four corners to eliminate interference and enable larger movements of the feet (Fig. 12).

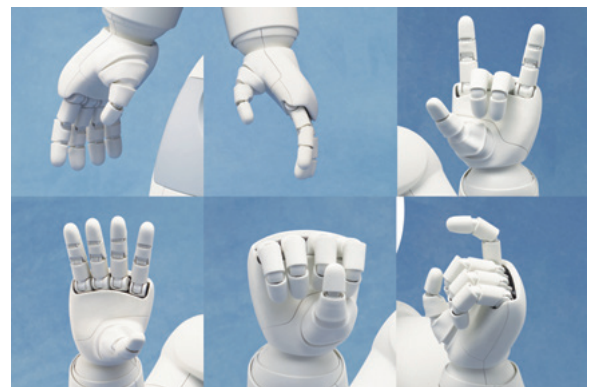


Fig. 11 Compact, hydraulic multi-axis hand



Fig. 12 Legs that move more freely

4.2. Comprehensibility

As occasions for ASIMO and people to have encounters and interact directly increased in number, circumstances began to occur in which people would wonder whether or not ASIMO was listening to them, or whether or not it was in a state ready for them to talk to it, and in which people could not readily know what state ASIMO was in at that time.

The robot's expressions made with gestures, therefore, were augmented by supplementary use of light emission by LEDs. The aim was to draw people in by using changing lights to provide opportunities to initiate interaction. It was decided, therefore, to implement displays by readily understandable light emission in locations that are readily understandable to people in an intuitive manner, as described below.

- (1) Light-emitting display at the front of the top of the head indicates that the robot is aware

The robot was given the capability to react to a person's inquiry by emitting continuous light as though flowing. The light is projected from behind the exterior facing so that people will not sense the presence of LEDs when the lights are not on.

- (2) Light-emitting display at the front of the ear indicates how the robot is hearing

ASIMO makes these lights blink when it is ready to listen. This gives people active encouragement. The use of phased lights in accordance with the robot's sound collecting state also indicates the level at which ASIMO is listening to a person's voice.

- (3) Light-emitting display in chest indicates ASIMO's current state

In order to convey a harmonious, cooperating feeling when the robot is talking with people or operating close to people, it emits a pale green light. When ASIMO is carrying out a task on its own, it conveys a sincere, calm feeling by means of a pale blue light. At other times, it emits a neutral white light. It indicates waiting time by a gentle alternation of light and dark that is like breathing.

A process of searching for better illumination designs is under way while these light-emitting displays are used in actual operation. Figure 13 shows images of the lights in use.

4.3. Distinguishability

It has been found that when multiple ASIMOs are operating simultaneously in the same office, lobby or other such space, they can be difficult to distinguish one from the other because they are the same in external appearance. Accent colors of light blue, light green, and light yellow were therefore applied at the chest, shoulders, and thighs in a way that is readily identifiable and in line with ASIMO image, to make it acceptable to anyone. These colors are used in conjunction with the regular ASIMO colors (Fig. 14 and Fig. 15).



Fig. 13 Expression of the light



Fig. 14 Two new ASIMOs receiving a visitor

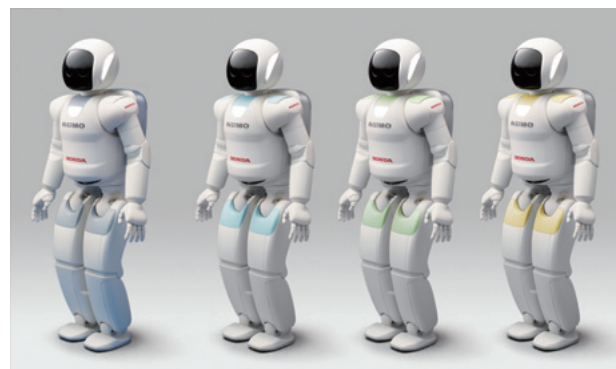


Fig. 15 Normal color and accent colors

At the same time, the basic overall colors of white, grey, and silver used for ASIMO from the beginning to express the qualities of being advanced and people-friendly, as well as to look at home in a variety of different environments, were updated to more refined, cleaner, and brighter color tones.

5. Conclusion

Design development for the new ASIMO further assimilates the basic “harmony of being advanced and being people-friendly” concept at a higher level. The design was carried forward with an emphasis on the perspectives of gentleness, comprehensibility, and distinguishability.

As a result, the new ASIMO has taken a step toward the goal of functioning around people less as an automatic machine and more as an autonomous machine. This was achieved by a design that people will receive sympathetically.

6. Closing

In November 2011, Honda decided on “Honda Robotics” as the collective name for its development of applied robot technology. This name embodies Honda’s powerful commitment and desire to contribute to society through robot research.

The “Honda Robotics logo mark” is placed on this technology (Fig. 16). It has ASIMO’s profile, head tilted slightly up to gaze into the bright future, as its graphic motif. This graphic symbol is accompanied by the “Honda Robotics logotype” in lower case letters with initial capitals. The image resonates with the “ASIMO logotype” that is in all capital letters. This logotype expresses the quality of being advanced and friendly, and a line in the red that is Honda’s corporate color expresses the expansiveness of robotics.

Consonant with the thoughts and feelings embodied in these marks, Honda’s desire is to further evolve ASIMO toward the conception of its goal as “a robot that coexists and cooperates with people and performs useful functions in society.”



Fig. 16 Honda Robotics logo mark

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Development of 2013 Model Year U.S. Accord

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ABSTRACT

The Honda Accord sedan and coupe for the North American market, both best-selling mid-sized cars, have undergone a full model change with the 2013 models and are now in their ninth generation. The Accord has a 37-year history, and this model in particular has incorporated a large amount of new technology. An efficient packaging technology based on Honda's unique "man maximum, machine minimum" concept has achieved larger cabin space and greater cargo capacity than previous models, while total length of the sedan has been shortened by 70 mm and of the coupe by 40 mm. The line-up of available power trains includes the newly developed L4 2.4-liter direct-injection engine, the V6 3.5-liter variable cylinder management engine equipped with added VTEC mechanism, and the three transmission options of a continuously variable transmission, a six-speed manual transmission, and a six-speed automatic transmission. The combinations of engine and transmission achieve a high-level balance of driving performance and fuel efficiency performance. This model also achieves a high level of safety performance and was given the highest ranking of five stars in the United States New Car Assessment Program (US-NCAP). (Based on Honda in-house data.)

1. Introduction

In the midst of a worldwide economic slump and increasing competition among automobile manufacturers, users have become more demanding in their selection of cars. There is an increasing tendency to purchase cars that have superior performance. In the middle-sized segment where the Accord is positioned, manufacturers have been placing one highly product-competitive new model after another on the market in recent years. Competition in this category has grown more intense than ever before, and car performance has been evolving conspicuously as a result.

When surveys of sedan users were conducted in the course of developing the 2013 model year U.S. Accord (new model Accord), it was found that the frequency of sedan use is increasing. Commuting to work is still the main use, but even on the weekend, for example, people are more likely than before to take their sedan rather than a minivan when they go out with their family. Sharply rising gasoline prices and growing awareness of environmental issues are among the background factors in this situation. The Honda team feels certain that consumers have high expectations for the Accord, which already has an established reputation for good fuel efficiency and spaciousness.

The team therefore decided to create a sedan that would address user expectations regarding both work and leisure

needs. This sedan would enable even more intelligent use and have styling that expresses a high-quality lifestyle. Development of this new model Accord began with the goal of a car possessing outstanding performance in every area.

2. Concept

The grand concept for the ninth-generation new Accord has been formulated as Excels in All Areas. This model further embodies intelligence, high quality, and advanced innovativeness (Fig. 1) through these three concepts: Its Intelligence follows the design sense and feeling of spaciousness of the previous model Accord, its High



Fig. 1 Exterior design (Sedan)

* Automobile R&D Center

Quality is based on the reliability and sense of security of the Honda sedan, and its Advances come from its position as the next-generation middle-sized sedan.

(1) Intelligence

The aim was to create an “Exhilarating and Efficient Package” in the form of a smart package with intelligence, high quality, and advanced innovation that is well thought-through down to the smallest detail.

(2) High Quality

The aim was to achieve advances in quick, sporty handling by means of the car’s high-rigidity body and newly developed suspension. These were also intended to achieve greater quietness and refined, dynamic performance.

(3) Advances

Advances were made in basic performance, including the achievement of environmental performance at the top of its class. Taking that as a basis, development was carried out with the aim of achieving a sedan that is superior in the area of safety.

The previous model of the coupe, on the other hand, achieved popularity, especially among personal users in their late 20s, for its styling as well as its spacious rear seats and roomy trunk. The new model Accord coupe incorporates these qualities while also introducing the evolutions in advanced basic performance achieved in the sedan. The aim with the coupe, therefore, was to achieve a car for adults that conveys a more dynamic, tighter sense of intelligence with the concept of Aggressive and Athletic (Fig. 2).



Fig. 2 Exterior design (Coupe)

3. Intelligence

3.1. Package

“Exhilarating and Efficient Package” was adopted as the concept for the package layout. The aim was to achieve a balance between optimizing the body frame to support fuel consumption and other such environmental requirements together with safety performance, and carrying on the car’s comfortable interior space while creating new value as a sedan.

For the exterior, the aim was to build on the “man-maximum, machine-minimum” concept of Honda’s human-centered styling philosophy to assure outstanding visibility performance. On that basis, the Honda team wanted to express the essence generated by functionalities such as good fuel consumption and aerodynamic performance as well as spaciousness and comfort at levels one rank above its class. The exterior concept was formulated as “exhilarating” to express the comfortable space and “concentrated” to express the tightly contained dynamic feeling and excellent aerodynamic performance. An uncompromising approach was taken to the quality of the body surface, and this model was imbued with functional significance as it was formed. The overall form was made to express a powerful presence and high-quality appearance by envisioning a three-box design (with engine compartment, cabin, and trunk) for its sedan-like quality while adding horizontal design keynotes that emphasized an extended lengthwise appearance when viewed from the side. Viewed from above, the car body was narrowed at the front and rear while the center was slightly widened to make a barrel-like shape. The front and rear pillars were also drawn inward while the center pillars were pushed outward, and deeply curved glass was used throughout to enhance aerodynamic performance while making the exterior appearance convey the spaciousness and comfort of the interior.

The interior expresses the concept of the luxurious modern interior with the themes of exhilarating, advanced, and high quality. Specifically, this involves the three aspects of generating a further sense of spaciousness in an ample interior space; realizing a next-generation interface that allows greater ease of operation; and conveying a sense of high quality that is one rank above its class. By achieving these three, a comfortable new sedan value was created.

3.2. Triple-Gauge Cluster

As a conventional sedan, the emphasis was placed on visibility. A triple-gauge cluster was therefore adopted with a large-diameter speedometer in the middle. The metallic trim-rings were shaped to appear even more metallic and further heighten the sense of detailed precision. An information display was placed in the center of the speedometer, and an ambient light was positioned around the periphery to display fuel efficiency status by color (Fig. 3).



Fig. 3 Gauge cluster

4. High Quality

4.1. Body Rigidity

The rigidity of the body frame was reinforced throughout (Fig. 4) in order to achieve linear handling as well as greater ride comfort and quietness. Rigidity was increased in the rear chassis, in particular, which contributes significantly to running stability and ride comfort, and this yielded high performance. The development effort made good use of computer-aided engineering (CAE) and also employed the test course to put the car through an intensive run and optimize the structure. The structure was examined with care for human sensibility, with attention given even to slight deformation and vibration and corrective measures taken. Weight reduction measures were also taken, including the use of even stronger high-strength steel (high-tensile steel), to produce a body frame that is lighter in weight and greater in rigidity.

As a result, as shown in Fig. 5, static rigidity rose by 34% for bending rigidity and 42% for torsional rigidity. In terms of dynamic rigidity, a 16% increase was made in front left and right rigidity, which affects handling responsiveness, and a 39% increase in rear upper and lower rigidity, which affects rear tire contact feel and ride comfort. In addition, lateral rigidity at the rear ground contact points was raised 25% and rear damper torsional rigidity was raised by 40%, suppressing rear suspension geometry change during cornering. A high-level balance of handling and ride comfort was achieved by strengthening

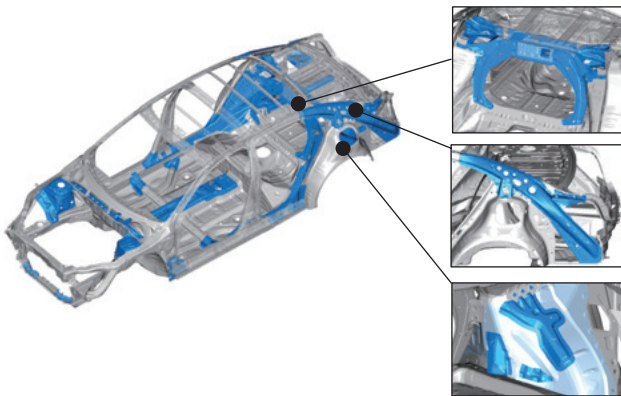


Fig. 4 Illustration of body strengthening

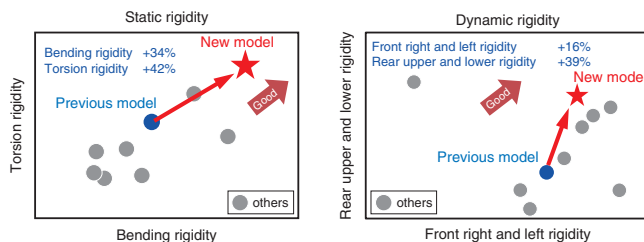


Fig. 5 Body rigidity comparison image

the front chassis while significantly increasing the rigidity of the rear chassis in this way.

4.2. Expanded Application of High-Tensile Steel

As shown in Fig. 6, higher strength high-tensile steel makes up 17.2% of the steel used. This includes the higher strength grade 1500 MPa steel achieved by hot stamping technology as well as grade 780 MPa and grade 980 MPa high-tensile steel. The result was an 8 kg reduction in the weight of the body frame.

4.3. Chassis

The chassis system was renovated with the aim of further evolving the quick, linear handling performance that was a strength of the previous model, as well as to achieve a balance between that and superior ride comfort.

A strut-type front suspension was adopted to enable a lighter weight and more efficient package. For the rear suspension, the basic structure of the previous model was retained and the suspension bushing was optimized to achieve a balance between dynamic performance at the top of its class and ride comfort with high-quality performance. The new model also included electric power steering (EPS), a lightweight, high-rigidity hybrid front subframe, and tires that provide outstanding grip performance while reducing rolling resistance. Thus, the developed chassis system could also contribute to reduction of fuel consumption.

4.3.1. Suspension

A MacPherson strut-type front suspension was adopted for dynamic performance exceeding that of the previous model. This new model suspension achieves a reduction in weight of 15 kg while also contributing to a higher-efficiency package by enabling a shortened front nose. In order to provide a further evolution in its top-class dynamic performance while achieving greater ride

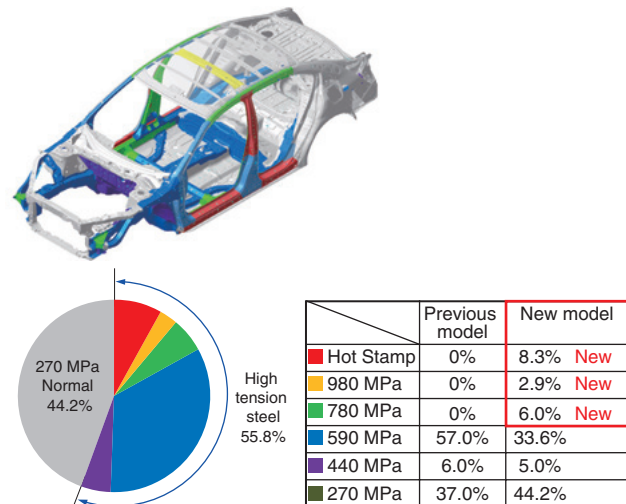


Fig. 6 High-tension steel utilization

comfort, suspension dampers with a new structure were adopted together with hydro-compliance bushings arranged coaxially with movable lower arms (Fig. 7, Fig. 8). High-frequency road noise was also reduced.

Newly adopted rebound springs in the front dampers act on the extension side and reduce vehicle roll by suppressing the lift of the inside wheels during cornering. As a result, this measure assures high turn stability while lowering the spring rate of the main damper springs and increasing ride comfort. Steps were also taken in the interior structure of the dampers, where Teflon rings were added to the oil seal in order to reduce upward movement on the springs and enhance responsiveness to the start of rolling movements during fine steering operations. In addition, a new valve structure matched to a newly developed oil was adopted in order to optimize the attenuation characteristics, achieving a flat ride. The system can change characteristics for lower spring rate in response to input from suspension stroke, and higher spring rate in response to input from cornering force, achieving a balance of handling and ride comfort. The adoption of a hydro-compliance bushing reduced steering vibration at high speeds (Fig. 9).

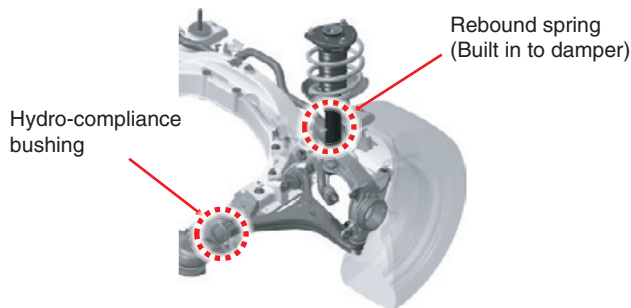


Fig. 7 MacPherson front suspension

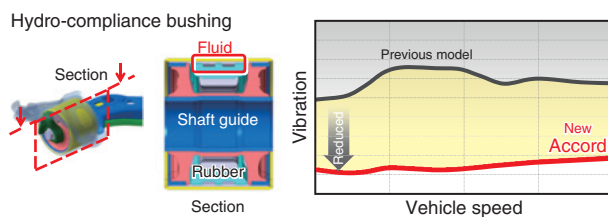


Fig. 8 Hydro-compliance bushing

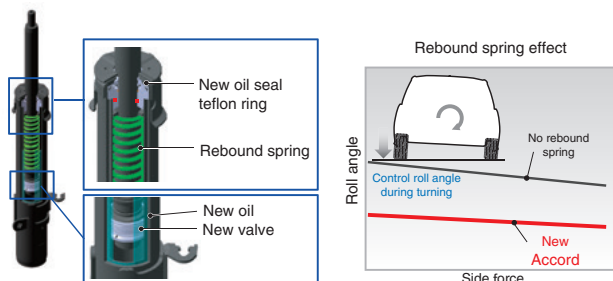


Fig. 9 Rebound spring

The multi-link double-wishbone rear suspension was continued, with the suspension bushings and dampers tuned to achieve superior running stability and greater ride comfort.

4.3.2. Steering system

EPS with superior fuel efficiency performance was adopted as the steering system. The adoption of a non-contact type as the new torque sensor equipped to the EPS increased sensor detection accuracy at the start of steering operation and achieved a more linear steering feel (Fig. 10). The quick, light handling performance that was a strength of the previous model was evolved by the adoption of a quicker steering ratio. Moreover, the use of a highly rigid steering column reduced the amount of vibration transmitted to the steering wheel and enhanced the rigid feel when steering.

4.3.3. Front subframe

For the front subframe, Honda has developed a proprietary manufacturing method of joining steel and aluminum that is a world first. The resulting hybrid steel and aluminum structure enabled a 25% reduction in weight and an 20% increase in rigidity relative to the previous model, in which this structure was made entirely of steel⁽¹⁾. Newly developed Friction Stir Welding (FSW) was used for the dissimilar-metal welding of steel and aluminum. As this joining technology involves the application from above of a rotating tool under high pressure against an aluminum piece lapped over a steel piece, it results in the creation of an

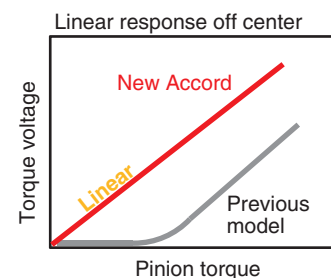
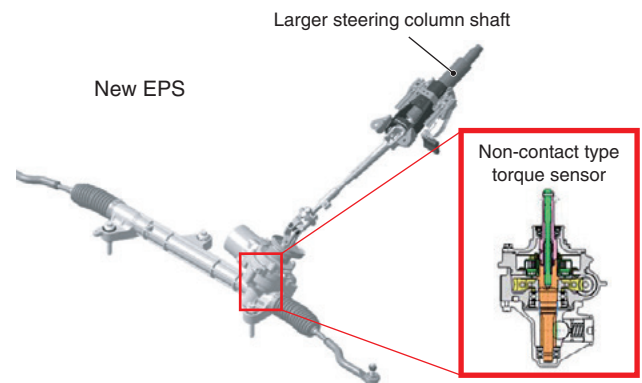


Fig. 10 EPS system

intermetallic compound of steel and aluminum as the tool advances (Fig. 11).

4.4. Noise and Vibration (NV) Performance

In order to realize high-quality driving, detailed measures were taken to suppress unpleasant noise and vibration under a variety of driving and road conditions.

First, active noise control (ANC) was adopted in both the 2.4-L and 3.5-L vehicles. Active sound control (ASC) was newly adopted in addition to ANC in order to add generated sounds to the engine sound during acceleration and produce a pleasant acoustic quality. This reduces booming noise when the car starts moving and also realizes a linear engine sound synchronized with acceleration, producing a sporty feel (Fig. 12).

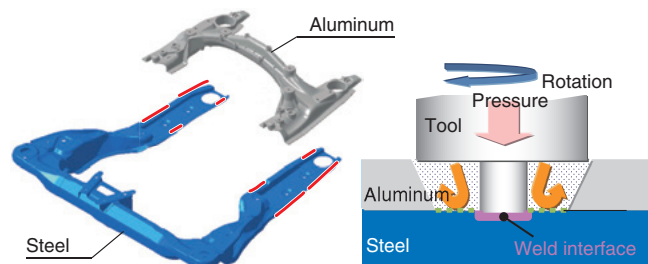


Fig. 11 FSW technology

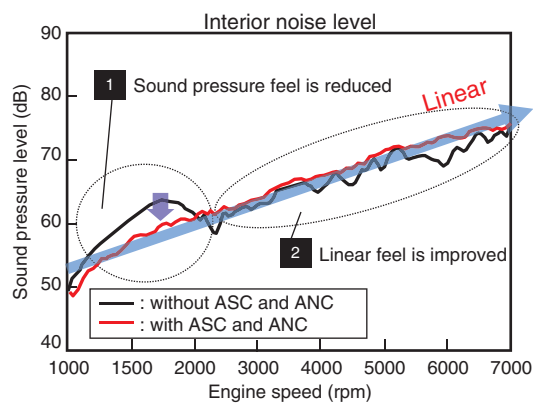


Fig. 12 Active noise control and active sound control

A specially designed muffler is used in the 3.5-L coupe in order to tune the exhaust sound. The ASC was also set to produce a sportier acoustic quality and volume, achieving a balance between impact during acceleration and refined quality when cruising.

In order to support the new model V6 engine that has higher mid-to-low-speed torque, the actuators of the active control engine mount (ACM) adopted for use on 3.5-L vehicles have been upgraded from 12V to 28V specifications for greater power (Fig. 13). Superior NV performance enables cylinder resting over a greater range and contributes to enhanced fuel efficiency performance.

Next, renovation of the front suspension, addition of dynamic dampers to the rear subframe, increased body rigidity for the suspension attachment structures, and other such measures have suppressed the transmission of vibrations that cause road noise. Then, the use of sound-absorbent material and acoustic insulation was expanded and various structural measures were devised. This involved providing the dashboard insulator with acoustic insulation functionality in addition to sound absorption by means of a triple-layer construction with a rubber layer sandwiched between felt layers. The number of places where urethane foam is inserted in the pillars was increased from six to 10, and thicker glass was used for the side windows as measures to reduce airborne road noise.

In addition, the wind noise suppression in the previous model, which was already at the top level for its class, was further increased. Superior body geometry was adopted for aerodynamic performance, the resting position for the front windshield wipers was lowered, and the flow of air from the engine hood over the front window glass was made smoother. The change in level between the front pillars and the front window glass was reduced by 20% or more, suppressing the turbulence in the air flow to the sides.

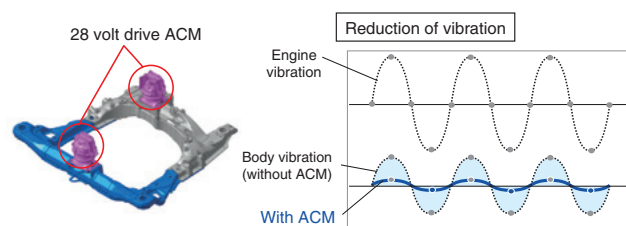


Fig. 13 Image of active engine mount control

5. Advances

5.1. Engine

The new model Accord engine applies EARTH DREAMS TECHNOLOGY, which is based on outstanding environmental performance and which pursues the joy

of driving found only with Honda. Specifically, it offers an L4 2.4-L DOHC i-VTEC engine that uses a direct injection system and is newly developed from the block up, or a newly developed V6 3.5-L i-VTEC engine that switches between low-speed and high-speed valve timing in addition to providing a cylinder cutoff system for increased power and torque relative to the previous model. Transmission options include a newly adopted Continuously Variable Transmission (CVT) (L4) and a six-speed automatic transmission (6AT) (V6), which together have also enhanced fuel efficiency performance (Fig. 14). These configurations (the CVT and 6AT for California) have also acquired designation as Super Ultra-Low Emission Vehicle (SULEV) and Partial Zero-Emissions Vehicle (PZEV), recognized for their clean performance under the California Air Resources Board (CARB) emissions regulations.

The L4 engine has Honda's proprietary VTEC and Variable Timing Control (VTC) systems, and in addition has a direct injection system (Fig. 15). By narrowing the valve inclination angle, the surface area of the combustion chamber was reduced to lower thermal loss. Steps were also taken to optimize the geometry of the intake ports and combustion chamber. These measures produced a powerful tumble flow within the cylinders, which promoted homogeneity of the air-fuel mixture and enhanced combustion stability. Detailed measures were also taken to reduce friction and lower weight, including the adoption of offset cylinders and a double-arm tensioner to reduce tensioner load from the timing chain.

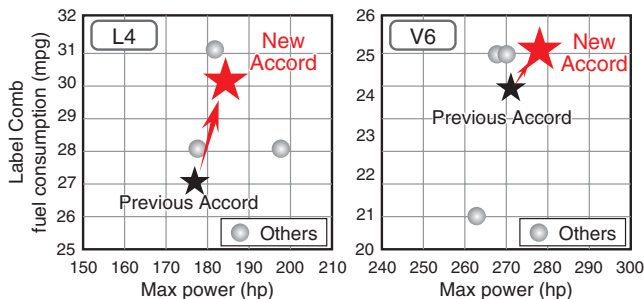


Fig. 14 Label FE-Power

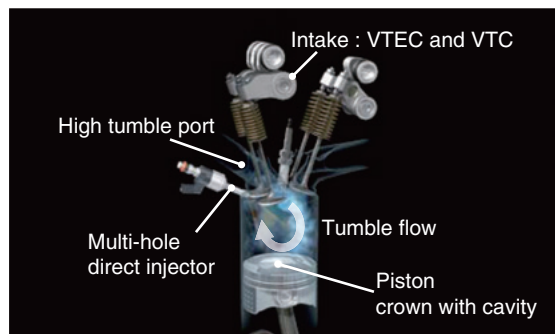


Fig. 15 Direct injection system

The V6 engine has a VTEC system that switches between low-speed and high-speed cams on the intake side of the front bank of cylinders, and a three-stage VTEC system that switches between a low-speed cam, a high-speed cam, and a cutoff cam on the intake side of the rear bank (Fig. 16). The low-speed cam has increased torque during three-cylinder combustion and expands the cylinder cutoff range. The high-speed cam has also increased power in the high-speed range.

5.2. Transmission

A high-precision CVT (Fig. 17) was newly developed for the L4 engine. The highly efficient CVT is capable of supporting the high torque of direct injection engines and making effective use of the range of greater engine efficiency, as well as advantageous use of its characteristic enhanced fuel efficiency. In addition, the G-Design Shift transmission control system, which provides immediate response to accelerator operation, gives drivers a feeling of linear acceleration (Fig. 18).

A new six-speed manual transmission was also developed to support the high torque of the direct injection engine and enhance both the driving experience and fuel

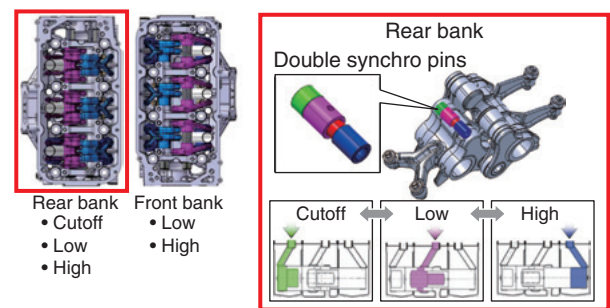


Fig. 16 VTEC structure

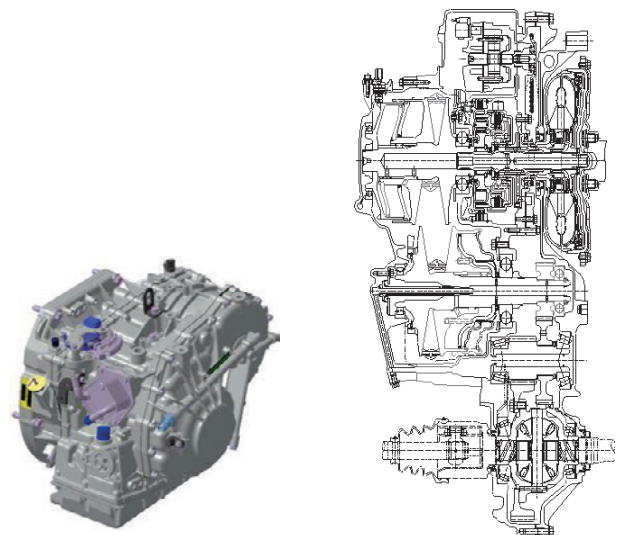


Fig. 17 CVT structure

efficiency. Inventive engineering of the structure and placement of every component achieved increases in strength and transmission efficiency while also realizing light weight and compactness on a par with previous five-speed manual transmissions.

A six-speed automatic transmission was newly adopted for the V6 engine in order to maximize the engine characteristics that enhance mid-to-low-speed torque and power. Expanding the gear ratio by multi-staging and expanding the lock-up range by means of a multi-plate lock-up clutch enhanced fuel consumption performance while also achieving driving with a solid, direct feel.

5.3. LED Headlights

LED headlights for the new Accord were developed to provide approximately twice the amount of illumination as halogen headlights for one-half or less of the power consumption (Fig. 19). The use of two projector headlights

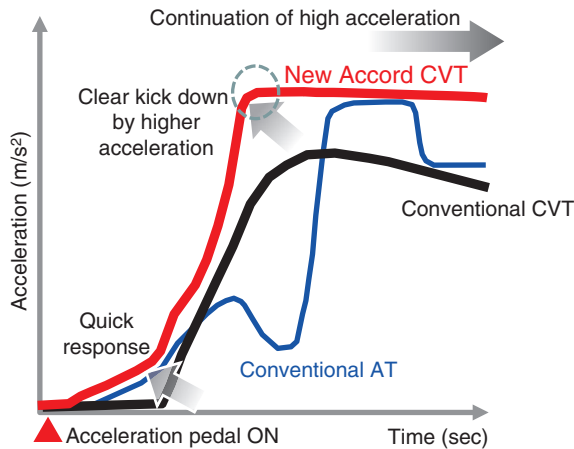


Fig. 18 Kick down characteristics



Fig. 19 LED headlight

employing five LEDs each achieved outstanding light distribution characteristics, increasing the driver's sense of security when driving at night by providing illumination over a wider area and greater distance (Fig. 20).

5.4. LaneWatch

The new model Accord was the first in the world to adopt the LaneWatch system (Fig. 21). This system has a camera built into the bottom part of the door mirror on the front passenger side that captures imagery across a range that cannot be covered with the door mirror and displays the imagery on an eight-inch high-resolution Wide-Video Graphics Array (WVGA) display. Compared to the approximately 20° angle of visibility with the door mirror, LaneWatch displays a range of approximately

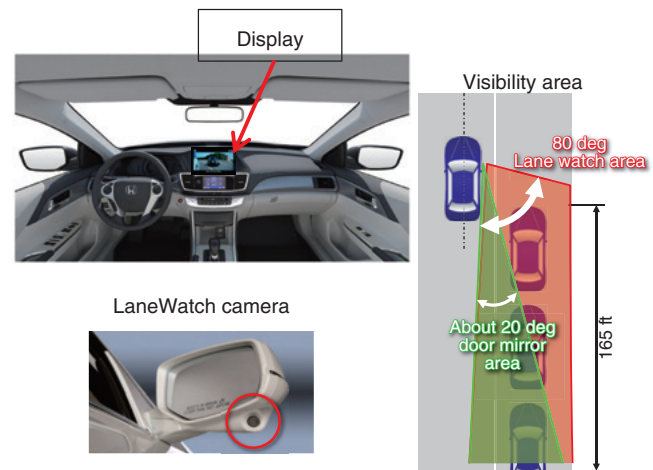


Fig. 21 LaneWatch system

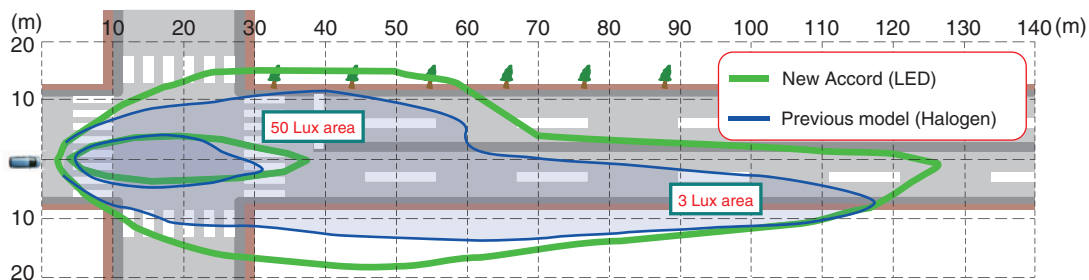


Fig. 20 Distributed headlight comparison

80°, enabling support in checking for vehicles to the rear side. Furthermore, the display is placed at a readily visible location on the upper instrument panel so that drivers can check without having to significantly divert their attention from the road. The display of guidelines also makes it easier to judge distances. Pressing the switch at the tip of the turn signal lever or turning on the right turn signal will bring up the camera imagery. Usability on vehicles equipped with navigation systems has been enhanced by enabling dual display of LaneWatch and the navigation system.

5.5. Forward Collision Warning System

The new model Accord has forward collision warning (FCW), which is a support technology for preventing collisions. In this system, a millimeter-wave radar placed in an opening in the front bumper or a monocular camera mounted inside the front windshield can detect vehicles ahead. When the car nears a detected vehicle and the system determines that there is a risk of collision, it warns the driver by blinking a dedicated heads-up warning (HUW) projected on the windshield, causing an FCW indicator in the gauge cluster to start blinking, and sounding a warning tone. The system allows for customization to enhance usability, so that system operation can be set to Normal, Short, or Long timing, and it can also be turned OFF.

5.6. Lane Departure Warning System

The new model Accord has lane departure warning (LDW), which is a support technology for preventing a car from inadvertently moving out of its current traffic lane. This system shares use of the FCW camera to detect lane markers on the left and right of the car's lane. If the system determines from the car's lateral velocity that it is about to depart from a detected lane, it causes an LDW indicator to blink in the gauge cluster and sounds a warning tone that is different from the FCW warning. In order to enhance usability, the system has a switch that allows it to be turned OFF or ON (Fig. 22).

5.7. Audio with Touch Screen

An audio system with touch screen (Fig. 23) was developed for the new model Accord. It has a 5.8-inch touch-sensitive color LCD panel in an easy-to-reach location. Only the necessary buttons and icons are displayed in large sizes on the touch screen so that users can select the desired operation quickly and easily. The system was also made capable of connecting with a variety of different sources such as Pandora, Aha (an Internet radio service), short message service (SMS), e-mail, portable audio players, and mobile telephones (by Bluetooth and USB connections).

5.8. Honda Link

The Honda Link used in the new model Accord supports Aha Radio, through which it can provide a variety of Internet services. When users connect their smartphone

(iPhone or Android) to it, Aha Radio can access web content such as Facebook, Twitter, Podcasts, Slacker Radio, traffic information, and restaurant information, and it also allows people to enjoy using software applications in the car.

5.9. Safety Performance

The highly evolved Advanced Compatibility Engineering (ACE™) body is one instance of the high standard of passive safety realized by the enhancements in this model. It represents a further advance in safety performance over the previous model, which had acquired five stars in the United States New Car Assessment Program (US-NCAP) and was named a Top Safety Pick by the Insurance Institute for Highway Safety (IIHS). The highly efficient body frame construction, the adoption of high-tensile steel with even greater strength, and other such measures have upgraded the impact absorption performance. This model has also been early in supporting the new small overlap collision rating introduced by IIHS, and the Federal Motor Vehicle Safety Standard (FMVSS) for rollover testing, which will be introduced in stages from September 2013.

5.9.1. Small overlap support

The small overlap test is for a mode that envisions collision with standing trees and other such barriers. The car is run into a pole-shaped barrier with a 25% overlap at 64 km/h (40 mph).

Since the barrier impacts with the outside of the side frame, the issue is how the collision energy is to be

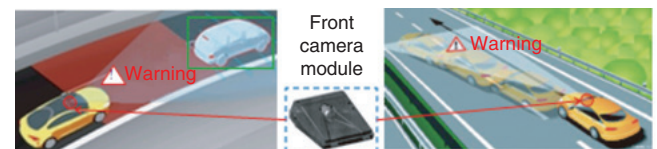


Fig. 22 FCW (Camera) and LDW image



Fig. 23 Audio with touch screen and 8-inch display

absorbed within the engine compartment. The new model Accord supports this by means of the Next Generation ACE™ body, which is an evolution of the ACE™ body adopted from the current model. The efficiency of the joint between the side frame and lower member has been increased and the lower members have been made higher-strength. Thus, collision energy can be absorbed effectively within the engine compartment by also using the side frame. Deformation of the cabin has also been limited by the use of high-tensile steel to reinforce the pillars, side sills, and roof rails (Fig. 24). These measures have achieved support of the small overlap impact with a lighter and more efficient body frame than vehicles without an ACE™ body.

5.9.2. Rollover support

Rollover support was provided by strengthening the cabin for small overlap support, increasing the roof strength to bear a greater load, and installing a rollover sensor. In the event of a roll, this helps increase occupant survivability while the rollover sensor provides optimized timing for deployment of the side-curtain airbags. In order to protect occupants' heads, the side-curtain airbags are constructed to enable them to remain in the deployed state (Fig. 25).

5.9.3. New side airbag construction

For the new side airbags (SmartVent™), two-layer construction was adopted to provide a high-pressure portion that deploys promptly and a low-pressure portion

that comes in contact with the occupant. This achieves a balance between mitigation of occupant injury during high-speed collisions and reduction of potential hazards to children and other occupants of the front passenger seat. As a result, there is no longer any need for the Occupant Position Detection System (OPDS) at the front passenger seat (Fig. 26).

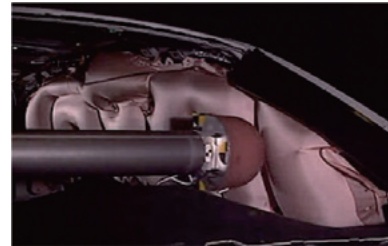


Fig. 25 Side curtain airbag

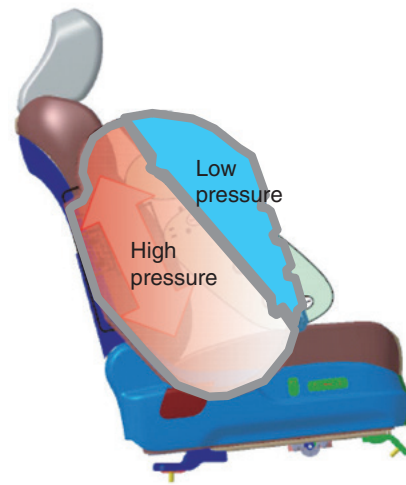


Fig. 26 Smart vent

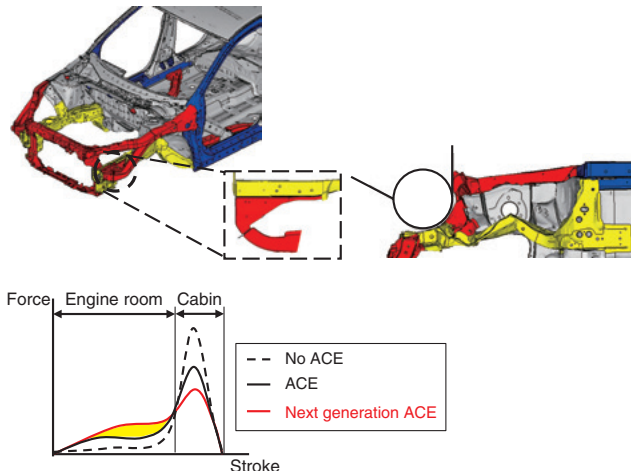
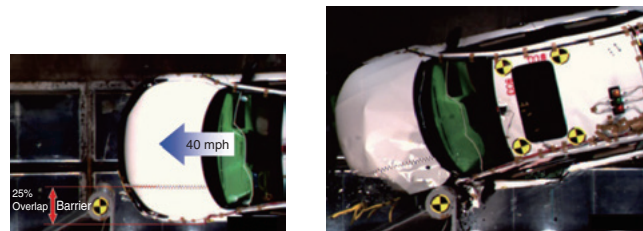


Fig. 24 Small overlap collision

6. Conclusion

The new 2013 model Accord for the North American market has a newly developed engine and platform and incorporates numerous new technologies. The above describes how the Honda team took on the challenge of building the world's most balanced sedan that would be chosen by users around the world. The vehicle that was produced as a result is an optimal family sedan that is matched to the needs of the future. Moreover, the coupe was given high-quality styling of dynamism and refinement within which the technology of the sedan is contained to ensure the achievement of a more aggressive vehicle.

The Honda team feels certain that customers will enjoy driving the new model Accord, and that it will enable them to experience the joy of driving and the joy of mobility in safety and comfort.

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ABSTRACT

In recent years, it has become easy to obtain the desired information in our lives with the enhanced network environment, the spread of smartphones, and so forth. Telematics, together with other services, have created the convenience of a variety of information being available for car users as well. However, thus far, there has been almost no service providing these types of information for motorcycle users. At the same time, motorcycle users have been experiencing strong needs for information that is unique to motorcycle use, such as rain and snowfall information appropriately provided in advance while riding, and parking information for motorcycles as such parking facilities are increasing rapidly.

To provide the information needed for motorcycle users, new services for motorcycle users have been developed such as mapping information of rain and snowfall, and information to inform them in a timely manner as to when to put on rainwear. In addition, the existing services for car users such as maintenance information and fuel economy information have been modified to fit motorcycle users. These applications have been incorporated into Honda Moto LINC service to be released as the first known membership system of telematics services for motorcycle users.

This development has helped motorcycle users to manage the difficulties they face while riding, and has made a significant contribution to expand the joy of using motorcycles.

1. Introduction

The rapid spread of smartphones in recent years is creating a society in which networks and information can be accessed anytime and anyplace. Honda has provided its internavi Premium Club telematics service for some time via vehicle-borne navigation systems and mobile phone data communications. This service increases the comfort and convenience of cars by using real-time information to enable users to avoid traffic congestion, and at the same time provides information useful for road administration^{(1),(2)}. In addition, provision of the internavi LINC service using smartphones also started from the spring of 2011⁽³⁾.

On the other hand, Honda developed the first known built-in-type navigation system for motorcycles exclusively for the large touring model GOLDWING⁽⁴⁾. In addition, other vehicle types were supported using general-purpose retrofitted-type navigation systems. However, none of these systems supported network connections, so there were no telematics services for motorcycle users. Therefore, it was thought that a telematics service for motorcycle users could be provided using smartphones in the same manner as internavi LINC, so efforts commenced to develop a

telematics service for motorcycle users.

The aim when developing motorcycle versions of applications was to provide information services that help address motorcycle user difficulties and also expand the joy of using motorcycles.

Motorcycle structure and usability differ greatly from those of cars, so there are also user difficulties and social demands that are unique to motorcycles. Examples of unique difficulties for motorcycle users include weather issues since the vehicle body structure does not enclose the rider, and countermeasures for the lack of motorcycle parking that has become a social issue in the national capital region in recent years. Unique information for motorcycle users is needed to address these difficulties.

When naming the service, it was thought important to select a name that instantly evokes the image of a Honda brand service for motorcycle users. Therefore, following in the brand image of internavi LINC for car users, the developed service was named Honda Moto LINC to clearly establish it as a unique service for motorcycle users.

Operation of Honda Moto LINC, which is the first known membership system telematics service for motorcycle users, commenced on April 12, 2012. This paper introduces the service contents and functions.

* Motorcycle R&D Center

2. Development of Service Applications to be Provided

To provide the information needed by motorcycle users, it was necessary to develop new services not included in internavi LINC for car users. In addition, applications already being provided to car users needed to be partially changed for motorcycle users.

The investigations and developments performed to address these issues are described in detail below.

2.1. Original New Services for Motorcycle Users

The five items of rain and snowfall map information, information on timing for putting on rain gear, motorcycle parking information, motorcycle dealer information, and touring information were developed as new services specialized for motorcycle users. Each new service is described separately below.

2.1.1. Development of service providing rain and snowfall map information

A service that provides local weather information at the estimated arrival time at the destination already exists as a service for car users. However, in the case of motorcycle touring it is not uncommon for the destination to change due to weather conditions along the intended route. Therefore, providing weather forecasts for the area around the destination and also current weather conditions and forecasts along the touring route in an easy-to-understand format can help address this difficulty for motorcycle users⁽⁵⁾.

When providing rain and snowfall information for use during touring, it is important that current conditions and forecasts be updated frequently, and that weather conditions along and around the route be easy to understand.

In addition, display screens for existing Internet and mobile website weather information services are often divided into prefectural and regional units. Furthermore, prefectural borders and other administrative districts are displayed, but there are few or no road shape indications.

For these reasons, existing weather information services are not suited for applications that view current weather conditions and forecasts along the touring route or for providing enlarged, detailed views of arbitrary areas.

Therefore, the specifications overlay rain and snowfall information onto the road map display screen, and enable the map to be scrolled, enlarged and reduced on the screen.

In addition, the weather information displayed by the rain and snowfall map uses 10-minute increment forecast information provided by the Japan Weather Association (JWA). Rainfall and snowfall information is color-coded into the seven types of 1, 5, 10, 20, 30 and 50 mm or more per hour, and snowfall, and displayed in 2.5 km-square mesh units (Fig. 1).

In addition to current conditions, forecasts are provided in one-hour intervals up to three hours in the future (Fig. 2).



Fig. 1 Rain and snowfall map (enlarged)

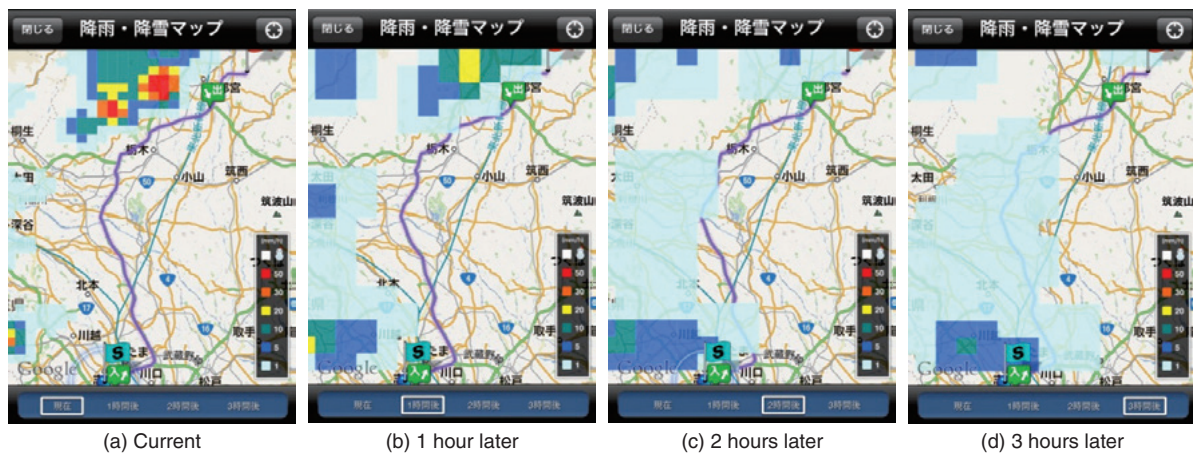


Fig. 2 Rain and snowfall map (Current condition and forecast)

Development of this rain and snowfall map information began as an original new service for motorcycle users, but partway through development it was recognized that this information is also useful for car users and in everyday situations where vehicles are not used, so it was also added as a new application to the internavi LINC for car users.

2.1.2. Information on timing for putting on rain gear

When rain falls while riding a motorcycle, it is necessary to put on rain gear. In this case riders need to secure a space where they can stop the motorcycle safely until they have finished putting on their rain gear, so if the timing when rain gear will become necessary is known in advance, riders can take the necessary measures without having to rush. This type of information can help address this difficulty for motorcycle users⁽⁵⁾.

The internavi LINC for car users provides a heavy rain warning (Fig. 3) for local downpours and other heavy rain as one of its disaster prevention information provision services. A method for effectively using that service as weather information for motorcycle users was investigated.

If the judgment threshold of the heavy rain warning for car users is used as is, it is easy to imagine that the rider's body will already be wet by rain by the time the warning sounds. Therefore, the specifications addressed this issue by using the heavy rain warning judgment logic as is but changing the judgment threshold. The judgment threshold was set at 1 mm or more per hour, which is the minimum rainfall value for the rain and snowfall forecast.

It was understood from past experience that motorcycles cannot be driven comfortably when there is rainfall of 1 mm or more per hour and rain gear is not worn. However, rain clouds were searched for using a prototype version of the rain and snowfall map and motorcycles were actually ridden to confirm just how wet riders actually get and how motorcycle users feel in that condition. The results showed that while there is some difference due to the length of the

time riding in the rain, the changed value of 1 mm noted above was determined to be appropriate as a judgment threshold for the timing to put on rain gear for motorcycles.

In addition, to enhance usability during touring, notification of rain in the direction opposite the direction of travel is unnecessary. Conversely, a function that adjusts the notification timing for rain in the direction of travel according to the vehicle speed would increase the value of the information for motorcycle users. Therefore, the specifications were set so that when the route is set, an alarm is displayed when rainfall of 1 mm or more per hour is forecast in the direction of travel along the route to the destination and the motorcycle position after 20 minutes is predicted to be within that range. This enabled notification of distant rain when traveling at high speeds on expressways, and nearby rain when traveling at slower speeds on ordinary roads. In addition, when the route is not set, the specifications are such that the judgment is made for a range centering on the vehicle's position. This enables notification of approaching rainfall even before setting the route prior to departure (Fig. 4).

This service was named the “Kappa” alarm as a familiar and easy-to-remember name. This name is associated with “kappa,” which is both a type of rain gear and an imaginary water creature that appears in Japanese folk tales.

2.1.3. Development of service providing motorcycle parking information

Parking for motorcycles (excluding bicycles and scooters with engine displacement of 50 cc or less) could be said to almost not exist before 2006. Revision of the Road Traffic Law in June 2006 brought about stricter enforcement of illegal parking rules for both cars and motorcycles. Prior to this revision, motorcycles (engine displacement exceeding 50 cc) were exempt from the Parking Places Act, so there was little construction of motorcycle parking, and enforcement of parking rules combined with this lack of

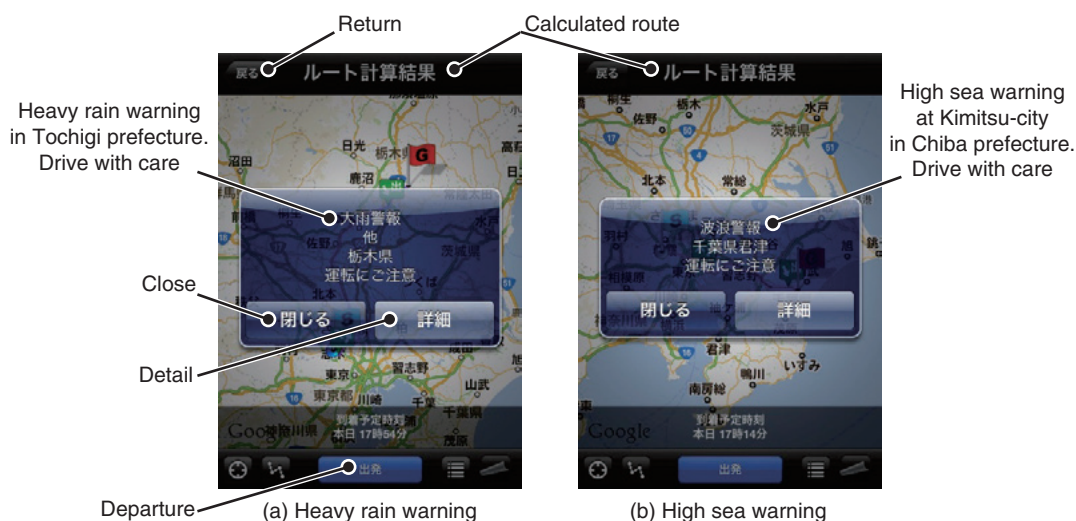


Fig. 3 Warning of bad weather conditions

motorcycle parking became a social issue centering on the national capital region around Tokyo as well as Osaka and other metropolitan areas.

As a result, various municipalities initiated promotional measures such as subsidies, and construction of motorcycle parking has begun⁽⁶⁾, but 60% or more of motorcycle users still experience trouble searching for parking^{(5), (7)}.

It is important to update parking information with regards to crowding, new establishment, discontinuance and other statuses. Construction of motorcycle parking has just started, so there are many changes such as new establishment and discontinuance, which made it an enormous burden to update a parking information database. The motorcycle parking information database released over the Internet by the motorcycle industry group Nippon MotorCycle Association (NMCA) currently boasts the largest data volume in Japan, and is continuously updated.

It was thought desirable to make this information easily available and usable to motorcycle users. With that reason, the Honda Moto LINC top menu (Fig. 5) included an additional link icon for NMCA motorcycle parking information (Fig. 6). This specification enabled motorcycle users to search for parking with simple operations as needed.

2.1.4. Motorcycle dealer information

The ability to easily search for the nearest motorcycle dealer in the event of vehicle trouble while touring gives great peace of mind to motorcycle users. However, information on motorcycle dealers in Japan is not currently compiled into a database, which made it difficult to search for motorcycle dealers even with existing motorcycle navigation systems. Nevertheless, a database specializing in Honda Motorcycle authorized dealers has already been created, and this search page is available on the Honda website.

Therefore, a link icon for dealer search (Fig. 7) was added to the Honda Moto LINC top menu (Fig. 5), and searches using icon button operations were enabled on

the Honda website. This means that whenever the Honda website data is updated, the latest information can even be viewed using smartphones.

At the same time, steps were also taken on the Honda website to support functions linking multiple smartphone applications. This enables jumping the position of the searched dealer to the destination setting of the smartphone locator simply by pressing an operation button one time.

This development enabled continuous operations from searching for a dealer to calculating the route, and enabled providing the service without placing a load on smartphones.

2.1.5. Development of service providing touring information

The drive information provided by internavi LINC includes enhanced content such as recommended spots, but this information is mainly for car drivers. As described above, preferences and purposes for motorcycle touring differ from those for car driving, so original content for touring is demanded.

The required information is described using Fig. 8 and the example of planning motorcycle touring from starting point IC1 to destination Pd.

When driving a car, the destination Pd is generally the final arrival point of the trip or travel, and expressway A or arterial road B is selected as the route from starting point IC1 to destination Pd. On the other hand, the purpose of touring is often simply to enjoy the process of traveling to the destination, such as over hilly winding arterial road C1, small passage D1, or graveled road D3. Therefore, the destination Pd may only have meaning as a turnaround point or a place to stop for convenience.

Drive information for car users is generally edited for use as a reference when selecting the destination Pd, so information on roads such as hilly winding arterial road C1 or graveled road D3 is often not shown. This makes it insufficient as a collection of information to be referenced for motorcycle touring.



Fig. 4 "Kappa" alarm

of imported motorcycles also differ.

Therefore, the member information database was changed to a digit configuration that also enables registration of motorcycle VIN, and a vehicle type database of the 500 or more domestic and imported vehicle types sold from 1980 onward was prepared. This enabled registration of Honda motorcycles regardless of domestic or imported models.

3. User Interface

3.1. Icon and Menu Design Concepts

In addition to content, high added value is also demanded of the menu design to enhance the appeal of applications. Good menu design is also important as a trigger for users to feel that they want to try using the application⁽¹⁰⁾.

The Honda Moto LINC icon appears alongside the icons of other applications on the top menu of smartphones, so it could be said to be the “face” of Honda Moto LINC. The design evokes an image of a running motorcycle, while maintaining design coordination with the internavi LINC icon (Fig. 9).

The Honda Moto LINC top menu image was based on the internavi LINC top menu, and the icon for each function was designed with motorcycles in mind (Fig. 5).

3.2. Transition from the Top Menu to Each Service

The function icon layout on the top menu is based on that of the internavi LINC, with the functions thought to be more frequently used for motorcycles arranged on the top level of page 1 of the top menu, and functions with lower frequencies of use arranged on page 2.

Response times to status transitions and operations were verified from the viewpoint of easy operability. As a result, changes were made such as button designs that facilitate understanding of whether a tap operation or slide operation is required.



Fig. 9 Honda Moto LINC icon

4. Post-release Support

This application was released as the first known membership system telematics service for motorcycle users on April 12, 2012. The total number of downloads exceeded 10 000 downloads in the first month following the release, and 20 000 downloads after six months (Fig. 10).

The total number of downloads as of November 10, 2012 was 14 300 downloads for the iPhone version and 8 500 downloads for the Android version, for a total of some 22 800 downloads.

Prior to release, the application was demonstrated using smartphones at the Automotive Telecommunication Technology Tokyo (ATTT) and the Motorcycle Show (Tokyo, Osaka) held in March 2012, and was greeted with anticipation by many attendees. The number of downloads exceeded 5 000 downloads in the first five days immediately following the release, and this is thought to be due to the anticipation effects of these demonstrations.

The number of downloads per day gradually decreased thereafter, but the total exceeded 10 000 downloads in the first-month period, which included the Golden Week holidays. The number of downloads trended stably at several 10 per day from June onward, and was not seen to change greatly even during the summer vacation period.

The next noticeable change occurred when the e-book format touring map was announced on October 10, with the number of downloads trending at over 100 per day for a five-day period. This increase in the number of downloads even though actual touring map access would not start for another three weeks indicates the high expectations of motorcycle users toward the touring map. Three weeks later, the number of downloads also exceeded 100 per day for about a week starting from the day prior to the start of touring map access on October 31.

Following its release on April 12, Honda Moto LINC received favorable marks from motorcycle journal writers and other journalists as “revolutionary” and “highly

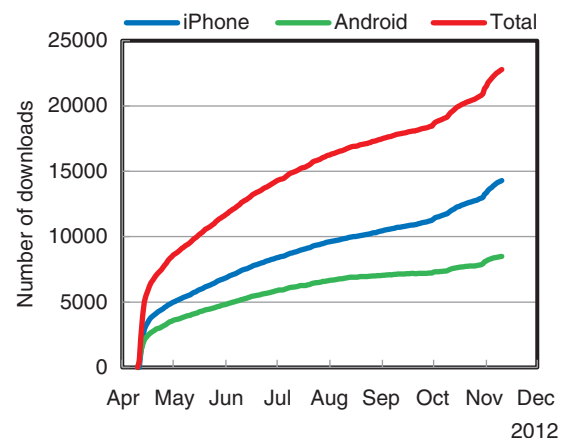


Fig. 10 Download counts

convenient,” including one comment in particular regarding the “Kappa” alarm that the journalist could “feel the love toward riders.”⁽¹¹⁾

In addition, many messages of support were received from Honda motorcycle users for this first approach for motorcycles, along with requests for further enhanced applications and services.

5. Conclusion

Honda Moto LINC is the first known membership system telematics service for motorcycle users, and was developed by creating new services for motorcycle users and partially changing existing services for motorcycle users in order to provide information needed by motorcycle users.

The newly developed services are as follows:

- (1) Provision of rain and snowfall map information
- (2) Provision of information on timing for putting on rain gear (“Kappa” alarm)
- (3) Provision of motorcycle parking information
- (4) Provision of motorcycle dealer information
- (5) Provision of touring information

In addition, services such as the following were developed by partially changing existing services:

- (1) Maintenance information
- (2) Fuel economy information
- (3) Member registration

These developments helped to address difficulties for motorcycle users and contributed to expanding the joy of using motorcycles.

6. Afterword

In order to respond to the high marks and requests for further enhancement received from users following the release of this service, Honda intends to continue proposing new value by further enhancing this application as a new point of contact between Honda and motorcycle users.

The authors would like to take this opportunity to express their profound gratitude to the Nippon MotorCycle Association (NMCA) for their generous cooperation in providing information on motorcycle parking.

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Development of Battery System for EV

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ABSTRACT

A battery system for EV use has been developed with the aim of realizing compactness and high durability. The system employs Toshiba SCiB lithium-ion batteries, which display good durability and power regeneration characteristics, as the power source. Sufficient numbers of batteries to provide the energy necessary for the driving range demanded for the vehicle have been connected in a densely packed configuration and fitted in a battery box case employing a seal structure providing good environmental resistance. Together with battery fans for cooling and high-voltage safety devices, this comprises a battery system designed to be installed compactly under the floor of the vehicle without sacrificing the interior comfort of the FIT EV. In addition to demonstrating that the batteries themselves displayed good durability, verification of the durability of the battery system indicated that the cooling system helped to ensure the maintenance of a high level of performance over the entire period of vehicle use.

1. Introduction

As one response to the depletion of fossil fuel reserves and the need to protect the global environment, Honda is advancing technological development efforts in the area of vehicle electrification, and is providing appealing products to the market. The FIT EV, which was made available for lease in the U.S. and Japan in summer 2012, achieves the highest electrical energy efficiency performance in its class (as of the commencement of leasing of the vehicles; figures approved by the EPA), and in addition combines this achievement with the driving pleasure offered by three-mode driving. Standing behind this achievement is a high-power 100 kW battery system with a capacity of 20 kWh. The fitting of the battery system under the vehicle floor in a high-density package also contributes to the interior comfort of the FIT EV. The battery system has also been provided with a cooling system in order to realize a higher level of durability, offering convenience and peace of mind to users throughout the entire period of vehicle use. This paper will discuss the characteristics of the batteries forming the foundation of the FIT EV battery system, the battery cooling concept and configuration based on those characteristics, and battery system durability.

2. Development Goals

Because batteries provide drive power for EV, if driving range is increased, it is generally necessary to fit an

increased number of batteries in the vehicle. This increases the volume and weight of the batteries. The FIT, which served as the base vehicle for the FIT EV, was well received for balancing a compact body with a high level of interior comfort. It was therefore necessary in the development of the battery system for the FIT EV to realize the essential level of energy while also focusing on reducing the size of the system in order to maintain this characteristic of the base vehicle.

In addition, it is necessary for the battery systems employed in EV, which consist of large numbers of batteries connected together, to maintain the vehicle performance over the entire period of use of the vehicle. Given that EV are fitted with numerous batteries, differences in the battery use environments can result in differences in the changes to battery characteristics. Changes in battery characteristics are affected by factors such as the batteries' temperature, state of charge (SOC), and current flow. Battery temperature is one of the factors exerting an effect on changes in battery characteristics, but because battery temperature is unevenly distributed within a battery system, differences in battery characteristics occur. A cooling system developed on the basis of an understanding of changes in battery performance in relation to battery heat generation characteristics and battery unit temperature, able to realize a uniform temperature environment for multiple batteries, is therefore an important device in terms of maintaining vehicle performance.

The following two main goals were set for the

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development of the EV battery system.

- (1) A battery layout contributing to the realization of a high level of interior comfort
- (2) A battery system that maintains a high level of durability

3. System Configuration and Performance

3.1. Cooling System Configuration and Performance

In designing the cooling system, particular consideration was given to environmental temperatures and the status of traffic infrastructure in the U.S., quick-charging needs in Japan, and factors that would affect battery temperature. The battery system used in the FIT EV is positioned under the vehicle's floor, and uses external air as a cooling air flow. Because of this, the system would be affected by exhaust heat from the drive system while the vehicle was in operation (Fig. 1) and radiant heat from the road surface when it was parked (Fig. 2). In addition, when the vehicle was parked outdoors the battery system would also be affected by heat as the temperature inside the vehicle increased due to the effect of heat from the sun (Fig. 3). In an EV, which employs a larger number of batteries than an HEV, it is essential to develop cooling systems based on consideration of these effects.

In the case of the battery cooling method, two methods of cooling, direct and indirect, were considered as candidates. Direct cooling would use external air to provide a cooling air flow in the interior of the battery box case, and the air would directly exchange heat with the batteries. In the case of indirect cooling, heat would be transmitted from

the lower surfaces of the batteries to heat transfer members and the battery box case, and heat would be exchanged with cooling air passing through the two-fold structure of the case. Direct cooling would offer a higher level of battery cooling performance, but it would be essential to attempt to prevent foreign matter, dust and grit, and moisture in the external air introduced in this method from infiltrating the battery connections. Based on study of the amount of heat generated by the batteries as a result of user modes of use, indirect cooling, which would make it possible to use a compact battery box configuration without any structures to assist in preventing dust and moisture, was selected for application in the FIT EV.

Figure 4 shows the structure used for indirect cooling. Heat transfer materials are positioned between the bottoms of the cells and the battery case, and transmit the heat generated by the batteries to the battery box case. Fin-shaped guides that increase the cooling surface area and create a homogeneous cooling air flow across the battery box case are positioned in the cooling air channels in the two-fold structure of the battery box. When the cooling air flow, introduced by two battery fans, passes through the cooling air channels in the battery box, it exchanges heat with the batteries indirectly.

In designing the cooling structure, repeated simulations were conducted to design a structure able to efficiently cool all of the batteries and control battery temperature within a prescribed range. Figure 5 shows the flow of cooling air

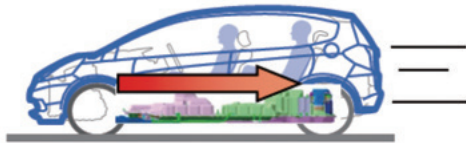


Fig. 1 Influence of exhaust heat during vehicle operation



Fig. 2 Influence of radiant heat from road

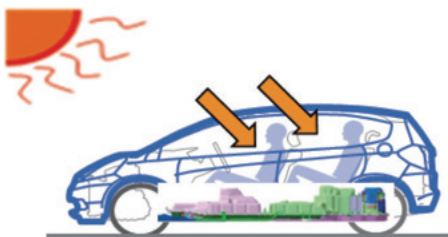


Fig. 3 Influence of sunlight when vehicle is parked

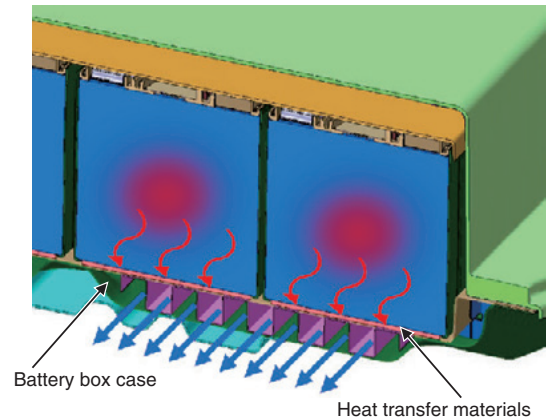


Fig. 4 Indirect cooling configuration

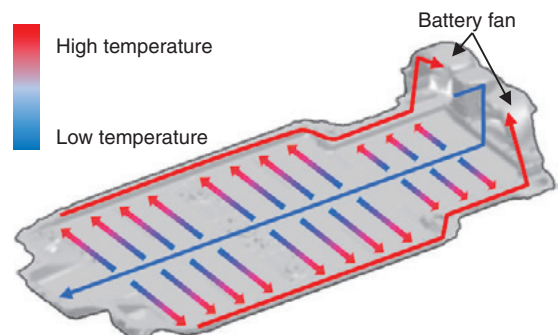


Fig. 5 Cooling air flow in battery box case

through the battery box case. The cooling air provides heat exchange as it passes through the battery box case. Figure 6 shows a comparison of the results of a test conducted to verify battery system cooling performance with results obtained when the cooling system was not employed. When the cooling system was not employed, it was not possible to control the increase in the temperature of the batteries with repeated vehicle operation and battery charging, and the upper limit temperature for the maintenance of battery durability was exceeded. By contrast, in the case of the battery system using the cooling system, temperature increase was controlled and the upper limit temperature was not exceeded, helping to enable continuous vehicle operation without power save.

3.2. Battery Layout

Because the average voltage of the SCiB is lower than that of a typical lithium-ion battery, it would be essential to use a large number of cells in order to increase voltage and help ensure the necessary energy. Because of this, the volume of the FIT EV battery system, composed of large numbers of SCiB, would be greater than that of a system using typical lithium-ion batteries. At the same time, it was also important to install the battery system in the vehicle without sacrificing the interior comfort of the base FIT. This necessitated study of densely packed battery layouts. Figure 7 shows a diagram of a section of a battery module, the basic component of the battery system. The use of

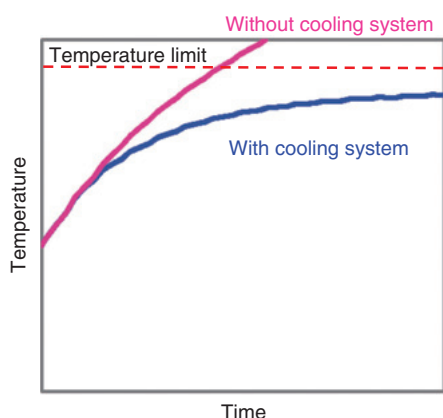


Fig. 6 Results of cooling tests

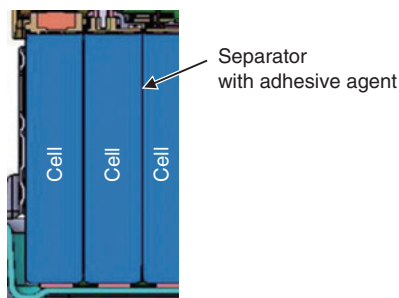


Fig. 7 Structure of battery module connection (Overview)

indirect cooling made spaces between the cells for the flow of cooling air unnecessary, and it was therefore possible to bond the cells together using only separators between them to create a densely packed layout while also realizing strength as a module. By this means, it was possible to balance the battery layout with good interior comfort.

3.3. Features of Lithium Titanium Oxide (LTO) Batteries

Typical lithium-ion batteries use metal oxides and carbon materials as the active materials at the cathode and anode, respectively. The SCiB employed in the new FIT EV resembles a typical lithium-ion battery as it also uses metal oxides as the active materials at the cathode, but is distinctive in using LTO at the anode. Figure 8 shows the results of a differential thermal analysis of the anode materials during occlusion of Li with the electrolyte as a means of evaluating the thermal stability of LTO as an active material for use in electrodes⁽¹⁾. In Fig. 8, results for hard carbon are added to a graph drawn in Reference (1) in the References section. The vertical axis of the graph shows the reaction heat capacity with the electrolyte. The graphite used in typical lithium-ion batteries displays a high heat-generation peak at around 240°C, but the heat generation of the LTO is less than one-tenth this level. Similarly, the LTO's level of heat generation is lower than that of the hard carbon used as an anode material in lithium-ion batteries, and it thus displays high heat stability.

Figure 9 shows the electrical potential characteristic of a typical lithium-ion battery, and Fig. 10 shows the potential characteristic of the SCiB.

While the electrical potential of the anode of a typical lithium-ion battery drops to close to 0 V when the battery is charged, the potential of the anode of the SCiB remains constant at around 1.5 V. Because of this, the average voltage of the SCiB is 2.3 V, a low figure compared to the average of 3.6 V for a typical lithium-ion battery. When the electrical potential of the anode of a lithium-ion battery drops to a level close to the potential for lithium plating, lithium metal can precipitate on the anode. Plating of lithium metal is a factor in the formation of lithium

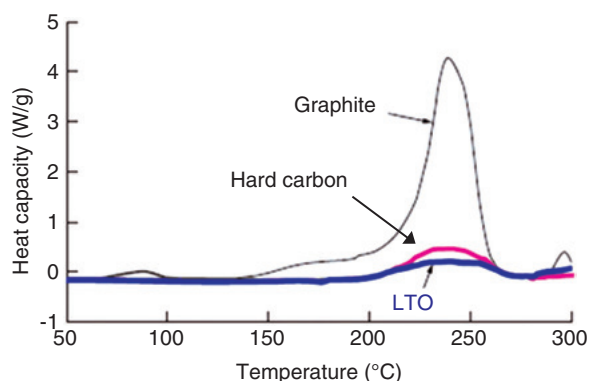


Fig. 8 Reaction heat capacity of cathode material with electrolyte during Li occlusion

dendrites (needle-shaped crystals), and this can result in declines in the battery's reversible capacity and durability. In the case of a typical lithium-ion battery, when the battery is fully charged, particularly at low temperatures, fine control of battery voltage is necessary to attempt to prevent plating of lithium metal, and this can prevent a full charge from being attained or can extend the time taken to reach full charge, among other issues. By contrast, in the case of the SCiB, the potential of the anode does not drop to close to the potential for plating of lithium metal. Because of this, plating does not occur, meaning that the battery maintains its reversible capacity and displays increased durability.

3.4. SCiB Performance

Figure 11 shows the regeneration characteristics of different types of batteries at each SOC at -10°C . As the graph indicates, the SCiB displays good regeneration performance.

By contrast with standard Li-ion batteries, SCiB are reported as displaying a high level of durability⁽¹⁾, but they are prone to performance decline in the form of decline in capacity and increase in resistance due to

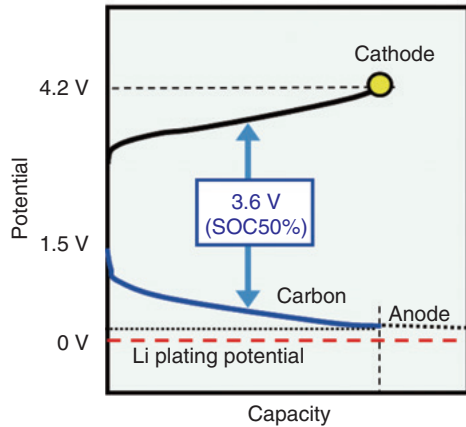


Fig. 9 Potential characteristics (Typical lithium-ion battery)

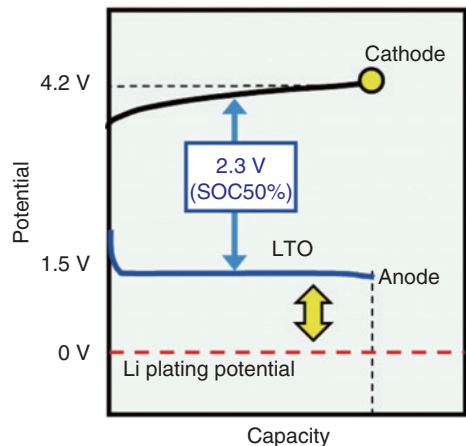


Fig. 10 Potential characteristics (SCiB)

the effect of SOC and temperature. As a representative example, Figure 12 shows the results of test verification of the change in resistance for two standard temperature settings. It is clear that the increase in resistance is greater at 45°C . The fact that performance decline increases in proportion to temperature is a characteristic the SCiB shares with typical lithium-ion batteries. And like a typical lithium-ion battery, the performance change in the SCiB can be considered using the 1/2-power rule⁽²⁾.

Figure 13 shows the results of applying the 1/2-power rule to the time in the results shown in Fig.

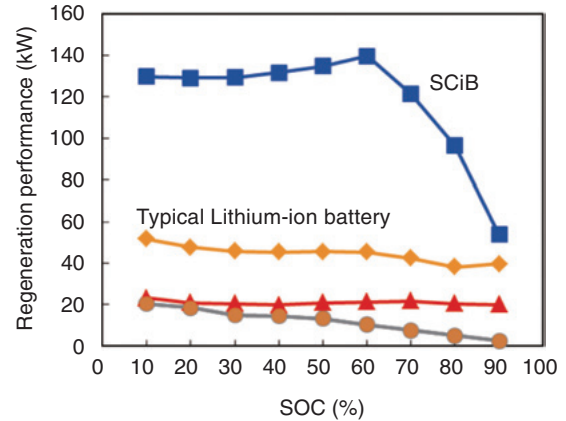


Fig. 11 Regeneration performance (-10°C)

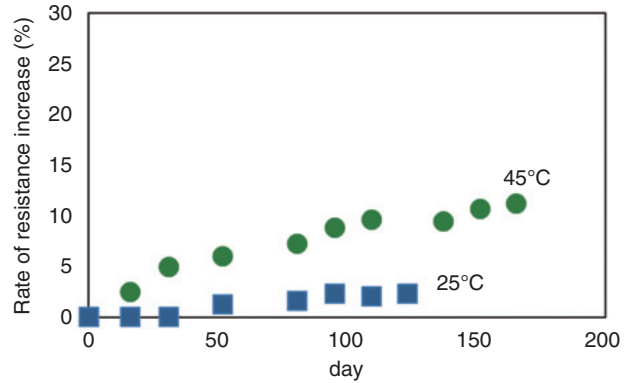


Fig. 12 Change in resistance

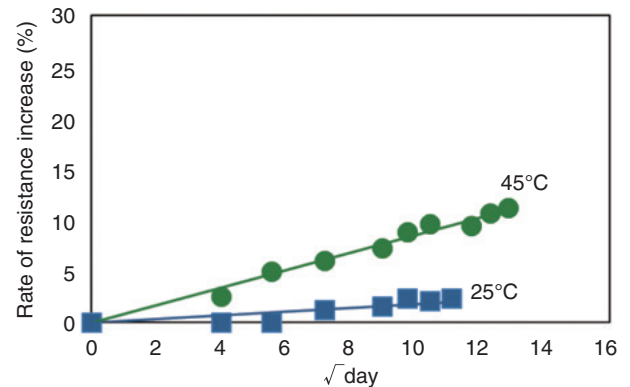


Fig. 13 Change in resistance

12. The increase in resistance displays linearity in relation to the 1/2-power of time. The coefficients of change in resistance performance for the respective temperatures can be obtained from the gradient of the line. Figure 14 shows performance change coefficients calculated from the results of tests of changes in capacity and resistance for a charge-discharge cycle and storage for battery units, based on modes of vehicle use. The performance change for the SCiB is greater for the charge-discharge cycle (vehicle operation) than for storage. An appropriate range of use was set for the SCiB based on this type of understanding of performance change.

In addition, it is possible to estimate the performance change for the SCiB during vehicle use based on the performance decline coefficients obtained from cycle tests and storage tests. This makes it possible to estimate and study changes in the performance of the SCiB in relation to modes of use for a variety of users. Furthermore, it is also possible to conduct tests of accelerated performance decline by setting high-load conditions based on user modes of use.

3.5. Battery System Durability

In developing a battery system for EV use it is essential to sufficiently verify the durability of the batteries based on factors including the temperature environment and road and charging infrastructure of the region in which the vehicles will be used. Because test results have indicated that temperature affects battery performance changes, it is also necessary to consider battery heat generation. EVs differ from HEVs in that batteries are their sole power source and, because of this, more current flows through the batteries the higher the vehicle speed, and the level of heat generated by the batteries is consequently greater (Fig. 15). During the development of the battery system, a range of vehicle speeds encompassing vehicle use conditions for different users was set based on the distribution of average speeds and battery current flow produced by vehicle use patterns

including highway driving, and EV durability tests were conducted with thermal loads applied to EV.

Figures 16 and 17 show the results of these EV durability tests for the SCiB and a typical lithium-ion battery.

The effect of the application of high-durability SCiBs and the ability of the cooling system to control battery temperature were indicated in test results. The batteries displayed around one-fifth the decline in capacity and one-half the increase in resistance of a typical lithium-ion battery over the period of vehicle use, indicating that the potential effect on the vehicle's driving range had been

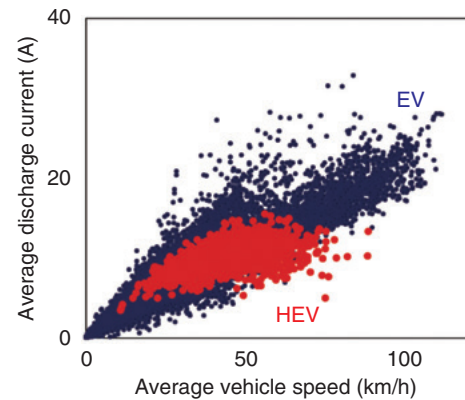


Fig. 15 Correlation between vehicle speed and battery discharge current

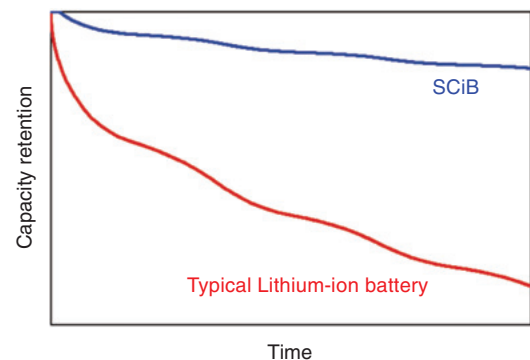


Fig. 16 Results of durability tests (Capacity)

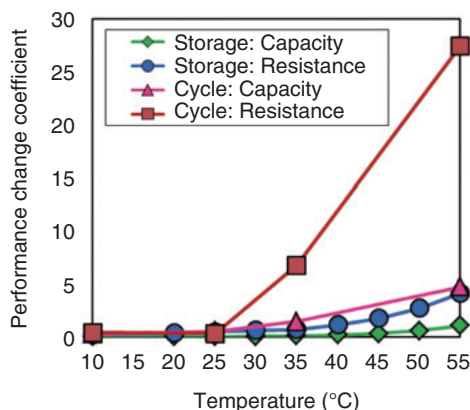


Fig. 14 Performance change coefficients (Storage, cycle)

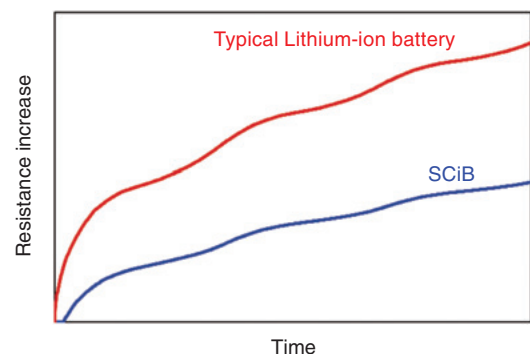


Fig. 17 Results of durability tests (Resistance)

controlled to a significant degree. In terms of the increase in resistance also, the results showed that the level was such that it would not affect the FIT EV's dynamic performance.

4. Conclusion

An EV battery system has been developed for use in the FIT EV. Employing a densely packed module configuration in order to increase compactness, the battery system is fitted under the floor of the vehicle, contributing to the good interior comfort of the FIT EV. The battery system employs indirect cooling, making it possible to efficiently cool all of the batteries. This has been verified as helping to control increases in battery temperature with repeated vehicle operation and charging. The combination of the high level of durability of the SCiB themselves and the effect of the cooling system have made it possible to offer a battery system that displays minimal battery performance decline resulting in a reduction in the vehicle's driving distance and offers a high level of durability for the entire period of vehicle use.

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Development of Battery Traceability System for EV

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ABSTRACT

The EV market is expected to expand, and as this happens, battery usage will increase. Batteries deteriorate with time and usage, and when battery capacity declines as a result of age-related deterioration, the cruising distance over which a vehicle can travel decreases. Therefore, it will be important mainly to monitor batteries during use but also to assess the performance of batteries during the resale of used cars and during secondary usage such as conversion to stationary batteries. Honda has equipped the FIT EV first leased in the summer of 2012 with a system that regularly sends mainly battery-related data to a server when the vehicle is travelling and when the battery is charging. Customer consent is obtained to collect data from Japan and the U.S. on the Honda R&D Japan server. The goal of this system was to determine battery performance and to build a model for estimating battery deterioration, so that a foundation for successful data analysis and visualization could be built.

1. Introduction

Batteries deteriorate with time and usage (repeated charging and discharging). Deterioration mainly involves the decline of battery capacity and output; however, when the capacity of an EV battery, in particular, declines, the distance that the vehicle powered by the battery can travel decreases. This means that a decline in battery performance causes a decline in vehicle performance⁽¹⁾. Therefore, the ability to accurately assess battery status is considered to be necessary for vehicle operation, the used-car market, and secondary use, and this ability is expected to be an important factor for developing services capable of responding to market needs in the future EV market (Fig. 1).

The battery traceability system for EVs that Honda developed is provided with a mechanism for collecting mainly battery-related data from individual vehicles that are travelling or charging and regularly sending the data to a server. In other words, battery-related data from all vehicles that are leased is aggregated on this system, and traceability data (hereinafter, “trace data”) is gathered. This trace data is important for determining the status of the battery and the status of the vehicle. The data is analyzed in real time and needs to be effectively utilized in developing batteries and expanding business and services.

Methods of collecting and analyzing massive quantities of data that are known as “big data” (large data size, multiple data sources, unstructured data, high-volume transactions) are additionally being proposed. To utilize big data is to rapidly and in real time process data that has been collected

thus far but has not yet been fully analyzed, as well as data that took time to analyze and process. Such data processing enables information from massive quantities of data to be visualized, analyzed, extracted, and then utilized in business and services. Big data utilization is gaining attention as a means of swiftly responding to and establishing action plans to address changing values and a changing market in which various products continue to be made electronic. Trace data can also be thought of as a part of big data.

Based on the background described above, a system foundation was built to analyze and visualize gathered trace data in order to determine battery performance in EVs and create a model for estimating battery deterioration.

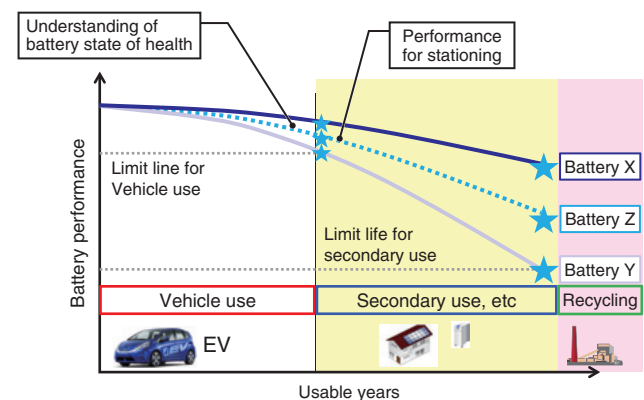


Fig. 1 Importance of battery performance understanding (EV)

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2. System Configuration

2.1. Concept

The usage status of a battery is influenced by changes in traffic and differences in battery charging equipment. In order to observe differences in battery information resulting from differences in the travelling region and battery usage status, it is believed that data should be collected and analyzed in one place. Therefore, after data is collected from various regions, Honda aggregates the data on the Honda Japan R&D internal server via an internal intranet and then consecutively analyzes that data.

2.2. Vehicle System

The FIT EV, which is the vehicle in which the traceability function has been installed, possesses a telematics communication unit that uses a smartphone application to enable the air conditioning timer to be operated, charging to be controlled, and the status of charging to be determined. The battery traceability system consists of this system foundation and a data collection function (Fig. 2).

Vehicle information sent by the telematics communication unit (TCU) is received and collected in real time on the regional server using pre-existing equipment for communication-type vehicle navigation systems. In this system, an intranet is utilized in communicating between central servers, and sent vehicle information is compressed into files of binary data on the TCU and data is periodically uploaded (Fig. 3).

Data is sent from the vehicle, especially when the vehicle is travelling, but also when it is charging.

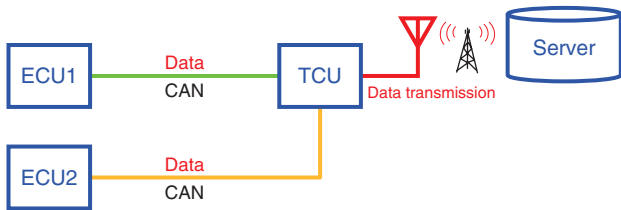


Fig. 2 Vehicle system composition

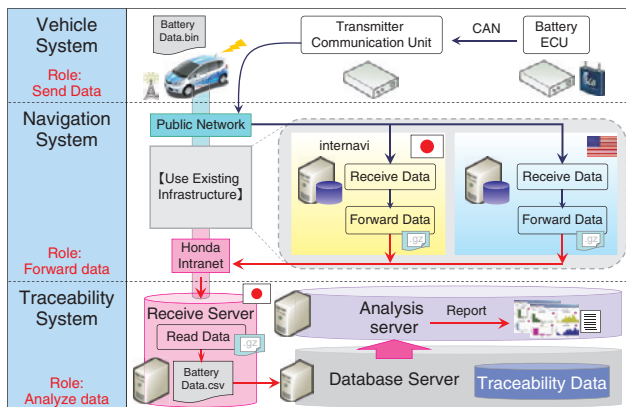


Fig. 3 Transmitting system configuration

2.3. Server System

The battery traceability server is configured with blade server architecture and a virtual platform. Web service technology (SOAP over HTTP) is used in communicating between the battery traceability server and the vehicle navigator server, the data is received on a reception server and received binary data is turned into text and accumulated into a database for analysis.

3. Building a Model for Estimating Battery Deterioration

Batteries differ greatly in capacity retention (capacity maintenance rate) depending on their history, such as the duration of use, temperature, and state of charge (SOC). In addition, previous research has shown that even batteries with the same capacity maintenance rate can develop differences in remaining battery life if their history differs (Fig. 4)⁽²⁾.

3.1. Conventional Method of Estimating Battery Life

The capacity maintenance rate that is generally used to estimate battery life is expressed by Eq. (1), which is proportional to the square root of time (root principle). When using this equation, it is impossible to express differences in the rate of battery capacity decline resulting from differences in usage history. The reason is that, in the root principle equation, the change rate of the capacity retention rate: differential equation of Eq. (1) is represented by Eq. (2) and the usage conditions and the capacity retention rate at that time are expressed. Therefore, as Fig. 4 shows, if batteries with the same capacity retention rate but different histories are later used in the same way, the later capacity retention rate will be mistakenly deemed the same. Thus, this equation cannot be used to perform an estimate that accounts for the deterioration rate of individual batteries.

$$y = 1 - a \times t^{0.5} \quad (1)$$

$$y' = -\frac{0.5}{1-y} a^2 \quad (2)$$

y : Capacity retention rate

y' : Change rate of the capacity retention rate

a : Deterioration factor decided by the usage conditions

t : Integrated usage time

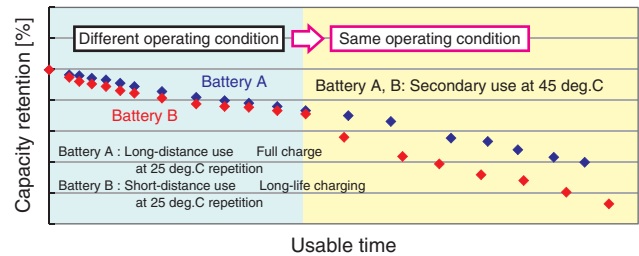


Fig. 4 Capacity retention variation by specification operating condition

3.2. Battery Life Estimation Method that Accounts for Usage History

By including the past battery usage history in the estimation equation, it is believed to be possible to represent differences that result from usage history. If factors that influence future deterioration can be uncovered from the past usage history, then it is possible to index the remaining value of battery during use, in the used vehicle market, and in secondary use, as shown in Fig. 1. It is also expected to be possible to index categorization criteria when remanufacturing battery modules or cells that must be investigated from the standpoint of efficient resource usage. Battery life estimation methods that incorporate usage history are shown in Eqs. (3) and (4).

$$y' = -\frac{0.5}{1-y} (\alpha + b \times \beta) \quad (3)$$

$$y = 1 + \sum_{t=0}^T y' \quad (4)$$

b : Correction factor determined by previous usage history

α, β : Deterioration factor decided by usage conditions

T : Time elapsed since battery production date

Equation (3) shows the deterioration factor a^2 from Eq. (2) substituted by the first-order equation $(\alpha + b \times \beta)$, which incorporates correction factor b that is determined by usage history. It is possible to express the difference in the deterioration rate caused by usage history and, as Eq. (4) shows, the integral of Eq. 3 can be used to estimate the decline of the capacity retention rate. For the usage history, traceability data that was collected by the traceability system is used to derive correction factor b . Because the value of correction factor b differs according to usage history, deterioration rate y' also differs according to usage history. This method allows the capacity retention rate to be estimated even for batteries with different usage histories.

3.3. Verification Results for the New Battery Life Estimation Method

A single-cell battery was used to derive the first order equation coefficient $(\alpha + b \times \beta)$ in above-mentioned Eqs. (3) and (4) from battery property data obtained through shelf tests, cycle tests, and other parameter studies of general battery properties, and a method of estimating battery life was built. The states in which the vehicle was charging, travelling, or parked were all considered when the accuracy of this estimation equation was verified through endurance tests that simulated the actual device load. The measured battery capacity value and the estimated value derived from the battery life estimation method were plotted and compared. The change over time in the values of both plots was nearly equivalent (Fig. 5 and Fig. 6).

In addition, batteries installed in test vehicles were also

verified. The actual value and expected value of the battery capacity retention rate when endurance and driving tests were conducted in extremely hot regions were plotted and compared. Because the change in capacity over time was proximate to the change in actual values over time, as was the case with the single-cell battery tests, the accuracy of the estimation method has been demonstrated (Fig. 7).

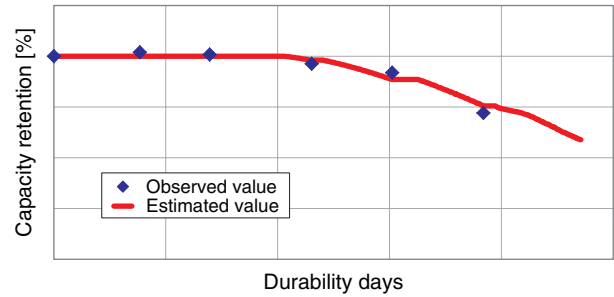


Fig. 5 Normal 200 V charge SOC limit 100% at 55 deg. C

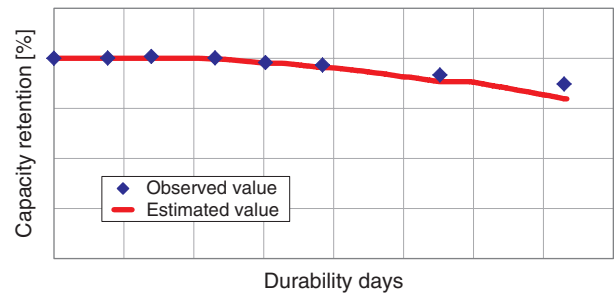


Fig. 6 Quick DC charge SOC limit 100% at 55 deg. C

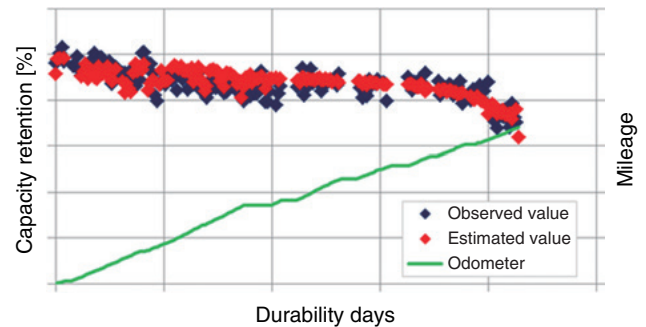


Fig. 7 Verification of life estimation method

4. Analytical Verification of Demonstration Test Data

Collaborative demonstration tests are being conducted by Kumamoto Prefecture, Saitama Prefecture, and Saitama City in order to identify and respond to issues facing the thorough dissemination of the EV and in order to verify future personal mobility and the CO₂ reduction effect under actual traffic conditions or with equipment such as chargers.

Starting in December 2011, vehicles equipped with this system were actually used for demonstration tests, and test driving by local government officials and general users was started. Honda is currently conducting region-specific demonstration tests including park and ride, car sharing, and ride sharing (Fig. 8)⁽³⁾.

The operation of this system began with these demonstration tests, and various criteria have been verified.

4.1. Creating a Data Analysis and Visualization Function

This system is a report system that automatically analyzes and visualizes trace data from vehicles without human intervention. It works by performing pre-set processes, analyses, and extraction processing to enable results to be displayed on a Web browser.

4.2. Trace Data Analysis and Visualization Processing

The trace data analysis and visualization processing in this system can be broadly divided into the following two categories.

- (1) Processing aimed at determining the present battery performance and vehicle status

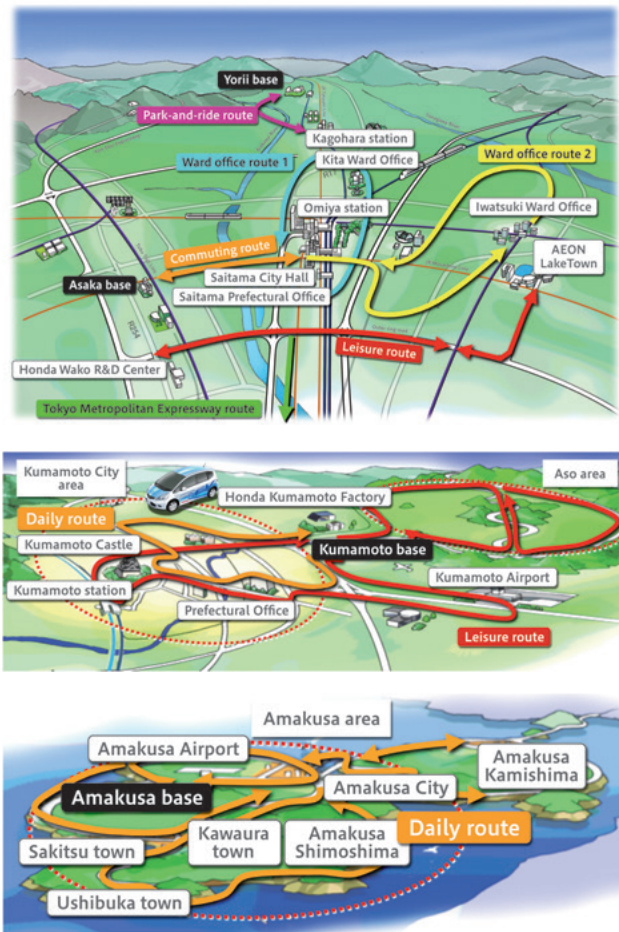


Fig. 8 Honda electric vehicle demonstration program operation area

- (2) Statistical processing that incorporates past travelling data and other data for each vehicle (battery)

For (1), a report system (cockpit) was built that processes data sent from a vehicle in real time, analyzes the present travelling mode such as travelling, charging, or parked, extracts the vehicle status, cumulative driving information summaries, as well as the battery capacity transition, and visualizes battery problem detection, etc. A cockpit report that was developed based on vehicle data for demonstration tests is shown below [Fig. 9(a) and Fig. 9(b)].

The cockpit system displays the total cumulative travelling distance of vehicles in the demonstration tests, histograms of the battery temperature, SOC, battery current, etc., and comparisons with the averages of other vehicles. As a result, changes in the test vehicles can be easily determined.

For (2), an analytical report system was built in which various categories of trace data sent from vehicles are used as indexes, and various analytical reports were developed, including finite and non-finite reports that identify factors arising from battery performance assessments and battery deterioration analyses [Fig. 10(a) and Fig. 10(b)]. In Fig. 10(a), vehicles are continually operated, and the fact that the total distance travelled has exceeded 7700 km in approximately 9 months of operation can be determined in real time. Also, in Fig. 10(b), it is possible to determine

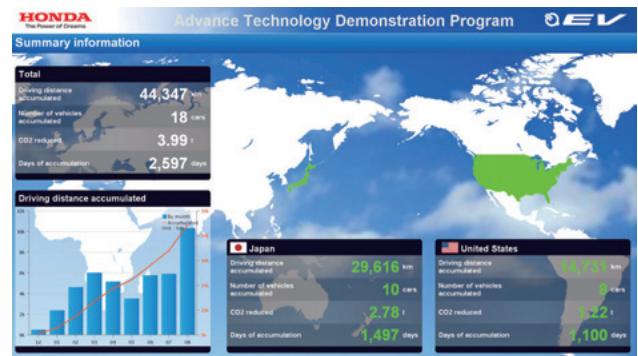


Fig. 9(a) Cockpit report: Vehicle information summary



Fig. 9(b) Cockpit report: Battery state information

differences in how each vehicle was used, and by analyzing the distance travelled in one day, it is clear that vehicle A travels longer distances per day than vehicle B.

Building a report system requires a scalable report system foundation capable of easily selecting factors influencing or possibly influencing battery performance and visualizing and extracting analytical information. Honda's system uses the IBM software IBM Cognos® and IBM SPSS® Modeler to achieve the above-mentioned analytical report system. This software not only enables many types of analyses and reports of various data in an online graphical user interface (GUI), but also allows statistical model analyses to be performed.

4.3. Utilizing the Results of Data Analysis

As for the data analytical system that was developed, Honda is developing and verifying a model for estimating battery deterioration by utilizing the analytical results of data sent from demonstration test vehicles and test vehicles. In addition, the data analysis results can be utilized in many ways; for example, by collecting and analyzing trace data in real time, it is possible to consecutively analyze the status of the battery in the travelling vehicle. As a result, not only is it possible to address mechanical problems, it is also possible to monitor progress after addressing those problems. A specific example is the fact that it is possible to use vehicle data for early detection when battery cell

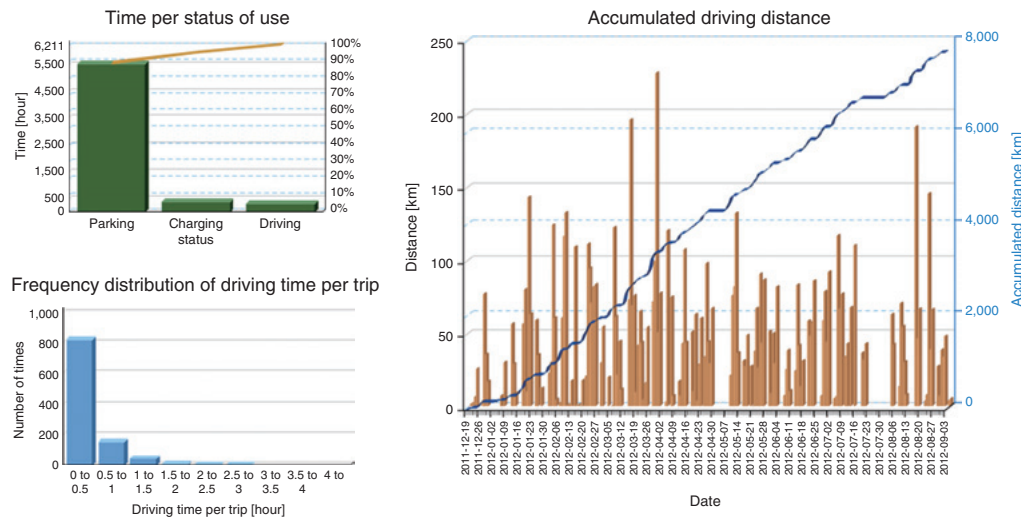


Fig. 10(a) Analysis report total mileage and operation state

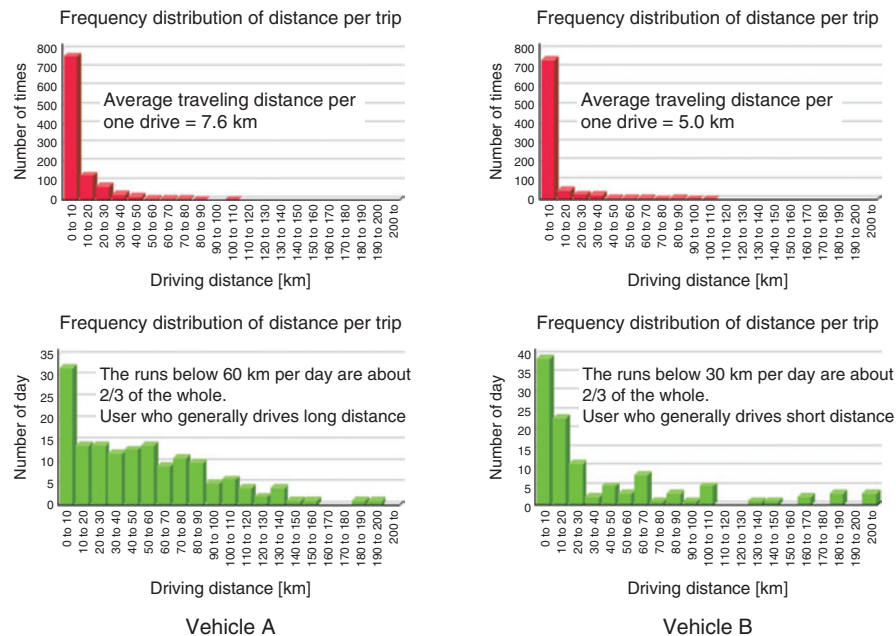


Fig. 10(b) Analysis report distribution of distance

voltage varies more than expected. Considering this, the test vehicle's control unit program was changed to reflect the content of countermeasures for addressing problems. Similar to progress monitoring, it is possible to confirm the change in voltage over time, and thus, the validity of the results of the countermeasure program can be confirmed (Fig. 11).

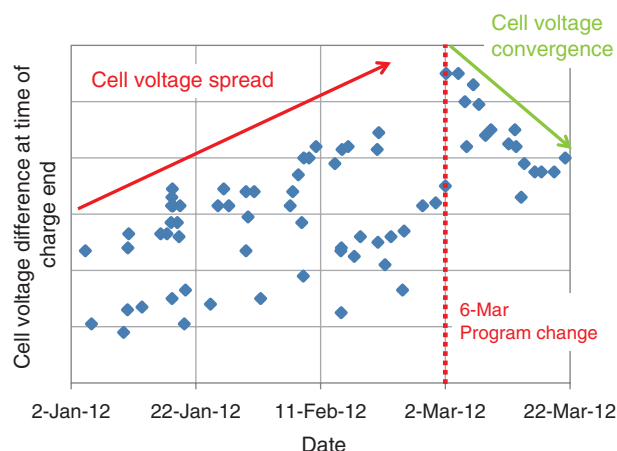


Fig. 11 Early detection and correspondence of vehicle condition

5. Conclusion and Future Outlook

The present vehicle status and battery status were determined by analyzing and visualizing trace data sent from vehicles. Problematic events could also be identified prior to mechanical problems actually occurring by visualizing past data and statistical data. In addition, an expandable trace data analysis foundation was established, which is capable of not only making the above-mentioned determinations but also extracting anomalous data points and implementing specific measures to address problems. Furthermore, by incorporating a battery deterioration assessment capability in addition to a data analysis capability, in the future it will be possible to detect and discover warning signs of capacity reduction, etc. from the battery status.

The initial leasing of the FIT EV to customers in the U.S. began in July 2012. Honda plans to statistically accumulate and analyze trace data collected at the data center with the owner's consent, utilize that data in future development efforts, and investigate control that accounts for differences in battery performance and promote the development of new services.

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Development of Charging System for FIT EV

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ABSTRACT

A high-efficiency plug-in charging system has been developed for the FIT EV.

The goal of development of the system was to promote the more widespread use of EV. The development was conducted in order to increase the user-friendliness of EV with the aim of offering rapid, high-efficiency, and safe charging. The rapid charging was realized by setting the power of the on-board charger to maximize the capacity of the power distribution equipment employed in the Japanese and U.S. charging environments and expanding the range of the constant power charging control that maximizes the charger's performance. In addition, the minimization of the operating time of the charger has made it possible to realize high-efficiency charging. As a result, the developed system has realized the target of a minimum charging time of within three hours.

With regard to safe charging, the system employs protection against abnormal excess heat produced as a result of greater deterioration of the charge connector terminals under the conditions of use of the system than initially projected. The optimal positions for the charge lids were determined based on studies of the garages in users' homes and public charging stations.

1. Introduction

Today, increased environmental awareness on a global scale and concern over energy security originating in the instability of crude oil supply are stimulating technological development towards the achievement of increased energy efficiency in automobiles, which account for 20% of CO₂ emissions. As is well known, response to the Zero Emission Vehicle (ZEV) regulation in California necessitates the further electrification of automotive drive, and manufacturers are turning to active development of electric vehicles (EV) as the next stage beyond the hybrid electric vehicle (HEV).

The merits of EV are not limited to their environmental performance; the vehicles possess a variety of advantages over internal combustion engine (ICE) vehicles, including quietness and low running costs. Honda was the first major automaker to begin leasing an EV with the EV Plus in 1997. The EV Plus employed plug-in charging⁽¹⁾, utilizing a 4.2 kW on-board charger. Plug-in charging involves electrically connecting the vehicle and a commercial power source, and charging the traction battery with the necessary energy for vehicle operation. In the FIT EV, the aim was to achieve the world's highest figures for miles per gallon of gasoline-equivalent (MPGe) and increase user-friendliness. To determine MPGe, the amount of power consumed from the commencement of charging to the completion of mode driving is measured. This makes it important to increase the efficiency of the on-board charger. The efficiency and

power of the charger were therefore increased, its charging time was decreased, and the unit was further reduced in size.

This paper will discuss the aims for the development of the new charging system and the characteristics of the system.

2. Charging System: Issues and Targets

The plug-in charging necessary for EV corresponds to the supply of fuel to an ICE vehicle. Because the energy density of batteries is lower than that of fossil fuels, in order to achieve an equivalent supply of energy, the charging time for an EV is longer and the charging frequency is higher than the corresponding supply of fuel to an ICE vehicle. These are issues to be overcome in the quest to achieve more widespread use of EV.

Targets were established for the present development in an attempt to address these issues. These were:

(1) Charging in a short period

Minimum charging time target: Within three hours

(2) High level of charging efficiency

Charging efficiency target: 87% or more

(3) Safe charging

3. Overview of Charging System

This section will provide an overview of the charging system employed in the FIT EV.

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3.1. FIT EV Charging Environment

This section will discuss the charging environments in use in Japan and the U.S.

3.1.1. Specifications for Japanese charging environment

Figure 1 shows Japan's charging environment. Japan employs a quick-charging network based on the CHAdeMO standard, and, a quick-charging system that complies with this standard was employed in the Japanese FIT EV, given that it would therefore be sufficiently practical for charging when the vehicle was on the road. Making use of quick charging at 50 kW, the system can bring the battery to an 80% state of charge (SOC) within 20 minutes.

The main method of normal charging is overnight charging at home, and a specification with an output of 3.3 kW, able to charge the batteries at 200 V within six hours from the time the vehicle is returned home to the time it is driven out again, was therefore employed.

With regard to the on-board charging cable, in addition to a 200 V specification compatible with the recommended charging voltage, the vehicle is also fitted as standard with a 100 V specification for emergency charging.

3.1.2. Specifications for U.S. charging environment

Figure 2 shows the U.S. charging environment. Because there is basically no quick-charging network in the U.S. at present, a quick-charging system was not employed in the U.S. FIT EV. In its place, charging time was reduced by maximizing the output of normal charging as much as

	Type	Charge time	Electric Vehicle Supply Equipment (EVSE)	
Normal charge	100 V 12 A	23 hours	Portable type	
	200 V 15 A	6 hours	Installed type	
Quick charge	DC 50 kW MAX	20 min.		

Fig. 1 Overview of charging (Japan)

	Type	Charge time	Electric Vehicle Supply Equipment (EVSE)	
Normal charge	120 V 12 A	15 hours	Portable type	
	208 V 32 A	3 hours	Installed type	
	240 V 32 A			

Fig. 2 Overview of charging (USA)

possible.

The power distribution equipment fitted as standard in homes has a 32 A continuous rating. A 6.6 kW output that would maximize the capacity of the existing power distribution equipment was therefore set, making it possible to charge the batteries within three hours.

A 120 V specification, stipulated by Article 625 of the National Electrical Code (NEC) as suitable for fitting in a vehicle, was employed as the on-board charging cable.

3.2. Composition of Charging System

Figure 3 shows the basic composition of the plug-in charging system. The section outlined by the broken line in Fig. 3 shows the Japanese specifications of Quick charge system. Figure 4 shows a diagram of the vehicle layout of the system.

Main contactors are positioned at each of the poles of the traction battery. The traction system and the auxiliary equipment system, on the secondary side of the main contactors, are connected to the battery by the main contactors.

In the traction system, traction is obtained by transmitting electricity through the traction inverter to a traction motor.

A DC/DC converter and an on-board charger are connected to the auxiliary equipment system. The DC/DC converter converts power to be sent to the 12 V battery, and the on-board charger converts AC commercial electric power.

The traction battery, DC/DC converter and on-board

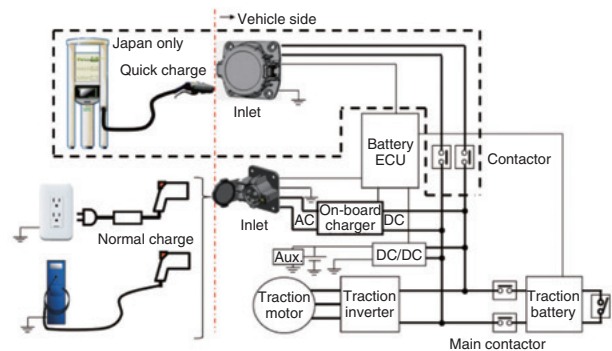


Fig. 3 Charging system

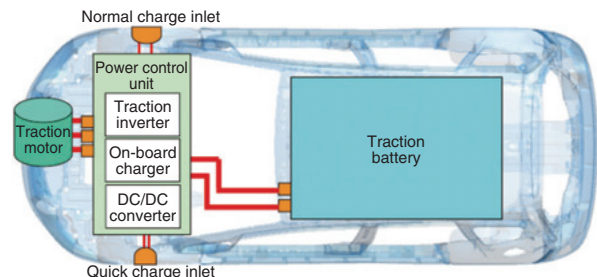


Fig. 4 Vehicle layout for FIT EV (Japan)

charger are all controlled by a battery Electric Control Unit (ECU). All of the communications wires necessary for charging are therefore connected to the battery ECU. Because the state of the traction battery and charge control are intimately related, it is necessary for a fail-safe function to go into effect rapidly in the event of a failure of the charging system. The charge control function was therefore incorporated in the battery ECU, which also has the twin functions of detecting the state of the battery and controlling the distribution of high-voltage power.

The control system for plug-in charging is made up of a communications function between electric vehicle supply equipment (EVSE) and the vehicle, charging output instructions and control, and charging system failure detection and fail-safe action.

3.2.1. Composition of normal charging system

AC commercial electric power input from the EVSE is transmitted via power wires from the vehicle inlet to the on-board charger. The on-board charger converts the AC power to DC, and charges the traction battery. The EVSE is connected to the battery ECU by means of communications wires in the inlet.

3.2.2. Composition of quick-charging system

The quick-charging system is compatible with CHAdeMO V0.9.

DC power input from the charger is transmitted from the vehicle inlet to the traction battery via the contactors. The charger is connected to the battery ECU by means of communications wires in the inlet.

3.2.3. Position of charge lids

Table 1 shows the position of the charge lids for each sales area.

(1) Normal charging

In both Japan and the U.S., home charging is the standard method of normal charging. The user plugs the vehicle in after parking it in the garage, and the charge lids for normal charging were therefore positioned on the driver's seat side in order to help ensure sufficient space.

(2) Quick charging

Because numerous quick-charging stations are located at the side of the road (for example in rest areas on expressways), the quick-charging lid was positioned on the passenger seat side.

4. Technologies Employed to Realize Charging in a Short Period

In order to realize charging in a short period, it was necessary to increase the output of the on-board charger. Figure 5 shows an external view of the on-board charger.

Table 2 shows a comparison of the specifications of the on-board chargers used in the FIT EV and the EV

Table 1 Location of charge lid

Sales area	Normal charge	Quick charge
Japan	Right side	Left side
USA	Left side	

Table 2 Comparison of on-board charger specifications

	EV Plus	FIT EV
Volume	8.6 L	8.1 L
Weight	11 kg	10 kg
Power	4.2 kw	6.6 kW
Cooling	Water	Water
Efficiency	84.4%	91.9%

PLUS. The FIT EV charger is more compact than the unit employed in the EV PLUS while realizing increased power. The charger produces 6.6 kW from a smaller unit than a competitor's 3.3 kW on-board charger. Figure 6 shows the relative positioning of the FIT EV charger against other models.

The rest of this section will discuss the technologies employed to reduce the size of the charger despite its increased output.

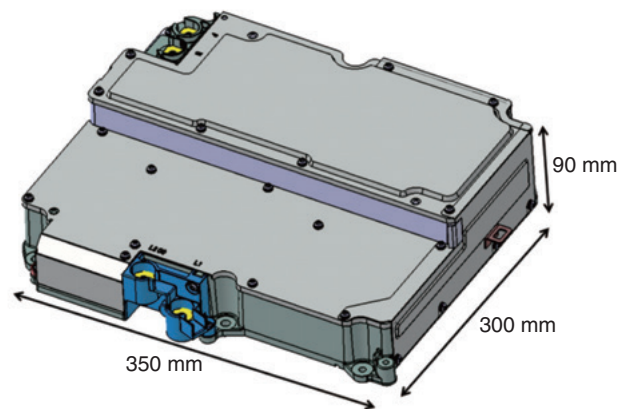


Fig. 5 External view of 6.6 kW on-board charger

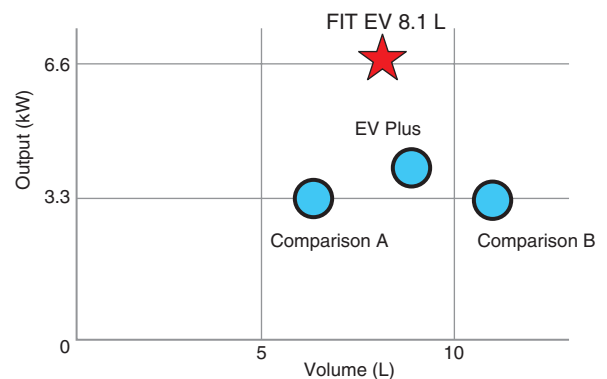


Fig. 6 Volume-power ratio

4.1. Cooling Configuration

The on-board charger is positioned inside the Power Control Unit (PCU). The PCU is fitted with a water jacket for cooling, and the on-board charger is attached to the water jacket to cool it. Figures 7 and 8 show the configuration.

4.2. Reduction of Capacity of Power Factor Correction (PFC) Circuit Output Electrolytic Capacitors

Figure 9 shows a block diagram of the on-board charger circuits.

The electrolytic capacitors used in power factor correction (PFC) circuits have the function of helping to ensure the ripple voltage and minimum voltage that are essential to the normal functioning of the DC/DC converter

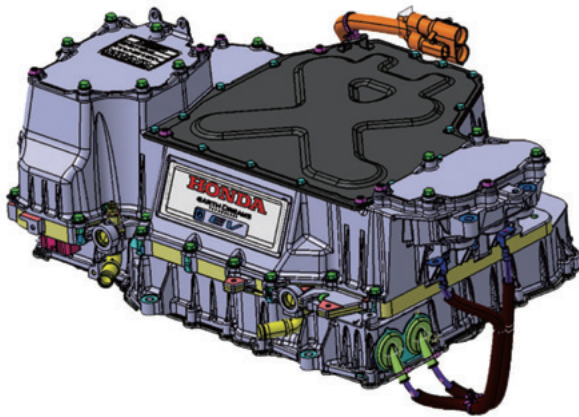


Fig. 7 External view of power control unit

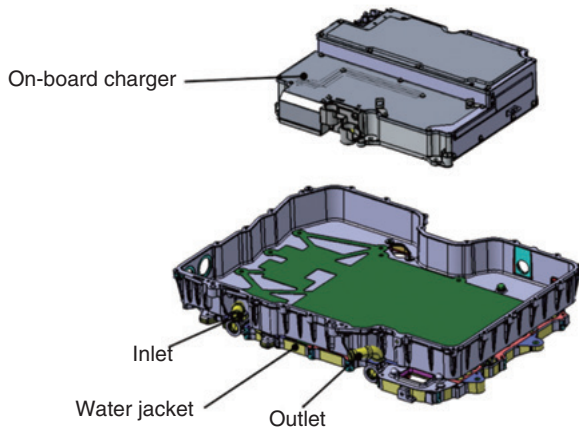


Fig. 8 Cooling structure of on-board charger

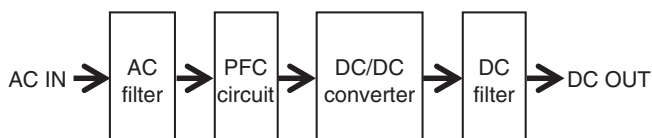


Fig. 9 Block diagram of on-board charger circuits

circuits in the on-board charger. The following methods were employed to address the issue of reducing size by reducing their capacity to the minimum.

4.2.1. Measures in response to increased ripple voltage

When the capacity of the electrolytic capacitors was reduced, ripple voltage superimposed on the commercial power supply sine waves increased. A measurement method in which the value was averaged for every commercial half-cycle was adopted in order to cancel the amount of increase by averaging during measurement.

4.2.2. Measures in response to increase in voltage fluctuation with sudden changes in input voltage

When the capacity of the electrolytic capacitors was decreased, the ability of the system to absorb fluctuations due to sudden changes in the voltage of the commercial power supply declined. To help ensure that this did not result in the termination of charging by the on-board charger due to the operation of over-voltage protection, the system was configured to suspend operation at a lower voltage than the level for operation of the over-voltage protection in order to help prevent voltage increase.

4.3. Increased Switching Frequency

The use of a switching frequency of 90 kHz reduced the magnetic flux density of the windings of the transformer and other parts, making it possible to reduce the size of the cores.

In addition, because the ripple current in the windings was reduced, it was also possible to reduce the size of the smoothing capacitors.

5. Technologies to Realize Increased Charging Efficiency

The energy efficiency (MPGe rating) of EV is evaluated as plug-to-wheel efficiency, based on the tank-to-wheel efficiency (fuel efficiency) used as a benchmark for ICE vehicles. Charging efficiency, an aspect of the MPGe rating, is the power ratio between the output end (the outlet) and the input end (the traction battery). Charging time is the time necessary to proceed from a state in which the low charge indicator is lit to a state of full charge.

Given this, the ideal measure to increase the MPGe rating via the range covered by plug-in charging would be to reduce power loss from the outlet to the traction battery to zero.

As will be discussed below, comprehensive reduction of charging-related loss during the development of the FIT EV charging system resulted in the achievement of a charging efficiency of 87.9%.

5.1. Reduction of Loss via Reduction of Charging Time

The reduction of charging time not only has considerable benefits from the perspective of users, but also significantly

affects charging efficiency.

During plug-in charging, devices including the on-board charger and battery cooling system are in operation. If the charging time is lengthy, the energy loss associated with the operation of these devices reduces charging efficiency per charge.

In order to reduce charging time in the FIT EV charging system, both hardware-based measures, in particular the realization of increased output from the on-board charger, and software-based measures to realize charging control able to maximize charger performance, were employed.

On the hardware side, the output of the charger was determined based on consideration of compliance with legally stipulated regulations and standards and potential increase in the size of the charger.

On the software side, a control specification that would maximally increase the range of constant power charging, making it possible to maximize the performance of the hardware, was realized by means of accurate prediction of the relationship between the voltage characteristic during charging and the SOC by the battery ECU. Switching to constant voltage charging during the final stage of the charging process controls overshoot, which reduces the durability of the battery, in addition to increasing the accuracy of full charge. Figure 10 shows the relationship between SOC and charging power.

5.2. Design of On-board Charger for High Efficiency

In a plug-in charging system, loss from the on-board charger represents the major factor in loss. Reduction of loss from the on-board charger contributed to increased efficiency, reduction of size and weight, and reduction of power consumption for cooling.

Figure 11 shows efficiency for different output levels of the on-board charger.

5.2.1. Use of soft switching

Soft switching has been employed to reduce switching

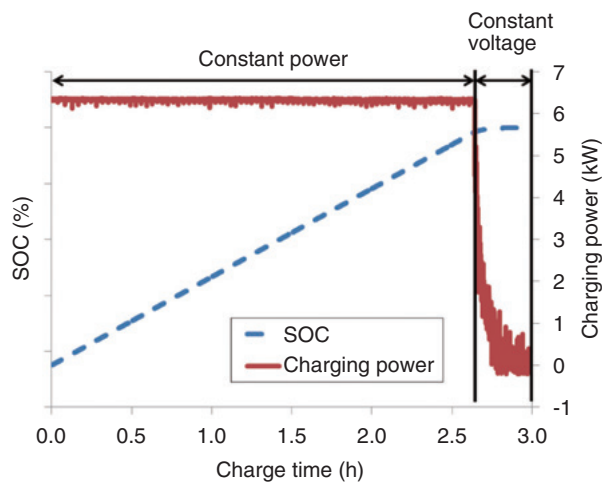


Fig. 10 Charging power control

loss. Figure 12 shows the soft-switching circuit employed in the FIT EV. As the circuit diagram shows, soft switching has been realized through the use of a resonance choke in the circuit.

5.2.2. Reduction of loss from rectifying diodes

The reduction of surge voltage due to the application of soft switching made it possible to reduce the withstand voltage of the diodes from 1200 V to 600 V. This reduced the diode recovery current, increasing the efficiency of the DC/DC converter in the on-board charger by approximately 1%. Figure 13 shows the switching waveform and loss.

5.2.3. Reduction of loss in transformer and chokes

(1) Reduction of loss in coils

Due to the skin effect, high-frequency alternating current does not flow for a depth equivalent to the skin depth. The surface area of the electrical wires can be increased in order to reduce the resistance of the alternating current as a result of this phenomenon. Litz wire was therefore employed. Figure 14 shows a structure of a Litz wire.

For the cores, a dispersed arrangement that reduced individual gap widths without changing the total gap length

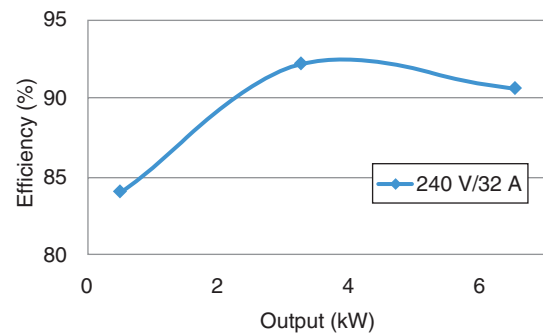


Fig. 11 On-board charger efficiency

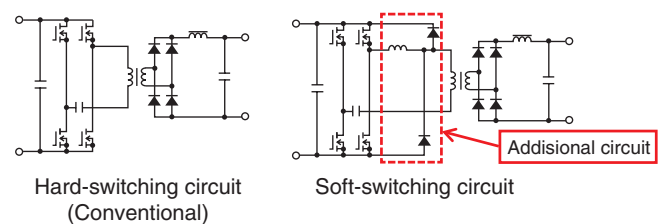


Fig. 12 Comparison of switching circuits

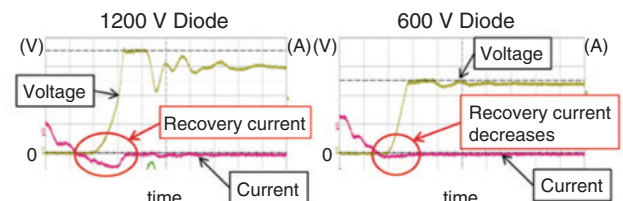


Fig. 13 Comparison of diode recovery current

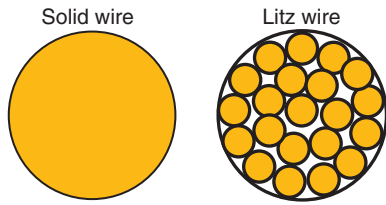


Fig. 14 Structure of solid wire and litz wire

was adopted. This reduced the degree of linkage of the leakage flux with the coils, reducing loss.

(2) Reduction of loss in the cores

Dust cores were employed in order to reduce loss. When dust cores are employed, heat is dissipated from the surface into the air due to the toroidal shape of the cores, thermal resistance becomes the issue. Resin molding of the choke was considered as a means of addressing the issue, but half-molding was finally adopted due to the increase in weight associated with the former method.

6. Technologies for Increased Safety

When charge connectors installed in vehicles deteriorate, connection resistance increases and excessive heat may be generated. A thermistor that detects excessive heat has therefore been fitted in the terminal employed in the vehicle inlet; when a preset temperature threshold is reached, charging current is reduced to help prevent excessive heat in the charge connector. Figure 15 shows the mounting configuration for a thermistor in an inlet for normal charging.

This measure is employed for both quick charging and normal charging. With consideration of the heat resistance of the structural parts of the charge connectors, a value that would not result in mistaken detection was employed as the threshold value for heat protection.

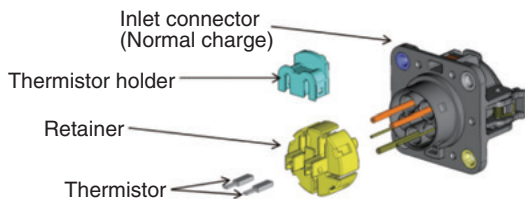


Fig. 15 Structure of inlet thermistor

7. Conclusion

A compact and lightweight on-board charger has been developed that increases the efficiency of the FIT EV plug-in charging system as a whole and introduces the latest technologies to the system. As a result, a charging efficiency of 87.9% has been achieved, and targets for the realization of a short charging time, high efficiency, and a

high level of safety have been met. According to a Honda survey conducted at the end of August 2012, the system achieved the world's highest MPGe rating. The system is also sufficiently competitive as a product, able to charge a battery within three hours.

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Development of Electric Servo Brake System

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ABSTRACT

An electric servo brake system realizing good brake feel while making effective use of regenerative braking has been developed in order to increase power efficiency in electric vehicles and fuel efficiency in hybrid electric vehicles. This system controls brake pressure using hydraulic pressure cylinders with an electric motor as the drive source. By means of this configuration, it realizes high-accuracy brake pressure control that cooperates with regenerative braking, and its use in the Fit EV has increased power regeneration by 8% against a conventional system based on EPA fuel economy standards. In addition, the use of a configuration in which the pedal operation section is independent from the brake pressure generation section helps to prevent the pedal force from being affected by brake pressure control, making it possible to realize good brake feel.

1. Introduction

Against the background of increasing demands for the reduction of fuel consumption and CO₂ emissions in order to protect the global environment, electric vehicles (EV), hybrid electric vehicles (HEV), and other vehicles fitted with electric power trains are attracting increased attention year by year. EV and HEV are able to realize increased power and fuel efficiency by recovering kinetic energy as regenerated energy through the use of regenerative brakes when decelerating. It is therefore important to efficiently produce regenerative braking energy during deceleration in order to increase the amount of regenerated power. What is needed in order to realize this goal is brake systems provided with a cooperative regenerative braking function, making it possible, in response to the state of the system, to cooperatively allocate friction and regenerative braking force for the braking force demanded by the driver. In order to realize the braking feel intended by the driver while producing regenerative braking force, it is necessary to constantly coordinate the total of regenerative braking force and friction braking force with the value demanded by the driver^{(1), (2)}. To realize the same good brake feel as a normal brake system using a vacuum booster at this time, it is desirable that the pedal force should not be affected by brake pressure control.

A variety of systems have been proposed to date in order to realize a cooperative regenerative braking function at the same time as achieving a good brake feel. These include brake-by-wire systems⁽³⁾, in which the brake pressure is controlled independently for each wheel, and systems that employ identical pressure for each wheel while being capable of freely generating brake pressure as demanded;

these latter systems include hydraulic booster types^{(4), (5)}, vacuum booster types⁽⁶⁾, and electric booster types⁽⁷⁾. The realization of a good brake feel during cooperative regenerative braking has represented an issue for systems in which the pedal operation section and the brake pressure generation section are linked, because pedal force can be affected by brake pressure control. An electric servo brake system has therefore been newly developed in order to contribute to increased power efficiency in EV and fuel efficiency in HEV by realizing a cooperative regenerative braking function while addressing this issue. The developed system employs a configuration in which the pedal operation section and the brake pressure generation section are independent. It employs a pedal stroke and pedal force simulation function in the pedal operation section. The brake pressure generation section generates friction braking force by means of brake pressure control using hydraulic cylinders, with an electric motor as the drive source, and is able to freely control brake pressure.

This paper provides an overview of the developed electric brake system, and discusses element technologies related to brake pressure control and the cooperative regenerative braking function.

2. Development Goals

In order to reconcile the reduction of power consumption in EV and fuel consumption in HEV with brake performance, the following main aims were set for the development of the electric servo brake system:

- (1) Realization of a cooperative regenerative braking function and the achievement of a high level of regeneration efficiency

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- (2) Realization of good brake feel
- (3) Realization of better non-servo performance than a conventional brake system

3. System Overview

The developed system is composed of a pedal feel simulator, a tandem motor cylinder, a brake pedal, a pedal stroke sensor, and an ECU, as shown in Fig. 1. The pedal feel simulator produces a pedal stroke and pedal force when the brake pedal is operated. The tandem motor cylinder generates brake pressure as commanded by the ECU based on the degree of pedal operation detected by the pedal stroke sensor.

3.1. Pedal Feel Simulator

As Fig. 2 shows, the pedal feel simulator is made up of a master cylinder, a stroke simulator, a solenoid valve, and a hydraulic pressure sensor.

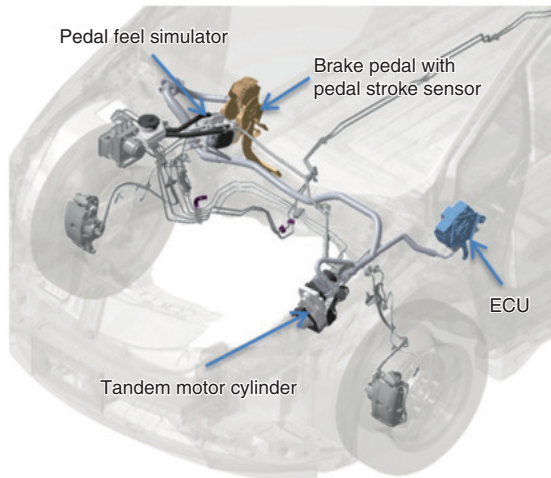


Fig. 1 Overview of electric servo brake system

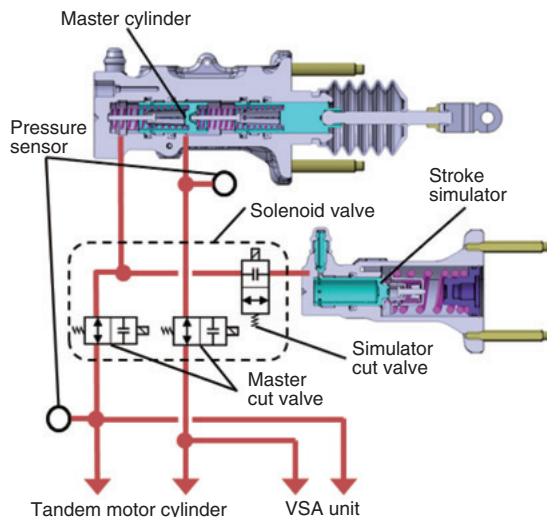


Fig. 2 Structure of pedal feel simulator

The master cylinder is linked to the brake pedal, and connected to the stroke simulator by a hydraulic channel. A simulator cut valve and a hydraulic pressure sensor are positioned in this hydraulic channel. The simulator cut valve is a normally closed solenoid valve. The pedal stroke and pedal force simulation functions are realized by means of this configuration. The pedal force characteristic can be adjusted by setting the brake pedal link ratio and the spring reaction force characteristic of the stroke simulator.

The master cylinder and the tandem motor cylinder are also connected by a hydraulic channel, and a master cut valve and hydraulic pressure sensor are positioned in this hydraulic channel. The master cut valve is a normally open solenoid valve. In addition, the tandem motor cylinder and the Vehicle Stability Assist (VSA) unit are connected by a hydraulic channel downstream from the master cut valve. Because the brake pressure generated by the tandem motor cylinder is isolated in this configuration, pedal force is not affected by brake pressure control. In addition, the hydraulic circuits are switched at system start-up and shutdown.

3.2. Tandem Motor Cylinder

As shown in Fig. 3, the tandem motor cylinder is made up of a brushless motor, a reduction gear, a ball screw and a slave cylinder. A resolver is built into the brushless motor.

The brushless motor and the slave cylinder are connected by the reduction gears and the ball screw. Brake pressure is generated by direct pressure of the ball screw on the slave cylinder piston, making possible high-accuracy brake pressure control.

Figure 4 shows the operation of the electric servo brake system when it is generating brake pressure. Based on the degree of pedal operation detected by the pedal stroke sensor, the ECU decides on a brake pressure target, and the brake pressure control, which will be discussed below, generates the brake pressure by simultaneously operating the brushless motor and closing the master cut valve. The brake pressure is transmitted to the wheel cylinder via the VSA unit.

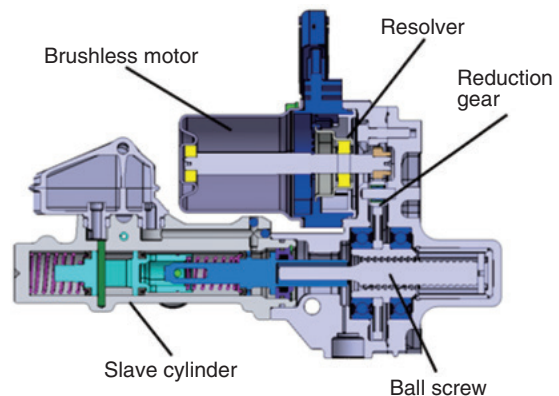


Fig. 3 Structure of tandem motor cylinder

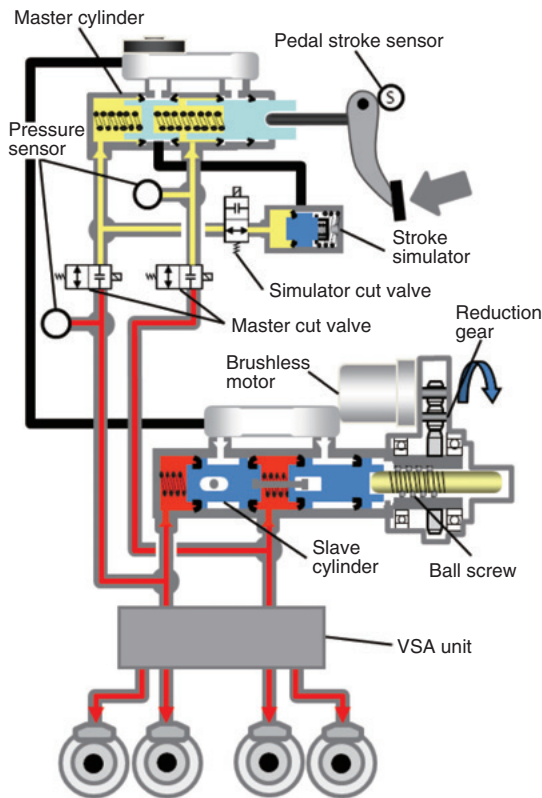


Fig. 4 System configuration diagram

4. Brake Characteristics

4.1. Basic Brake Characteristics

In order to realize good brake feel, the brake pedal link ratio and the spring reaction force characteristic of the stroke simulator were adjusted in order to achieve a pedal force characteristic similar to that of a standard brake system. Figure 5 shows a comparison of the pedal force characteristic of the electric servo brake system and a standard brake system. Appropriate adjustment of the parameters discussed above has made it possible to realize a similar pedal force characteristic in the electric servo brake system as displayed by a standard brake system.

The setting for the brake pressure target in relation to the degree of pedal operation was also adjusted with the aim of achieving a similar characteristic in terms of the effectiveness of braking to a standard brake system. Figure 6 shows a comparison of the brake effectiveness characteristic of the electric servo brake system and a standard brake system. The electric servo brake system realizes a similar brake effectiveness characteristic to the standard brake system. Together, this brake effectiveness characteristic and the pedal force characteristic produce a good brake feel.

4.2. Non-servo Performance

The system employs a configuration in which the pedal input is transmitted directly to the master cylinder and brake

pressure is generated and transmitted to the wheel cylinder when the servo is not in use, for example when the system is shut down. Adjustment of the brake pedal link ratio and the cross-sectional area of the master cylinder piston has doubled the hydraulic pressure generated at a pedal force of 500 N when the servo is not in use against a vacuum booster brake system.

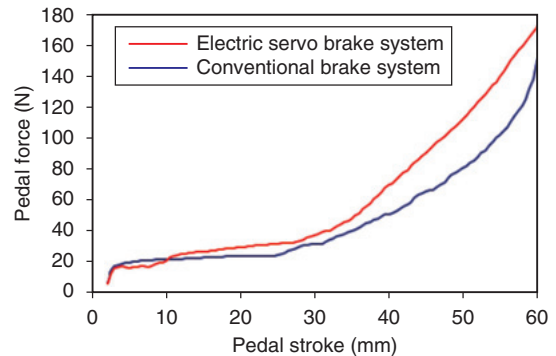


Fig. 5 Pedal force characteristic

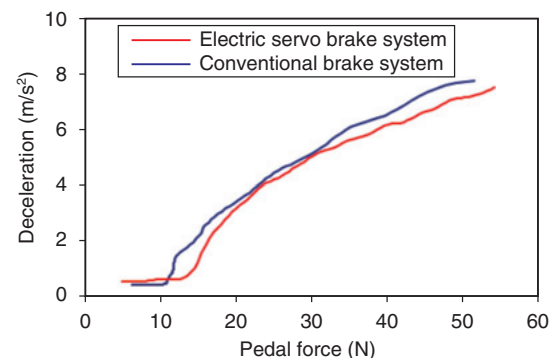


Fig. 6 Braking effectiveness

5. Brake Pressure Control

Brake pressure control functions as follows. First, a brake pressure target is decided on the basis of the degree of pedal operation and data concerning the upper limit value of regenerative torque able to be produced by the power train. Next, the brushless motor is operated by means of the hydraulic pressure control section and the motor angle control section shown in Fig. 7, and the target brake pressure is generated.

The hydraulic pressure control section outputs a stroke target for the slave cylinder piston in response to the brake pressure target. The hydraulic pressure control section is made up of a feedforward section and a feedback section. The feedforward section uses the slave cylinder piston stroke and a characteristic map of the generated brake pressure. Values from the hydraulic pressure sensor positioned downstream from the master cut valve are input to the feedback section. Figure 8 shows the brake pressure response. As Fig. 8 shows, this hydraulic pressure control

configuration has made it possible to realize good response to, and good tracking performance of, brake pressure. Figure 9 shows the slave cylinder pressure response during ABS operation. Even in a situation in which the brake load fluctuates, such as ABS operation, the system displays adequate control stability and realizes high-accuracy hydraulic pressure control.

The motor angle control section employs angle feedback control using a resolver, and outputs a motor drive voltage. In addition, in order to realize a high level of responsiveness from the compact brushless motor, Flux Weakening Control (FWC) has been employed to make it possible to increase the motor's speed⁽⁸⁾. Figure 10 shows a comparison of the step response of brake pressure and motor speed with and without flux weakening control. The use of flux weakening control increases motor speed by 30% and reduces response time by 25%, making it possible to realize a high level of responsiveness from a compact motor.

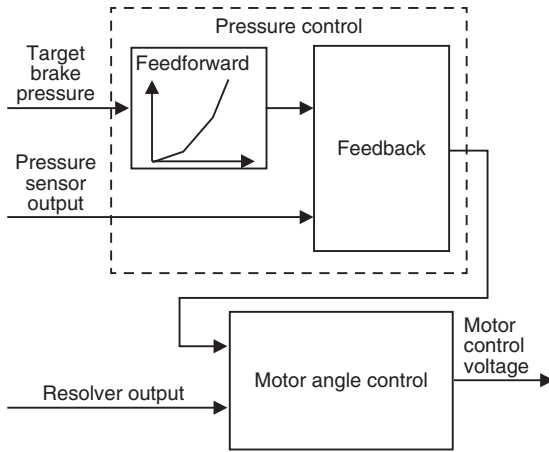


Fig. 7 Block diagram of brake pressure control

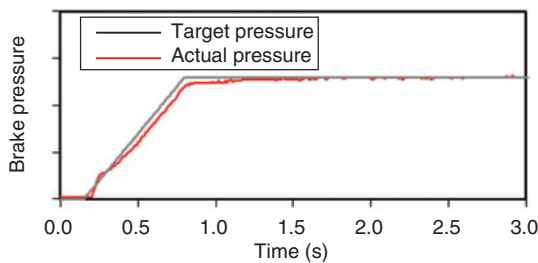


Fig. 8 Response of brake pressure

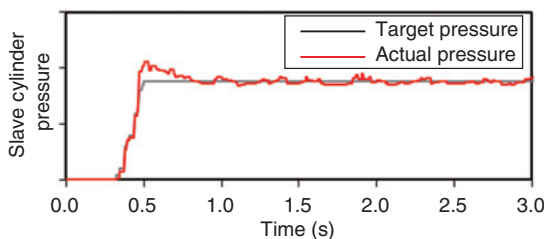
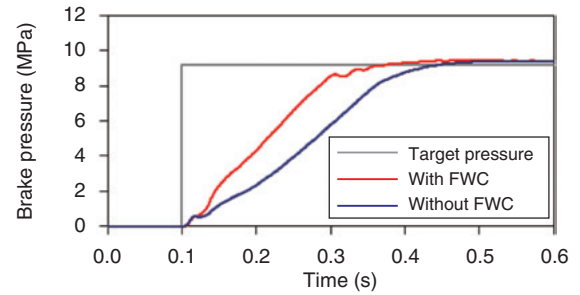
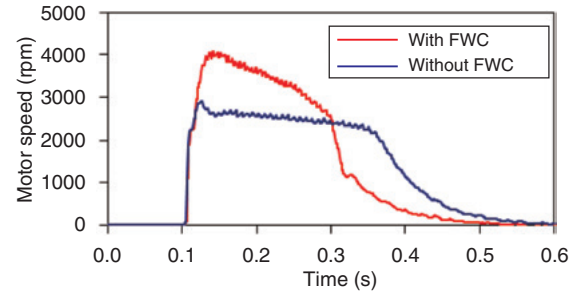


Fig. 9 Slave cylinder pressure response during ABS operation



(a) Brake pressure



(b) Motor speed

Fig. 10 Step response with flux weakening control (FWC)

6. Cooperative Regenerative Braking

The reduction of energy loss when friction brakes are employed during vehicle operation is an important factor in increasing the power and fuel efficiency of EV and HEV. One of the demands on cooperative regenerative braking is therefore to make as little use of the friction brakes as possible. However, there are situations in which regenerative brakes are not able to generate power due to the status of the power train, for example when the vehicle is traveling at high speeds and immediately before the vehicle stops. Unless it is possible to rapidly switch to the friction brakes in these cases, fluctuations in the degree of deceleration contrary to the driver's expectations will occur. Because of this, a high-accuracy brake pressure control system is essential to the realization of increased fuel efficiency by means of cooperative regenerative braking. The use of a configuration in which the pedal operation section and the brake pressure generation section are independent, in addition to the high-accuracy brake pressure control discussed above, have helped to enable high regeneration efficiency to be reconciled with the realization of good brake feel in the electric servo brake system.

Figure 11 shows time history data for cooperative regenerative braking control in the electric servo braking system during regular braking from 70 km/h. Until immediately before the vehicle stops, the frictional brakes are not used at all, and the vehicle's kinetic energy is recovered using the regenerative brakes alone. The degree of deceleration is constant and consistent with the driver's expectation, and there is no change in the brake feel due

to regeneration. Because the regenerative brakes cannot be used immediately before the vehicle stops and while it is stopped, it is necessary to switch to the friction brakes. The characteristics of the electric servo brake system's high-accuracy hydraulic brake pressure control have made it possible to reduce vehicle speed at the completion of the switching, V_e , to 2 km/h. In the FIT EV, the use of the electric servo brake system has increased regeneration amount by 8% against a conventional system as measured by EPA fuel efficiency rating standards.

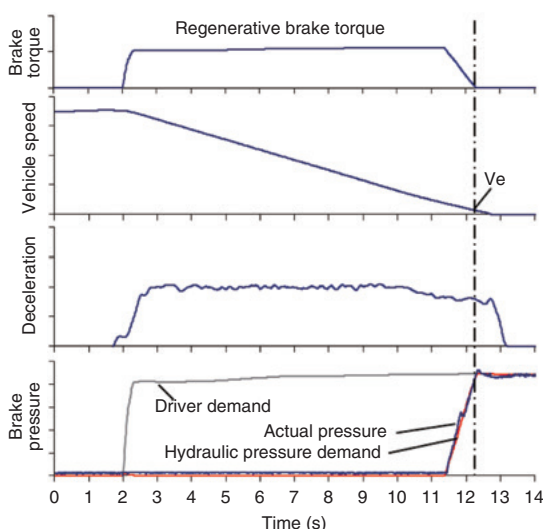


Fig. 11 Time chart of regeneration braking

7. Conclusion

An electric brake system realizing good controllability has been developed. The system generates brake pressure using a configuration that separates the pedal operation and brake pressure generation sections and employs an electric motor, a ball screw, and hydraulic cylinders. The system both realizes increased energy recovery efficiency during the use of cooperative regenerative braking and offers the driver smooth, natural braking feel. It is hoped that the development of this electric servo brake system will contribute significantly to the further spread of electric vehicles.

Details of the main technologies employed in the system can be summarized as follows.

- (1) The use of motor angle control, in addition to hydraulic pressure control comprising feed forward and feedback control and flux weakening control, has made it possible to realize high-accuracy and highly responsive electric brake control, thus realizing high regeneration efficiency.
- (2) The use of a configuration in which the pedal operation section is independent of the brake pressure generation section, optimization of the simulator spring reaction force characteristic and the brake pedal link ratio, and the use of high-accuracy and highly responsive electric brake control has made it possible to realize a brake feel equivalent to that offered by a standard brake system.

- (3) Optimization of the brake pedal link ratio and the cross-sectional area of the master cylinder pistons has made it possible to realize a higher level of non-servo performance than a conventional vacuum booster brake system.

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Development of Double-arm Chain-tensioner System

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ABSTRACT

Research was conducted with a focus on chain-tensioner systems in order to enhance the functions of chain systems. A double-arm chain-tensioner system was developed that differs from a conventional system in adding a sub-tensioner arm (“sub-arm” below) to make use of a lever ratio effect.

As a result, cost and weight have been reduced to levels below those of a conventional system, and chain friction and chain tension have been reduced across the entire operating range. In addition, the rate of chain elongation has been reduced compared to that of a conventional system.

1. Introduction

Among engine cam drive systems, chain drive systems are both more durable and more compact than belt drive systems, and are therefore used in the majority of engines. However, chain drive systems also present the disadvantage of greater friction and elongation due to wear compared to belt drive systems. Chain drive systems use chain guides and tensioner arms, which are not found in belt drive systems. As the chain slides against the chain guide and tensioner arm, resistance is generated. It has been reported that this sliding resistance with the chain guide and tensioner arm represents 90% or more of chain friction⁽¹⁾. Given this, the reduction of sliding resistance would be an effective means of reducing chain friction. Figure 1 shows friction in a cam drive system against engine displacement at an engine speed of 1500 rpm. The chain drive system displays approximately twice the friction loss of the belt

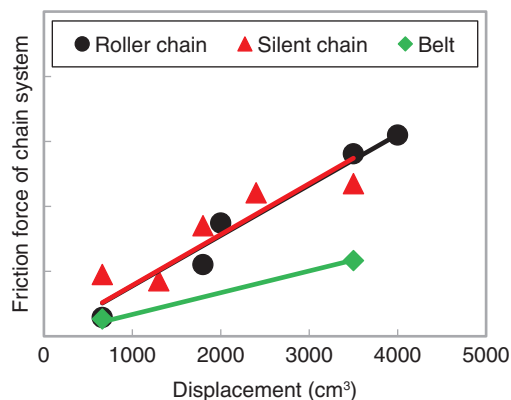


Fig. 1 Friction force of camshaft drive system

drive system. Friction resistance reduction methods such as the application of surface treatments to sliding parts are available to reduce sliding resistance in chain drive systems, but significant friction reductions cannot be expected using these methods. The reduction of average chain tension and the amplitude of chain tension, the sources of sliding resistance, are therefore important considerations in the reduction of chain friction.

Figure 2 shows a chain drive system using a hydraulic chain tensioner (termed a “chain tensioner” below). Chain drive systems use tensioner arms and chain tensioners positioned on the chain-unmeshing side of the crank sprocket in order to control chain vibration produced by factors such as torque fluctuations when the engine is in operation. In addition, a spring is situated inside the chain tensioner in order to apply a load to the chain. The chain tensioner controls the amplitude of chain tension by drawing engine oil into the interior of a plunger when the chain is slack and the plunger is extended, and attenuating the load exerted on the plunger when the chain is under

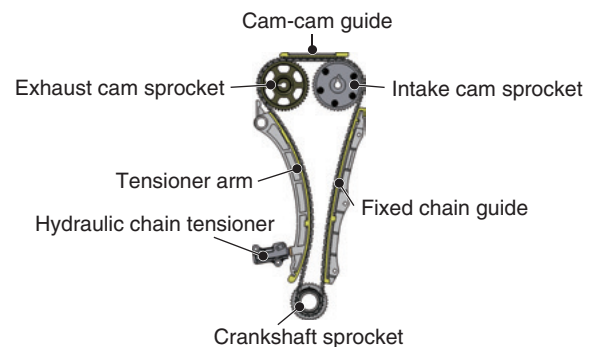


Fig. 2 Camshaft drive chain system

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tension. With the repetition of engine operation, the chain becomes worn and elongated. When the chain becomes elongated, the chain-tensioner plunger extends further to follow the chain. When the plunger extends further, the load placed on the chain by the spring in the chain tensioner is reduced. If the load becomes too low, the behavior of the chain becomes unstable, resulting in functional issues such as poor meshing between the chain and the sprocket. Because of this, it is essential to set the spring load such that the necessary load can be applied even when the chain becomes elongated. However, if the spring load is set too high, friction increases at the initial stage of chain life, before it has become elongated. As this indicates, the chain-tensioner system significantly affects the average chain tension due to the load placed on the chain by the spring inside the tensioner and the amplitude of chain tension during engine operation.

Consideration of chain-tensioner systems indicates that if it were possible to reduce the level of extension of the plunger following the chain elongation, and to reduce the rate of chain elongation itself, it would be possible to reduce the average tension on the chain by reducing the setting of the load placed on the chain by the spring inside the chain tensioner when the chain is in its initial state, before it has become elongated. In addition, if it were possible to reduce the amplitude of chain tension, chain friction would be reduced and the level of fatigue strength demanded from the chain would also be reduced, making it possible to reduce the cost and weight of the chain by reducing the number of plates.

Using a 2.4 L engine with a chain drive system, the research discussed in this paper set out to reduce friction by 50% in order to achieve friction loss equivalent to a belt drive system, while realizing a cost and weight equivalent to or lower than in a conventional chain drive system. In order to realize these goals, the project developed a double-arm chain-tensioner system making it possible to reduce the average chain tension, the amplitude of chain tension, and the rate of chain elongation.

2. Double-arm Chain-tensioner System

Figure 3 shows an overview of the double-arm chain-tensioner system. The system differs from a conventional system in positioning a sub-arm between the tensioner arm and the chain tensioner. The positioning of the sub-arm produces lever ratios at the points of contact between the sub-arm and the tensioner arm and between the sub-arm and the chain tensioner due to the difference in their distance from the sub-arm support point. The effect of these lever ratios results in the following main differences between the double-arm chain-tensioner system and a conventional system:

(1) The degree of movement of the plunger in relation to the degree of movement of the tensioner arm due to the chain elongation is reduced.

- (2) The stroke and the speed of variation of the amplitude of the plunger in relation to the oscillation of the tensioner arm due to the amplitude of chain tension are reduced.
- (3) The load input to the plunger due to the amplitude of chain tension is increased.
- (4) The load applied to the chain by the chain-tensioner spring is reduced.

In addition, the positioning of the tensioner has the following effects:

- (5) The positioning of the chain-tensioner plunger facing upward enhances air-venting performance.
- (6) The positioning of the chain-tensioner plunger facing upwards helps to prevent oil leakage when the engine is stopped.

Making use of these effects, the following goals were set for functional enhancements in the development of a double-arm chain-tensioner system:

Friction: Based on the effect described in (1), reduction of friction by means of the reduction of the degree of movement of the plunger in response to the chain elongation.

Amplitude of chain tension: Based on the effect described in (2), reduction of the amplitude of chain tension with the increased ability of the tensioner to follow the chain at high engine speeds.

Reduction of chain elongation: Based on the effects described in (1) and (2), reduction of chain elongation by means of reduction of the surface pressure between the pins and the plates of the chain.

Cost and weight: Reduction of cost and weight through reduction of the number of chain plates by means of reduction of the amplitude of chain tension and chain elongation.

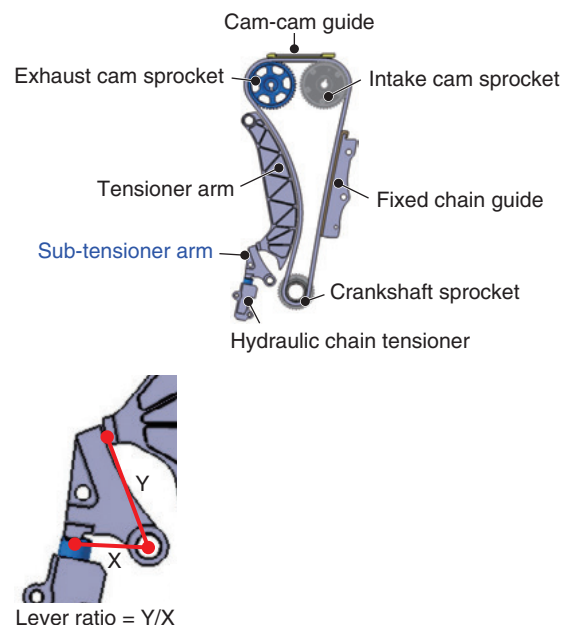


Fig. 3 Double-arm chain-tensioner system

Figure 4 shows changes in the load applied to the chain by the chain-tensioner spring due to chain elongation for a conventional chain-tensioner system and the double-arm chain-tensioner system. The load applied to the chain by the double-arm chain-tensioner system was set in order to make the load at the maximum chain elongation rate equivalent to that of a conventional system. It was possible to set the load on the chain in its initial state prior to elongation at a lower level in the double-arm chain-tensioner system as compared to the conventional system, and the changes in load in response to chain elongation are also less pronounced in the double-arm system. In addition to the fact that the degree of movement of the chain-tensioner plunger in relation to the degree of movement of the tensioner arm due to the chain elongation is lower in the double-arm system, this is also a result of the fact that the load setting took into consideration factors including changes in the lever ratios and changes in the efficiency of load transmission from the plunger spring to the chain due to chain elongation.

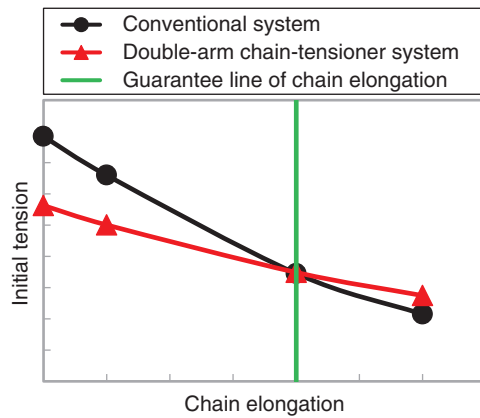


Fig. 4 Relation between chain elongation and initial tension

3. Basic Performance of Double-arm Chain-tensioner System

In order to verify the basic performance of the double-arm chain-tensioner system, the double-arm system and a conventional system were tested in engines of the same specifications; only the chain drive systems were changed. Table 1 provides an overview of the engine specifications. Figure 5 shows a comparison of measurements of chain tension during engine operation for the conventional system and the double-arm chain-tensioner system. Chain tension measurements were conducted by turning the crank sprocket into a load cell. Tension was regarded as positive on the fixed-guide side and negative on the tensioner-arm side. Figure 5 indicates that the amplitude of chain tension is lower for the double-arm chain-tensioner system across the entire range of engine operation. At high engine speeds, the

Table 1 Engine specifications

Cylinder configuration	Inline 4
Bore × Stroke (mm)	87.0 × 99.0
Displacement (cm ³)	2354
Compression ratio	10.5
Chain type	Silent chain
Chain pitch (mm)	6.35
Number of link plates	4/3
Number of links	176
Tensioner type	Hydraulic tensioner
Number of sprocket teeth	Crankshaft sprocket : 23 Camshaft sprocket : 46

increase in the ability of the chain tensioner to follow the chain due to the reduction of the amplitude of chain tension has resulted in a reduction of chain tension of approximately 35%. In addition, the results indicate that the reduction in the amplitude of chain tension would make it possible for this specification to satisfy fatigue strength demands even if the number of plates in the chain was reduced.

Figure 6 shows a comparison of the results of measurements of friction in the chain systems. The

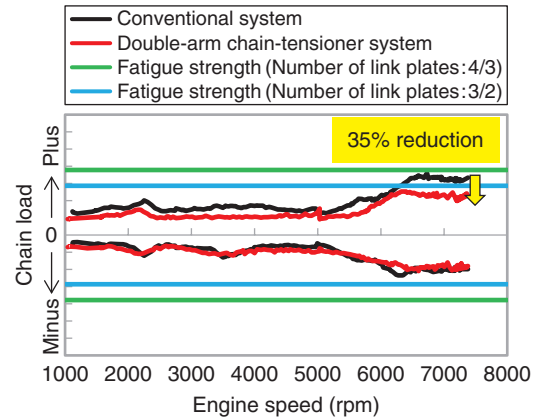


Fig. 5 Comparison of chain load

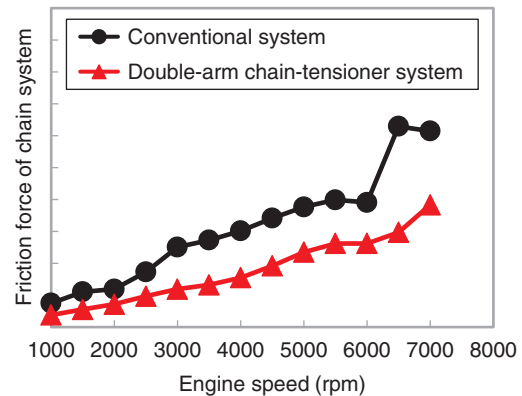


Fig. 6 Comparison of chain friction force

reduction of the amplitude of chain tension and average chain tension realized through the use of the double-arm chain-tensioner system reduces friction across the entire engine operating range. In addition, friction is reduced by 50% in the range up to 4000 rpm.

4. Evaluation of Reliability

In order to evaluate the reliability of the double-arm chain-tensioner system, the system was subjected to chain system durability tests and chain elongation evaluation tests using conditions in which the loading on the sub-arm support points would be at its maximum. Figure 7 shows a comparison of the surface pressure between the chain pins and plates calculated from the results of chain tension measurements. Figure 8 shows the results of the chain elongation evaluation tests. Figure 5 indicated that the

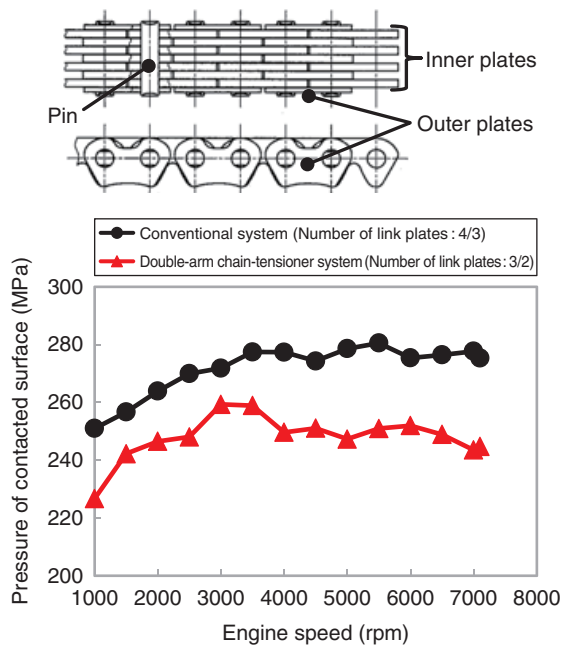


Fig. 7 Comparison of pressure of contacted surface

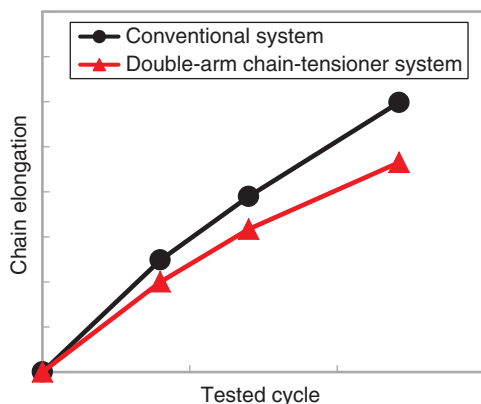


Fig. 8 Comparison of chain elongation

reduction of the amplitude of chain tension in the double-arm chain-tensioner system would make it possible for the system to satisfy fatigue strength demands despite the reduction of the number of plates in the chain. Figure 7 indicates that as a result of the reduction of average chain tension and the amplitude of chain tension, the surface pressure between the chain pins and plates would be lower in the double-arm chain-tensioner system than the conventional system even if the number of plates in the chain was reduced. Given that these results demonstrate that the use of the double-arm chain-tensioner system increases reliability, tests for the evaluation of reliability were performed on a specification in which the number of plates was reduced against the conventional system. Figure 8 indicates that despite the reduction in the number of plates, the effect of the reduction in surface pressure reduces the chain elongation rate by 30% in the double-arm chain-tensioner system as compared to the conventional system. In addition, the results of durability tests using maximum loading on the sub-arm showed that no malfunctions occurred under these conditions.

5. Cost and Weight

Figure 9 shows a comparison of the cost and weight of a conventional system and the double-arm chain-tensioner system. The reduction of the number of plates in the double-arm chain-tensioner system reduces cost by 2.5% and weight by 0.8% against the conventional system.

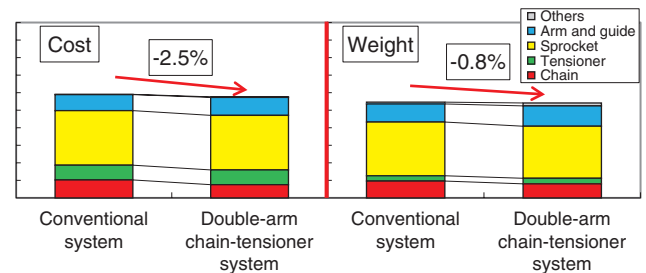


Fig. 9 Cost and weight

6. Conclusion

The following results were achieved in the development of a double-arm chain-tensioner system.

- (1) Friction in the system has been reduced across the entire range of engine operation. In particular, friction has been reduced by 50% in the range up to 4000 rpm.
- (2) The amplitude of chain tension has been reduced across the entire range of engine operation. In particular, the amplitude of chain tension has been reduced by a maximum of 35% in the high-speed range.
- (3) The reduction of the average chain tension and the amplitude of chain tension made it possible to reduce

the number of plates in the chain. This has reduced cost by 2.5% and weight by 0.8% against a conventional system. In addition, despite the reduction in the number of plates, the chain elongation rate has been reduced by 30%.

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Masao SEKI

Development of Gear Case for Large Outboard Motor

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ABSTRACT

As part of the development process for its flagship large outboard motor, the BF250, Honda has developed a gear case for large outboard motors in order to realize increased reliability and power performance against existing models.

Enhanced tooth contact has been achieved through redesign of the structure of the bearing mountings, and pitting resistance has been increased by modifying the gear materials and heat treatment. In addition, gear specifications making it possible to realize high durability under high loads have been employed, and the optimization of tooth contact via finite element model (FEM) analysis with consideration of stiffness and manufacturing variations has assisted in the realization of a tooth grinding process that reduces manufacturing variations.

These innovations have resulted in the realization of a gear case that achieves class-topping power performance while satisfying targets for durability performance.

1. Introduction

Today, there is a tendency towards the use of increased horsepower capacity in offshore boats manufactured mainly in Europe and North America. A high proportion of these outboard motors are employed under conditions of prolonged use at high loads in boats for professional use, and durability and reliability are therefore demanded.

This paper will discuss the technologies applied in a gear case for a large outboard motor as part of the development of the BF250⁽²⁾ in response to these demands. Figure 1 shows the external appearance of the gear case.



Fig. 1 External view of gear case

2. Demands on Drive Systems of Large Outboard Motors and Development Goals

Professional (commercial and government) use of boats fitted with large outboard motors represents approximately 30% of the market, with standard leisure use making up the remainder. This proportion of professional use is high in comparison to other horsepower ranges (Fig. 2).

A great deal of professional use is conducted under severe conditions of high loads for extended periods, making a high level of durability an important factor.

While use time in the high-load range (3850 rpm to

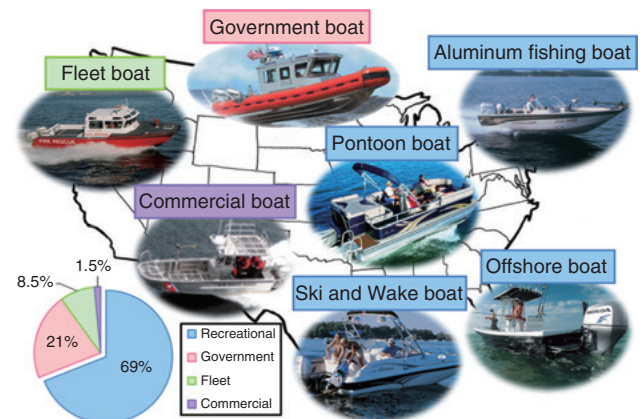


Fig. 2 Variety and percentage of large boats in the market

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WOT) is thought to represent 30% or less of total use time for conventional professional use, in the case of boat taxis (tourist boats) the figure is double this at 60% (Fig. 3).

Figure 4 shows boats used for applications such as boat taxis and coastal patrols as examples of this type of extended high-load operation.

Because the submerged gear case represents a resistance when the boat is in operation, it is necessary to increase durability while satisfying power performance demands.

Based on the above considerations, the following aims were set for the development discussed here:

- Realization of class-topping power performance (Balancing maximum speed performance and acceleration performance)
- Realization of adequate durability for high-load operation for extended periods (Realization of level of durability twice or more as high as a conventional gear case in component durability tests)

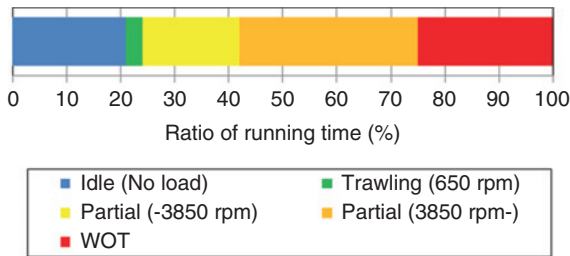


Fig. 3 Ratio of running time represented by different loads for boat taxi



Fig. 4 Boat taxi and coastal patrol boat

3. Basic Structure of Outboard Motor Gear Case

The basic structure of an outboard motor gear case orthogonally transforms power from the engine shaft by

means of pinion and bevel gears, and transmits the power to the propeller shaft. The structure considers the support of propeller thrust, the minimization of water resistance, and the realization of watertightness. Figure 5 shows this basic structure.

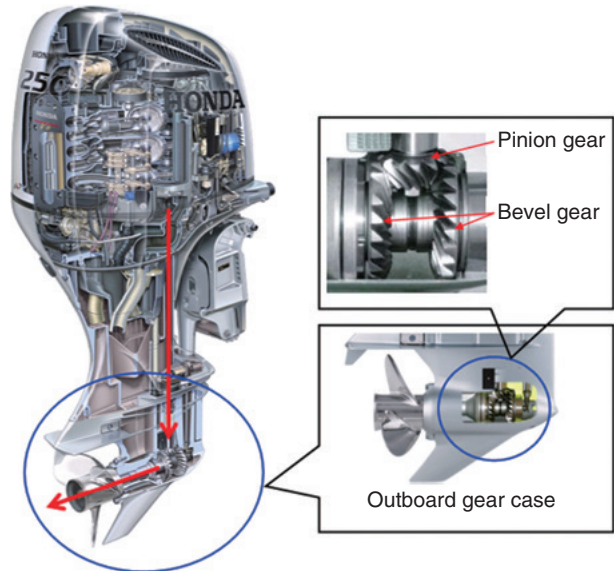


Fig. 5 Standard outboard gear case

4. Main Technologies

This section introduces the element technologies employed in the developed gear case.

4.1. Gear Case Specifications

The setting of a reduction ratio at the commencement of the design process is an important factor in the design of a gear case. Outboard motors differ from four-wheeled and two-wheeled vehicles in that the ultimate transmitter of power is a propeller, and the diameter of the propeller produces differences in propulsion efficiency. For this development project, the propeller diameter and reduction ratio producing the highest efficiency in the representative speed range for professional use were selected. Figure 6 shows the relationship between these parameters, with a comparison with a conventional gear case.

In order to increase durability while modifying the reduction ratio of the gear case, the diameter of the bevel gears was increased by 10% against a conventional gear case. This increased underwater resistance by 2.4% against a conventional gear case, but optimization of the case's shape based on a computational fluid dynamics (CFD) analysis of the pressure on the surface of the unit made it possible to reduce its C_d by 5%.

Figure 7 shows the results of CFD analysis of the pressure on the surface of the gear case.

4.2. Configuration of Bearings

4.2.1. Basic structure of bearings

Prior to discussing the basic structure of the bearings, this section will provide a brief discussion of methods used when fitting large outboard motors. Normally, the propeller of an outboard motor rotates clockwise when seen from behind. This is known as the standard rotation specification. In some cases, however, a boat might be fitted with two outboard motors, as shown in Fig. 8. When this is the case, if the direction of rotation of the propellers were the same, the torque reaction force would be high, affecting the boat's straight-line stability. Because of this, when two motors are fitted, opposite directions of rotation are employed for the propellers of the right and left motors, mutually canceling out torque reaction force and helping to maintain straight-line stability. The specification of the motor in which the propeller is rotating counter-clockwise when viewed from behind is termed the counter-rotation specification⁽¹⁾.

Turning to the current basic structure of the bearings, because the direction of propeller rotation differs for the standard and counter-rotation specifications even when the direction of engine rotation is the same, it is necessary to use a different position for the forward gear for each specification. Because of this, the structure of the bearings differs for each specification. At present, tapered bearings,

which are able to bear thrust load, are employed for the frequently used forward gear, while ball bearings are employed for the less frequently used reverse gear.

Figure 9 shows the difference between these structures.

4.2.2. Modification of structure of bearings for counter-rotation specification

A consideration of the direction of the tapered bearings in Fig. 9 shows that they face in the same direction for both the standard and counter-rotation specifications.

In the case of the standard rotation specification, both the gear thrust force and the propeller force act in the same direction; both loads are able to be received by the outer ring of the tapered bearings, and gear tooth contact is stable regardless of propeller force. By contrast, in the counter-rotation specification, because the gear thrust force and the propeller force act in opposite directions, unless the propeller force is greater than the gear thrust force, the loads cannot be received by the outer ring of the tapered bearings, and gear tooth contact becomes unstable. In the newly developed gear case, rather than supporting the forces separated by

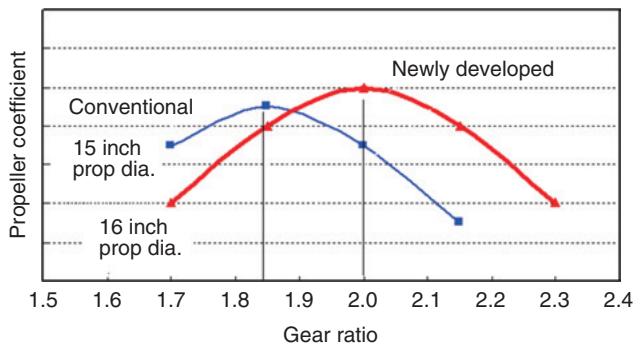


Fig. 6 Gear ratio and propeller coefficient

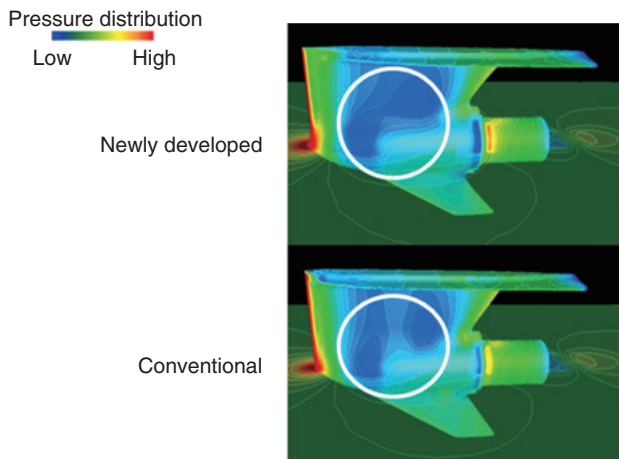


Fig. 7 Pressure distribution

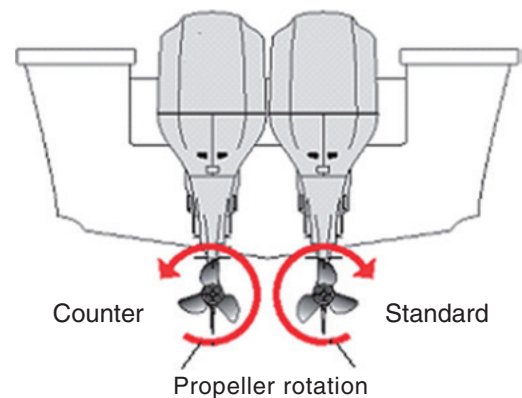


Fig. 8 Twin outboard motors

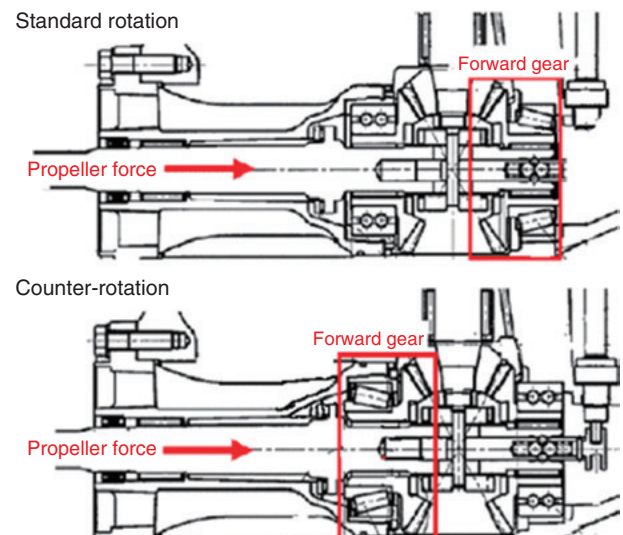


Fig. 9 Structure of gear case

the gears by a single set of bearings, different bearings are employed to receive each load. In this configuration, the radial force is received by needle bearings, and thrust force is received by thrust bearings. This has helped to realize constant stability in gear tooth contact, and has increased durability. This configuration is shown in Fig. 10.

4.3. Gear Materials and Heat Treatment

A setting was established for the hardness of the gears in order to help ensure pitting resistance. Because the tooth shape is adjusted by grinding in the developed gear case, the degree of grinding was taken into consideration, and it was judged that a hardness distribution that maintained a high level of hardness to a depth of around 0.3 mm from the surfaces of the teeth in comparison to the conventional specifications would be necessary to help ensure resistance to both pitting originating at the surfaces and pitting originating internally. New heat treatment conditions were set in order to realize this goal, and the use of a high-alloy steel (containing, among other components, 2% Ni and 0.75% Mo) made it possible to achieve the necessary toughness for the gears while realizing the target hardness distribution (Fig. 11).

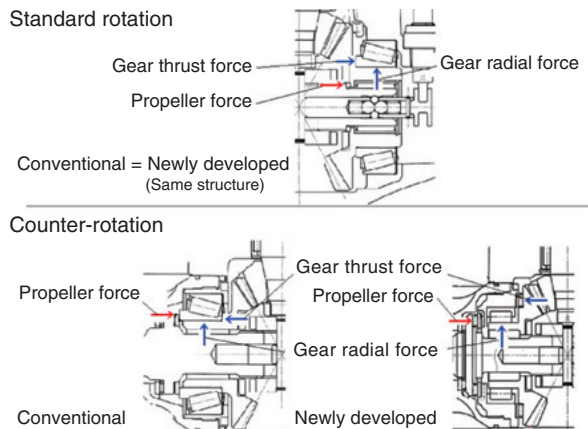


Fig. 10 Detail of gear case

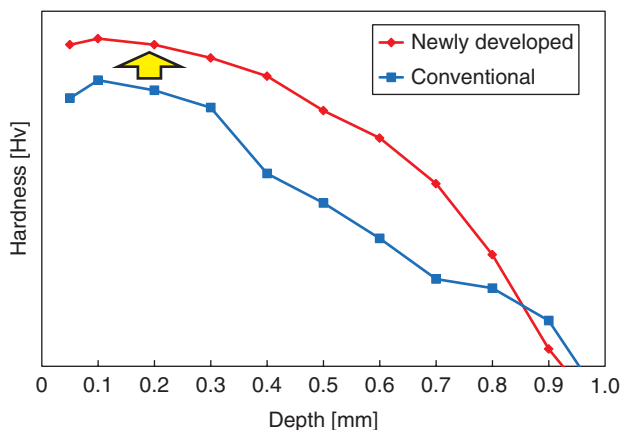


Fig. 11 Hardness distribution

Figure 12 shows the results of roller pitting fatigue tests conducted to verify the effectiveness of these measures. The production of the test specimens modeled actual gear manufacturing methods as much as possible, including grinding of the teeth in order to remove their surfaces.

The developed specifications achieved an increase in pitting strength of approximately 10% against the conventional specifications.

4.4. Gear Specifications and Tooth Shape

One of the characteristics demanded from the gears employed in a large outboard motor is sufficient durability for operation under high loads for extended periods. The development discussed here therefore optimized the gear module, using FEM analyses in order to achieve a balance between tooth face pressure, tooth root stress, and stiffness. Figure 13 shows the difference in the shapes of the conventional and the developed gears.

Because the gears in large outboard motors are used under high loads, the gear case undergoes considerable distortion, and the level of displacement of tooth contact from the clearance of the bearings is high. Measurements of gear displacement and FEM analyses were therefore conducted in order to correct the shape of the teeth and realize appropriate tooth contact. In addition, grinding is employed to reduce variations in tooth shape during production to the greatest possible extent, thus realizing reproducibility of tooth contact. Figure 14 shows an example of the FEM analyses that were conducted.

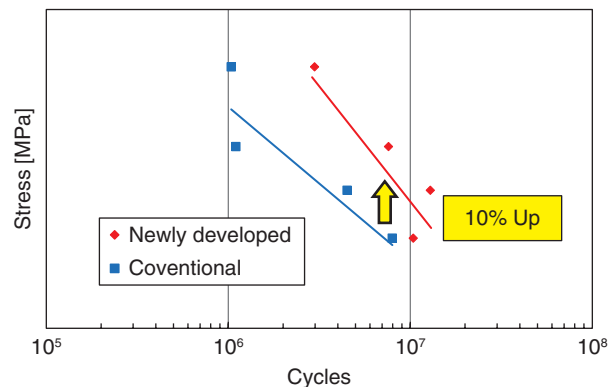


Fig. 12 Results of fatigue test for pitting

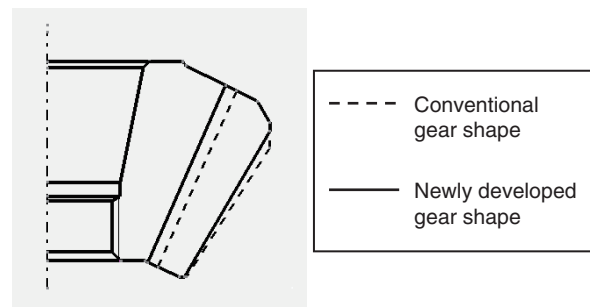


Fig. 13 Comparison of pinion gear shape

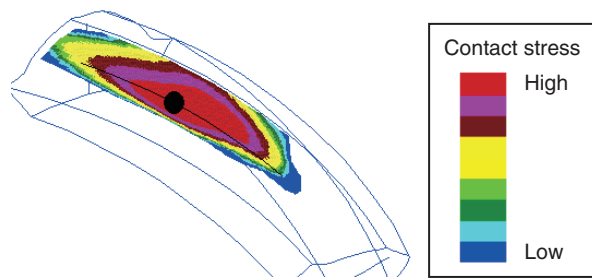


Fig. 14 Example of FEM

5. Durability Performance and Power Performance

5.1. Durability Performance

In component durability tests, the newly developed gear case displayed three times the durability time of a gear case using a conventional design, exceeding the target of twice the durability time (Fig. 15).

5.2. Power Performance

Optimization of the gear ratio and reduction of the water resistance of the gear case, while realizing increased durability, made it possible to achieve class-topping power performance, balancing maximum speed performance and acceleration performance (Fig. 16).

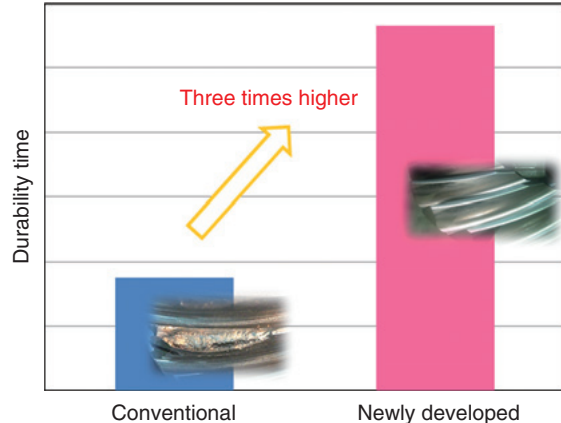


Fig.15 Durability

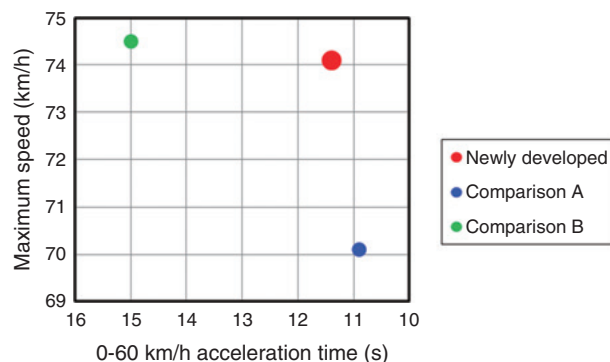


Fig. 16 Maximum speed and 0-60 km/h acceleration time

6. Conclusion

The development of a gear case for a large outboard motor resulted in the achievement of

- Class-topping power performance
- Three times greater durability as demonstrated in component durability tests

These results were achieved as follows:

- (1) Optimal specifications were selected based on the relationship between reduction ratio and propulsion efficiency, and water resistance was reduced.
- (2) The bearing structure for the counter-rotation specification was redesigned, helping to ensure stable tooth contact irrespective of operating conditions.
- (3) Gear materials and the heat treatment applied were reconsidered, making it possible to realize a high level of hardness on the outermost surfaces and the surface layers of the gear teeth, increasing pitting resistance by approximately 10%.
- (4) The gear module was optimized with consideration of durability under high-load operation. In addition, FEM analyses considering stiffness and manufacturing variations were conducted in order to optimize tooth contact, and grinding has been applied to finish the gear teeth.

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Development of New Seat Belt E-pretensioner System

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ABSTRACT

The previous seat belt E-pretensioner that was developed possessed a function for retracting the seat belt with a motor in a pre-crash area when the Collision Mitigation Braking System or brake assistant was operated. However, that operation scenario was limited.

Development efforts to address this issue were conducted to expand the range of operation of the pre-crash area and enhance the comfort of routine use of the seat belt. As for the pre-crash area, according to an analysis of serious injury accidents in which seat belts were worn, the system operation range was increased from 18% to 45%. Regarding the comfort area, by halving the occupant movement during sports driving and the pull-out load on the seat belt, and by reducing the retraction load by 30% when the seat belt is worn, the pull on the seat belt was lightened and the feeling of pressure when wearing the seat belt was reduced, improving comfort.

1. Introduction

Normal seat belts are provided with a locking function called Emergency Locking Retractor (ELR). This lock operates when accelerating or decelerating at approximately 3.9 m/s^2 or when there is a sudden pull on the seat belt. Because there is slack in the seat belt due to the clothing of the occupant, there is a possibility that the occupant will move forward until restrained by the seat belt. If the position or posture of the occupant is not suitable during a collision, the passenger might not be sufficiently restrained by the seat belt. For this reason, seat belt retractors have an emergency retraction function called a pretensioner. If a front collision is detected at a certain deceleration value or greater, the force of a gunpowder explosion causes the seat belt to retract in a short period of time in order to control how much the occupant moves during the collision.

However, because this pretensioner operates after detecting a collision, it is not effective against changes in the posture of the occupant prior to the collision. According to the North American National Automotive Sampling System Crashworthiness Data System (NASS/CDS)⁽¹⁾, approximately 20% of vehicles perform a pre-crash avoidance operation, such as emergency braking or sudden turning of the steering wheel. Due to this avoidance operation, occupants receive an inertial force with a maximum acceleration of approximately 10 m/s^2 . Unless occupants maintain their posture through their own strength, the inertial force can cause a change in posture.

The E-pretensioner reduces these changes in posture prior to a collision, and if there is a collision, the E-pretensioner has a function that quickly retracts the seat belt using an electric motor.

The conventional E-pretensioner has a Collision Mitigation Braking System (CMBS)⁽²⁾ that determines whether there is a risk of a collision and retracts the seat belt if the emergency automatic brakes are activated or if the brake pedal is pushed strongly, activating the electronic braking assistant.

The purpose of development was to further enhance the functions of the conventional E-pretensioner by expanding the pre-crash function, lessening the degree to which the occupant sways during sports driving, decreasing the pull-out load and retraction load on the seat belt, and reducing the feeling of pressure when wearing the seat belt, all of which had been problematic in the conventional system.

2. System Configuration

Figure 1 shows the vehicle layout of the E-pretensioner system and a system configuration diagram. The primary components added to the regular seat belt are the seat belt retractor drive unit and the motor control ECU. In addition, the system is configured such that CMBS and Vehicle Stability Assist (VSA) ECU, etc. are connected to the Controller Area Network (CAN). Also, in contrast to the conventional E-pretensioner, a rotation angle sensor that detects seat belt displacement and speed has been

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added to the ELR in the new E-pretensioner. The rotation angle sensor is provided in both the driver seat and front passenger seat.

2.1. Retractor Drive Unit

The E-pretensioner retractor is a normal seat belt retractor to which a motor, deceleration gear, and clutch mechanism have been added. When the system is not operating, the motor and deceleration gear are separated from the spool. If electric current for motor rotation is turned on in the forward direction, the clutch mechanism activates, the motor and spool are joined via the deceleration gear and retract the belt. If the electric current for motor rotation is turned on in the reverse direction, the clutch is released, and the motor and spool are separated.

The gear material and shape of the gear structure have been revised in order to reduce the gear-sliding sound during comfort operation.

2.2. Rotation Angle Sensor

When spool rotation is detected, the displacement and speed of the seat belt are acquired (Fig. 2).

2.3. Motor Control

There are three methods of motor control, and the optimal control that is utilized depends on the purpose of the operation.

(a) Motor current control

The motor current is controlled by Proportion Integral Differential (PID) feedback so that the current approaches a target value. When the motor is stalled, torque is proportional to current, so belt tension can be controlled.

(b) Motor voltage control

When the clutch is released, there is no load applied to the motor, so the motor is in its no-load state, motor drive current does not flow, and current feedback control is not possible. Therefore, motor voltage is controlled at a fixed voltage.

(c) Constant speed control

Using a rotation angle sensor, duty is controlled by feedback so that the output from the rotation angle sensor within a fixed period of time approaches a target value.

2.4. Electronic Control Unit (ECU)

An ECU that controls motors in the new E-pretensioner was developed, with one ECU controlling one motor. Using an Application Specific Integrated Circuit (ASIC) in which a calculation processing circuit, a drive circuit, and a CAN interface are integrated onto one IC, the newly developed ECU has been miniaturized and is lighter in weight in comparison to the conventional ECU (Fig. 3).

The ECU communicates with the ECUs of affiliated systems, such as the CMBS, VSA, and SRS, via the CAN. The ECU receives signals required for pretensioner motor control. Such signals include CMBS and VSA operating state, the yaw rate, longitudinal acceleration, lateral acceleration and the seat belt-wearing state, all of which trigger pre-crash operation. The ECU is also provided with a receiving unit for rotation angle sensor signals and buckle

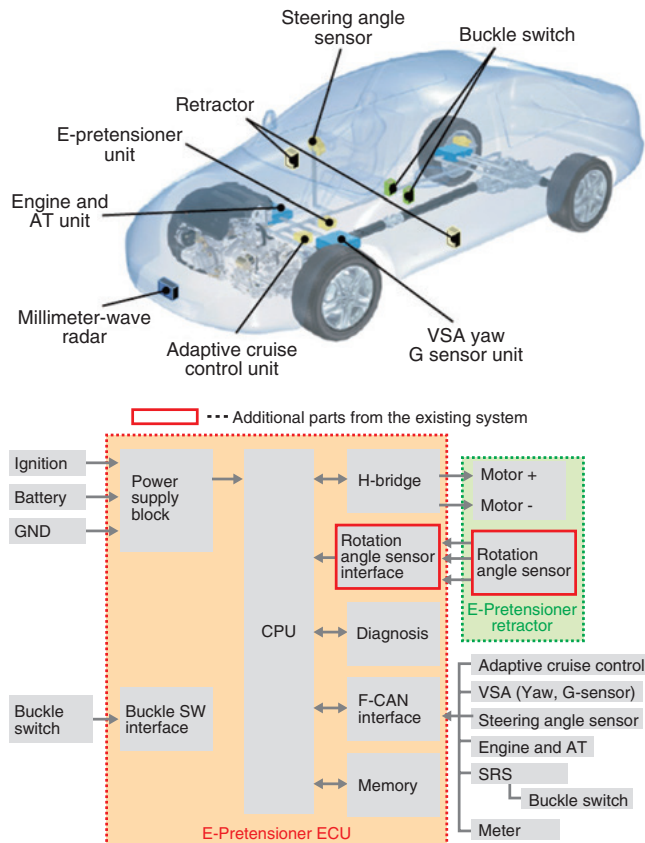


Fig. 1 Vehicle layout and system configuration diagram

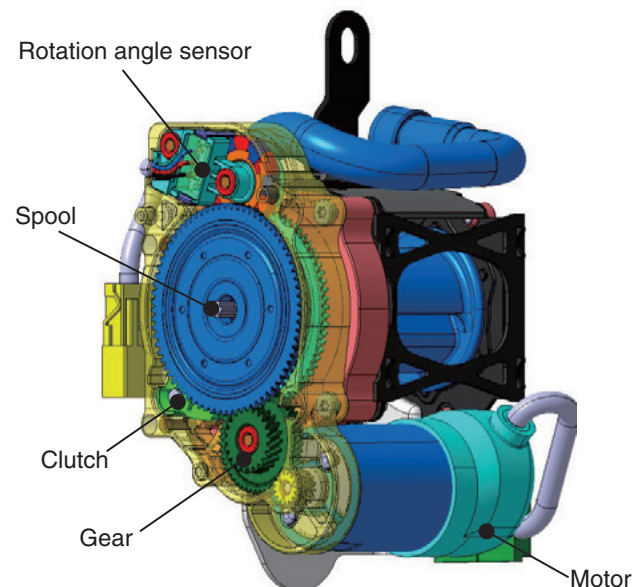


Fig. 2 E-pretensioner retractor

switch signals. These signals are processed by the CPU and they control the motor through Pulse Width Modulation (PWM).

2.5. Buckle Switch

A seat belt-wearing state signal is required for the motor to retract the seat belt when the ignition is OFF. The buckle switch has been added because the seat belt-wearing signal cannot be received from the CAN when the ignition is OFF.

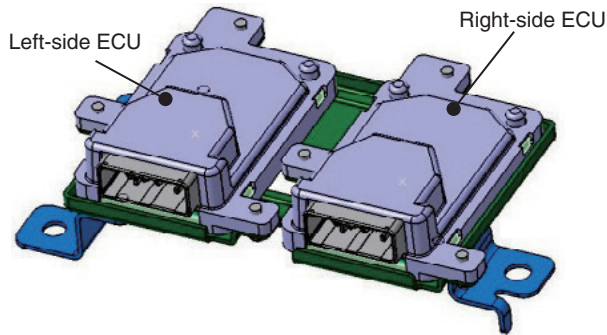


Fig. 3 E-pretensioner ECU

3. Explanation of the New E-pretensioner Functions

The new E-pretensioner functions can be classified into functions during pre-crash emergency retraction, which occurs during emergencies, and comfort functions that increase the comfort of routine use of the seat belt. Because the conventional E-pretensioner only operated in a pre-crash area during an emergency, the operation scenario was limited. However, because a comfort function has been added to it, the new E-pretensioner functions even during routine use and enhances the comfort of the seat belt.

The details of the functions are described below.

3.1. Emergency Retraction Function during Pre-crash

This is a function for retracting the seat belt during an emergency to control the changes in posture of the occupant prior to the collision when the risk of a collision is judged to be higher than that of the normal driving state. The seat belt is retracted by electric current control. Retraction in the shortest amount of time, approximately 100 msec, is possible at a shoulder belt tension force of 100 N. When the inertial force that acts on an occupant is low, it would feel very uncomfortable to the occupant to have the seat belt retract at a high tension force. Therefore, the target value of motor current is set according to the situation, and retraction occurs in the following cases.

- During lateral slip or sudden turns: Retraction occurs when the VSA lateral slip control Active Yaw Control (AYC) function operates and when the steering wheel is turned abruptly (Fig. 4).
- During shock input: When the SRS sensor detects a

collision, seat belt retraction occurs to prepare for a possible secondary collision (Fig. 5).

3.2. Comfort Function

By decreasing swaying of the occupant during sports driving, lightening the pull of the seat belt during routine use, and reducing the seat belt retraction load, the feeling of pressure when wearing the seat belt was mitigated. Also, when the seat belt is worn, there is a function that automatically retracts slack in the seat belt and a function that assists the seat belt storage operation.

The following operation modes exist:

(a) Dynamic mode

When vehicle speed has reached or exceeded a set value and when braking deceleration and turning acceleration have reached or exceeded their set values, the seat belt tension is increased and the swaying of the upper body of the occupant is decreased (Fig. 6).

(b) Welcome mode

When the seat belt is worn, after the seat belt tongue is inserted into the buckle, the seat belt is retracted at a constant speed and the slack is retracted.

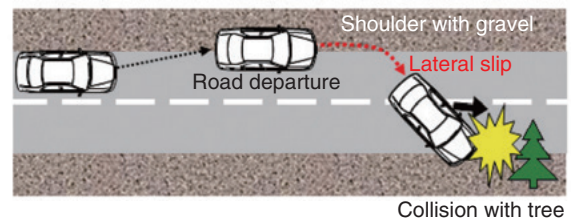


Fig. 4 Example of collision with lateral slip

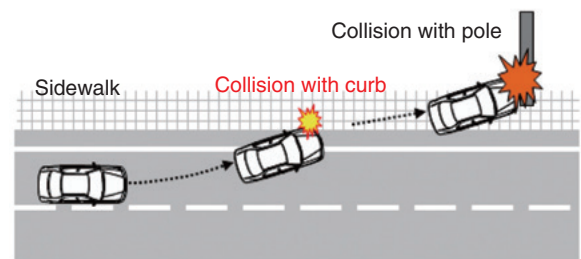


Fig. 5 Example of multiple collisions

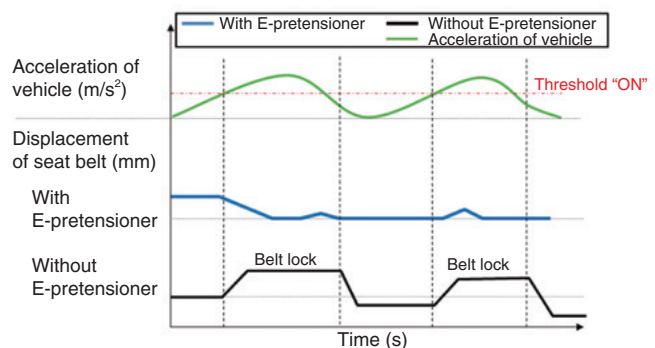


Fig. 6 Dynamic control

(c) Seat belt storage function

When the seat belt is unbuckled or after the seat belt is withdrawn from its fully retracted state, the seat belt is retracted by constant speed control in order to store the seat belt. In order to prevent seat belt retraction from feeling uncomfortable, retraction speed is set according to the quantity of seat belt that is to be retracted. Up to 150 mm from the seat belt-wearing position, the seat belt is retracted at a relatively rapid speed. At 150 mm and greater, the seat belt is set to retract slowly so that it is difficult for the tongue of the seat belt to become caught on the occupant's arm or clothing (Fig. 7).

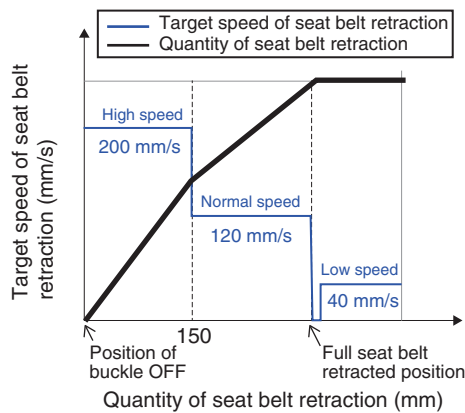


Fig. 7 Speed control in retraction function

4. Application Effect

The effects of the new E-pretensioner functions are shown below.

4.1. Pre-crash Area

By adding a pre-crash function, this newly developed E-pretensioner is estimated to be capable of increasing the range in which the system operates from 18% to 45%, according to an analysis of serious injury accidents occurring when seat belts were worn that are recorded in the NASS/CDS database (Fig. 8).

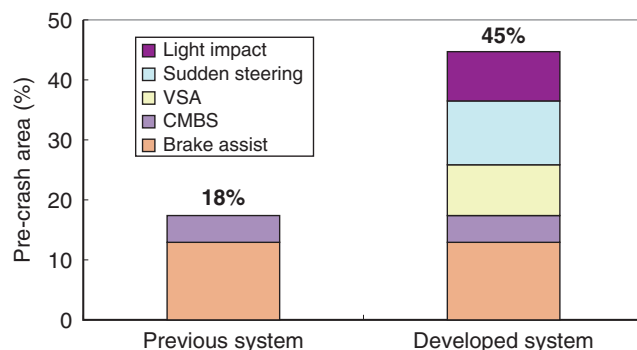


Fig. 8 Comparison of pre-crash area

4.2. Comfort Area

By adding the Dynamic Mode, seat belt displacement during sports driving on a winding test course was able to be reduced by a maximum of 42 mm with respect to the normal amount of seat belt extension, and the amount of movement of the upper body of the occupant could be controlled to a maximum of 100 mm (Fig. 9).

Also, when the seat belt is stored, retraction by a spring is assisted with a motor, making it possible to halve the force of the spring, halve the seat belt pull-out load, and reduce the retraction load when the seat belt is worn by 30%. This lightens the pull on the seat belt, and reduces the feeling of pressure when wearing the seat belt, thereby increasing seat belt comfort (Fig. 10).

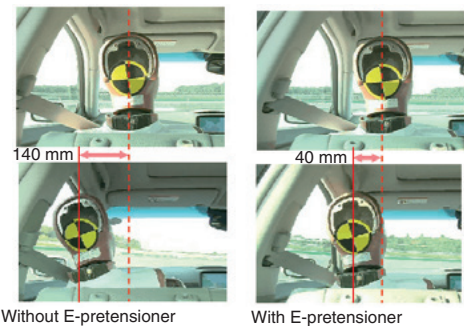
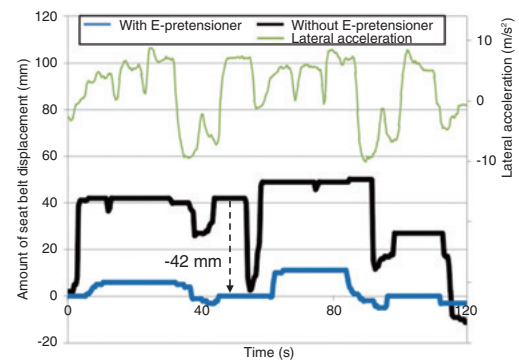


Fig. 9 Photographs of occupant posture in turn test

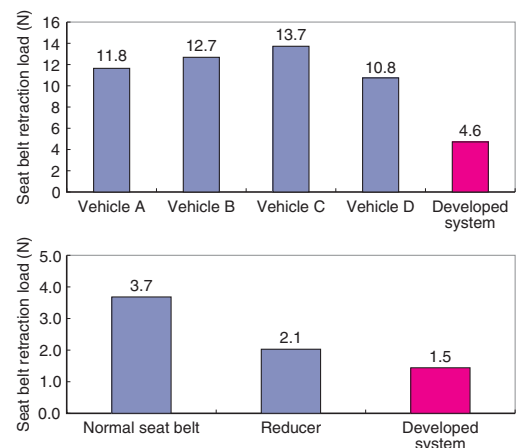


Fig. 10 Seat belt pull-out load and seat belt retraction load

5. Conclusion

A new E-pretensioner function was developed and its effects were confirmed. Those effects are as follows.

- (1) In addition to the conventional CMBS and brake assistant functions, in affiliation with the VSA and SRS systems, the system operation range in the pre-crash area was increased from 18% to 45% compared to that of the conventional system.
- (2) By adding a comfort function that operates during normal use of the seat belt, the occupant movement amount during winding and sports driving was halved.
- (3) By halving the seat belt pull-out load and reducing the retraction load by 30%, the seat belt pull was lightened and the feeling of pressure when the seat belt is worn was reduced, thereby increasing seat belt comfort.

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■ Author ■



Shotaro ODATE



Naotoshi TAKEMURA

Bimetallic Corrosion Prevention in Hybrid Doors by Means of Watertight Structure using Low-elasticity Sealer

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ABSTRACT

Seeking to realize weight savings in vehicle bodies, Honda has developed a lightweight and high cost-efficiency aluminum/steel-sheet hybrid door. Bimetallic corrosion in the area of connection between the two metals represented an issue for this door. The difference in electric potential between the two metals would promote corrosion, raising the possibility of a decline in commercial appeal. In order to address this issue, a sealer was employed to help prevent infiltration of water. In addition to providing watertightness in the use environment, it was necessary for this sealer to possess a low elastic modulus in order to help prevent distortion from being generated in the skin panel during the sealer curing process. A watertight structure was realized through the development of application specifications using a low-elasticity sealer.

1. Introduction

Concern regarding global warming originating in greenhouse gas emissions today demands the reduction of CO₂ emissions on a global scale. For the automotive industry, the major issue is the realization of increased fuel efficiency, and a wide variety of means of reducing fuel consumption are being researched and put into practical application, including the achievement of increased engine efficiency, engine electrification, and the reduction of rolling resistance. The achievement of weight savings in the vehicle body is an important element in the realization of increased fuel efficiency, prompting automakers to replace certain parts of their vehicle bodies with low-density materials. Honda has led its competitors in advancing the realization of weight savings in automotive bodies, for example in its early mass production of all-aluminum bodies. The company is also working to develop low-cost weight-saving technologies to help enable the presentation of higher-quality products to its customers at lower prices. In the area of vehicle doors, Honda is already mass producing lightweight all-aluminum doors, but has also proceeded with the development of a hybrid door that uses aluminum only for the door skin in order to realize a lightweight door with a high cost-to-weight-reduction ratio (Fig. 1).

Normally, when two different metal materials such as aluminum and steel sheets come into contact, bimetallic corrosion becomes an issue. In an aqueous solution, the metal with an electric potential further in the base direction becomes an anode, and the speed of corrosion increases in comparison to a situation in which the metal is used in

isolation⁽¹⁾. Attempting to prevent decline in commercial appeal due to this phenomenon in the lower edge of the door, where rainwater and other fluids tend to accumulate, was also an issue in the development of the hybrid door. In seeking to prevent bimetallic corrosion, attempting to prevent contact with water is the important first consideration, and the realization of a watertight structure in the area of the joint between the aluminum and the steel sheets was therefore examined.

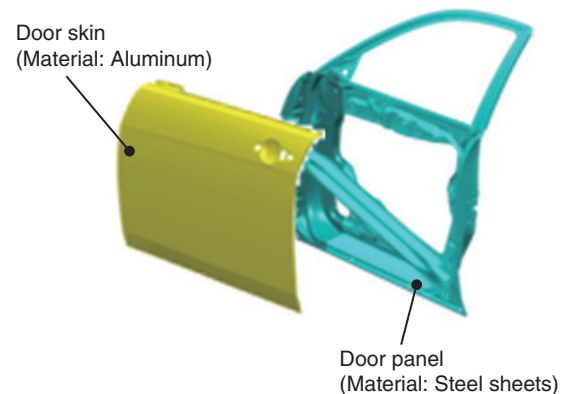


Fig. 1 Structure of hybrid door

2. Issues and Methods of Realizing Goals

Figure 2 shows the hemming structure that joins the door skin and the door panel at the lower edge of the door and the path of infiltration of water. When water infiltrates into the interior of the door, bimetallic corrosion occurs, and

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corrosion products result in the deterioration of the joint, swelling of the surface of the exterior sheet, and perforation corrosion, as shown in Fig. 3. As a countermeasure, it was therefore decided to coat the mating faces of the door skin and the door panel with a sealer to help prevent the infiltration of water. However, filling the space between the mating faces with a conventional sealer would result in distortion occurring in the door skin during production. A sealer material and an application specification that would not produce distortion were therefore examined.

2.1. Reduction of Distortion

Sealers are hardened in a baking process. In the case of the hybrid door, the difference in the coefficients of linear expansion of the aluminum and the steel sheets would result in a displacement of a maximum of 1.2 mm between the mating faces when the materials were heated. When the sealer hardens in this state, a shear force is generated in the sealer when the door skin contracts as the door is cooled. Distortion occurs because the door skin is maintained in a position in which the return force of the door skin and the shear force of the sealer are in equilibrium. Because conventional sealers have high elastic moduli and the shear force is strong, equilibrium is reached with significant displacement remaining in the door skin, as shown in Fig. 4, and this results in distortion exceeding benchmark values (Fig. 5). In order to reduce the distortion in the door skin,

it would be necessary to reduce the residual displacement of the mating faces to 0.4 mm or less. It would therefore be necessary for the sealer to be capable of responding to displacements of 0.8 mm or more. Given that the thickness of the sealer at the mating faces is only around 0.1 mm, the material would have to display a rate of expansion of 800% or more, an unrealistic figure. It was therefore decided to induce breakage of the sealer in order to reduce distortion. Equation (1) shows the breaking displacement of the sealer, and Eq. (2) shows the necessary conditions for breaking of the sealer.

$$x_B = \sqrt{\{(\epsilon_B + 1)^2 - 1\} t^2} \quad (1)$$

$$F_B = E \frac{\sqrt{t^2 + x_B^2} - t}{t} S < 76.05 \times (1.2 - x_B)^{0.56} \quad (2)$$

x_B : Sealer breaking displacement

ϵ_B : Sealer breaking distortion

t : Sealer thickness

F_B : Minimum shear force per unit length at which sealer breaks

E : Sealer elastic modulus

S : Area of sealer

A sealer with a low modulus of elasticity satisfying Eqs. (1) and (2) was selected (termed “low-elasticity sealer” below). Table 1 shows the physical properties of the sealer.

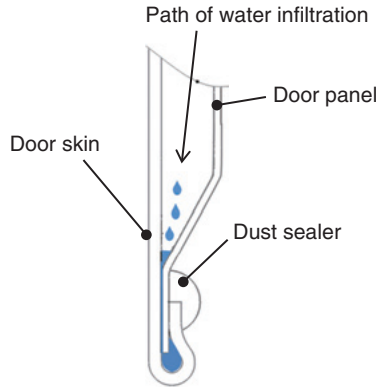


Fig. 2 Path of water infiltration into hemming structure

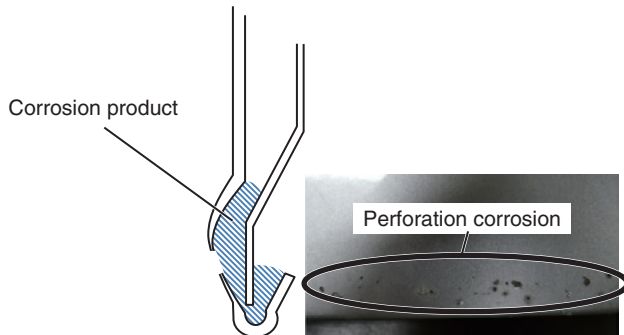


Fig. 3 Example of galvanic corrosion

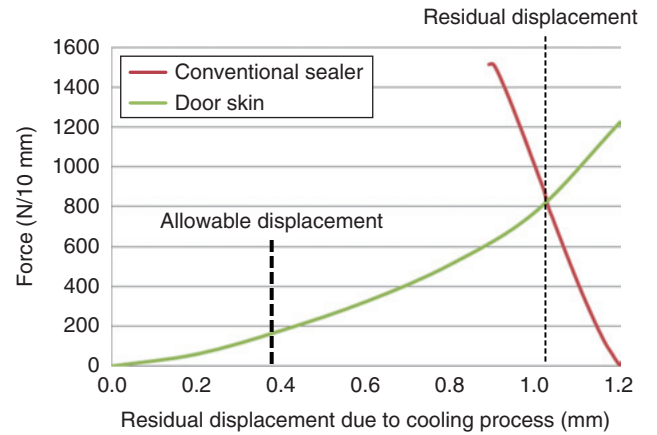


Fig. 4 Returning force of door skin and shear force of sealer

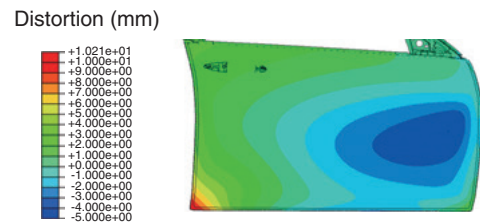


Fig. 5 Distortion in door skin with conventional sealer

Table 1 Mechanical properties of low-elasticity sealer

Young's modulus	Maximum elongation	Guaranteed elongation after degradation
0.22 MPa	200%	50%

2.2. Measures to Realize Watertightness

The sections of the low-elasticity sealer filling the space between the mating faces would be broken in order to reduce distortion. A projecting application structure was therefore considered in order to help ensure watertightness (Fig. 6). This projecting section would respond to displacement without breaking. As displacement occurring in the mating faces, it would be necessary to consider not merely displacement occurring during the baking process, but also displacement occurring during the use of the vehicle.

Both static displacement due to the difference in coefficient of linear expansion between the two materials and dynamic displacement due to the opening and closing of the door occur during use, and composite displacement resulting from these two types of displacement occurs in the mating faces. Because the degree of expansion of the aluminum door skin would be greater than that of the steel-sheet door panel in a high-temperature environment, displacement would occur in the sliding direction of the door panel away from the hemming structure of the door skin. At the maximum temperature in extremely hot regions, the maximum displacement would be 0.3 mm. In low-temperature regions, contraction would be greater in the door skin than the door panel, and displacement would occur in the sliding direction of the door panel into the hemming structure. At the minimum temperature in extremely cold regions, the maximum displacement would be 0.3 mm. With regard to dynamic displacement, the maximum displacement occurs when the door is closed hard, as shown in Fig. 7, and the maximum displacement under these conditions was found to be 0.6 mm in the sliding direction of the door panel away from the hemming structure. Therefore, the maximum displacement during use would be 0.9 mm in the sliding direction of the door panel away from the hemming structure, occurring when the door was closed hard in an extremely hot environment.

Based on the above, it was determined that the

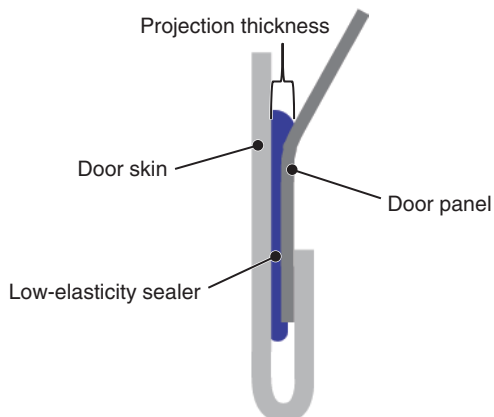


Fig. 6 Watertight structure

maximum displacement occurring in the joint section would be 1.2 mm in the sliding direction of the door panel into the hemming structure during manufacture.

As Fig. 8 and Eq. (3) show, the ability of the low-elasticity sealer to respond to the maximum displacement of 1.2 mm is dependent on the elongation characteristic of the sealer and the thickness of its application.

$$x = (\varepsilon^2 - 1)^{0.5} t_0 \quad (3)$$

t_0 : Sealer thickness

x : Sealer shear displacement

ε : Sealer rate of elongation

Because the guaranteed rate of elongation of the low-elasticity sealer was 50%, the thickness of the projecting section was set at 1.0 mm to help enable the material to respond to a maximum shear displacement of 1.2 mm.

This projecting section would also produce a shear force in the sealer of the type described in the previous section. Equation (4) shows the shear force F in the projection and the sealer at the mating faces.

$$F = E \frac{x - 0.1}{0.1} S + \int_0^y E \frac{x + \sqrt{r^2 - y^2} + r}{-\sqrt{r^2 - y^2} + r} y w dy \quad (4)$$

r : Radius of bend in door panel

y : Height of projection

w : Width of coating

Figure 9 shows the relationship between the shear characteristic of the projection during application and the return force of the door skin. The projecting section of the low-elasticity sealer is in a state of equilibrium with the return force of the door skin without breaking at a displacement of close to 0 mm. The adopted specifications made it possible to reconcile the reduction of distortion in the door skin with the achievement of watertightness.

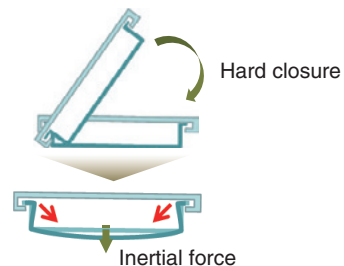


Fig. 7 Displacement by hard closure

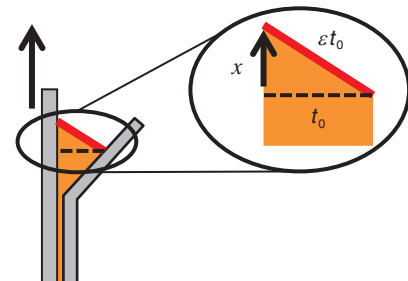


Fig. 8 Shear displacement and sealer elongation

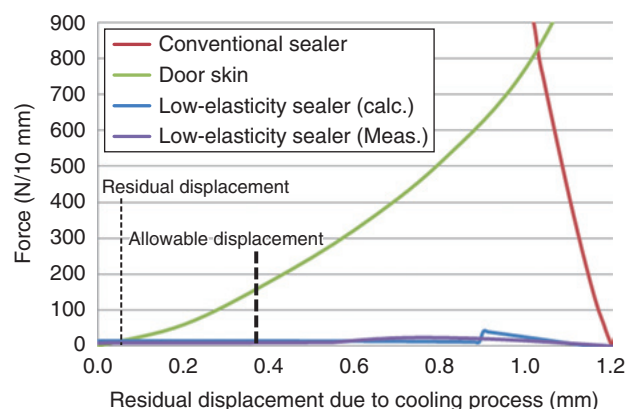


Fig. 9 Returning force of door skin and shear force of watertight structure

3. Verification Results

3.1. Verification of Door Skin Distortion with Low-elasticity Sealer

Figure 10 shows the results of measurements of distortion in the door skin for the connecting structure shown in Fig. 6, as used in an actual door. Distortion in the door skin has been reduced compared to results for a conventional sealer (Fig. 5).

3.2. Verification of Corrosion Resistance of Hybrid Door in Actual Vehicle

The corrosion resistance of a hybrid door using the low-elasticity sealer was studied in vehicle driving tests used to evaluate compound durability. Figure 11 shows an external view of the door following the test, and Fig. 12 shows a cross-section of the lower edge of the door following the test. Bimetallic corrosion has been controlled, and no reduction in commercial appeal is apparent.

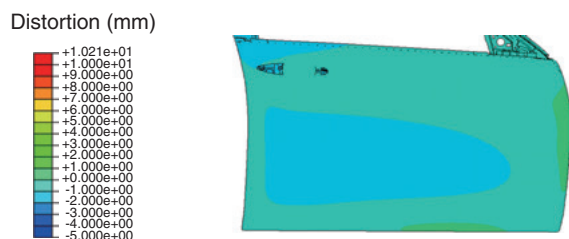


Fig. 10 Distortion in door skin with watertight structure

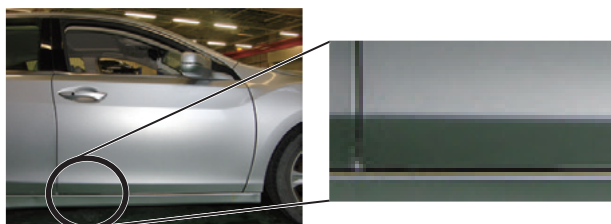


Fig. 11 Appearance after vehicle durability test

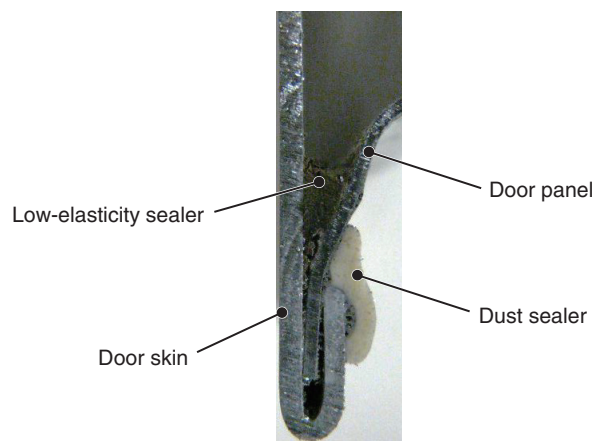


Fig. 12 Door lower edge section after vehicle durability test

4. Conclusion

Honda has developed an aluminum/steel-sheet hybrid door that uses aluminum rather than steel sheets for the door skin as a high-productivity lightweight door. To help prevent the bimetallic corrosion that would represent an issue in this project, a low-elasticity sealer that would help to control distortion in the skin panel was developed, and sealing specifications were set to assist in enabling the material to respond to displacement, resulting in a watertight sealer structure. Comprehensive vehicle durability tests verified that the developed structure provided sufficient corrosion resistance performance.

This aluminum/steel-sheet hybrid door is being employed in the 2014 new model ACURA RLX.

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Development of Steel and Aluminum Continuous Welding Technology Using FSW and Application to Subframes

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ABSTRACT

Continuous welding technology that uses Friction Stir Welding and has a welding strength equal to or above that of MIG welding was developed to weld together steel parts and aluminum parts, and at the same time technology was also developed to rustproof areas where dissimilar metals are welded. In order to apply this welding technology to mass production, an original welding system using an industrial articulated robot was developed, and application was realized with an efficiency and investment equivalent to that of conventional MIG welding. In addition, an inspection system using a highly sensitive infrared camera and laser beam was also developed as a non-destructive in-line system to assure joint strength. This technology was applied to the front subframe, which is a structural component of mass-production vehicles. As a result, compared to the previous model, the weight was reduced by 25%, the rigidity of the suspension mounting point was increased by 20%, the vehicle's dynamic performance was enhanced, and electrical power consumption during the welding process was reduced by 50%.

1. Introduction

Recently, needs are increasing to reduce vehicle weights and to conserve energy in the production stage from the viewpoints of reducing carbon emissions, conserving energy, and enhancing fuel economy performance and dynamic performance in order to protect the global environment. The function required of the front subframe is to possess sufficient support rigidity to support the suspension, power plant, and steering gear box. In addition, the subframe must also contribute to steering stability performance, vibration and noise performance, and collision performance. For these reasons, automotive manufacturers are actively researching and developing welding, structural, material and other technologies to efficiently achieve a good balance of the required performances with reduced weight and low cost⁽¹⁾. One method of reducing weight is to mechanically join aluminum members to aluminum subframes and steel subframes. Conventional subframes have mostly used MIG welding, so there was a limit to how much energy consumption could be reduced in the manufacturing stage. Furthermore, mechanical joining

methods are point connections, so weight reduction and increases in rigidity are also limited, which makes it a challenge to greatly enhance dynamic performance.

This development focused on Friction Stir Welding (hereafter "FSW"), which consumes less electrical power and has a lower running cost in the manufacturing process compared to MIG welding, and developed dissimilar-metal welding technology to take the place of conventional MIG welding and mechanical joining. Generally speaking, sufficient joint strength cannot be obtained when welding together steel and aluminum, so these dissimilar metals could only be joined mechanically or bonded using adhesive. In addition, dissimilar-metal welding technology using FSW is being actively researched^{(2), (3)}, but there are no known cases of practical application, and insulating technology for rustproofing also has not yet been established. Furthermore, the known cases of FSW use large welding equipment, which requires a large installation area and high infrastructure investment. Therefore, a space-saving welding system was developed that can be applied to existing plants and production lines with low infrastructure investment, and which also has a production tact equivalent

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to that of MIG welding in consideration of application to mass production of 1500 units or more per day. In addition, ultrasound tests and other techniques are known as non-destructive methods of observing the inside of FSW joints, but in subframes the joints are located inside the various members, so they cannot be easily checked. This made it necessary to develop non-destructive inspection technology that can accurately inspect the inside of joints in a short time, so an original inspection system was constructed.

2. Dissimilar-Metal Welding Technology Development Goals and Parts Development Target Values

The goals of this technical development were the following four items.

- (1) Establishment of technology for welding together the dissimilar metals of steel and aluminum
- (2) Establishment of rustproofing technology
- (3) Establishment of a mass-production welding system using industrial robots
- (4) Establishment of non-destructive inspection technology and an inspection system

Next, the following target values were set to realize mass production of hybrid subframes comprising both steel and aluminum members.

- (1) Weight: reduction of 10% or more compared to the previous model
- (2) Number of component parts: reduction of 30% or more compared to the previous model
- (3) Rigidity of the suspension mounting point: increase of 10% or more compared to the previous model
- (4) Electrical power consumption during the welding process: reduction of 30% or more compared to the previous model
- (5) Cost: equivalent to that of the previous model

3. Steel and Aluminum Welding Method and Welding Mechanism

FSW, which enables bonding with low heat input, was selected as the method of welding the dissimilar metals of steel and aluminum. As shown in Fig. 1, FSW is solid-

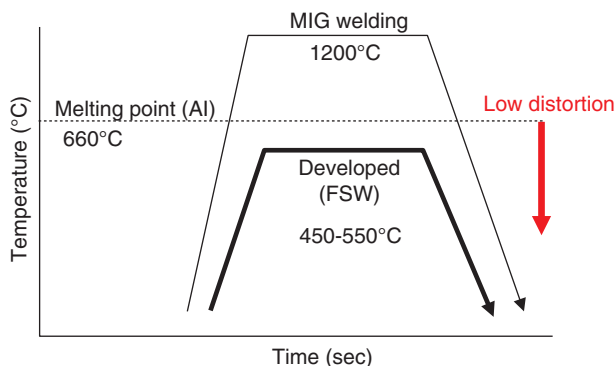


Fig. 1 Welding temperature of FSW

state welding at a temperature below the melting point, which enables suppression of the formation of coarse and brittle intermetallic compounds (hereafter “IMC”). The joint is formed as shown in Fig. 2 by applying pressure from the aluminum side to insert the tool while rotating the tool. As a result, the aluminum side is stirred, the tip of the probe destroys and removes impurities such as the galvanization, cathodic electrodeposition, and sealer on the steel surface to reveal the bare surface of the steel, and IMC are formed to bond the metals. The IMC composition was confirmed by the electron diffraction pattern to be $\text{Fe}_4\text{Al}_{13}$. In addition, it was also confirmed that the IMC thickness is suppressed to approximately $1\ \mu\text{m}$. Figure 3 shows the confirmation results using a Scanning Electron Microscope (SEM) and a Transmission Electron Microscope (TEM). In order to set the appropriate FSW joining conditions, it is important to stabilize the plastic flow and control the probe insertion position, and consideration must also be given to structural advantages due to differences in the shape of the hook created by the plastic flow in the FSW stirring area. Regarding the joining condition settings, the

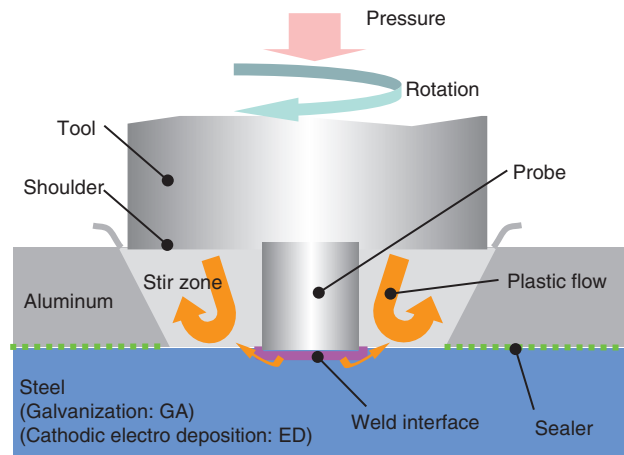


Fig. 2 Cross-sectional structure of Steel and Aluminum FSW

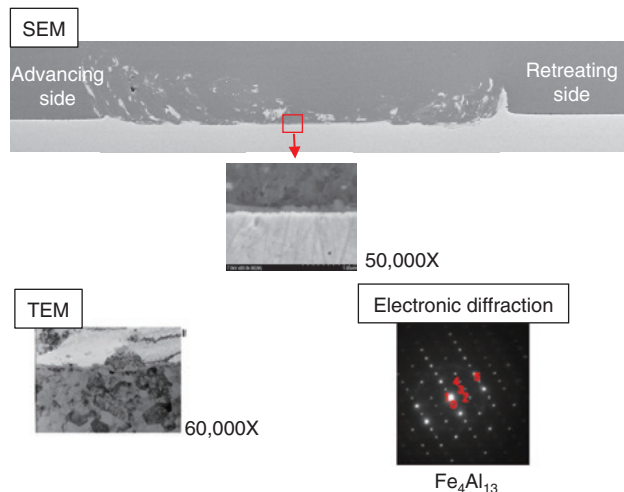


Fig. 3 Observation of IMC

shaded area in Fig. 4 was found to be the appropriate range based on the relationship between the rotations and the feed speed. Insertion position control is particularly important when welding dissimilar metals. When the probe insertion position is too shallow, impurities are not adequately discharged from the boundary, and a stable joint boundary cannot be obtained. Conversely, when the probe insertion position is too deep, even if impurities can be adequately discharged from the boundary surface, the contact between the steel and the probe is too strong, resulting in an excessively large hook or other factors that may reduce the strength. In addition, this also leads to a drop in probe durability, so it was necessary to set an appropriate range.

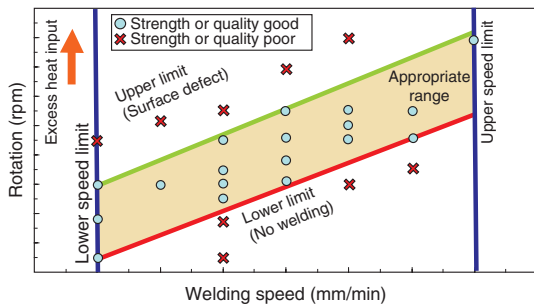


Fig. 4 Range appropriate FSW conditions

4. Rustproofing Technology

Figure 5 shows a schematic cross-section of a welding part. There is no cathodic electrodeposition around the joint between the dissimilar metals or on the steel surface in the edge area, and the galvanization and aluminum are in direct contact with each other. A structure that fills the gap between the steel and aluminum with sealer was used to prevent water or other substances from entering and corroding this type of welded joint between dissimilar metals. Table 1 lists the functions and physical properties required of the sealer. From these viewpoints, a single fluid denatured silicon-based sealer material was selected. Next,

Table 1 Required functions and properties for sealer

	Required function	Required property
(1)	Shut off from corrosion factors (water and corrosion current)	Water tightness, adhesion, and insulation
(2)	No failure to depend on frictional heat	Thermal stability
(3)	No failure to depend on market degradation	Durability over environment and vibration
(4)	Minimum investment in equipment and minimum cost for production control	One part agent providing cold curing

Fig. 6 shows the welding procedure. The sealer is applied before welding in order to completely fill the gap, and the coating diameter and position are also optimized. The joint strength with and without sealer was compared, and the sealer was confirmed to have no effect on the strength. In addition, Fig. 7 shows cross-sectional photos of the area around the joint after a 120-cycle test in a Cyclic Corrosion

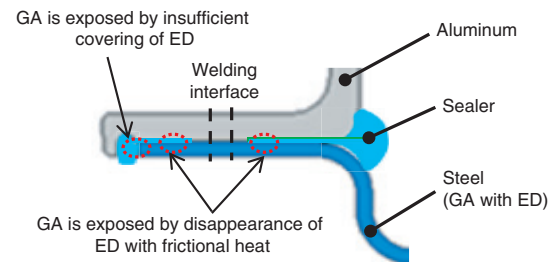


Fig. 5 Schematic cross section of welding part

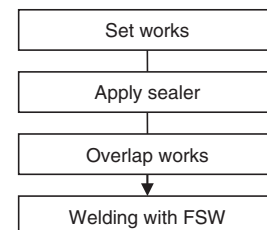


Fig. 6 Welding flow process

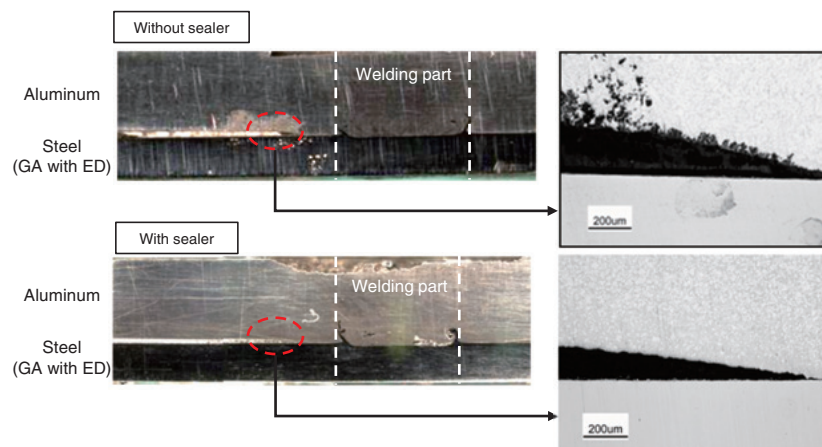


Fig. 7 Cross-sectional photos of welding test pieces with and without sealer after 120 cycles CCT

Tester (CCT). Corrosion with a maximum depth of 0.4 mm occurred on the aluminum side of the test piece without sealer, but corrosion did not occur on the test piece with sealer. In this manner, joint rustproofing effects can be obtained by using sealer. Furthermore, sacrifice corrosive protection effects maintain the rustproofing effects even if the cathodic electrodeposition on the back side of the joint peels off.

5. Robot FSW Technology

FSW continuous welding is generally performed using a large gate-type welding machine^{(4), (5)}. This is because a large load is applied to the tool during welding, so a highly rigid machine is necessary. Applying this type of large machine to mass production requires a large installation area (Fig. 8) and large investment, which makes introduction to existing plants and production lines difficult. Therefore, it was decided to develop a welding unit that requires a small installation area and suppresses investment by using an industrial articulated robot. There are cases of research on FSW continuous welding using articulated robots^{(6), (7)}, but to date it has been considered necessary to develop special

robots with a high rigidity able to withstand the large loads during welding. In particular, when performing FSW continuous welding on the subframe structure in this study, the load in the axial direction of the tool is 10 kN or more. However, instead of developing a special robot that can withstand this load, the FSW welding system shown in Fig. 9 was developed with the aim of not placing a high load on the robot. This welding system eliminates the robot load in the axial direction of the tool by mounting an FSW unit with a highly rigid C-shaped frame structure onto a robot. In this welding method, a steel part and an aluminum part are set in a supported welding jig, and the FSW welding unit welds together the parts by clamping the work together with the jig. In addition, the jig side of the C-shaped frame is a roller, which enabled sufficiently reducing the load generated in the weld advancing direction to within the allowable load range of the robot. Furthermore, the welding speed is also equivalent to that of general MIG welding, thereby achieving the production tact in conventional mass production.

The robot used performs 6-axis control, but the developed system incorporates two axes (welding tool rotation and insertion position) into the robot's own control system,

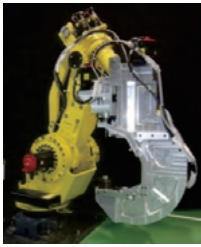
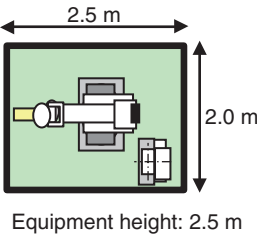
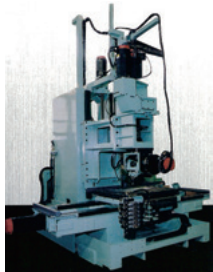
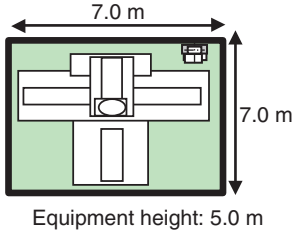
FSW unit of C-gun type robot		FSW unit of marketing	
Appearance	Ground contact size	Appearance	Ground contact size
			

Fig. 8 Comparison of FSW units

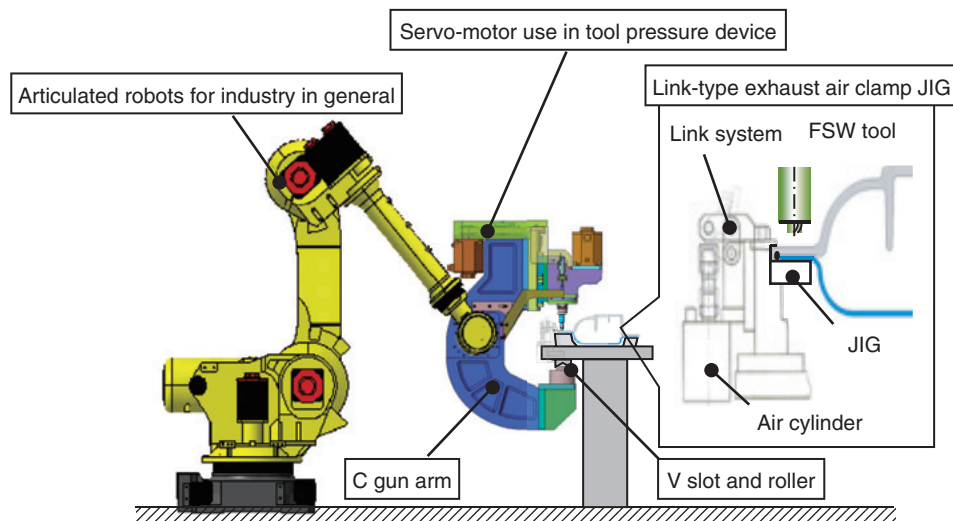


Fig. 9 FSW equipment appearance

and performs integrated control. This realizes a simple system that does not require a special control unit or other devices for welding. This system can be taught both online and offline in the same manner as a normal robot, and two- or three-dimensional welding programs can be easily created. In addition, the welding tool rotations and insertion position, the welding speed, and other parameters are controlled collectively, which enables constant monitoring of the welding status and real-time quality control.

The ability to use industrial robots enables production with stable quality in existing plants and production lines. In addition, this FSW welding system is highly versatile and can also be used to weld aluminum-to-aluminum structures.

6. Non-destructive Inspection Technology and Inspection System

It was decided to perform both condition control that includes monitoring of the welding status, and in-line non-destructive inspection that enables inspection of every unit as quality control for areas where dissimilar metals of steel and aluminum are welded. Existing non-destructive inspection technology is unable to accurately judge the quality of dissimilar-metal joints in a short time, so a new inspection system was developed by incorporating a laser excitation system onto the base of a thermal image analyzer using optical excitation. This system uses the principle of active thermography, and includes a heat source suited to the subject materials. Active thermography is a technology for visualizing the information in the lamination direction (plate thickness direction) when heat energy is applied to a material. When a material absorbs applied heat energy, locations (cavities, defects, boundaries, etc.) that differ from the base material (normal areas) radiate heat energy as self-energy. This phenomenon is measured using a high-performance infrared camera, and visualized by performing frequency analysis. Conventional excitation methods had the issue that when applied to materials with high thermal conductivity such as aluminum, the heat diffusion was greater than the heat transfer speed in the depth direction,

so accurate measurement was not possible. Therefore, a laser device with high energy irradiation and easy output adjustment (power density, time control) was incorporated to construct a system for welding the dissimilar metals of steel and aluminum parts (Fig. 10). In addition, the following features were taken into account for application.

6.1. Cross-sectional Structure of Joints

The cross-sectional structure of dissimilar-metal joints can be classified into three areas as follows (Fig. 11).

- (a) An area with high thermal conductivity where IMC are formed
- (b) An area where IMC are not formed, but the dissimilar metals are closely adhered and the thermal conductivity is lower than that in area (a)
- (c) An area with greatly reduced thermal conductivity due to the presence of sealer, cathodic electrodeposition, an air layer, and other factors

This system detects the differences in the reflected energy phase from each of these areas, and is used to determine the welding quality.

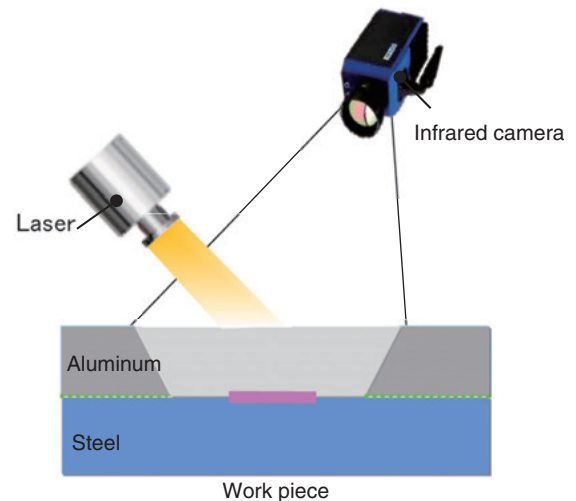


Fig. 10 Non-destructive inspection equipment

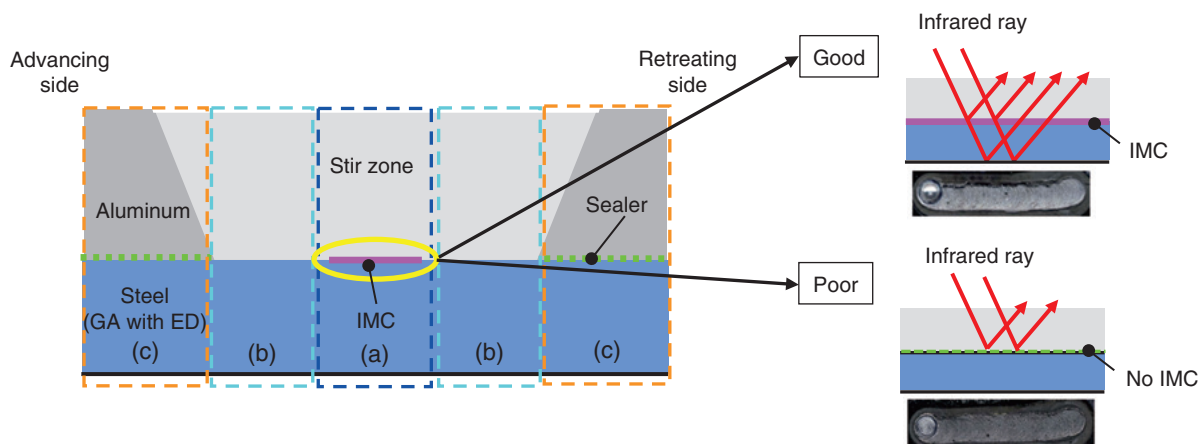


Fig. 11 Cross-section structure and judgment of welding

6.2. Steel and Aluminum Joint Structure

FSW joints are used for two-part structures with a steel part and an aluminum part, and three-part structures with two steel parts and one aluminum part. Therefore, the reflection method was adopted to enable measurement of all structures in the same manner (Fig. 12). In addition, the reflection method is affected by noise due to surrounding reflected light, so measurement conditions and judgment values were set and a re-measurement method was incorporated to increase the measurement accuracy with respect to reflected light.

Judgment criteria were set based on the relationship between the joint strength and the phase angle (Fig. 13), and adoption of this system enabled in-line inspection of all units.

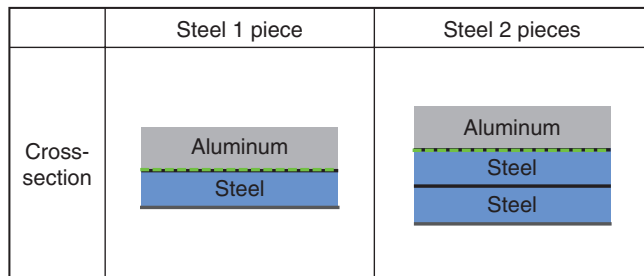


Fig. 12 Steel construction

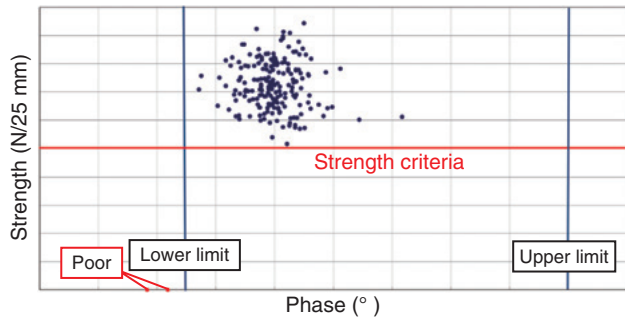


Fig. 13 Measurement result of non-destructive inspection

7. Structure of Hybrid Subframe Using Dissimilar Metal Welding Technology

The subframe functions to support the suspension, power plant and steering gear box (Fig. 14), and it also provides sufficient support rigidity and contributes to control in collision mode. Figure 15 and Table 2 show the structure and the material composition. The rear member is made of aluminum diecast to provide and increase support rigidity for the power plant and steering gear box with the aim of reducing both the weight and the number of component parts. In addition, this aluminum diecast rear member was also positioned to increase the rigidity of the suspension lower arm mount that has a high contribution to dynamic

Table 2 Materials of hybrid subframe for FSW

Steel parts		Aluminum parts	
Part name	Material	Part name	Material
Side member upper right-hand side	JAC270C	Rear member	Equivalent ADC3
Side member upper left-hand side			
Side member lower right-hand side			
Side member lower left-hand side			

performance. Steel sheet material is used for the side member lower to stabilize energy absorption and prevent the subframe from separating in the event of a collision. In the previous model, steel sheet member and aluminum diecast member components were joined mechanically using multiple bolts. The hybrid subframe has a structure with flanges formed from the various steel sheet members and aluminum diecast members, and these dissimilar metal flanges are welded using FSW. This reduces both weight and cost and saves energy.

The newly developed rear member is 3.2 times heavier and has a product projection area 2.4 times that of the member of the previous model (Fig. 16). Modification of the product shape and use of a compact casting plan enabled

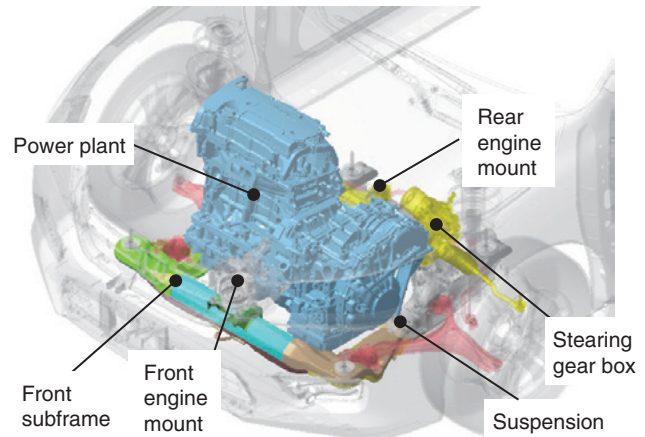


Fig. 14 Front subframe position of vehicle

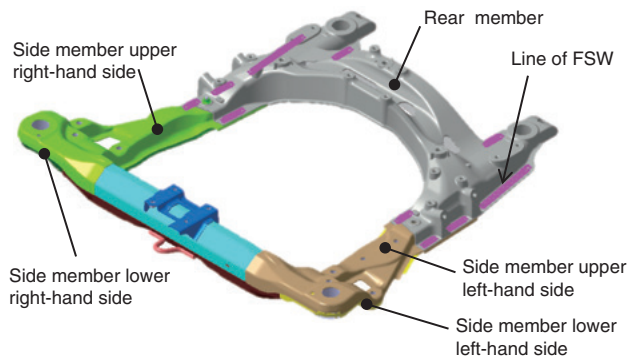


Fig. 15 Hybrid subframe components (after FSW)

using the same casting machine as the previous model, casting of two parts with a single die, increasing casting quality and reducing costs. In addition, the FSW welding surface thickness accuracy has a large influence on welding quality. Therefore, the required mechanical properties were secured and thickness accuracy was enhanced by optimizing the casting plan and casting conditions, modifying the cross-section and rib shapes required for the subframe functions, and enhancing the fluidity. This eliminated machining of the FSW joint surface, which reduced costs. As a result, by adopting an aluminum diecast rear member, and by establishing and applying dissimilar metal welding technology using FSW, the number of component parts was reduced by 40%, the weight was reduced by 25%, and the rigidity of the suspension mounting point was increased by 20%. Compared to the previous model, the MIG welding length was also reduced, which enabled reduction of electrical power consumption during the welding process by 50%.

Part name	Bolt tightening member	Rear member
Model year	2008	2013
Figure		
Weight [kg]	2.62	8.41
Projection area [cm ²]	1008	2446
Casting machine/cavity	High vacuum diecast 3500 ton/2 cavity	High vacuum diecast 3500 ton/2 cavity

Fig. 16 Comparison of bolt tightening member and rear member

8. Conclusion

Technology for continuously welding dissimilar metals using FSW was developed, and mass-production technology was established. As a result, the following results were obtained.

- (1) The IMC mechanism of welded joints was clarified, and appropriate welding conditions were constructed.
- (2) Technology was established for rustproofing structures that use dissimilar metal welding technology.
- (3) A C-shaped frame and jig structure using an industrial articulated robot and robot control technology were developed, and a highly versatile mass-production welding system was established.
- (4) Non-destructive inspection technology was developed that incorporates a laser excitation system onto the base of a thermal image analyzer using optical excitation, and an in-line inspection system for mass production was established.

Compared to the previous model, these effects reduced the weight by 25%, reduced the number of component parts

by 40%, increased the rigidity of the suspension mounting point by 20%, and reduced electrical power consumption during the welding process by 50%. In addition, costs could also be reduced by 5%. Expanded application of dissimilar-metal welding technology and hybrid structure subframes in the future can contribute further to reducing vehicle weights and conserving electrical power in the manufacturing process.

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Development of Road Input Prediction Technology for Setting of Suspension Durability Conditions

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Shunsuke KATAOKA*
Shingo YOKOYAMA*

ABSTRACT

Conventionally, durability input conditions for suspension durability performance tests have been set based on wheel input data obtained from actual vehicles. The research discussed in this paper developed a technology for the prediction of the wheel input data necessary for setting durability input conditions without using actual vehicles by means of full vehicle analysis using multi-body simulation. Rough road, curb stone road, winding road, panic braking and pot hole braking were set as the target driving modes, and digital roads were created and an ABS control model was incorporated in order to reproduce the driving modes. In addition, the fact that performing the vehicle set-up assuming the loading of gravity increased accuracy was verified, and it was possible to realize good accuracy in all the target driving modes using a single model. The developed technology makes it possible to set suspension durability conditions without using an actual vehicle.

1. Introduction

Complete vehicle simulations employing multi-body simulation have been utilized in a variety of fields, making it possible to respond rapidly to customer needs through the reduction of development costs. In addition, they have reduced development time⁽¹⁾⁻⁽⁸⁾. Reducing the number of prototypes used during development by means of complete vehicle simulations necessitates a technology able to compute all the items that should be obtained from actual vehicle tests. The suspension is one of the parts subject to durability evaluation of the vehicle body. The flow of evaluation of suspension durability is as follows. Using a prototype vehicle, the loads generated at the centers of the wheels are measured, and these figures are used in calculating the load allocated to each part. Next, with these allocated loads as input loads, unit component evaluations and evaluations of components in combination are conducted using bench test devices (Fig. 1). As this

indicates, a vehicle test is necessary only to obtain the load input to the centers of the wheels. If a simulation could be substituted for this measurement of input via a vehicle test, it would be possible to realize a development flow that did not use actual vehicles. This paper discusses the reproduction of driving modes, the adjustment of vehicle parameters and the creation of tools for the realization of increased accuracy, and the verification of accuracy towards the creation of a complete vehicle simulation technology using multi-body simulation and the realization of results of sufficient accuracy to be substituted for vehicle tests.

2. Overview of Complete Vehicle Simulation

A full vehicle model employing multi-body simulation was created in order to determine the inputs generated when vehicles travel on roads (Fig. 2).

A model of the 2008 FIT, a CVT vehicle with a 1.5-L engine, was formulated for study. For rough roads and curb stone roads, methods developed for use in previous studies were employed as references in the development of the model⁽¹⁾⁻⁽³⁾. The body, suspension arms, knuckles, hubs and other parts were modeled as rigid parts, while the torsion beams were modeled as flexible parts. A wheel force transducer model was created at the wheel center for evaluation of the input load.

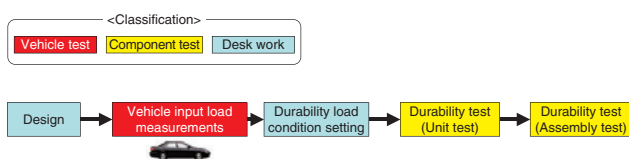


Fig. 1 Suspension development flow for durability

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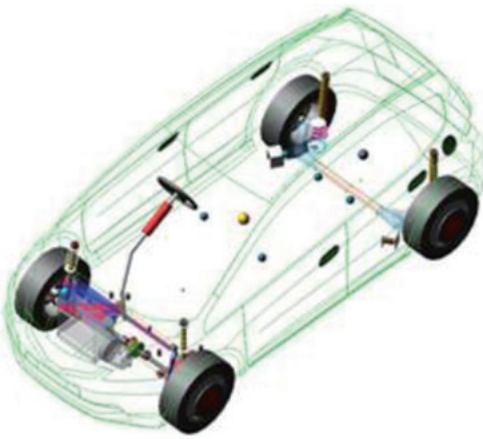


Fig. 2 Full vehicle simulation model

3. Creation of Model Environments for Reproduction of Driving

Loads from the road surface are input to the suspension in a variety of directions depending on the driving conditions. Driving modes have therefore been established for various directions of input to help ensure durability. In order to determine conditions for durability using a simulation, it is necessary to reproduce driving in all of these modes. Therefore, the organization of the necessary driving mode, digitalized the road surface in response to each driving mode, and select the variety of tire model and coupling technology between the multi-body simulation with a MATLAB/Simulink model for reproducing brake ABS behavior were applied in this project

3.1. Extraction of Driving Modes

Five driving modes are necessary for the determination of conditions for suspension durability. These are: winding road, panic braking, pot hole braking, curb stone road, and rough road. Figure 3 shows the driving modes and the corresponding target load directions.

3.2. Creation of Digital Road Surfaces

Three standard methods of creating a digital road surface are available.

Method (A): Shapes are expressed as mathematical formulas. This method is used for two-dimensional road surfaces or 2.5-dimensional road surfaces that reproduce the road surface shape only in the direction of the vehicle movement.

Method (B): Shapes are reproduced using triangular patches. This method is used for road surfaces for which detailed three-dimensional reproduction is necessary. Because ASCII format is used for descriptions and the road shapes are complex, computation costs tend to be high for this method (Fig. 4).

Method (C): Using a grid shape as the basis, three-dimensional shapes are reproduced by means of data in the direction of height for each point of intersection. This

method is used for road surfaces for which the reproduction of complex three-dimensional shapes is necessary. Increases in computational cost can be minimized by using binary data and computing three-dimensional road surfaces after simplifying the expression of the shape by means of the grid concept⁽⁹⁾ (Fig. 5).

Based on considerations of computation cost and the necessity of reproducing the road surfaces, the research discussed here selected the following model creation processes for the generation of digital road surfaces (Fig. 6).

Method (A) was selected for the reproduction of panic braking, for which a flat plane can be computed. Method (B) was selected for curb stone road, pot hole braking, and winding road modes, which necessitate detailed three-dimensional shapes but do not feature continuous concavities and convexities. For rough roads on which complex concavities and convexities occur continuously and it is necessary to reproduce shapes with a

Driving mode	Target load direction
Winding	Mx
Panic braking	Fx, My
Pot hole braking	Fx
Curb stone	Fx, Fz
Rough road	Fx, Fz

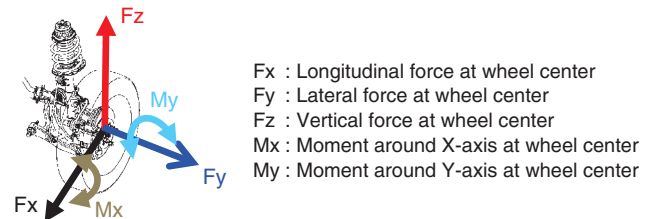


Fig. 3 Driving mode for each load direction

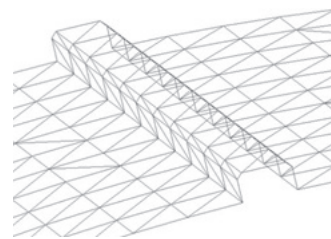


Fig. 4 Image of format (B)

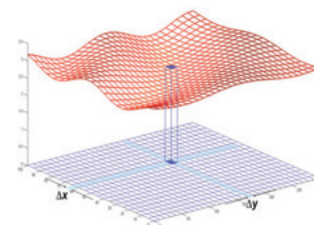


Fig. 5 Image of format (C)

high level of accuracy, the marked increase in computation cost occasioned by the use of method (B) represented an impediment to the performance of the computations, and method (C), which is slightly inferior to method (B) in terms of reproduction of shapes but offers a sufficient level of reproduction for the evaluation of durability in addition to realistic computation costs, was therefore selected.

For curb stone and rough roads, road surface measurement data obtained in previous studies was used as the basic data for the generation of road surface models. In the case of pot hole braking, drawing data was employed, while for winding road, road surface measurements were conducted and the data was digitalized.

3.3. Selection of Tire Models

From among the tire models available for multi-body simulation, the present research selected FTire⁽⁹⁾ and Magic Formula tires from considerations of compatibility with a software in multi-body simulation and cost of introduction. These two types of tire model were employed in response to the characteristics of the specific driving mode in use. Magic Formula reproduces lateral force, longitudinal force and other forces generated on the tire as tire models using mathematical formulas, making it possible to control computation costs. It is generally employed to conduct studies in vehicle dynamics analyses. FTire models reproduce the steel belts of the tires as a mesh, making it possible to reproduce the tires' envelope characteristics. Because FTire models are able to reproduce the behavior of the tires around their points of contact with the road surface in detail, computation cost increases. On the other hand, the models are also able to reproduce the envelope characteristic, the tire's property of wrapping around small projections, with a high degree of accuracy. FTire models are used in a range of applications, from computations for the area of strength and durability, as in the present research, to calculations for the area of passenger comfort.

In selecting tire models, the research focused on the

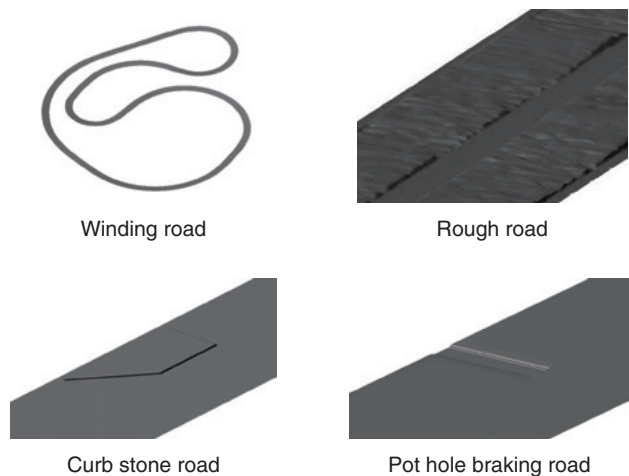


Fig. 6 Road shape for each driving mode

Table 1 Varieties of tire model

Driving mode	Tire model	
	FTire	Magic Formula
Winding		✓
Panic braking		✓
Pothole braking	✓	
Curb stone	✓	
Rough road	✓	

necessity for accuracy in envelope characteristics in each driving mode. In the case of winding road and panic braking, while it was necessary to consider height differences on the road and the gradient in the lateral direction, the mode featured driving on a comparatively flat road, and it was therefore considered that the necessity for accuracy in reproducing the tires' envelope characteristics was low. A Magic Formula tire model was therefore selected. By contrast, pot hole braking, curb stone road and rough road are modes in which the tires pass over projections of various sizes, and the reaction force generated by the tires would differ depending on the shape of the projection and the state of the tire. Given this, it was considered that it was necessary to reproduce envelope characteristics accurately, and an FTire model was therefore selected (Table 1).

3.4. Application of Technology for Coupled Analysis of Multi-body Simulation and Control

For the panic braking and pot hole braking modes, it is important to reproduce ABS characteristics. The model was therefore coupled with a MATLAB/Simulink model that modeled the ABS characteristics. Figure 7 provides an overview of the model.

The coupled simulations between multi-body simulation and control were conducted using MATLAB/Simulink. The Simulink model was composed of a vehicle model block created by changing the format of a multi-body simulation model to S-function format, a controller block that applied ON/OFF control to the brake pistons as ABS control, and a hydraulic unit block that controlled hydraulic pressure. Vehicle behavior was obtained from the multi-body simulation model for use in computations in the respective blocks, and ABS behavior was reproduced by calculating the brake piston pressure generated for each wheel.

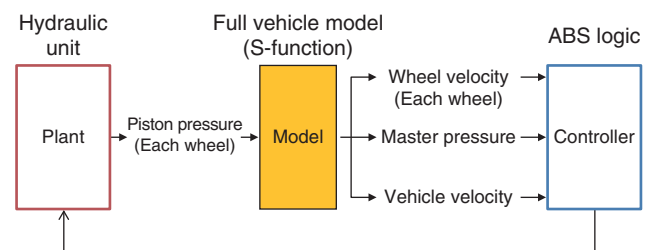


Fig. 7 Co-simulation flow in MATLAB/Simulink

4. Adjustment of Model Parameters

4.1. Verification of Accuracy by means of Constant-radius Cornering

Previous papers have treated the verification of accuracy for rough roads and curb stone roads⁽¹⁾⁻⁽³⁾, and the present research therefore restricted its focus to winding roads, panic braking and pot hole braking, and narrowed down the necessary items for adjustment of parameters by means of constant-radius cornering (Fig. 8).

Figure 9 shows a comparison of results for the change in the load on the outer front wheel during cornering.

Based on the simplicity of the constant-radius cornering driving mode, the complexity of the target driving mode, and the final target for accuracy, a target accuracy of within 5% was set for efforts to increase accuracy.

4.2. Contribution Analysis for Increased Accuracy

A contribution analysis was conducted in order to narrow down the parameters for enhancement of the model towards the realization of increased accuracy. The analysis was conducted for the parameters shown below. Figure 10 shows the results of the analysis.

- (1) Mass, height of center of gravity, and moment of inertia of complete vehicle
- (2) Initial toe and initial camber
- (3) Damper damping, spring stiffness, bump stop stiffness, stabilizer stiffness
- (4) Bush stiffness and damping

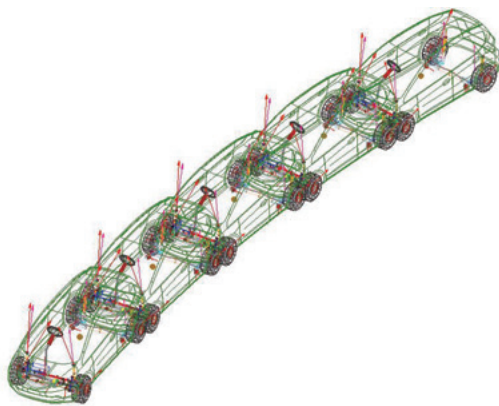


Fig. 8 Constant radius cornering

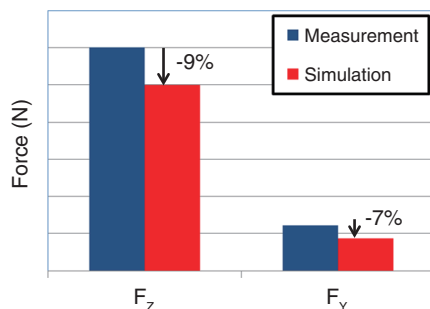


Fig. 9 Comparison of front wheel load at wheel center

The results showed a high level of contribution for the complete vehicle mass, initial camber, height of center of gravity and initial toe. These results indicated that the parameters defining the initial state of the vehicle displayed a high level of contribution. The next highest contribution was displayed by parts stiffness items, such as the stiffness of the rear bump stop and the front stabilizer. In the present research, efforts were made to minimize error, for example by using the measurement data for rear bump stop stiffness and, in the case of front stabilizer stiffness, employing parts of the same specifications as had been used in analyses in an actual vehicle. For the reasons discussed above, the research focused on the reproduction of the initial state of the vehicle in attempting to increase accuracy.

4.3. Verification of Reproduction of Initial Vehicle State

Figure 11 shows the computation process for conducting driving computations from the stage of construction of a full vehicle model.

- Step 1: Vehicle parameters for the state of the vehicle at the stage of creation of the model are adjusted to match the vehicle characteristics that are the subject of computations.
- Step 2: Static equilibrium calculations are conducted with 1 g of gravity ($1\text{ g} = 9.8\text{ m/s}^2$) imparted to the vehicle model. This state is defined as the state in which the vehicle is stabilized by 1 g of gravity ("1 g condition").
- Step 3: The desired driving mode is calculated using the vehicle state described in Step 2.

Normally, vehicle parameters for the reproduction of the initial vehicle state are adjusted in Step 1. However, given

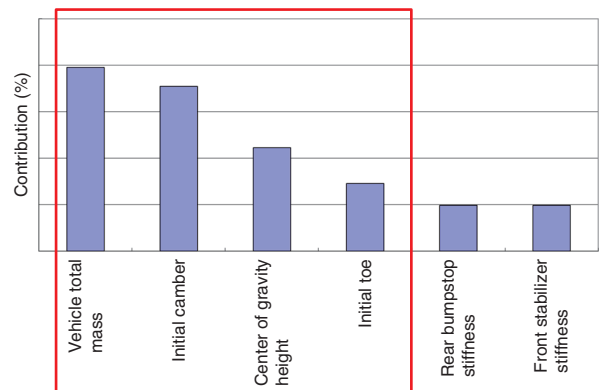


Fig. 10 Results for contribution of parameters

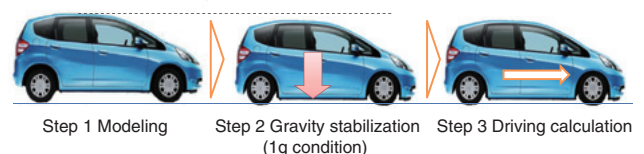


Fig. 11 Process for driving calculation

Table 2 Variation in set parameters due to 1 g load

Set parameter		Modeling (Step 1)	1g condition (Step 2)	Error
Center of gravity height (mm)		494	449	-45
Initial toe (deg)	Front	0.00	0.01	+0.01
	Rear	0.10	0.34	+0.24
Initial camber (deg)	Front	0.00	-0.56	-0.56
	Rear	-1.00	-1.39	-0.39

that gravity of 1 g is acting on an actual vehicle at all times, the state in Step 2 (1 g condition) can be considered to be the initial state to which the model should be matched. The amount of change in the vehicle parameters from Step 1 to Step 2, with a load of 1 g of gravity applied, was therefore studied. The results are shown in Table 2.

Based on the results shown in Fig. 10, the height of the center of gravity, the initial camber, and the initial toe were selected as the vehicle parameters to be focused on. Vehicle mass, which had displayed the highest degree of contribution, was excluded as an item because it did not change under gravity of 1 g.

The results obtained in this study confirmed that the vehicle parameters changed from the target values for vehicle characteristics set for the initial state in the model creation process (Step 1) under a load of 1 g gravity. The height of the center of gravity and the initial camber displayed a particularly significant degree of change, and these were therefore items to be enhanced in the quest for increased accuracy in the analysis. This was considered to result from the fact that the model had been created with reference to drawing data, but the vehicle state used in durability evaluations differs from the vehicle state assumed in this data.

4.4. Enhancement of Method of Reproduction of Vehicle Characteristics

It was considered that in order to increase the accuracy of driving analysis, it would be necessary to adjust vehicle parameters under the 1 g condition rather than at the stage of model creation. However, because the 1 g condition was the result of static equilibrium calculations, it was not possible to directly adjust model parameters in relation to it. In the case of the height of the center of gravity, which displayed a significant difference from its initial value under the 1 g condition, the initial value was re-set with consideration of the degree of change under the 1 g condition, as shown in Fig. 12.

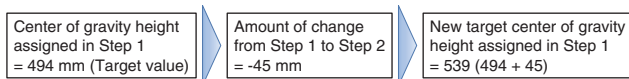


Fig. 12 Flow of adjustment of target value

Figure 13 shows a comparison of loads with a model in which the parameters had been adjusted based on the concept described above.

The results show an enhancement of load transition, with a difference in load of 5% for F_z and 2% for F_y , and a consequent increase in the accuracy of lateral force, realizing reproduction that meets accuracy targets.

4.5. Development of Tool for Automation of Process

Results had indicated that adjustment of vehicle characteristics under the 1 g condition was essential for the realization of increased accuracy.

Attempting to increase the accuracy of analysis as much as possible, the present research aimed to set all of the vehicle set-up parameters that it is necessary to manage in actual vehicle tests under the 1 g condition. Vehicle height, bump stop clearance, initial toe, initial camber, height of the center of gravity and moment of inertia around the center of gravity were selected as the vehicle parameters. Because there is mutual interference between vehicle height, bump stop clearance and height of the center of gravity, it was necessary to conduct iterative calculations similar to convergence calculations for these parameters. This made it impossible to use the simple method discussed in Section 4.4, which significantly increased the person-hours for procedures during the adjustment of the model, making the method unsuitable for use in the automotive development process.

Given this, a tool to make it possible to automate the vehicle parameter adjustment process while maintaining the 1 g condition was developed in order to save time. The resulting tool reduced the person-hours involved in adjustment of parameters and made it possible to apply the method in the automotive development process.

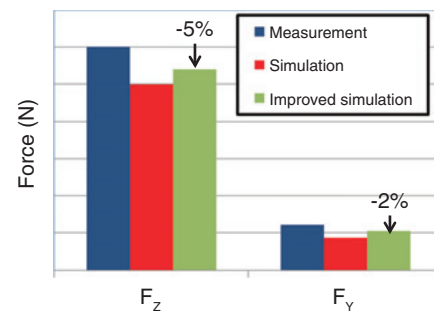


Fig. 13 Reevaluation of front wheel load at wheel center

5. Verification of Accuracy in Target Driving Modes

Accuracy in the target driving modes was verified using the models in which accuracy had been enhanced. The results are shown in Figs. 14 to 20.

(a) Winding

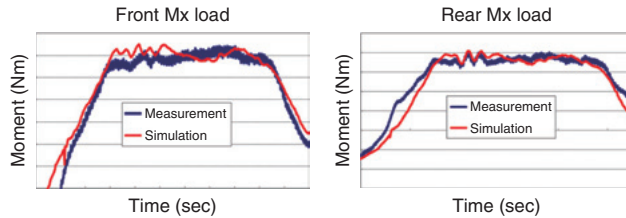


Fig. 14 Moment around X-axis at front and rear wheel center

(b) Panic braking (ABS control)

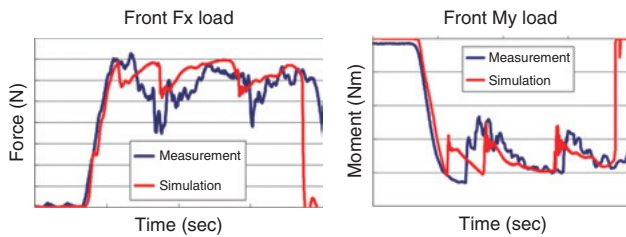


Fig. 15 Longitudinal force and moment around Y-axis at front wheel center

(c) Pot hole braking (ABS control)

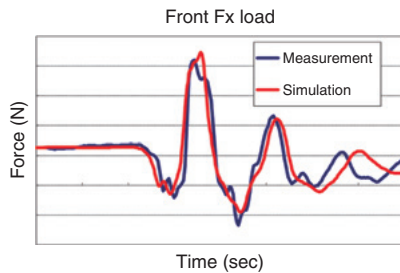


Fig. 16 Longitudinal force at front wheel center

(d) Curb stone

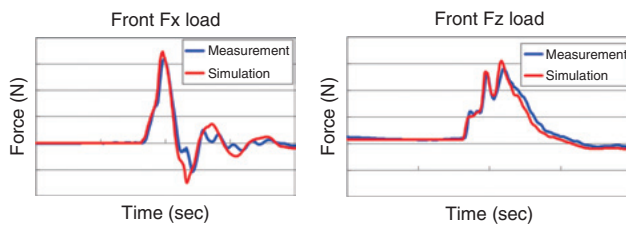


Fig. 17 Longitudinal and vertical force at front wheel center

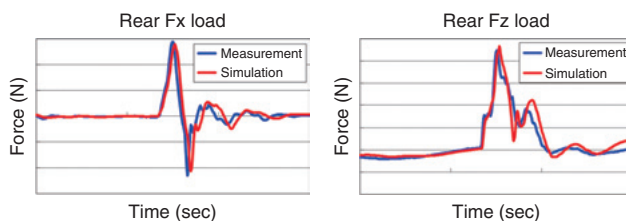


Fig. 18 Longitudinal and vertical force at rear wheel center

(e) Rough road

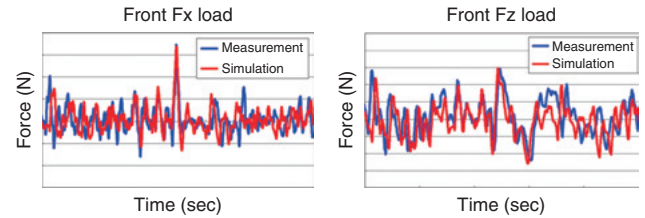


Fig. 19 Longitudinal and vertical force at front wheel center

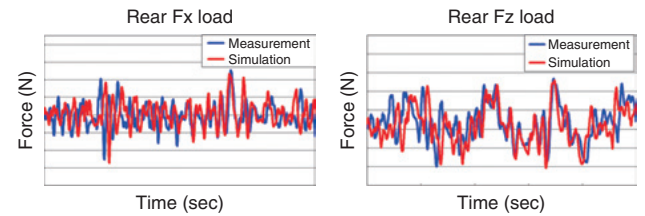


Fig. 20 Longitudinal and vertical force at rear wheel center

A good correlation with vehicle test results was achieved for target loads for all target driving modes. This was considered to demonstrate that the models achieved sufficient accuracy for use as a substitute for vehicle tests.

6. Conclusion

The following technologies were developed as a substitute for the measurement of inputs in vehicle tests in order to determine suspension durability conditions, and were refined for application in the automotive development process.

- (1) Digital roads were created and tire models were selected in order to reproduce all the driving modes necessary for vehicle input measurement tests via multi-body simulation using a single model.
- (2) The importance and effectiveness of employing vehicle parameters incorporating the 1 g condition in increasing accuracy was verified, and a tool was developed in order to reduce adjustment time.
- (3) The accuracy of the method was tested, and it was verified as offering sufficient accuracy for use as a substitute for vehicle tests.

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Introduction of Solar Hydrogen Station Installed in Japan

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ABSTRACT

A high differential pressure water electrolysis-type solar hydrogen station has been installed on the campus of the Saitama Prefectural Office in Japan. This is Japan's first demonstration with this type of system, and the first such system in the world to be installed in a public location. A station of the same type, conforming to U.S. standards, had previously been installed at Honda R&D Americas, Inc. Los Angeles Center in Torrance, California, but it was necessary to make modifications in introducing the system to Japan in order to conform to Japanese laws and regulations. Thus, a fire-protection wall was put in place, the system ventilation airflow was increased and pressure proof tests were conducted on the high differential pressure water electrolysis stack. In the course of accommodating the Japanese laws and regulations, it was realized that under the present laws, the same configurations would be necessary when considering the household installation of such hydrogen stations. The study has also shown that this hydrogen station is capable of stably producing hydrogen in a verification test conducted in coordination with a fuel cell electric vehicle being used as an official vehicle by Saitama Prefecture.

1. Introduction

Viewing hydrogen fuel cells as the ultimate next-generation automotive powerplant, providing an alternative to fossil fuels and functioning to reduce exhaust emissions and assist in mitigating global warming, Honda has been engaged for some time in development efforts related to fuel cell electric vehicles.

Honda also takes research on technologies for the supply of hydrogen to fuel cell electric vehicles with minimal carbon emissions very seriously, and, looking towards the realization of a future hydrogen energy-based society, is conducting R&D in the area of solar hydrogen stations that use solar cells as their power source. As the first stage in this process, Honda commenced experimental tests on a solar hydrogen station produced using commercially available components at Honda R&D Americas, Inc. Los Angeles Center in Torrance, California, from July 2001. In October 2003, the company introduced a uniquely developed normal pressure water electrolysis-type solar hydrogen station (Solar Hydrogen Station 1, or SHS1)⁽¹⁾, and in January 2010 replaced the SHS1 with a high differential pressure water electrolysis-

type solar hydrogen station (Solar Hydrogen Station 2, or SHS2)^{(2), (3)}. This paper will introduce a version of the SHS2 established to domestic specifications on the campus of the Saitama Prefectural Office (termed "domestic SHS2" below). This station is Japan's first demonstration of high differential pressure water electrolysis system, and the first such system in the world to be situated in a public location. The system was jointly established by the Saitama prefectural government, Iwatani Corporation, Honda Motor Company, Ltd., and Honda R&D Co., Ltd, as part of the project "CO₂ emission reduction system which combined small scale solar hydrogen station and fuel cell electric vehicles," itself part of the Ministry of the Environment's FY2011 Low Carbon Technology Research and Development Program. On March 27, 2012, the station was delivered to the Saitama prefectural government, and the project was commenced for a projected period of two years. The domestic SHS2 has been modified in relation to the U.S. specifications ("U.S. SHS2" below), chiefly in order to conform to laws and regulations related to domestic facilities. This paper will discuss the design of the domestic SHS2, responses to laws and regulations, and the operating status of the station.

* Automobile R&D Center

2. Design

2.1. Concept

Figure 1 shows the design concept of the SHS2.

The concept for the station was FLUID FUNCTION, as in the case of the U.S. SHS2. The flow of energy production by the system is expressed by symmetrical curved surfaces that offer a sense of stability. The finishing of the area around the station, the color choice, and the positioning of the station were designed to help ensure that it would blend in with the surrounding environment in the grounds of the prefectural office with no sense of incongruity, and the exterior of the station overall sought to offer a feeling of advanced technology. Figure 2 shows the exterior of the domestic SHS2 that embodies these design concepts.

2.2. Layout

Looking towards the widespread diffusion of small hydrogen stations in the near future, the domestic SHS2 is a facility that conforms to Japanese laws and regulations concerning the design of high-pressure gas refueling stations, and that is being used in tests and evaluations of equipment for the supply of hydrogen fuel

for fuel cell electric vehicles. Because the station is used to supply fuel for the fuel cell electric vehicle that serves as the Saitama Governor's official vehicle every working day, the station was designed to be able to be used safely and effectively by any user. The layout of the station is compact but offers a sense of openness, while also taking into consideration a variety of design features considered necessary for the future use of similar equipment in ordinary households. Figure 3 shows a CG layout of the domestic SHS2.

The vehicle to be fueled backs up to the filling position within the guide lines, and stops at the optimum position as indicated by a guide bump positioned at the rear of the vehicle. The curb around the island on which the refueling dispenser is located is elevated 150 mm above ground level, and the distance of the refueling dispenser from the curb was determined with consideration of preventing impact with the dispenser even in the event of mis-steering. In consideration of rainy climates, a roof has been added to the area where the vehicles park, to help protect them from rain during the refueling process. To help ensure that there is no interference between the roof and the doors of vehicles being fueled when occupants get out, and in addition to maintain the harmony of the overall design, the roof uses cantilevered pillars, creating a comfortable refueling area with a sense of openness.

2.3. User Interface

The U.S. SHS2 employs a user interface (UI) screen offering simple and intuitive operations; the UI screen was modified for the Japanese station. Figure 4 shows the UI screen. First, the display language was changed entirely to Japanese. Instructions for fueling operations are presented in an animated form, making them more intuitively understandable. In addition, while the U.S. SHS2 provides a numerical display of the status of hydrogen fueling, the domestic version offers a graphic display in which a gauge fluctuates depending on the state of fueling, and also indicates the remaining fueling time, increasing convenience for users.

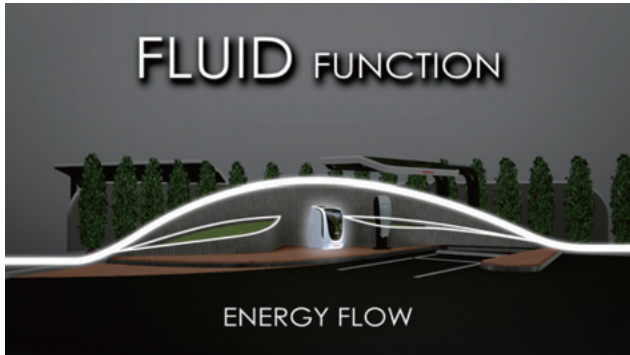


Fig. 1 Design concept of SHS2



Fig. 2 External view of SHS2 installed on Saitama prefecture property

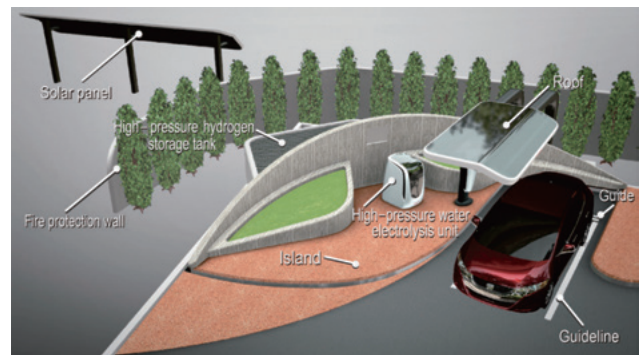


Fig. 3 System layout of SHS2



(a) Connecting nozzle



(b) Refueling

Fig. 4 UI screens of SHS2

3. System Specifications

Like the U.S. SHS2 (as previously reported⁽²⁾), the domestic SHS2 combines a high differential pressure water electrolysis system with solar cells, making it possible to generate hydrogen without any associated CO₂ emissions. Table 1 shows the specifications of the U.S. and the domestic SHS2 stations. In terms of basic specifications, the two stations are identical. The specifications differ in the hydrogen storage amount, the area of the solar cells, and the ventilation flow rate. Because the U.S. SHS2 was developed in order to conduct proving trials of hydrogen home refueling appliances, the hydrogen storage amount was set at a low 4 kg (35 MPa, 167 L). At this hydrogen storage amount, final refueled pressure is too low to complete the hydrogen refueling process exclusively by means of fast fill, which uses the pressure differential between the hydrogen storage tank and the fuel cell electric vehicle tank. Because of this, after hydrogen has been fueled using fast fill, the U.S. SHS2 employs slow fill, a method in which hydrogen is supplied directly to the vehicle's tank from the high differential pressure water electrolysis stack. In Japan, however, the law stipulates that a person must be present while the vehicle is being fueled with hydrogen, and slow fill, which would take several hours at the hydrogen

Table 1 Specifications of SHS2

		United States	Japan
Hydrogen production	Flow rate	0.7 Nm ³ /h	
	Filling pressure	35 MPa	
	Purity	>99.99%	
	Storage	4 kg	20 kg
Component	Photovoltaic module	CIGS 6 kW	CIGS 10.5 kW
	Electrolysis unit	High differential pressure electrolyzer	
Utility	Electricity	240 VAC / 60 Hz	200 VAC / 50 Hz
	Water	Tap water	
	Ventilation	3.5 Nm ³ /min	7 Nm ³ /min
Unit volume		Approx 0.33 m ³	

production rate of the SHS2, is therefore impracticable. The domestic SHS2 therefore does not employ slow fill. A high pressure is necessary in order to complete hydrogen fueling using fast fill, and it was also necessary to take into consideration the fact that the station would be situated in a public area, and therefore might be used to fuel multiple vehicles. Given this, the hydrogen storage amount for the domestic SHS2 was set at approximately 20 kg (35 MPa, 828 L). Like the U.S. SHS2, the domestic SHS2 employs CIGS thin-film solar cells⁽⁴⁾ manufactured by Honda Soltec in its solar cell panels; the increase in the hydrogen storage amount necessitated a higher hydrogen production, and the output of the cells was therefore increased from 6 to 10.5 kW. In addition, the system is also able to utilize commercial power in order to produce hydrogen when power output from the solar cells declines at night, due to rainy weather, or simply when necessary.

4. Responses to Laws and Regulations

4.1. Overview of Laws and Regulations

Table 2 shows the applicable laws, regulations and standards in Japan and the U.S. Laws, regulations and standards concerning the installation of hydrogen stations differ between the two countries. ISO 22734-2⁽⁵⁾, for residential applications of hydrogen generators using a water electrolysis process, has already been established as an international standard for this type of hydrogen generator, and this was the main standard applied to the U.S. SHS2. Other than ISO 22734-2, the U.S. NFPA55⁽⁶⁾ standard for the handling of high-pressure gas and NFPA497⁽⁷⁾ standard for the classification of hazardous locations were also applicable, and the system received approval for operation as a hydrogen station as a result of satisfying these standards. The SHS2 had therefore already been designed to satisfy U.S. standards, but because domestic standards based on the High Pressure Gas Safety Act and the Building Standards Act were more stringent, it was necessary to introduce new responses in order to comply with these regulations.

Table 2 Codes and standards for US and Japan

	United States	Japan
Water electrolysis applications	ISO22734-2	None
High-pressure gas applications	NFPA55	High Pressure Gas Safety Act Building Standards Act
Classification	NFPA497	JIS60079-10

4.2. Differences in Responses in U.S. and Japan

4.2.1. Responses to Japanese laws and regulations

The High Pressure Gas Safety Act classifies high-pressure gas production equipment into Class 1 Producers, the operation of which must be approved, and Class 2 Producers, on which there are comparatively few restrictions, and that can be installed after the relevant authorities are notified of the fact. Based on its specifications and volume of hydrogen production, the SHS2 corresponds to a Class 2 Producer. Nevertheless, because the Saitama Prefectural Office, where the domestic SHS2 would be installed, is located in a commercial district, the restrictions on areas able to be used stipulated by Article 48 of the Building Standards Act would, in principle, prohibit the production of flammable and compressed gases. However, in accordance with the government's measures to stimulate more widespread use of fuel cell electric vehicles, hydrogen stations are able to be installed if they are Specific Compressed Hydrogen Stations, which have received approval as Class 1 Producers. It was therefore necessary to design the domestic SHS2 to conform to the technical standards related to Specific Compressed Hydrogen Stations as stipulated in Article 7, Part 3(2) of the Security Regulation for General High-Pressure Gas. The main points of the Security Regulation to which it was necessary to respond in the domestic SHS2 as a high differential pressure water electrolysis system were as follows:

- (1) Installation of a fire-protection wall around the area in which the system is installed
- (2) Installation of a firefighting sprinkler system in order to control temperature increase in the tank
- (3) Positioning of the system at least 6 m from any equipment or facility in which fire is employed
- (4) Demonstration by calculation that the components have sufficient strength; otherwise, in pressure proof tests that the components would not rupture at more than four times normal operating pressure
- (5) Guarantee of explosion-proof performance in the unit

4.2.2. Area of installation of system

In selecting the area for installation of the unit, it was necessary to consider items (1) Installation of a fire-protection wall, (2) Installation of a sprinkler system, and (3) Distance from equipment of facilities using fire in Section 4.2.1 above. A fire-protection wall that wraps around the back of the island on which the system is positioned was therefore installed, as shown in Fig. 3. A

sprinkler system was installed at the tank section in order to help prevent fires and control temperature increase. In addition, as will be discussed below, the SHS2 maintains explosion-proof performance by means of ventilation, but the system contains relays and other sources of potential ignition, and these can be regarded as fires. Given the necessity of satisfying item (3) in Section 4.2.1, the air inlet for system ventilation was positioned more than 6 m from the system. By contrast, in the case of the U.S. SHS2 it was not necessary to install a fire-prevention wall or a sprinkler system, and requirements regarding distance from equipment or facilities that use fire did not apply to the system. However, NFPA55 regards vent pipes for the release of hydrogen as a source of hazard, and stipulates a distance of at least 15 ft (approx. 4.6 m) from the vent pipe to the system. Response to this stipulation was the only response necessitated for the U.S. SHS2.

4.2.3. Stack

The High Pressure Gas Safety Act stipulates that the Security Regulation for General High-Pressure Gas is applicable to any container for which the product of the designed pressure expressed in MPa and the internal volume expressed in m³ is 0.004 or less. This is expressed in equation form as Eq. (1). The Designated Equipment Inspection Regulations apply to all other high-pressure containers, and these regulations place major restrictions on the structure of the container and the materials used in comparison to the Security Regulation for General High-Pressure Gas.

$$P \times V \leq 0.004 \quad (1)$$

P : Designed pressure (MPa)

V : Interior volume (m³)

Because the designed pressure of the high differential pressure water electrolysis stack used in the SHS2 is 38.5 MPa, if the internal volume was 103.9 cm³ or less, the Security Regulation for General High-Pressure Gas would apply. The internal volume of the water electrolysis stack is 100.8 cm³, meaning that it was the Security Regulation for General High-Pressure Gas for which compliance was necessary.

As indicated in item (4) in Section 4.2.1, it is necessary for a high-pressure vessel to have sufficient strength corresponding to the normal operating pressure. An illustration of the Security Regulation for General High-

Pressure Gas specifies calculation methods for pressure resistance when a container is manufactured from metal, and it is also possible to use calculations to indicate the ability of components such as pipes, relief valves, check valves, and solenoid valves that possess pressure-resistant structures made exclusively of metal to have sufficient strength. However, because a part of the pressure-resistant structures in the high differential pressure water electrolysis stack is composed of non-metallic proton exchange membranes, the formulas specified by the illustration of the Security Regulation for General High-Pressure Gas cannot be applied. The ability of the stack to resist rupture was therefore verified in pressure resistance tests at 160 MPa, which is more than four times normal operating pressure.

4.2.4. Pressure resistance design for components

Like the high differential pressure water electrolysis stack, it was also necessary to demonstrate that the components used in high-pressure sections were able to resist four times normal operating pressure. By contrast, the U.S. regulations stipulated that pressure-resistant containers in high-pressure sections should resist 2.4 times normal operating pressure, and plumbing should resist three times normal operating pressure. The components employed in the SHS2 had been selected on the assumption of being used in a U.S. system, and it was therefore necessary to verify that each device was able to resist four times design normal operating pressure. The performance of each component used in the SHS2 was investigated and the results showed that components including the filters and various fittings did not satisfy pressure resistance demands. These components were therefore redesigned to give them sufficient thickness to resist four times normal operating pressure.

In addition, to help ensure space, the high-pressure hydrogen separator and the absorber vessel were coupled by square flanges. However, the Security Regulation for General High-Pressure Gas does not specify any formulas for square flanges, and these were therefore redesigned as cylindrical flanges, for which the Security Regulation does specify formulas.

4.2.5. Ventilation flow rate

As indicated by item (5) in Section 4.2.1, because the system contains non-explosion-proof components, the Security Regulation for General High-Pressure Gas specifies that the system unit should have a structure in which ventilation helps to ensure explosion proofness. A continuous ventilation system has been employed to help ensure explosion proofness in the SHS2, and this system is also used in the U.S. SHS2. The concept behind this use of ventilation for explosion proofness is to employ sufficient ventilation such that even if a hydrogen leak occurs in the system, the hydrogen content concentration in the air will be less than 1%, or less than one-quarter of the lower

flammable limit.

In the case of the U.S. NFPA497 regulations and ISO 22734-2, it is not necessary to consider ruptures, disconnection, or similar phenomena in parts that have been designed for pressure resistance. The Japanese High Pressure Safety Act, however, does demand consideration of sudden escape of hydrogen due to the rupture of a pipe. The scenario to be assumed is that a pipe ruptures when the system is operating at full pressure, and all of the hydrogen produced by the system leaks into the system. At this time it is necessary to maintain a high degree of ventilation by means of a formula stipulated by JIS C 60079-10⁽⁸⁾. The conditions for this maintenance of ventilation are shown in Eq. (2). Here, a high degree of ventilation is a degree of ventilation helping to ensure that the gas concentration in the source of the gas release is instantaneously reduced. Satisfying Eq. (2) makes it possible to avoid the danger associated with a release of gas. Given that the system's hydrogen production rate is 0.7 Nm³/h and its interior volume is 0.17 m³, from Eq. (2), it is necessary to maintain a ventilation flow rate of 5.6 Nm³/min. A ventilation fan capable of realizing a ventilation flow rate of 5.6 Nm³/min or higher was therefore installed in the domestic SHS2.

$$\frac{f \times (dG/dt)_{\max} \times T}{C \times k \times LFLm \times 293} < 0.1 \quad (2)$$

f	: Ventilation efficiency
$(dG/dt)_{\max}$	: Maximum hydrogen production rate
T	: Surrounding temperature
C	: Ventilation flow rate / Volume to be ventilated (Interior volume)
k	: Safety rate in relation to $LFLm$
$LFLm$	: Lower flammable limit

4.3. Issues Related to Household Installation

By making modifications in order to conform to Japanese laws and regulations, as described above, it was possible to establish a domestic version of the SHS2 in Japan. However, under the terms of the Building Safety Act, in Japan at present the installation of hydrogen stations and other equipment that produce flammable gas is restricted in principle to industrial districts. It is possible to obtain a special ruling from the relevant authorities to locate a facility outside an industrial district, as in the case of the domestic SHS2, but in such cases the technical standards relating to the large facilities classified as Class 1 Producers are applied. Therefore, for household installation of a system of this type, it would be necessary to install a fire-prevention wall and a sprinkler system, and to position the system 6 m or more from any flames, as discussed above. It would be challenging to satisfy these conditions in an ordinary household. In addition, by contrast with the U.S., where components are expected to display resistance to 2.4 times and three

times normal operating pressure, Japanese laws stipulate that components should be able to resist four times normal operating pressure, resulting in increased cost. Given this, while it would be possible to install the domestic SHS2 in households under existing Japanese law, it would be costly and necessitate the use of a large amount of space. It is clear that the introduction of further legislation will be necessary in order to promote widespread use of the system.

The goal of the present research is ultimately to promote the installation of small, inexpensive hydrogen supply systems in households, and in future it will be necessary for Honda to influence the authorities to make legal changes to enable domestic installation of the systems.

5. Operating Status

Despite the fact that a number of modifications were made to the domestic SHS2, for example to increase pressure resistance and the ventilation flow rate, as indicated in Chapter 3, the basic makeup of the system is identical to that of the U.S. SHS2, and the system therefore displays identical levels of system efficiency and hydrogen quality to the U.S. system. Table 3 shows the quality of the hydrogen produced by the SHS2. While some components of the gas cannot be adequately measured due to limit values for detection, overall, the figures are identical to figures for the U.S. SHS2⁽³⁾.

The proving trial discussed here is being conducted in order to study patterns of use when fuel cell electric vehicles are used as the official vehicles of a local administration, and the consequent demands on a hydrogen station. Figure 5 shows a profile of hydrogen production over a typical month. The current domestic SHS2 is being used to fuel a single fuel cell electric vehicle used as an official vehicle by Saitama Prefecture, and is demonstrating the ability to stably produce hydrogen in response to the distance driven by the vehicle. The findings produced by the proving trial will be compiled at the completion of its two-year period.

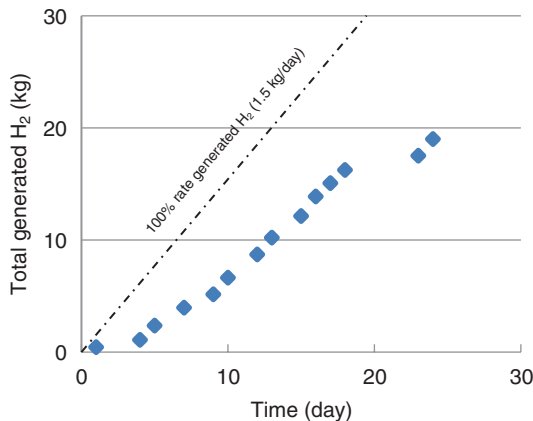


Fig. 5 Typical H₂ production at Saitama

Table 3 Hydrogen quality of SHS2

	ISO14687-2	SHS2 US	SHS2 JP
H ₂ (%)	>99.99	>99.99	>99.99
H ₂ O (ppm)	<5	<1	<5
Hydrocarbon (ppm)	<2	<0.1	<0.3
O ₂ (ppm)	<5	1	<0.1
He, Ar, N ₂ (ppm)	<100	<16	<0.1
CO ₂ (ppm)	<1	<0.1	<0.1
CO (ppm)	<0.2	<0.001	<0.1
Sulfur (ppm)	<0.004	<0.0008	<0.8*
HCHO (ppm)	<0.01	<0.001	<0.01
HCOOH (ppm)	<0.2	<0.001	<5*
NH ₃ (ppm)	<0.1	<0.02	<0.01
Halogenates (ppm)	<0.05	<0.004	<0.5*

* below detectable limit

6. Conclusion

A high differential pressure water electrolysis-type solar hydrogen station has been established in Japan. The following are the salient points of this study.

- (1) Modifications were made to the system in order to respond to Japan's current laws and regulations relating to the installation of hydrogen stations; this is Japan's first demonstration with this type of system, and the first to be installed in a public location anywhere in the world.
- (2) A survey of relevant laws and regulations indicates that Japanese laws do not make adequate provision for domestic installation of systems of this type. It will be necessary to identify the issues related to future domestic installation of the systems, and to work towards the modification of existing laws.
- (3) The results of the initial operation of the domestic SHS2 indicate that its system efficiency and hydrogen quality are identical to those of the U.S. system. In addition, the system is able to be operated in coordination with a single fuel cell electric vehicle with no particular issues.

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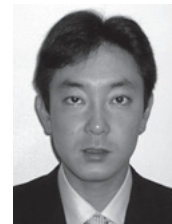
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Study of Low-engine-speed Self-ignition in Gasoline Engine Using Elementary Process Simulation

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ABSTRACT

The mitigation of noise originating in self-ignition in gasoline engines at low speed (“self-ignition noise” below) is an issue in realizing increased commercial appeal. Low-speed self-ignition noise occurs sporadically, producing significant torque fluctuation and engine vibration in each cycle. Synchronizing measurements with the occurrence of the noise and reproducing the phenomenon has therefore represented a challenge, and progress has not been made in its analysis. The research discussed in this paper verified that an elementary process simulation was able to model combustion conditions in which self-ignition noise occurs with sufficient accuracy. Using this simulation, the research studied phenomena occurring at the level of elementary processes when self-ignition occurs, changes in self-ignition characteristics with changes in conditions, and methods of ameliorating self-ignition. The conditions varied in the simulations were air-fuel mixture temperature, compression ratio, excess air ratio, EGR, residual gas and octane number. The results of these studies showed that reducing fuel density by increasing leanness is an effective means of controlling self-ignition when the engine is restarted. In addition, it was shown that it is possible to control self-ignition by reducing the temperature of the air-fuel mixture through reduction of the ratio of residual gas. Further, the simulation can be utilized to predict self-ignition characteristics produced by changes in factors such as the octane number of the fuel and the temperature of the external air in different regions, and to set appropriate specifications, on the desktop.

1. Introduction

Today, issues including the need to combat global warming demand the realization of increased efficiency in internal combustion engines. Methods including increasing the compression ratio, increasing the temperature of the cooling water, and boosted downsizing are available as methods of increasing efficiency in gasoline engines. Because these technologies increase the pressure and temperature of the combustion chamber, they also increase the possibility of abnormal combustion including knocking and pre-ignition due to factors such as high intake air temperature. As Fig. 1 shows, abnormal combustion can be classified into knocking, low-speed self-ignition, low-speed pre-ignition, and runaway pre-ignition. Because low-speed self-ignition results in significant torque fluctuations in each cycle, producing severe engine vibration, conducting stable measurements has represented a challenge to date, and progress has not been made in the analysis of the problem. Previous research^{(1), (2)} has considered changes in conditions and the timing of self-ignition, but has not mentioned the conditions for self-ignition noise.

In addition, very few studies have utilized elementary process simulations in the analysis of low-speed self-ignition. Research on self-ignition at close to idling speed conducted by Nakajima et al.⁽³⁾ elucidated the mechanism of low-speed self-ignition noise. Figure 2 shows the necessary combustion conditions for the occurrence of low-speed self-ignition noise. The engine speed is 750 rpm

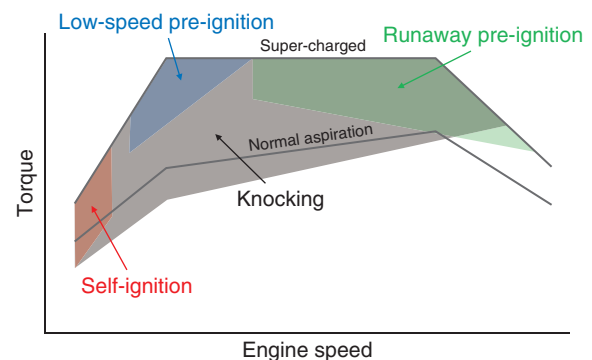


Fig. 1 Classification of abnormal combustion

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and the intake air pressure is atmospheric pressure. The horizontal axes of the graphs show the crank angle, while the vertical axes show mass burnt fraction (MBF), $dP/d\theta$, and in-cylinder pressure. At a crank angle of 0 deg, that is, close to TDC, a low-temperature oxidation reaction commences; at 37 deg ATDC, the remaining approximately 60% of the air-fuel mixture self-ignites, and at close to 40 deg ATDC, the angle at which $dP/d\theta$ is at its maximum, the pistons make contact with their sleeves and self-ignition noise occurs. This paper discusses research based on this mechanism that simulated the occurrence of low-speed self-ignition noise using elementary process simulations to verify the mechanism of occurrence of self-ignition, thus establishing guidelines for the development of countermeasures.

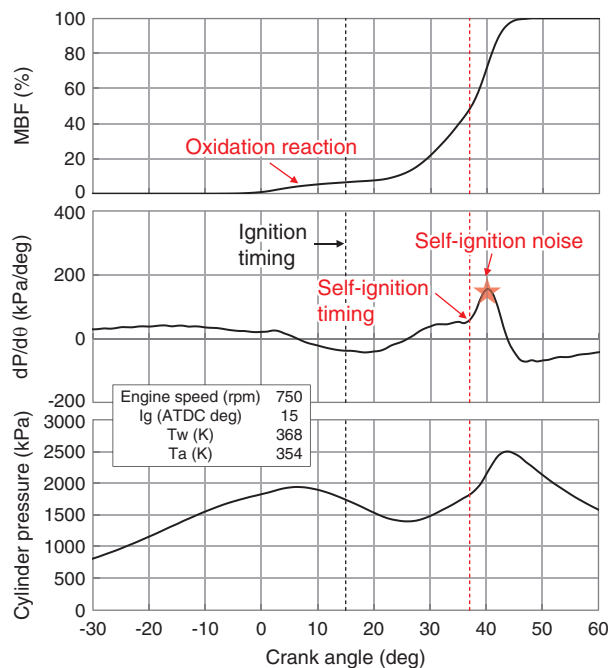


Fig. 2 Combustion generating self-ignition noise

2. Simulation Model of Low-speed Self-ignition

2.1. Application of Elementary Process Simulation

The challenge represented by conducting measurements in the case of low-speed self-ignition makes the use of simulations in conducting various studies a desirable method. Elementary process simulations are a type of simulation making it possible to obtain more detailed data than would be the case in an actual engine test. However, because elementary process simulations are zero-dimension simulations, they are unable to reproduce the propagation of flame by spark ignition. Knocking caused by the self-ignition of the remaining end gas due to the pressure generated by the combustion of the greater part of the air

Table 1 Fuel specifications (RON91)

n-Heptane	0.1823
iso-Octane	0.6642
Toluene	0.1535

Table 2 Engine specifications

Fuel	Gasoline	
Cylinder configuration	In-line 4-cylinder	
Displacement (cm ³)	1799	
Bore (mm)	81.0	
Stroke (mm)	87.3	
Compression ratio	10.5	
Crank offset (mm)	12.0	
Valve diameter (mm)	In.	32
	Ex.	26

fuel mixture as a result of spark ignition is an example of a case in which the application of elementary process simulation represents a challenge. However, because the combustion condition producing self-ignition is self-ignition of the majority of the fuel when the ignition timing is retarded, it is considered that the amount of combusted air-fuel mixture at the time is low, and the effect of factors such as combustion pressure due to spark ignition is limited. It was therefore considered possible to simulate actual tests using elementary process simulation.

2.2. Specifications of Elementary Process Simulation

The research discussed in this paper studied low-speed self-ignition using the type of elementary process simulation applied in simulations such as studies of homogeneous-charge compression ignition (HCCI). The specifications of the simulation were as follows:

- (1) Elementary process simulation
CHEMKIN-PRO (Ver. 2010/06/28)
- (2) Model: Ignition combustion (IC) engine

2.3. Surrogate Fuel

The properties of gasoline can be modeled by varying the ratios of n-heptane, iso-octane, and toluene in a surrogate fuel^{(4), (5)}. The ratio of components was designed based on measurement results to be as close as possible to the fuel used in engine tests. The ratio of components is shown in Table 1.

2.4. Engine Specifications

Table 2 shows the specifications of the engine used in the engine tests and the simulation.

2.5. Comparison of Elementary Process Simulation and Engine Tests

Figure 3 shows the results of engine tests and the elementary process simulation with adjusted parameters. For the tests, the engine speed was 750 rpm, intake pressure was atmospheric pressure, and the intake air

temperature was set at the lowest temperature at which self-ignition noise occurs. In the elementary process simulation also, the intake air temperature was the lowest temperature at which self-ignition occurs. The horizontal axes in Fig. 3 show the crank angle and the vertical axes show MBF, $dP/d\theta$, and in-cylinder pressure. Judging from the mass burnt fraction, a low-temperature oxidation reaction occurred from close to TDC in the engine tests, and the elementary process simulation reproduced this phenomenon. It was considered that in the engine test, after combustion proceeded up to 37 deg ATDC, the remaining approximately 60% of fuel self-ignited, and self-ignition noise occurred at close to 40 deg ATDC, when $dP/d\theta$ was at its maximum. A hot flame also occurred at close to 40 deg ATDC in the elementary process simulation. Therefore, at the lowest intake air temperature at which self-ignition occurs, the low-temperature oxidation reaction commences at almost exactly the same crank angle in the engine tests and in the elementary process simulation. In addition, self-ignition occurs and $dP/d\theta$ reaches its maximum in the engine tests and hot flame is generated in the elementary process simulation at almost exactly the same crank angle. Therefore, if the crank angle at which the hot flame occurs in the elementary process simulation is considered to be the crank angle at which $dP/d\theta$ reaches its maximum in the engine tests, that is, at which self-ignition noise occurs, in the elementary process simulation, the simulation can be considered to reproduce the engine test fairly well, despite its inability to model flame propagation by spark ignition.

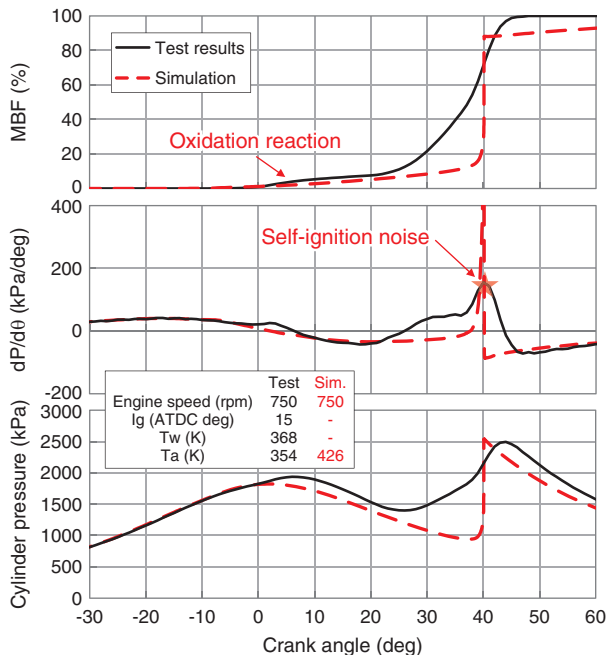


Fig. 3 Results of elementary process simulation

3. Verification of Correlation of Simulation with Engine Test

3.1. Effect of Intake Air Temperature

The correlation between the elementary process simulation with parameters set to make it possible to model combustion in which low-speed self-ignition noise occurs and the engine test was verified by varying parameters. Figure 4 shows the simulation results when the intake air temperature is varied. The horizontal axes show the crank angle with 0 deg as TDC, and the vertical axes show the mole concentration of the three components of the surrogate fuel and oxygen, and the temperature and pressure of the air-fuel mixture. As the compression stroke proceeds and temperature and pressure increase, n-heptane and iso-octane are oxidized, and the amount of oxygen declines. Toluene is not prone to oxidation, and at this stage the toluene is almost completely unoxidized. When the oxidation reaction proceeds further, the volume of fuel and oxygen declines rapidly and temperature and pressure increase rapidly. This crank angle is defined as the crank angle when self-ignition occurs. Because the oxidation reaction takes time when the intake air temperature is low, the engine speed at which self-ignition occurs is lower. Figure 5 compares the engine speeds at which self-ignition occurs in the elementary process simulation and the engine test when the intake air temperature is varied. There is a good

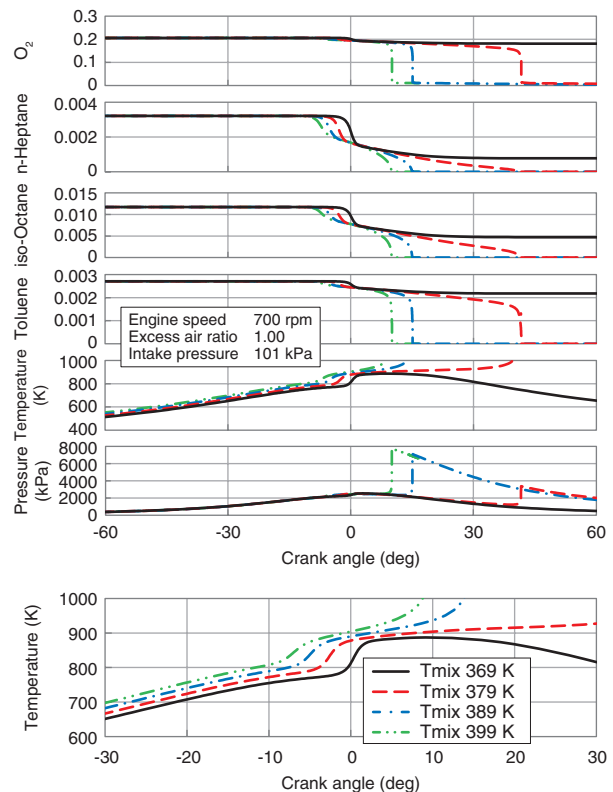


Fig. 4 Results of elementary process simulation

correlation between the engine test and the elementary process simulation.

3.2. Effect of Compression Ratio

Because the compression ratio changes pressure, it is considered to exert a significant effect on self-ignition. Figure 6 shows simulation results when the compression ratio is changed from 9.5 to 12.5. The higher

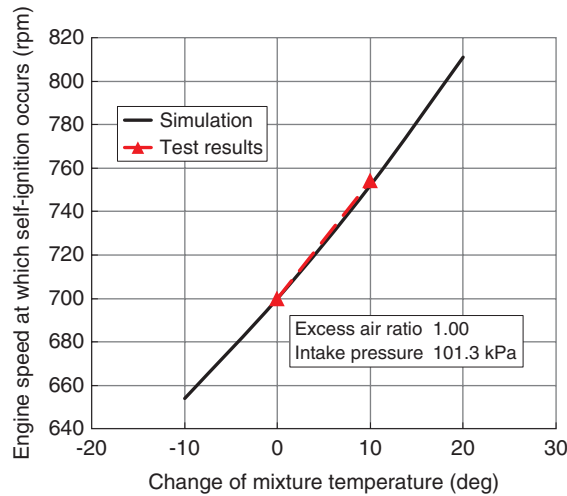


Fig. 5 Relationship between mixture temperature and self-ignition

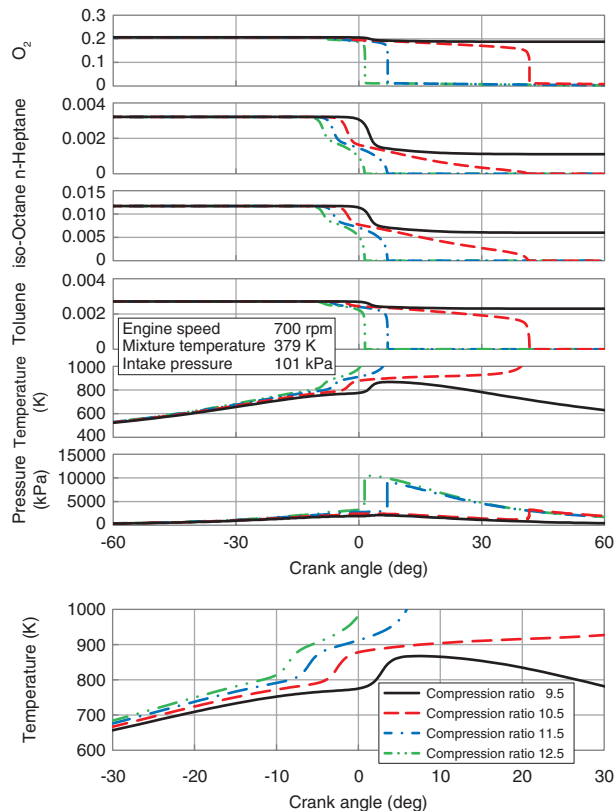


Fig. 6 Elementary process simulation of compression ratio change

the compression ratio, the earlier the commencement of the oxidization reaction and the earlier the occurrence of self-ignition. Figure 7 shows the relationship between the compression ratio and the engine speed at which self-ignition occurs. As when the intake air temperature was varied, there is an almost complete match between engine test results and the results of the elementary process simulation when the compression ratio is varied. This elementary process simulation model, able to reproduce engine test results when temperature and pressure, the important parameters for the elementary process, were varied, was employed to study the effect of operating conditions on self-ignition and countermeasures to apply to self-ignition.

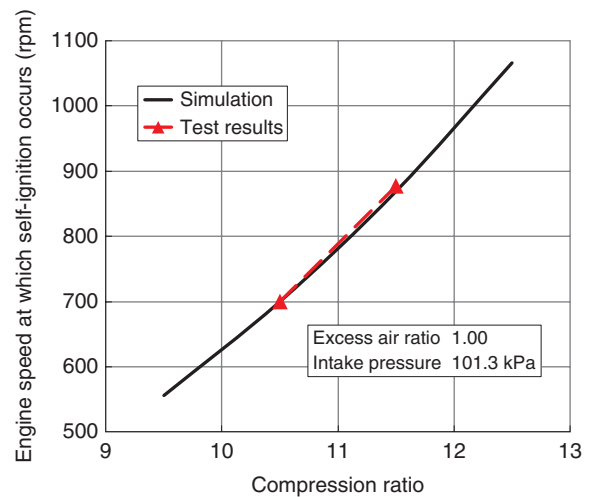


Fig. 7 Relationship between compression ratio and self-ignition

4. Mechanism of Low-speed Self-ignition

4.1. Effect of Excess Air Ratio

The excess air ratio is considered to have a significant effect on self-ignition characteristics. The richness or leanness of the mixture has been shown to have an effect in engine tests conducted under a limited range of conditions, but because it is challenging to conduct the tests, the mechanism of these effects has not been clarified. Figure 8 shows the relationship between the excess air ratio in engine tests and the intake air pressure under which self-ignition occurs and the torque produced under those conditions. While the intake air pressure at which self-ignition occurs is higher for both rich and lean conditions than for the theoretical air-fuel ratio, because rich mixtures produce a greater amount of torque for the same amount of intake air, the rich mixture produces a greater amount of torque. It is believed that the improvement in the self-ignition characteristic on the rich side is due to a significant cooling effect resulting from the latent heat of vaporization,

but the detailed causes of the decline on the lean side are unknown. A detailed study was therefore conducted using the elementary process simulation. Figure 9 shows the simulation results for the theoretical air-fuel ratio, $\lambda = 1.20$ and $\lambda = 0.83$. Factors such as the latent heat of vaporization and residual gas components were not considered. Table 3 shows the temperature of the air-fuel mixture before and after TDC.

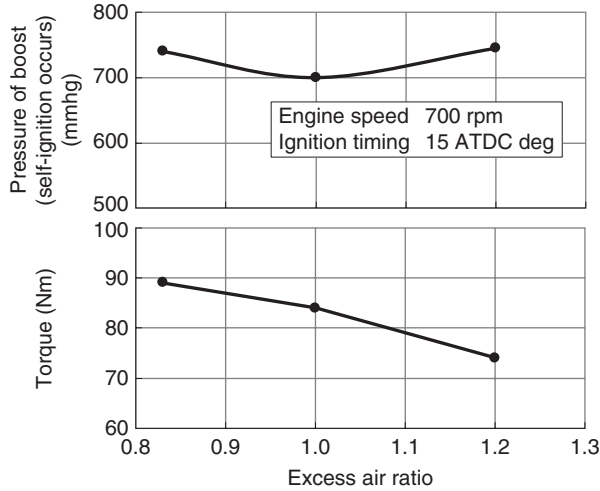


Fig. 8 Relationship between excess air ratio and pressure boost of self-ignition

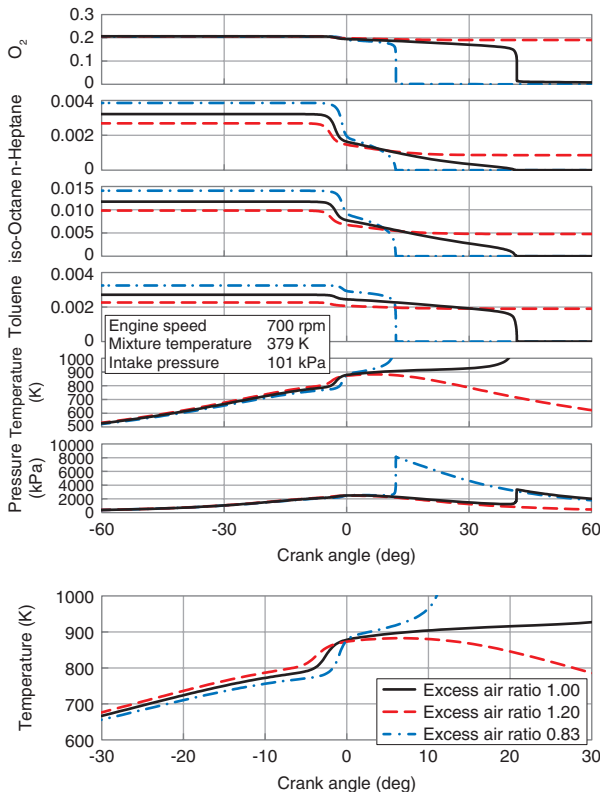


Fig. 9 Simulation results of change in excess air ratio

Table 3 Comparison of cylinder temperature

Excess air ratio		0.83	1.00	1.20
Mixture temperature (K)	BTDC 10 deg	753	772	788
	ATDC 10 deg	1000	905	880

Table 4 Conditions of simulation

Excess air ratio		0.83	1.00	1.20
Same intake pressure	Intake pressure	101	101	101
Same torque	(kPa)	84.4	84.4	101

(1) Temperature during compression

Because hydrocarbons are polyatomic molecules, their ratio of specific heat is low. Therefore, because the ratio of specific heat is higher at $\lambda = 1.20$, when the ratio of fuel is lower, the temperature during compression is also higher at this air-fuel ratio.

(2) Temperature following commencement of oxidization reaction

When the oxidization reaction begins, the situation is reversed, with the temperature becoming higher at $\lambda = 0.83$, when the ratio of fuel is higher, than at $\lambda = 1.20$.

In this case, it is the rich mixture, $\lambda = 0.83$, which is most prone to self-ignition. Figure 10 shows changes in the engine speed at which self-ignition occurs when the air-fuel ratio is varied. The black line shows results when the excess air ratio is varied for the same intake air pressure, and the red line shows results when the excess air ratio is varied for the same torque. Table 4 shows the conditions employed. Increasing the leanness of the mixture is effective in controlling self-ignition at the same intake air pressure, but promotes self-ignition at the same torque. This is believed to be because the effect of increasing the leanness of the mixture in controlling the oxidization reaction is overwhelmed by the effect of the increased pressure in promoting the oxidization reaction. The results of the simulation that did not consider the cooling effect

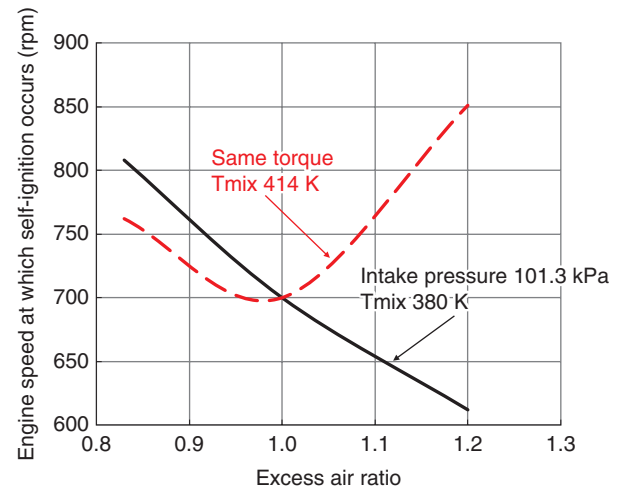


Fig. 10 Relationship between excess air ratio and self-ignition

of the latent heat of vaporization also showed that the rich mixture was more prone to self-ignition, and this is believed to be due to the fact that the ratio of fuel is higher and the temperature increase due to the oxidization reaction is greater.

4.2. Effect of EGR

Countermeasures against self-ignition utilizing the excess air ratio are acceptable as temporary measures, but constant application could result in increased emissions. Control of self-ignition by means of EGR was therefore studied as an alternative measure. In addition to possessing an identical effect to the other method in terms of diluting the fuel, EGR presented a greater advantage in having a low ratio of specific heat due to the high amounts of the three-atom molecules CO_2 and H_2O . Figure 11 shows results for the elementary process simulation. The temperature of the EGR is the same as that of the air-fuel mixture. Table 5 shows the temperature of the air-fuel mixture close to TDC when the fuel is diluted by increasing the leanness of the mixture and by EGR. The temperature of the mixture during compression is lower when EGR is employed than when the leanness of the mixture is increased. Figure 12 shows the relationship between the EGR rate and the engine speed at which self-ignition occurs. The black line shows results when the EGR rate is varied for the same intake air pressure, and the red line shows results when the EGR rate

Table 5 Comparison of cylinder temperatures

Excess air ratio	1.00	1.00	1.20
EGR ratio	0.0	0.167	0.0
Mixture temperature (K)	BTDC 10 deg	772	778
	ATDC 10 deg	905	853

Table 6 Conditions of simulation

EGR ratio	0	0.1	0.2
Same intake pressure	Intake pressure (kPa)	101	101
Same torque		81.1	90.1

Table 7 Fuel specifications

RON	87	91	95	100
n-Heptane	0.2247	0.1823	0.1602	0.1256
iso-Octane	0.6215	0.6642	0.5781	0.5183
Toluene	0.1538	0.1535	0.2617	0.3561
C/H ratio (weight)	5.6717	5.6724	5.9951	6.3145

is varied for the same torque. Table 6 shows the conditions used in the simulation. At the same torque, the effect of improvement in self-ignition due to EGR is minimal. It is believed that this is due to the fact that the intake air pressure increases in the percentage corresponding to the ratio of EGR, promoting the oxidization reaction and canceling the effect.

4.3. Effect of Octane Number

The phenomenon of self-ignition of the air-fuel mixture is known to be affected by the chemical structure of the fuel^{(6), (7)}. It was therefore considered that the octane number could provide an indicator for the self-ignition characteristics of the fuel. When changing the octane number of the surrogate fuel, it is necessary to change both the octane number and the C/H ratio to the correct figure. Table 7 shows the ratios of components in RON and the

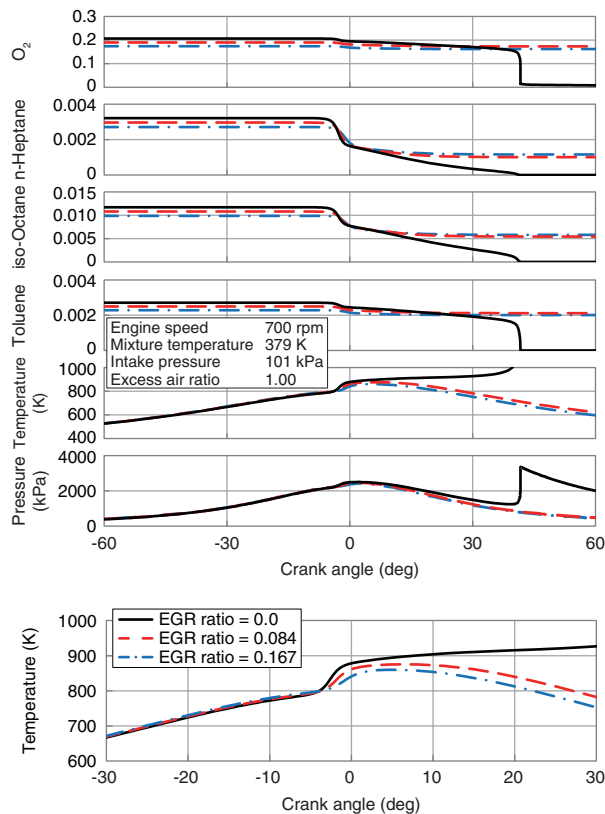


Fig. 11 Simulation results of EGR

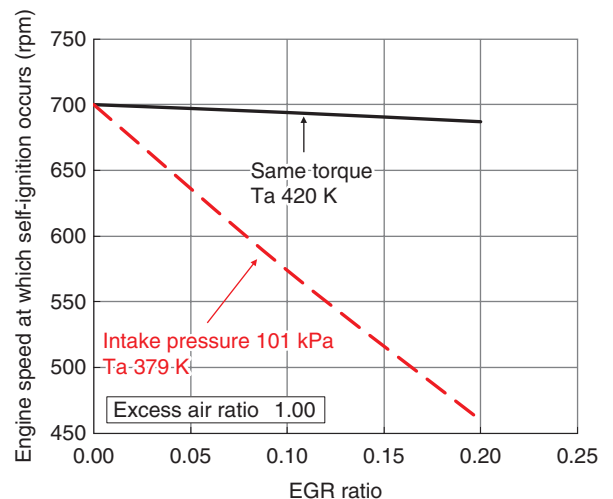


Fig. 12 Relationship between EGR ratio and self-ignition

surrogate fuels. Figure 13 shows the results of simulations conducted to study the differences between these fuels. There is a close relationship between the octane number and self-ignition characteristics, and this is mainly due to differences in the ratio of toluene, which remains almost entirely unoxidized in the low-temperature oxidation reaction. Figure 14 shows the relationship between RON and the engine speed at which self-ignition occurs. This shows that as in the case of knocking, RON provides an

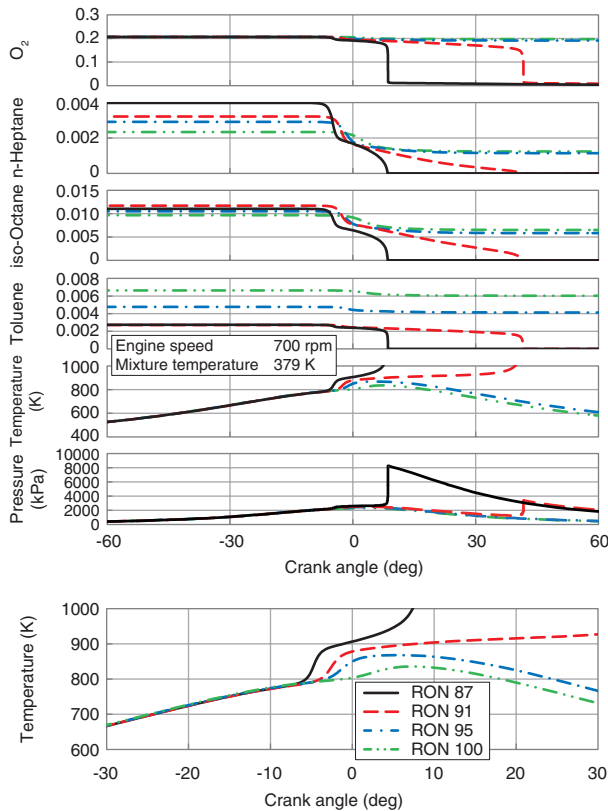


Fig. 13 Simulation of octane number

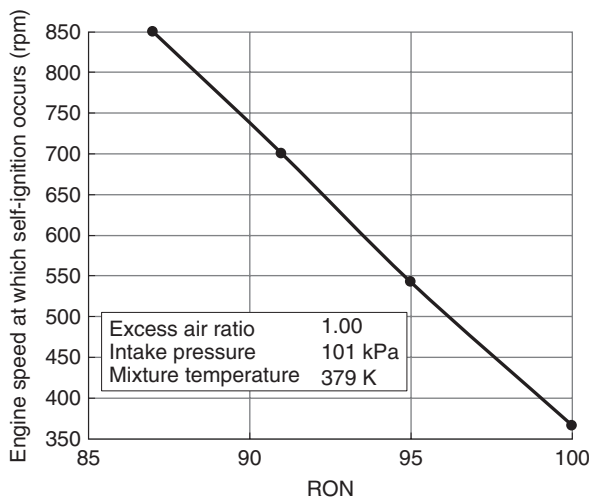


Fig. 14 Relationship between RON and self-ignition

indicator of the likelihood of self-ignition of the fuel.

From Figs. 4, 6, 9, 11, and 13, no matter which parameters are varied, the oxidation reaction will proceed even during the expansion stroke when the temperature of the air-fuel mixture exceeds 900 K, and hot flame is generated suddenly when it exceeds 950 K.

5. Countermeasures against Low-speed Self-ignition

5.1. Reduction of Residual Gas

When it was cooled to the same temperature as the air-fuel mixture, EGR displayed little effect in mitigating self-ignition. It was therefore considered that the negative effect of residual gas, a factor in increasing the temperature of the air-fuel mixture, would be high. An engine test was conducted using a cam profile that reduced valve overlap, as shown in Fig. 15. Figure 16 shows the relationship between intake air temperature and the ignition timing

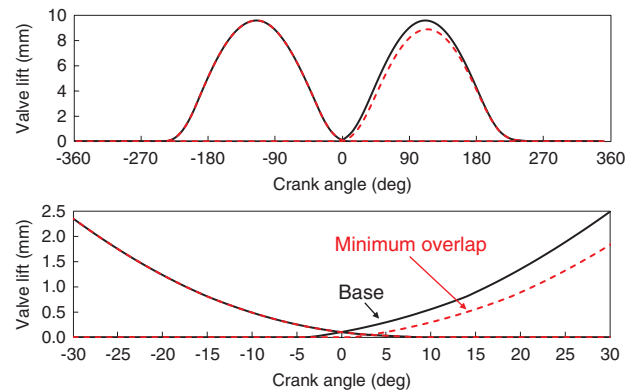


Fig. 15 Profile of reduced overlap cam

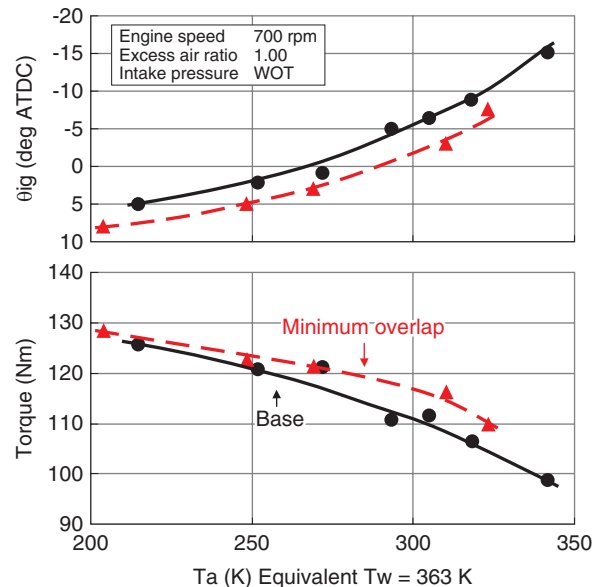


Fig. 16 Effect of residual gas reduction

immediately before the occurrence of self-ignition or knocking and the torque produced under these conditions. The self-ignition characteristic is improved across the entire temperature range, representing a maximum improvement of 8% when converted to torque. In order to conduct a detailed analysis of these test results, a simulation was conducted with the temperature difference between the residual gas and the air-fuel mixture set at 700 K, corresponding to wide-open throttle. Figure 17 shows the results of this simulation. When the amount of residual gas increases, the temperature of the air-fuel mixture increases, and the oxidization reaction commences earlier. Figure 18 shows the relationship between the ratio of residual gas and the engine speed at which self-ignition occurs. The black line shows results obtained for an assumed temperature difference of zero between the residual gas and the air-fuel mixture, and the red line shows results for an assumed temperature difference of 700 K. The residual gas has a greater effect in promoting self-ignition due to the increase in the temperature of the air-fuel mixture than the effect of EGR in controlling self-ignition. Based on these results, it was considered that the factor behind the control of self-ignition when the valve overlap was reduced was the reduction of the temperature of the air-fuel mixture by the reduction of the amount of residual gas. In addition, it was conjectured that there was almost no promotion of self-ignition by increase in the density of the fuel.

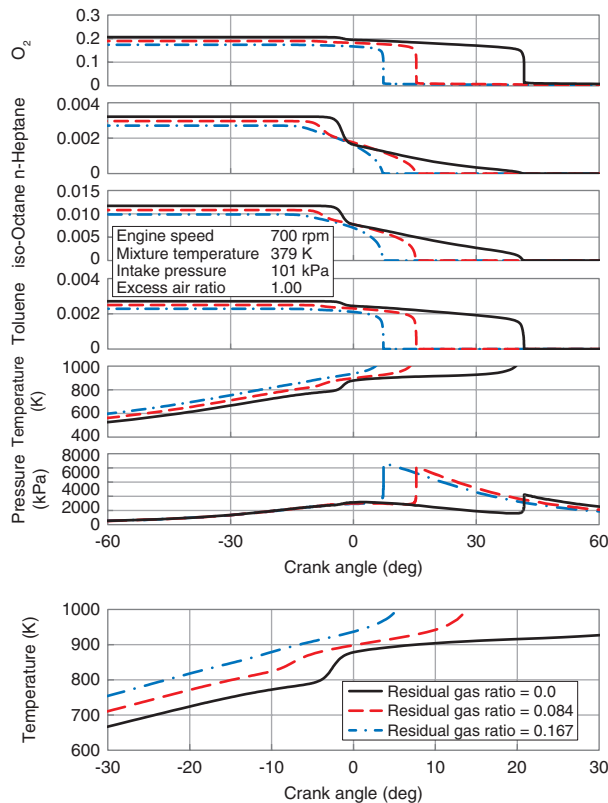


Fig. 17 Simulation of residual gas ratio change

5.2. Controlling Self-ignition when the Engine is Restarted

Self-ignition when the engine is restarted represents an important issue, for example in engines employing idle-stop mechanisms, which are becoming more widespread. The conditions prevailing when the engine is restarted are

- (1) High in-cylinder temperature
- (2) Intake air pressure: Atmospheric pressure
- (3) Low friction

A reduction in torque to a degree that did not have a major effect on the increase in engine speed would therefore be allowable as a measure to attempt to prevent self-ignition. Increasing the leanness of the excess air ratio, which had been effective in controlling self-ignition at the same intake air pressure, was therefore considered as an improvement measure. For the basic simulation conditions, intake air temperature was adjusted with consideration of changes in charging efficiency to make it possible to obtain results corresponding to those obtained from engine tests. Figure 19 shows a comparison of results for 250 rpm, corresponding to engine restart, and for 700 rpm, corresponding to idling. At 250 rpm, corresponding to engine restart, the crank angle at which the oxidization reaction commences is retarded, but self-ignition occurs because the engine speed is low and the time for the compression stroke is extended. The temperature and pressure at which self-ignition occurs differ, but the states of oxidization of the three components of the surrogate fuel are almost identical. Figure 20 shows results for the simulation when the excess air ratio is made leaner. Because the engine speed for restart is set, these simulation results show the air-fuel temperature at which self-ignition occurs. At the appropriate leanness for engine restart, the air-fuel mixture temperature at which self-ignition occurs increases, and self-ignition is controlled.

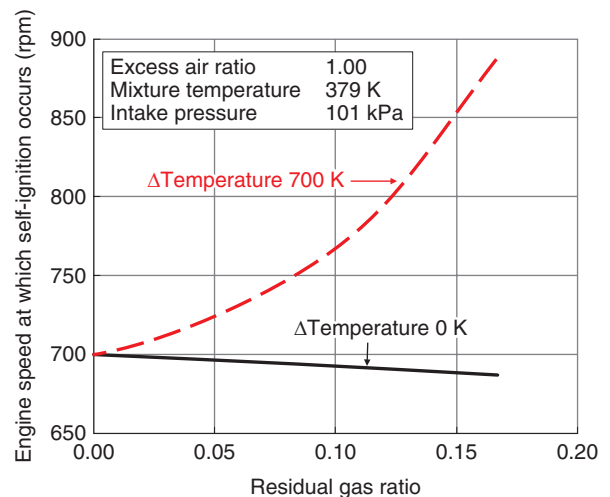


Fig. 18 Relationship between residual gas ratio and self-ignition

5.3. Prediction by Application of Simulation Results

By combining various simulation results, it is possible to simulate factors such as how self-ignition characteristics change in response to such changes as the country of sale and the type of gasoline used. For example, combining simulation results for the effect of the compression ratio and the effect of RON makes it possible to estimate the compression ratio to be employed in order to achieve the same self-ignition characteristic to an RON of 91 and

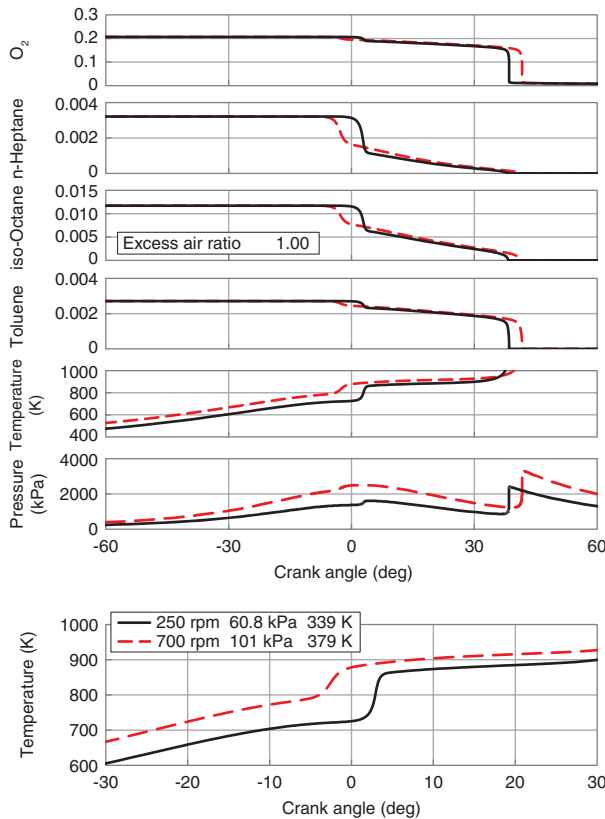


Fig. 19 Simulation at engine restart

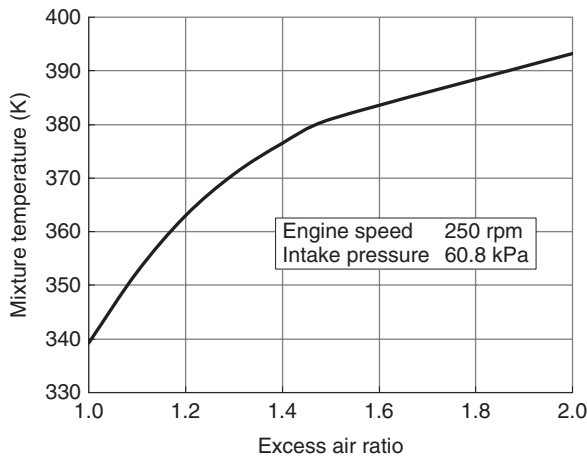


Fig. 20 Relationship between excess air ratio and self-ignition

a compression ratio of 10.5 when the RON is changed, as shown in Fig. 21. This is merely one example. The simulations will be effective in such procedures as conducting desktop studies of changes in specifications in relation to changes in use conditions in different regions.

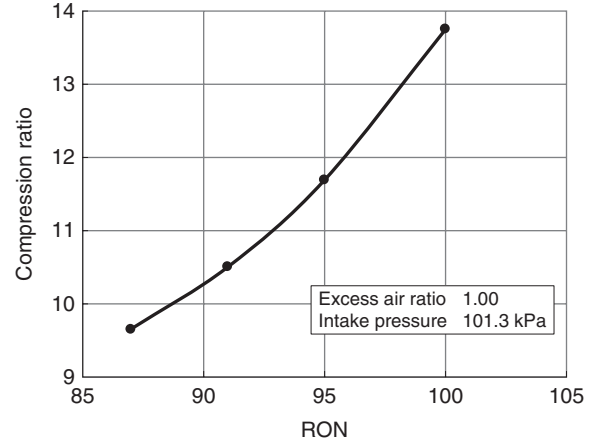


Fig. 21 Simulation of relationship between RON and compression ratio

6. Conclusion

An elementary process simulation model capable of modeling combustion conditions in which low-speed self-ignition noise occurs has been formulated and employed in analyses, producing the following findings.

- (1) Simulation of combustion producing low-speed self-ignition noise

Using CHEMKIN-PRO, it was possible to simulate measurement results for the timing of the commencement of low-temperature oxidation reactions and self-ignition and the changes in self-ignition characteristics with changes in parameters. This model was employed in conducting a variety of studies of low-speed self-ignition for which actual measurements would represent a challenge.

- (2) Mechanism of self-ignition

The self-ignition that occurs close to 40 deg ATDC and is the cause of low-speed self-ignition noise results from the following mechanism: A low-temperature oxidation reaction commences due to the increase in temperature and pressure during compression; there is a further increase in temperature, and when the temperature of the air-fuel mixture exceeds 900 K, the oxidation reaction proceeds irreversibly, and at a temperature in excess of 950 K, the reaction proceeds in a burst and a hot flame is produced. The reduction of the temperature and pressure of the air-fuel mixture can be considered as a general orientation for the control of self-ignition. For the same amount of air, the reduction

of the concentration of fuel in the mixture by the excess air ratio or EGR can be indicated. In addition, because the readily oxidizable straight-chain components of the fuel, as typified by n-heptane, are reduced in volume when the RON increases, the tendency towards self-ignition declines. RON can be considered a good indicator of the tendency of a fuel to self-ignite.

(3) Method of control of self-ignition

Reduction of the ratio of residual gas can be employed as a method of reducing the temperature of the air-fuel mixture. The elementary process simulation showed that the effect of this reduction in temperature exceeds the increase in the temperature of the mixture during the low-temperature oxidization reaction due to the increase in the concentration of the fuel. In addition, at engine restart, because the intake air pressure is basically atmospheric pressure, it is acceptable to reduce torque by using excess air, and the increased leanness of the mixture is effective in controlling self-ignition.

(4) Usefulness of simulation in studying specifications

Combining the results of various simulations makes it possible to simulate the ways in which self-ignition characteristics will change in response to factors such as different areas of sale and changes in the gasoline employed, making the simulation useful in studying changes in specifications.

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■ Author ■



Susumu NAKAJIMA

Elucidation of Deterioration Mechanism and Improvement of Durability for Lean NOx Catalyst

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ABSTRACT

The exhaust gas temperatures of diesel engines have been falling gradually due to engine efficiency improvement and body weight reduction. Thus, an NOx aftertreatment system that has high NOx conversion at lower temperature is required for these engines. To meet demand, the development of a new Lean NOx Catalyst with a new De-NOx mechanism has been carried out. However, it has recently been found that Lean NOx Catalyst has a durability issue during aging in an actual exhaust gas environment. The deterioration mechanism of Lean NOx Catalyst was clarified and durability enhancement of the material was conducted. As a result of an analysis using a combination of surface analysis and ab-initio calculation, it was found that the sulfur in exhaust gas was irreversibly adsorbed onto platinum in the form of sulfate, which reduced catalyst activity. A new Lean NOx Catalyst that has high durability even after aging with exhaust gas was developed by improving NO₂ adsorption activity and NO₂ reduction activity, which had been degraded by sulfur poisoning.

1. Introduction

Diesel engines are promising as low fuel consumption engines because of their high thermal efficiency. In order to comply with strict emission regulation in America, Japan, Europe, and other places, exhaust gas after-treatment technology that provides high performance and high efficiency is required. Since the exhaust gas of a diesel engine is oxygen-rich atmosphere (lean), purification of the nitrogen oxide (NOx) in the exhaust gas is an issue, and various NOx purification systems have been proposed (Fig. 1).

One system is the Urea Selective Catalytic Reduction^{(1),(2)} (Urea-SCR) NOx purification system. Urea-SCR is a system that uses aqueous urea to purify NOx by an NH₃-selective reduction reaction, and it has high NOx catalytic reduction performance. However, Urea-SCR involves such issues as high system costs because of urea tank, injector, and other added devices, high weight, and so on. When exhaust gas temperature is low, an increase in exhaust gas temperature is needed to convert urea water to NH₃.

By contrast with this, the NOx Storage Catalyst (NSC) system has been developed⁽³⁾⁻⁽⁷⁾ as a simple, low-cost system that does not involve added devices. In an NSC system, after the NOx in exhaust gas is stored in an alkaline earth metal or other such material in the catalyst, the stored NOx

is purified by such reducing agents as carbon monoxide (CO) and hydrocarbon (HC), which are produced by rich control. However, the alkaline earth metal that is mainly used as an NOx storage material needs a temperature of approximately 300°C to store NOx. In order to obtain high catalytic reduction performance at low temperatures, large amounts of precious metals are required.

In recent years, exhaust gas temperatures of diesel engines have been decreasing gradually due to the engine efficiency improvement and body weight reduction. Therefore, the Urea-SCR system and the NSC system face the issues of catalytic reduction performance at low temperatures. Thus, an NOx aftertreatment system that has high NOx conversion at lower temperatures and offers low cost is required.

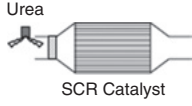
	System configuration	Equation
Urea-SCR	 SCR Catalyst	$4\text{NO} + 4\text{NH}_3 \rightarrow 4\text{N}_2 + 6\text{H}_2\text{O}$ $6\text{NO}_2 + 8\text{NH}_3 \rightarrow 7\text{N}_2 + 12\text{H}_2\text{O}$ $4\text{NH}_3 + 2\text{NO} + 2\text{NO}_2 \rightarrow 4\text{N}_2 + 6\text{H}_2\text{O}$
NSC	 NSC Catalyst	Lean: $\text{NO} + 1/2\text{O}_2 + \text{MO} \rightarrow \text{MNO}_3$ Rich: $\text{MNO}_3 + 2\text{CO} \rightarrow \text{MO} + 2\text{CO}_2 + 1/2\text{N}_2$ $\text{MNO}_3 + 2\text{H}_2 \rightarrow \text{MO} + 2\text{H}_2\text{O} + 1/2\text{N}_2$ $9/2\text{MNO}_3 + \text{C}_3\text{H}_6 \rightarrow 3\text{CO}_2 + 3\text{H}_2\text{O} + 9/2\text{MO} + 9/4\text{N}_2$

Fig. 1 Conventional NOx reduction technology

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** N. E. CHEMCAT CORPORATION

A Lean NOx Catalyst (LNC) system⁽⁸⁾ that has high catalytic reduction performance at low temperatures using a new purification concept has been proposed by Honda. The LNC system is like the NSC system in the respect that additional equipment is unnecessary and rich spike control is necessary to purify NOx. Whereas the NSC approach has issues with NOx purification at low temperatures, the LNC approach provides high catalytic reduction performance at low temperatures by means of a new NOx mechanism of catalytic reduction.

On the other hand, automotive catalyst is required to be highly durable under severe exhaust gas environment conditions, and durability is also an important issue for LNC. In the course of confirming LNC durability, it became clear that NOx catalytic reduction performance diminished when the LNC was aged under an exhaust gas environment. This report therefore considers the causes of LNC deterioration, sets forth methods for countering it, and describes the beneficial effects of those methods.

2. LNC Characteristics and Durability Issues

2.1. LNC Characteristics

LNC yields high NOx catalytic reduction performance at low temperatures by means of a new NOx mechanism of catalytic reduction. The mechanism of NOx purification by the LNC and details of its NOx purification performance have been previously reported⁽⁸⁾. The mechanism of NOx reduction with LNC is shown in Fig. 2. The LNC has a two-layered configuration, comprising a layer employing a NOx adsorption material, which possesses NOx adsorption and reduction functions, and a layer employing zeolite, which functions to reduce NOx by means of NH₃. This two-layered configuration realizes efficient NOx purification. With regard to the mechanism of NOx purification, first, NO is oxidized to NO₂ by Pt in the NOx adsorption material layer, and the NO₂ adsorbs on the NOx adsorption material. Following this, the NO₂ desorbs from the NOx adsorption material when the exhaust gas becomes rich, and is converted to N₂ or NH₃ on the Pt and the NH₃ adsorbs on the zeolite layer. In the lean phase after that, the NH₃ in the zeolite layer purifies NOx. As NOx adsorption material, the catalyst uses cerium (Ce) compound, which has high NOx

Table 1 Model gas composition for LNC and NSC

	Lean (55 sec)	Rich (5 sec)
O ₂ (%)	6	0
CO (%)	0.09	2
C ₃ H ₆ (ppmC)	500	
CO ₂ (%)	6	
NO (ppm)	110	
H ₂ O (%)	7	
N ₂ (%)	Balance	
GHSV (h ⁻¹)	50000	

adsorption performance from low temperatures, and also uses zeolite, which is capable of NH₃-selective reduction even at low temperatures. This NOx mechanism of catalytic reduction enables the LNC catalyst to yield high catalytic reduction performance from low temperatures. Figure 3 shows a comparison⁽⁸⁾ of the purification performance of conventional NSC and LNC. Table 1 further shows the test conditions that were used in the comparison. The results show that where the peak of NSC catalytic reduction performance occurs at 400°C, LNC has its peak at 250°C and possesses high catalytic reduction performance at low temperatures. Thus LNC is a catalyst system that yields high NOx catalytic reduction performance from low temperatures by means of a new NOx mechanism of catalytic reduction.

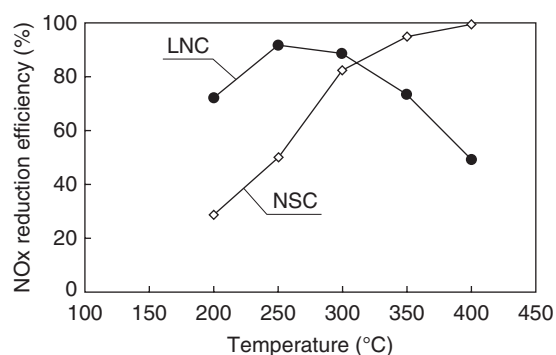


Fig. 3 Comparison of LNC and NSC in NOx reduction efficiency

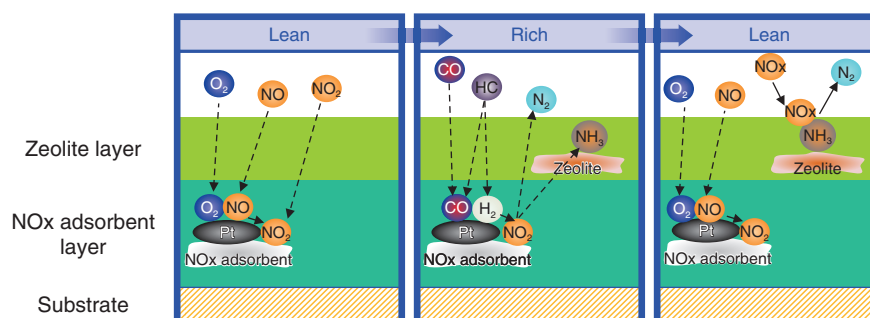


Fig. 2 Reaction mechanism of LNC

2.2. NOx Emission Behavior in Vehicles Testing

Figure 4 shows the configuration of an emission gas purification system that uses LNC. This system places a diesel oxidation catalyst (DOC) and catalyzed diesel particulate filter (cDPF) in a position close coupled with the engine, and places the LNC under the floor. In this system, CO and HC are purified by the DOC, soot is removed by the cDPF, and NOx is purified by the LNC. The durability of this emission gas purification system was verified by carrying out emission testing with actual vehicles using aging catalyst in FTP-75, which is a US emission test mode. Two types of aging were conducted. One was engine emission gas aging (engine aging), in which the system was placed downstream of the engine emission gas and controlled to bring the LNC to a certain temperature. The other was aging by an electric furnace to verify only the influence of heat on the catalyst (thermal aging). For both methods, the heat load was made equivalent to both aging temperature and time at 120000 miles of operation. The gas conditions for the thermal aging were oxygen 10% and water 6%.

Figure 5 shows a comparison of NOx emission behavior of thermal aging catalyst and engine aging catalyst in FTP-75 mode. The thermal aging catalyst shows virtually zero NOx emissions following the warm-up time of approximately 200 seconds after start-up. The engine aging catalyst, however, shows NOx emissions throughout the entire mode procedure. The amount of NOx emissions was particularly large at the rich spike times marked by arrows in Fig. 5. It can be inferred that this occurs because the NOx desorbed from the adsorption material during the rich spikes is emitted without undergoing full reduction, and it appears that NOx catalytic reduction performance is reduced under rich conditions. Furthermore, the engine

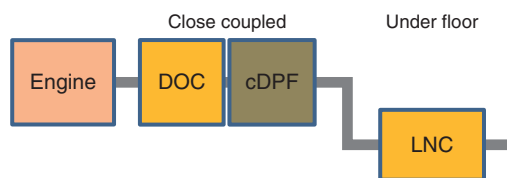


Fig. 4 Exhaust aftertreatment system using LNC
Engine: 2.2 L direct-injection diesel, close coupled catalysts: DOC 1.0 L, CSF 2.2 L, under floor catalyst: LNC 2.0 L

Table 2 Model gas composition

	Lean (55 sec)	Rich (5 sec)
O ₂ (%)	6	0
C ₃ H ₆ (ppmC)	0	4000
CO (%)	0	1
CO ₂ (%)	10	
NO (ppm)	200	
H ₂ O (%)	7	
N ₂ (%)	Balance	
GHSV (h ⁻¹)	75000	

aging catalyst shows NOx emissions even in the high gas flow rate region due to high acceleration at around 1600 seconds, so it appears that NOx catalytic reduction performance is also reduced under lean conditions.

2.3. Elementary Reaction Characteristics in Model Gas Testing

Next, the LNC used in the emission test was made into a test piece and a catalytic reduction performance test was carried out using the model gas exposure test system manufactured by Horiba in order to clarify the deterioration characteristics of the catalyst. Catalyst performance was evaluated by alternately passing lean gas and rich gas through the LNC. The average purification rate for lean and rich gases was used as a criterion in these evaluations. Figure 6 shows the model gas testing results. Table 2 shows the test conditions. The fresh catalyst achieves a

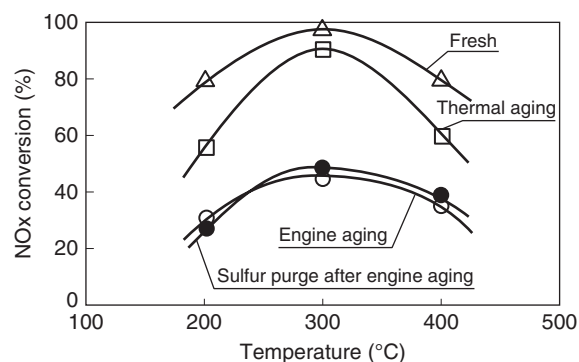


Fig. 6 Comparison of NOx conversion between thermal aging and engine aging in model gas test

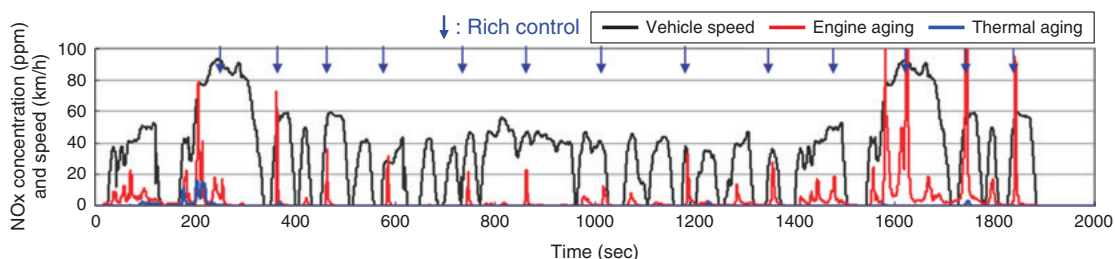


Fig. 5 Comparison of NOx emission profile between thermal aging and engine aging in FTP-75 mode

Table 3 Performance in elementary reactions

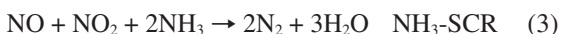
	Thermal aging (%)	Engine aging (%)	Material	Model gas composition
(1) NO oxidation	52	68	Pt	NO 500 ppm, O ₂ 0.5 %, N ₂ balance
(2) NO ₂ adsorption	75	45	Pt, Ce-compound	NO ₂ 500 ppm, O ₂ 0.5 %, N ₂ balance
(3) NH ₃ -SCR	71	69	Zeolite	NO 100 ppm, NO ₂ 100 ppm, NH ₃ 200 ppm, O ₂ 6%, H ₂ O 7%, N ₂ balance
(4) NO ₂ reduction	98	32	Pt	NO ₂ 500 ppm, CO 1000 ppm, N ₂ balance

high purification rate of around 80% at every temperature, but the purification rate shows a decline for both aging catalysts. The purification rate of the engine aging catalyst in particular is low, showing significant deterioration with a decline of from 30% to 50%. In addition, it is known that the type of catalyst that adsorbs or stores NO_x accumulates sulfur from the exhaust gas, and this causes a reduction in NO_x catalytic reduction performance. Therefore the engine aging catalyst was subjected to sulfur removal treatment by rich gas at 650°C for 10 minutes to investigate whether the diminished NO_x catalytic reduction performance would recover. The results show that catalytic reduction performance was largely unchanged, suggesting that this was not ordinary deterioration due to sulfur accumulation.

This suggested that NO_x purification performance of engine aging LNC was due to some irreversible factor other than heat.

LNC is a catalyst in which the NO_x purification reaction proceeds by a combination of elementary reactions. Therefore engine aging catalyst and thermal aging catalyst were subjected to model gas testing for the elementary reactions in Eq. (1) through (4). These tests were used to determine which elementary reactions performance was declining and which materials this performance decline originated. Table 3 shows the test conditions and test results. According to these results, the NO oxidation reaction in Eq. (1) and the NH₃-SCR reaction in Eq. (3) had generally the same performance with both forms of aging, but the NO₂ adsorption reaction in Eq. (2) and the NO₂ reduction reaction in Eq. (4) showed diminished performance with the engine aging catalyst. Since platinum (Pt) or the cerium (Ce) compound is related to these performance parameters, it was considered that the functions of these materials were declining.

Lean phase



Rich phase



3. Elucidation of LNC Deterioration Mechanism

In order to clarify the causes of LNC deterioration, analyses of the platinum and the cerium compound considered to be declining in performance were conducted. Morphological analysis was conducted by surface analysis and configuration simulation using ab-initio calculation was carried out to obtain theoretical corroboration.

3.1. Clarification of Deterioration Configuration by Surface Analysis

The effects of thermal deterioration in platinum and the cerium compound were first investigated. It is known that crystal growth in Pt and the cerium compounds occurs. X-ray diffraction (XRD) was therefore employed in order to measure crystallite diameter, an index of agglomeration. The results show that crystallite diameter in both platinum and cerium compound had generally the same values under both thermally aging catalyst and engine aging catalyst conditions. This result suggested that there was no difference in degradation due to heat for the different types of aging. Next, the influence of poisoning deterioration due to the accumulation of sulfur (S) and phosphorus (P) of exhaust gas was investigated. Inductively Coupled Plasma-Atomic Emission Spectrometry (ICP-AES) was used to quantify the amounts of sulfur and phosphorus accumulated on the engine aging catalyst. As a result, the amounts were found to be very small, with sulfur at 0.02% and phosphorus at 0.02%. This is the equivalent, converted from fuel and oil consumption, of the amounts from driving several tens of kilometers. However, it was conceivable that very small amounts of poisoning components were adhering to certain specific areas, and so the state of the poisoning components in the platinum and the cerium compound was investigated. The engine aging catalyst was first examined using a Scanning Transmission Electron Microscope (STEM). This identified platinum particles and cerium compound particles, and the various particles were subjected to element analysis by Energy Dispersive X-ray Spectrometry (EDS) to investigate where the poisoning substances were located. Figure 7 shows the EDS results. A peak of sulfur at around 2.3 keV was detected in the EDS

spectrum for the platinum particles in Fig. 7(a), indicating that platinum and sulfur existed together. In Fig. 7(b), however, no sulfur was detected in the particles of cerium compound, indicating that cerium compound and sulfur did not exist together. This suggested that sulfur was poisoning the platinum particles, and it was inferred that sulfur was being selectively adsorbed onto the surface of the platinum particles.

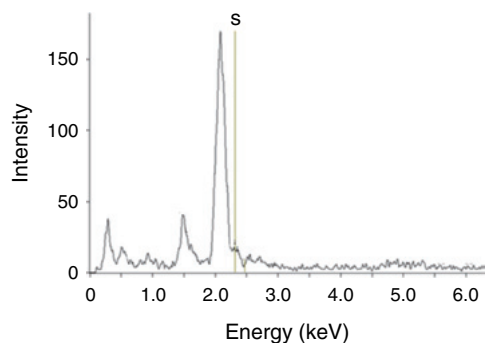
In the case of selective adsorption of sulfur onto the surface of platinum particles, a reduction in the active surface area of the platinum particles can be expected. Therefore the amount of CO adsorption was measured by Fourier Transform Infrared Ray spectroscopy (FT-IR), which is a method for measuring the active surface area of a metal, and the change in the active surface area due to deterioration was investigated. Figure 8 shows the results. The peak at around 2090 cm^{-1} can be identified as the peak of Pt-CO adsorption. As a result, it was found that the amount of Pt-CO adsorption by the engine aging catalyst had become smaller than that by the thermally aging catalyst, and that the active surface area of the platinum particles had grown smaller. The results of XRD analysis described above showed that platinum crystallite diameter was the same under the two types of aging, so that the extent of platinum particle aggregation is thought to be about the same. Therefore this reduction in the active surface area is considered to be due to the adsorption of

sulfur on the surface of the platinum particles.

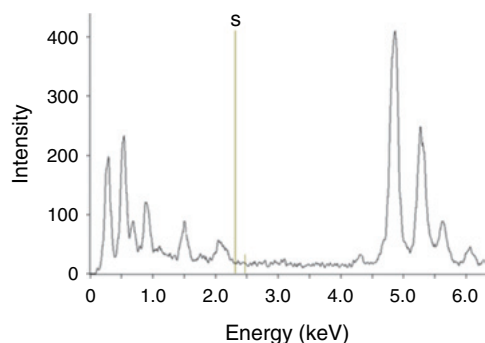
A structural analysis of surface chemical species was then carried out by X-ray Absorption Near Edge Structure (XANES) in order to investigate the structure of sulfur covering the surface of platinum particles. A structural analysis of the sulfur in engine aging catalyst was carried out by comparison with calcium sulfide (CaS) and calcium sulfate (CaSO_4), for which the structure was already known. Figure 9 shows the sulfur K-edge XANES spectra. It was apparent that the engine aging catalyst peaks at around 2.48 keV, resembling the CaSO_4 spectrum. This suggested that the sulfur on the surface of the platinum particles is adsorbed in the form of a sulfate (SO_4).

3.2. Verification of Deterioration Configuration by Ab-Initio Calculation

Ab-initio calculation relating to the bonding of platinum and sulfur was carried out in order to obtain theoretical



(a) Pt particle



(b) Ce-compound particle

Fig. 7 EDS patterns of catalysts under engine aging

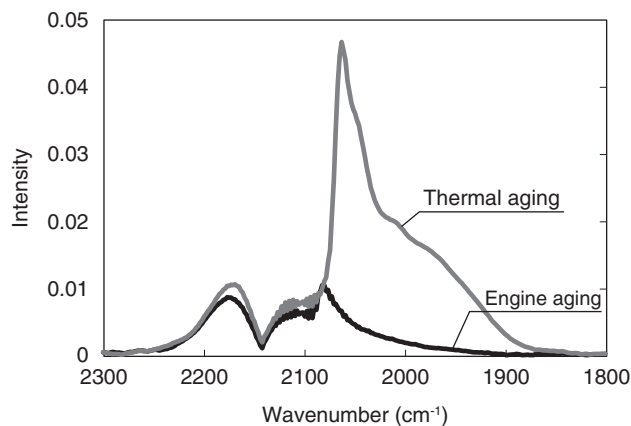


Fig. 8 FT-IR spectra of CO adsorption
Temperature: 300°C, gas condition: CO
1000 ppm, He balance

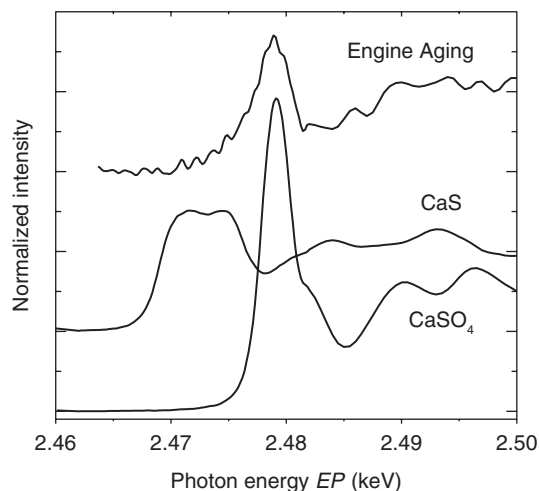
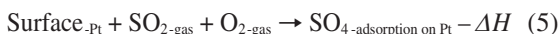


Fig. 9 XANES spectra of various samples

Table 4 Condition for ab-initio calculation

Item	Contents
Program	D-Mol3 ⁽⁹⁾
Functional	VWN (LDA) ^{(10), (11)}
Basic function	DND basis set
Inner core calculation	ECP (Effective core potential)
Model	Pt (111) surface with periodic boundary condition

verification of the structure of sulfur adsorption described in the previous section. A model was fabricated of a cluster of 12 platinum atoms with 111 facets exposed on the surface of which one sulfur oxide (SO_x) molecule (S, SO, SO₂, SO₃, SO₄, SO₅) is adsorbed. This was examined with the adsorption enthalpy in a reaction equation such as that in Eq. (5) (taking the case of SO₂ as an example) as the index of stability.



When the adsorption enthalpy (ΔH) derived by this calculation is smaller, then the state of SO_x adsorption on the platinum is thought to be more stable. Figure 10 shows the results of this calculation, and Table 4 shows the conditions for the calculation. As the number of oxygen (O) atoms increases with respect to the sulfur, the adsorption enthalpy is reduced, reaching its lowest point with SO₄ and then increasing again with SO₅. From this it was inferred that SO₄ is the most stable form of sulfur poisoning of platinum. The adsorption enthalpy with SO₄ on CeO₂ is also shown for purposes of comparison, but the adsorption enthalpy with platinum is even smaller than with CeO₂, suggesting that SO₄ is more strongly adsorbed onto platinum than onto CeO₂. This suggests that the reason there was no recovery of catalytic reduction performance in Fig. 6 even when sulfur removal treatment was carried out was that the SO₄ poisoning of the platinum

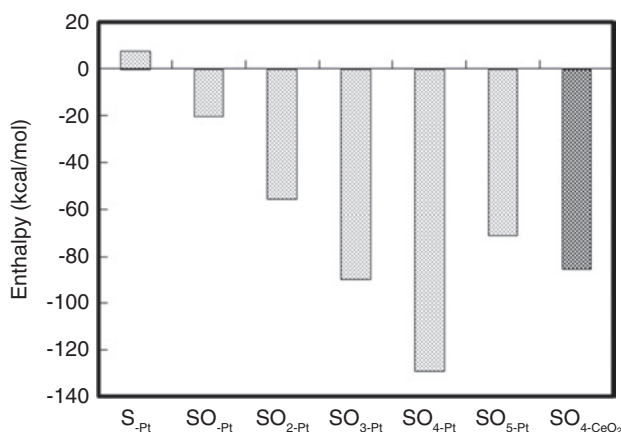


Fig. 10 Comparison of adsorption energy among various sulfur oxides on Pt and CeO₂

was so stable that the sulfur was not desorbed. Platinum poisoning by SO₄ can be considered to cause irreversible deterioration in the temperature range of actual diesel operation. Figure 11 shows a conceptual image of the structure of SO₄ bonded to platinum as derived from ab-initio calculation. The poisoning of platinum by SO₄ takes the structure of three oxygen atoms bonding each to one of three platinum atoms. One SO₄ molecule therefore poisons three atoms on the platinum surface, so that even a small amount of SO₄ can be expected to have a significant influence.

The results of surface analysis and ab-initio calculation considered in combination suggest that the causal factor in deterioration of LNC performance due to engine aging is that the platinum particle surface, which is an activation site, is poisoned by the strong adsorption structure of SO₄. This appears to be the case even though the amount of sulfur poisoning is very small.

4. Countermeasures to LNC Deterioration

4.1. Conceptual Approach to Deterioration Countermeasures

This suggested that the degradation of the LNC due to engine aging resulted from the covering of the Pt by SO₄. The ab-initio calculation suggests that the state of SO₄ adsorption on the platinum surface is thermodynamically very stable. It seems to be difficult to prevent SO₄ poisoning of the catalyst, it was therefore considered that the introduction of modifications to reduce the effect of SO₄ poisoning on performance would be desirable. In Section 2, the reactions that resulted in a decrease in function due to SO₄ poisoning were shown to be the NO₂ adsorption reaction in Eq. (2) and the NO₂ reduction reaction in Eq. (4). Therefore catalyst improvement that would lessen the impact of SO₄ poisoning on these functions was investigated. Figure 12 shows the reaction mechanism in deteriorated LNC. First, in the NO_x adsorption reaction under lean conditions, the Eq. (1) reaction that forms NO₂ from NO progresses even on the platinum that has been poisoned. However, since the migration of NO₂ on the surface of Pt is interfered with by the adsorbed SO₄

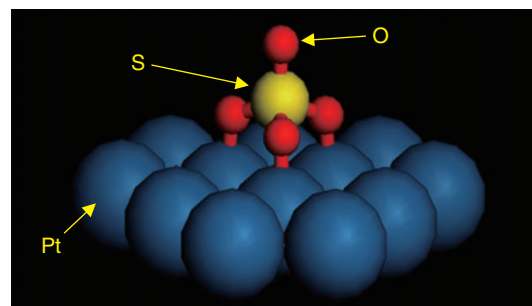


Fig. 11 Image of SO₄ structure on Pt surface

molecules, the rate of NO_2 adsorption on the Ce compound declines. The presence of SO_4 molecules on the platinum in the NO_2 reduction reaction also similarly impedes the adsorption of the desorbed NO_2 and the reducing gas such as CO on the platinum surface. As a result, it is inferred that NO_2 is emitted because the Eq. (4) reduction reaction of NO_2 and CO does not progress. A structure like that shown in Fig. 13 was therefore considered as a high-durability LNC.

First of all, a key consideration is how to cause NO_2 to move smoothly from the platinum surface to the cerium compound in the course of NO_2 adsorption. When the platinum particles are large, the NO_2 formed on active sites that are distant from the boundary surface between the platinum particle and the cerium compound has farther to travel to reach the cerium compound, so there is greater likelihood that the movement will be obstructed by SO_4 . Consequently, it appears that the key to improvement will not be a method of simply increasing the platinum content in order to increase the platinum active surface area, but rather in making the platinum particles small in order to lessen the distance moved by the NO_2 that is formed. A cerium compound had previously been used as the support for the Pt, however, high-surface-area Al_2O_3 was now employed in order to promote the formation of small platinum particles. In addition, a higher quantity of Al_2O_3 than the cerium compound was employed. Figure 14 shows the results from STEM confirmation of this trial. These results show that the average diameter of the platinum particles is 22 nm when Al_2O_3 is not employed, and 8 nm when it is employed, verifying that it was possible to form small platinum particles.

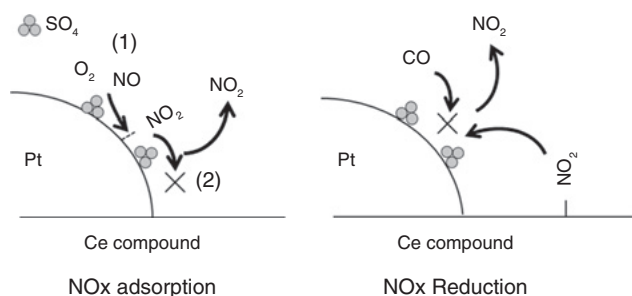


Fig. 12 Reaction mechanism of initial LNC

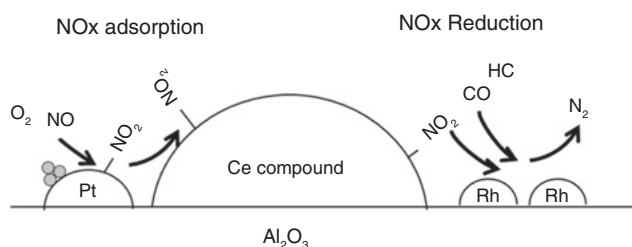
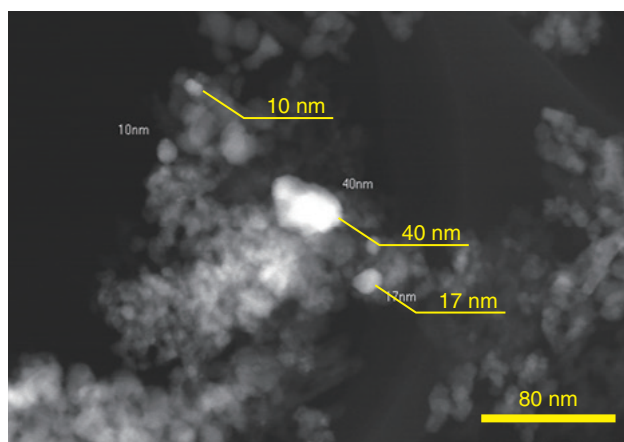


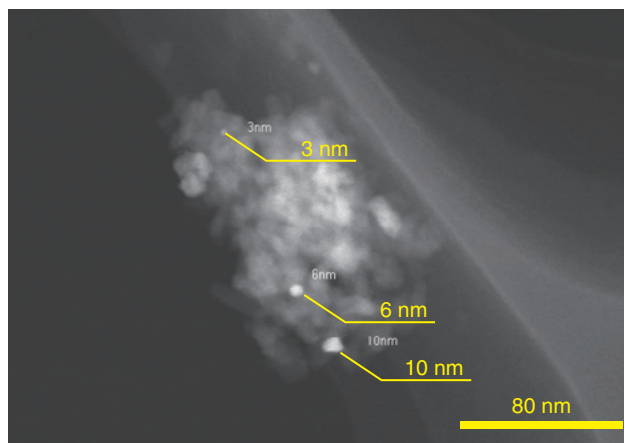
Fig. 13 Reaction mechanism of improved LNC

Next, in the course of reduction, it is important that the NO_2 desorbed from the cerium compound be caused to react immediately on a precious metal. Rhodium was therefore employed to promote a reduction reaction. Because rhodium is a precious metal which is not prone to agglomeration, it produces small particles. The use of small-particle rhodium increases the boundary surface between the rhodium and the cerium compound. This effect may enable the NO_2 desorbed from the cerium compound to move smoothly onto the rhodium and be reduced. Figure 15 shows a STEM image of $\text{Rh}/\text{Al}_2\text{O}_3$ after engine aging. It is apparent from these results that the rhodium maintains a small-particle state at about 2 nm, so that high NO_x reduction performance can be expected.

This effect was confirmed by conducting a model gas temperature-raising test with NO_2 and CO and investigating the activating temperature. Figure 16 shows the results. After thermal aging of the platinum-only LNC, it yielded high catalytic reduction performance from around 250°C. After engine aging, however, it yielded catalytic reduction



(a) Without Al_2O_3



(b) With Al_2O_3

Fig. 14 STEM images of initial LNC and improved LNC under engine aging

performance from around 400°C, showing a significant decrease in activity. On the other hand, the rhodium-added LNC showed virtually no difference after thermal aging and after engine aging, and it yielded high activity from low temperatures. In this way, the addition of rhodium was able to prevent the decrease in reduction performance after engine aging.

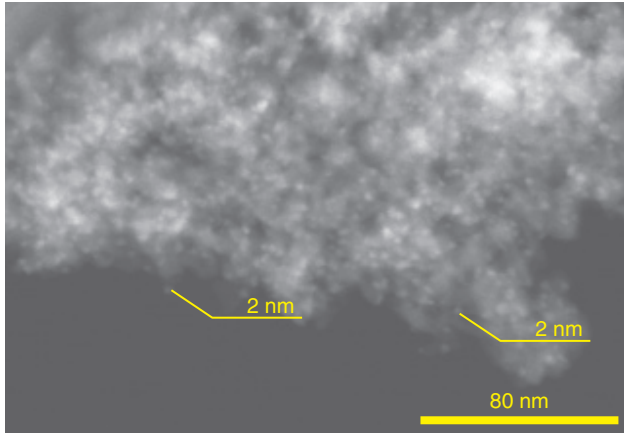


Fig. 15 STEM images of Rh/Al₂O₃ under engine aging

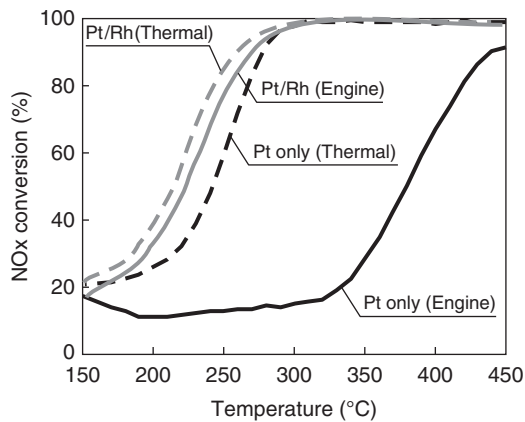


Fig. 16 Effect of Rh for improving durability under engine aging
NO₂ reduction: NO₂ 500 ppm, CO 1000 ppm, N₂ balance, GHSV 50000 h⁻¹

4.2. NO_x Purification Performance of a High-Durability LNC

The performance of the original LNC and the high-heat-resistance LNC was compared in order to determine the extent to which the modification boosted catalyst performance. The comparison was also extended to include the performance of the initial LNC after thermal aging. Figure 17 shows the results. The testing was conducted under the conditions shown in Table 2. The high-durability LNC after engine aging yields generally the same level of performance as the initial LNC after thermal aging. This confirms that the influence of SO₄ poisoning has been successfully reduced.

Vehicle tests were conducted in order to study the exhaust gas NO_x purification performance of the LNC. The original LNC and the modified LNC were aged in an engine to the equivalent of 120000 miles. The purification performance of the catalysts was compared in vehicle tests using FTP-75 mode. Figure 18 shows the results. With initial LNC catalyst, NO_x emissions are observed under both rich and lean conditions as stated above. With the high-durability LNC, however, emissions during rich spikes can largely be reduced, and performance during lean operation with high exhaust gas flow has also been improved.

These test results also verify the effectiveness of the modification of the catalyst in increasing durability in relation to engine aging.

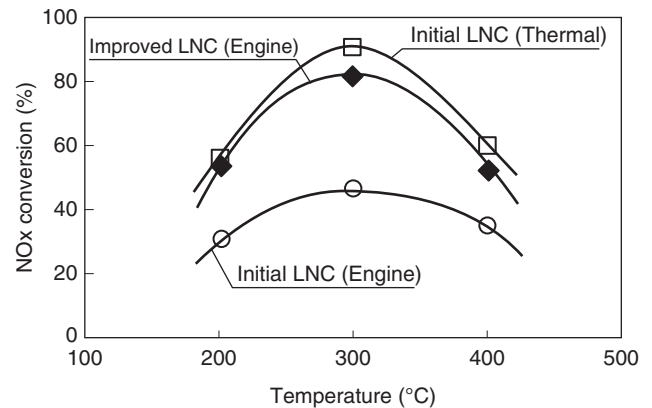


Fig. 17 Comparison of initial LNC and improved LNC in model gas test

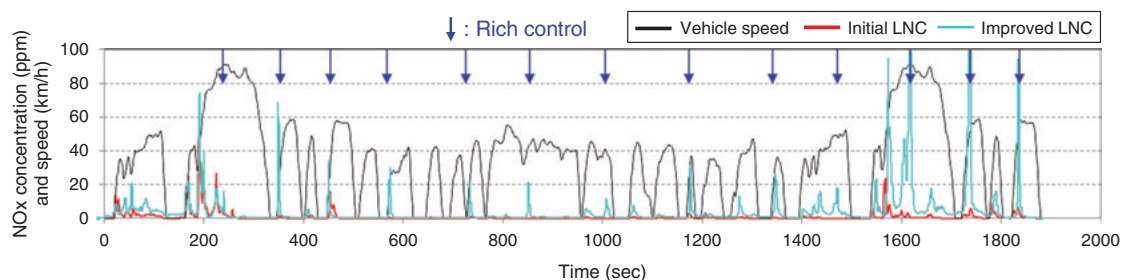


Fig. 18 Comparison of NO_x emission profile in FTP-75 mode between initial LNC and improved LNC under engine aging

5. Conclusion

An original Honda LNC system for diesel vehicles employing a new NOx purification mechanism was developed. It was observed that NOx purification performance declined when the catalyst was aged in an exhaust gas environment. The mechanism of catalyst degradation was elucidated and the catalyst materials were modified. As a result, a high-durability LNC was developed. This process yielded the following findings:

- (1) The factors resulting in a decline in the NOx purification performance of LNC when aged in an exhaust gas environment were analyzed. The results of this analysis suggested that the main factors in the degradation of the LNC were irreversible adsorption of the sulfur in the exhaust gas on the surface of the Pt as SO₄, and a decline in NO₂ adsorption performance and NO₂ reduction performance due to this SO₄.
- (2) Improvement of materials was carried out on the basis of the deterioration mechanism. As a result, the microparticulation of platinum promoted the movement of NO₂ to the cerium compound and the addition of rhodium enhanced the reduction function. By combining these results, a high-durability LNC catalyst that limits the decrease in NOx catalytic reduction performance was developed.
- (3) The NOx purification system using high-durability LNC catalyst was able to reduce NOx emissions during rich and lean operation, and the enhancement of purification performance was confirmed in vehicle testing.

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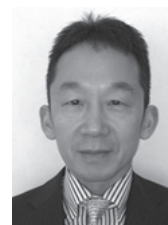
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Study of NO_x Reduction Behavior on Iron Ion-exchanged Zeolite Catalyst for ULEV50

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ABSTRACT

In response to further strengthening of emission regulations in recent years, the Urea Selective Catalytic Reduction system is gaining attention as a technology for reducing the NO_x emitted from diesel engines. The reaction between NO_x and NH₃ is known to exhibit high reduction performance (Fast-SCR) when the ratio of NO₂ to NO_x is 50%; that is to say, when the NO and NO₂ quantities are equal. On the other hand, the dynamic NO_x reduction behavior of an iron ion-exchanged zeolite (Fe-Zeolite) catalyst was analyzed in detail, and the results showed that high reduction performance can be obtained by intentionally varying the NO₂ ratio.

The behavior mentioned above is thought to occur because NO₂ adsorbs to the Fe-Zeolite catalyst and manifests a buffer function with respect to NO₂ ratio variation, so the effective NO₂ ratio near the catalyst reaction active sites is 50% even when the NO₂ ratio of the feed gas to the catalyst is not constantly maintained at 50%. Application of this intentional variation of the NO₂ ratio to vehicle control enabled clearance of the LEV-III ULEV50 standard.

1. Introduction

Recently, movements are accelerating to reduce carbon dioxide (CO₂) emissions that are said to be a cause of global warming. In addition, various regions have decided to introduce stricter emission regulations such as the LEV-III standard (California, U.S.A.) and the European Euro6 standard. Furthermore, regulations are also considered to limit the emission of nitrogen dioxide (NO₂), which is said to have an adverse effect on respiratory organs and to be a substance contributing to acid rain and photochemical smog. Moreover, European emission regulations require emission conversion technology that can adapt to a wider range of driving conditions, and European authorities have proposed and are investigating a new driving mode that adds high-load driving conditions to better represent real driving behavior of typical drivers. This new driving mode will replace the conventional New European Driving Cycle (NEDC) mode, which mainly consists of low-load driving.

Three-Way Catalyst (TWC) systems are widely used as an exhaust gas purification technology in gasoline engines, which mainly use stoichiometric air-fuel ratio control. On the other hand, TWC is not an effective exhaust gas purification technology in diesel engines, which use lean air-fuel ratio control. Carbon monoxide (CO) and hydrocarbons (HC) can be oxidized by a Diesel Oxidation Catalyst (DOC) as a method for converting exhaust gas at a lean air-fuel ratio, i.e., in an atmosphere with excess oxygen. However,

nitrogen oxides (NO_x) need to be converted by reduction, and it is a challenge to promote the reaction in a lean air-fuel ratio atmosphere, so exhaust gas conversion technology other than TWC is needed. Typically known technologies for converting NO_x in a lean air-fuel ratio atmosphere are NO_x Storage Catalyst (NSC) systems that store NO_x at lean air-fuel ratios and then reduce the stored NO_x by rich spike (varying the air-fuel ratio by increasing the fuel injection quantity), and Urea Selective Catalytic Reduction (Urea-SCR) systems that dose Urea (NH₂CONH₂) water into the exhaust gas and use the generated ammonia (NH₃) to reduce NO_x. The NSC system does not require major changes to the aftertreatment system, but the catalyst capacities and precious metal quantities need to be increased to maintain sufficient NO_x reduction performance at high NO_x concentrations, and optimum rich spike control and other technologies are required for NO_x reduction, which results in a technically challenging aftertreatment system that uses large quantities of expensive precious metals. On the other hand, due to the chemical reaction characteristics of the Urea-SCR system, high NO_x reduction performance can be expected even at high NO_x concentrations, and R&D has been promoted as a promising technology for vehicle types with high NO_x emissions that make clearing strict emission regulations a challenge.

The authors⁽¹⁾ proposed a combination of stoichiometric combustion and a Urea-SCR system as a low emission technology for diesel engines that require high NO_x

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reduction performance. That study indicated that high NO_x reduction in the initial stages of the driving mode is the most important factor for clearing strict emission regulations. To reduce NO_x from the initial stages of the driving mode, it is necessary to rapidly heat up the entire catalyst system to the activation temperature, and to generate an appropriate quantity of NO₂ for NO_x reduction. One rapid heat-up method is to control the engine fuel injection to emit large quantities of CO and HC, oxidize this CO and HC with a DOC and a Catalyzed Diesel Particulate Filter (cDPF) located immediately after the engine, and use that reaction heat to heat up the catalyst system⁽¹⁾. In this case it is necessary to strengthen the oxidation performance of the DOC and cDPF, but this also creates the issue that the oxidation performance is too strong in the middle of the driving mode when the system is sufficiently warmed up, so excess NO₂ is generated and the NO_x reduction performance of the Urea-SCR system is reduced.

To simultaneously address both of the above conflicting issues, this study hypothesized and verified an NO₂ ratio (percentage calculated by NO₂/NO_x) buffer concept based on NO₂ adsorption on the catalyst. As a result, it was found that the high NO_x reduction performance of the Urea-SCR system can be maintained by intentionally varying the NO₂ ratio, even under conditions where excess NO₂ is generated. In addition, prospects were obtained for clearing the LEV-III ULEV50 standard by applying the NO₂ ratio buffer concept to engine control, and those contents are also reported herein.

2. NOx Reduction Characteristics of Urea-SCR Catalysts

2.1. Urea-SCR System

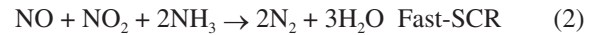
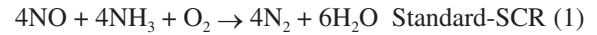
The Urea-SCR system generates NH₃ by dosing Urea water into the exhaust gas and performing hydrolysis⁽²⁾, and then uses the generated NH₃ to selectively reduce NO_x on the catalyst. Most on-board catalysts are iron (Fe) or copper (Cu) ion-exchanged zeolite catalysts. To obtain high NO_x reduction performance using a Urea-SCR catalyst, the following conditions need to be met.

2.1.1. Exhaust gas temperature

Generation of NH₃ from Urea water by hydrolysis requires an exhaust gas temperature of 180°C or more. When Urea water is dosed at less than that temperature, byproducts are generated, urea deposition occurs inside the exhaust pipe, and the Urea water cannot be effectively used as an NO_x reduction agent. However, the zeolite used in the Urea-SCR catalyst has NH₃ adsorption capability, so the adsorbed NH₃ can be used to reduce NO_x even under temperature conditions of less than 180°C. Nevertheless, the NH₃ quantity that can be adsorbed is limited, so to obtain high NO_x reduction performance, the catalyst temperature should quickly reach the Urea water hydrolysis temperature or higher.

2.1.2. NO₂ ratio in the Urea-SCR catalyst inlet gas

The following three reactions are known as NO_x reduction reactions by NH₃ in the steady state⁽³⁾.



Of the above three reactions, the reaction that proceeds from a low temperature and can be expected to exhibit high reduction performance (>95%) is the Fast-SCR reaction⁽³⁾ shown in Eq. (2). To realize high reduction performance in a Urea-SCR system, it is necessary to maintain the NO₂ ratio at the optimum ratio (NO₂ ratio = 50%).

Figure 1 shows a schematic illustration of the reaction at different NO₂ ratios. When there are equal quantities of nitrogen monoxide (NO) and NO₂ in the exhaust gas as shown in Fig. 1 (b), NO_x is reduced by the Fast-SCR reaction shown in Eq. (2). On the other hand, when there is excess NO as shown in Fig. 1 (a), a quantity of NO equal to the NO₂ quantity is reduced by the Fast-SCR reaction, and the excess NO is reduced by the Standard-SCR reaction. When there is excess NO₂ as shown in Fig. 1 (c), the excess NO₂ is reduced by the Slow-SCR reaction.

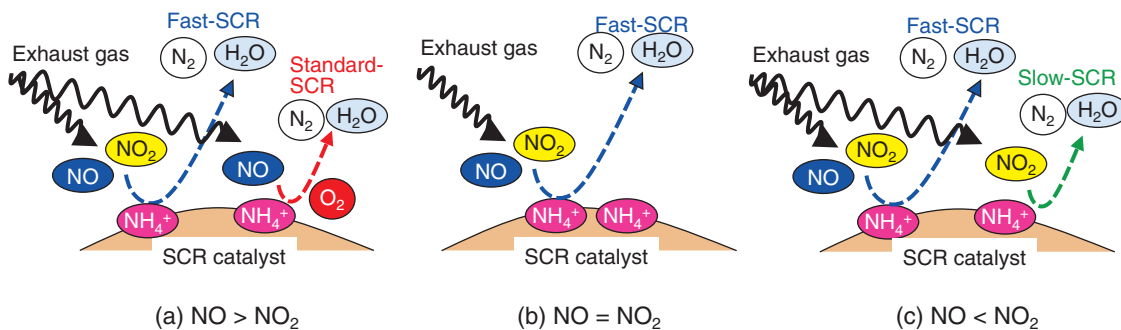


Fig. 1 Schematic illustration of NO_x reduction under various steady-state conditions, (a) NO > NO₂, (b) NO = NO₂, (c) NO < NO₂

2.2. Steady-state NO_x Reduction Characteristics of Fe-Zeolite Catalysts

The NO_x reduction performance of an iron ion-exchanged zeolite (Fe-Zeolite) catalyst at various NO₂ ratios was measured using a HORIBA model gas tester. Figure 2 shows an overview of the test apparatus. The model gas for the test was prepared by controlling the flow rates of water vapor (H₂O), oxygen (O₂) and nitrogen (N₂) gases using a Mass Flow Controller (MFC) to mix the designated quantities, passing these base gases through a heater to heat the mixture to the designated temperature, and then similarly controlling the NO, NO₂ and NH₃ component gases to the designated ratios using the MFC and mixing these component gases with the base gases. The prepared model gas was fed to the test catalyst, and the reactant gas after the test catalyst was measured using a HORIBA MEXA-6000FT that can simultaneously measure NO and NO₂. All model gas tests described hereafter measured the NO_x reduction performance using this apparatus.

The model gas was based on a gas composition simulating that of the exhaust gas during vehicle mode driving, and the gas conditions composition was set to increase the NO_x concentration in order to facilitate observation of NO_x behavior (Table 1). Figure 3 shows the NO_x reduction performance measurement results at that time.

It was confirmed that the NO_x reduction performance in the steady state shows the highest conversion rate at an NO₂ ratio of 50%, and that the Fast-SCR reaction exhibits high NO_x reduction characteristics. On the other hand, the NO_x conversion rate dropped when the NO₂ ratio deviated from 50%. The NO_x emitted from the engine is mostly NO, so in

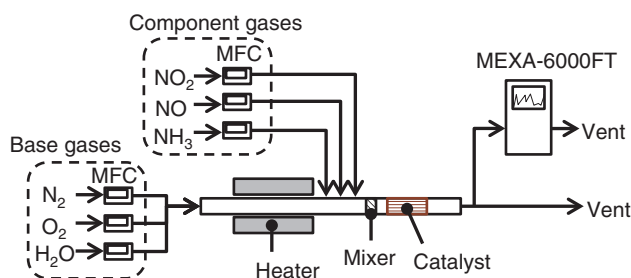


Fig. 2 Synthetic gas bench

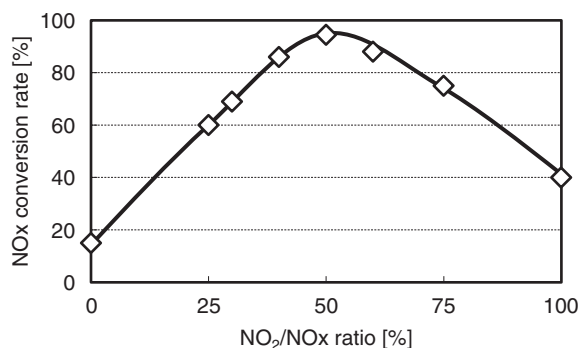


Fig. 3 NO_x conversion rate at each NO₂/NO_x ratio

Table 1 Gas conditions for various NO₂/NO_x ratios

NO _x [ppm]	200
NO ₂ /NO _x ratio [%]	0, 25, 30, 50, 60, 75, 100
NH ₃ [ppm]	200
O ₂ [%]	6
H ₂ O [%]	7
N ₂	Balanced
Space velocity [1/h]	37500
Temperature [°C]	250

order to control the NO₂ ratio fed to the Fe-Zeolite catalyst to 50%, it is important to appropriately oxidize the NO to NO₂ using an oxidation catalyst located upstream of the Fe-Zeolite catalyst.

2.3. Issues for Using a Urea-SCR Catalyst System in a Vehicle

The previous section described the features of the Urea-SCR catalyst system and the NO_x reduction behavior on an Fe-Zeolite catalyst in the steady state. However, the exhaust gas temperature, flow rate, NO_x concentration and other factors related to the Urea-SCR reaction vary greatly in the vehicle condition, making it a challenge to keep a stable NO_x reduction reaction such as in steady state. With regards to this issue, Tokui et al.⁽³⁾ and Murata et al.⁽⁴⁾ conducted studies based on the mechanism in an Fe-Zeolite catalyst whereby ammonium nitrate (NH₄NO₃) is generated from NO₂ and NH₃ at a high NO₂ ratio and the generated NH₄NO₃ and NO react at a low NO₂ ratio⁽⁵⁾. These studies confirmed that keeping the NO₂ ratio during the driving mode within the range of 50 ± 10% leads to high NO_x reduction performance, and reported the need to select oxidation catalysts with appropriate oxidation performance.

On the other hand, a recent trend in emission regulations is to emphasize test methods that are more deeply based on actual driving, and demands are increasing for low emissions during driving with a wider range of engine loads than current driving modes. In addition, as emission regulation values become stricter, high NO_x reduction performance is required from the initial stages of each mode. This means that it is necessary to enhance oxidation catalyst performance to generate NO₂ from lower temperatures, which makes it a challenge to select an oxidation catalyst that achieves an appropriate NO₂ ratio (50 ± 10%) after warm-up. These demands for conflicting catalyst characteristics are a major issue facing the Urea-SCR catalyst system.

2.4. SCR Catalysts Concept for Optimized NO_x Reduction at Transient Vehicle Operation

To develop emissions technology that better conforms to the vehicle condition, it is considered necessary to understand dynamic NO_x reduction behavior in which the NO₂ ratio varies, unlike the steady state shown in Fig. 1.

Regarding the dynamic NO_x reduction characteristics of SCR catalysts, NO₂ adsorption behavior on an Fe-Zeolite catalyst has been reported⁽⁶⁾, which suggests the existence of NO_x reduction behavior via NO₂ adsorption.

Therefore, as a result of examining the phenomenon of NO₂ adsorption to an Fe-Zeolite catalyst in conjunction with NO_x reduction behavior, it is thought that the adsorbed NO₂ acts as a buffer for the varying NO₂ ratio. Figure 4 shows a schematic illustration of the NO_x reduction behavior due to adsorbed NO₂. This concept is noted hereafter as the NO₂ ratio buffer concept.

When the NO₂ ratio is more than 50% as shown in Fig. 4 (a), the excess NO₂ is adsorbed near the catalyst active sites. Conversely, when the NO₂ ratio is less than 50% as shown in Fig. 4 (b), the excess NO contacts the adsorbed NO₂ and reacts by Fast-SCR. Therefore, a high NO_x conversion rate can be obtained without limiting the NO₂ ratio variation width by alternately repeating the states in Fig. 4 (a) and Fig. 4(b). In this manner, the NO₂ ratio buffer concept is a new approach that establishes local Fast-SCR on the catalyst via adsorbed NO₂ by intentionally repeating the high and low NO₂ ratio conditions, attaining high NO_x reduction performance even when the NO₂ ratio varies.

3. Investigation of Dynamic NO_x Reduction Behavior of Urea-SCR Catalysts

3.1. Verification of NO_x Reduction Behavior when the NO₂ Ratio Varies

The dynamic NO_x reduction behavior when the NO₂ ratio varies was measured using a model gas tester to verify the NO₂ ratio buffer concept. Test conditions (NO₂ ratio variation cycle test) were set that simulate NO₂ ratio variation in the vehicle condition, and the NO_x reduction characteristics were confirmed. When accelerating in the vehicle condition, the NO_x quantity increases and the NO₂ ratio drops, so the NO₂ ratio variation method in the cycle test was set to increase the NO gas concentration. In consideration of the Urea dosing control delay due to rapid acceleration, the NH₃ concentration was maintained at the high NO₂ ratio side condition. The NO₂ ratio variation time

Table 2 Synthetic gas conditions for cycle test (Mode 1)

	High NO ₂ /NO _x ratio	Low NO ₂ /NO _x ratio
NO [ppm]	50	350
NO ₂ [ppm]	150	150
NO ₂ /NO _x ratio [%]	75	30
NH ₃ [ppm]	200	200
O ₂ [%]	6	6
H ₂ O [%]	7	7
N ₂	Balanced	Balanced
Time [s]	30	10
Space velocity [1/h]	37500	
Temperature [°C]	250	

was set to a cycle with a high NO₂ ratio for 30 s and a low NO₂ ratio for 10 s in reference to the vehicle driving mode pattern. Table 2 and Fig. 5 show the test conditions, and Fig. 6 shows the test results.

The NO_x conversion rate when the NO₂ ratio was varied between 75% and 30% was between 80% and 100%, and indicated an average conversion rate of 90% (red line in Fig. 6). The average conversion rate for the NO₂ ratios of 75% and 30% obtained from the steady-state test results was 73% (blue dotted line in Fig. 6), so a 17% increase in the conversion rate was confirmed. That is to say, higher NO_x reduction performance than that in the steady state is exhibited by varying the NO₂ ratio, which suggests that the

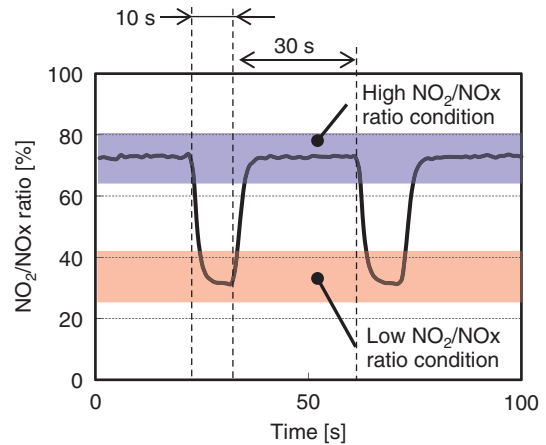


Fig. 5 Cyclic curve of NO₂ ratio (Mode 1)

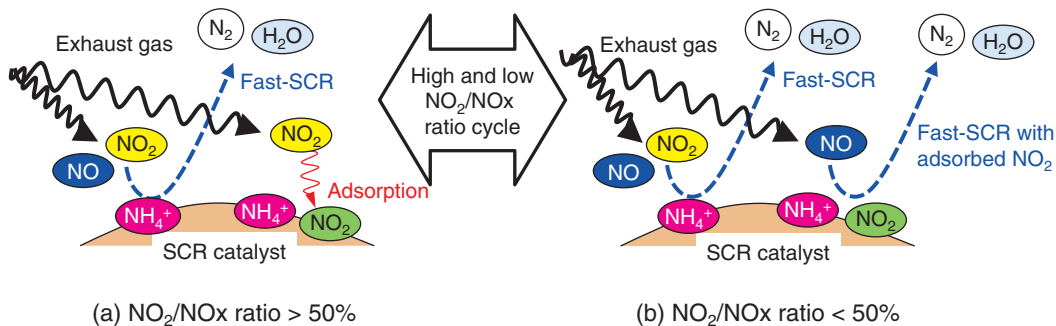


Fig. 4 Schematic illustration of NO_x reduction behavior during NO₂/NO_x ratio cycle, (a) high NO₂/NO_x ratio, (b) low NO₂/NO_x ratio

inferred NO_2 ratio buffer concept is manifesting.

Examination of the NO_x reduction behavior in Fig. 6 reveals that there is a time difference between the NO_2 ratio behavior and the NO_x conversion rate behavior. That is to say, when the NO_2 ratio changes from a higher state to a lower state, the NO_x conversion rate shifts accordingly to a higher conversion rate. However, when the NO_2 ratio changes from a lower state to a higher state, the NO_x conversion rate does not change immediately, and instead exhibits behavior to shift to a lower conversion rate after a delay. This is thought to indicate the following behavior: at a low NO_2 ratio, the NO_2 adsorbed to the Fe-Zeolite catalyst is consumed by local Fast-SCR on the catalyst, and at a high NO_2 ratio, the excess NO_2 is adsorbed to the Fe-Zeolite catalyst and the NO_x conversion rate then gradually drops to the same level as that in the steady state as the adsorption sites on the Fe-Zeolite catalyst become filled with NO_2 .

In order to verify the inferred dynamic NO_x reduction behavior described above, that is to say NO_2 adsorption at a high NO_2 ratio and local Fast-SCR on the catalyst at a low NO_2 ratio, NO_2 ratio variation cycle tests were performed with different conditions to confirm the effects on NO_x reduction performance.

3.2. Verification of NO_2 Ratio Variation Buffering Effects of Adsorbed NO_2

The effects on reduction performance when the NO_2

Table 3 Synthetic gas conditions for cycle test (Mode 2)

	High NO_2/NO_x ratio	Low NO_2/NO_x ratio
NO [ppm]	50	150
NO_2 [ppm]	150	150
NO_2/NO_x ratio [%]	75	50
NH_3 [ppm]	200	200
O_2 [%]	6	6
H_2O [%]	7	7
N_2	Balanced	Balanced
Time [s]	30	10
Space velocity [1/h]	37500	
Temperature [°C]	250	

ratio was set to vary between 75% and 50% were verified as conditions under which excess NO is not generated at a low NO_2 ratio. Table 3 lists the test conditions, and Fig. 7 shows the test results and compares them with the results of the steady-state test.

In the Mode 2 test, the NO_2 ratio variation width was changed to between 75% and 50%. This mode exhibited almost the same NO_x conversion rate performance as that in the steady state shown in Fig. 3 at both a high NO_2 ratio and low NO_2 ratio, and it was confirmed that the high NO_x reduction performance at a high NO_2 ratio described in section 3.1 above was not exhibited. That is to say, it is thought that high reduction performance could not be manifested because the conditions were such that excess NO was not generated

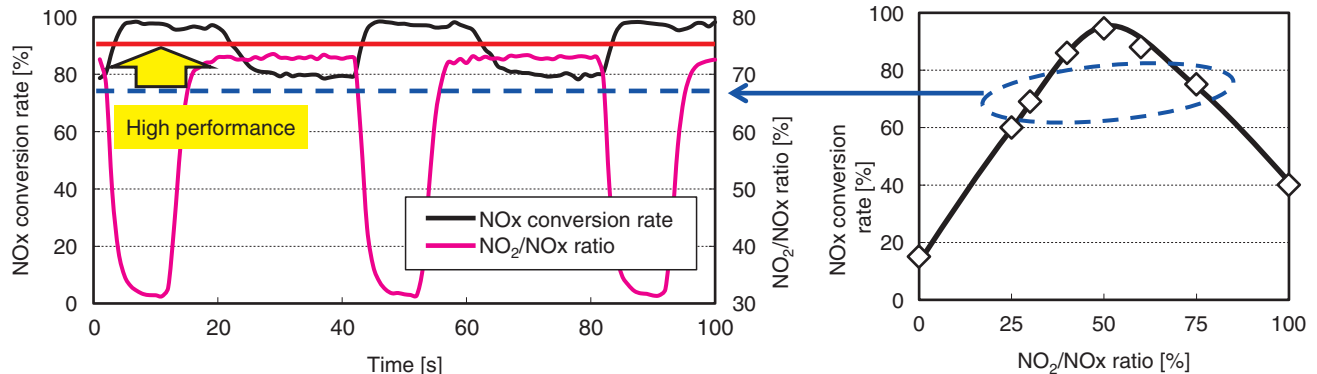


Fig. 6 Results of synthetic cycle gas test (Mode 1)

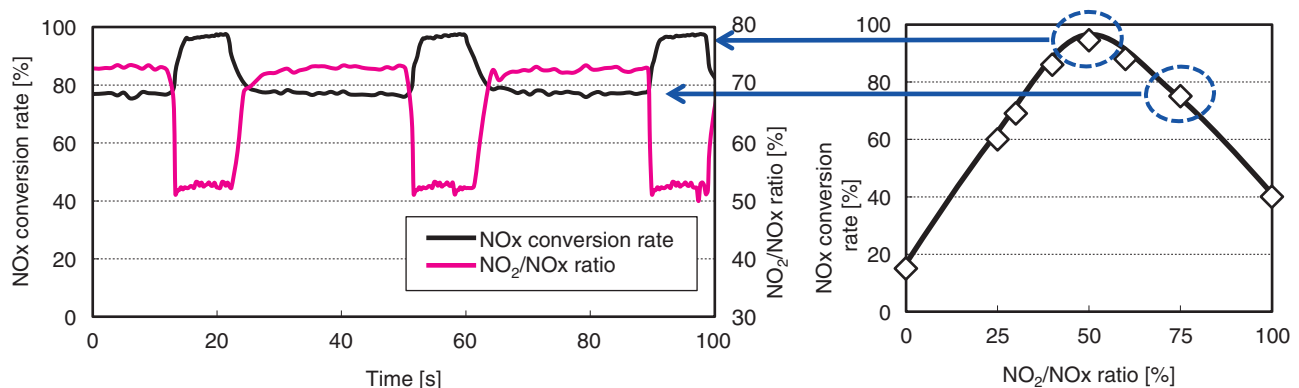


Fig. 7 Results of synthetic cycle gas test (Mode 2)

on the low NO₂ ratio side, so the adsorbed NO₂ was not consumed, and the excess NO₂ could not be adsorbed at a high NO₂ ratio. Therefore, the Mode 2 results could be said to support the NO₂ ratio buffer concept approach.

Next, the NO₂ behavior was verified when the time on the low NO₂ ratio side (NO₂ ratio = 30%) was changed to vary the excess NO feed quantity. Compared to Mode 1 in Table 2, Table 4 shows Mode 3 in which the low NO₂ ratio time was shortened to 5 s, and Table 5 shows Mode 4 in which the low NO₂ ratio time was lengthened to 30 s. Figure 8 shows the time-base trend in the NO₂ gas concentration in each test mode in order to investigate the detailed NO₂ gas behavior.

The NO₂ concentration behavior in Mode 1, which uses the same conditions as those described in the previous section, was as follows. The NO₂ concentration remained low for approximately 7 s after switching to the high NO₂ ratio, and then rose to reach a steady state at around 50 ppm (NO_x conversion rate 78%). The NO_x conversion rate in this steady state matched the NO_x conversion rate in the NO₂ ratio 75% steady state shown in Fig. 3, indicating that only the reduction reactions (Fast-SCR and Slow-SCR) were occurring. The low NO₂ concentration range behavior before reaching the steady state is thought to result from other than reduction reactions, and the NO₂ concentration rises over time, so this can be understood as the NO₂ adsorption behavior to the Fe-Zeolite catalyst.

In contrast, the time to reach the steady state was shorter in Mode 3, which had a shortened low NO₂ ratio time. Conversely, in Mode 4 which had a lengthened low NO₂ ratio time, the time until the NO₂ concentration began to rise was also longer, and it was observed that the steady

Table 4 Synthetic gas conditions for cycle test (Mode 3)

	High NO ₂ /NO _x ratio	Low NO ₂ /NO _x ratio
NO [ppm]	50	350
NO ₂ [ppm]	150	150
NO ₂ /NO _x ratio [%]	75	30
NH ₃ [ppm]	200	200
O ₂ [%]	6	6
H ₂ O [%]	7	7
N ₂	Balanced	Balanced
Time [s]	30	5
Space velocity [1/h]	37500	
Temperature [°C]	250	

Table 5 Synthetic gas conditions for cycle test (Mode 4)

	High NO ₂ /NO _x ratio	Low NO ₂ /NO _x ratio
NO [ppm]	50	350
NO ₂ [ppm]	150	150
NO ₂ /NO _x ratio [%]	75	30
NH ₃ [ppm]	200	200
O ₂ [%]	6	6
H ₂ O [%]	7	7
N ₂	Balanced	Balanced
Time [s]	30	30
Space velocity [1/h]	37500	
Temperature [°C]	250	

state was not reached by the 30 s mark.

Based on these NO₂ behavior results, it is thought that the NO₂ adsorption quantity at the high NO₂ ratio increases as the low NO₂ ratio time becomes longer, that is to say as the excess NO feed quantity increases. This is thought to indicate that the excess NO undergoes local Fast-SCR

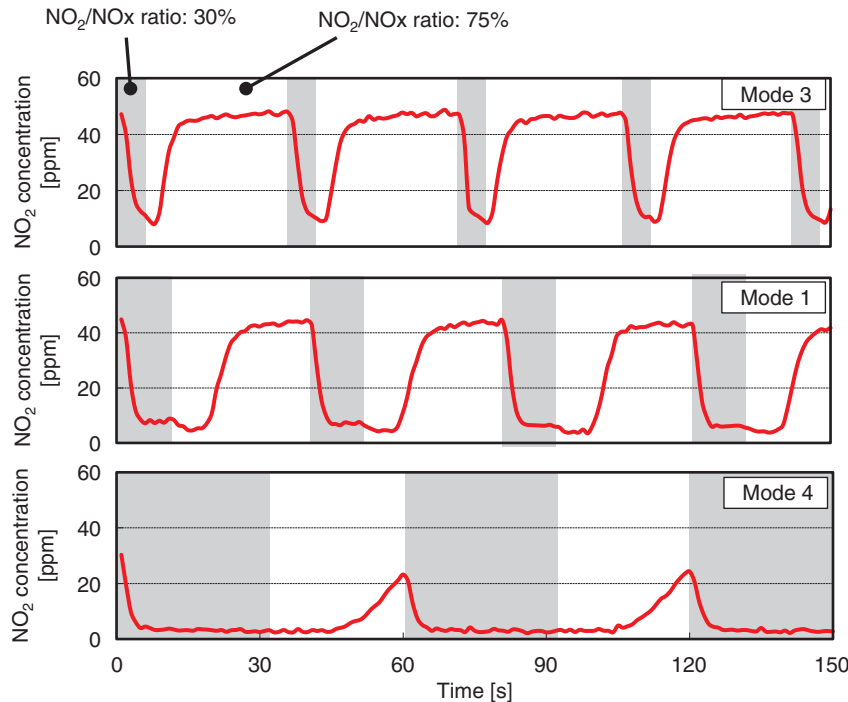


Fig. 8 Results of synthetic cycle gas test (Mode 1, Mode 3 and Mode 4)

reactions with the NO_2 adsorbed to the catalyst, with the result that the NO_2 is consumed, returning the NO_2 adsorption sites to a state that enables adsorption.

The above verification results and observations enabled to experimentally confirm the NO_2 ratio buffer concept proposed in the previous section. That is to say, the method of taking into account the NO_2 adsorption capacity of the Fe-Zeolite catalyst and intentionally supplying excess NO to the Fe-Zeolite catalyst before the NO_2 slips is an effective method for bringing out the maximum NOx reduction performance of the Urea-SCR system.

The following sections describe application of the NO_2 ratio buffer concept to a vehicle system, and the results of verifying these effects.

4. Urea-SCR System for LEV-III ULEV50

4.1. Urea-SCR System in a Vehicle

A mid-size passenger car (inertia weight: 4 000 lbs) equipped with a 2.2-liter inline four-cylinder intercooler turbocharged diesel engine was used as the test vehicle to measure emissions. Figure 9 shows the layout of the

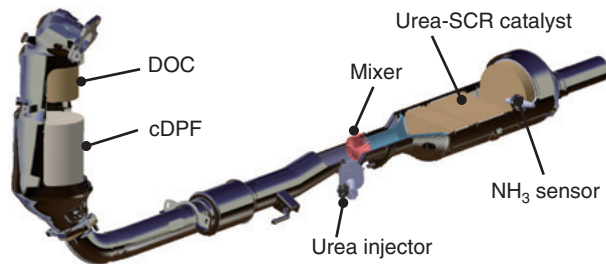


Fig. 9 Hardware configuration of Urea-SCR system

emission system in this study. A DOC and cDPF are located immediately after the engine, and an injector for Urea water dosing, a mixer for feeding the dosed Urea water to the catalyst in a homogeneous manner, and the Urea-SCR catalyst are located in that order under-floor. An NH_3 sensor is installed between the SCR catalysts, enabling optimum NH_3 supply and NH_3 slip control.

The feature of this system is that it assigns a role to each catalyst. The DOC immediately after the engine oxidizes the CO and HC, and the heat from these reactions is used for the initial heat-up of this system. The cDPF collects and reduces particulate matter (PM), and generates the NO_2 required for the Urea-SCR reaction by the platinum (Pt) based catalyst. The Urea-SCR catalyst has the role to reduce NOx by NH_3 . This system is an aftertreatment system that is able to reduce emissions effectively when all of these roles function properly.

4.2. Technical Issue for Clearing the LEV-III ULEV50 Standard

To clear the LEV-III ULEV50 standard, it is necessary to reduce 90% or more of the emitted NOx in FTP-75 mode. To clear this high requirement, emissions reduction was investigated by dividing the NOx reduction areas in FTP-75 mode (Fig. 10) and setting the catalyst functions required in each area (Table 6).

The approach to each area in Fig. 10 is as follows. In area (A), the catalyst system is rapidly heated up to the activation temperature, and in area (B), NOx is reduced using the NH_3 adsorbed to the Urea-SCR catalyst. Urea water is not dosed in area (B) because the catalyst system has not yet reached the Urea water hydrolysis temperature range. In the following areas (C) and (D), the exhaust

Table 6 Catalytic functions in each NOx reduction area

Catalyst	Area (A)	Area (B)	Area (C)	Area (D)
DOC	CO and HC oxidation	CO and HC oxidation to keep catalysts active		
cDPF	-	NO oxidation (NO_2 generation)		
SCR	-	NOx reduction by adsorbed NH_3	NOx reduction by adsorbed NH_3 and urea dosing	

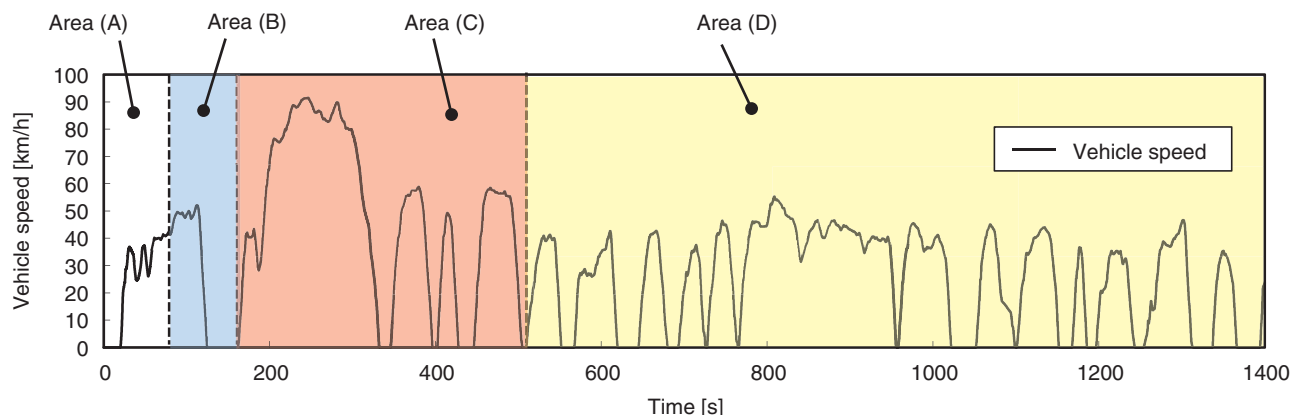


Fig. 10 Image of NOx reduction areas in Cold Transient (CT) and Cold Stabilized (CS) phases of FTP-75 mode

temperature is sufficiently high, so Urea water is dosed to generate NH₃, and NO_x is reduced at the high conversion rate of 95% or more.

To realize this approach, low temperature CO and HC oxidation activity in area (A) is required of the DOC, and low temperature NO₂ generation capability in area (B) and NO₂ generation capability under high gas space velocity (SV) conditions in area (C), in which the engine operates at high load, are required of the cDPF. In this test it was important to achieve high NO_x reduction performance from the initial stages of the driving mode, so DOC and cDPF with high oxidation performance were set in consideration of the required performances for area (A) and for areas (B) and (C). Figure 11 shows the vehicle emission results at that time.

The expected NO_x reduction results were obtained in areas (B) and (C) of FTP-75 mode, but the NO_x reduction performance dropped in area (D) in the middle of the mode. Figure 11 shows that the NO₂ ratio trended at 50% or more in area (D), and almost all of the NO_x that passed through the Urea-SCR catalyst was exhausted as NO₂. This is because the catalyst is sufficiently warmed up in area (D) and the engine load and SV are also low, so excess NO₂ generated by the strong oxidation DOC and cDPF flows to the Urea-SCR catalyst. That is to say, setting DOC and cDPF with high oxidation performance to emphasize NO_x reduction in the initial stages of the driving mode creates the issue that the NO_x reduction performance drops in the low load area after warm-up. This means that the technical issue to clear LEV-III ULEV50 is NO₂ ratio control in the low-load area after catalyst warm-up.

Therefore, the NO₂ ratio buffer concept using an Fe-Zeolite catalyst confirmed in the previous section was applied to vehicle control, and the emissions were evaluated.

5. Application of NO₂ Ratio Buffer Concept to Vehicle Control

To apply the NO₂ ratio buffer concept to a vehicle, an engine system control method that controls the NO₂ ratio is needed. Therefore, the NO₂ ratio was adjusted by controlling the exhaust gas recirculation valve (EGR valve), the fuel injection quantity and other parameters to change the exhaust gas from the engine. Figure 12 shows the results of applying NO₂ ratio control under cruise driving conditions where excess NO₂ is emitted from the tailpipe. It was confirmed that by performing control to intentionally lower the NO₂ ratio (green bands in Fig. 12), NO₂ emissions from the tailpipe were reduced, and high NO_x reduction performance was obtained.

Next, Fig. 13 shows the results of applying NO₂ ratio control while driving in FTP-75 mode to verify the NO₂ ratio buffer concept during mode driving. It was confirmed that by applying NO₂ ratio control, NO₂ emissions during mode driving that involves acceleration and deceleration are suppressed in the same manner as during cruise driving conditions, and high NO_x reduction performance could be achieved even in area (D). These results indicate that the NO₂ ratio buffer concept is also effective at typical driving conditions, and that this concept is an effective means for addressing the technical issues described above.

Application of a Urea-SCR system incorporating the Fe-Zeolite catalyst NO₂ ratio buffer concept examined in this study achieved NO_x emissions of 0.025 g/mile and non-methane organic gas (NMOG) emissions of 0.015 g/mile in FTP-75 mode, which enabled to clear the LEV-III ULEV50 level emissions values (Fig. 14).

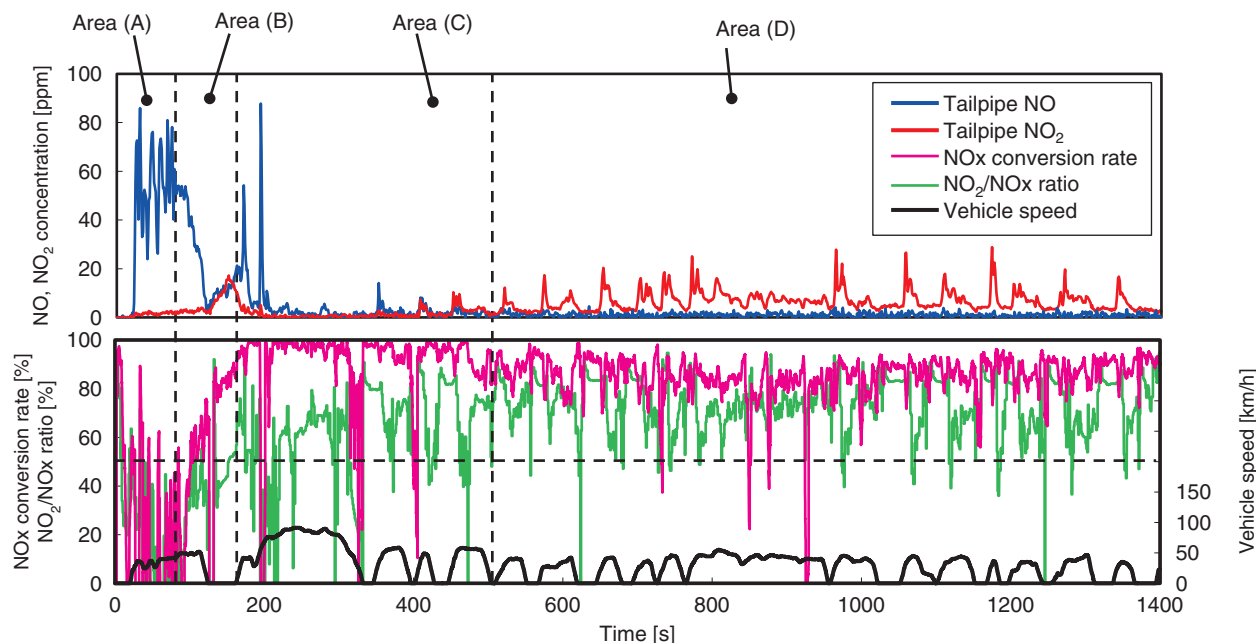
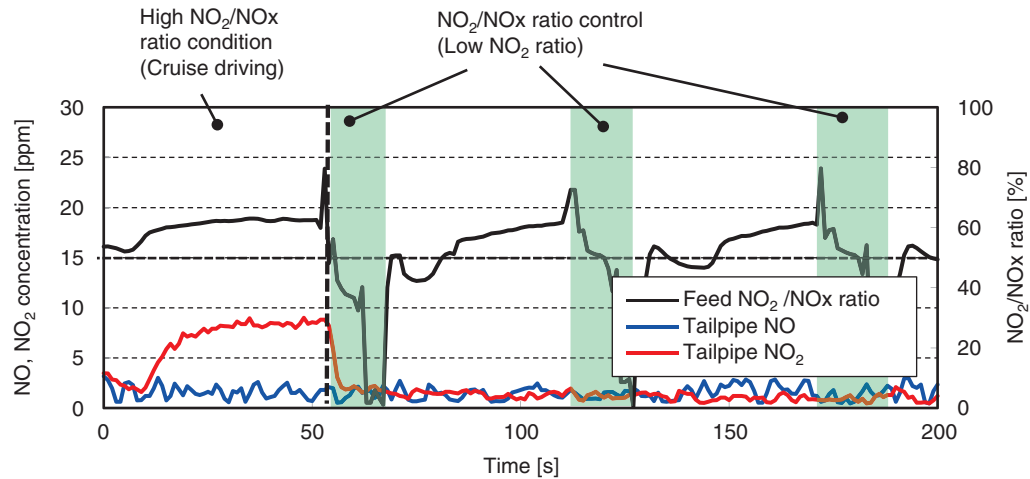
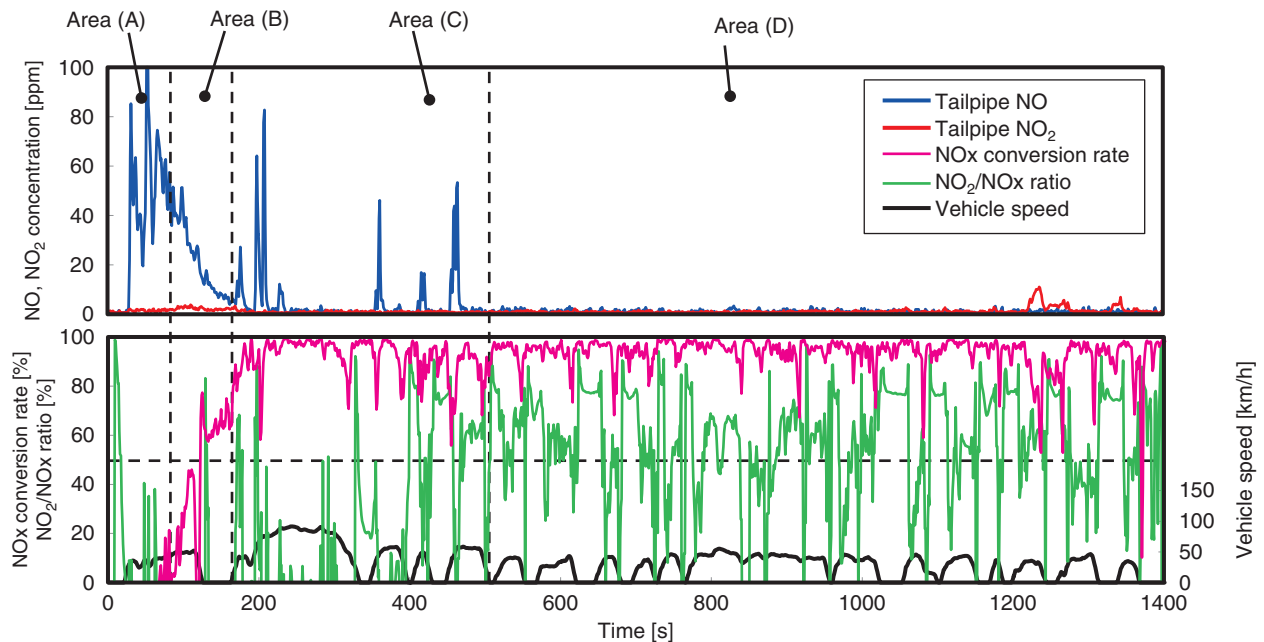
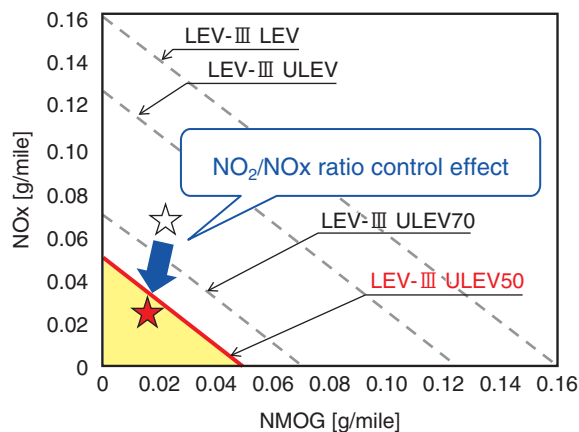


Fig. 11 Vehicle emission results by using strong oxidation catalysts (DOC and cDPF) in FTP-75 mode (CT and CS)

Fig. 12 Effect of new NO_2/NO_x ratio control in vehicle cruise conditionFig. 13 Vehicle emission results by introducing new NO_2/NO_x ratio control in FTP-75 mode (CT and CS)Fig. 14 NO_x and NMOG emission results in FTP-75 mode

6. Conclusion

The NO_x reduction behavior of an Fe-Zeolite catalyst was understood, and an NO_2 ratio buffer concept was proposed.

This NO_2 ratio buffer concept makes use of the mechanism that excess NO_2 is adsorbed to the Fe-Zeolite catalyst at a high NO_2 ratio, and local Fast-SCR reactions occur on the catalyst between the excess NO and the adsorbed NO_2 at a low NO_2 ratio, enabling to manifest high NO_x reduction performance even when the NO_2 ratio deviates from around 50%. This concept enabled to manifest high NO_x reduction performance even in the vehicle condition without limiting the NO_2 ratio variation width, by intentionally varying the NO_2 ratio.

This concept was applied to a vehicle, and it was proven that the NO₂ ratio can be controlled by EGR and fuel injection. As a result, high NOx reduction performance could be obtained even at dynamic vehicle conditions.

These measures addressed the technical issue of the drop in NOx reduction performance in the low-load area after catalyst heat-up that appears when catalysts with the high oxidation performance required in the initial stages of the driving mode are set, enabling to clear the LEV-III ULEV50 standard.

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Optimization of Fuel Cell Generation Environment by Water Observation using Neutron Radiography

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ABSTRACT

Electric generation by a polymer electrolyte fuel cell requires an appropriate quantity of water to help ensure proton conductivity of the membrane electrode assembly. However, excess water in the gas channels and the gas diffusion layer is thought to be a factor impeding gas supply, so it is necessary to appropriately control the water inside the fuel cell during electric generation.

This study used neutron radiography to observe the state of the water accumulated in a fuel cell during electric generation in order to clarify the mechanism by which water accumulates and its effect on voltage behavior, and set the optimum electric generation conditions.

Observation using neutron radiography was performed from the two directions of the through-plane direction and the in-plane direction of the fuel cell. Observation in the through-plane direction of an actual cell clarified the correlation between the accumulated water and the voltage behavior. On the other hand, observation in the in-plane direction of a test cell determined the gas channels in which accumulated water exists and the reason for that water accumulation, and suggested directions for countermeasures. Based on these results, the optimum electric generation environment was set in terms of water and electric generation performance, and applied to the control of an actual vehicle.

1. Introduction

1.1. Fuel Cells

Fuel cell electric vehicles, which are the ultimate form of electric vehicle that does not emit CO₂ or hazardous substances, are being developed to address environmental issues such as global warming and air pollution, and energy issues due to depleting fossil energy such as petroleum resources. Honda is also promoting the development of fuel cell electric vehicles using polymer electrolyte fuel cells (PEFC) that can start from low temperatures and also be made smaller.

Fuel cells are power generating devices that produce electricity by the electrochemical reaction of oxygen and hydrogen. The actual reaction comprises the oxidation reaction of hydrogen at the anode shown in Eq. (1), and the oxygen reduction reaction at the cathode shown in Eq. (2).

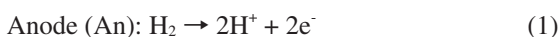


Figure 1 shows a schematic image of a fuel cell. The fuel cell structure consists of a membrane electrode assembly (MEA) with catalyst layers and gas diffusion layers (GDL)

positioned around a polymer electrolyte membrane (PEM), and separators that form gas and coolant channels.

1.2. Importance of Water Management in a Fuel Cell

To obtain high power output during electric generation by a fuel cell, the MEA needs to be moistened with an appropriate quantity of water to help ensure proton conductivity. However, excess water accumulates in the GDL and gas channels, which impedes gas supply, lowers electric generation performance, and results in unstable voltage behavior. In addition, impedance of the gas supply increases the electric potential, which can cause corrosion of

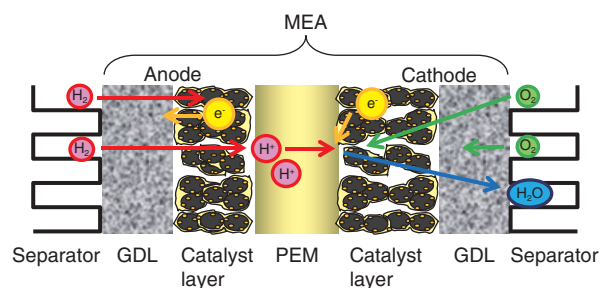


Fig. 1 Schematic image of fuel cell

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the carbon and platinum, and lower durability performance. Therefore, water management to maintain an appropriate MEA water content and to discharge the water from the gas channels is important for the electric generation of fuel cells. Recently, various methods are being investigated to visualize accumulated water due to the importance of this water management. For example, there are numerous reports of methods such as making an acrylic window and directly observing the inside of a fuel cell, or using X-ray radiography measurement and magnetic resonance imaging (MRI) to observe the water⁽¹⁾⁻⁽⁴⁾.

Table 1 shows the measurement speed, spatial resolution and observation range of various visualization technologies. Direct observation via an acrylic window is advantageous for continuously understanding changes in the accumulated water, but this method is unable to observe the water inside the MEA. The other methods can understand the water in the MEA, but neutron radiography and X-ray radiography have an advantage over MRI in terms of measurement speed, and can easily understand time-base changes in the water.

Figure 2 shows the mass attenuation coefficients of various atoms when irradiated by neutron rays and X-rays⁽⁵⁾. The mass attenuation coefficients of water and the atoms that constitute water are higher than those of Al, Fe and other fuel cell component materials, which is effective for observing small quantities of water inside a fuel cell. X-rays have the characteristic that the mass attenuation coefficient increases gradually as the atomic number increases, which makes it a challenge to isolate small quantities of water. On the other hand, neutron rays exhibit higher mass attenuation characteristics for elements with smaller atomic numbers such as hydrogen, and the mass attenuation coefficient of water in particular is an order of magnitude or more higher than that of Fe and Al. This characteristic means that small quantities of water can be observed more accurately using neutron radiography measurement than with X-rays. This study used neutron radiography as the water measurement technology in consideration of the overall balance between the water detection accuracy, measurement speed, and observation range. Neutron radiography measurement

Table 1 Comparison of visualization methods

Method	Time resolution	Spatial resolution	Observation region
Neutron radiography	5-10 sec	10-200 μm	Below 300 mm
X-ray radiography	2-10 sec	1-10 μm	As required
MRI	5 minutes	10 μm	100 mm
Direct visualization	—	—	As required

technology that was formerly limited to small cells was applied to a cell of the size mounted in the Honda FCX Clarity, and the accumulated water state was observed in an actual cell size.

1.3. Research Goals

The goal of this study was to use neutron radiography to clarify the relationship between accumulated water and voltage behavior by checking the state of water accumulated in a cell of the size mounted in the Honda FCX Clarity, and apply the optimum stack electric generation environment to actual vehicle control.

2. Principle of Water Measurement Using Neutron Radiography

Neutron radiography is a method for irradiating the observation subject with neutron rays, converting the transmitted neutron rays to light with a scintillator, and visualizing this light with a CCD camera or other detector. The attenuation coefficient of the neutron rays differs according to the atoms through which the neutron rays are transmitted, so locations consisting of atoms with large attenuation coefficients appear dark, and locations consisting of atoms with small attenuation coefficients appear bright.

The water is quantified by first extracting an image of the water by dividing an image of the state without water (background image) from the neutron radiography image during electric generation. Next, the water can be quantified by using the neutron attenuation characteristics given in Eq. (3) to derive the thickness t [cm] of the measurement subject⁽¹⁾.

$$I_{\text{out}} = I_{\text{in}} \exp(-\alpha t) \quad (3)$$

Here, I_{in} is the incident neutron intensity, I_{out} is the transmitted neutron intensity, and α indicates the neutron mass attenuation coefficient [cm^2/g] of the measurement subject. This study quantified the liquid water inside the fuel cell by multiplying the thickness t [cm] of the liquid water inside the fuel cell under various electric generation conditions by the area to be quantified [cm^2], and expressing the liquid water as a volume [cm^3]. Note that the neutron radiography images in this report indicate regions without water as yellow, and regions with water as blue.

Water measurement using neutron radiography was performed at the NEUTRA beam line of the Paul Scherrer

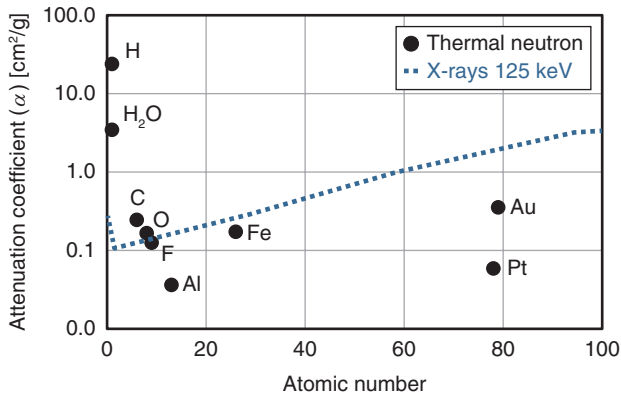


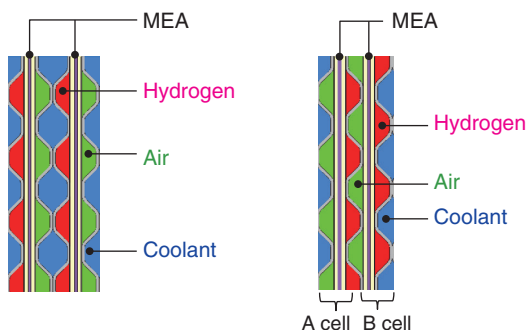
Fig. 2 Attenuation coefficient of elements

Institute (PSI) in Switzerland, in collaboration with the PSI Electrochemistry Laboratory (ECL) and Neutron Imaging and Activation Group (NIAG).

3. Tests

3.1. Directions for Observing Accumulated Water Using Neutron Radiography

Typical fuel cell stacks have a structure in which cooling is performed separately for each cell [Fig. 3(a)]. In contrast, the fuel cell stack mounted in the FCX Clarity uses a structure that performs cooling in two-cell units as shown in Fig. 3(b) in order to reduce size and weight⁽⁶⁾. For this reason, there are two types of cell: A-cells in which the cathode surface is cooled, and B-cells in which the anode surface is cooled. Observation was performed from both the through-plane and in-plane directions to accurately observe the distribution and locations of accumulated water in this fuel cell stack. Figure 4 shows a schematic image of the neutron radiography observation directions. Observation from the through-plane direction enables understanding of the water distribution over the entire electric generation surface, and observation from the in-plane direction enables accurately isolating the positions of the accumulated water in the MEA and each gas channel.



(a) Two cooling layers per two cells (b) One cooling layer per two cells

Fig. 3 Schematic image of two kinds of stacking

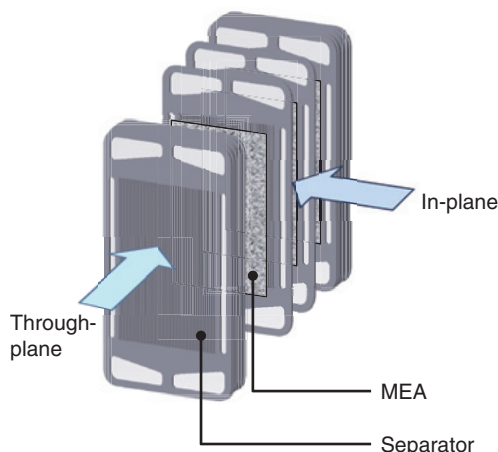


Fig. 4 Schematic image of neutron irradiation direction

3.2. Introduction of Neutron Radiography Cells

3.2.1. Actual cell used for observation in the through-plane direction

Observation in the through-plane direction was performed using an actual cell in order to observe the accumulated water distribution conditions in the cell mounted in the FCX Clarity. Figure 5 shows the cell holder used for observation and the neutron radiography image. The imaging range is wide, so the cell holder is made of materials that take into account both neutron transmittance and the strength to prevent structural warping. This enabled clear observation of accumulated water while maintaining structural integrity. In addition, two-cell observation is possible using a one cooling layer per two cells structure (Fig. 5), and each cell can also be observed separately to check the difference between the A-cell and B-cell.

3.2.2. Cell used for observation in the in-plane direction

The actual cell has a long neutron transmission distance with low transmittance in the in-plane direction, which makes it difficult to measure the water. Therefore, an in-plane observation cell was designed that simulates a one cooling layer per two cells structure (Fig. 6). The MEA

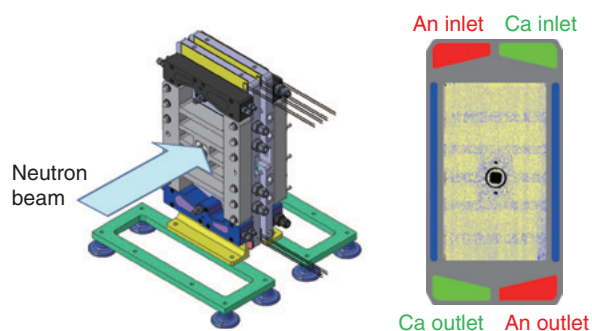


Fig. 5 Schematic and neutron images of through-plane cell

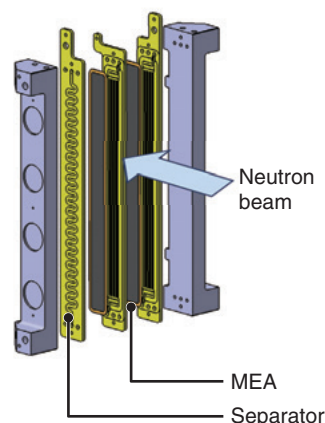


Fig. 6 Schematic image of in-plane cell

seal structure and sealer material of this cell were modified and the number of gas channels was reduced to increase the neutron transmittance in the in-plane direction. All A-cell and B-cell gas channels have the same structure. In addition, the temperature of each separator can be measured using thermocouples.

Figure 7 shows an image of the in-plane observation cell visualized using neutron radiography. The structure simulates the one cooling layer per two cells structure, and both sides of two adjacent cells are cooled by coolant. The red frames in Fig. 7 indicate the anode channels, and the green frames indicate the cathode channels. The region between the anode and cathode channels is the MEA.

3.3. Test Conditions

3.3.1. Neutron measurement conditions

Table 2 shows the neutron radiography measurement conditions. The conditions for the in-plane observation cell were set to increase the resolution in the sideways direction, and the conditions for the actual cell were set to obtain a wide measurement field of view. In addition, a long exposure time was set for the in-plane observation cell to increase the image resolution.

3.3.2. Power generation conditions

The electric generation performed during neutron radiography measurement was low-load electric generation in which the accumulated water has a large effect on the voltage behavior. The standard electric generation conditions in the development stage were set as the base electric generation conditions.

The accumulated water quantity in the actual cell was observed at different cathode stoichiometry values and coolant temperatures as parameters. In addition, tests were also conducted that simulated anode purge, which is performed as control when mounted in a vehicle. A compound test that combined the above conditions was also added as a proving test for application to a vehicle. The electric generation apparatus at the PSI facility was used^{(7), (8)}.

In this paper, the gas flow rate is expressed by the stoichiometry. The stoichiometry is the value calculated

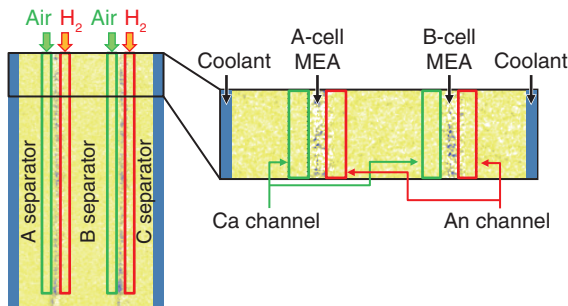


Fig. 7 Neutron radiography image of in-plane cell

Table 2 Neutron imaging condition of each cell

Imaging condition	Through-plane	In-plane
Pixel size (H × W) [μm]	100 × 100	75 × 15
Field of view (H × W) [cm]	25 × 35	25 × 5
Exposure time [sec]	5	10

according to Eq. (4), and is expressed as a relative value, with 1 being the case when the exact quantity of gas required for the fuel cell reaction is supplied.

$$\text{Stoichiometry} = \text{Feed-gas quantity} / \text{Gas quantity required for reaction} \quad (4)$$

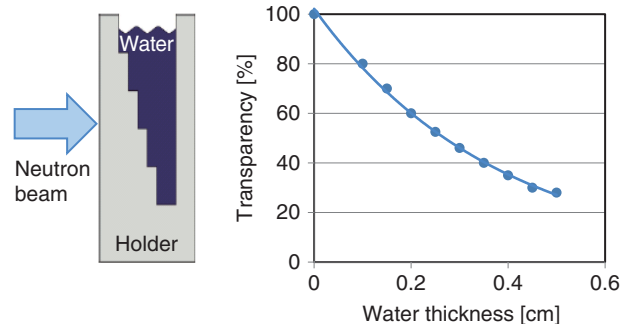
3.3.3. Anode purge

Anode purge refers to periodically opening the anode discharge side valve during electric generation to discharge the nitrogen and other gases not required for electric generation. This operation increases the purity of the hydrogen in the anode gas, and is also thought to be effective from the viewpoint of discharging water. Therefore, a test that simulated anode purge was conducted using the actual cell to check the effects of anode purge on discharging accumulated water.

4. Neutron Radiography Test Results

4.1. Quantitativity Using Neutron Radiography

Different water thicknesses were measured and the neutron transmittance was checked in order to check the quantitativity of water using neutron radiography. Figure 8 shows the measurement results. The transmittance was observed to decrease exponentially as the water thickness increased, which confirmed the validity of Eq. (3). The quantitative values in this study indicate the water using volume [cm³] conversion by first obtaining the water thickness using the quantitative constant obtained through tests, and then multiplying the results by the quantification area.



(a) Schematic representation of water scale (b) Transparency of water filled with different thicknesses

Fig. 8 Validation for water quantification

4.2. Observation of Current Accumulated Water Status and Factor Analysis

4.2.1. Observation of accumulated water during electric generation by an actual cell

Neutron radiography measurement was performed during electric generation using an actual cell to check the accumulated water status in a fuel cell of the size mounted in the FCX Clarity. Figure 9 shows the voltage behavior when electric generation was performed at the standard conditions. The voltage behavior instability was confirmed to be greater for the B-cell than for the A-cell.

Neutron radiography measurement was performed to analyze the difference in the A-cell and B-cell voltage behavior from the viewpoint of accumulated water. Figure 10 shows the images after 30 minutes of electric generation, at which point the accumulated water quantity is constant. More water was observed to have accumulated in the B-cell than in the A-cell. In addition, the results also clearly showed that more accumulated water was present in the lower part of the channels.

Figure 11 shows the change in the accumulated water when electric generation is performed at the standard

conditions. In the initial stages of electric generation, the accumulated water in both the A-cell and B-cell increases at a similar rate. However, after around 10 minutes of electric generation, more water tends to accumulate in the B-cell than in the A-cell, and the accumulated water quantity was observed to stabilize in both cells from around the 30-minute mark. This difference in the accumulated water quantity in the A-cell and the B-cell is due to the differences in the amount of condensation and the water discharge capability. In addition, the time period during which the B-cell exhibited unstable voltage behavior in Fig. 9 is the same as the time during which the quantities of water accumulated in the A-cell and B-cell diverge in Fig. 11. Therefore, the unstable voltage behavior of the B-cell is thought to have occurred because the large quantity of accumulated water in the B-cell impeded the gas supply.

Figure 12 shows the relationship between the standard deviation (SD) at 10-minute intervals of the voltage behavior shown in Fig. 9 and the average value of the accumulated water quantity during the same time period shown in Fig. 11. These results verified that the voltage behavior became unstable as the accumulated water increased.

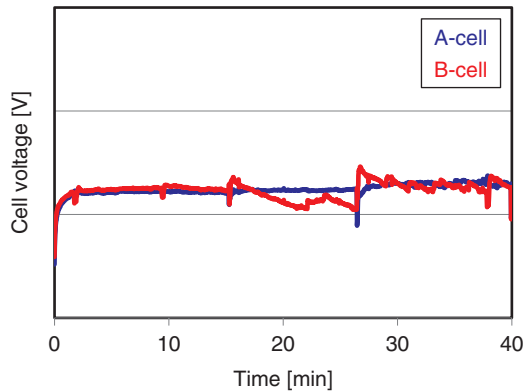


Fig. 9 Cell voltage of each cell under generation

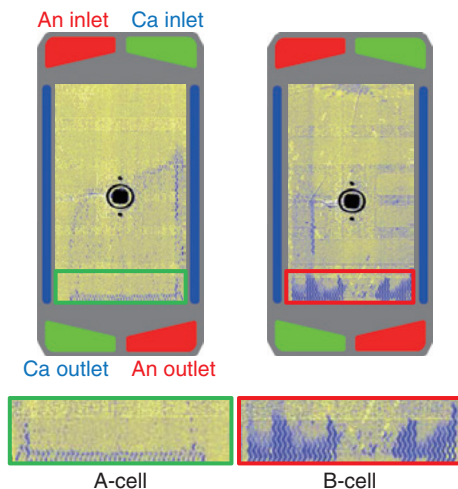


Fig. 10 Neutron radiography image of each cell

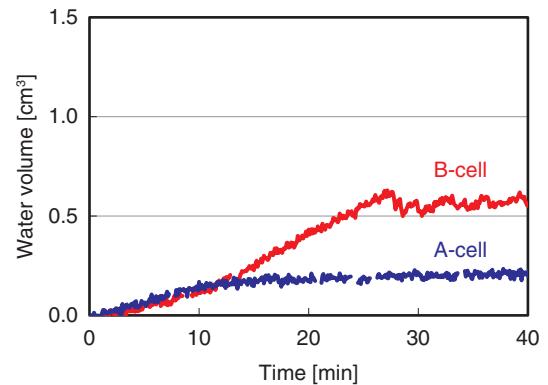


Fig. 11 Water volume transition for each cell under generation

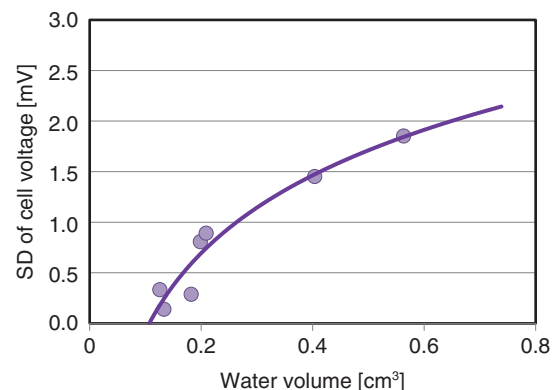


Fig. 12 SD of cell voltage at various water volumes

4.2.2. Results of observation in the in-plane direction of the cell

Figure 13 shows the neutron radiography image when electric generation is continued until the accumulated water quantity becomes constant at the standard conditions. Water was observed to be generated on the cathode side, and also to accumulate on the anode side. This is thought to be due to the effects of back diffusion whereby water moves from the cathode to the anode via the MEA. Figure 14 shows the results of quantifying the accumulated water in each gas channel. This clarified that more accumulated water was present in the B-cell than in the A-cell for both the anode and cathode sides. The separator structure is the same for both the A-cell and the B-cell, so the difference in the accumulated water quantity between the A-cell and the B-cell was determined to be due to the difference in the electric generation environment.

4.2.3. Examination of differences in accumulated water quantities in a one cooling layer per two cells structure

Examination focused on the environment temperature during electric generation in order to investigate the difference in accumulated water quantities between cells in a one cooling layer per two cells structure. Figure 15 shows the temperature of each separator during low-load electric generation. The temperatures of the A and C separators on both sides, which have cooling surfaces, are lower than the temperature of the B separator located in the center. In addition, the C separator

temperature is lower than the A separator temperature. This is thought to be because the C separator is located on the anode side and the A separator is located on the cathode side. That is to say, the C separator is located on the anode side that has lower overvoltage and heat generation, and also has a cooling surface, so this separator is the lowest temperature.

These results confirmed that the B-cell has a lower cell environment temperature than that of the A-cell, and is an environment in which condensation occurs more easily. This is thought to be why more accumulated water is present in the B-cell on both the anode and cathode sides. Accumulated water impedes the supply of gas to the cell, which further reduces water discharge performance, and is thought to result in unevenness in the quantity of gas supplied to each cell.

The cooling surfaces for the anode and the cathode differ in a one cooling layer per two cells structure, which makes it a challenge structurally to maintain a uniform environmental temperature in each cell. Therefore, in order to supply gas evenly to each cell during electric generation, it is necessary to actively discharge the accumulated water and reduce the difference in the quantities of accumulated water. In addition, accumulated water is present not only on the cathode side but also the anode side, which suggests that it is necessary to reduce the accumulated water at both the anode and the cathode.

4.3. Investigation of Application to Control Using the Actual Cell

The results obtained from the actual cell and the in-plane observation cell confirmed that the presence of accumulated water in the gas channels during electric generation and the difference in accumulated water quantities between cells resulted in uneven gas supply, which in turn produced instability in the cell voltage behavior. Therefore, countermeasure methods to reduce the accumulated water quantity were investigated using the actual cell.

4.3.1. Effects of increasing the cathode stoichiometry

The effects of increasing the cathode stoichiometry on reducing the accumulated water were investigated. Tests were conducted using a two-cell structure in

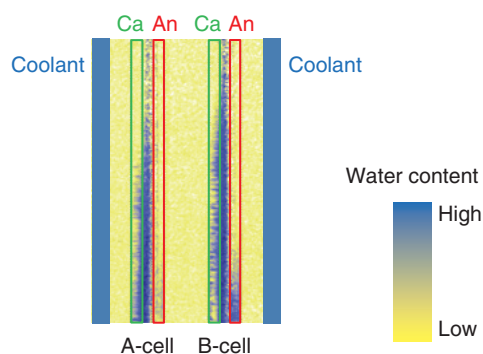


Fig. 13 Neutron radiography image of in-plane cell

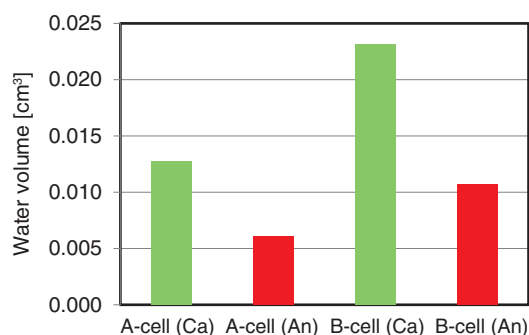


Fig. 14 Water volume of each channel

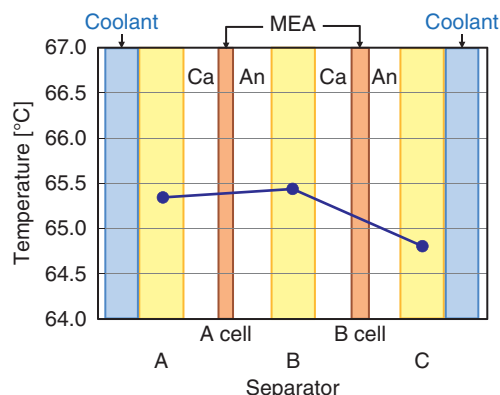


Fig. 15 Separator temperature of each cell

which the A-cell and B-cell perform electric generation simultaneously. Figure 16 shows the relationship between the voltage behavior and the stoichiometry in terms of the standard deviation. It was observed that the voltage behavior tended to stabilize at a stoichiometry of 3.4 or more.

The accumulated water at various stoichiometric quantities was measured to check the relationship between stabilization of the voltage behavior and the accumulated water. Figure 17 shows the neutron radiography image when electric generation is performed at various cathode stoichiometric quantities. The accumulated water in the electric generation area was observed to decrease as the stoichiometry increased.

Figure 18 shows the results of quantifying the accumulated water in the lower part of the gas channels of the electric generation area. This shows that the accumulated water quantity can be reduced by 50% by increasing the stoichiometry from 2.6 to 6.0. This is thought to be because the cathode flow rate increased, which increased the differential pressure and enhanced the water discharge performance.

Figure 19 shows the cell voltage and the resistance overvoltage when electric generation was performed while varying the stoichiometry. An increase in the resistance overvoltage was not observed, but enhanced electric generation performance was confirmed. These results

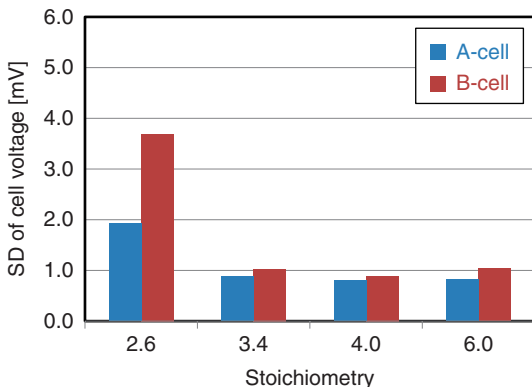


Fig. 16 SD of cell voltage at various stoichiometric quantities

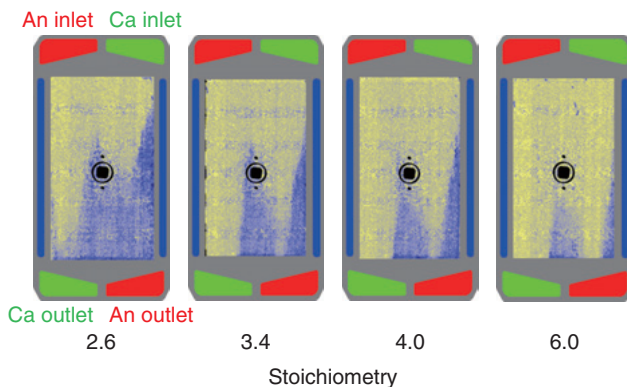


Fig. 17 Neutron radiography image at various stoichiometric quantities

confirmed that increasing the cathode stoichiometry reduces the accumulated water, which stabilizes the voltage behavior and also enhances electric generation performance.

4.3.2. Effects of increasing the coolant temperature

The effects of increasing the coolant temperature on reducing the accumulated water were checked. Figure 20 shows the relationship between the coolant temperature and the voltage behavior as the standard deviation. It was confirmed that the voltage behavior tended to stabilize at a coolant temperature of 72.5°C or more.

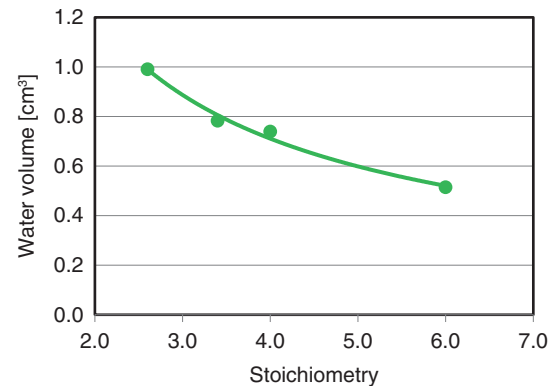


Fig. 18 Water volume at various stoichiometric quantities

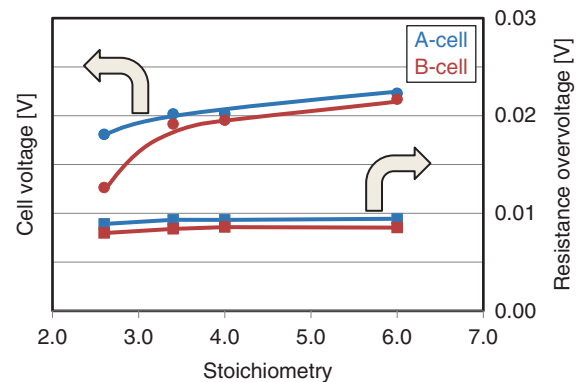


Fig. 19 Cell voltage and resistance overvoltage at various stoichiometric quantities

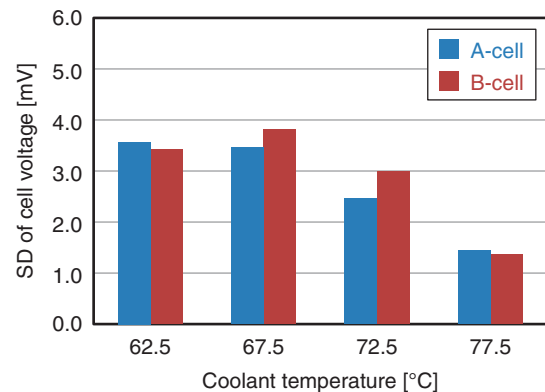


Fig. 20 SD of cell voltage at various temperatures

Figure 21 shows the results of quantifying the accumulated water in the lower part of the gas channels of the electric generation area. Increasing the coolant temperature was confirmed to reduce the accumulated water. This is presumed to be because the cell internal environment temperature rose, so the saturation vapor pressure also rose and the amount of condensation decreased.

Figure 22 shows the relationship between the cell voltage and the resistance overvoltage at various coolant temperatures. Unlike the case when increasing the stoichiometry, increasing the coolant temperature was confirmed to increase the resistance, which reduced the electric generation performance. This is thought to be because while increasing the cell temperature reduced the accumulated water in the gas channels, it also simultaneously reduced the water content of the MEA. This clearly showed that while increasing the coolant temperature effectively reduces the accumulated water, the rise in the resistance overvoltage results in a drop in electric generation performance.

4.3.3. Effects of reducing the accumulated water quantity by anode purge

The effects of anode purge performed in an actual vehicle on the accumulated water quantity were checked. Figure 23 shows the change in the standard deviation of the cell voltage during anode purge control. These results showed

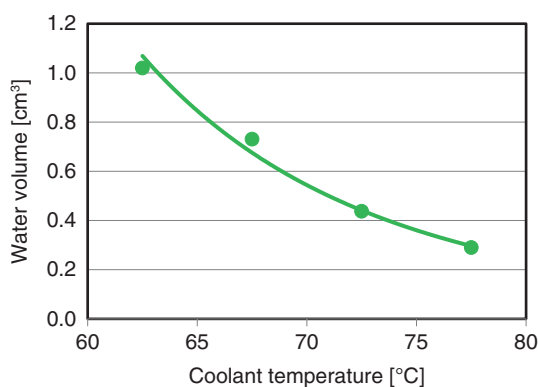


Fig. 21 Water volume at various coolant temperatures

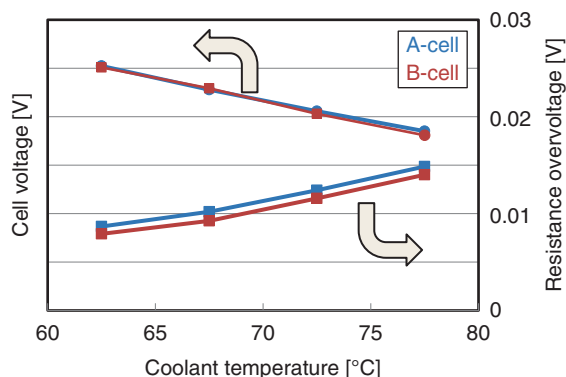


Fig. 22 Cell voltage and resistance overvoltage at various temperatures

that voltage behavior can be enhanced by anode purge.

Figure 24 shows the change in the accumulated water quantity in the lower parts of the A-cell and B-cell gas channels when anode purge control is started. Anode purge was confirmed to reduce the accumulated water in the B-cell to the same level as that in the A-cell.

In addition, Fig. 25 shows the effect of anode purge on the cell voltage. No drop in electric generation performance was observed due to anode purge. Therefore, it is thought that only the accumulated water in the gas channels was discharged, and the MEA water content was maintained.

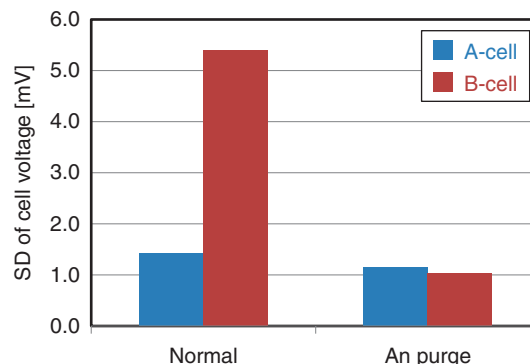


Fig. 23 SD of cell voltage before and after An purge

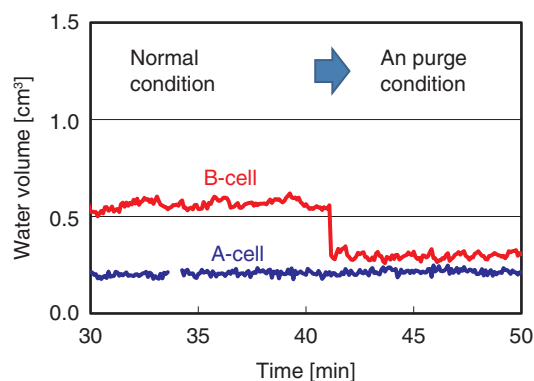


Fig. 24 Transition of water volume before and after An purge

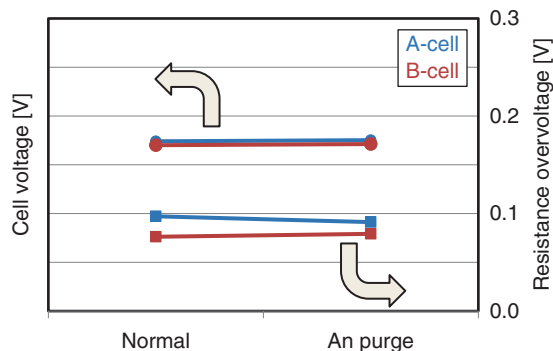


Fig. 25 Cell voltage and resistance overvoltage before and after An purge

These results clarified that anode purge effectively reduces the accumulated water, and can also help stabilize voltage behavior and maintain electric generation performance.

5. Investigation of Methods for Application to an Actual Vehicle System

Application of methods to reduce accumulated water to an actual vehicle system was investigated based on the above results. The results from the actual cell showed that increasing the cathode stoichiometry and performing anode purge are effective methods that can reduce the accumulated water without adversely affecting performance. When applying the above methods as control, the parameters were adjusted within the ranges that do not lower the electric generation efficiency or fuel economy.

Figure 26 shows the neutron radiography images during electric generation before and after the accumulated water countermeasures. It was confirmed that the countermeasures effectively reduce the accumulated water during electric generation.

Figure 27 shows the effects of this countermeasure control in terms of the relationship between the accumulated

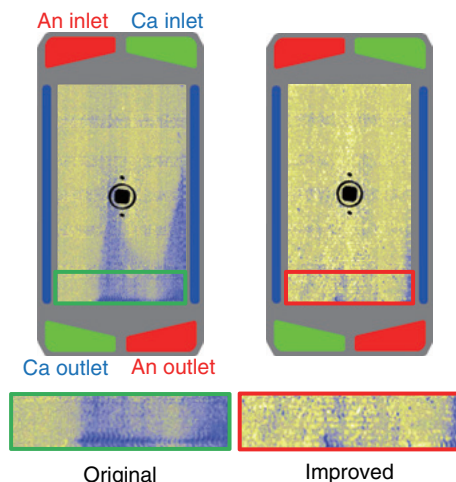


Fig. 26 Neutron radiography image of original and improved generation environment

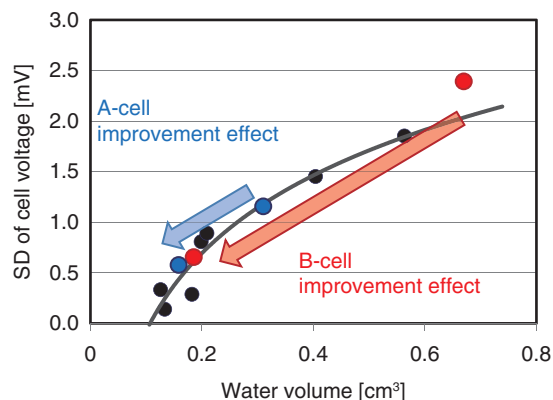


Fig. 27 Relationship of water volume and cell voltage

water and voltage behavior indicated in section 4.2.1. The countermeasure control enabled reduction of the accumulated water quantity in both the A-cell and the B-cell, and reduced the accumulated water quantity in the B-cell to nearly the same level as the A-cell. As the accumulated water quantity decreased, the stability of the voltage behavior also increased.

This countermeasure control controlled the accumulated water quantity and helped to stabilize the voltage behavior, thereby maintaining electric generation performance.

6. Conclusion

The accumulated water was observed by neutron radiography using an actual cell and an in-plane observation cell, the relationship between the accumulated water and the voltage behavior was determined, and methods of reducing the accumulated water were applied to an actual vehicle system. The results are as follows.

- (1) Neutron radiography was used to observe the water behavior during electric generation, and the effects of accumulated water on the cell voltage behavior during electric generation were clarified.
- (2) The water accumulation positions were isolated and analyzed using an in-plane observation cell, and it was proven that the difference in the accumulated water quantity between cells is due to the electric generation environment.
- (3) Methods of reducing accumulated water were investigated while checking the electric generation performance and accumulated water using an actual cell, and increasing the cathode stoichiometry and performing anode purge were found to be effective methods.
- (4) The above methods were applied to the control of an actual vehicle, and enabled stabilizing of voltage behavior while maintaining electric generation performance by simultaneously realizing both an appropriate MEA water content and accumulated water discharge.

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Projected Texture Stereo Vision System in Sunlight

Chiaki AOYAMA*

ABSTRACT

A binocular stereo vision camera system able to measure relative distance to a non-textured floor in real time without contact even under direct sunlight has been developed. A camera that alternately captures images with and without a projected texture at high speed, integrates the respective images within the pixels, and outputs the difference between them outside the pixels was developed for the system. The combination of this camera and a texture projector using a Class 1 laser that is safe for the eyes for pulse light emission has made it possible to measure the distance to a subject with a non-textured surface even under sunlight of 100000 lx if its reflectivity is at least 5%, achieving a measurement accuracy of 1 mm in a range between 0.9 m and 1.5 m.

1. Introduction

To help enable a bipedal walking robot to walk freely, it is necessary to measure the relative distance to the floor in real time.

Honda therefore developed a floor detection sensor for walking indoors⁽¹⁾ (Fig. 1) and applied the unit in its ASIMO robot. The floor detection sensor uses the light-section method, which is one method of realizing stereo vision. In this method, a line of light is projected, and distance is measured from its position on an image by means of triangulation⁽²⁾ (Fig. 2). The use of a projector makes it possible to measure distance even on a non-textured surface. In the normal light-section method, the distance to a flat surface is measured by scanning a single line of light. However, if a single beam is scanned while the robot is walking, the distance to the flat surface changes with the position of the robot during the scan, and thus cannot be measured. Multiple lines of light are therefore projected at the same time and the simultaneity of each line is maintained, making it possible to measure the distance to the flat surface while the robot is walking.

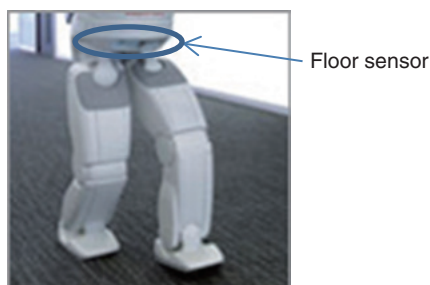


Fig. 1 Floor sensor of ASIMO

To help enable the lines to be detected using simple image processing, everything other than the lines of light is imaged as black, as shown in Fig. 3. A laser emitting narrow-band light is used as the light source, and the camera is fitted with a band pass filter that allows only narrow-band light to pass. Using near-infrared as the wavelength of the laser makes it possible to obtain an image of only the lines of light under fluorescent light, which emits almost no near-infrared light.

However, in sunlight, near-infrared light is extremely strong, and the texture of the floor can be seen, as shown in Fig. 4. If the texture of the floor is linear, the system is unable to distinguish the lines of light and the texture, and it therefore cannot be used close to a window that is admitting sunlight.

Increasing the intensity of the light source reduces the relative effects of environmental light, but this produces safety issues. Safety conditions for the eyes are stringent for light sources such as lasers that concentrate light on a single point when a lens is employed, and increasing their intensity therefore represents a challenge. In the case of light sources other than lasers, it is challenging to project a clear line

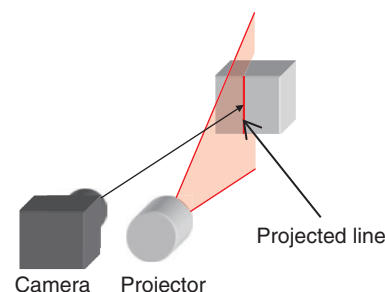


Fig. 2 Light-section method

* Fundamental Technology Research Center

texture regardless of distance, and the light-section method cannot be used.

The binocular stereo vision used in the head section of ASIMO, which does not use projected light, is safe for the eyes, but is unable to measure distance to non-textured surfaces. This is because when triangulation is used to determine distance, clear image characteristics are necessary for the determination of correlations between the images. Because a non-textured or virtually non-textured floor without clear characteristics is not a special case, it is not possible to use binocular stereo vision as a distance sensor in relation to a floor.

The issue of determining distance even when there is no texture can be resolved, if a distance of several meters is available, by employing a texture projector to apply a texture to the surface of an object⁽³⁾ (Fig. 5). However, in systems using conventional cameras, the projected textures become unclear under sunlight, and it is not possible to measure distance. (An example will be discussed below.) If it was possible to obtain images even in sunlight in which a clear texture was projected and that could be used in correlation processing for stereo vision, it would be possible to measure distance even to non-textured surfaces.

Given this, the following targets were established for system development:

- (1) Ability to measure distance to an irregular, non-textured surface within a range from 0.9 to 1.5 m with sufficient accuracy for control purposes
- (2) Ability to be used in sunlight of 100000 lx
- (3) Use of a light source that is safe for the eyes (conforming to Class 1)

This paper will discuss the principle of the system, its configuration, and the measurement results obtained.

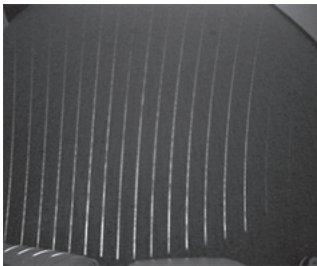


Fig. 3 Good image for floor sensor

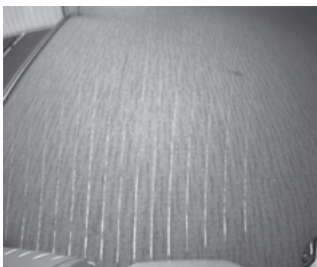


Fig. 4 Bad image for floor sensor

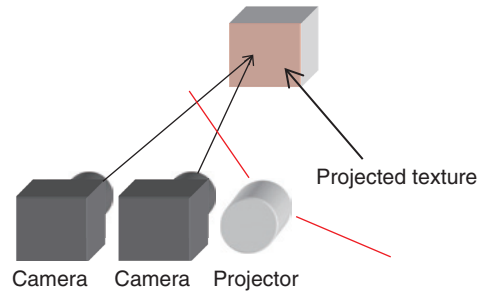


Fig. 5 Projector stereo camera system

2. Principle of System

The research discussed in this paper developed a projected texture binocular stereo vision system using two cameras and one or two texture projectors. The project also developed a camera and projector that would make it possible to see a texture under sunlight while maintaining safety for the eyes.

This section will discuss the principles underpinning the realization of increased resistance to environmental light in each of the elements of the system.

2.1. Difference Camera

In order to mitigate the effect of environmental light, the system compares the difference between images in which a texture has been projected and images in which a texture has not been projected, and does away with environmental light by canceling the shared environmental light in the images (Fig. 6). If, hypothetically, two images featuring only an environmental light component, with no projection, were taken consecutively and the differences between the images compared, there would be a difference of zero if no movement had occurred. In other words, a completely black image would be obtained.

If movement had occurred, because the position of the subject would have changed due to the time difference between the images, when the difference between the images was compared, it could not rigorously be judged as zero. In the case of images taken while a robot is moving, if the images are taken at intervals of 10 μ s or less, at the robot's walking pace the error due to misalignment of the images would be equivalent to or less than noise even at the boundaries between black and white sections, and could thus be ignored. However, in order to obtain images of sufficient brightness at an exposure of only 10 μ s, a light source able to instantaneously output an extremely bright light is necessary. It is challenging to install a light source of this type in a robot.

In order to obtain images of sufficient brightness while realizing a time difference approaching zero, the research therefore adopted a method in which a number of short-exposure images are integrated. Using this method, if the integration period is 5 ms, the error due to movement will be equivalent to or less than noise and can be ignored if

the time difference is 50 μs or less. The project was able to obtain a projector with a light emission interval of 100 μs and a light emission period of several μs , making it possible to obtain images of sufficient brightness to be integrated 50 times in a 5 ms period.

However, it is challenging to read and integrate images in 100 μs or less each time a short-exposure image is taken. This is equivalent to reading 100000 images per second, and such high-speed reading is only possible using a large, high-speed camera.

The research therefore focused on realizing the ability to integrate images within the pixels at high speed and without noise, conducting charge reading and integration operations within the pixels and outputting differences outside the pixels.

Figure 7 shows a simple model of the structure of a pixel, and Fig. 8 shows the flow of integration of the electrons within a pixel. In the center of the simplified pixel there is a photoelectric conversion section (a photodiode), and to the left and right there are charge accumulation sections (capacitors). Transfer gates control the transfer of electrons between the photoelectric conversion section and the charge accumulation sections.

Figure 8 shows the potential of the path between A and A' in the structure of the pixel. The higher in the figure, the higher the potential. Electrons travel in the direction of lower potential, that is, downwards. If there are areas of high potential on the transfer path, the electrons accumulate near these areas without crossing them.

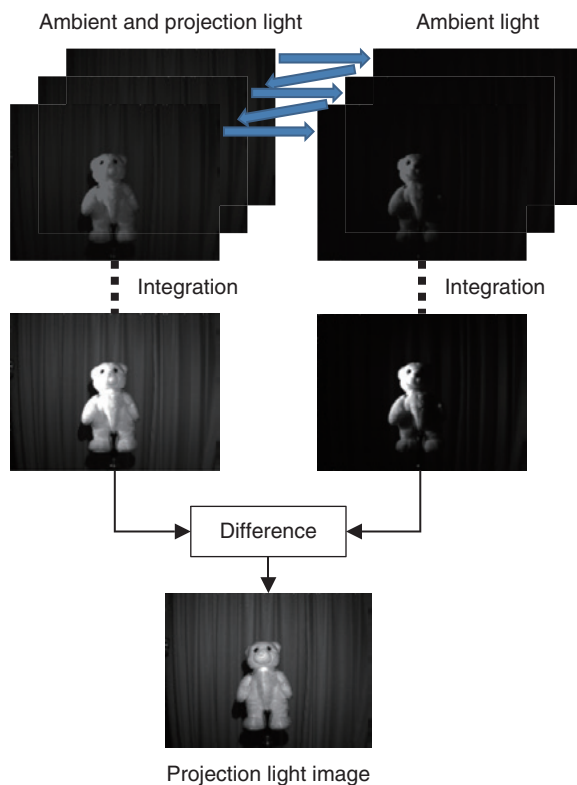


Fig. 6 Integration difference method

The series of integration operations commences with processing that discharges the electrons from the charge accumulation section, making the charge zero. Following this, the left and right transfer gates are opened alternately, and, depending on their state, the electrons produced in the photoelectric conversion section are transferred alternately to the left and right, and are integrated in the charge accumulation sections. After the integration processing has been repeated a number of times, a pixel signal is obtained by reading the level of the accumulated electrons. The integration of the electrons by alternation between the transfer gates can be repeated 100000 times or more per second, but the integrated pixel signals for the entire screen can only be read a maximum of 50 times.

Correction is applied to the signals that have been read to help ensure that inputs of the same light intensity are

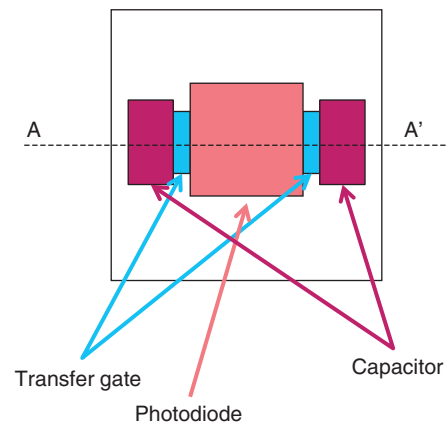


Fig. 7 Simple pixel model

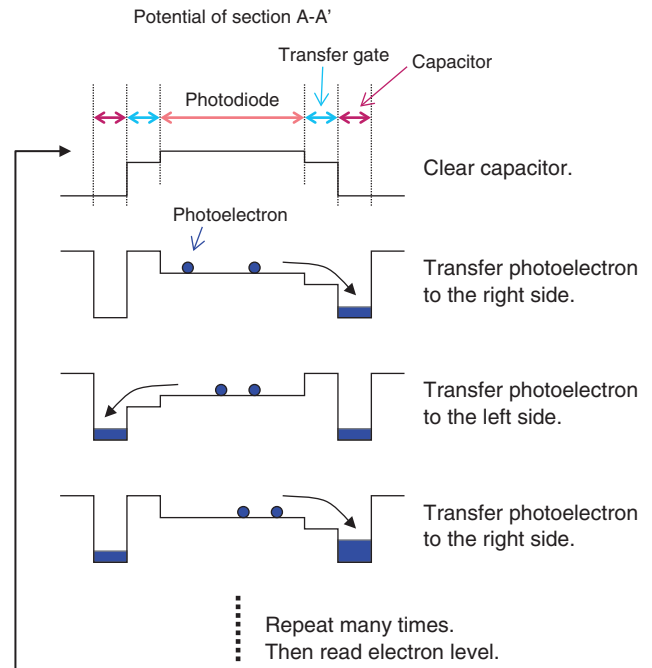


Fig. 8 Integration sequence

output at the same intensity. Following this, the difference between the images is identified. It is necessary to apply correction to the pixel signals, because if the signals were left uncorrected, differences in sensitivity and offset would result in differences in output for each pixel even if light of the same intensity had been incident on the entire screen. The application of correction means that output would be zero if there were no changes in the environment in which the images were taken.

It was possible to identify differences but not to apply correction within the pixels, and differences were therefore not identified within the image sensor.

2.2. Projector

Given that a light source able to emit narrow-band light at high power in a short period was necessary for the projector, a laser was selected for use. A diffuser was positioned in front to produce a broad light source despite the use of a laser, and to help ensure that light would not be concentrated on a single point even when concentrated by a lens. The use of the diffuser increases safety by a factor of several tens by helping to ensure that light is not concentrated on the retina.

The moiré method was employed as the texture projection method (Fig. 9). The use of a parallel grid pattern separated at

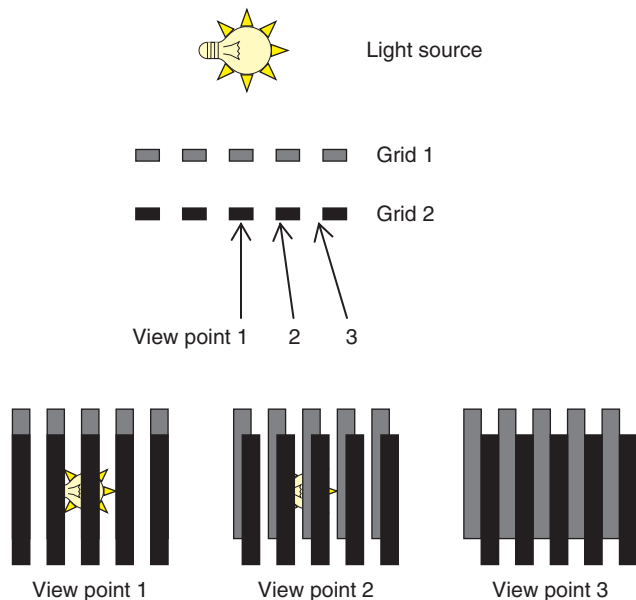


Fig. 9 Moiré method



Fig. 10 Moiré image

regular intervals means that the light source is either visible or obscured depending on the position, resulting in the projection of a striped texture. When seen from position 1, the grids appear to overlap and the light source is seen through slit-like gaps. Changing the position of viewing reduces the amount of the light source that can be seen, and from position 3 the slits obscure the view, and the light source cannot be seen. If a subject for imaging was present, it would be imaged as bright at position 1 and as dark at position 3. This means that when a white plate is positioned at the viewing positions, a vertical striped texture is projected (Fig. 10).

3. Difference Camera System

This section will provide an overview of the elements composing the difference camera system and the basic processing employed.

3.1. System Configuration

To create the system, one or two projectors were fitted in two difference cameras (Fig. 11). The image sensors had 640×480 effective pixels, with each pixel being $12 \mu\text{m}$ square. The baseline length was 170 mm, and the focal length of the lenses was 8.5 mm.

The difference cameras consisted of two camera heads, a camera control unit, and a camera interface unit (Fig. 12). The image signals output from the image sensors were sent to a PC via the camera control unit and camera interface unit. The projector emitted light from a laser diode (LD) in response to instructions from the camera control unit.

After the integrated images were output from the cameras and linear correction had been applied to them by the PC, the differences between the images were identified and distance information was obtained by stereo vision processing.

3.2. Pixel Circuits

The circuits within a pixel are divided into a photoelectric conversion section, transfer and drain gates, a charge accumulation section, and reading circuits (Fig. 13). The photoelectric conversion section converts light into electrons. The transfer gates connect to the charge accumulation section, where the electrons produced are accumulated (integrated).

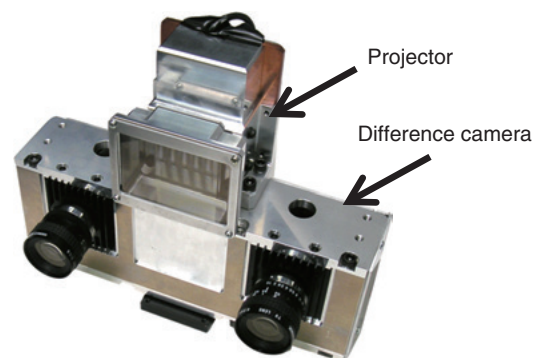


Fig. 11 Difference camera system

The drain gate is connected to the power source, where the electrons produced are extinguished. The electrons accumulated in the charge accumulation section are converted to magnitudes of current by the reading circuit and read out to the exterior of the pixel. The sensors developed for this project have four charge accumulation sections, and two of these are used when the sensor is operating as a difference camera.

The signal read out to the exterior of the pixel is converted from analogue to digital in the sensor, and output to the exterior as a 14-bit digital signal.

3.3. Pixel Configuration

Each pixel is a square with $12\ \mu\text{m}$ sides. The photoelectric conversion section is positioned at the focal point of a micro lens. The use of a pinned photodiode structure for the photoelectric conversion section reduces noise production. The electrons produced in the photoelectric conversion section are sent to any of the four charge accumulation sections via the transfer gates or to a drain via the drain gate.

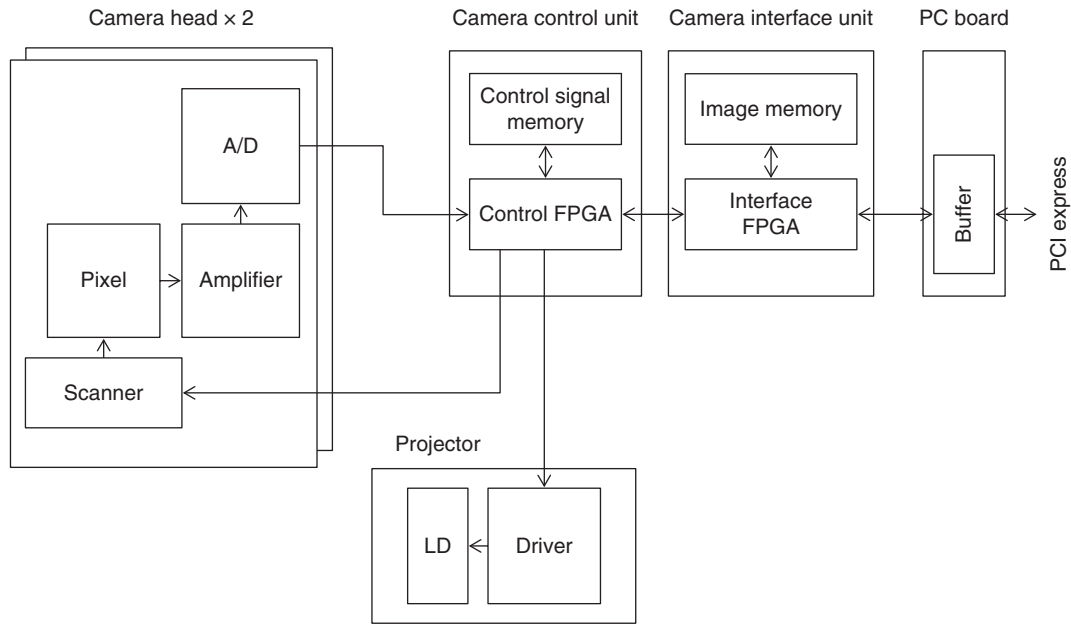


Fig. 12 Difference camera block diagram

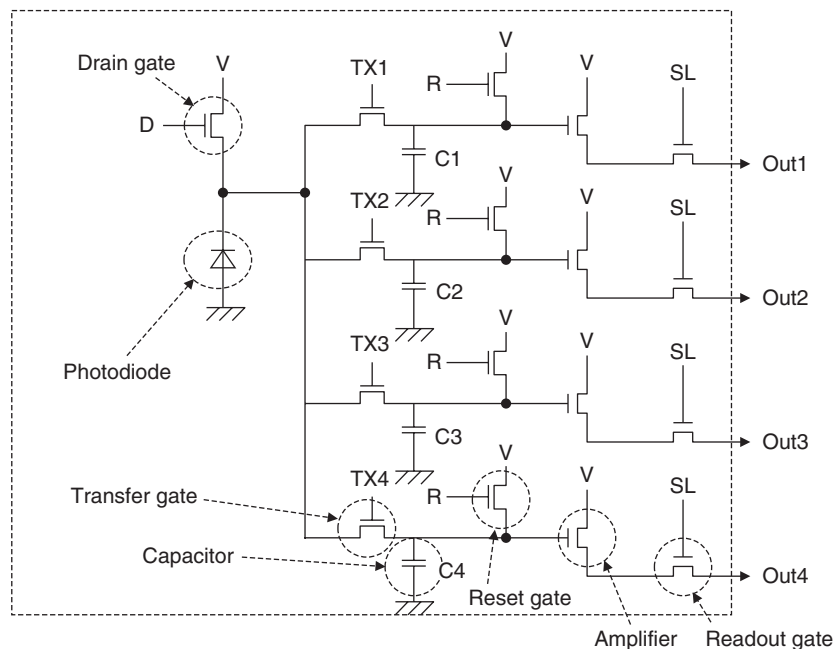


Fig. 13 Pixel circuit

3.4. Projector

In the projector, a diffuser is positioned in front of a semiconductor laser, and two grids are positioned beyond the diffuser. A semiconductor laser with a peak output of 300 W, a wavelength of 800 nm, and a minimum light emission interval of 100 μ s is used as the light source for the projector. The grids used to project a striped texture are vertical stripes with line widths of 0.25 mm and intervals of 0.25 mm transferred onto glass plates. There is a 20 mm gap between the grid plates.

For binocular stereo vision, two types of projector (projecting vertical stripes and diagonal stripes) are combined to help prevent mistaken correlation. Using vertical stripes only, identical textures appear repeatedly in the search range when a left-right correlation is attempted, and a single distance cannot be determined. The addition of diagonal stripes restricts the texture to a single example, making it possible to determine the distance. When the distances are roughly known in advance, the search range can be narrowed and vertical stripes alone are sufficient for the determination of distance.

3.5. Shutter Control

One exposure period is 100 μ s, and there are 50 exposure periods (Fig. 14). The time from commencement to completion of exposure (the exposure period) is 5 ms. The transfer gates are opened for a period that allows for the transfer of a sufficient number of the electrons produced by the received light pulse, and for the remaining time the electrons are discarded via the drain gate.

3.6. Camera Image Signal Processing

To help ensure that input of the same light intensity is output at the same light intensity, each charge accumulation section is corrected via broken-line approximation. By means of this correction, the variation in output values between pixels

becomes equivalent to or less than the shot noise generated by photon number fluctuation, which cannot be removed.

Differences are identified for linearized signals. Negative results are sometimes obtained when differences are identified, and an offset is applied to results for difference to help prevent this.

3.7. Stereo Vision Processing

Following distortion correction, conversion to match epipolar lines, and brightness correction, correlation processing is applied. The sum of absolute difference (SAD) is employed in order to identify correlations. The domain for identification of correlations is 13 horizontal pixels by 11 vertical pixels.

Brightness correction normalizes the luminance of the central pixels based on the average luminance of the domain. By making the intensity of domains with a low reflectivity (dark domains) and domains with a high reflectivity (bright domains) identical, this correction makes it possible to judge the reliability of SAD correlation processing results by means of a fixed threshold value.

Domains that are too dark are not corrected because this would increase noise to too great an extent.

4. Distance Accuracy of Camera System

The distance accuracy of the binocular stereo vision camera system was measured.

The subjects of the measurements were black flat surfaces and white flat surfaces. The reflectivity of the black flat surfaces was 5%, and the reflectivity of the white flat surfaces was 83%. A halogen light was used as ambient light. The light was adjusted to irradiate the subjects at a power corresponding to 100000 lx at close to 800 nm. The light intensity for the projection was 70%, the maximum

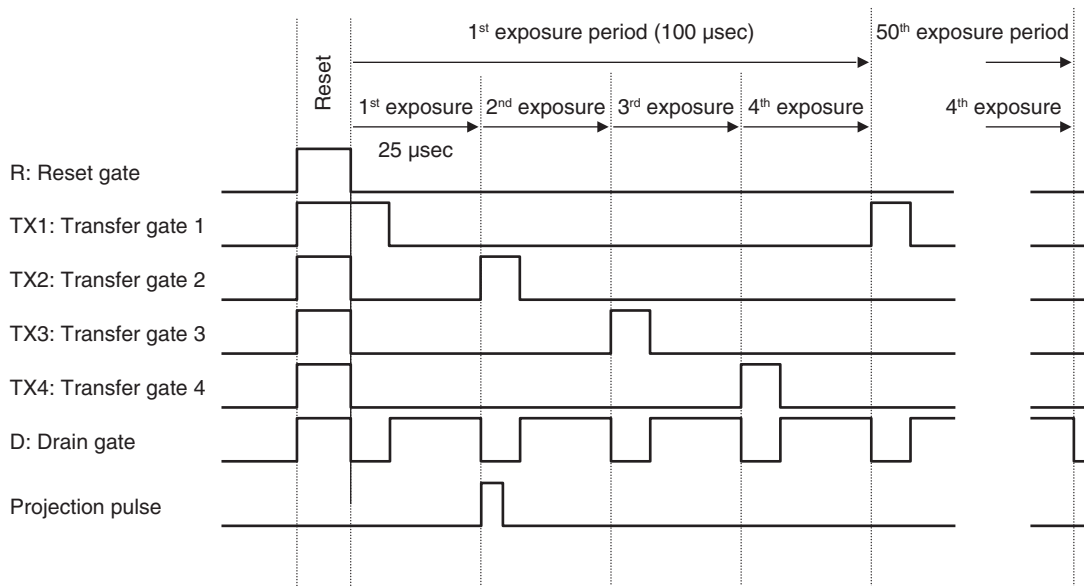


Fig. 14 Shutter timing

intensity allowed for a Class 1 laser.

The distance between the plane and the camera was varied in 2 cm increments using a linear stage. Among the distance data detected by the system, the average value within a domain extending 20 cm in all directions in actual space close to the center of the plane was taken to be the distance to the plane.

Figure 15 shows the images used in measurements and images taken using an ordinary camera with an exposure timing that continued to expose the image after the period of light emission. When the difference camera was used, identical images were captured irrespective of the presence

or absence of ambient light. When the normal camera was used, either the image was not captured due to saturation by ambient light, or, when the sensitivity was reduced to help prevent saturation, the texture became unclear. Because integration processing was conducted for the normal camera for longer than the period of emission of light by the light source, when the brightness of the image was made the same using only projected light, if environmental light was present, the image became brighter. This is to say, the image was significantly affected by environmental light.

Figure 16 shows the relationship between distance and error in measurement results. Error is expressed as

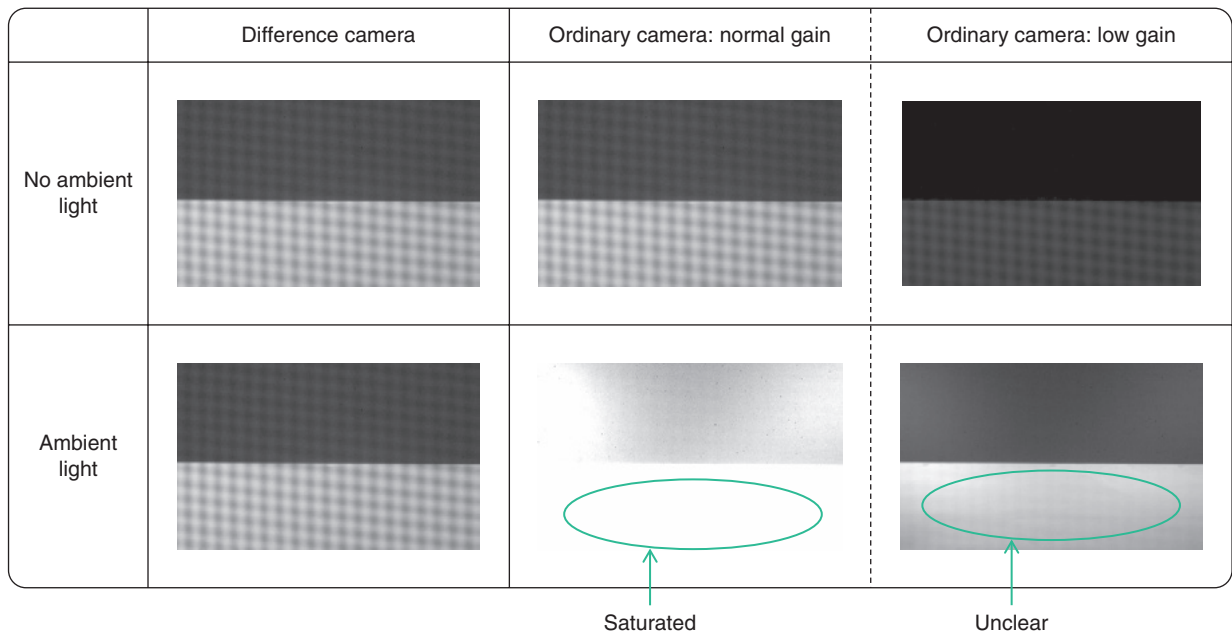


Fig. 15 Difference image and ordinary image

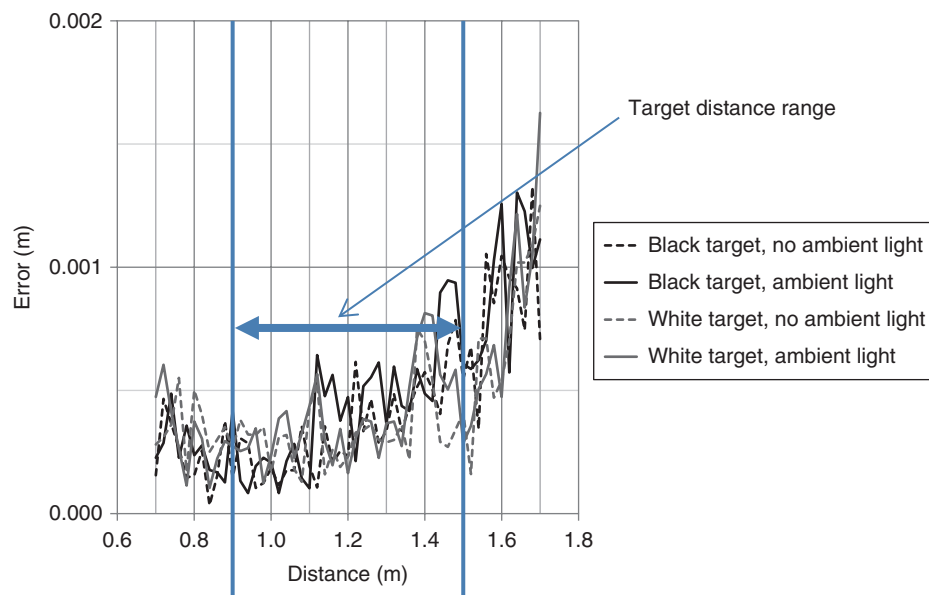


Fig. 16 Measurement error

3σ (three times σ , the standard deviation of distance data measured ten times), added to the absolute value of the difference between the average distance and the actual value of distance. Results for the black flat surface with no environmental light are shown as the black broken line, and results with environmental light as the black solid line; results for the white flat surface with no environmental light are shown as the gray broken line, and results with environmental light are shown as the gray solid line.

The graph indicates that in the range between 0.9 m and 1.5 m, error is 1 mm or less irrespective of reflectivity or the presence of environmental light. This represents sufficient accuracy for walking control.

5. Example of Application of Difference Camera

Because the difference camera makes it possible to remove ambient light and obtain images using only projected light, one application for the camera other than as a distance sensor is in enhancing the resistance of recognition functions to environmental light. The camera would make it possible to obtain performance of the type offered by appropriate image processing under indoor illumination even outdoors.

The grids were removed from the projector to do away with the texture, and images were taken outdoors with the subject in sunlight from different angles. The facial recognition processing used in ASIMO was applied to these images, and facial recognition results for images taken with a normal camera and the difference camera images were compared (Fig. 17).

The square green frame shows the domain for facial recognition results. The green or red bars at the left of the images show the degree of match with the pre-recorded subject (that is, the “score”). The longer the bar, the better the match. If the bar is green, the face is judged as being the same person; if the bar is red, the face is judged as being a different person.

The results show that the difference images are not affected by sunlight, and the facial recognition results are similarly unaffected. The ghosting visible in the upper section of the image taken under back-lighting by the normal camera has also been removed in the difference image.

6. Conclusion

A difference camera system able to measure distance to a non-textured surface under sunlight has been developed.

The system achieves a distance accuracy of 1 mm in the range from 0.9 m to 1.5 m, sufficient for use in robot walking control, irrespective of the reflectivity of the subject or the presence of environmental light.

When used with non-textured projected light, the system also enhances the accuracy of facial recognition.

Acknowledgements

The author wishes to express his sincere gratitude to Professor Shoji Kawahito of Shizuoka University for his guidance in the development of the image sensor discussed in this paper.

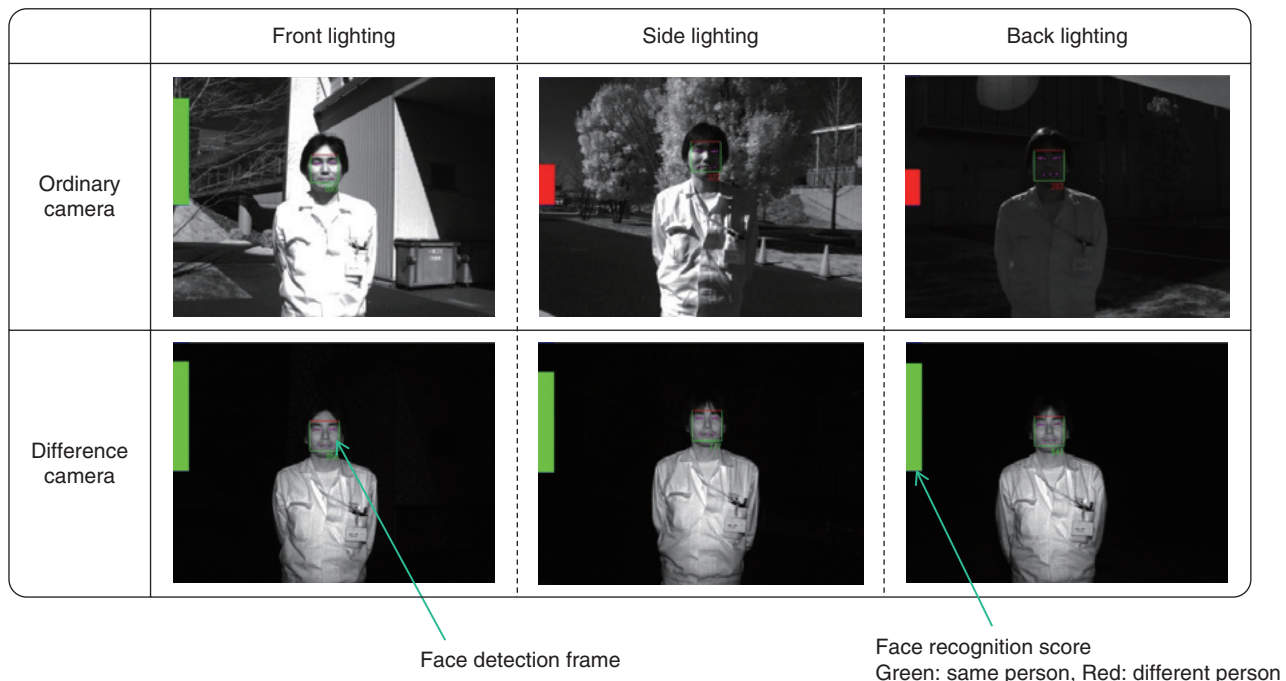


Fig. 17 Face recognition results

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Chiaki AOYAMA

Development of Prediction Technology for Resonance Stress of Exhaust System by Considering Nonlinear Properties of Spherical Joints

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ABSTRACT

A highly accurate and versatile prediction technology that provides results equivalent to stress measurement in the vehicle condition was developed. The purpose of this technology was to predict high-cycle resonance stress of the exhaust system in the fully assembled state. The phenomenon of high-cycle vibration of the exhaust system was divided into the two systems of an input system and a response system. The input system was accurately reproduced by considering the multiple degrees of freedom and multiple orders of phase of the engine excitation force transmitted to the exhaust system. The response system was experimentally modeled by conducting standalone tests of the spherical joint and obtaining the stick-slip characteristics. Application of the above enabled accurate prediction for various exhaust system shapes.

1. Introduction

Recently, enhanced fuel economy, emissions and output power performance are being demanded of automobiles. Lighter weight and good heat-up characteristics are demanded of exhaust systems in order to enhance these performances, but at the same time, high reliability is also required.

One issue concerning exhaust system reliability is high-cycle fatigue breakdown due to resonance phenomena induced by the engine excitation force, and designs that take this issue into account are necessary.

In addition, many vehicle types share the same hot-end (parts mainly including the catalyst) from the viewpoint of parts sharing, but the cold-end (parts mainly including the chamber and the silencer) exhaust system variations are tending to increase, and measures to support increased development and test process times are demanded. At the same time there are also demands for shorter development periods, and the development of Computer Aided Engineering (CAE) technology that can accurately predict high-cycle vibratory stress in a short time in the drawing stage is needed.

Previous Finite Element Method (FEM) analysis of exhaust systems in the fully assembled state consisted of

performing frequency response analysis using the FEM CAE software NASTRAN, and comparing these analysis results with the results of stress measurement in the vehicle condition for use in investigating countermeasures.

As an example, Fig. 1 compares the stress results obtained by the previous analysis method with the results of stress measurement in the vehicle condition. While there are noticeable differences in the calculated and measured stress peaks in each mode, the level is such that enables use for investigation of countermeasures.

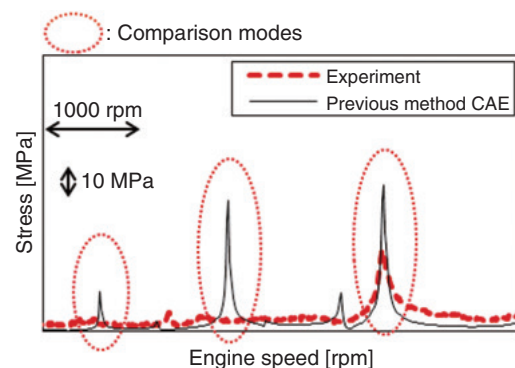


Fig. 1 Comparison of experiment and CAE by previous method

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However, absolute value evaluation of stress is needed to determine the feasibility of specifications in the drawing stage, and increasing this accuracy is an issue.

Three issues for promoting accurate analysis of the exhaust system assembly using FEM are appropriate modeling of couplings between parts, accurate modeling of the engine excitation force, and creation of mesh models that accurately reproduce responses.

In particular, setting appropriate part coupling conditions has been an issue for some time, and especially modeling of the spherical joint that acts to absorb the engine oscillation (Fig. 2). The spherical joint uses a spherical gasket to seal the exhaust gas, and this spherical gasket has characteristics such that the stick state is maintained at small inputs. However, when a certain input or more is applied, the slip state results and the gasket slides with some friction.

As examples, Fig. 3 shows an image of a shaker excitation test for an exhaust system with a spherical joint, and Fig. 4 shows the change in the Frequency Response Function (FRF) due to differences in the excitation input level. In consideration of the input transmission characteristics, the input point was located near the center of gravity of the exhaust pipe used in this study, and the response point used the average value of seven locations

spread over the entire exhaust system to equalize the response effect for comparison.

The results in Fig. 4 confirm that the response level decreases as the input level increases. This indicates that the spherical joint enters the slip state, so the exhaust system is divided and the rigidity drops. The change in the response level is closely related to the stress level, so to increase the stress accuracy it is necessary to construct a model in

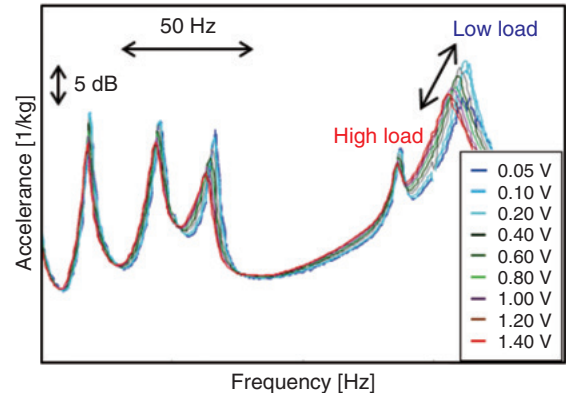


Fig. 4 FRF characteristics under different input conditions

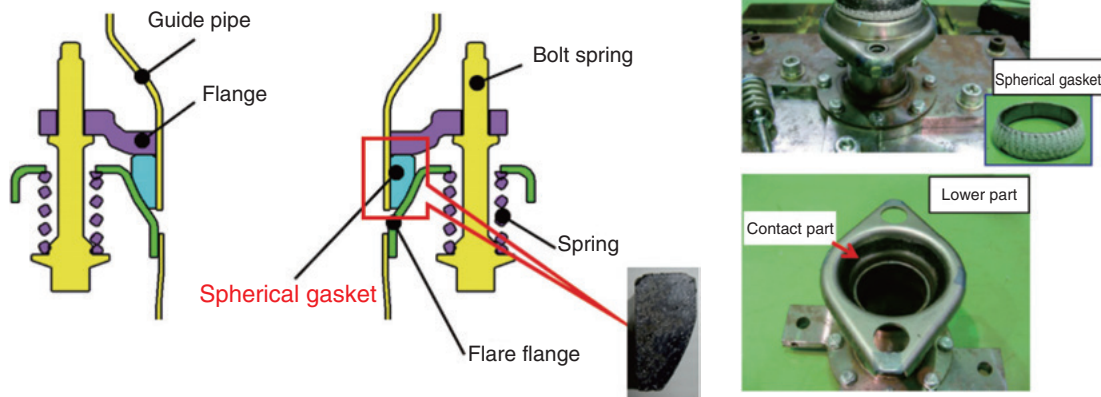


Fig. 2 Section and parts of spherical joint

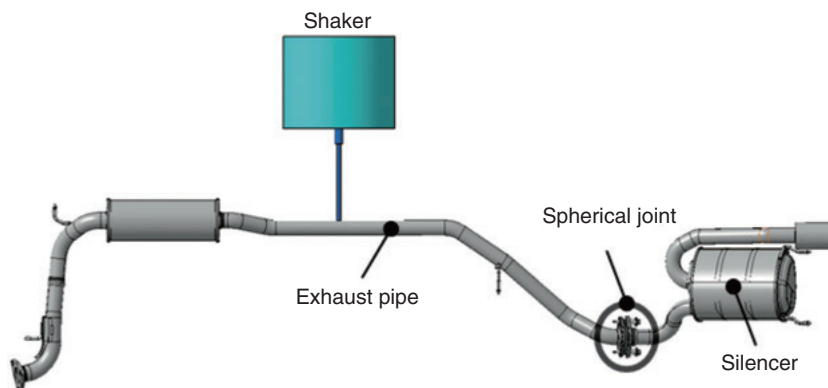


Fig. 3 Shaker test of exhaust system with spherical joint

which the stick and slip behavior of this location changes in accordance with the change in input. To construct this model, friction needs to be considered in the spherical joint model, but the previous analysis method uses the frequency axis, and was unable to take into account the change in the friction behavior over time.

Previous studies have discussed the influence of spherical joint behavior on high-cycle vibration behavior, and modeling has also been attempted⁽¹⁾⁻⁽³⁾. However, there is no known analysis method that takes into account the time-base change in behavior, and the subject frequencies have been limited to relatively low frequencies, and the high-frequency range that is the subject of this study has not been covered.

This paper describes the method of increasing the CAE accuracy and developing stress prediction technology for high-cycle vibratory stress on the exhaust system that is equivalent to stress measurement in the vehicle condition. This is achieved by creating a model that takes into account the nonlinear properties of the spherical joints, modeling the engine excitation force, creating an accurate mesh model, and applying these models to transient response analysis that performs analysis on the time axis.

2. Development of Analysis Method

The parts subject to analysis were the close-coupled converters, exhaust pipe and silencer that comprise the exhaust system in the fully assembled state. This study aimed to develop an analysis method using the simple exhaust system for verification shown in Fig. 5.

Spherical joints are used as the coupling between parts, so an experimental model was defined that considers the friction on the sliding surfaces.

To consider friction it is necessary to understand the changes in overall exhaust system behavior over time, and transient response analysis is given as this analysis method.

However, mesh models have many nodal degrees of freedom, so transient response analysis using all nodal degrees of freedom increases the calculation load. To address this issue, modal reduction was performed on the mesh model to reduce the degrees of freedom, and

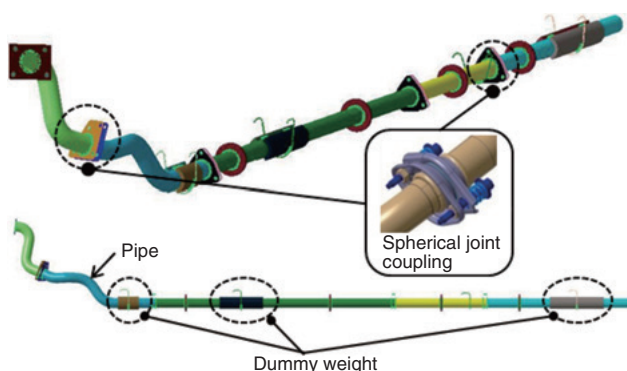


Fig. 5 Simple exhaust system layout for verification

a modally reduced model was assembled using a mode synthesis method with good analysis efficiency. The overall exhaust system behavior was then calculated from the results of dynamic analysis using this assembly model, and the displacement output was applied to a detailed mesh model to obtain the stress.

2.1. Construction of Models

2.1.1. Experimentally modeling the spherical joint

When modeling the spherical joint, it is necessary to consider the two characteristics of stick characteristics during small input, and slip characteristics during large input. These characteristics were obtained by creating a test jig with single pipes welded to a spherical joint, and measuring the rotational stiffness and the friction torque by a shaker excitation test. Figure 6 shows the test equipment used to obtain the friction characteristics of the spherical joint. This study attempted to experimentally model the stick-slip characteristics by performing calculations using an analysis model similar to the test jig, and identifying the spring constant during stick and the friction torque during slip from the test results.

The friction characteristics were obtained by performing a shaker excitation test with two rotational degrees of freedom in the topple direction and one rotational degree of freedom in the direction of rotation around the axis, and obtaining the hysteresis loop. The shaker excitation frequency should be set so that the jig resonance does not influence the friction torque. To avoid this, the FRF resonance frequency was confirmed by a hammering test, and the results (Fig. 7) confirmed that there was no jig resonance at 100 Hz, so that frequency was used for the excitation test.

Figure 8 shows the friction measurement results. Stick characteristics are exhibited during small input, but slip characteristics appear at a certain input or more, creating a hysteresis loop. The slope during small input expresses the gasket stiffness in the stick state, and the hysteresis loop amplitude expresses the friction torque in the slip

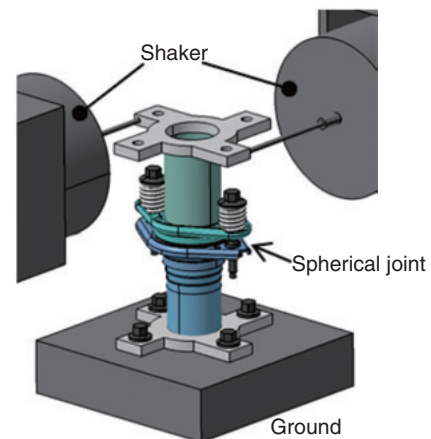


Fig. 6 Test equipment for measurement of friction characteristics

state. These results were used to calculate the stick spring constant and the slip friction value. At the same time, the effect of different inputs on the friction torque was checked, and it was found that the displacement increases as the input increases after slip, but the friction torque indicates a constant value.

Next, Fig. 9 shows an outline of the spherical joint analysis model. In this model, when the input is smaller than the friction torque, the stick model shown on the right side of Fig. 9 does not slide, and only the stick model that expresses the gasket stiffness shown on the left side of Fig. 9 is used. When the input is larger than the friction torque,

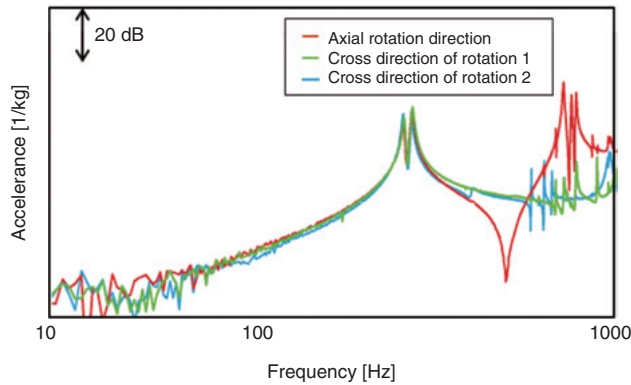


Fig. 7 FRF characteristics of spherical joint

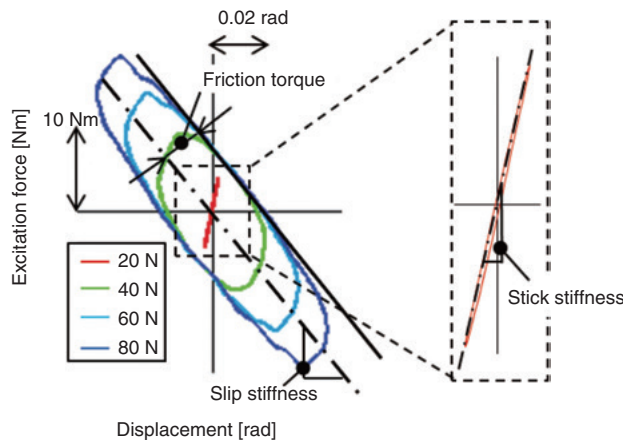


Fig. 8 Method of extracting spherical joint characteristics

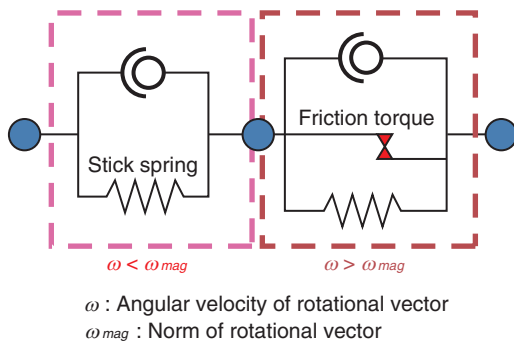


Fig. 9 ADAMS model outline of spherical joint

the slip friction torque is added to the stick model, and sliding occurs. Here, it is necessary to use the total friction force of the three rotational degrees of freedom, so a friction ball model was defined as shown in Fig. 10. The moments of the three rotational degrees of freedom of this model are expressed by the following equations.

$$T_x = T_{x0} \cdot \frac{\omega_x}{\omega_{mag}}$$

$$T_y = T_{y0} \cdot \frac{\omega_y}{\omega_{mag}}$$

$$T_z = T_{z0} \cdot \frac{\omega_z}{\omega_{mag}}$$

T_{x0}, T_{y0}, T_{z0} : Friction around the x, y and z axes

T_x, T_y, T_z : Moment around the x, y and z axes

$\omega_x, \omega_y, \omega_z$: Angular velocity around the x, y and z axes

ω_{mag} : Norm of angular velocity

Figure 11 shows an example comparing results with the results calculated by the mechanics analysis software ADAMS using the above models and the experiment results. Comparison of the change in behavior and the slip torque for different inputs shows that the change in behavior relative to the input and the friction torque width in the

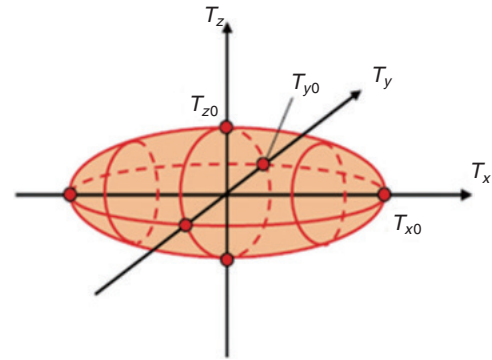


Fig. 10 Friction ball model

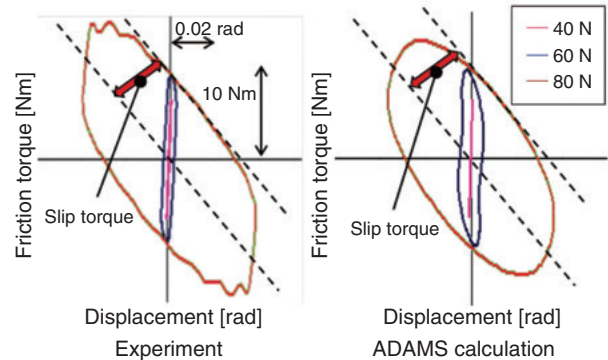


Fig. 11 Comparison of experiment and ADAMS calculation

slip state calculated by this model accurately reproduce the vehicle condition.

2.1.2. Modeling the engine excitation force

The major orders of engine vibration contribute greatly as the main excitation source for high-cycle vibration of the exhaust system. The vibration characteristics transmitted from the engine, which is an excitation source, to the exhaust system are mainly the excitation of the combustion pressure, the excitation of the reciprocating system inertial force, and excitation due to power plant resonance. One method of estimating the excitation force is to calculate the reciprocating system behavior and obtain the power plant behavior using FEM. However, this method takes time for model creation and calculation, so it is not suitable for this analysis. Therefore, this study calculated the input boundary conditions by measuring the excitation force of an actual engine.

To accurately measure the engine excitation force, the acceleration of the exhaust system flange on the engine side was measured using 3-axis accelerometers. However, engine displacement has three translational and three rotational degrees of freedom, so rotational behavior cannot be measured by normal 3-axis accelerometers. Therefore, accelerometers were located at the three flange bolt connection points shown in Fig. 12, simultaneous measurement was performed, and the rigid body motion with phase of each point was estimated with three translational and three rotational degrees of freedom from the measured three points with three translational degrees of freedom using the least squares method.

Normal sensors cannot perform stable measurement due to the effects of noise within the engine bench. To minimize the effects of this noise, accelerometers with charge amplifiers were used to amplify the signal on the sensor side to reduce the effects of noise.

All loads governed by the combustion pressure from idling to the red zone and motoring conditions governed by the inertial force were measured as the operating conditions. The excitation force of major orders that contribute to

exhaust system resonance were calculated from each measured operating condition.

2.1.3. Creating a mesh model of the exhaust system

To accurately reproduce the parts subject to analysis, the mesh model was modeled using shell mesh tetrahedral first order elements, and a mesh size was selected that can adequately reproduce the shape. Reducing the element size increases the accuracy, but the size was adjusted from the viewpoints of shortening the calculation time and reducing the calculation load on the analysis machine. The plate thickness and material characteristics extracted from the drawing values were defined as the physical property values of this mesh model. In addition, to increase the convenience of modeling, the Rigid Bar Element2 (RBE2) was used for welded joint elements and the Rigid Bar Element3 (RBE3) was used for the concentrated mass definition to reduce the number of elements and strive for simple modeling.

As an example of verifying the accuracy of this model, the correlation with hammering test results in the vehicle condition was obtained using Mode Assurance Criterion (MAC). The subject correlation frequency range was up to 450 Hz in the exhaust system assembly. Figure 13 shows the comparison results. These results show that a high correlation is obtained.

2.2. Development of a Dynamic Analysis Method

Transient response analysis using ADAMS was performed as the dynamic analysis method as described below.

Dynamic analysis that takes into account the friction of sliding parts was performed by modally reducing the mesh model and then assembling this reduced model on ADAMS using the spherical joint model. Figure 14 shows the analysis procedure using ADAMS. The assembly resonance characteristics can be accurately reproduced up to the high-frequency range by increasing the upper limit frequency of the modally reduced frequencies of each analyzed part. In consideration of shortening the calculation time and reproducibility of the assembly resonance characteristics,

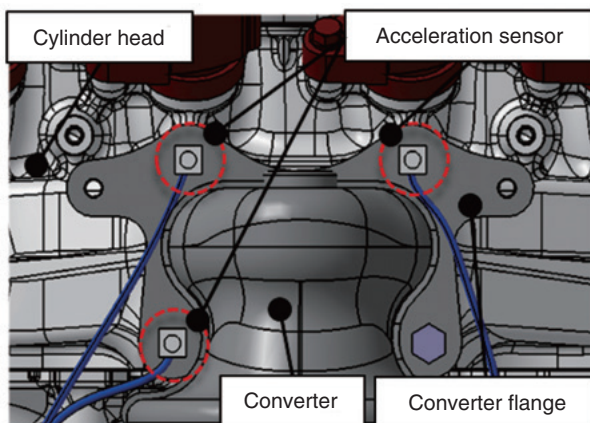


Fig. 12 Acceleration sensor installation locations

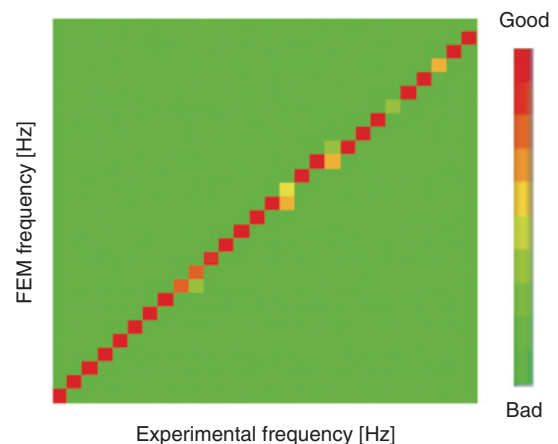


Fig. 13 MAC correlation results

this study adjusted the modally reduced frequencies of the analysis model, and set an upper limit of 1 000 Hz. As an example of verifying the accuracy of the reduced model, assembly eigenvalue analysis was performed separately by NASTRAN in consideration of all nodal degrees of freedom of the mesh model and by ADAMS using the modally reduced model, and the eigenvalue difference was confirmed. Figure 15 shows the comparison results. These results show that the maximum eigenvalue difference was 3.5 Hz, indicating that the reduced model has sufficient accuracy.

This reduced model was assembled on ADAMS using the spherical gasket model as the coupling condition. Figure 16 shows the ADAMS assembly state.

Regarding the input excitation data, waveforms that take into account the displacement excitation and phase of each measured order component were added together on ADAMS and applied as the time-base input waveform.

Stress recovery was performed by NASTRAN transient response analysis based on the displacement output of each

reduced model obtained by transient response analysis performed under the above conditions, and the stress at each evaluation point was evaluated.

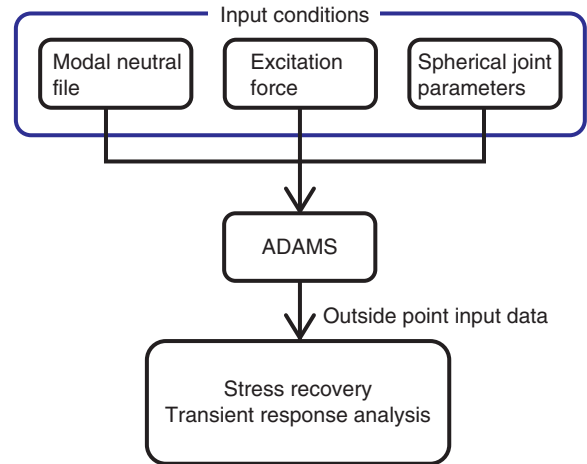


Fig. 14 Analysis procedure using ADAMS

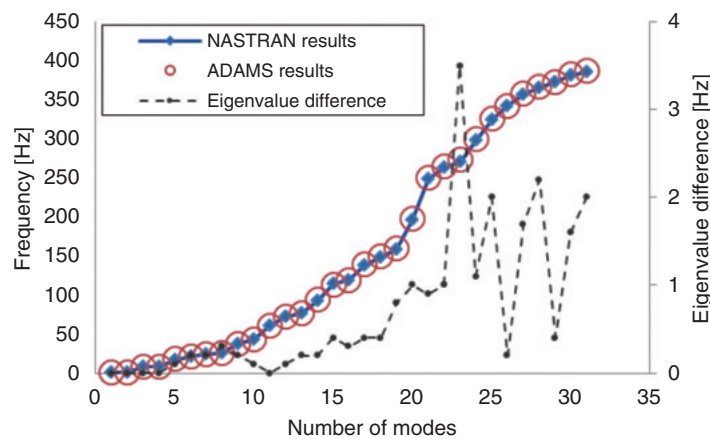


Fig. 15 Comparison of NASTRAN and ADAMS eigenvalue results

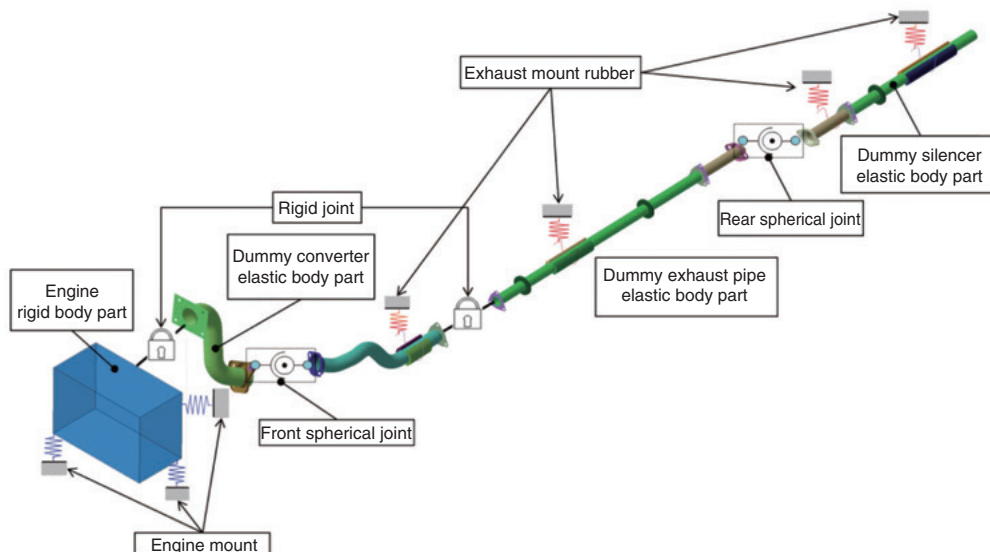


Fig. 16 ADAMS assembly model

3. Verification of Analysis Results

3.1. Verifying the Validity of the Developed Method

The graph shown in Fig. 17 compares the stress results obtained by the previous and new analysis methods and by testing of an actual vehicle as verification of the analysis results. Comparison of the stress peak in each mode shows that in modes where the previous analysis method indicated that high stress was generated, the new analysis method indicates the same values as the actual testing. This shows that force transmission is divided by the spherical joint stick-slip phenomenon, and indicates that the newly developed spherical joint model is a valid model.

3.2. Verifying the Versatility of the Developed Method

The actual stress was measured using the exhaust system of a mass-production vehicle, and the measurement results were compared with the analysis results obtained using the developed method to confirm the versatility of the new analysis method. Figure 18 shows an overall view of the subject exhaust system.

In order to predict high-cycle resonance stress in the drawing stage, the spherical joint characteristics used the newly developed models and parameters. The mesh model was modeled based on 3D-CAD data, and the material properties were defined as specified in the drawings without correcting the stiffness or density. Figure 19 shows graphs comparing the stress results obtained by stress measurement in the actual testing and by the new calculation method.

The results showed a resonance frequency error of 3% and stress error of 20%. This is sufficiently accurate for

practical application as drawing-based prediction results, and indicates the validity and versatility of this analysis method.

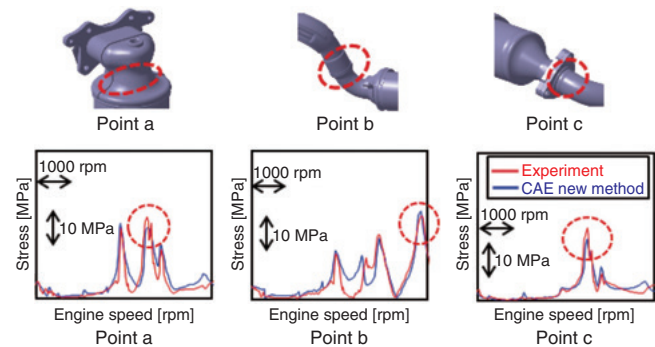


Fig. 19 Comparison of experiment and CAE by new method

4. Conclusion

Technology for accurately predicting the resonance stress of the exhaust system in the engine utility range was developed by the following method.

- (1) A spherical joint model was experimentally modeled by obtaining the stick-slip characteristics of the actual vehicle and reflecting these characteristics to an ADAMS model.
- (2) The input excitation force was accurately measured, and an input model that considers the engine behavior and phase at multiple orders was created and applied as the input conditions for analysis.
- (3) The mesh model was modeled using a mesh size enabling sufficiently accurate analysis, and the correlation was increased by performing MAC correlation with the actual vehicle.

This method enables model-based prediction of the stress on various different exhaust systems, and it is thought that this will contribute greatly to increasing development speeds and reducing costs by reducing the number of prototypes, and also helping to reduce weight. As a future possibility, it is thought that this new method will also enable prediction of exhaust system stress for vehicle types with different spherical joints.

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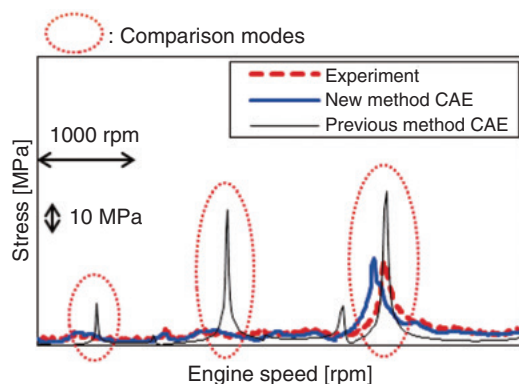


Fig. 17 Simple exhaust system model stress-analysis result

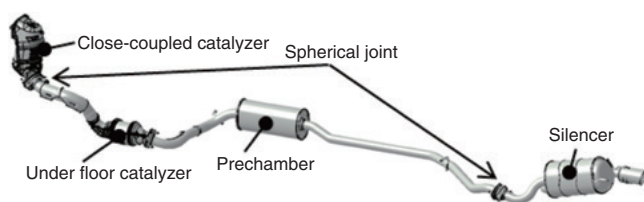


Fig. 18 Exhaust system for verification

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Oil Temperature Classification Method for Hydraulic Electronically Controlled AWD System

Hiroki YOSHINO*

ABSTRACT

A low computational complexity and high-accuracy method for clarification of oil temperature in a hydraulic electronically controlled rear differential was developed for implementation to a vehicle control unit. Because the target system uses a motor-driven electronic hydraulic pump to increase hydraulic pressure, a correlation was considered to exist between oil temperature and motor current. Theoretical analysis of the structure of the system demonstrated that a linear model was sufficient for clarification of the state of oil temperature from motor current, and a clarification model for the state of oil temperature using logistic regression was formulated. In addition, in order to balance accuracy and frequency of clarification, a method of calculation of posterior probability when continuous multiple clarifications were executed using only the output of the logistic regression model was developed. The results of application of the method to data obtained from the actual target system confirmed that the proposed method was able to clarify the state of oil temperature with a high degree of accuracy.

1. Introduction

The performance of systems that use oil for control or cooling is significantly affected by the characteristics and the state of the oil in the system. Focusing on the effect of oil temperature on the output of an AWD system⁽¹⁾, the research discussed in this paper obtained oil temperature data using an oil temperature sensor, and used this data as a control parameter. Figure 1 shows the configuration of the rear differential used in the AWD system. Using an FF vehicle as a base, this system transmits drive power from the front wheels to the rear wheels, and is able to switch between 2WD and AWD by means of appropriate allocation of drive power between the front and rear wheels. The drive power of the front wheels is input to the rear differential via a propeller shaft; the input torque is transmitted to the rear wheels via hypoid and differential gears by means of the engagement of a clutch positioned between a clutch guide and a clutch hub. Clutch connection force can be generated as desired by controlling oil pressure using an actuator (motor and solenoid valve), and using the generated oil pressure to change the clutch control oil pressure in the system. The system has three operating statuses: pressure increase, pressure maintenance, and pressure reduction. When clutch control oil pressure is needed, the solenoid valve is closed and the motor is operated in order to

increase pressure (pressure increase); when the system judges that the control operation has been completed, the motor is stopped and oil pressure is maintained (pressure maintenance). Following this, when there is no longer any need for clutch control oil pressure, the solenoid valve is opened and pressure is reduced (pressure reduction).

If the viscosity of the oil changes, the shear resistance and friction force of the oil film between the clutch plates also changes, and this results in changes in the torque transmitted to the rear wheels. Because of this, readings from an oil temperature sensor are used as one of the parameters to determine the necessary clutch control oil

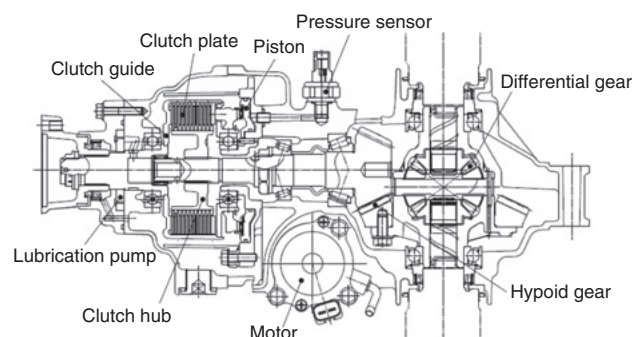


Fig. 1 Structure of rear differential system

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pressure (“target clutch control oil pressure”). When the oil temperature sensor is operating normally, no issue is presented, but if, for example, a sensor malfunction were to result in an abnormal value being output, there would be a divergence between the true target clutch control oil pressure and the target clutch control oil pressure that is calculated on the basis of a variety of parameters. If the vehicle were driven when the target clutch control oil pressure was not calculated correctly, vehicle behavior could become unstable. In particular, if the oil temperature sensor outputs an abnormal value when the oil temperature is low, it is possible that excessive torque might be applied to the gears in the rear differential, and this could affect their durability. It is therefore necessary to judge whether the present oil temperature in the system is low using a separate method to the oil temperature sensor.

A variety of estimation methods and judgment methods have been proposed in order to obtain data concerning the physical characteristics of the oil. Of these, previous research has indicated the practicality of methods employing machine learning. Because when machine learning is used it is possible to create a model based solely on the data that the user wishes to model, this method has the merit of making it possible to produce models even in situations in which obtaining direct knowledge of the internal structure of the system under study represents a challenge. Balabin et al. proposed methods of classifying mineral and synthetic oils and of classifying viscosity in oils used in an engine by applying the k-nearest neighbor (k-NN), neural network (ANN) and support vector machine (SVM) methods to absorption spectrum data obtained by means of near-infrared spectroscopy⁽²⁾. In addition, a number of reports have been published concerning methods of estimating the temperature of the insulating oil in oil-immersed transformers utilizing ANN that learns from the transformer load and the ambient air temperature⁽³⁾⁻⁽⁵⁾. Other research focusing on oil-immersed transformers has taken the amount of gas in the oil as a feature quantity, and classified abnormal states in on-load tap switches using logistic regression and ANN⁽⁶⁾. This paper indicates that the ANN approach is the more effective, given the fact that this method displays higher classification accuracy than logistic regression.

These reports demonstrate that it is possible to estimate oil characteristics with a high degree of accuracy using ANN and SVM. Because ANN and SVM employ non-linear models, high-accuracy estimation and classification is possible even in the case of data for which relationships of correlation are complex. However, the other side of this is that when results are output, it is necessary to calculate large numbers of non-linear functions (activation functions and kernel functions) by means of numerical calculation. In addition, in the case of SVM, it is necessary to store support vector data in the memory. Given the fact that in many cases real-time calculation is prioritized in the field of automotive control, these methods cannot be considered desirable from

the perspectives of computational complexity and allocation of resources. In addition, if the methods are applied, how to judge the reliability of the output figures remains an issue.

The research discussed in this paper therefore developed a highly accurate and reliable method for the classification of oil temperature in AWD systems, while controlling computational complexity. The details of this development are discussed below.

2. Oil Temperature Classification Method

2.1. Relationship between Oil Temperature and Motor Current

The research discussed here focused on the relationship between oil temperature and motor current, and developed a method of classifying the state of oil temperature by modeling the correlation between data for the two parameters. The pressure adjustment mechanism in the target system rotates rotors attached to the end of the motor by transmitting current to the motor, and draws up oil using the principle of a trochoid pump (Figs. 2 and 3). As

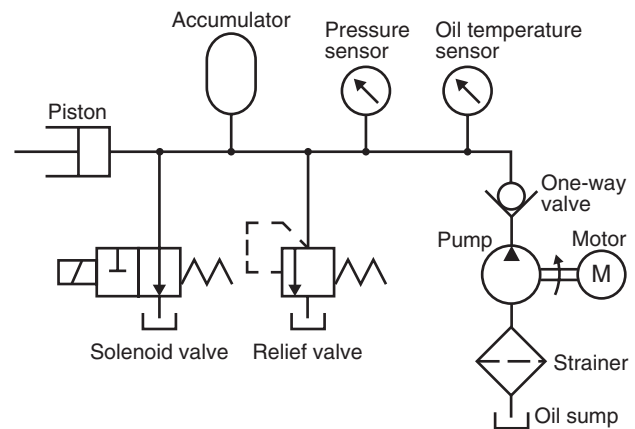


Fig. 2 Schematic of target system

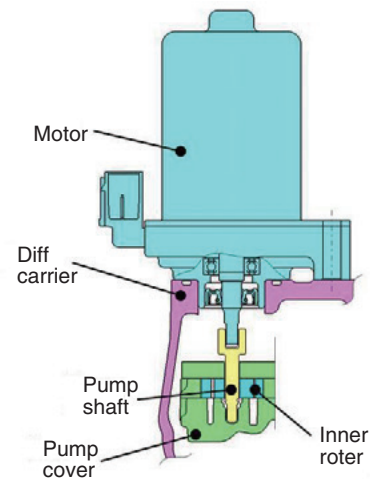


Fig. 3 Cross-section view of motor pump

indicated above, the viscosity of oil displays a temperature characteristic, with viscosity being higher the lower the temperature of the oil. It is considered that the friction in the rotors is also affected by this characteristic. In addition, due to the characteristics of the motor, there is a linear relationship between the current supplied to the motor and the torque that it produces. Because of this, it is considered that there is also a correlation between motor current and oil temperature when oil pressure is being increased (Fig. 3).

First, in order to clarify the relationship between oil temperature and motor current, the relationship was formularized using system parameters. Taking the transient state directly following the transmission of the operational command to the motor, motor current $I_1(t)$ and $I_2(t)$ can be expressed by

$$I_1(t) = \frac{V}{R} \left(1 - e^{-t/\tau_1} \right) \quad (1)$$

$$I_2(t) = \frac{1}{R} \left\{ V + \left[\frac{R}{K_T} Tq(t, temp) + V \right] \times \left[1 - e^{-t/\tau_2} \right] \right\} \quad (2)$$

where

- V : Motor drive voltage
- R : Motor resistance
- K_T : Motor torque constant
- Tq : Motor output torque
- τ_1, τ_2 : Time constants dependent on motor characteristics
- $temp$: Oil temperature
- t : Time

Figure 4 shows an outline of the current waveforms for Eqs. (1) and (2). Next, using Eqs. (1) and (2), the integrated value for current for the period from $t=0$ to $t=T$ is calculated. Because a current corresponding to the lock current is flowing at $t=T_1$, T_1 can be regarded as a constant; by further taking the integrated time T to be an arbitrary constant, the integrated value for current can be expressed as follows.

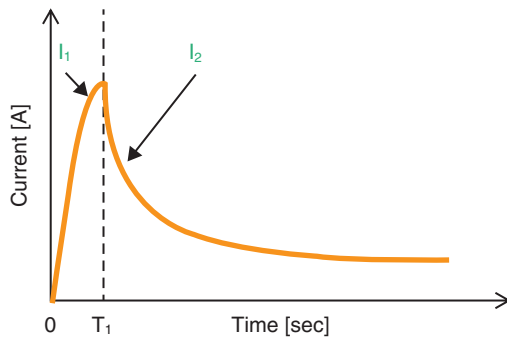


Fig. 4 Current waveform model

$$\begin{aligned} & \int_{t=0}^T [I_1(t) + I_2(t)] dt \\ &= \frac{V}{R} \left[T_1 + \tau_1 \left(e^{-T_1/\tau_1} - 1 \right) \right] + \frac{V}{R} (T - T_1) \\ &+ \frac{1}{R} \left[\frac{R}{K_T} \int_{t=0}^{T-T_1} Tq(t, temp) dt - V \right] \left\{ (T - T_1) + \tau_2 \left[e^{-(T-T_1)/\tau_2} - 1 \right] \right\} \\ &= c_1 V + c_2 \int_{t=0}^{T-T_1} Tq(t, temp) dt \end{aligned} \quad (3)$$

Here, c_1 and c_2 are constants.

Taking into consideration situations in which temperature is an arbitrary value in Eq. (3), the integral term for motor drive torque can be regarded as a constant, and Eq. (3) becomes

$$\int_{t=0}^T [I_1(t) + I_2(t)] dt - c_1 V = \text{const} \quad (4)$$

Equation (4) indicates that when data for which the oil temperature is identical has been isolated from three-dimensional data for the integrated value of current, motor drive voltage, and oil temperature, the two-dimensional data for the integrated value of current and the motor drive voltage included in that data will be linear. Put another way, it shows that when an oil temperature classification model that uses the integrated value of current and motor drive voltage as feature quantities is formulated, the boundaries for classification will be linear. This fact indicates that rather than using non-linear models such as ANN or SVM for the classification model, it is sufficient to use linear models that entail lower computational complexity.

2.2. Creation of Classification Model

Based on the results discussed above, a two-class classification model (classifying oil temperature as low or not low) using logistic regression was formulated that outputs the state of oil temperature following input of the integrated value of current and the motor drive voltage. Logistic regression is a form of standard linear modeling, and is a classification method that is robust against outliers (values significantly removed from other values)⁽⁷⁾.

The logistic regression model is expressed as

$$LOG = \frac{1}{1 + \exp(-\mathbf{w}'\mathbf{x})} \quad (5)$$

where

- $\mathbf{x}$: Input data vectors
- $\mathbf{w}$: Parameter vectors
- LOG : Class posterior probability

$\mathbf{x}$ represents the two-dimensional vectors of the integrated value of current and the motor drive voltage, and LOG is the probability of the oil temperature being above (or below) the threshold value. $\mathbf{w}$ represents the parameters that determine the behavior of the model, and its value can be calculated from data for the integrated value of current,

motor drive voltage, and state of oil temperature by means of the process described below.

First, the true state of oil temperature in each set of data is designated t . Because there are two classification classes, if $t \in \{0, 1\}$, the likelihood function for L data can be expressed by the following equation.

$$p(\mathbf{t}|\mathbf{w}) = \prod_{i=1}^L LOG_i^{t_i} \{1 - LOG_i\}^{1-t_i} \quad (6)$$

Here, $\mathbf{t}$ represents the vectors for the true state of oil temperature, and LOG_i is the figure generated by Eq. (5) for $\mathbf{x}_i$. If the negative logarithm of the likelihood function is defined as the error function $E(\mathbf{w})$, then from Eq. (6), $E(\mathbf{w})$ is expressed as

$$E(\mathbf{w}) = -\ln p(\mathbf{t}|\mathbf{w}) \\ = -\sum_{i=1}^L \{t_i \ln LOG_i + (1-t_i) \ln(1-LOG_i)\} \quad (7)$$

The value being sought is the value for $\mathbf{w}$ that will minimize Eq. (7), and this can be determined using the Newton-Raphson method by means of the following iterative calculation.

$$\mathbf{w}_{s+1} = \mathbf{w}_s - \mathbf{H}^{-1} \nabla E(\mathbf{w}) \quad (8)$$

Here, s is the number of iterations, and $\mathbf{H}$ is the Hessian matrix found from $\nabla^2 E(\mathbf{w})$. Because $\mathbf{H}$ is a positive definite matrix in the case of the logistic regression model defined by Eq. (5), in Eq. (7) $\mathbf{w}$ is a convex function, and the global optimum solution and the local optimum solution correspond. Therefore, the value of $\mathbf{w}$ determined by convergence via iterated calculation using Eq. (8) is the parameter to be sought. When $\mathbf{w}$ is determined, $\mathbf{x}$ is substituted in Eq. (5), and judgment of whether LOG is higher (or lower) than the threshold value makes it possible to output an oil temperature classification result.

In the case of the normal logistic regression equation, Eq. (5), probabilities can only be output for inputs of single units of data. This results in a trade-off, in which the potential for mistaken classification is increased if the threshold value for classification is lowered, but classification represents more of a challenge if the value is raised. A calculation method able to output probabilities when multiple data is input was therefore introduced in order to balance classification accuracy and classification frequency.

First, two units of input data, $\mathbf{x}_1$ and $\mathbf{x}_2$, were introduced in order to expand Eq. (5), and the determination of the posterior probability when oil temperature is classified as low for both (expressed as $y=1$) was considered. From Bayes' theorem, the posterior probability $p(y=1|\mathbf{x}_1, \mathbf{x}_2)$ can be expressed as

$$p(y=1|\mathbf{x}_1, \mathbf{x}_2) \\ = \frac{p(\mathbf{x}_1|y=1)p(\mathbf{x}_2|y=1)p(y=1)}{p(\mathbf{x}_1, \mathbf{x}_2)} \quad (9)$$

Because, despite the fact that $\mathbf{x}_1$ and $\mathbf{x}_2$ are in the same class, the oil temperature can be considered to change with the passage of time, conditional independence is assumed for $\mathbf{x}_1$ and $\mathbf{x}_2$ here.

Next, the output of logistic regression of $\mathbf{x}_1$ is considered as LOG_1 , and transformed as follows in accordance with Bayes' theorem.

$$LOG_1 = p(y=1|\mathbf{x}_1) \\ = \frac{p(\mathbf{x}_1|y=1)p(y=1)}{p(\mathbf{x}_1)} \quad (10)$$

Similarly, the output of logistic regression of $\mathbf{x}_2$ is considered as LOG_2 , and can be expressed as

$$LOG_2 = \frac{p(\mathbf{x}_2|y=1)p(y=1)}{p(\mathbf{x}_2)} \quad (11)$$

In addition, using Eqs. (10) and (11), Eq. (9) can be expressed as follows.

$$p(y=1|\mathbf{x}_1, \mathbf{x}_2) = \frac{LOG_1 \times LOG_2}{p(y=1)} \frac{p(\mathbf{x}_1)p(\mathbf{x}_2)}{p(\mathbf{x}_1, \mathbf{x}_2)} \quad (12)$$

Further, by means of identical calculations, $p(y=0|\mathbf{x}_1, \mathbf{x}_2)$ can be expressed as

$$p(y=0|\mathbf{x}_1, \mathbf{x}_2) \\ = \frac{(1-LOG_1) \times (1-LOG_2)}{p(y=0)} \frac{p(\mathbf{x}_1)p(\mathbf{x}_2)}{p(\mathbf{x}_1, \mathbf{x}_2)} \quad (13)$$

Based on Eqs. (12) and (13) and $p(y=1|\mathbf{x}_1, \mathbf{x}_2) + p(y=0|\mathbf{x}_1, \mathbf{x}_2) = 1$,

$$\frac{p(\mathbf{x}_1)p(\mathbf{x}_2)}{p(\mathbf{x}_1, \mathbf{x}_2)} \\ = \frac{p(y=1)}{LOG_1 \times LOG_2 + (1-LOG_1) \times (1-LOG_2)} \quad (14)$$

Because data concerning the posterior probabilities $p(y=1)$ and $p(y=0)$ cannot be obtained, here it is assumed that $p(y=1)=p(y=0)$. Substitution of Eq. (14) into Eq. (12) makes it possible to express the posterior probability when $\mathbf{x}_1$ and $\mathbf{x}_2$ are given by means of the following equation.

$$p(y=1 | \mathbf{x}_1, \mathbf{x}_2) = \frac{LOG_1 \times LOG_2}{LOG_1 \times LOG_2 + (1 - LOG_1) \times (1 - LOG_2)} \quad (15)$$

In addition, conducting identical calculations makes it possible to express the posterior probability when n units of data $(\mathbf{x}_1, \dots, \mathbf{x}_n)$ are input as

$$p(y=1 | \mathbf{x}_1, \dots, \mathbf{x}_n) = \frac{\prod_{i=1}^n LOG_i}{\prod_{i=1}^n LOG_i + \prod_{i=1}^n (1 - LOG_i)} \quad (16)$$

A value higher than the classification threshold for logistic regression is introduced as the threshold value for Eq. (16), and the classification result generated by Eq. (16) is considered to be the final classification result. By this means, an algorithm capable of instantaneously determining a classification result when data of a high degree of reliability is input, and of determining a classification result when data of ordinary reliability is input continuously, has been realized using only the output of logistic regression. Figure 5 shows the increase in the posterior probability as data is added, in accordance with Eq. (16). Figure 5 shows a case in which data containing values higher than the classification threshold value for logistic regression is continuously input. Classification is determined when the posterior probability exceeds the high threshold value. Figure 6 shows the complete flow of the algorithm.

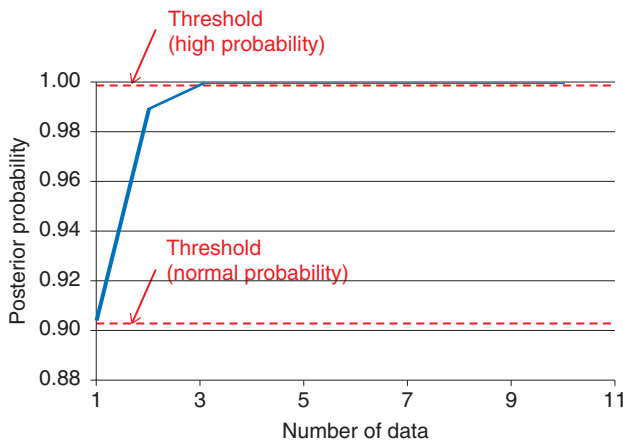


Fig. 5 Example of transition of posterior probability

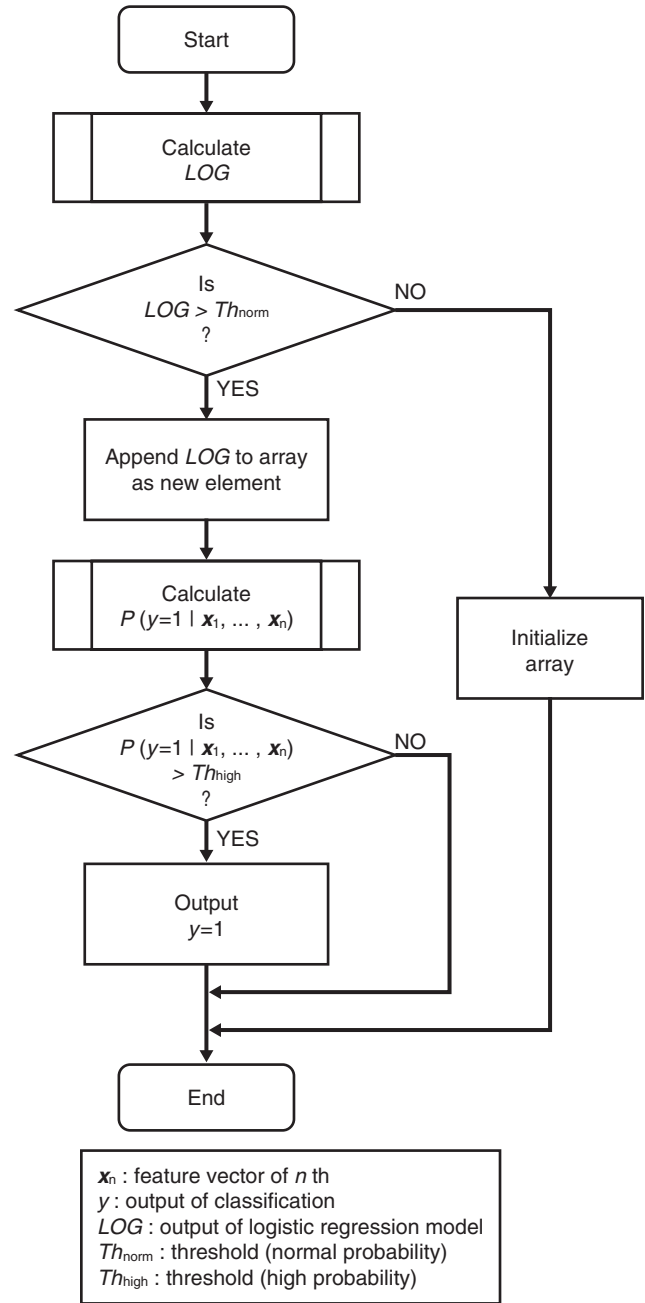


Fig. 6 Algorithm of proposed method

3. Results of Tests

Using data obtained from an actual system, a model for the classification of oil temperature as exceeding the threshold value or not was created, and the classification rate was evaluated. For the formulation and evaluation of the model, three-dimensional data for the integrated value of current, motor drive voltage, and oil temperature was used, and ten-fold cross-validation was conducted. (In this method, the data set is divided into 10, with nine units of data being used in the formulation of the model and the

Table 1 Classification rate of logistic regression

Threshold (°C)	Training		Test	
	Mean	Min	Mean	Min
20	0.974	0.973	0.973	0.960
10	0.985	0.984	0.985	0.982
0	0.982	0.982	0.982	0.977
-10	1.000	1.000	0.999	0.999
-20	0.999	0.999	0.999	0.997
-30	0.996	0.995	0.996	0.992

Table 2 Computing time comparison

Logistic regression	ANN	SVM
0.14 ms	0.23 ms	0.41 ms

remaining one unit of data used for the evaluation of the model's performance. This is repeated 10 times. The nine units of data are termed "training data," and the remaining one unit of data is termed test data). The period for integration of the current was set at 150 ms. Table 1 shows the classification results generated by the logistic regression model. The "Training" columns show the classification rate for the training data, and the "Test" columns show the classification rate for the test data. In the respective categories, the "Mean" column shows the median value for the classification rate obtained by cross-validation, and the "Min" column shows the lowest classification rate.

First, the results for the training data indicate that the state of the oil temperature can be classified with a high degree of accuracy even when a linear model is used. For the system under consideration, high classification accuracy at low temperatures is a specific goal, and the results indicate the realization of this goal, with classification accuracy increasing with decreasing temperature. This is considered to be due to the fact that the change in the viscosity of the oil in relation to temperature becomes greater the lower the temperature, which results in an increase in the rate of classification of data close to the threshold value.

The classification rate for test data also compared favorably with the results for training data for all threshold temperatures. This indicates that the generated model possesses a high level of generalization capability, and demonstrates the fact that the method has successfully produced a model able to make correct classifications even in relation to new, previously unknown data. The combination of the output of a logistic regression model and the algorithm provided by Eq. (16) has made it possible to realize more accurate classification.

Next, classification models for this specific classification challenge were formulated using logistic regression, ANN and SVM, and their calculation time for classification calculations in relation to 3000 units of data was measured. The results are shown in Table 2. For the measurements, non-linear models that would maintain classification accuracy while reducing computational complexity were selected for ANN

and SVM. For the ANN model, the evaluation was conducted using a two-layer network with two hidden units (two inputs, two hidden units, and one output), in order to indicate the differences between the model and a standard linear model. The results demonstrate that the selection of a logistic regression model as a classification model is able to reduce computational complexity in comparison to the selection of ANN or SVM models.

4. Conclusion

A method of classifying the state of oil temperature in an electronically controlled AWD system was developed and its effectiveness verified. Formularization of the motor current waveform and determination of the fact that the relationship between the integrated value of current and the motor drive voltage when the oil temperature is fixed is linear in the data space made it possible to select a classification method possessing the necessary functions to satisfy the purpose at the same time as controlling computational complexity. In addition, the reliability of multiple classification results was quantitatively evaluated, and an algorithm that outputs final classification results was formulated using only output from the classification model, resulting in the realization of a method that controls computational complexity and balances the reliability of final classification results and classification frequency.

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Hiroki YOSHINO

Creation of Technology for Prediction of Self-Excited Vibration under Operating Conditions for Multiple-Plate Clutches

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ABSTRACT

It is important to grasp the dynamic frictional characteristics of facing materials, their methods of use, and the damping characteristics of the drivetrain in order to predict self-excited vibration in multiple-plate clutches used in transmissions. A dedicated torque transmission characteristic test apparatus capable of equalizing the surface pressure was built to deal with the dynamic frictional characteristics of the facing materials, which are affected by the temperature and contact pressure of the friction surfaces. With this apparatus, the sliding velocity dependence was found for each temperature and surface pressure. The behavior of the actual vehicle was also measured when vibration occurs and loss torque is generated by friction in the drivetrain, and damping characteristics were determined from the measurement values. An FEM model of the clutch for the purpose of gaining a quantitative grasp of the dynamic frictional characteristics and methods of use of the facing materials obtained in this way was created, and a simulation model for dynamic behavior analysis of the characteristics of torque transmission from a transmission that incorporates drivetrain damping characteristics to the vehicle body was constructed. In order to confirm the usefulness of this method, the construction of a clutch that suppresses self-excited vibration was examined by simulation and the reduction of vibration in an actual vehicle was confirmed.

1. Introduction

The rise in user needs for enhanced fuel economy has been accompanied by demand for transmissions that are lightweight, compact, and low-friction⁽¹⁾. The size of the clutch imposes limits on demands for lighter weight and further compactness, and it is also necessary to consider durability in terms of absorbing a certain amount of heat. There is a known measure for addressing this by maximizing the difference between the internal and external diameters of the disk friction surfaces, thus limiting the amount of heat produced per unit area⁽²⁾. It is known that surface pressure, temperature, and sliding velocity within the friction surfaces are distributed unequally in a radial direction and that these quantities are considerably affected by the friction characteristics of the facing materials of the clutch⁽³⁾. In designing a clutch, therefore, it is important to clarify the distribution of the surface pressure, temperature, and sliding velocity of the clutch friction surfaces and to grasp the friction characteristics of the facing materials themselves.

Another conceivable response to the demand for reduced

friction would be to reduce friction in the drivetrain. This, however, involves a reduction of damping in the drivetrain, which can be disadvantageous in terms of vibration. The vibration that is an issue in the drivetrain includes forced vibration, the frequency of which varies dependently on the speed of rotation, and self-excited vibration, the frequency of which depends on the eigenvalue of the drivetrain and remains fixed.

In the case of forced vibration, the shape and precision of component parts often provide ways of examining the causes of vibration and measures against it. In the case of self-excited vibration, the amplitude of the vibration tends to continue increasing as long as slippage occurs in the clutch. Whether or not self-excited vibration will be produced is determined both by the influence of the slope of the coefficient of friction with respect to the sliding velocity of the clutch sliding surfaces and by the damping characteristics of the drivetrain⁽⁴⁾. Many of the factors involved in quantification of the slope of the dynamic frictional characteristics of the clutch and the damping characteristics of the drivetrain are unknown. In the current state of technology for simulating vehicle vibration, the

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clutch is most often represented by a single coefficient of friction, so that the technology is not yet at the point of allowing for investigation of vibration countermeasures by means of clutch structure at the design stage. Consequently, countermeasures against self-excited vibration to date have been taken from the perspective of limiting the deterioration of facing materials or oil, which affects friction characteristics to a considerable extent^{(5), (6)}. For this paper, therefore, attention was focused on the dynamic frictional characteristics of the clutch facing materials and methods for their use. The clutch was created as a finite element method (FEM) model, and technology for dynamic behavior analysis capable of simulating torque transmission characteristics during self-excited vibration by means of a plant model that includes the damping characteristics of the drivetrain was developed. This clarified the mechanism by which the torque transmission characteristics of the drivetrain are determined and enable quantitative prediction of self-excited vibration.

2. Obtaining a Clear Understanding of Vibration Phenomena

Figure 1 shows the main section of the transmission used for analysis. It has a wet-type starting clutch placed above the driven pulley shaft, and it enables the vehicle to move smoothly from a standing start by controlling the pressure in a hydraulic circuit placed within the rotating shaft.

Figure 2 shows the results for reproduction of vibration in an actual vehicle by the use of oil that has no friction modifiers added to it. The output speed during acceleration from a standing start was subjected to frequency analysis by fast Fourier transform (FFT), and the results indicated that in addition to a forced vibration with a frequency that rises

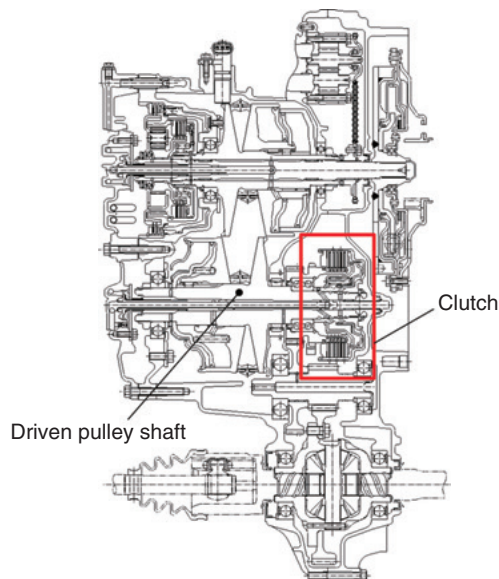


Fig. 1 Main section of transmission

in proportion to the speed of rotation, there was also a self-excited vibration with a frequency that reached a fixed peak independent of the speed of rotation in the range where vibration occurred in the clutch (Fig. 3). Verification was also conducted of torque transmission characteristics from the torque measured on an actual vehicle and the rotational difference in the clutch. This showed a negative slope in the range of rotational difference where vibration occurred, suggesting that torque transmission characteristics with a negative slope with respect to rotational difference were contributing to the vibration (Fig. 4).

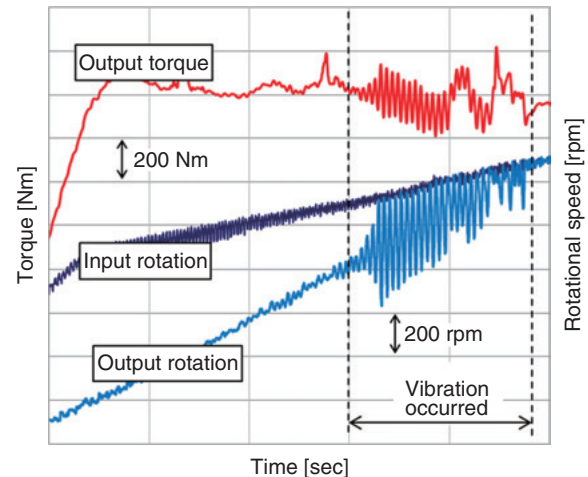


Fig. 2 Self-excited vibration measurement results

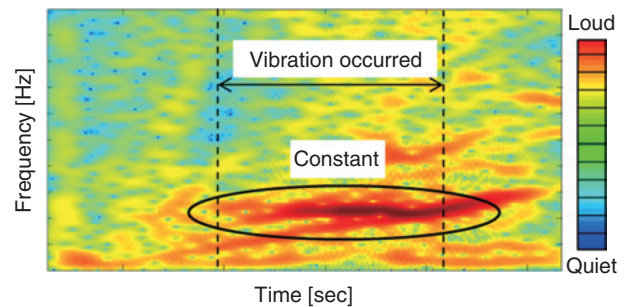


Fig. 3 Frequency analysis of vibration at starting acceleration

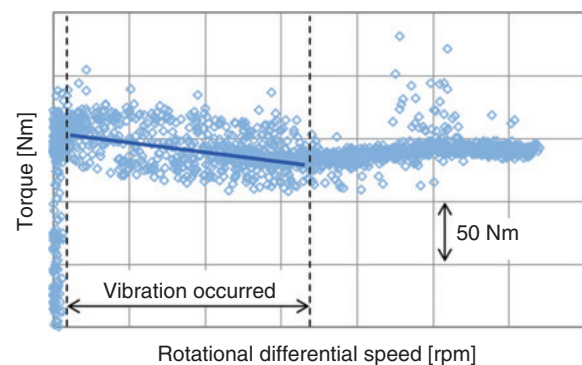


Fig. 4 Torque transmission characteristics

Next, the distribution of surface temperature and surface pressure was investigated in order to determine the state of friction surfaces when vibration occurs in an actual vehicle. In order to measure the temperature of friction surfaces in the course of clutch engagement, thermocouples were embedded in the plates (Fig. 5). The rates of plate temperature increase were different at points A and C on the inside diameter and points B and D on the outside diameter, suggesting that temperature is distributed in a radial direction (Fig. 6).

The distribution of surface pressure was measured using pressure-scaling paper. The results confirmed that, like surface temperature, surface pressure is distributed in a radial direction (Fig. 7).

The surface temperature and surface pressure on the clutch friction surfaces were found from the above to be distributed unequally in a radial direction.

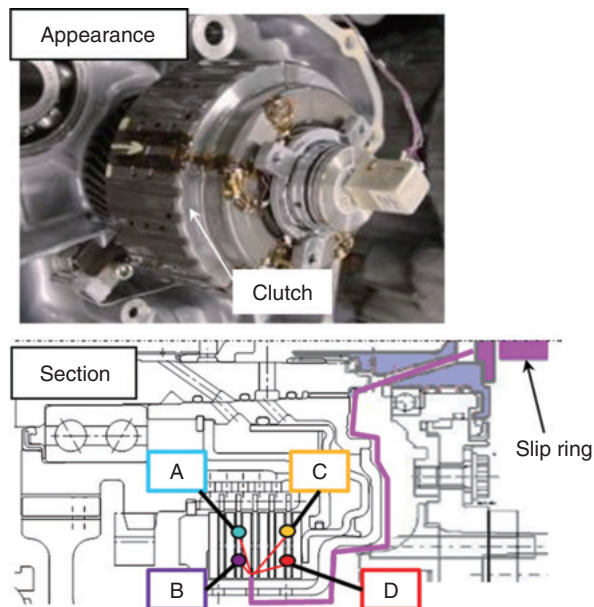


Fig. 5 Measurement points

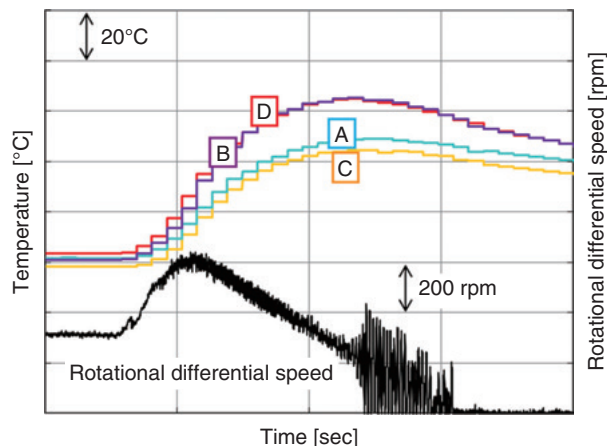


Fig. 6 Plate temperature measurement results

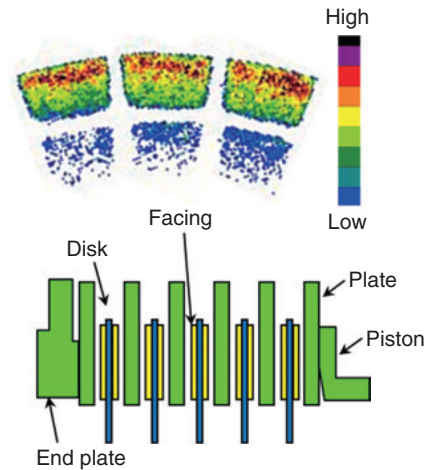


Fig. 7 Facing contact pressure distribution

3. Construction of an Analytical Model of Self-Excited Vibration

The self-excited vibration of the clutch can be described by the equation of motion for the spring-mass system of the drivetrain as a whole (Fig. 8). Here, I is the inertia of the vehicle body downstream from the clutch, K is the torsional rigidity of the drive shaft, T is the transmission torque of the clutch, T_0 is the transmission torque of the clutch in the coefficient of static friction, and θ_1 and θ_2 are the speed of rotation of the driving side and driven side of the clutch, respectively. This calls for the torque transmission characteristics determined by the facing

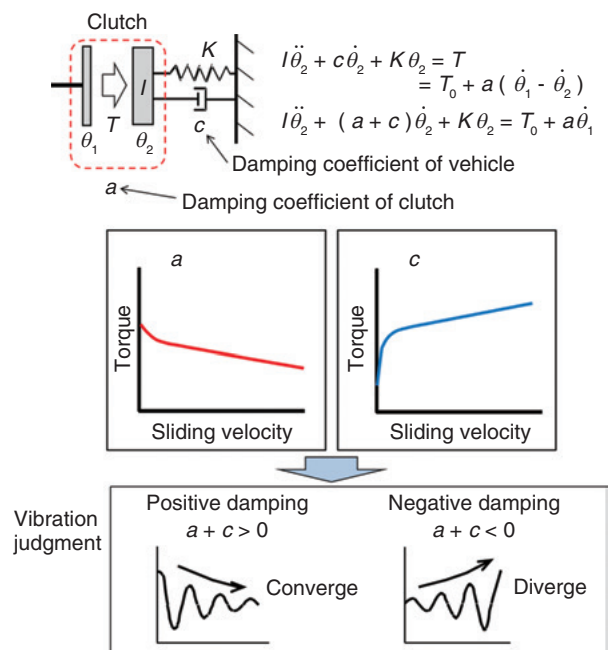


Fig. 8 Hypothesized mechanism of self-excited vibration

materials (slope a in Fig. 8) and the torque transmission characteristics determined by drivetrain damping (slope c in Fig. 8) to be taken into consideration. These characteristics affect the convergence or divergence of the vibration.

The torque transmission characteristics determined by drivetrain damping are thought to always have a positive slope. There are some cases, however, when the coefficient of friction has a negative slope with respect to the sliding velocity in the various parts of the friction surfaces due to variations in friction characteristics and clutch structure as they influence the torque transmission characteristics determined by facing materials. It was inferred that vibration occurs when the sum of those torque transmission characteristics have a negative slope.

The state of the clutch friction surfaces during actual operation was therefore reproduced and a computational model that enables consideration of drivetrain damping characteristics was constructed (Fig. 9). The clutch portion was created as an FEM model, and the distribution of surface pressure, temperature, and sliding velocity of the clutch friction surfaces was taken into consideration. Furthermore, the drivetrain elements before and after the clutch were linked by a one-dimensional torsional vibration system model, which yielded a model for dynamic behavior analysis capable of predicting the vibration of an actual vehicle.

3.1. Dynamic Frictional Characteristics of Clutch Facing Materials

SAE #2 test equipment and LVFA test equipment are known testing devices for measuring the frictional characteristics of clutch facing materials. However, these testing equipment are not readily able to measure frictional characteristics under surface pressure conditions equivalent to those in an actual vehicle. In addition, it is no easy matter to maintain the facing materials as a whole at a fixed temperature. A dedicated torque transmission characteristic test apparatus was therefore created (Fig. 10). By the use of this apparatus, the surface pressure could be maintained at a uniform high range equivalent to that in an actual vehicle, and the friction surfaces could be kept at a uniform overall temperature. This apparatus was used to vary the sliding

Table 1 Conditions of torque characteristics measurement

Contact pressure	0.5 – 6.0 [MPa]
Plate temperature	40 – 180 [°C]
Rotational speed differential	0.5 – 600 [rpm]

velocity and test torque transmission characteristics. The test conditions are shown in Table 1.

Figure 11 shows an example of the torque measurement results. First, hydraulic pressure was added, and then the speed of rotation was raised to the target value. The transmission torque and the temperatures in the various parts of the friction surfaces were measured when the hydraulic pressure and speed of rotation reached a constant state. This test was conducted multiple times under the surface pressure, temperature, and speed of rotation conditions shown in Table 1.

Next, the relationship between dynamic coefficient of friction and temperature at various surface pressures was systematically organized from the measurement results at

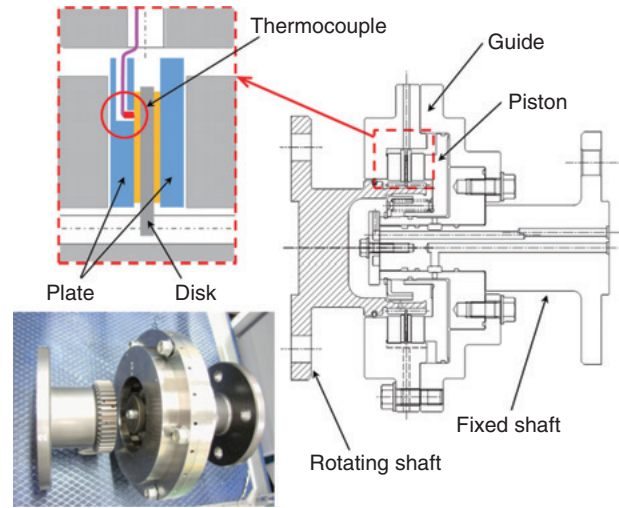


Fig. 10 Test apparatus for torque transmission characteristics

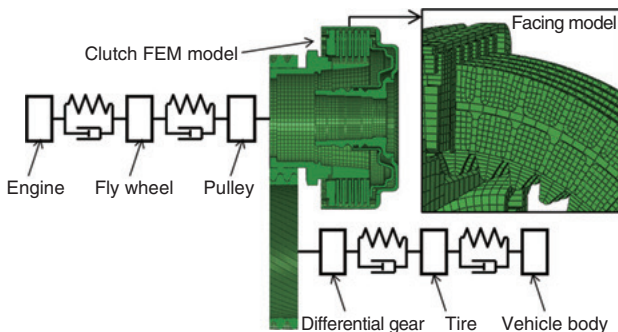


Fig. 9 Self-excited vibration prediction model

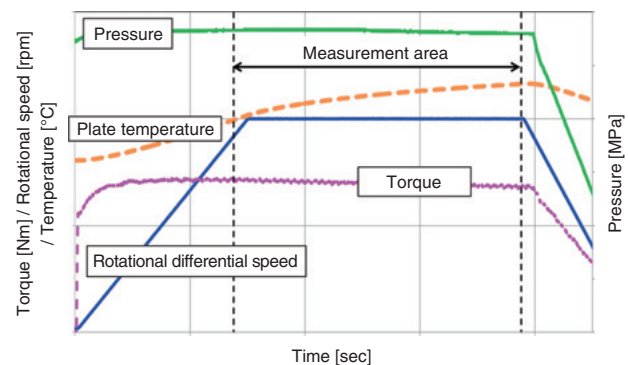


Fig. 11 Results of measurement

various rotational differential speeds as shown in Fig. 12. This was further organized to show the relationship between dynamic coefficient of friction and rotational difference at various temperatures for the various surface pressures.

The relationships between dynamic frictional characteristics of clutch facing materials and rotational difference obtained in this way were organized by surface pressure (Fig. 13) and by temperature (Fig. 14). The dynamic frictional characteristics showed a dependence on surface pressure and rotational difference. A positive slope tended to be found on the lower side of surface pressure and on the higher side of temperature, respectively.

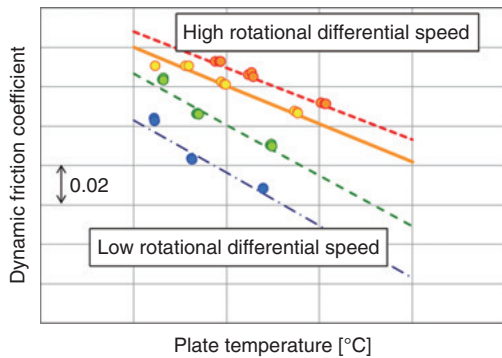


Fig. 12 Friction characteristics

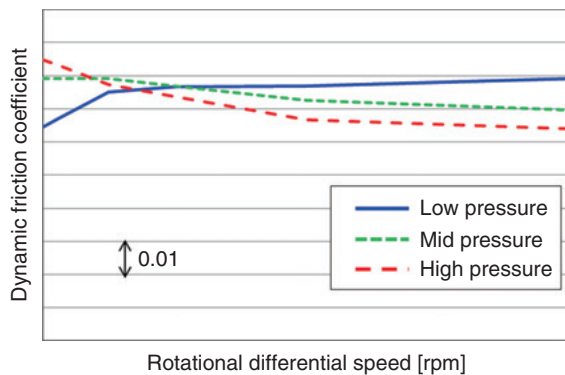


Fig. 13 Influence of pressure in dynamic friction coefficient

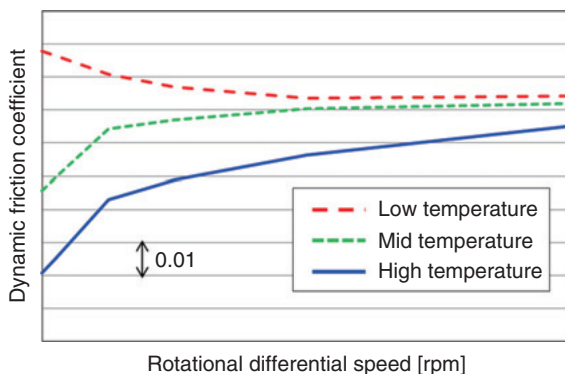


Fig. 14 Influence of temperature in dynamic friction coefficient

3.2. Obtaining a Clear Understanding of Drivetrain Damping Characteristics

When considering damping of the drivetrain as a whole, from the engine to the tires, attention was focused on damping between the clutch and the tires. This is because damping from the engine to the clutch does not affect vibration.

Damping between the clutch and the tires includes friction in the interior of the transmission and differential movement in the differential gear. For the interior friction, the loss torque was measured with the clutch disk and plate removed and the differential gear locked. The damping coefficient was then derived from the relationship between loss torque and rotational speed (Fig. 15).

The damping that accompanies differential movement of the differential gear was identified by both measurement and analysis. Figure 16 shows the results for calculation of the vibration behavior from the speed of rotation of the left and right drive shafts. The drive shafts on left and right are of unequal length, so there is a difference in rigidity. Consequently, when there is no friction in the differential gear, then when vibration occurs, the rotational speed and torque distributed to the left and right drive shafts will be affected by the right-left difference in rigidity even if

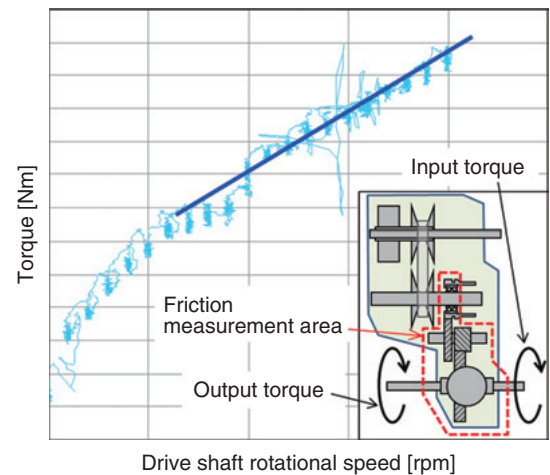


Fig. 15 Measurement of friction loss in transmission

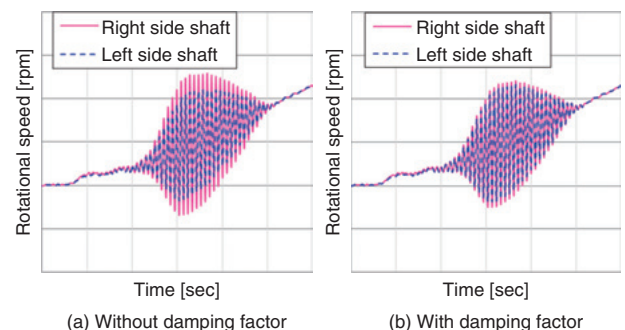


Fig. 16 Effect of differential gear damping factor

the vehicle is moving straight forward, and a difference in vibration behavior will become evident. On the other hand, when there is friction in the differential gear so that a damping effect operates, then the right-left difference in vibration behavior will be reduced, and the effect will be to reduce the influence of the right-left difference in drive shaft rigidity.

Measurements were therefore made of the rotational behavior on the inboard side of the right and left drive shafts of the actual vehicle from which the vibration had been reproduced, with the vehicle moving straight forward. The results showed that the difference in vibration behavior between right and left was infinitesimal, confirming the effect of the friction (Fig. 17). Taking this into account, a theoretical computational model was made of the differential gear vicinity, including damping parameters, and damping parameters that would produce the same rotational behavior as in the actual vehicle were identified (Fig. 18).

3.3. Confirmation of Consistency with Actual Phenomenon

A comparison of simulation results from the analytical model of self-excited vibration using the friction characteristics and damping characteristics described above and the measurement results from an actual vehicle are

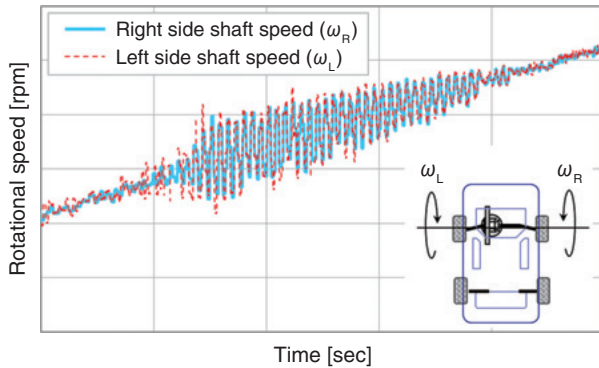


Fig. 17 Measurement of drive shaft speed

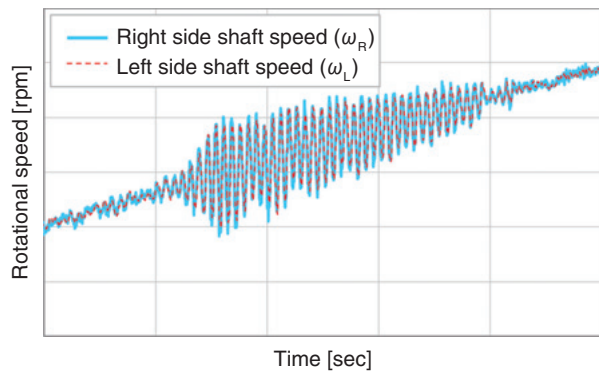


Fig. 18 Identification of damping factor

shown in Fig. 19 and Fig. 20. Verification was conducted using two types of oil, one with and one without friction modifiers, and the occurrence and non-occurrence of vibration equivalent to that of the actual vehicle were reproduced. The verification conducted with oil not containing friction modifiers, in which vibration did occur, showed that frequency, amplitude, and time vibration started were all consistent with the actual measurements.

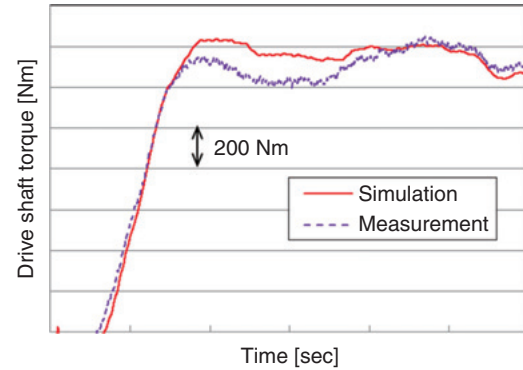


Fig. 19 Torque characteristics (oil with friction modifiers)

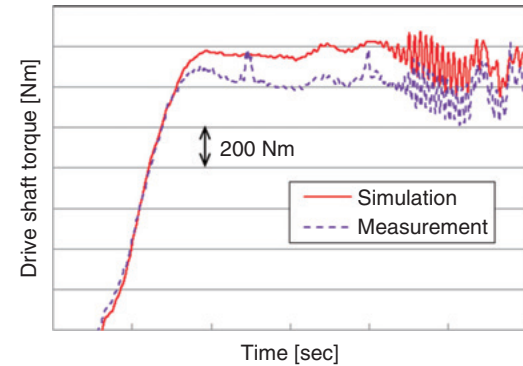


Fig. 20 Torque characteristics (oil without friction modifiers)

4. Validation of Construction Technology

The suppression of vibration was examined from the perspective of construction and based on the conceptual approach of maximizing the use of the range in which variations in dynamic frictional characteristics due to temperature, surface pressure, and sliding velocity produce a positive slope.

With the dynamic frictional characteristics obtained by measurement, the use of the lower range of surface pressure is effective in suppressing self-excited vibration. In terms of temperature, the use of the higher range is effective. However, since the use of the high temperature range has a considerable impact on durability performance, attention was focused on reducing surface pressure.

One conceivable approach to reducing surface pressure without diminishing torque capacity, which is a fundamental aspect of clutch performance, is to increase the number of friction surfaces. However, simply adding disks and plates, and expanding the clutch in a radial direction, would conflict with the move toward compactness. Therefore, the approach of affixing facing materials to one side of both the disk core metal and plates with reduced thickness as well as increasing the number of disks from five to seven was examined as a method that could maintain cooling function and durability without increasing the overall dimensions of the clutch (Fig. 21).

The effectiveness of this approach was validated using oil without friction modifiers so that self-excited vibration could be made to occur more readily. The effect of suppression of self-excited vibration due to the specifications incorporating these countermeasures was verified first by simulation. The results confirmed the reduction of vibration by specifications incorporating countermeasures in desktop estimations (Fig. 22).

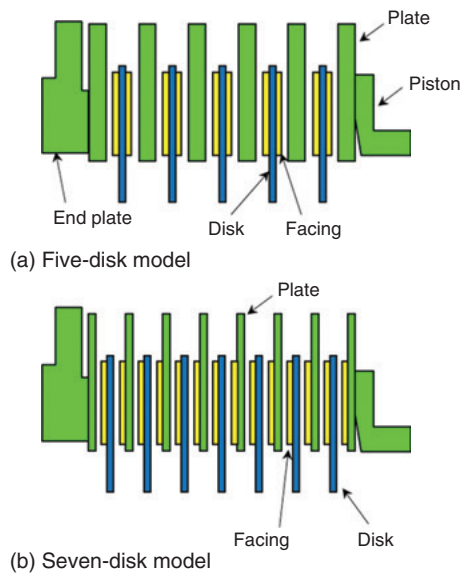


Fig. 21 Section of current and review specifications

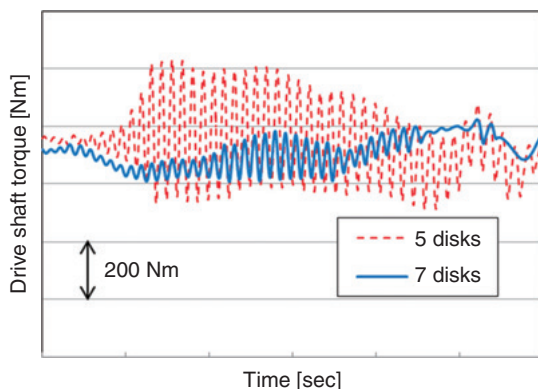


Fig. 22 Calculation results of torque characteristics

This effect was then confirmed on an actual vehicle using these specifications. The results show that vibration was reduced just as in the simulation (Fig. 23). The result of frequency analysis of this vibration waveform also shows a reduction of the peak amplitude in the frequency range where self-excited vibration occurs. This confirmed the usefulness of this method of approach (Fig. 24).

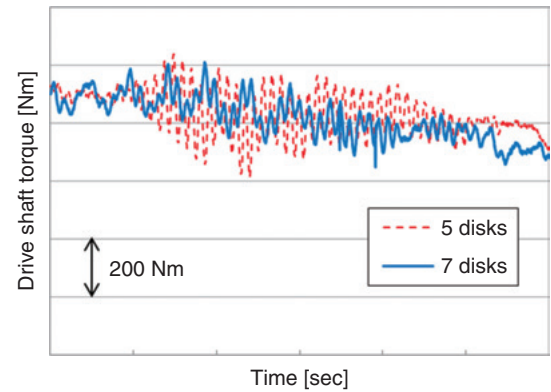


Fig. 23 Measurement results of torque characteristics

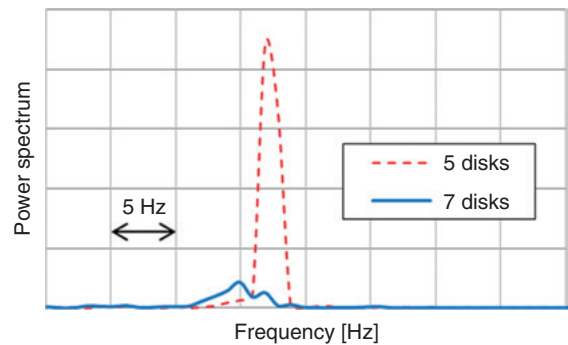


Fig. 24 Power spectrum of measurement results

5. Conclusion

The following conclusions were reached regarding this technology for prediction of self-excited vibration under operating conditions for multiple-plate clutches.

- (1) A model for dynamic behavior analysis was developed with the capability for calculating the torque transmission characteristics of the drivetrain as a whole by using the torque transmission characteristics of the clutch friction surfaces together with the damping values of the drivetrain. This enabled quantitative prediction of self-excited vibration in actual vehicles.
- (2) A clutch construction with an increased number of disks and facing materials affixed to one side of the disks was examined as a method of suppressing vibration. The vibration-suppressing effect this achieved was confirmed both by a model built to perform dynamic behavior analysis and by testing on an actual vehicle.

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Research on Dog Teeth Wear in Dog Clutch Transmissions

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ABSTRACT

In dog clutch transmissions, the excessive wear of dog teeth affects the functions of transmission and becomes a factor to obstruct smooth operation. Therefore, a study was conducted to clarify the wear mechanism of dog teeth by analyzing the operations in transmissions during dog clutch engagement.

In addition, a method of estimating the contact stress applied to the dog teeth was developed, and it was confirmed that contact stress is highly correlated with the degree of dog teeth wear.

Use of the developed method enabled accurate investigation in the early stage of designing on specifications which are related to dog teeth wear.

1. Introduction

Dog clutches are widely used in the stepped transmissions of motorcycles because of the demands for reductions in size and weight, and cost merits. Dog clutch transmissions do not have a synchronizing mechanism, which is applied to synchronized transmissions, for absorbing the rotational speed differences when shifting. Accordingly, a larger load is normally applied to a dog clutch.

In recent years, the displacement of motorcycle engines and their output power have increased, and the moments of inertia of drive train parts and the rotational speed of the transmission operations have been boosted. Through those changes, the load on a dog clutch has been increasing and this increased load makes the dog teeth wear (Fig. 1) larger. Excessive dog wear leads to difficulties in shifting and/or unexpected gear disengagements while driving affecting the smooth riding of a motorcycle.

Meanwhile, the conventional method for mathematical

evaluations on dog teeth wear were generally conducted only by the simplified calculations followed by repeated trials and errors to determine the specifications. With those backgrounds, in this study, we made an attempt to clarify the causal mechanism of dog teeth wear on dog clutches to develop a method of numerical evaluation that enabled the precise estimation of the dog teeth wear progression in order to enhance the development efficiency.

2. Overview of Dog Clutch Transmission

Figure 2 shows a typical dog clutch transmission of a motorcycle. The rider operates the shift lever with his or her left foot to rotate the shift drum via a shift spindle linked to the shift lever. When the shift drum rotates, the shift fork slides in the axial direction of the transmission while the pin on the fork traces the guiding groove situated around the outside of the shift drum. The shift fork arms are positioned to fit the grooves on the shifter with dog teeth, and the shifter slides along the spline on the shaft making the dog teeth engaged.

Before the dog teeth engagement occurs, there is a difference in rotational speeds between the engaging target gear (rotates freely relative to the shaft) and the shifter (spline-fitted onto the shaft), and torque impact is generated when the dog teeth on the gear and the teeth on the shifter make motions for the engagement. At this point of time, if the dog teeth are worn, the axial force is generated (Fig. 3) and the dog teeth jump out without engagement resulting in an incomplete shift maneuver.

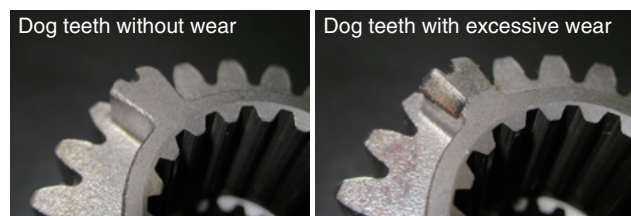


Fig. 1 Dog teeth wear

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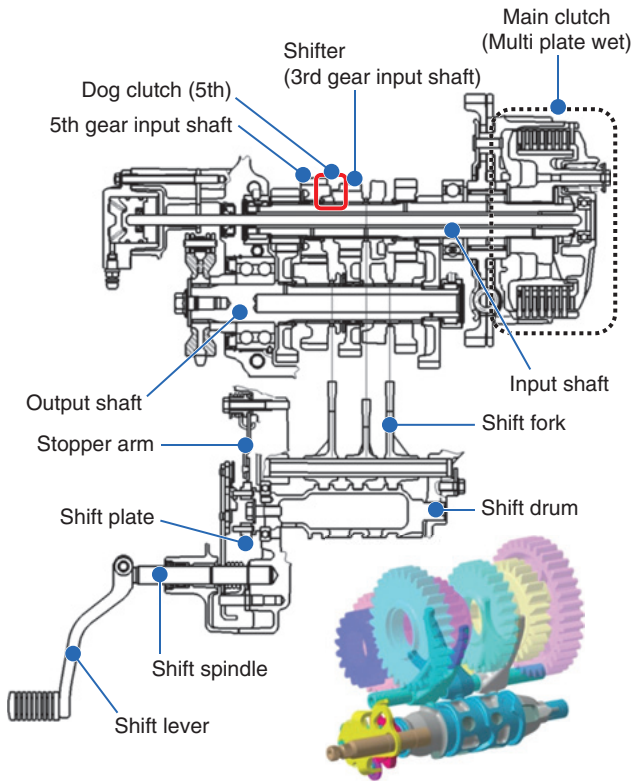


Fig. 2 Structure of dog clutch transmission

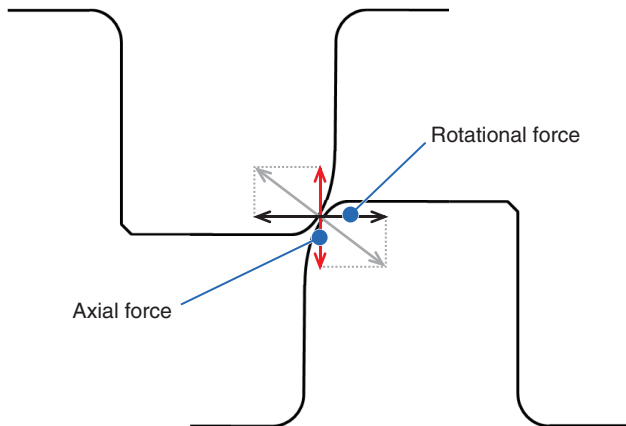


Fig. 3 Axial force on dog teeth

3. Transmission Behavior when Shifting

The behaviors of the transmission when shifting were investigated to clarify the dog teeth wear mechanism. Figures 4 and 5 show a typical example. Data which appear on these graphs are measured at the same time but are separately shown in the different graphs just for ease of comprehension except for the curve showing input shaft torque which is commonly appeared on both graphs.

Figure 4 shows the measurement data for the rotational angles of the shift drum and shift spindle, and the input shaft torque. When shifting, in the first place, the power

delivery between the engine and the transmission is cut off by the main clutch located between the crankshaft and the input shaft. This isolation makes the transmission free from the load and the shift drum begins to rotate in accordance with the rider's operation. When the shift drum rotates, the shifter slides and the dog teeth engage.

When the dog clutch engages, torque fluctuations are generated twice in succession on the input shaft. The shift drum continues to receive rotational force from the rider's operation, but the rider cannot generate large force enough to surpass the resisting torque produced on the input shaft. As a result, the shift drum rotation is stopped. (In the example shown in the figure, the shift drum rotation is stopped at a depth of approximately 30% of the maximum contact depth.) Then, when the torque produced on the input shaft is released, the shift drum begins to rotate again and the dog clutch engages completely.

In this manner, the torque which is large enough to stop the shift drum rotation is occurring at the moment of the dog clutch engagement. The dog teeth contact depth does not change until this torque is released. For this reason, when the dog teeth engage with small contact depth, a large contact stress is generated on the dog teeth contacting surfaces. Since dog teeth wear progresses gradually from the teeth top ends, this load at a small contact depth is thought to be a primary factor of producing wear.

Figure 5 shows the measurement data for the angular velocities of the crankshaft, the input shaft, the output shaft and the rear wheel, and the input shaft torque, where the angular velocities of individual shafts are converted into the equivalent values as those reflected on the velocities on the shaft on which the dog clutch installed. When the drive force is cut off by the main clutch, the input shaft maintains a constant angular velocity with its inertia.

Output shaft maintains its angular velocity as the rear wheel keeps rotating constantly with the stability derived from the inertia of vehicle body mass. The angular velocities of the input and output shafts change abruptly at the instant of the dog clutch engagement, and the two shafts exchange angular momentum, generating the first impact torque. The angular velocity of the rear wheel does not change at this time. It is because the output shaft and the rear wheel are connected by the drive chain that has some slack having them virtually separated at the moment of this first impact (Fig. 6).

Following that, the slack on the drive chain is tightened up with the increase of output shaft angular velocity caused by the first torque impact. At the moment when the slack chain is tightened up, the transmission system and the body system confront to each other, generating the second torque impact as a result (Fig. 6). After the second torque impact is vanished, the third and the fourth torque impacts in negative direction follow. However, those impacts have nothing to do with the wear of the dog teeth because the dog clutch has already fully engaged with the maximum engagement depth (i.e. the angular position of the shift drum has reached to the angle of 60 degrees which is for the next gear position). Consequently,

the first and the second torque impacts are the influencing factors that causes of the dog teeth wear. The second torque impact is smaller than the first one because the torsional rigidity of the body, which has a damper on the rear wheel, is far less than that of the transmission system. Based on this fact, the first torque impact was used in our study for the estimation on the wear amount of the dog teeth by calculating

the contact stress (force per unit area) on the dog teeth.

Meanwhile, it should be noticed that the wear which matters in this case is the one along the direction of the dog clutch engagement motion and the wear amount of dog teeth was defined as shown in Fig. 7, where the wear amount to be evaluated was the sum of the dog teeth wear on the engaging target gear and the wear on the shifter.

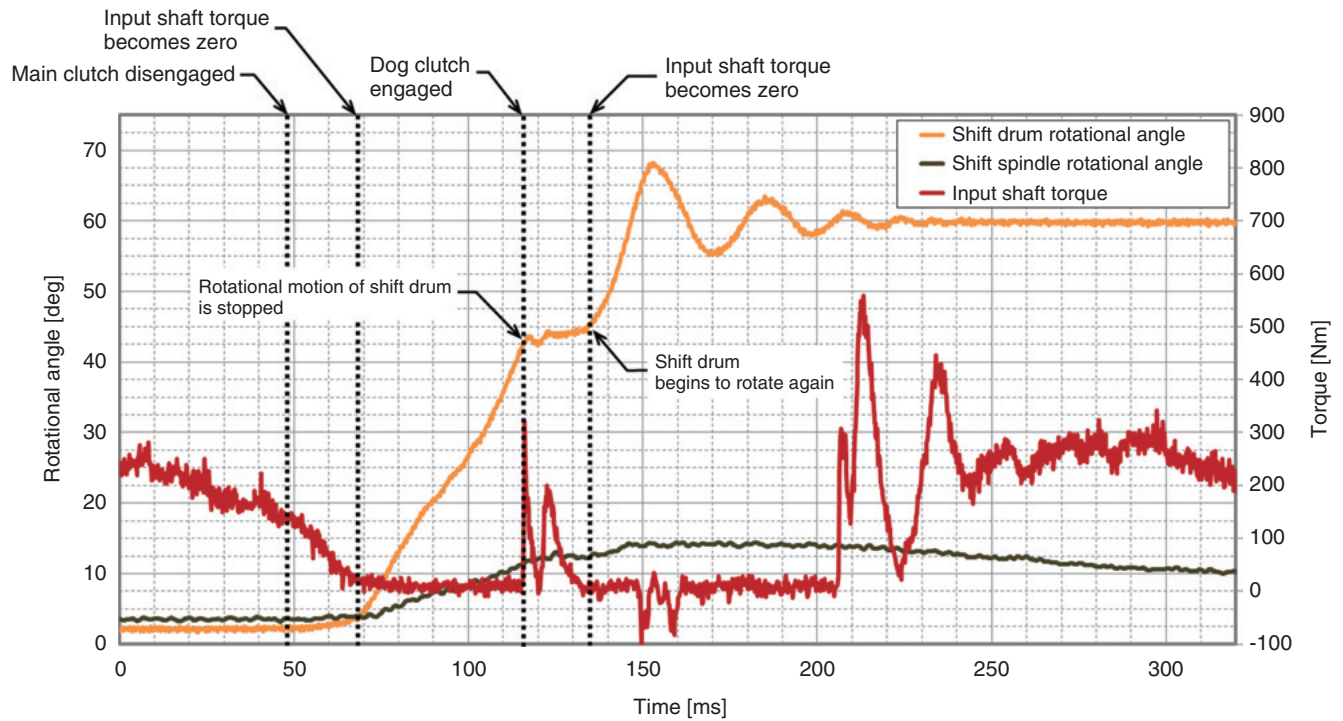


Fig. 4 Time history of rotational angle and torque when dog clutch engages

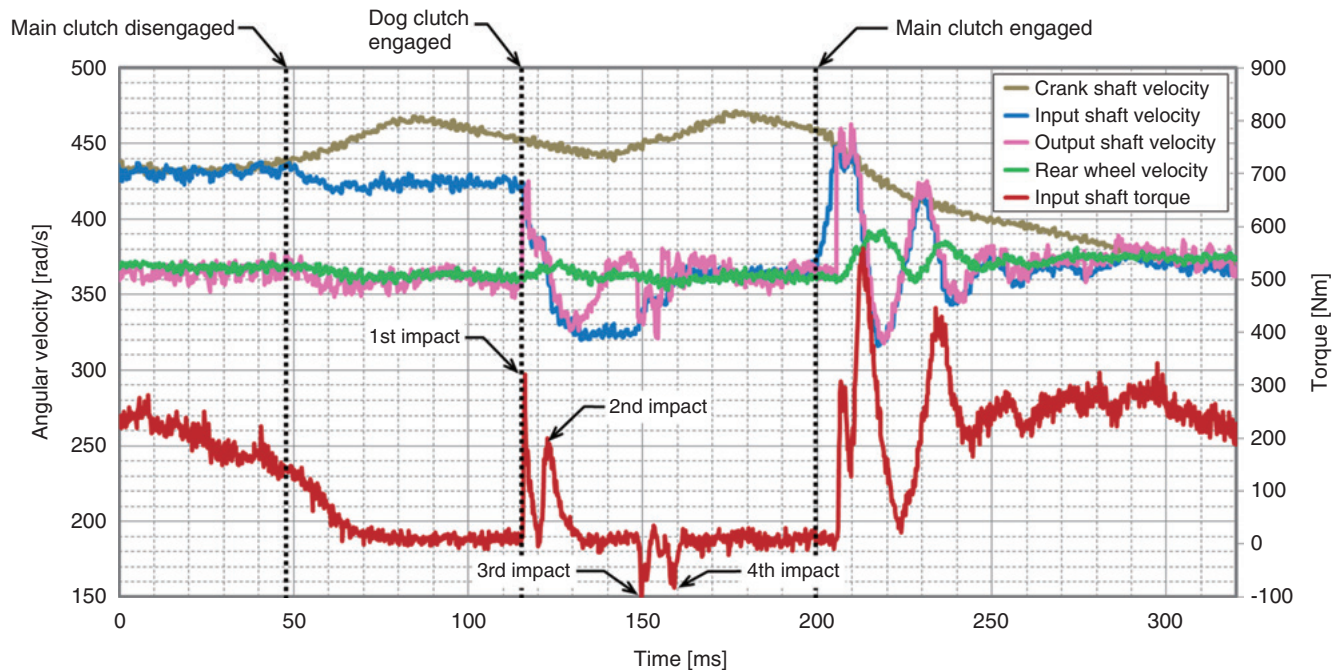


Fig. 5 Time history of angular velocities and torque when dog clutch engages

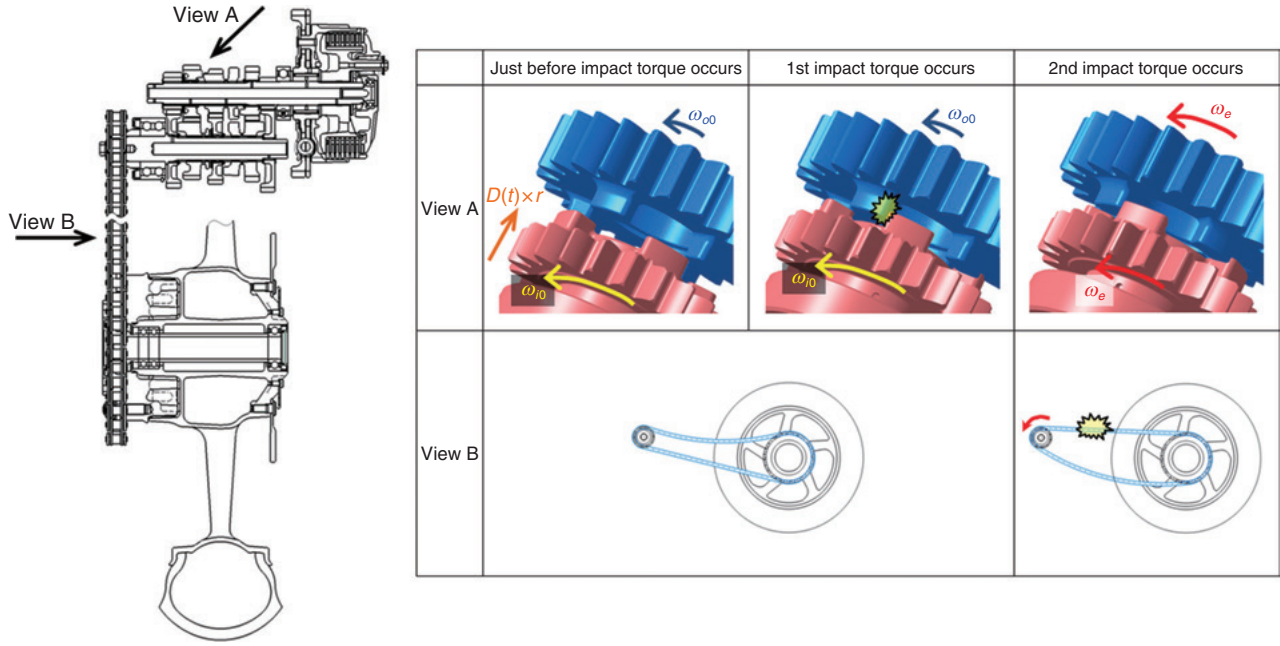


Fig. 6 Transition of drive-system action

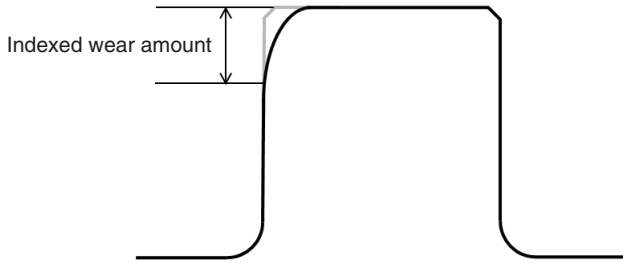


Fig. 7 Definition of Indexed wear amount

4. Torque Impact Associated with Dog Clutch Engagement

To begin with, we considered the first torque impact, which is generated through the engagement of the dog clutch, to analyze the contact stress applied to the dog teeth. The function of a transmission involves multiple dynamic elements, as shown in Fig. 2, with a lot of degrees of freedom, and represents the extreme complexity in its operational behaviors. However, when it is analyzed from macroscopic viewpoint, our research subject can be treated as the collision derived from rotation speed differences between the systems associated with the input shaft and that with the output shaft. Based on this approach of analysis, we estimated the torque impact for the total system converted into just two degrees of freedom for the input shaft system and the output shaft system.

It is easier to understand the kinetics of transmission which is actually a rotational dynamics if it is described as linear motion dynamics model as shown in Fig. 8. Firstly, we made an investigation on the coefficient of restitution to verify

whether or not the attenuation factors have to be taken into account when considering the dog clutch engagement takes place. The test parts were made of low carbon chromium molybdenum steel, applied with carburizing process, which is the same material generally used in the production of transmissions. The coefficient of restitution was 0.98 which is yielded from the data on the angular velocities before and after the collision by conducting the tests excluding the drive chain from the system to eliminate the effects from the body (rear wheel) to the angular velocities of the transmission. Consequently, the equation of motion representing the dog clutch engagement phenomena can be given disregarding the damping terms and is described as below.

$$I_i \cdot \ddot{\theta}_i = K \cdot (\theta_o - \theta_i)$$

$$I_o \cdot \ddot{\theta}_o = K \cdot (\theta_i - \theta_o)$$

Where $t = 0$ at the moment when the input shaft system and the output shaft system just start contacting.

Meanwhile, the torsion spring constant K is defined as the combination of the equivalent torsion spring constant for the input shaft system k_i and the equivalent torsion spring constant for the output shaft system k_o in series and given as:

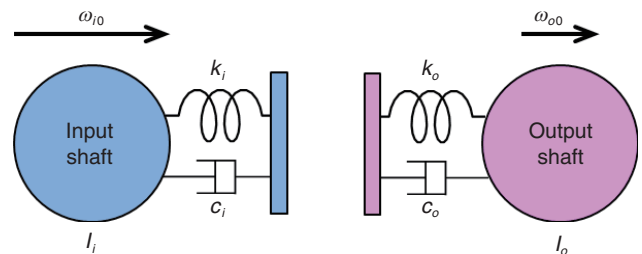


Fig. 8 Impact model

$$K = \frac{k_i \cdot k_o}{k_i + k_o}$$

For the next step, putting the relative angle of torsion between input shaft system and the output shaft system signified as θ ($\theta = \theta_i - \theta_o$), and setting:

$$I = \frac{I_i \cdot I_o}{I_i + I_o}$$

Then the equation of motion can be simply expressed as:

$$I \cdot \ddot{\theta} + K \cdot \theta = 0$$

The initial condition when $t = 0$ can be set as $\theta = 0$ and $\dot{\theta} = (\omega_{i0} - \omega_{o0})$ and the solution of the equation above is given as follows.

$$\theta = \frac{(\omega_{i0} - \omega_{o0})}{\omega_n} \cdot \sin(\omega_n t)$$

$$\omega_n = \sqrt{K/I}$$

Subsequently, by setting the impact torque, experienced when the dog clutch engages, to T and with the assumption that the torque is defined positive when its direction is to accelerate the vehicle, T is described as:

$$T = \omega_n \cdot \frac{I_i \cdot I_o}{I_i + I_o} \cdot (\omega_{i0} - \omega_{o0}) \cdot \sin(\omega_n t)$$

In addition, the maximum impact torque T_{max} is given when $t = \pi/(2\omega_n)$, and T_{max} is presented as follows.

$$T_{max} = \sqrt{\frac{k_i \cdot k_o}{k_i + k_o} \cdot \frac{I_i \cdot I_o}{I_i + I_o}} \cdot (\omega_{i0} - \omega_{o0})$$

I_i : Equivalent moment of inertia of the input shaft system [kg·m²]

I_o : Equivalent moment of inertia of the output shaft system [kg·m²]

θ_i : Equivalent angle of torsion of the input shaft system [rad]

θ_o : Equivalent angle of torsion of the output shaft system [rad]

k_i : Equivalent torsional spring constant of the input shaft system [Nm/rad]

k_o : Equivalent torsional spring constant of the output shaft system [Nm/rad]

ω_{i0} : Equivalent initial angular velocity of the input shaft system [rad/s]

ω_{o0} : Equivalent initial angular velocity of the output shaft system [rad/s]

ω_n : Natural angular frequency [rad/s]

t : Time [s]

T : Impact torque when the dog clutch engages [Nm]

T_{max} : Maximum impact torque when the dog clutch engages [Nm]

By using the actual motorcycle with a displacement of 1000 cm³, the torque was measured, while varying the angular velocity difference when shifting, to verify the accuracy of this equation for maximum torque impact. The measurement was carried out using a strain gauge at the point between the main clutch and the 1st gear on the input shaft. It was confirmed that the measurement error was around 10% in any measurement conditions, including the situations with different gear positions, through the comparisons between the measurement results and the calculated values. As a representative case, the results for the shift-up to the 2nd gear are shown in Fig. 9. The measurement results and the calculated values are showing good correspondence and the calculation method we developed for the maximum impact torque was validated.

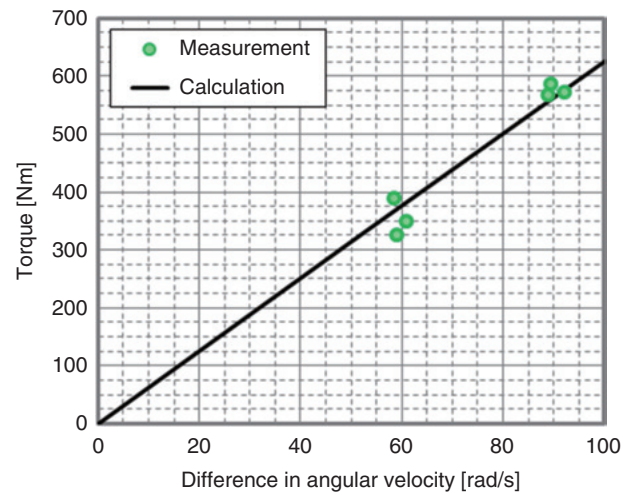


Fig. 9 Relation between impact torque and difference in angular velocity

5. Dog Clutch Engagement Area and Contact Stress

For the next step, we considered the dog tooth engagement area to evaluate the contact stress applied on the tooth. The dog tooth engagement area is defined as the multiplied value of the radial length by the depth of engagement (Fig. 10).

$$A = L \times H$$

A : Dog tooth engagement area [mm²]

L : Dog tooth radial length [mm]

H : Dog tooth depth of engagement [mm]

The depth of engagement H varies depending on the maneuvering characteristics of a rider (the transitional manners of the shift lever movement to the time progression) and the specifications of the dog clutch, while the radial length L can be determined simply from the set value in the design process. Therefore, in our study, we

moved forward to think about the way to calculate the contact stress through the organized investigations on the factors that involve the depth of engagement.

5.1. Shift Lever Operation Characteristics

It is already well recognized that the dog teeth experience significant wear when the rotational speed of engine is high. It is because the higher the engine rotational speed when shifting is the larger the relative rotational speed is, when the dog teeth engage. As the relative rotational speed goes up, the engagement depth gets smaller with the increased torque impact at the engagement. That's why the contact stress on the dog teeth becomes larger and the dog teeth undergo more wear as a result.

Accordingly, in our study, we conducted the tests to analyze the shift lever maneuvering characteristics, which has significant influences on the dog teeth engagement depth, in the tests under conditions of the shift changes at the maximum engine speed with full open throttle when accelerating and, when decelerating, under conditions of the shift changes at the engine speed that is supposed to generate the maximum engine speed after shifting with complete closed throttle.

5.1.1. Differences between situations with and without teeth edge faces contacts

There are two cases for dog teeth engagement. One of them is the case in which the dog teeth experience the contacts on their edge faces to each other. The other case is the case in which the dog teeth engagement occurs without the teeth edges' contacts. The typical data is shown in Fig. 11 for those two cases presenting the shift drum rotation angles with reference to the time. These angular velocity of the shift drum are equivalent until the dog teeth edges reach the position where their contact are about to occur. After that moment, the shift drum continues to rotate at about the constant angular velocity if the dog teeth edges

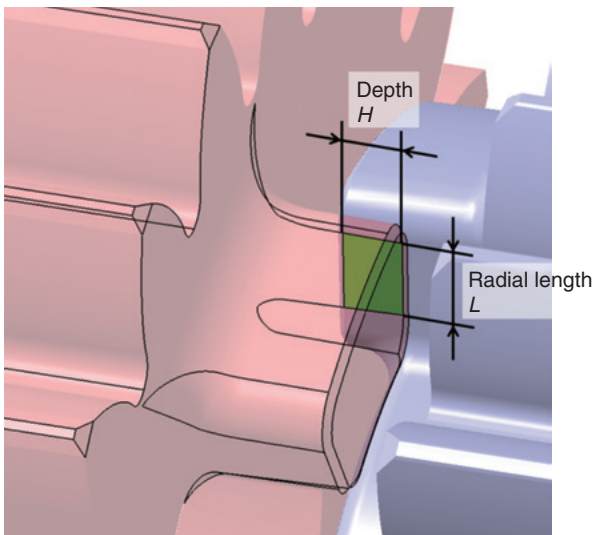


Fig. 10 Contact area of dog teeth

have no contacts. On the other hand, if the dog teeth edges (described as “tips of dog teeth” in figures) experience the contacts, the angular velocity of the shift drum goes down to zero for a short period because the movement of the shifter is inhibited. The shift drum angular velocity gradually increases after the contacts are dissolved. With those facts, it can be clearly understood that the situation with the contacts of the dog teeth edges are different from that of without them. Thus, the characteristics of the shift lever maneuvering are to be separately investigated depending on whether or not the dog teeth edges experience the contacts.

In our study, we conducted the averaging processes to the data of shift lever maneuvering on 10 riders of subjects to get the standardized maneuvering characteristics for shift lever operation. Based on that result, the approximation equation was yielded as shown below.

$$D_1(t) = \alpha \cdot t^4 + \beta \cdot t^3 + \gamma \cdot t^2 + \delta \cdot t$$

$$D_2(t) = \varepsilon \cdot t$$

$D_1(t)$: Displacement of shift lever peg when the dog clutch engaged with the teeth edge faces' contact [mm]

$D_2(t)$: Displacement of shift lever peg when the dog clutch engaged without teeth edge faces' contact [mm]

The time t is set as $t = 0$ at the moment when the dog teeth edges' confronting contacts finish in the case that the dog teeth edges have contacts, and is set, in the case that the dog teeth edges have no contacts, at the moment when the dog teeth advances in the axial direction and reach to the point that the teeth edge surface planes of both opposing dog teeth are aligned in a single plane. Where α , β , γ , δ and ε are the constants derived from the measurement results of the subjects tests.

5.1.2. Shift lever maneuvering characteristics when passing neutral position

In the next step, we examined the situation of shifting accompanied by the passing of the neutral position (hereafter

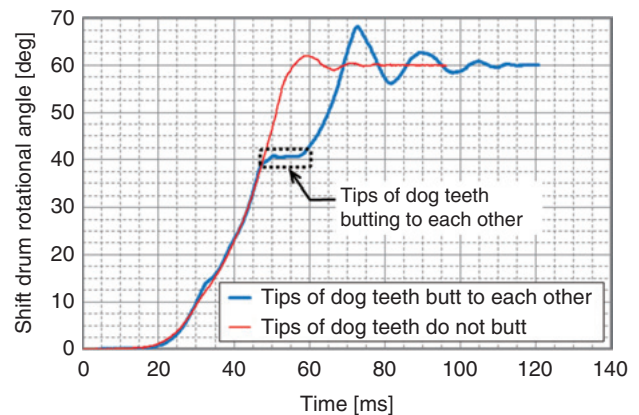


Fig. 11 Comparison of angular transition of shift drum

denoted as N) which is located between the 1st gear position and the 2nd gear position. In Fig. 12, the data of rotational angle of the shift drum is shown with reference to time when shifting from 1st gear position to 2nd gear position. Where 1st gear position corresponds to the shift drum rotational angle of 0 degree, 2nd gear position corresponds to that of 60 degrees and N corresponds to that of 30 degrees. As shown in Fig. 12, it is clearly noticed that the angular velocity of the shift drum varies at around 35 degrees which is the point just after passing N. This is because the roller placed on the stopper arm advances to contact the protuberance corresponding to N on the shift plate on its 2nd gear position side (Fig. 13).

During the shifting, the roller travels over the shift plate smoothly when it is not going through N. However, when there is N on its course, the roller is pushed toward the center of the shift plate by the spring placed on the stopper arm after passing over the protuberance located on the side of the gear position prior to N (i.e. 1st gear position side illustrated in Fig. 13). Consequently, the angular velocity of the shift drum decreases because of the collision of the roller against the protuberance located on the gear position subsequent to N (i.e. 2nd gear position side illustrated in Fig. 13). At this moment, the shift drum angular velocity goes down to zero for an instant and gradually goes up. This phenomenon shows the resembling behavior to the case where the dog teeth edges collide. Accordingly, we

compared the data of the shift drum rotation angle when the dog teeth edges had contacts, as shown in Fig. 11, with the data of that angle when shifting passing through N (Fig. 14). The results showed that the characteristics of those two shift drum rotational angles were equivalent. As a result, we have concluded that the characteristics for the case of passing through N can be evaluated by the characteristics equation $D_1(t)$ which is for the case that the dog teeth engage after edges' collision.

5.2. Contact Stress when Dog Clutch Engaged after Teeth Edge Faces Collision

When the dog teeth edges collide and confront to each other before engagement, the shifting maneuver is restrained during that confronting status lasts. The engagement occurs with the next immediate set of dog teeth when the confrontation of the dog teeth edges is terminated, because of the rotational speed difference, followed by the shifter advancement. The depth of dog teeth engagement can be increased only during the time from the termination of the dog teeth edges' confrontation until the engaging surfaces of the dog teeth reach to the contacting surfaces on the next dog teeth. Setting this time duration as the time available for dog clutch engagement t_s , it is defined by using the difference of angular velocities, the number of dog teeth and the angle of a dog tooth (Fig. 15).

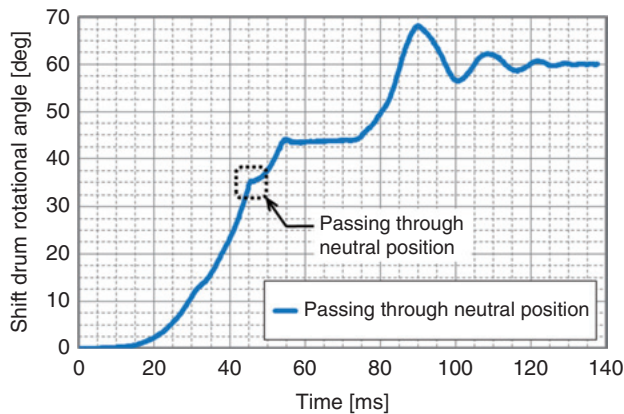


Fig. 12 Angular transition of shift drum in process of passing through neutral position

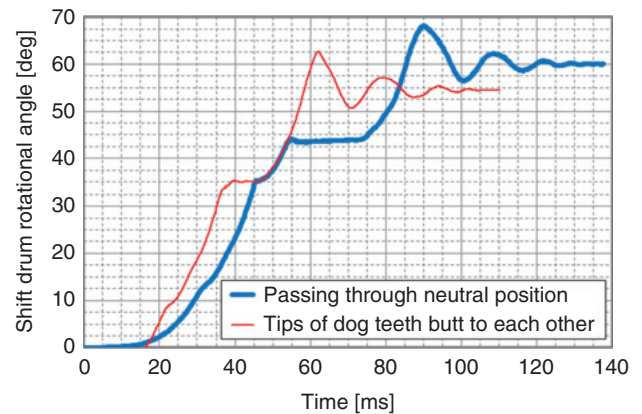


Fig. 14 Comparison of angular transition of shift drum with and without passage through neutral position

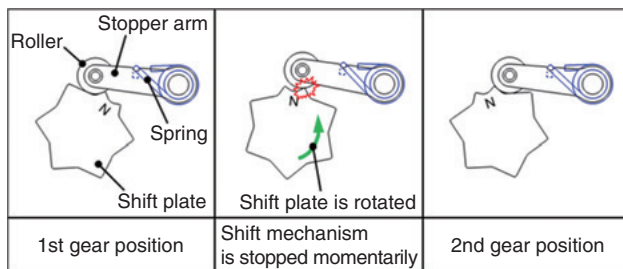


Fig. 13 Transition of shift mechanism action

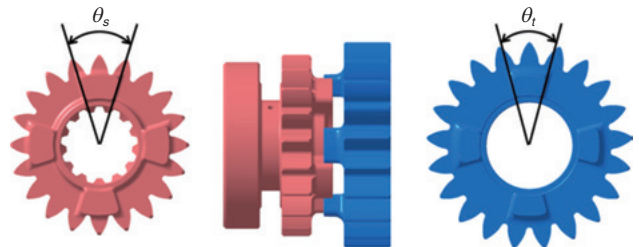


Fig. 15 Angles of dog teeth

$$t_s = \left\{ \frac{2\pi}{n} - (\theta_t + \theta_s) \right\} / (\omega_{i0} - \omega_{o0})$$

- t_s : Time available for dog clutch engagement [s]
 n : Number of dog teeth arranged at even angular intervals around the circumference
 θ_t : Dog tooth angle of engaging target gear [rad]
 θ_s : Dog tooth angle of shifter [rad]

The amount of displacement of the peg on the shift lever caused by a rider's maneuver, in the case that the dog teeth engage after experiencing the teeth edges' contact, can be determined by applying t_s , which is the amount of the time practically used for shifting, to $D_1(t)$. By using the ratio, which can be derived from the values set in the design process, between the displacement of the shifter and the displacement of the shift lever peg, together with the amount of chamfer at the top end perimeters of the dog teeth before use, the area that can be used for the dog teeth engagement is defined as shown in the next equation.

$$A_1 = L \cdot (D_1(t_s) \times r - 2c)$$

- A_1 : Dog tooth engagement area when the dog clutch engaged after the teeth edge faces' contact [mm²]
 r : Ratio between the displacements of the shifter and the shift lever peg
 c : Amount of chamfer at the perimeters of dog teeth top ends of the engaging target gear and the shifter before use [mm]

Consequently, the contact stress σ_1 is given as the equation shown below when the dog clutch engagement occurs accompanying the dog teeth edges' contact.

$$\sigma_1 = \frac{T_{max} \cdot 10^3}{(P/2) \cdot n} \cdot \frac{1}{L \cdot (D_1(t_s) \times r - 2c)}$$

- σ_1 : Contact stress when the dog clutch engaged with experiencing the teeth edge faces' contact [MPa]
 P : Effective diameter of dog teeth [mm]

However, there exists the case in which the value of $(D_1(t_s) \times r)$ exceeds the designed value of maximum possible depth of dog teeth engagement under the condition that the time available for dog clutch engagement is amply long. In this particular case, the area for engagement is defined using the designed value of the maximum dog teeth engagement depth instead of applying $(D_1(t_s) \times r)$.

5.3. Contact Stress when Engaged without Dog Teeth Edge Faces Confrontation

In the case that the dog teeth get engaged without contacts of tooth edges, the depth of the dog teeth engagement varies depending on the angular phase when the dog teeth reach to the point of the opposing dog teeth edge surfaces aligned in a single plane. The maximum engagement depth is brought about when the shifter is

advanced for full of the time available for the depth increase of the dog clutch engagement, which has been mentioned above. The minimum engagement depth is just above 0 mm (where the exact 0 mm cannot be regarded as an actual engagement). The depth of the dog teeth engagement can be any value with the same probability of occurrence within the range between these maximum and minimum engagement depths if the dog teeth do not experience the edges' contacts. Thus dog teeth contact stress can be evaluated using the averaged value calculated based on this range for the dog teeth engagement depth when the dog teeth get engaged without having edges' contacts.

However, the chamfered portions on the dog teeth generate a force component parallel to the gears axis if the engagement depth is smaller than the size of the initial chamfer at unused condition (Fig. 3). In consequence, the dog teeth cannot get engaged in the first chance as the shifter is repelled. The case with this repellent phenomenon should be separately considered and the only cases without repellent are picked up for the investigations of the contact stress here. The contact stress σ_2 is given as in the equation shown below for the engagement cases where the dog teeth edges do not experience the face contacts, by applying 10^{-5} mm for the engagement depth as a adequately small value but larger than 0 mm.

$$\sigma_2 = \frac{T_{max} \cdot 10^3}{(P/2) \cdot n} \cdot \frac{1}{L} \cdot \frac{1}{(D_2(t_s) \times r - 2c) - 10^{-5}} \cdot \int_{10^{-5}}^{D_2(t_s) \times r - 2c} \frac{1}{H} dH$$

- σ_2 : Contact stress when the dog clutch engaged without experiencing the teeth edge faces' contact [MPa]

5.4. Contact Stress when Engagement is Associated with Repellence against Shifter

As described before, the repellence occurs when the engagement depth is smaller than the size of the initial chamfer at the dog teeth edges. Through the investigations of the behaviors of the transmission parts when the repellence occurs, after the disengagement of the dog teeth which follows after the repellence, the dog teeth edges immediately establish the confronting contacts with the counterpart dog teeth edges because the continuing force, delivered through the shift mechanism from the rider's operation, is being applied. Following that status, the dog teeth get engaged with the next immediate counterpart dog teeth with advancement of the shifter when the dog teeth edges' contacts are dissolved. Consequently, when a shifting is accompanied by the repellence of dog teeth, the torque impacts are generated twice for one shifting maneuver but the first torque impact at the dog teeth repellence is not the primary concern about the dog teeth wear in this study because, at that impact, the load is applied only to the limited area of the chamfered portions on the dog teeth edges. With these considerations, our analysis

can be focused on the second torque impact. However, we will firstly describe about the results of the analysis on the repulsion phenomenon that leads to the first impact because the contact stress generated by the second torque impact is under influence of the physics that happens when the dog teeth experience the repulsion.

Through the analysis on the data measured for the cases with dog teeth repulsion (described as “jumping in and out” in figures), the magnitude of the first torque impact varies a lot ranging from a trivial amount to the extremely large quantity (Fig. 16). This variation range of the first torque impacts is defined by the variation range of how much angular momentum can be exchanged at the occurrence of the repulsion. With this relationship, the larger the torque associated with the dog teeth repulsion is the bigger the amount of the exchanged angular momentum is, and that is also translated as the larger exchanged amount of the angular velocity. Based on this analysis, it is likely to be assumed that the second torque impact becomes small when the first torque is large, as the angular velocity difference should have been decreased compared when

the first impact has occurred. However, the reality is that a large torque impact is still observed at the second collision in the dog teeth engagement even when the dog has already experienced a large torque influence through the dog teeth repulsion phenomenon. Accordingly, we decided to investigate the behaviors of the transmission parts when the dog teeth repulsion is happening (Fig. 17).

The angular velocity of the output shaft, which has relatively small moment of inertia, goes up abruptly when the repulsion happens. Therefore, at the moment just after experiencing the repulsion, the difference of the angular velocities sure is small as it is expected. However, the angular velocity, which has once increased after the repulsion, is brought back to the initial velocity before experiencing the repulsion. This is why the difference of angular velocities at the second torque impact becomes much larger on the contrary to the expectation. This is because of the influence from the inertia of the vehicle body delivered through the drive chain which is tightened getting rid of the slack by increased angular velocity of the output shaft. The moment of inertia of the body system is far larger compared to that of the output shaft system, and the angular velocity of the output shaft is brought back to the same speed as the one that corresponds to the movement of the vehicle body, that is to say, the angular velocity before experiencing the repulsion. The amplitudes of the angular velocities of the output shaft are the same at the first and the second collisions of the dog teeth. However, the difference in angular velocities at the second dog teeth collision becomes slightly smaller compared to that for the first collision because the change in the angular velocity caused on the input shaft through experiencing the repulsion has been maintained.

The amplitude of the change in input shaft angular velocity caused by the first impact with the dog teeth



Fig. 16 Time history of torque when dog clutch engages

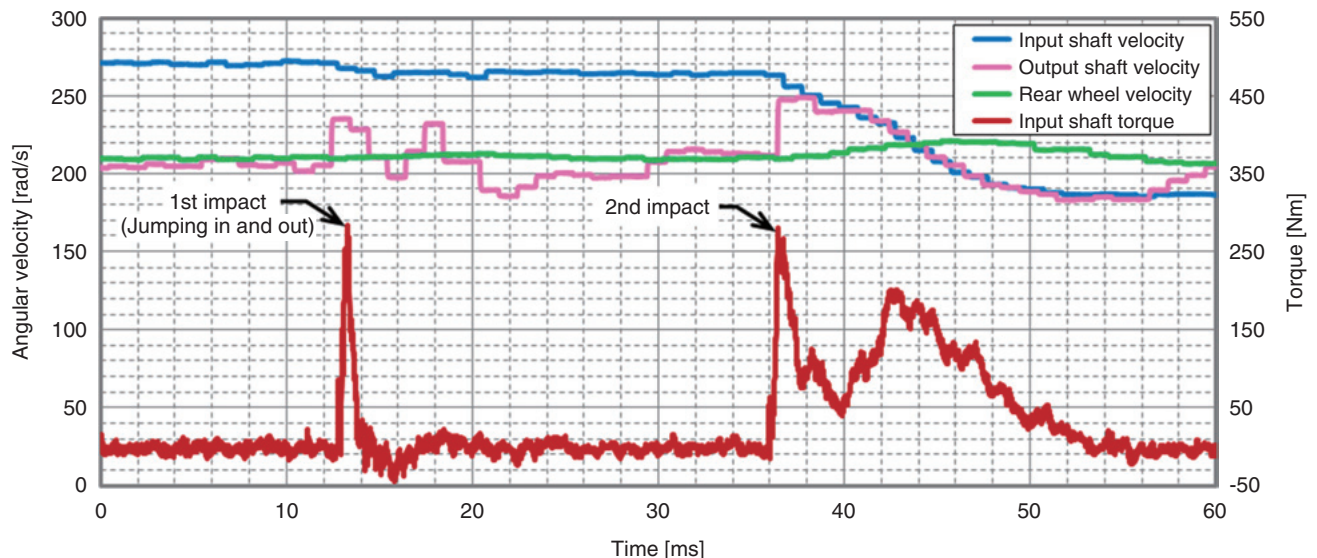


Fig. 17 Behavior of angular velocities and torque when jumping in and out

repellence varies depending on the magnitude of the exchanged angular momentum generated by this repellence. The first torque impact takes the value between 0 to T_{max} because the moment, at which the dog teeth get disengaged with the repellence, appears on the way that the impact torque increases to the maximum value. The angular velocity of the input shaft is the same as the initial angular velocity when the torque impact from the repellence is 0 without accompanying any angular momentum exchange. This is virtually the same as the case without occurrence of the repellence and, in this case, the second impact torque (it is actually not the second impact but the first impact) is T_{max} .

As the next subject, the case in which the magnitude of the first impact torque caused by the repellence reaches to the maximum value (T_{max}) will be considered. This case is the case that the angular velocity of the input shaft decreases the most. It is the moment when the angular velocities of the input shaft and the output shaft turn into the same value that the impact torque comes to the maximum amplitude associated with the first impact from the repellence. Setting the angular velocity to ω_e when this angular velocity match happens, the following formula is derived from the law of conservation of angular momentum.

$$\omega_e = \frac{I_i \cdot \omega_{i0} + I_o \cdot \omega_{o0}}{I_i + I_o}$$

The angular velocity of the input shaft gets ω_e just after the release of the dog teeth engagement when the first impact torque from the repellence becomes its maximum value. Accordingly, the second impact torque T_{min} which follows the first impact is described as in the formula shown below.

$$T_{min} = \sqrt{\frac{k_i \cdot k_o}{k_i + k_o} \cdot \frac{I_i \cdot I_o}{I_i + I_o}} \cdot (\omega_e - \omega_{o0})$$

Based on those mentioned above, T_{mean} which is the mean value of the second impact torque can be given as in the formula below.

$$T_{mean} = \frac{1}{2}(T_{max} + T_{min}) = \frac{1}{2} \cdot \sqrt{\frac{k_i \cdot k_o}{k_i + k_o} \cdot \frac{I_i \cdot I_o}{I_i + I_o}} \cdot (\omega_{i0} + \omega_e - 2\omega_{o0})$$

The area of the dog teeth engagement at the second collision is described as $L \cdot (D_1(t) \times r - 2c)$ because the dog teeth get engaged after experiencing the dog teeth edges' contacts. On the other hand, the angular velocity of the input shaft has already changed from the initial value and the time available for the dog clutch engagement is given as shown in the next formula.

$$t'_s = \left\{ \frac{2\pi}{n} - (\theta_i + \theta_s) \right\} / (\omega_e - \omega_{o0})$$

Consequently, the contact stress when the repellence occurs is shown as the following formula.

$$\sigma_3 = \frac{T_{mean} \cdot 10^3}{(P/2) \cdot n} \cdot \frac{1}{L \cdot (D_1(t'_s) \times r - 2c)}$$

ω_e : Angular velocity immediately after the maximum torque occurs [rad/s]

T_{min} : Minimum impact torque when the dog clutch engages after the repellence [Nm]

T_{mean} : Average impact torque when the dog clutch engages after the repellence [Nm]

t'_s : Time available for dog clutch engagement after the repellence [s]

σ_3 : Contact stress when the repellence occurs [MPa]

5.5. Contact Stress when Passing through Neutral Position

We will move on to the consideration in the case that the shifting maneuver goes over N. Because the characteristics of the shift lever maneuvers are different depending on whether or not the shift operation includes the passing through N, as mentioned above. However, there is no need to consider the influence of the passing through of N because the shifting maneuver is interrupted by the dog teeth edges' contact after passing through N when the engagement takes place following after the dog teeth edges' contact. Thus, we need to consider the influences of passing through N only for the cases in which the dog teeth get engaged without experiencing the edges' contact.

When the engagement is established without having the dog teeth edges' contact, the depth of the dog teeth engagement varies depending on the angular phase at the point that the teeth edge surfaces of the opposing dog teeth are aligned in a single plane. Through the former investigations, it is already known that the characteristic of the shift lever maneuver when passing through N can be evaluated by using the characterization value $D_1(t)$ for the case of the dog clutch engagement with the dog teeth edges' contact (Fig. 14). However, the time required to reach the point that the opposing dog teeth top edges get aligned in a plane should be taken into account because, when passing through N, the point from which the characteristic described by $D_1(t)$ is to be applied is not the same point at which the opposing dog teeth edges get aligned.

The starting point for the application of $D_1(t)$, when passing through N, is the point when the roller on the stopper arm establishes the contact with the protruded part of the shift plate on its side situated to encounter after passing N (Fig. 13). At this exact moment, the dog teeth top edge surfaces are not yet aligned to each other and there still some gap remains between the encountering dog teeth edges. Setting the time required to squeeze the gap to t_c , the maximum value of the dog teeth engagement depth, when experiencing the passage of N, is expressed by $(\{D_1(t_s + t_c) - D_1(t_c)\} \times r - 2c)$. (Where t_c can be defined by following the procedures shown below. In the first step,

the amount of the displacement of the shift lever, which provides the necessary advancement from the position that the roller on the stopper arm gets contact with the protrusion of the shift plate on its side coming after passing N to the position to reach the aligned position of the dog teeth top edge surfaces just before getting into the overlapped arrangement of the dog teeth, is defined by referring to the values already set in the system design. The value t_c can be obtained by putting this displacement to the inverse function of $D_1(t)$.

Consequently, σ'_2 can be defined as the following formula by applying the minimum amount of engagement depth of 10^{-5} mm which is assumed to be a sufficiently small value above 0 mm.

$$\sigma'_2 = \frac{T_{max} \cdot 10^3}{\left(\frac{P}{2}\right) \cdot n} \cdot \frac{1}{L} \cdot \frac{1}{\left(\{D_1(t_s + t_c) - D_1(t_c)\} \times r - 2c\right) - 10^{-5}} \cdot \int_{10^{-5}}^{\{D_1(t_s + t_c) - D_1(t_c)\} \times r - 2c} \frac{1}{H} dH$$

t_c : Time required to squeeze the gap between the edge faces of the dog teeth after passing through N [s]

σ'_2 : Contact stress when the dog clutch engaged without experiencing the teeth edge faces' contact after passing through N [MPa]

6. Estimating Degree of Dog Teeth Wear Caused by Contact Stress

The contact stress applied on the dog teeth varies depending on the cases whether it associated with the top edge surfaces' contact of the dog teeth before engagement, or it establishes the engagement without top edges' contact, or it is the case with the repulsion against the dog teeth. (When it is not associated with dog teeth top edges' contact, whether or not it is accompanied by the passing through of N makes the difference as well.) Accordingly, the combined total contact stress is defined as σ by adding up the contact stresses multiplied by each probability of occurrence of the individual cases.

$$\sigma = \sum_{j=1}^3 \sigma_j \cdot p_j$$

However, in the case of experiencing the passage of N, the evaluation is carried out using σ'_2 instead of σ_2 as the value of the contact stress when the dog teeth top edges' contact does not occur. On the other hand, the probabilities of occurrence for individual cases are provided by following formulae.

$$p_1 = \frac{(\theta_t + \theta_s)}{2\pi/n}$$

$$p_2 = (1 - p_1) \cdot \frac{D_2(t_s) \times r - 2c}{D_2(t_s) \times r}$$

$$p_3 = (1 - p_1) \cdot \frac{2c}{D_2(t_s) \times r}$$

σ : Combined total contact stress [MPa]

p_1 : Probability of the dog clutch engagement with experiencing the teeth edge faces' contact

p_2 : Probability of the dog clutch engagement without experiencing the teeth edge faces' contact

p_3 : Probability of the occurrence of the repulsion

We conducted the durability tests for the actual motorcycles with the displacements of 700 cm³, 900 cm³ and 1200 cm³ in order to verify the relationship between this combined total contact stress and the degree of dog teeth wear. The durability tests are carried out under the test conditions of the shift changes at the maximum engine speed with full open throttle when accelerating and, when decelerating, under conditions of the shift changes at the engine speed which is supposed to generate the maximum engine speed after shifting with complete closed throttle, and the same number of shift maneuvers were performed for each gear position. The carburized low-carbon chromium molybdenum steel, which is commonly used for the production of the transmission parts, was applied for the material used for the test parts. As for the measurement of the dog teeth wear amount, a profilometer was used for the investigations.

The relationship between the contact stress and the dog teeth wear amount after the durability tests is shown in Fig. 18. The significant correlation was confirmed with the correlation coefficient of 0.96 between the contact stress and the dog teeth wear amount.

Through the results mentioned above, the evaluation method based on the contact stress estimation has been proved to be an effective measure which enables the prediction of the dog teeth wear amount with high accuracy providing important information in the process of the design for transmissions employing dog clutches.

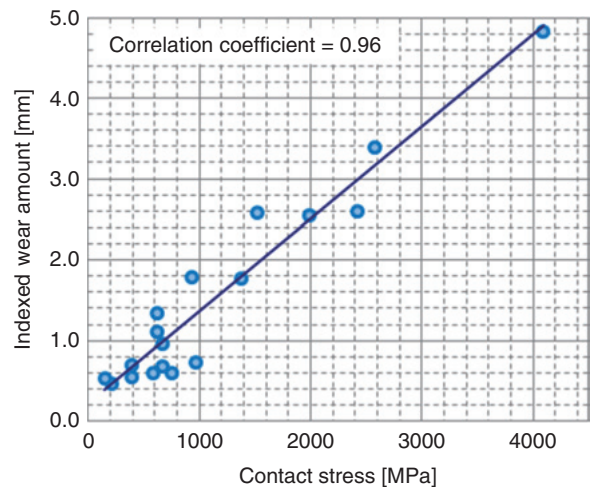


Fig. 18 Relation between indexed wear amount and contact stress

7. Conclusions

The study on the wear of dog teeth in transmission systems with dog clutches was conducted and the following results were obtained.

- (1) The mechanism of the wear associated with dog clutch engagement was clarified.
- (2) The procedure to calculate the impact torque associated with dog clutch engagement was established.
- (3) The evaluation method for the engagement depth associated with dog clutch engagement was established.
- (4) It was made possible to estimate the amount of dog teeth wear through establishing the calculation method to yield the contact stress applied on the teeth of a dog clutch.

Through those results, the investigation on specifications relating to dog teeth wears with high accuracy was made feasible in the earlier stages of the design process.

■ Author ■



Hiroyuki MAKITA

Fuel Consumption and Performance Prediction in Outboard Motors for High-speed Planing Boats using CFD Simulation

Shunsuke KURIOKA*

ABSTRACT

The research discussed in this paper set out to develop a simulator able to quantitatively predict the cruising fuel consumption and cruising performance of an outboard motor used for a small, high-speed planing boat; the target accuracy for prediction was within a range of $\pm 5\%$. The research focused on the fact that the thrust performance of a propeller has a significant impact on cruising fuel consumption and cruising performance. In order to evaluate the accuracy of the fuel consumption and performance simulator, simulation results were compared with the results of a full-scale cruising test. This comparison suggested that the simulator was able to satisfactorily predict the cruising fuel consumption and cruising performance of outboard motors. In particular, prediction results agreed with full-scale measurement results to within a range of $\pm 2.79\%$ (± 1.7 km/h) for maximum speeds and $\pm 4.55\%$ for cruising fuel consumption. These results indicate that the simulator is able to quantitatively predict the cruising fuel consumption and the cruising performance of an outboard motor. More importantly, the results demonstrate that the simulator can be used to identify optimal propeller specifications and gear reduction ratios, which have traditionally been based on experience. It was therefore concluded that the simulator possesses considerable potential as an effective tool able to be used in the development of environmentally friendlier outboard motors with outstanding cruising fuel economy and cruising performance.

1. Introduction

The world's first practical outboard motor was developed in the 1900s in the U.S. More than 100 years later, outboard motor engines offer pleasure and utility to users throughout the world as compact, lightweight, and high-power sources of drive power for small boats. Historically, the majority of outboard motors have used lightweight 2-stroke motors with simple configurations. However, there is a global demand for increased levels of environmental performance in internal combustion engines, and the engines used in outboard motors are no exception. Today, the use of 4-stroke engines in outboard motors in order to satisfy stringent exhaust gas regulations and environmental standards is increasing, and the manufacturers of engines for outboard motors have continued to rise to the technological challenge of balancing environmental performance and cruising performance⁽¹⁾⁻⁽³⁾. With an expansion in the scale of developments due to the recent trend towards larger displacement and increased horsepower in outboard motor engines, the reduction of development costs and the

realization of increased development efficiency are two of the most important issues facing manufacturers. One of the keys to the realization of these goals is the prediction of a variety of aspects of performance using simulations⁽⁴⁾⁻⁽¹⁰⁾.

Experience of outboard motor development to date has shown that even for identical motor engines, differences in propeller specifications and gear reduction ratios have a significant effect not only on cruising performance, but also on fuel consumption when the unit is in operation. The research discussed in this paper focused on the fact that propeller thrust performance significantly affects cruising fuel consumption, and conjectured that if it were possible to accurately determine propeller thrust performance, including the gear reduction ratio, by means of CFD simulation, and create a database combining engine performance data and fuel consumption maps obtained from tests, it would be possible to make numerical predictions of cruising fuel consumption, which have conventionally been obtained from tests or based on experience, at the initial stage of development.

The goal of the research was to develop a simulator

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capable of numerical predictions of cruising fuel consumption and cruising performance for an outboard motor used to power a small, high-speed planing boat to within an accuracy of $\pm 5\%$.

2. Overview of Simulator for Prediction of Fuel Consumption and Cruising Performance

2.1. Configuration of Simulator Database

The simulator for the prediction of fuel consumption and cruising performance in an outboard motor for a small, high-speed planing boat proposed in this research is made up of four databases, comprising engine power data, fuel consumption maps, hull resistance data, and propeller performance data (Fig. 1). Experimentally determined engine power curves are stored in the engine power database. The corresponding fuel consumption maps are similarly determined by experiment. The hull resistance database stores several types of data generated by a one-dimensional simulator created on the basis of experience from data derived from tests. The propeller performance database stores predictions and data based on CFD simulation.

The accuracy of the results of CFD simulation of propeller performance was verified by comparison with the results of model tests conducted in a towing tank. This process will be discussed later in the text.

2.2. Method of Prediction of Cruising Fuel Consumption and Cruising Performance

This section will discuss the prediction method employed by the simulator for the prediction of fuel consumption and cruising performance in an outboard motor for a small, high-speed planing boat proposed in this research. The method comprises the following three steps.

2.2.1. Prediction of propeller performance

Prediction of propeller performance is made up of

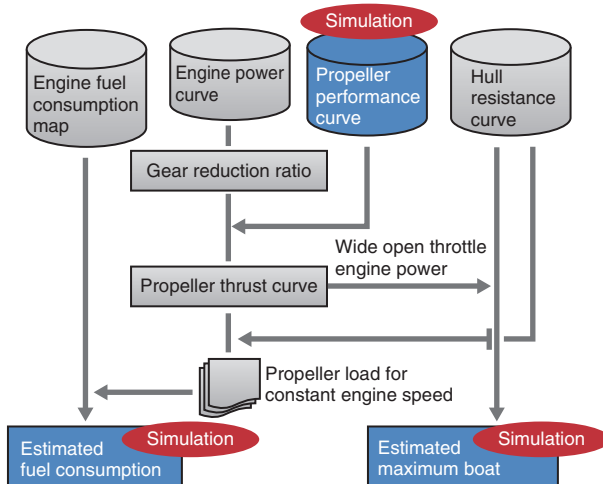
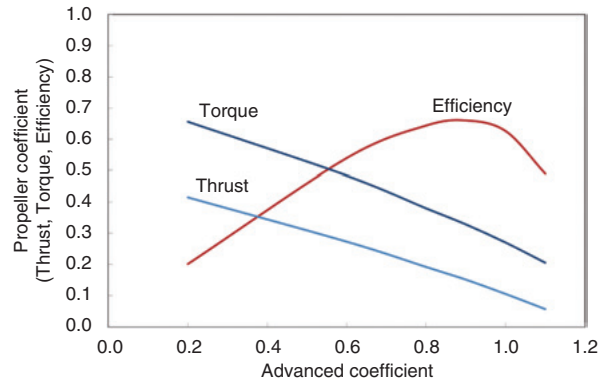


Fig. 1 Database components of simulator

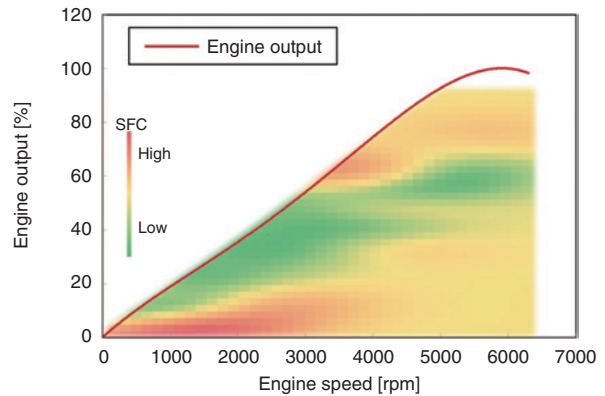
propeller performance curve [Fig. 2(a)] and engine power curves that take into consideration the horsepower loss from the engine to the propeller shaft and the gear reduction ratio of the propeller shaft [Fig. 2(b)]. Propellers used for planing boats generally convert input torque into thrust with a 60–80% range of propeller efficiency. Propeller thrust curves are estimated in relation to engine power on this basis.

2.2.2. Prediction of maximum cruising speed

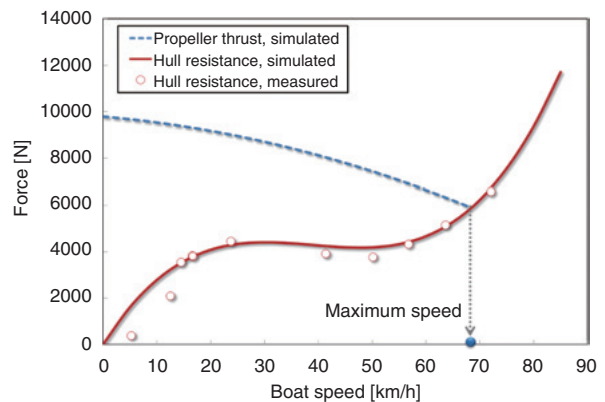
This prediction method combines the estimated propeller



(a) Propeller performance curves (Nondimensionalized)



(b) Engine power curve and fuel consumption map



(c) Hull resistance and propeller thrust curves

Fig. 2 Sample diagrams of database used in simulator

thrust curve and hull resistance curve. The propeller thrust curve combined with the engine power curve provides the thrust curve in relation to engine speed. In order to combine this curve with the hull resistance curve, it is necessary to convert it into a thrust curve in relation to cruising speed. The speed at which hull resistance and propeller thrust are in equilibrium is the final maximum speed of the planing boat [Fig. 2(c)].

2.2.3. Prediction of cruising fuel consumption

The propeller thrust curve calculated in Section 2.2.1 uses engine power at wide-open throttle. In order to evaluate cruising fuel consumption at a constant engine speed, that is, a state of partial throttle, propeller thrust curves for various engine speeds are formulated. Using the efficiency expressed in the propeller performance curve, this propeller thrust is converted to the engine horsepower absorbed by the propeller. Cruising fuel consumption is estimated by reading the absorbed horsepower for each engine speed off a fuel consumption map [Fig. 2(b)].

It was assumed that given the configuration of this simulator, realizing a level of accuracy of prediction of propeller performance exceeding that of prediction by CFD would represent a challenge. Based on the fact that the predictive accuracy of the propeller CFD used in the initial stages of the development process to date was around $\pm 5\%$, the target predictive accuracy for the simulator was set at $\pm 5\%$.

3. Method of Numerical Computation of Propeller Performance

3.1. CFD Solver

The research discussed here used commercially available CFD simulation software in order to predict propeller performance. The simulation solver was a CFD solver based on a finite volume method that handles unstructured grids, the efficacy of which in CFD analyses for propeller performance prediction has been indicated in a number of reports⁽⁶⁾⁻⁽⁹⁾.

The fluid equation was a standard incompressible Reynolds averaged Navier-Stokes (RANS) equation. The research adopted a method of steady-state analysis in which the coordinate system was rotated using the multi-rotating frame (MRF) method⁽¹¹⁾. The semi-implicit method for pressure-linked equations (SIMPLE) was used as the discretization method for pressure equations, while the high-accuracy quadratic upstream interpolation for convective kinematics (QUICK) scheme was used for momentum. A second-order central difference scheme was applied for space discretization accuracy apart from this.

3.2. Turbulence Model

This section will discuss the turbulence model used in the CFD simulation. The commercial software used in the research offered several types of turbulence model.

Turbulence models should be selected carefully with consideration of the flow fields of the subject of simulation. Because propeller performance was being databased in the present research, a $k-\omega$ SST model offering robust performance was selected based on the assumption that several hundred cases of CFD simulation would be conducted. The advantages presented by this turbulence model in research on CFD simulation for propeller performance prediction have already been reported by a number of researchers⁽⁶⁾.

3.3. Computational Domain

This section will discuss the analytic domain set for the CFD simulation. For this research the computational domain and the computational mesh size were rendered dimensionless by means of the propeller diameter, D ⁽⁶⁾. This offered the advantages of making it easier to realize accuracy and making it possible to compare results with the results of past research even in the case of predictions for propellers of differing diameters. The analytic domain was the domain shown by the cylinder in Fig. 3. With the propeller as the point of origin, the upstream side was set as distance $1.5D$, the downstream side as $3D$, and the radial direction as $2D$. The shape of the shaft at the front and rear of the propeller was the same as the shape used for the model in the towing tank test. The details of this test will be discussed later in the text.

3.4. Computational Mesh

This section will discuss the computational mesh used in the CFD simulation. Commercial grid generation software was used for the generation of the computational mesh. In terms of mesh type, tetra elements were employed for standard sections, while triangular elements called prism elements were used for boundary layers close to wall surfaces in order to increase analytic accuracy close to the wall surfaces. Propeller performance was calculated from thrust and torque calculated from the force obtained by integration of the pressure on the surfaces of the propeller blades. Given this, the computational resolution of the surfaces of the propeller blades was the most important factor. The mesh for the propeller surfaces was set at $D/200$, and the mesh for the first boundary layer was set at

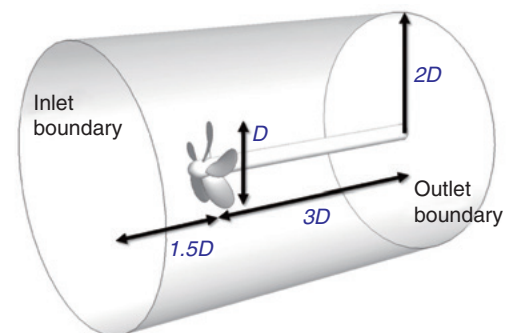


Fig. 3 Computational domain for propeller simulation

Table 1 Principal particulars of computational mesh for propeller simulations

Propeller diameter	D
Outside boundary surfaces	$D/4$
Propeller surfaces	$D/200$
1 st height of boundary layer	$D/1000$
Number of boundary layers	15
Growth rate of boundary layers	1.05
Growth rate of volume cells	1.20

$D/1000$. Fifteen boundary layers were employed, and the mesh growth rate for the boundary layer was set at 1.05. The computational mesh for the standard sections of the computational domain grew steadily larger at a growth rate of 1.2 from the surfaces of the propeller blades, and the outermost mesh of the computational domain was restricted to be $D/4$ at its maximum. Figure 4 shows an example of the computational mesh. Table 1 shows the mesh setting parameters.

3.5. Boundary Conditions

This section will discuss the boundary conditions set for the CFD simulation. A steady-state analysis was conducted, with a constant angular velocity imparted to the coordinate system. A velocity inlet boundary was employed in front of the propeller, and a pressure-fixed outlet boundary was employed behind the propeller. A free slip wall boundary was used for the walls of the cylindrical computational domain, and a non-slip wall boundary was used for the propeller blade surfaces. Propeller performance was derived from calculations of the thrust and torque coefficients, conducted while altering the speed of the inflow to the propeller.



Fig. 4 Propeller CAD model (left) and computational mesh (right)

4. Results and Considerations

4.1. Unit Performance of Propeller

For the simulator proposed by this research, propeller performance represented an important element that affects the cruising fuel consumption and thrust performance of an outboard motor and the selection of specifications for its

gear reduction ratio. Given this, high-accuracy prediction of propeller performance via CFD simulation was closely related to the realization of a high level of accuracy in the prediction results produced by the simulator. This chapter will discuss the accuracy of CFD simulation in relation to the prediction of propeller performance.

4.1.1. Benchmark computations (Propeller for displacement-type boats)

Prior to its use in predicting the performance of propellers used for planing boats, benchmark computations were conducted for the CFD solver employed in this research using publicly available test results for a five-bladed propeller used for displacement-type boats. The propeller used in the computations was the Seiumaru-I five-bladed propeller for displacement-type boats, generally termed a conventional-type propeller. This propeller has previously been used by numerous researchers in benchmark computations for model tests and numerical tests⁽⁶⁾. Figure 4 shows a CAD model and computational mesh for this propeller.

In this research, propeller performance was evaluated using performance curves that are in general use in the field of boat engineering. The performance curves were rendered dimensionless using the advance coefficient J , the thrust coefficient K_T , the torque coefficient K_Q , and propeller efficiency η , and were expressed using the following equations.

$$J = \frac{V_A}{nD}$$

$$K_T = \frac{T}{\rho n^2 D^5}$$

$$K_Q = \frac{Q}{\rho n^2 D^4}$$

$$\eta = \frac{J \cdot K_T}{2\pi \cdot K_Q}$$

Here, n is the propeller speed, D is the propeller diameter, V_A is the inflow velocity to the propeller, and ρ is fluid density. Figure 5 compares CFD simulation results with the results of model tests drawn from a published reference⁽¹²⁾.

The comparison between CFD simulation results and model test results for propeller thrust and torque coefficients indicated levels of predictive accuracy from the design point (approximately 80% of the point of maximum efficiency) to the point of maximum efficiency for CFD simulation of $\pm 2.8\%$ for thrust coefficient and $\pm 2.0\%$ for torque coefficient.

4.1.2. Benchmark calculations (Propeller for planing boats)

Performance predictions for a three-bladed planing boat

propeller were conducted using CFD simulation. In order to verify the predictive accuracy of the CFD simulation results, they were compared with the results of towing tank model tests. The towing tank model tests for this research were conducted at the Shipbuilding Research Centre of Japan in Mitaka City, Tokyo. Figure 6 shows the model propeller and the computational mesh used in the tests. The propeller was a 60% scale model manufactured by aluminum machining. As in the previous section, the performance of the three-bladed planing boat propeller was evaluated by means of performance curves. Figure 7 shows a comparison between the results of the towing tank model tests and the results of CFD simulation. The comparison between CFD simulation results and model test results for propeller thrust and torque coefficients indicated levels of predictive accuracy from the design point (approximately 80% of the point of maximum efficiency) to the point

of maximum efficiency for CFD simulation of $\pm 4.55\%$ for thrust coefficient and $\pm 4.58\%$ for torque coefficient. Generally, the effects of cavitation, which can be predicted to have an effect of several percent on efficiency, should be considered in the case of planing boat propellers. However, CFD simulation that takes cavitation into consideration presents issues of stability. Given that the present research prioritized computational stability, and conducted several hundred cases of CFD simulation in order to achieve stability, it was decided to ignore the effects of cavitation.

4.2. Prediction of Cruising Performance

In order to verify the predictive accuracy of the simulator proposed in this research, results for the cruising performance of a large-horsepower class outboard motor fitted to a high-speed planing boat were compared with the results of full-scale cruising tests. The planing boat used in this verification of the predictive accuracy of the simulator for cruising performance was a standard three-to-four passenger pleasure boat known as an offshore boat (Fig. 8). Any specific type of planing boat propeller will generally be sold with several propeller pitches set, making it possible for the user to select cruising performance characteristics depending on whether maximum cruising speed or acceleration performance is prioritized. Propeller selection will differ depending on the type of planing boat being used, and outboard motor manufacturers and sales outlets will normally offer assistance such as propeller matching charts that show recommended propellers.

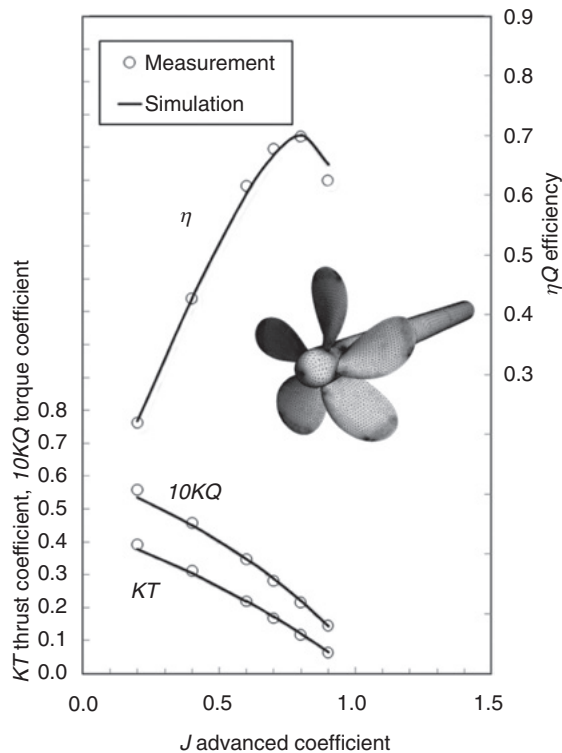


Fig. 5 Propeller performance curves compared with model test results

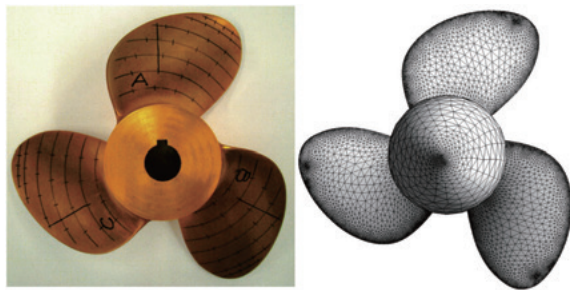


Fig. 6 Model propeller (left) and computational mesh (right)

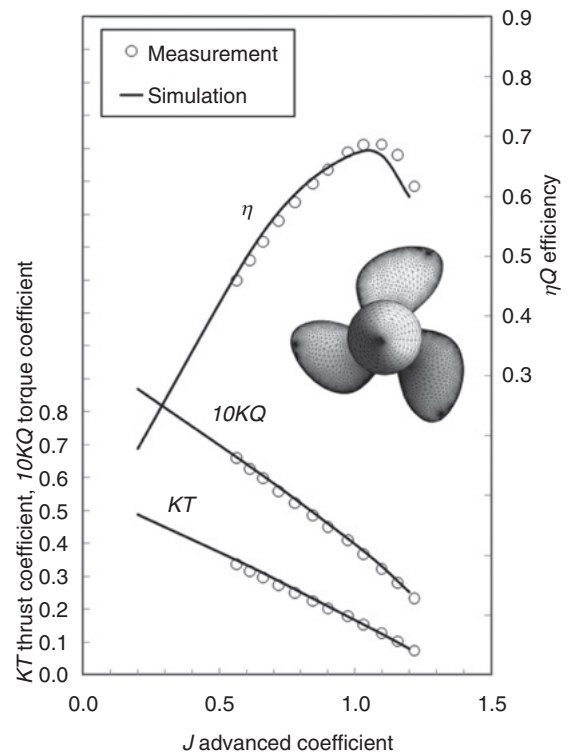


Fig. 7 Propeller performance curves compared with model test results

Table 2 Principal particulars of propellers used in evaluation of simulator

Propeller name	Photo	Diameter [inch]	Pitch [inch]	Gear reduction ratio
A-1		14.7	15.0	2.00
A-2			17.0	
A-3			19.0	
B		16.0	18.5	2.15
				2.33
				2.53
C		14.3	15.0	1.85
				2.15
				2.33

Given this, in the present research predictive accuracy was verified by conducting cruising performance predictions for planing boat propellers with different propeller pitches and two different types of hull. In addition, in order to study the effect on cruising performance of the gear reduction ratio, the framework of an outboard motor drive system, predictions were also conducted for cases in which the gear reduction ratio differed for the same propeller. Table 2 shows the planing boat propeller specifications employed. Propellers A-1, A-2, and A-3 were fitted to the hull shown in Fig. 8(a), and propellers B and C were fitted to the hull shown in Fig. 8(b).

Cruising performance predictions using the simulator proposed in this research demonstrated a predictive accuracy of $\pm 2.79\%$ (± 1.7 km/h) for maximum cruising speed. Figure 9 compares results for prediction of maximum cruising

speed and the results of tests using actual boats. These results indicate that the simulator is capable not only of predicting cruising performance in the case of differences in hull type and propeller pitch, but also in the case of differences in the gear reduction ratio, the framework of the drive system. The computations conducted by the simulator are automated via spreadsheet script software, and computations for a single propeller specification can be conducted in several seconds. In addition, because the different types of data can be managed by spreadsheet software or text data, the simulator also presents the advantage of simple handling of data. This indicates that the simulator will be sufficiently flexible for use at the design stage of the outboard motor development process.

4.3. Prediction of Cruising Fuel Consumption

In order to verify the predictive accuracy of the simulator developed in this research, results for the cruising fuel consumption of a large-horsepower class outboard motor fitted to a high-speed planing boat were compared with the results of full-scale cruising tests. The same hulls and outboard motor as described in the previous section were used in this verification. However, in order to use the same conditions as the test conditions for measurement of cruising fuel consumption, cruising fuel consumption at a fixed engine speed was evaluated at a variety of engine speeds. As in the previous section, predictive accuracy was verified by conducting predictions of cruising fuel consumption for planing boat propellers with different propeller pitches and two different hulls. In addition, predictions were also conducted for different gear reduction ratios for the same propellers in order to verify the effect of this parameter, the framework of outboard motor drive systems, on cruising fuel consumption. The planing boat propeller specifications employed were identical to those shown in Table 2.

The results for prediction of cruising fuel consumption by the simulator corresponded well with test results. In particular, comparison of prediction results for cruising fuel consumption at 40 km/h and above when the boat is planing with the results of cruising tests show an average predictive accuracy of $\pm 4.55\%$, within the target range of $\pm 5\%$ set for this research. This is equivalent to the accuracy of the benchmark computations of propeller performance discussed in Section 4.1.2. Figure 10 shows a graph comparing the results of prediction of cruising fuel consumption for propeller A with the results of tests using an actual boat. This is a case in which a difference in propeller pitch produces a difference in fuel consumption, but the simulator is able to reproduce the fact that propeller A-3 offers better fuel consumption figures than propeller A-1. Figures 11 and 12 show graphs comparing the results of predictions of cruising fuel economy for propellers B and C with the results of actual boat tests. These are cases in which a difference in the gear reduction ratio for the same propeller pitch results in differences in fuel consumption,

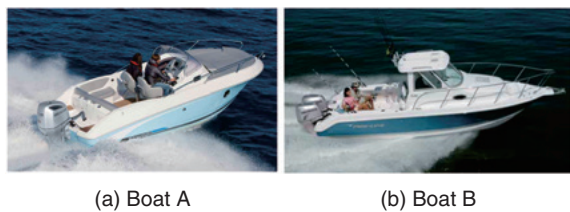


Fig. 8 Small planing boats used in the simulator

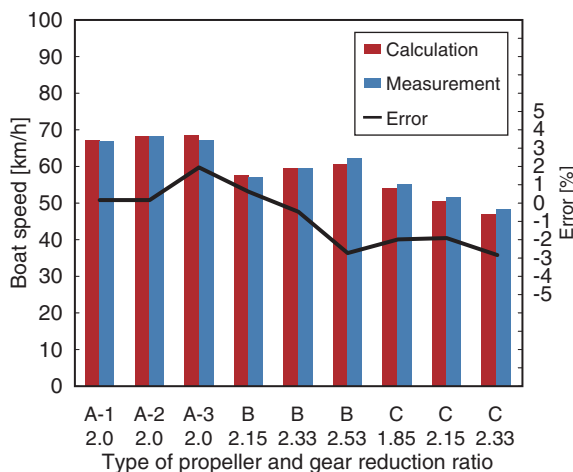


Fig. 9 Predicted maximum boat speeds and full-scale test results

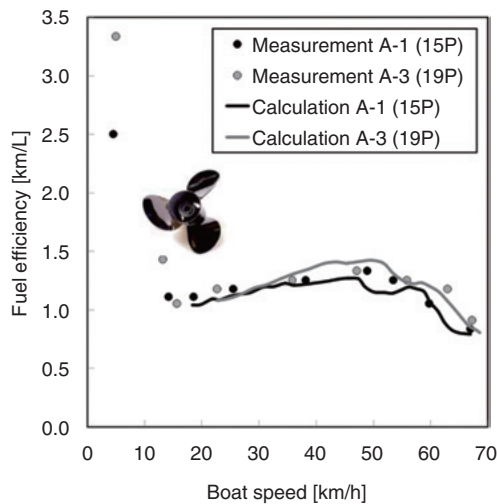


Fig. 10 Fuel consumption simulation compared with full-scale measurement results

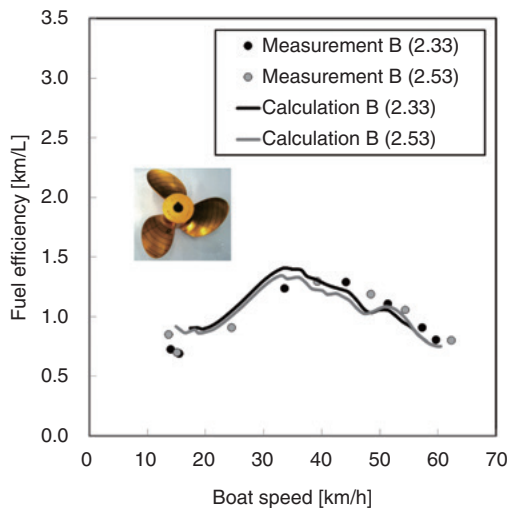


Fig. 11 Fuel consumption simulation compared with full-scale measurement results

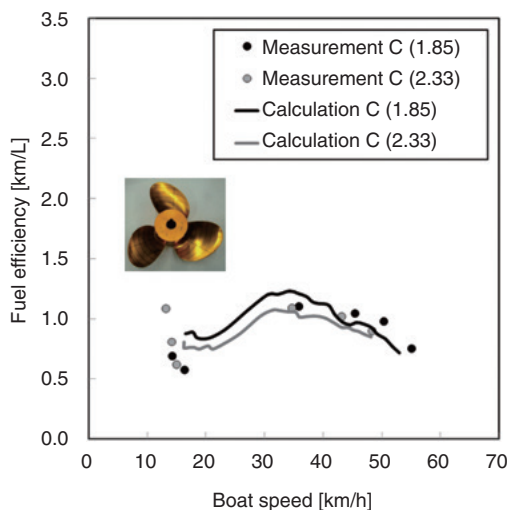


Fig. 12 Fuel consumption simulation compared with full-scale measurement results

and the simulator is able to reproduce the better fuel consumption results for the respective gear reduction ratios.

These results indicate that using the developed simulator, it is possible to numerically predict differences in cruising fuel consumption due not only to differences in propeller specifications, but also due to differences in gear reduction ratios, the framework of the drive system, within a fixed range of accuracy, without conducting boat tests. In addition, the fact that it has been possible to achieve a level of accuracy sufficient for practical use ($\pm 5\%$, the target predictive accuracy) without consideration of the effect of cavitation, as mentioned in Section 4.1.2., indicates that it will be possible to make effective use of the simulator in the initial stage of the outboard motor development process, the framework design stage, in order to develop motors displaying good environmental performance balancing cruising fuel consumption performance and cruising performance. The use of this simulator can be expected to make it possible to offer users not merely propellers possessing good cruising performance (maximum cruising speed and acceleration performance), but also propellers that possess good cruising fuel consumption performance.

5. Conclusion

The research discussed in this paper developed a predictive simulator with the goal of predicting the cruising fuel consumption and cruising performance of outboard motors used for small, high-speed planing boats with a range of error within $\pm 5\%$. Based on previous experience with outboard motor development, the research focused on the fact that propeller thrust performance has a significant effect on cruising fuel consumption, and created databases combining propeller performance predicted using CFD simulation and engine performance data and fuel efficiency maps obtained from tests.

In order to verify the predictive accuracy simulator for the prediction of fuel consumption and cruising performance proposed in this research, prediction results were compared with the results of full-scale cruising tests. These comparisons produced the following findings:

- (1) Comparison of the results of CFD simulation for the thrust coefficient and torque coefficient of propellers used for small, high-speed planing boats with the results of model tests conducted in a towing tank indicated that the simulator displayed a level of predictive accuracy of $\pm 4.55\%$ for the thrust coefficient and $\pm 4.58\%$ for the torque coefficient.
- (2) The simulator was used to predict the cruising performance of a large-horsepower class outboard motor fitted to a small, high-speed planing boat, and the results were compared with the results of cruising tests. The results of this comparison indicated that the simulator achieved a predictive accuracy of $\pm 2.79\%$ (± 1.7 km/h) for maximum cruising speed, and showed an extremely good correlation between simulator and cruising test results.

- (3) The simulator was used to predict the cruising fuel consumption of a large-horsepower class outboard motor fitted to a small, high-speed planing boat. The results showed a good correlation with the results of cruising tests. In particular, comparison of results for cruising fuel consumption when the boat is planing indicated that the simulator achieved an average predictive accuracy of $\pm 4.55\%$.

These results show that the simulator achieves a level of predictive accuracy within a $\pm 5\%$ range of error, the target set for the research. This indicates that the simulator will be able to be used to predict specifications offering good cruising fuel consumption performance and cruising performance, without conducting cruising tests, during the selection of propeller specifications and gear reduction ratios at the initial stage of the development process, which has previously been based on test results and experience. In addition, the results suggest that the simulator will be effective in the development of motors displaying good environmental performance balancing cruising fuel consumption performance and cruising performance.

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Development of Secondary Aluminum Alloy for Diesel Cylinder Head by Mg_2Si Precipitation Strengthening

Kosuke DATEKI*

Takuya TAKAGI*

Ryohei IKUTOMO*

ABSTRACT

A secondary aluminum alloy that achieves an identical level of thermal fatigue strength to a primary aluminum alloy has been developed as a material for use in the manufacture of diesel cylinder heads, parts for which high levels of thermal fatigue strength are demanded. Conventionally, primary aluminum alloys are employed for diesel heads, and their material properties are enhanced mainly by means of the Al_2Cu precipitate. The developed alloy newly applies a precipitation strengthening mechanism that functions mainly via the Mg_2Si precipitate, realizing an equivalent level of thermal fatigue strength and high-temperature strength to a primary aluminum alloy in a secondary aluminum alloy. In addition, the softening property of the developed alloy during the solution treatment process of the heat treatment, realized by means of the material's composition design, has reduced residual stress, a factor affecting the durability and reliability of cylinder heads, by a maximum of 30%.

1. Introduction

Amid increasing calls for the reduction of CO_2 emissions and other burdens on the environment, an initial necessity for automobiles is the realization of enhanced fuel efficiency performance. In order to realize this goal, the downsizing of automotive engines is proceeding for the realization of enhanced performance and increased compactness, but this trend also necessitates the realization of enhanced thermal fatigue strength (elongation) under high-temperature environments in engine structural parts such as the cylinder head. At the same time, social demands for the reduction of energy use in the automotive manufacturing process are becoming stronger. Therefore it is normal to use secondary alloys recycled from scrap as the aluminum alloys employed in automotive engine parts.

The cylinder head is a major engine part, and its main functions are to form part of the combustion chamber, to form the path for intake and exhaust, to form the path for coolant and to support peripheral parts. The majority of cylinder heads are formed by gravity die casting (GDC) or low-pressure die casting (LPDC) of aluminum alloy, and are put to use following the application of heat treatments to help ensure the desired mechanical characteristics⁽¹⁾. Cylinder heads for use in diesel engines (diesel heads) are exposed to high cylinder pressures and high-temperature environments, making thermal fatigue strength and high-

temperature strength particular demands⁽²⁾. Figure 1 shows a representative example of the microstructure of the secondary aluminum alloys used in cylinder heads. Within the microstructure the crystallization of a primary crystal aluminum matrix phase (the “ α phase” below), a eutectic crystal Si phase, and an Al-Fe-Si intermetallic compound known as K6 can be observed. This intermetallic compound is a factor in reducing the thermal fatigue strength of secondary aluminum alloys. To help prevent the crystallization of this intermetallic compound, primary aluminum alloys, to which elements essential to pure aluminum have been added and that display good thermal fatigue strength, are used for diesel heads^{(3), (4)}. To date, no examples of the application of secondary aluminum alloys to a diesel head have been reported. Honda also used

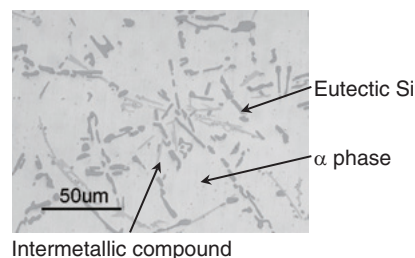


Fig. 1 Microstructures of secondary aluminum alloy for cylinder head

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primary aluminum alloy in its first diesel head for a four-wheeled vehicle. However, the primary aluminum alloy manufacturing process includes a refining process from alumina and its energy load is high. Therefore the first application of secondary aluminum alloy to a diesel head was realized.

As materials technologies employed towards this achievement, the paper will discuss the alloy composition design and the mechanism of improvement of the alloy's mechanical properties, and will consider the achieved effect in improving mechanical properties.

2. Development Goal

As indicated above, decline in thermal fatigue strength compared to primary aluminum alloys is a major issue in applying secondary aluminum alloys in order to reduce manufacturing energy. In addition, the residual stress built up when heat treatments are applied to materials used for casting complex shapes such as cylinder heads, like stress generated by such external factors as cylinder pressure when the engine is in operation, press-fitting of peripheral parts, and the tightening of the bolts, is a significant factor affecting durability and reliability, making the reduction of residual stress another important issue.

Thermal fatigue strength declines when secondary aluminum alloys are employed because cracks originating in intermetallic compounds that crystallize in the materials' microstructures tend to occur in the α phase, resulting in reduced elongation. Residual stress is high because the hardness of the material is high when the heat treatment is applied due to the use of large quantities of strengthening elements to help ensure high-temperature strength. In order to increase elongation and reduce the hardness of the material, the research discussed here decided to reduce the quantity of Cu, the main strengthening element added to the material. However, if the quantity of Cu added is reduced, the material does not achieve sufficient high-temperature strength. In order to enhance high-temperature strength, it was therefore decided to add Mg, an easily available and comparatively low-cost strengthening element, in order to recover material strength via the precipitation of fine Mg_2Si .

The project discussed here set out to develop a secondary aluminum alloy that would satisfy the criteria below, in order to make it possible for the material to realize equivalent material properties to the primary aluminum alloy (the "conventional material") used in Honda mass-production cylinder heads for diesel engines.

- (1) Equivalent level of high-temperature strength to the conventional material
- (2) Equivalent level of thermal fatigue strength to the conventional material
- (3) Equivalent or reduced level of residual stress to the conventional material

3. Material Composition Design

3.1. Concept of Material Composition Design

The material properties desired from the Al-Si-Cu-Mg heat-treated aluminum materials often employed in cylinder heads are normally obtained by means of aging precipitation strengthening of the Cu and Mg in the α phase. The mechanism of aging precipitation strengthening, shown in Fig. 2, is as follows: Atomic vacancies exist in the aluminum matrix during the solution treatment conducted as part of the T6 heat treatment, and during the aging process the strengthening elements Cu and Mg diffuse in these atomic vacancies and form precipitates. In the conventional material, the necessary material properties are obtained mainly by strengthening of the precipitate phase by Cu. Figure 3 shows the results of observation of the α phase in the microstructure of the conventional material using a transmission electron microscope (TEM). The image shows that an Al_2Cu precipitate phase with an average size of 118 nm exists in the α phase. TEM observation of the Mg_2Si phase in the α phase shows a fine Mg_2Si precipitate phase with an average size of 3.4 nm. Based on these studies of the material microstructure, it was considered that refinement of the precipitates in the α phase would represent an effective concept for maintaining high-temperature strength and increasing thermal fatigue strength in secondary aluminum alloys. The project set out to increase the number of precipitates formed by exploiting the properties of Mg, that

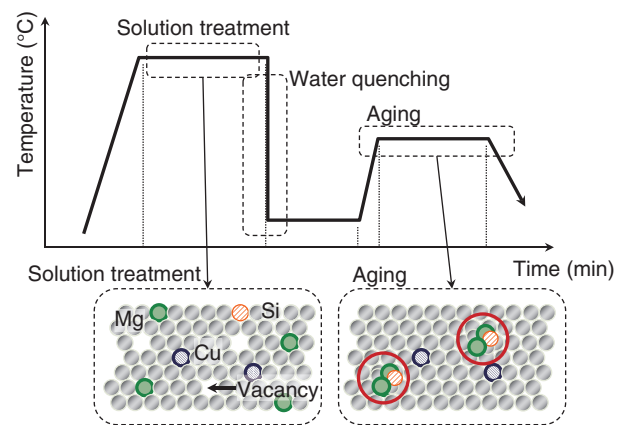


Fig. 2 Schematic illustration of Mg, Si clustering

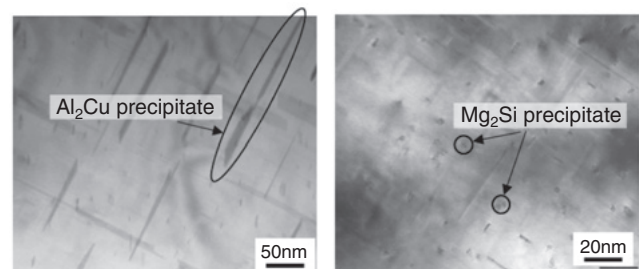


Fig. 3 Transmission electron micrograph of conventional alloy

displays a higher diffusion velocity than Cu in Al, for the refinement of the precipitates. As a result, the composition of the developed alloy used a reduced additive amount of Cu and an increased additive amount of Mg in comparison to the conventional alloy.

In addition, Cu and Mg additive amounts that would prioritize the precipitation of Mg_2Si over Al_2Cu were determined based on time-temperature-transformation curves for Mg_2Si and Al_2Cu precipitates obtained using a commercial multi-component aging precipitation simulation software (JMatPro), as shown in Fig. 4. For the developed material, the concept was to take the determined additive amounts into consideration in order to optimize the form of the precipitation phase by means of the main elements Cu and Mg, realizing the formation of fine Mg_2Si precipitates.

3.2. Test Material

To produce a test material in order to clarify the effect of aluminum alloy composition and the precipitate phase on material properties, 99.7% pure aluminum, an Al-25% master alloy, a magnesium-base master alloy, and impurity elements were added to a commercial AC2B secondary aluminum alloy, and the mixture was melted in an electric-resistance heating furnace using a graphite crucible. The melting temperature was 760°C or below and the holding temperature was $720 \pm 10^{\circ}\text{C}$. In addition, the material was subjected to molten metal treatment for 12 min in Ar gas in order to exclude the effect of hydrogen gas, inclusions and oxides. Following this, the material was poured at atmospheric pressure and room temperature into a JIS boat-shaped mold pre-heated to 200°C and maintained at a fixed temperature, in order to produce ingots for the manufacture of test specimens. After the application of a T6 heat treatment (conversion into a solution at 500°C for 4 hr, water quenching (WQ) at 80°C for 3 min, aging at 200°C for 2 hr), the specimens for material properties tests were obtained from a position 10 mm from the bottom of

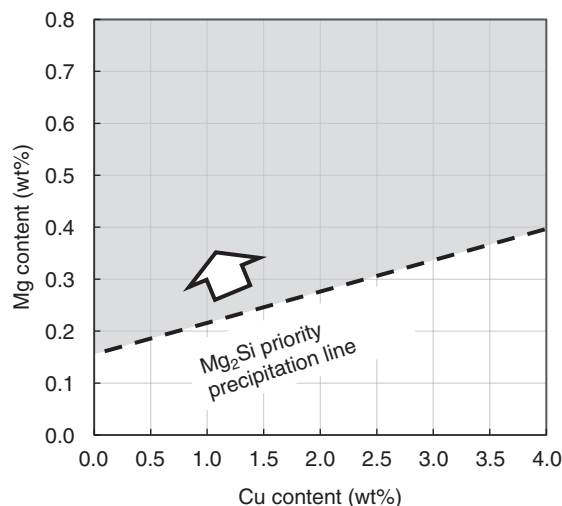


Fig. 4 Relationship of Cu, Mg content for Mg_2Si priority precipitation

the ingots. For the residual stress test, the material was cast in a dedicated mold, and melted and subjected to molten metal and heat treatments like the material used for test specimens for material properties tests. For the study of priority precipitation, a JIS 4 boat-shaped ingot identical to that used for the material properties test specimens was employed, but the material was aged for 10 min. Figure 5 shows the shapes and sizes of the test specimens used in the materials tests and the residual stress test.

3.3. Test Methods

The material properties tested were tensile strength, fatigue strength, thermal fatigue strength, and compression

	Shape and size of specimens (mm)
Tensile strength	
Fatigue strength	
Compression 0.2% proof strength	
Thermal fatigue strength	
Residual stress	

Fig. 5 Shape and size of specimens

proof strength. Average figures were determined at a range of temperatures. For residual stress, the average value was found at room temperature (RT). Table 1 shows the test temperatures for each material test. A tensile test device was used for tensile and compression testing and a test speed of 0.5 mm/min was employed. A rotary bending endurance test device was used for the fatigue strength test, and the test was conducted at a speed of rotation of 3000 rpm. The heat fatigue test used a fatigue testing machine employing an electro-hydraulic servo, and was conducted using sine waves in five-minute cycles under out-of-phase conditions such that heat deformation and mechanical deformation in the test specimens were in opposite phase. Residual stress was measured via the cutting method using a strain gauge. The study of priority precipitation employed a three-dimensional atom probe^{(5), (6)}.

3.4. Setting of Basic Composition

Table 2 shows the composition of the conventional and developed alloys. The effects of Cu and Mg, the main elements, were determined using alloys based on these compositions. The results are discussed below. First, the effect on the materials' thermal fatigue strength was judged in relation to their elongation properties. This was based on the fact that in the case of thermal fatigue strength, the range of strain is high, and in many cases the maximum strain occurs up to the plastic region. There is therefore a correlation between thermal fatigue strength and elongation.

Table 1 Test temperature conditions

	Test temperature (°C)						Soaking time (ks)
	R.T.	50	100	150	200	250	
Tensile strength	○			○	○	○	360
Fatigue strength				○	○	○	360
Compression 0.2% proof strength				○	○	○	360
Thermal fatigue strength						○	
Residual stress	○						

Table 2 Chemical composition of developed alloy (wt%)

	Si	Cu	Mg	Fe	Zn	Mn
Conventional alloy	6.5	3.0	0.2	0.10	0.01	0.01
Developed alloy	8.0	1.7	0.40	0.45	0.25	0.2

Figures 6 and 7 show the effect of the additive amount of Cu and Mg on tensile strength and elongation at room temperature, the temperature at which the tests were conducted. In the case of the effect of Cu on the material properties, tensile strength increased with an increased additive amount at a rate of 13 MPa/wt%, while elongation decreased at a rate of -0.4%/wt%. Turning to the effect

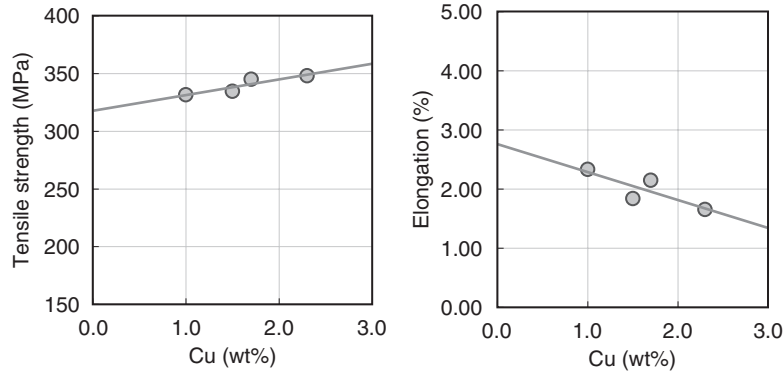


Fig. 6 Relationship between Cu content and tensile strength, elongation

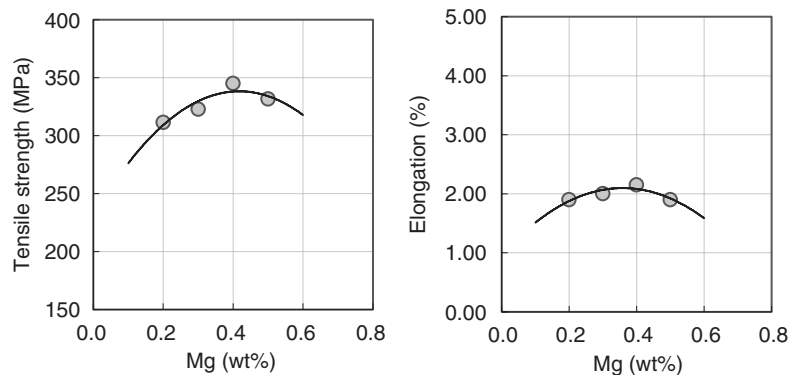


Fig. 7 Relationship between Mg content and tensile strength, elongation

of Mg on material properties, as in the case of Cu tensile strength increased with an increased additive amount, at a rate of 83 MPa/wt%. At the same time, elongation increased, in particular between 0.2wt% and 0.4wt%, at a rate of 1.25%/wt%. Based on these results, reduction of the additive amount of Cu, the main strengthening element employed in the conventional material, in order to increase elongation, and the exploitation of the effect of Mg in increasing elongation by means of its employment as the main strengthening element in place of Cu, would make it possible to increase elongation while maintaining tensile strength.

Next, the additive amounts of Cu and Mg were set in order to achieve an equivalent level of strength to the conventional material. Given that strength reached its peak at an additive amount of 0.40wt% of Mg, as shown in Fig. 7, the additive amount of Mg was set at 0.40wt%. Figure 8 shows the relationship between the additive amounts of Cu and Mg in order to achieve an equivalent level of strength to the conventional material. The additive amount of Cu was set at 1.7wt% in order to achieve this goal. The ability to realize the desired elongation at the additive amounts of Cu and Mg set with a focus on strength was also verified. Figure 9 shows the relationship between Fe content and elongation in the conventional alloy when the amount of Fe is increased. Increasing the Fe content of the material reduces elongation. This is due to the effect of the intermetallic compound discussed above. Figure 10 shows the elongation of the conventional alloy and the developed alloy with Cu and Mg additive amounts set with a focus on strength. The use of a secondary aluminum alloy results in a decline in elongation, but as indicated by the effect of Cu and Mg on elongation as shown in Figs. 6 and 7, decreasing the additive amount of Cu and increasing the additive amount of Mg increases elongation, making it possible to obtain a level of elongation equivalent to that of the conventional material.

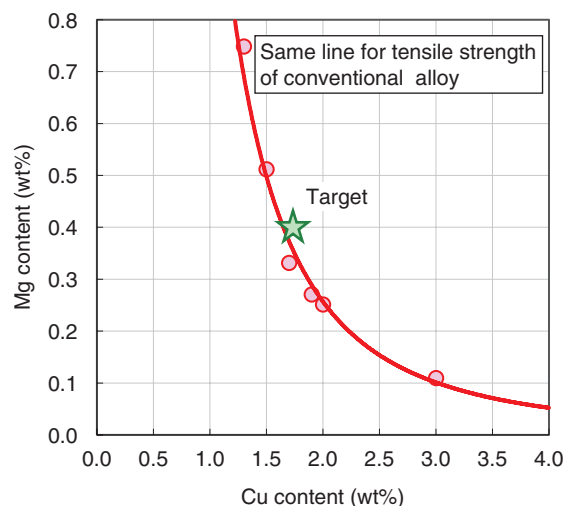


Fig. 8 Relationship between Cu, Mg content and tensile strength of conventional alloy

The composition of the alloy other than the main elements was set as follows. The additive amount of Si was set with consideration of ease of casting, in order to obtain an equivalent melting temperature and fluidity to the conventional alloy. The allowable amounts of the impurities Mn, Zn, Ti, Cr, Ni, Pb, and Sn were set at the same levels as secondary aluminum alloys, and the possibility of obtaining the desired properties at these levels was verified.

3.5. Study of Priority Precipitation of Mg_2Si

In order to verify the form of the priority precipitation of Mg_2Si , a study was conducted in order to determine whether Mg_2Si was precipitated by priority at the set additive amounts of Cu and Mg. A three-dimensional atom probe was used for the observations due to the challenge of conducting observations of the initial stages of formation of precipitates using TEM. The atom probe combined a field ion microscope (FIM) and a time-of-flight mass spectrometer (TOF-MS), making it possible to observe the distribution of atoms.

Figure 11 shows the distribution of Si, Mg and Cu atoms in a test specimen immediately following the commencement of the aging process, as observed using the three-dimensional atom probe. The results show that areas of concentration of Si and Mg atoms exist, and that these

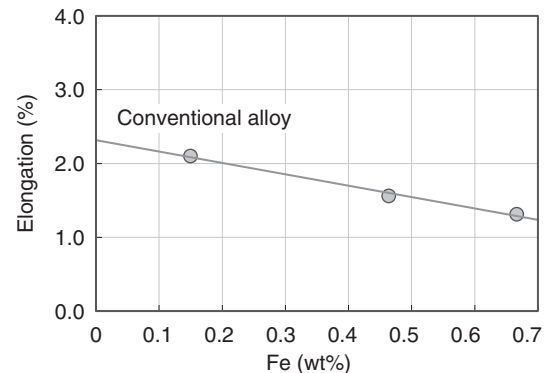


Fig. 9 Relationship between Fe content and elongation

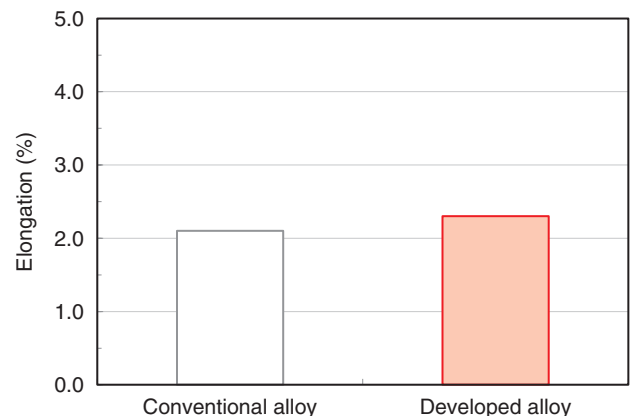


Fig. 10 Effect of elongation improvement by Cu content reduced and Mg content increased

areas correspond. No areas of concentration of Cu atoms were observed. These results indicated that at additive amounts of 1.7wt% of Cu and 0.4wt% of Mg, Mg_2Si is priority precipitated, realizing the form of precipitation projected by the development concept.

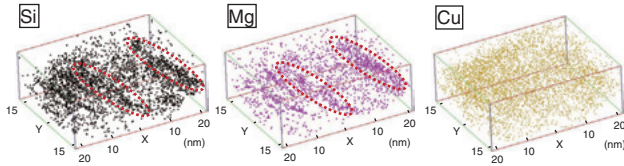


Fig. 11 Distribution of Si, Cu and Mg atoms by three-dimensional atom-probe

4. Material Properties

The developed alloy employing the composition shown in Table 2 in Section 3.4 was verified as possessing the following properties.

4.1. High-temperature Strength

Figures 12 and 13 show the results of tensile strength, fatigue strength, thermal fatigue strength, and compression proof strength. The developed alloy achieves the same levels of high-temperature tensile strength, high-temperature fatigue strength, and compression proof strength at 150°C, 200°C, and 250°C as the conventional alloy. In the case of thermal fatigue strength, the developed material displays equivalent mechanical properties to the conventional alloy in repeated tests conducted at temperatures ranging from 50°C to 250°C.

4.2. Residual Stress in a Simple Test Specimen

Simple test specimens that modeled the distribution of residual stress in a cylinder head were employed in order to compare the difference in residual stress between the alloys. Figure 14 shows the results of residual stress measurements. Residual stress, compared to the conventional alloy, is reduced by 30% in the developed alloy in both the area of occurrence of tensile stress and the area of occurrence of compression stress. Residual stress is produced by thermal strain due to differences in the temperature distribution between sections of the cylinder head, mainly during WQ from the solution treatment conducted as part of T6 heat treatment, and is affected by the hardness of the alloy when undergoing WQ from solution treatment. Figure 15 shows the Rockwell hardness of the alloy directly after WQ. Cu, Mg, and other additive elements that contributed to strengthening the α phase during the solution treatment are in a state of solid solution, but the significant reduction of the additive amount of Cu against the conventional alloy has reduced the hardness of the alloy directly after the solution treatment. This increased softness against the conventional alloy can be considered

to have reduced the material's Young's modulus, resulting in reduced residual stress. As indicated in Section 4.1, the developed alloy also displays an equivalent level of strength to the conventional alloy in the aging treatment that follows the WQ.

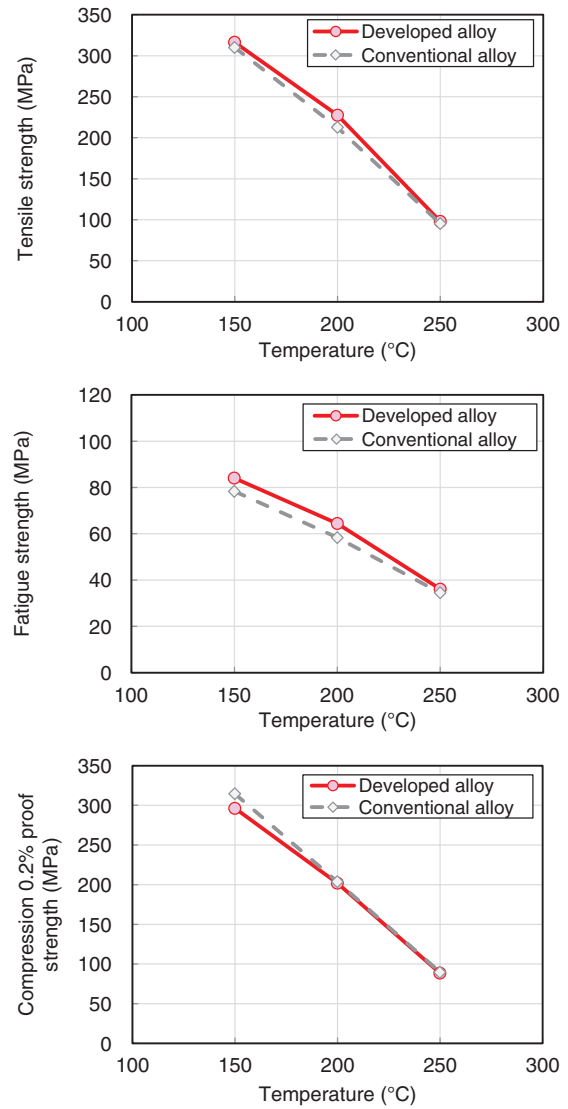


Fig. 12 Tensile strength, fatigue strength, and compression 0.2% proof strength

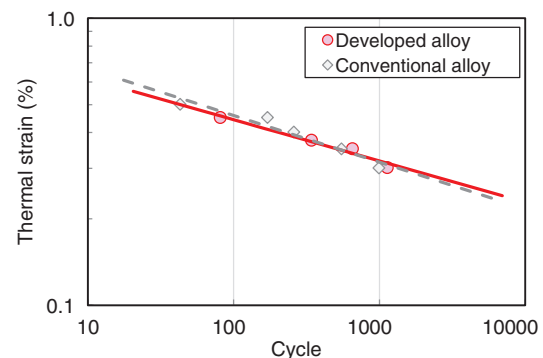


Fig. 13 Thermal fatigue strength

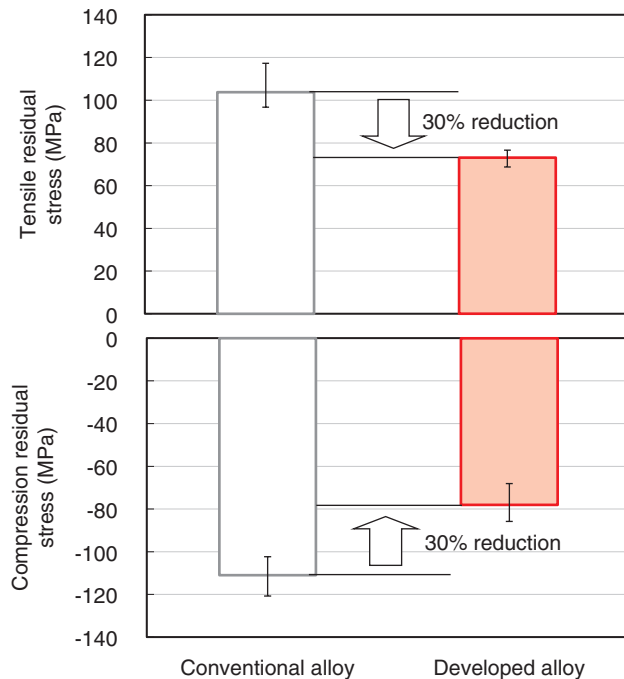


Fig. 14 Effect of developed alloy on residual stress

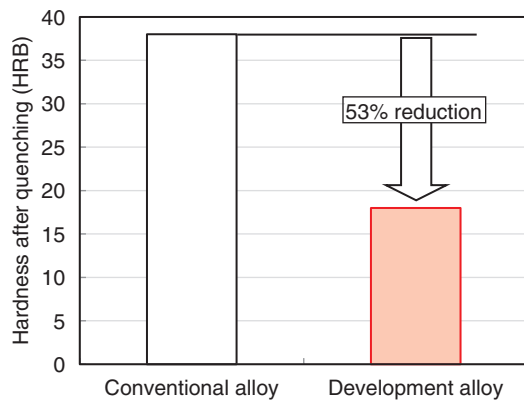


Fig. 15 Effect of developed alloy on Rockwell hardness after quenching process during T6 heat treatment

5. Conclusion

A secondary aluminum alloy has been developed that displays good thermal fatigue strength and high-temperature strength and is able to be employed in the manufacture of diesel heads. The development produced the following results:

- (1) The combination of a Cu additive amount of 1.7wt% and Mg_2Si precipitation strengthening made it possible to realize an equivalent level of high-temperature strength to the conventional alloy.
- (2) Priority precipitation of Mg_2Si made it possible to realize an equivalent level of thermal fatigue strength to the conventional alloy.

- (3) The reduction of the additive amount of Cu increased the softness of the alloy against the conventional alloy during the solution treatment conducted as part of the heat treatment, resulting in a 30% reduction in residual stress following the treatment.

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Editorial Afterword

The Honda R&D Technical Review was founded in 1989 to bear witness to the excellence of Honda's creative engineers. Ten years later, in 1999, the efficacy of the Technical Review as a tool for communication was enhanced by means of twice-yearly publication and the institution of an English edition, carrying us up to the present day. During this period, the pace of social change continued to increase, with the progress of information technology in particular outstripping expectations.

This social background demands that engineers not limit themselves to their own areas of specialization, but rather constantly adopt a broader perspective, striking out into new fields and fields born from a fusion of different elements. It is therefore important for engineers to make a mutual study of each other's fields of expertise. For this reason, the Honda R&D Technical Review, looking ahead to the next 10 years, made the move to electronic publishing on the Internet last spring. We naturally felt a strong sense of resistance to discarding paper media, but we reasoned that doing so would open up new possibilities for the journal.

One of the roles of this publication is to provide support for engineers grappling with new technologies at an academic level. Seeking to create an ethos in the R&D Center that enables all members of the team to experience a sense of enjoyment and pride together in even the laborious day-to-day grind of research, for the future of the Honda R&D Technical Review we intend to fully develop the modes of communication that only the Internet can offer.

For engineers, the writing of papers is an important means of explaining ideas. Remember also that the more intense the effort, the greater the reward. We invite you, then, to use this forum to take up the valuable challenge of writing a research paper.

—Akira Kaneko—

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ISSN 1881-6223

Published by:
Honda Motor Co., Ltd.
2-1-1, Minami-Aoyama, Minato-ku,
Tokyo 107-8556, Japan

Edited by:
Honda R&D Co., Ltd.
Honda R&D Technical Paper Committee

Printed:
Mar. 1, 2013

