

The background is a complex, abstract composition of various colors and textures. It features bold strokes of red, orange, yellow, green, blue, and purple. There are areas of fine, vertical lines, possibly representing a forest or a technical diagram, and other areas with more organic, painterly textures. The overall effect is one of dynamic energy and artistic expression.

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Comment on the Cover

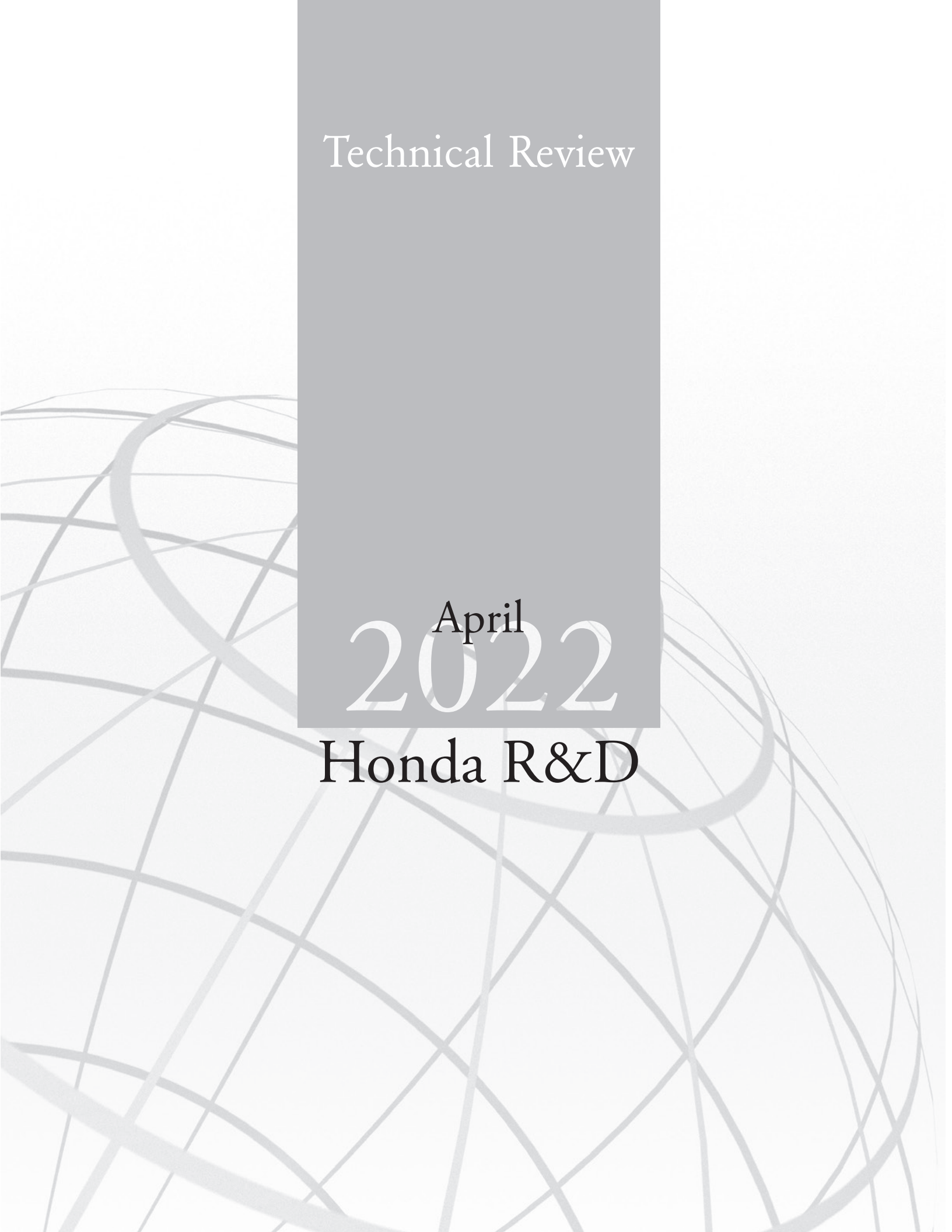
During the process of bringing new ideas and technology into being, a diversity of individualities and technologies are mixed together by serendipity, which feels to me like the repeated occurrence of various states, akin to chemical changes.

I expressed that process with the fortuitously beautiful blurring and overlapping of colors and *matière* that occur when lines, planes, and randomly selected digital effects are combined.

By making this cover an abstract image, I also expressed my own individuality as someone with a background in the study of fine arts, which is unusual for a product designer. In this, the cover also expresses diversity.



Design Center
Minami SAITO

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

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Styling Design of 2020 H'ness CB350

Yasushi TATEISHI¹

ABSTRACT

In the Indian motorcycle market, the popularity of 350 cc class mid-sized motorcycles has been increasing since around 2013. A field survey was conducted, and the target users were determined to be young people who are sensitive to changing trends and who adopt global trends. Based on their comments, we decided to use King Riding Position as the keyword. The design was to be based on the concepts of “the rider should look imposing,” “world-class modernity,” and “massive & shaped.” Based on the above, we were able to complete the styling design of the H'ness CB350 in the following manner: it balances the bike firmly supporting the rider confidently straddling it, which is an important value to the Indian customers, and the modernity of the vehicle that evokes global trends.

1. Introduction

The Indian motorcycle market is dominated by small, practical models in the 100 cc to 125 cc range. The reason for their popularity is that they are reasonably priced and match the daily needs of users. Then, from around 2013, mid-sized motorcycles in the 350 cc class have begun to gain popularity. A previously United Kingdom-based production for this particular medium-sized motorcycle model was moved to India in 1955, and the model continues to be produced today without major changes to the manufacturing method or form from that time. The reason for the increase in sales of this model is the enhancement in ease of handling due to the renewal of the engine in 2009, and the updates made to their dealer network that began around 2013. The updated, upscale appearance of the stores, combined with the model's design graphics following latest trends, has also contributed to a significant increase in sales. In towns full of small motorcycles, riders on their larger motorcycles attracted people's attention. Figure 1 shows the sales trend of the mid-sized motorcycle models.

In this market, customers' expectations for Honda's entry into this category have become clear through local dealers. In order to be accepted in India, our designers started a design study of a new model, the H'ness CB350 (Fig. 2).

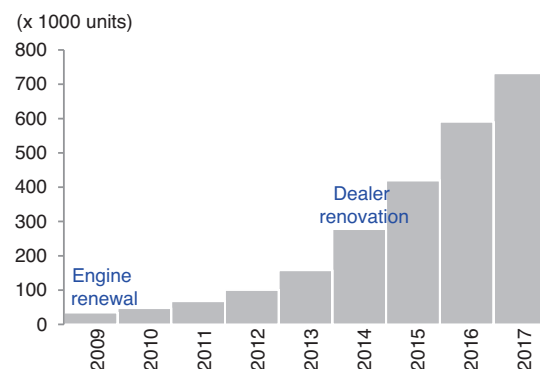


Fig. 1 Changes in sales volume of 350 cc class models



Fig. 2 The H'ness CB350

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2. Survey in India Market

2.1. Target Users

In order to determine the direction of the design, we went to India and visited the users of this class of models and interviewed them (Fig. 3). As a result, it was found that the users of this class of models were divided into two main ways of thinking.

The first group consists of the users who were proud of the existing 350 cc model of the British brand who did not show any interest in other brands because they felt the value of this brand. On the other hand, younger users⁽¹⁾, who are more sensitive to global fashions and changing trends, did not show any particular preference for the British brand. The ratio of the two types of users was about 50:50.

We judged that users with the latter orientation would be willing to accept our products, and designated them as our target users.

2.2. Comments for Existing 350 cc Model by Target Users

The following is a list of comments that we found particularly important.

(1) Purpose

- Practical use as a commuter for daily commuting to work, school, shopping, etc.
- Leisure activities on weekends to enjoy shopping and riding in the suburbs with partner or spouse.
- Touring purposes, from a day trip of about 300 km to a long tour of several days.

(2) What they like about the model

- “King Riding Position” where the rider can ride in an upright, dignified posture (Fig. 4)
- Styling that looks great even when riding at lower speeds
- Large, powerful-looking engine
- Masculine and shapely appearance

- Robust exterior metal parts that have a high quality feel, such as fuel tank and fenders
- Long seat that can seat two comfortably with room to spare

The so-called King Riding Position is the position of the rider. The term King Riding Position refers to the posture of the rider being upright, with their back straight, and chest is out in a confident manner.

2.3. Digging Deeper into User Values

We conducted an interview survey with the target users to identify their values.

They are relatively flexible in their thinking and keep abreast of global trends through the Internet such as SNS. They selected the existing 350 cc model as a motorcycle that can be used for practical purposes as well as for leisure and touring on holidays. Although they were interested in other foreign brand motorcycles sold in the Indian market, many of them are not suitable for practical use, one example being that they are designed for single-seater use in developed countries. Such motorcycles that are only ridden as a hobby on holidays are perceived as models for fun, and do not fit the local values.

Also, King Riding Position was a keyword that stood out in the comments, not only because of the attractiveness of the position itself, but also because it makes the rider look more confident. It seemed as if the respondents had the wish to appeal that they are diligent workers and can ride an expensive motorcycle.

2.4. Market Research Summary

Based on a survey of users of the existing 350 cc class, we decided to target young users who are sensitive to the changing trends and fashions of the world. They wish to ride with an imposing and confident King Riding Position and value practicality. We thought that the motorcycle, which would support such a rider, should be as elegant and imposing as the rider. We decided to work on a design that would balance and embody both Indian user values and global trend values at a high level.



Fig. 3 Scene of user interview



Fig. 4 King Riding Position

3. Styling Design Work

3.1. Styling Design Concept

Based on the comments and key words obtained from the field research, the design concept was defined as “the rider should look imposing” and “world class modernity.”

The first step was to incorporate modern expressions into the styling and detailing, in addition to styling that would suit the King Riding Position.

(1) Interpreting the King Riding Position as a styling direction

- A vehicle body styled with parallel lines that is stable and supports the rider in an upright, confident riding position
- A voluminous appearance that is not overpowered by the rider in King Riding Position
- A split construction seat that is long enough to comfortably accommodate two people
- Air-cooled upright single-cylinder engine with beautifully finished fins
- High-positioned headlight for a strong presence when viewed from front

(2) Structure and detailing that express modernity

- Styling balance with a forward center of gravity, which is the mainstream in modern motorcycles
- Detailing such as surface treatment and integration that makes each part look modern

3.2. Sketching

In order to create a wide range of ideas, sketches were made and discussed by many designers. To verify the balance with the key King Riding Position, we superimposed a photo of the target user in the said riding position on the sketch (Fig. 5).

After reading over the design concept, user comments, and keywords, we narrowed down the sketches to those that met these conditions. The final sketch was one that was simple, yet had a dignified air to it (Fig. 6). In order to confirm the volume and size of the sketch, we proceeded to examine it in a three-dimensional form.

3.3. 3D Study

Based on the sketches we had narrowed down, the overall balance was determined. First, the handles, the seat, and the steps were positioned at the designated points to show the King Riding Position.

The next step was to begin adding the clay, but as we began seeing the overall, three-dimensional image of the vehicle, we felt that the overall proportions were not good (Fig. 7).

The seat was to be made long enough to comfortably seat two people based on our field research, but the clay modeling study revealed that the seat appeared too voluminous and the fuel tank too small (Fig. 8). Since the seat length and riding position were already determined, it was not possible to simply make the fuel tank larger.

Therefore, the angle of the line where the tank meets the seat was adjusted to make the bottom line of the tank appear longer, which also makes the tank appear larger (Fig. 9).

In order to avoid the volume of each part overpowering the appearing of the engine, the size of each part and the mating angles were adjusted. The clay modeling study was thus concluded.



Fig. 5 Photo of target user in King Riding Position overlaid on design sketch



Fig. 6 Final sketch



Fig. 7 First clay model

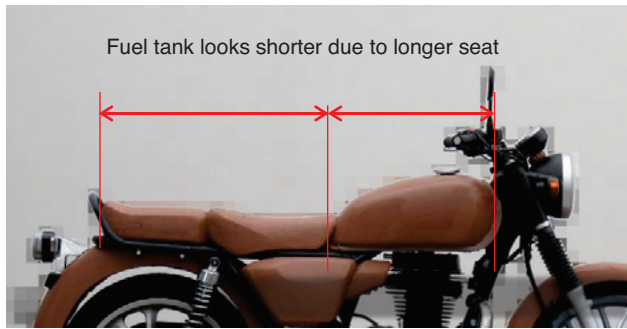


Fig. 8 Impression of double seat and fuel tank

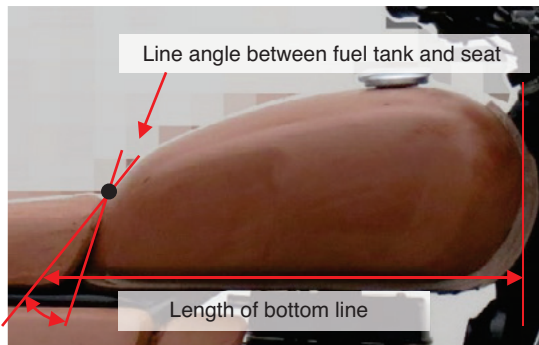


Fig. 9 Appearance adjustment by changing inclination of fuel tank rear end

4. Styling Design Survey in the Local Market

In order to confirm the styling design in the field, we replaced the clay with plastic and created a mock up with paint and other finishes to make it look like an actual motorcycle (Fig. 10). Using this mock up, we conducted survey in the field again to confirm the response of the target users.

4.1. Survey Using Mock Up

We invited users with various attributes from the target user group and held discussions while looking at the mock

ups (Fig. 11).

As a result, it became clear that the styling design presented here would not be accepted by many target users.

The following are the main comments from the users.

- The front view makes it look like a small motorcycle of about 100 cc
- The engine looks too small
- It looks like it is made of plastic and looks fragile
- It looks like it is trying to copy an existing model
- The headlight (Fig. 12) using the latest technology LED light source looks modern

4.2. User Comment Analysis

The team met in front of the mock up used in the survey and conducted a series of discussions to correctly understand and analyze the parts that were denied by users.

- (1) Front view looks like that of a small motorcycle in the 100 cc class

The possible cause of this is the thinness of the tires. Large 19-inch wheels were applied for a powerful look, but the exposed design of the tires made them look thinner. We thought it was necessary to increase the size of the front tire in the width direction.



Fig. 11 Survey state using mock up model



Fig. 10 Mock up model used in survey



Fig. 12 LED headlight

(2) Engine looks small

The size of the engine itself was sufficient, but its presence was overpowered by the surrounding wide frame and voluminous exterior parts. We thought it was necessary to revise the frame layout drastically to reduce the width of the frame and to increase the freedom of expression of the exterior inflection.

(3) Plasticky feel

It is inevitable that the mock up uses plastic even for the parts that are expected to be made with steel for the production model. However, if the shaping itself gives a plasticky feel, using actual steel would still have the same effect. It was therefore determined that the styling design should express the authentic feel of steel, such as the roundness and warmth of the press work.

(4) It looks like a copy of an existing 350 cc model

It is undeniable that following the same engine type and riding position of the existing 350 cc model would make our model to look similar. However, the main reason was that the styling design itself did not express the unique characteristics of this model. The entire team confirmed the need to go back to the basics and make enhancements.

5. Redesigning

Based on the analysis of user comments, we determined that the points for enhancement were extensive and that the design needed to be redone from the sketches. In addition to the design concepts of “the rider should look imposing” and “world-class modernity,” the words “massive & shaped” were added to create the design features. The target users are those who insist on the King Riding Position but do not adhere to the old models. We thought that the key to success would be to go beyond the boundaries of the Indian market and express originality that would be accepted worldwide.

5.1. Proportions

After the sketch redesigning started, the issue that came up again was the length of the seat. In order to achieve a styling balance with a forward center of gravity to express modernity, separate seats for the front and rear were too long. Although this was a request of the local community, we decided to shorten the overall length without changing the effective space for seating by changing to a one-piece 2-up seat. With this change, the proportions took a more modern direction.

5.2. Fuel Tank

For the fuel tank design, the challenge was how to express the sense of volume that Indian users prefer. The fuel tank is a key element of motorcycle design, and we wanted to give this model originality. We decided to try a design that combines hard, modern elements with a sense of volume and overhang in the width direction, which is a method that would give the impression of volume from the side, not just from the top down. The bottom line was raised slightly at the rear of the tank to emphasize the sense of volume at the front of the tank, which the Indian users prefer, resulting in a highly original tank design with a “Massive & Shaped” feel (Figs. 13 and 14).

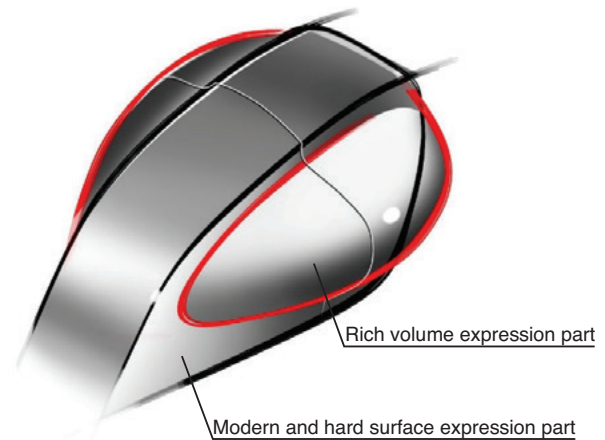


Fig. 13 Fuel tank shape with two different expressions



Fig. 14 Character line of fuel tank

5.3. Horizontal Character Lines Expressing Function

The external shape of the side cover is simple and in line with the frame. A horizontal character line was applied to match the character line of the tank, and the image of the air flow from the air cleaner inside was created. A functional flow connecting the throttle cover, fins of the engine cylinder head, exhaust pipe, and muffler was thus created (Fig. 15). By doing so, we were able to emphasize



Fig. 15 Character line evoking image of gas flow



Fig. 16 Updated final sketch

the expression of a horizontal base with a strong sense of stability.

These resulted in the completion of a new design sketch (Fig. 16). It was an achievement that was made possible by the entire development team working together and going through the process of revising the styling design balance to make an attractive product.

5.4. Three-dimensional Study Using a Clay Model

The shortening of the seat resulted in well-balanced overall proportions. The finished clay model had a well-balanced styling with the rider in the King Riding Position (Fig. 17). The front view, which was commented to look like a small motorcycle from the front, was enhanced by enlarging the tires to give an imposing impression. The stable trapezoidal form supports the rider well, and the parallel alignment of the down tube, pivot pipes, and rear suspensions creates a harmonized balance with the rider's legs and upper body (Fig. 18).

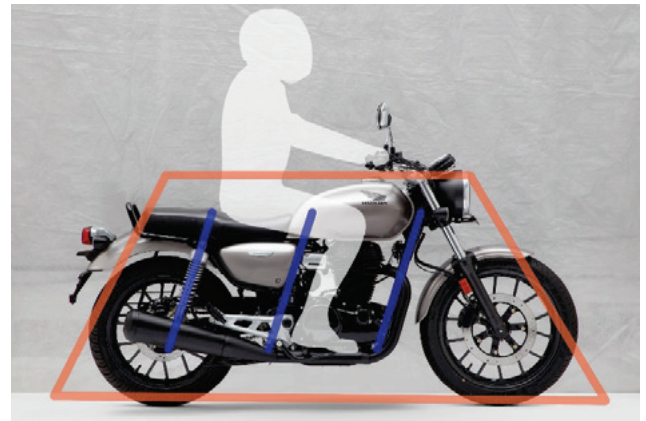


Fig. 18 Stable form and angular harmony between frame and rider



Fig. 17 Finished clay model

6. Styling Design Details

6.1. Engine

For the styling design of the engine, we tried to express the mechanism faithfully and simply. The forward balancer section, one of the most distinctive features of this engine, was emphasized to give the sense of volume in the front and rear. The fins, which are the highlight of the air-cooled engine, were shaped one by one for an overall well-designed look. In addition, the edges of the fins were machined to create a clear contrast of lightness and darkness, which enhances the modern, three-dimensional beauty of the engine. These techniques gave the engine itself a sense of massiveness and made it a symbolic part of this model (Fig. 19).

6.2. Meter Unit

In the styling design of the meter unit, the presence of the unit itself was kept minimal to prioritize an exhilarating riding sensation with a wide field of view for the rider. The display had the indicators arranged in a compact yet easy-to-read manner to reduce the case size. The lens and case were to be made with one-piece molding method to create a simple and fresh form with minimal steps at the mating area, positioning the lens as an important part to express modernity. With the analog speedometer, a gentle convex lens was combined with a flat lens for the digital display, contributing to a fresh appearance by separating the elements with minimalistic expression (Fig. 20).

6.3. Wheels

The wheels chosen for the vehicle have extremely thin spokes and are made of die-cast aluminum to be in line with the latest trends. The individual spokes are made as thinly as possible and in Y-shape, allowing to have lightness in weight and the suppleness to absorb the external force, as

well as strength through the number of spokes. This has resulted in a wheel design that fully supports the body and rider (Fig. 21).

6.4. Seat

The seat has a shapely form, with sufficient width to comfortably support the rider's body weight when riding, and a slim front section in the front to enhance footing when stopped. The slimmer front part also emphasizes the volume of the fuel tank and side covers (Fig. 22).

6.5. Fuel Tank

In order to realize the styling design of the fuel tank which we paid particular attention in the sketching stage, much time was spent to finish it in three dimensions. The character lines at the rear of the tank were also effective in achieving the desired effect, and we were able to express a "massive & shaped" look that is tight while emphasizing volume (Fig. 23).



Fig. 20 Appearance of meter unit



Fig. 19 External design of engine unit



Fig. 21 Exterior design of 14-spoke die-cast aluminum wheel



Fig. 22 Seat and peripheral parts



Fig. 23 Fuel tank

7. Coloring

The coloring is an important element that makes the overall form even more attractive. To make the rider look majestic and assertive, a high-quality, yet modern feel was created. The model comes in two types: the DLX, which is the standard mode, and the variant DLX Pro, which is equipped with a smartphone-linked system. The full color lineup is shown in Fig. 24.

7.1. DLX: Deep and Elegant Dark Colors

A special dark red color (Precious Red Metallic) was developed for this model which, when illuminated by strong light such as sunlight, the metallic shines in the deep areas of the paint, giving a sense of the thickness of the paint layer. In addition, the standard black color has transparent pearl flakes that glitters, emphasizing the depth of the color. Among the dark colors used, the semi-glossy matte green color, which is particularly conscious of global trends, is aimed at users who are more sensitive to trends.

7.2. DLX Pro: Two-tone Paint for a High-end Feel

The two-tone color application was the first specification for the Indian production line, which requires skilled manual work. Two patterns were created for the classic combination of black and silver to attract a wide range of users: one using glossy finish paint, and the other using semi-glossy matte finish paint that is in line with global trends. In addition, a combination of blue and white, the only bright color scheme in the otherwise dark color lineup,



Fig. 24 Full color lineup of the H'ness CB350 (2021 model)

was created to increase the number of choices for users. This color scheme combines a refreshing impression with a sense of luxury.

7.3. Colors, Materials, and Finishes for the Other Areas

By painting the surface of the engine case glossy black and the deeper parts such as the cylinders matte black, a sense of three-dimensionality was emphasized despite the uniformity of the black color. The metallic feel of the steel parts was also accentuated, including the machined edges of the cylinder head air-cooling fins and the chrome-plated engine cover (Fig. 25).

In addition, the front and rear fenders and mufflers were chrome-plated to showcase the beauty and form that are distinctive of pressed steel plates (Fig. 26). The overall finish is black and metallic (with chrome plating, etc.), giving it a dignified and luxurious feel.

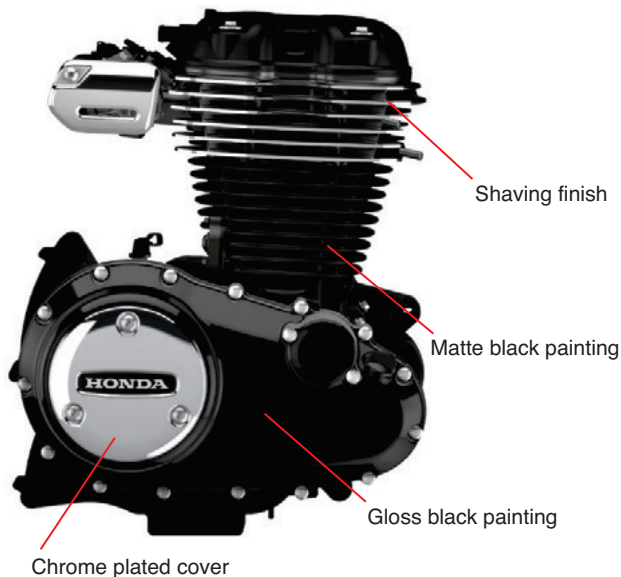


Fig. 25 Harmony of various surface treatments



Fig. 26 Designed shape accentuates raw strength and presence of pressed steel plate

8. Summary

What was consistent throughout the entire styling design process was the keyword King Riding Position, which was elicited from the users. The styling design is the result of understanding the feelings expressed in this phrase, and studying how to make the rider look majestic and elegant, as well as how to make the riders themselves feel confident and happy. It was an important experience for us to hear directly from users in the early stage of development. Through these efforts, we were able to complete the styling design of the H'ness CB350, balancing the value of satisfying Indian users with the modernity of global trends.

We look forward to seeing how the CB350 will develop around the world, and hope to be involved in its further evolution.

Reference

- (1) Yamamoto, K., Wakasa, H., Iguchi, T., Osanai, T., Teshima, S., Kariyasu, T.: Development of 2020 H'ness CB350, Honda R&D Technical Review, Vol. 34, No. 1, p. 10-21

■ Author ■



Yasushi TATEISHI

Development of 2020 H'ness CB350

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ABSTRACT

In India since the mid-2010s, middle and large size motorcycles are becoming popular and being enjoyed as a hobby, causing the market of such motorcycles to grow. This development is targeted at such a motorcycle that fills a wide variety of needs from daily use to long tours. In this newly developed model H'ness CB350, the chassis is designed for an upright riding position and a moderate steering response as well as ensuring stability and easy handling. The engine thumps to characterize strength. Another modern convenience is the connectivity that allows stress-free use of smartphone.

1. Introduction

While the global motorcycle sales dropped in 2020 due to COVID-19, an upward trend is seen in 2021. A single country contributing most greatly to the statistic is India, which is world's largest motorcycle market (Fig. 1). India has liberated international trades in 1990's and achieved remarkable economic growth later on. With the increase of labor force starting sometime around 2010, expectations are high as a hot market. Under such circumstances, a large number of people use motorcycles and scooters for

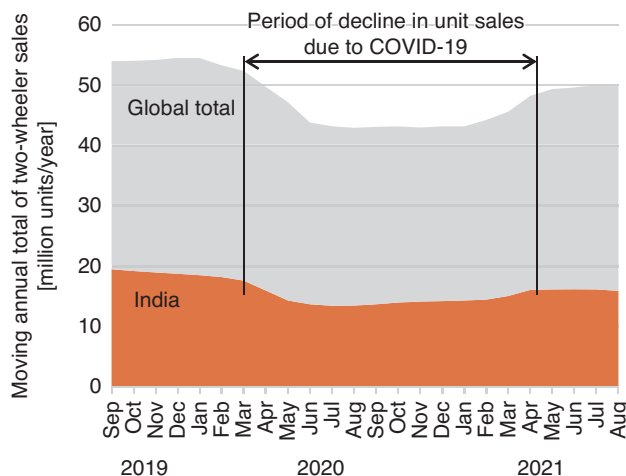


Fig. 1 Motorcycle sales trends (Market research data by Honda Motor)

daily commuting. Recently, in addition to smaller models for daily mobility, middle and large size motorcycles are increasing steadily. Figure 2 shows sales volume of middle and large size motorcycles having an engine displacement 161 cc or larger in India.

In this category, there are several models from a long-established Indian brand. The timeless styling design unchanged for tens of years and the thumping exhaust sounds from the single cylinder engine are well received by users. Such models still remain as a desirable item of consumers whose income is increasing due to the economic growth. The number of users of such motorcycles are increasing year by year because some models are in an affordable price range, creating a new market segment.

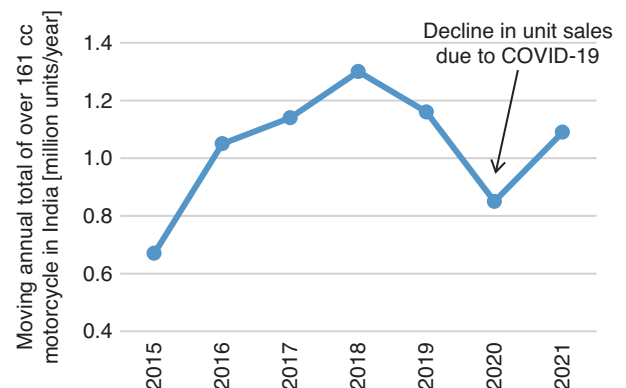


Fig. 2 Sales of over 161 cc motorcycle in India (Market research data by Honda Motor)

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In the meantime, Honda has striven for supplying of smaller motorcycles and scooters, which are convenient means of daily mobility, to the majority customers. The customer voices via dealers for high expectations of middle and large size motorcycles, however, are an indication of a potential desire of Indian customers for such motorcycles. What is expected from Honda at an entry into this segment is an advanced product rather than replication of existing model. To deal with such demands, we started development of H'ness CB350 for representation of high-tech identity of global Honda and beauty of classical motorcycle (Fig. 3).



Fig. 3 The H'ness CB350

2. Development Goals

In India, the core customers of middle and large size motorcycles having an engine displacement of 151 cc or larger are under 30 years old. Figure 4 shows the demography of purchasers in each segment in India. In view of the segment structure, we decided to analyze life style of young users at the start of this development.

First found is that approximately 80% of purchasers of middle and large size motorcycle live in an urban area (Fig. 4). It is evident that such users are influenced by world's trends on real-time basis via SNS information available from the widespread smartphones⁽¹⁾. Thereby changed is the value such customers seek from a motorcycle. Especially noteworthy is that motorcycle tours are starting to be included in weekend entertainment in that movies used to be predominant. Nowadays stories from a motorcycle tour are beginning to appear in SNS. Table 1 shows reasons to buy middle-class cruiser models, indicating a gradually increasing desire for comfort in long rides. Meanwhile the survey on the purpose of use of middle-size models shown in Fig. 5 shows that more than a half of the users frequently use the motorcycle for recreational purposes, which indicates that an increasing number of users enjoy long motorcycle travels. We consider from these findings that the purpose of use of motorcycles, which used to be a mere means of transport, is changing.

Table 1 Reason to buy middle class cruiser model

	2015	2016	2017
Top	Appearance	Appearance	Comfortable for long ride
2nd	Model reputation	Maker reputation	Appearance
3rd	Maker reputation	Exhaust sound	Riding comfort
4th	Riding comfort	Comfortable for long ride	Riding posture
5th	Exhaust sound	Riding comfort	Displacement cc

On the other hand, it has been revealed from the inhouse user interviews that such young users also find value in tradition. They often carry a family member or a friend as a passenger, want an awesome image when riding and prefer a timeless styling design rather than uniqueness. Similarly to users' adhesion to traditional values, the environment surrounding motorcycles has not yet undergone a complete renewal. Although infrastructures such as roads are rapidly developed recently in India to deal with urbanization, there are chronic traffic congestion in urban areas. A mixture of paved and unpaved roads is often seen. In view of the sense of value and environment new and old ones are mixed up, we set a target at new-generation users and filling of their diverse needs.

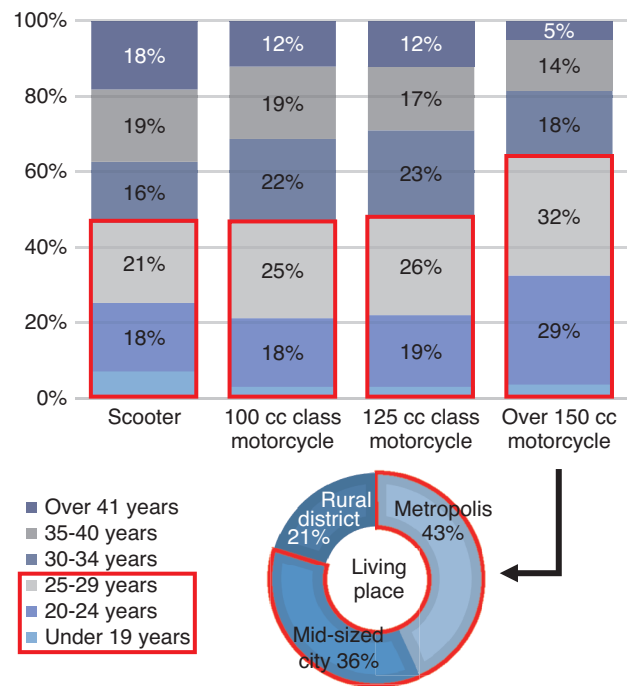


Fig. 4 Age analysis of people who bought new two-wheeler and living places of over 150 cc class motorcycle users (Market research data by Honda R&D India)

Usage of 350 cc class motorcycle

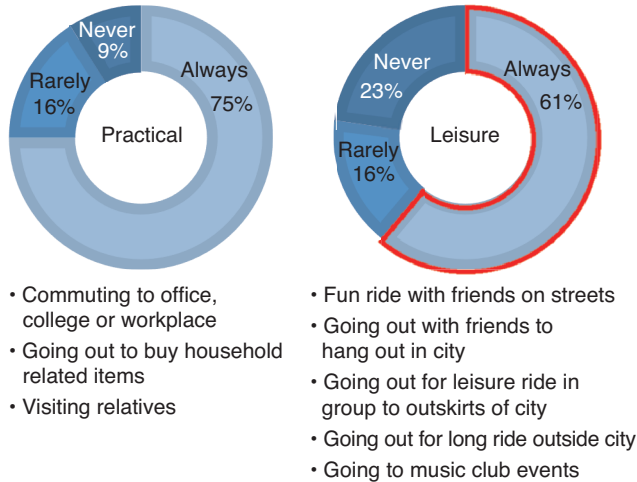


Fig. 5 Survey of usage applications for people riding motorcycles with displacement of 350 cc (Market research data by Honda R&D India)

3. Concept of Product

To satisfy needs of new-generation users having new/old mixed sense of value, we established such a theme “From daily use to long tours – The Honda Basic Roadster.” The following four key requirements have been set for realization of a motorcycle that meets the theme.

- (1) Allow the user to feel an imposing presence
A stately appearance full of characters and a relaxed riding position in that the key player is the rider.
Strong and massive thumping effect that provides the rider with such a sense of satisfaction that he or she is really motorcycling.
- (2) Allow the rider to ride with a peace of mind in long trips or even on unpaved surfaces

Table 2 Specifications of H'ness CB350

	H'ness CB350
Displacement (cm ³)	348.36
Bore x Stroke (mm)	70 x 90.5
Maximum power (kW/rpm)	15.5/5500
Maximum torque (Nm/rpm)	30.0/3000
Curb weight (kg)	181.1
Fuel consumption (km/L)	40.0 (WMTC)
Overall length (mm)	2163
Overall width (mm)	800
Overall height (mm)	1107
Wheel base (mm)	1440
Maximum steering angle (deg)	43
Fuel tank capacity (L)	15
Ground clearance (mm)	176

High stability and moderate steering response.

- (3) Easy to handle in a smaller space or congested roads in urban areas, pleasant to ride
Easy to handle in urban areas and fun-to-ride at lower speeds.

- (4) Provide high-tech images

Up-to-date connectivity to allow stress-free use of smartphone.

Table 2 shows major specifications. The major technical means for realization of the above requirements are sorted into three categories and explained from the next chapter.

4. Chassis

Though different from a region to another, the road conditions vary considerably in India. Figure 6 shows some examples of road conditions in India. In day-to-day



Fig. 6 Daily and leisure environments

use, the motorcycle is often used on unpaved roads and/or in a heavy traffic congestion. The motorcycle has to be easy to handle in a smaller motorcycle parking lot. For some recreational users, the beautiful Leh Ladakh in the Himalayan highland is a dream destination of motorcycle tour. To go there, it is presumable to ride hours on unpaved roads at a high elevation (as high as 5,000 meters). The technical considerations taken to realize a common riding position suitable for a wide variety of rider's build and riding conditions are explained hereafter.

4.1. Upright Riding Position

We aimed at a more upright riding position than an average middle-size naked motorcycle (Fig. 3). Comparison to an average middle-size naked motorcycle is shown along with the triangle composed of the locations of the handle grip, the hip point (H.P.), and the foot peg (Fig. 7). Setting the handlebar width at 616 mm and moving the handle grip position 44 mm to the rear of the motorcycle and 12 mm upward from the average position, rider's forward tilt is reduced. In this upright position, the broad front view provides the rider with a peace of mind. The foot pegs are moved 91 mm to the front of motorcycle and lowered 26 mm from the average position to bring the knee angle close to 90 degrees. With these considerations taken, the target riding position has been realized.

Figure 8 shows comparison of seat thickness from the H.P. As the handlebar position and the foot peg location are fixed, if the allowable seating position is limited to a smaller area on the seat, the riding position will be confined for some riders because rider's build varies. To realize comfortable upright riding position for riders of various builds, the flat seat bottom is employed along with the adequate pad thickness of 60 mm or more in the 120 mm range in the fore-and-aft direction of the motorcycle.

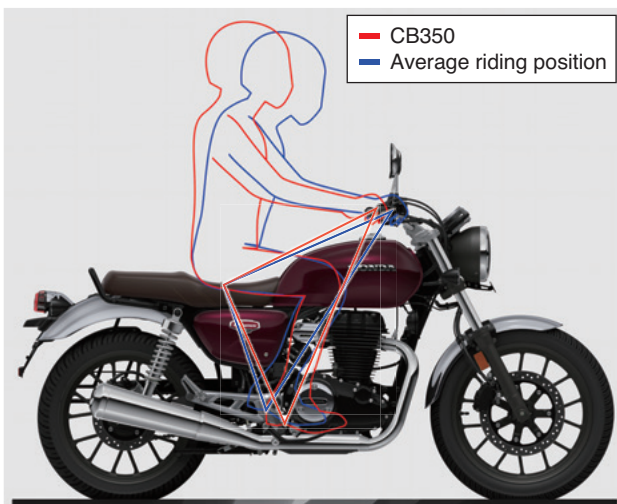


Fig. 7 Comparison of riding positions

Figure 9 is an image picture of the layout. The image lines are composed of the three parallel lines of frame components, i.e., the down tube, the pivot pipe, and the rear shock, the symmetrically-laid front fork line, and the horizontal line that splits the motorcycle into upper and lower parts. Thus, the comfortable upright riding position has been realized.

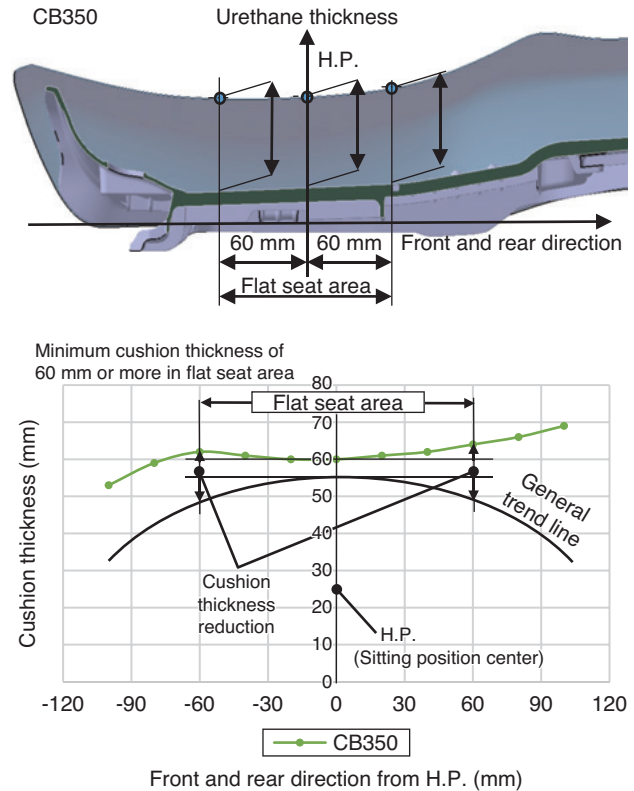


Fig. 8 Comparison of urethane thickness from center of sitting position

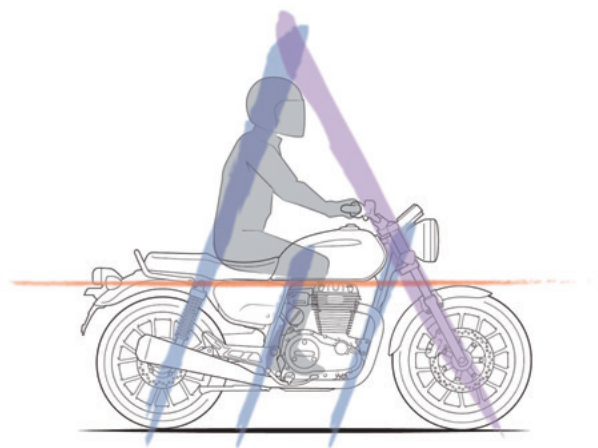


Fig. 9 Sense of unity between rider and vehicle

4.2. Moderate Steering Response

In conventional middle and large size motorcycles, the frame stiffness is maintained high for quick steering response by positively using the engine as a structural member. The mainstream construction is such in that the frame is connected to the cylinder head or the cylinder head cover (hereafter referred to as head hanger) in addition to the engine mount in the crankcase. As this model is aimed at a moderate steering response to deal with various road conditions, we consider that the frame requires flexibility that is unattainable from the abovementioned construction having a high stiffness. Accordingly, we used CAE and analyzed the construction to reduce frame body stiffness, to maintain flexible areas, and to achieve uniform deformation.

Figure 10 shows the frame constructions of this H'sness CB350 and CB400SS along with the relationship between the engine mount points and the flexible area. The red lines connect engine mount points. The CB400SS is an air-cooled, single cylinder model sold in Japan since 2001. This model uses a conventional design having a head hanger. Such a head hanger is non-existent in H'sness CB350, maintaining a long span from the lower end of the head pipe to the hanger mount point to allow the entire frame body to flex. Moreover, with the stiffness reduced by the inner pivot construction in that the frame body is laid inside of the swingarm, the torsional stiffness is lowered by 29% from CB400SS.

Figure 11 shows an example of frame body stiffness data analyzed by CAE when a lateral load is applied to the front and rear tire contact points. In this model, the stress is distributed widely from the lower end of the head pipe to the engine hanger mount. To equalize deformation, the cross-sectional areas of each frame member and the welded stays and other parts are designed to minimize influence on stiffness (Fig. 12). This flexible frame allows moderate convergence of body oscillation when subjected to an external force.

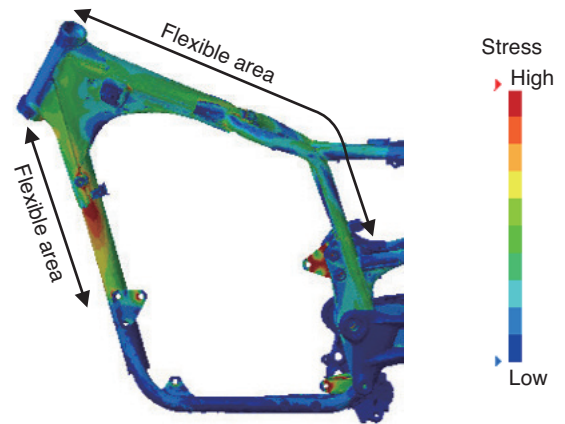


Fig. 11 Frame body analysis result

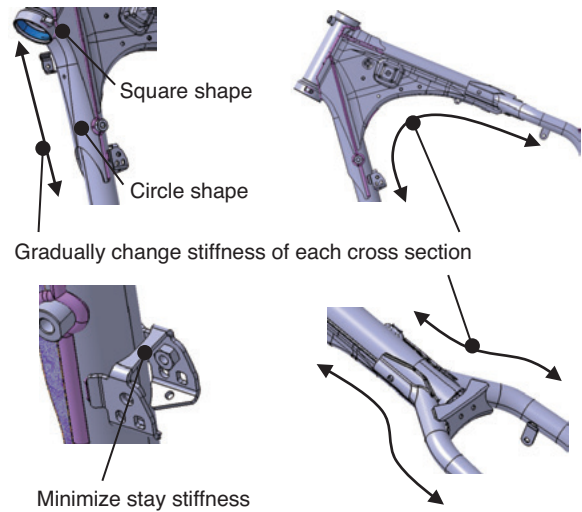


Fig. 12 Each part structure to make deformation uniform

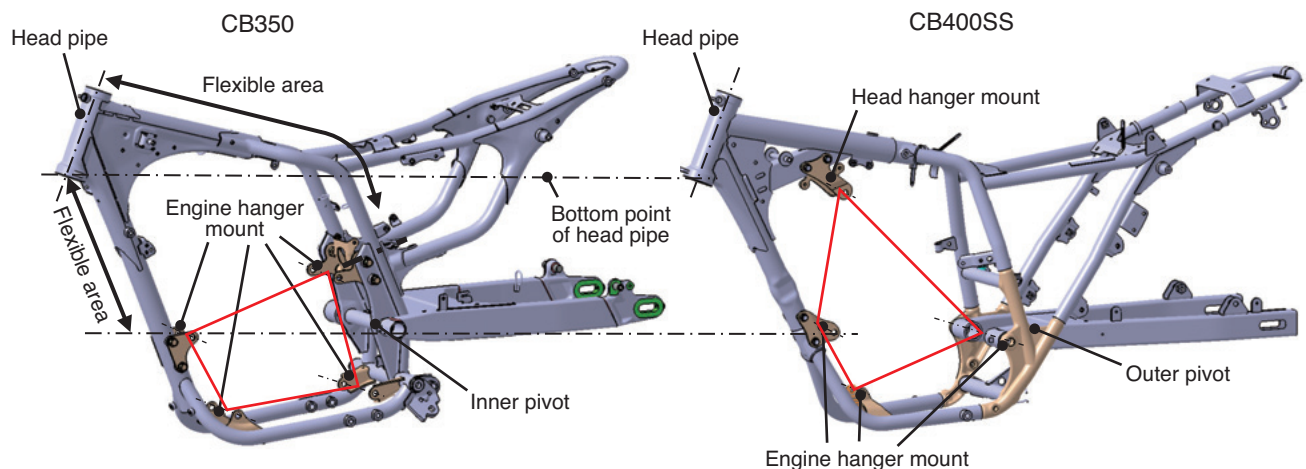


Fig. 10 Flexible areas in frame body

5. Engine

Figure 13 shows the relationship between the front wheel weight bias and the height of center of gravity (CG) of the motorcycle. The moderate steering response is achieved by reducing front wheel weight bias and lowering the CG from the basic sport motorcycles.

4.3. Stability and Easy Handling for Daily Use

In this model, the wheel base is set at 1,440 mm which is longer than other models in the same engine displacement category to enhance stability for relaxed riding. For easy handling in daily use, the maximum turning angle of steering is set large at 43 degrees. To realize the target turning angle, the caster is set at 27.5 degrees, the front fork pitch at 204 mm, and the fork offset at 45 mm. In addition to that, the fuel tank is shaped so as to allow easy handling as well as satisfying styling requirements. Thus realized is the turning radius as smaller as that of a smaller displacement motorcycle for daily commuting. Figure 14 shows the relationship between the wheel base and the turning radius.

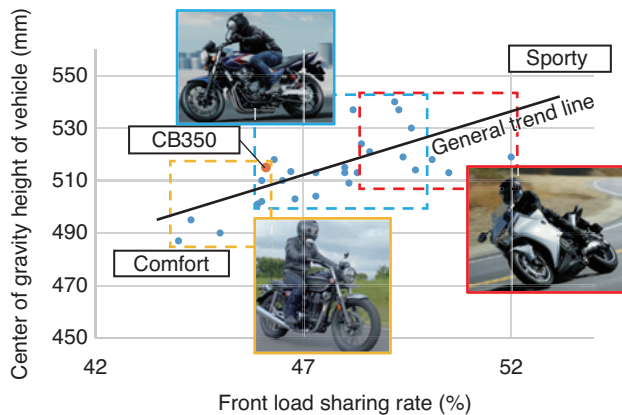


Fig. 13 Relationship between wheel weight bias and center of gravity height

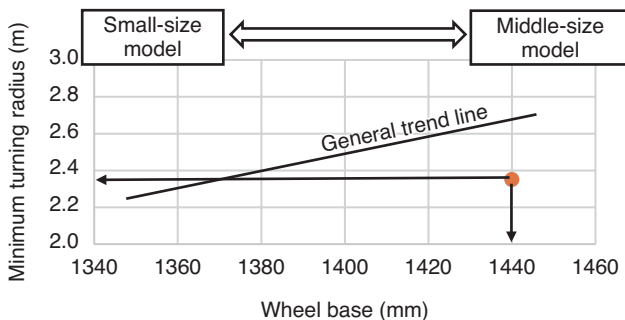


Fig. 14 Relationship between wheel base and minimum turning radius

Thumping sounds and sensation⁽²⁾ are important criteria for Indian customers when choosing a motorcycle for cruising. For in-town convenience, the engine has to be powerful and torquey in lower engine speeds, and the vehicle weight should be lower for easy handling. The configuration we have chosen to meet such demands is a single cylinder, long-stroke 350 cc engine that allows balancing of torque-to-weight ratio and engine weight. Such an engine is suitable for use in lower engine speeds. The rider can easily sense thumping of the engine of such a configuration because of its long combustion intervals. Employing the air-cooling system, the engine features simple appearance with an emphasis on the functional beauty of the cooling fins. In addition to the long-stroke configuration that excels in combustion efficiency, the upper part of the cylinder head is oil-cooled to meet BS6 emission norms enacted from 2020 in India.

5.1. Characteristics of Vibrations That Bring About Thumping Effect

The thumping effect is generally considered as a mixture of sensation from the sounds, especially exhaust sounds, and the vibrations. Both sounds and vibrations are important factors to provide the rider with a big thumping effect (Fig. 15). To provide strong thumping effect, studies were conducted in the frequency characteristics of the sounds and the vibrations. While listening to the exhaust sounds from the earphones at a constant speed of 3000 rpm, the tester was subjected to vibrations of various frequencies from the

	Exhaust sound	No sound
Engine vibration 	1 Good pulse feeling	2 Pushed up feeling (not pulse feeling)
No vibration 	3 Be able to feel pulse feeling, but not enough	

Test conditions

1. Feel the sound and vibration

Test vehicle: Single cylinder engine motorcycle
Test mode: Accelerate to speeds of 0-80 km/h

2. Feel only vibration

Test vehicle: Single cylinder engine motorcycle
Test mode: Accelerate to speeds of 0-80 km/h with earplugs

3. Feel only sound

Test vehicle: EV scooter
Test mode: Accelerate to speeds of 0-80 km/h
+ Play the exhaust sound with earphones

Fig. 15 Combination of exhaust sound and engine vibration for pulse feeling

large exciter machine, and assessed if the vibrations were matched with the exhaust sounds. The excitation was of a sine wave of 12.5 Hz, 25 Hz, 50 Hz, and 100 Hz. Three testers were deployed. An average of absolute assessment in 10 grades was taken. As shown in Fig. 16, it has been determined from the tests that the matching is best when the vibrations are at 25 Hz, which corresponds to the combustion interval, i.e., 0.5 order, of the engine speed. In other words, thumping effects can be enhanced if the vibrations from the 0.5 order excitation are emphasized and the vibrations of other orders are reduced. The vibrations from the 0.5 order excitation are hereafter referred to as 0.5 order vibrations.

5.2. Control of Vibrations to Achieve Maximized Thumping Effect

It was decided to attain the higher thumping effect in

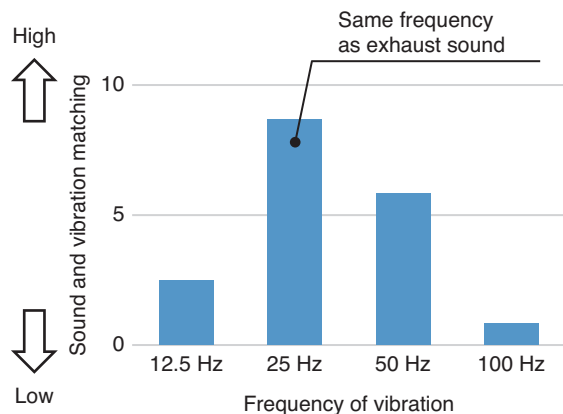
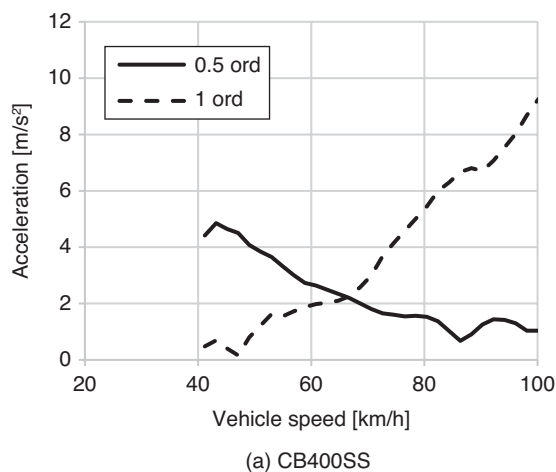


Fig. 16 Matching vibrations of different frequencies with exhaust sound

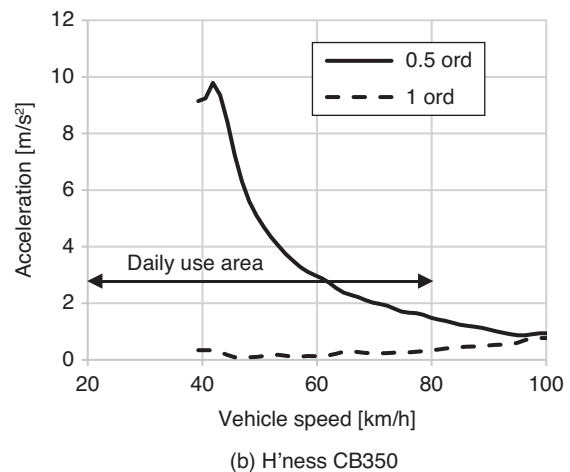
this model by setting the vibration characteristics in such a manner in that the 0.5 order vibrations are larger than those of the first order in the frequently-used speed range (up to 80 km/h in top gear). The vibration characteristics of CB400SS are shown in Fig. 17(a). The acceleration was measured at the frame pivot point in the full-throttle acceleration mode. The 0.5 order vibrations are mainly due to the torque fluctuations occurring predominantly from the combustion force that drives the rear wheel, and lowers along with an increase of engine speed. On the other hand, the first-order vibrations occur from reciprocating motions of the piston, and increase along with an increase of engine speed.

To increase 0.5 order vibrations, measures are taken to reduce loss of energy. First of all, the offset cylinder design is employed to reduce piston friction. Then, the sealed crankcase construction is applied to control oil agitation losses and air resistance from movements of the piston, crankshaft, and other parts. Figure 18 compares pressures at various places of this model to the 300 cc, air-cooled single cylinder engine without the sealed crankcase construction. It is evident that the crankcase pressure is lower in this model than the other model without the sealed construction.

The clutch damper and the rear hub damper are provided in most motorcycles to absorb shocks occurring from engine torque fluctuations when the power is transmitted to the rear wheel. While an excessively stiff damper may cause the motorcycle to become jerky, a soft damper reduces such a sensation as being kicked forward by the rear wheel. When assessing such a sensation as being kicked forward by the rear wheel, a number of factors have to be taken into account such as the combustion conditions, the inertia moment of power transmission parts, and the tire specifications. Conducting the multibody dynamic analysis⁽³⁾, a combination of damper characteristics for



(a) CB400SS



(b) H'ness CB350

Fig. 17 Frame pivot vibration acceleration of CB400SS and H'ness when throttle is fully open in top gear position

the highest stiffness without causing a jerky-ness was determined to control 0.5 order vibrations.

A new counter balancing system is employed to suppress the first order excitation, which is usually quelled by a single-shaft counter balancer like the one used in the CB400SS. In the single-shaft balancer, a first-order couple occurs because the balancer shaft and the crankshaft are on separate axes. To reduce the first order excitation from the couple, the two-shaft balancer is employed. Figure 19 is an image picture of the counter balancer in this model. An independent rear balancer shaft is not provided because the rear balancer is located on the transmission main shaft behind the clutch. While the rear balancer location is dependent on the distance from the crankshaft to the main shaft, there is more freedom in the front balancer shaft location. It is located as close to the crankshaft as possible to realize a compact and simple layout of the balancer system. Although the front and the rear balancer shafts

are located at a different distance from the crankshaft, the overall balance is maintained by unevenly setting the front and rear balancer weights and the location of the weights in the right and left direction of the engine. Thus, the first order excitation and the couple are controlled to reduce vibrations occurring at a frequency different from that of exhaust sounds.

Figure 17(b) shows the vibration characteristics of this model measured at the same measurement conditions as those of CB400SS. Compared to the CB400SS, the 0.5 order vibrations are higher in lower speeds due to the reduced pumping loss and the optimum setting of damper stiffness. The vibrations from the first order excitation are lower in the entire engine speeds due to the employment of two-shaft balancer. In the frequently-used engine speed zone, the 0.5 order vibrations are higher than the first order vibrations.

5.3. Sound Management to Achieve Higher Thumping Effect

To achieve strong and massive thumping effect at the highest level, the quality of exhaust sound is crucial. The exhaust sound can be assessed in a quantifiable manner using the deepness and the pulsive sensation as the indicators⁽⁴⁾. The deepness indicates such characteristics as thickness, lowness, and heaviness. The pulsive sensation is an index for the intermittence (discontinuity) such as the crispness. In other words, the larger the deepness and the pulsive sensation, the more strong and massive thumping effect is attainable.

To enhance deepness, it is necessary to boost sounds in lower frequencies under 200 Hz. Accordingly, the muffling of lower frequency sounds is reduced by the tail pipe having a larger diameter than that of the models in the same engine displacement category. To boost pulsive sensation, it is necessary to increase sound pressure amplitude deviation in

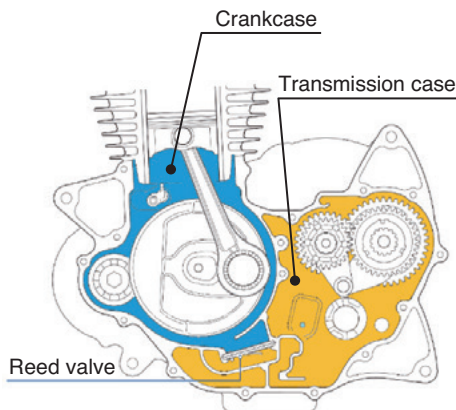
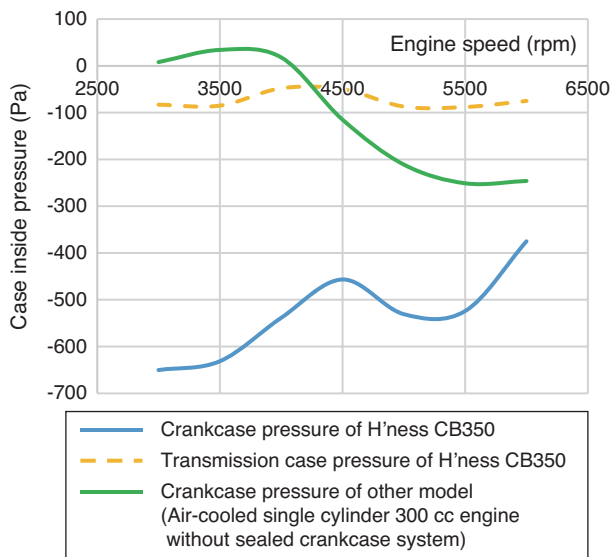


Fig. 18 Case inside pressure of H'ness CB350 (crankcase and transmission case) and other model (air-cooled single cylinder 300 cc engine)

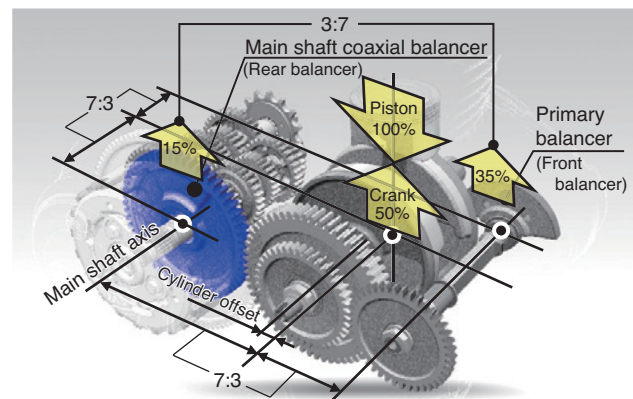


Fig. 19 General layout of single and dual axis balancers of H'ness CB350 engine for first order vibration

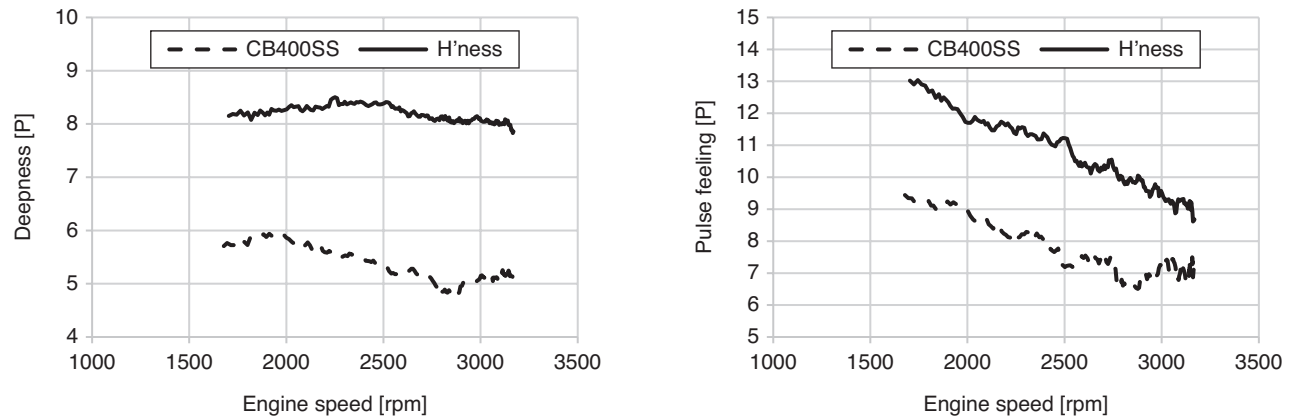


Fig. 20 Comparison of H'ness CB350 and CB400SS deepness and pulse feeling when throttle is 12.5% open

combustion cycle. As an increase of number of expansion chambers narrows the sound pressure amplitude, the single chamber muffler construction is employed to decrease excessive suppression of exhaust pressure pulsation and produce presence-full pulsive sensation. Moreover, the lower-revving, long-stroke, single cylinder engine configuration is employed to widen combustion interval for amplification of pulsive sensation and to achieve such engine characteristics that are strong yet torque-full in lower engine speeds. Figure 20 compares sound characteristics of this model to CB400SS measured during moderate acceleration at the throttle opening of 12.5%. It is evident that both the deepness and the pulsive sensation are greatly stronger in this model than the CB400SS. To meet the noise emission regulations, the noises other than the exhaust sound are suppressed as much noise as possible. The noise radiation from the engine, which affects most largely to the noise emission during running, is reduced by selecting appropriate shapes of engine cooling fins and optimum designing of engine cover stiffness using CAE.

Thus, the strong and massive thumping effect has been realized by the tuning of exhaust sound and the enhancement of the vibrations that best-match the exhaust sound.

6. Up-to-date Connectivity

With the number of users help predicted to exceed 60% of the population in India in 2022, the smartphone is an important item for everyday life⁽¹⁾. You may recognize quite a few riders using the smartphone when stopping at the traffic light or parking on the street in a city like New Delhi (Fig. 21). As the rider usually wears helmet, gloves, and other protective gears, there are many steps involved in just making a phone call. Along with the spread of smartphones in India where the motorcycle population exceeds 150 million units, the desire to use smartphone when riding has been growing. Accordingly, we have decided to

adapt up-to-date connectivity for interaction between the motorcycle and the smartphone.

6.1. System Concept

General-purpose headsets and related parts are already introduced in the market for control of smartphone when riding. Such products, however, require the rider to move a hand off the handlebar to take a phone call or to stand by for a certain period of time. To deal with such issues, we decided to develop a system that allows communication with the smartphone mainly by means of voice feedback. By using minimized hand switches together, the system allows stress-free operations when riding.

6.2. Basic Functions

A variety of smartphone functions such as text message, search, map, and navigation are generally used frequently⁽⁵⁾.



Fig. 21 Using smartphone while on motorcycle

After studying various options, we selected the voice navigation, music player, phone call, and message exchange as the basic functions that help riders when riding. The system is devised to give a priority to the phone call and message so that those functions interrupt other functions being used. Figure 22 shows detailed explanations of the functions.

6.3. System Structure and Hierarchy of Operations

Figure 23 shows overview of the system. The

smartphone and the motorcycle are connected via **Bluetooth** wireless. The rider can control the smartphone by operating the switch on the left handlebar. The headset and the smartphone are also connected via Bluetooth wireless, allowing the user to upload voice message and listen to music, phone call, and messages being read out loud.

The system is structured so as to permit reaching to the required command by repetition of selection from the simple menu of each hierarchy level. Though the hierarchy structure is different from a function to another, the

	Detail of function	Remark
 Navigation	Search places and set new destination	Can search destination by voice Navigate by using Google maps
	Set destination from favorites	Can register 5 favorite places (Home, work place, friend's house, and so on)
	Cancel navigation	-
 Call	Receive call	-
	Start call to your contacts	Can search contact by voice
	Start call to your favorites	Can register 5 favorite contacts
 Message	Read received SMS, WhatsApp, LINE, Facebook Messenger and reply by voice	Can input message by voice HSVCS changes voice message into text Can reply by using prepared quick messages as well (OK, I'm driving, etcetera)
	Start to send SMS to your contacts or favorites	Can search contact by voice Can register 5 favorite contacts HSVCS changes voice message into text
 Music	Play music (Google Play Music, Spotify, Amazon music, YouTube Music, and so on)	The availability of 3rd party apps depend on their updates
	Skip music (Next track)	-
	Adjust volume	Can manage volume of smartphone sound to 0, 20, 40, 60, 80, and 100%

Fig. 22 Basic function list

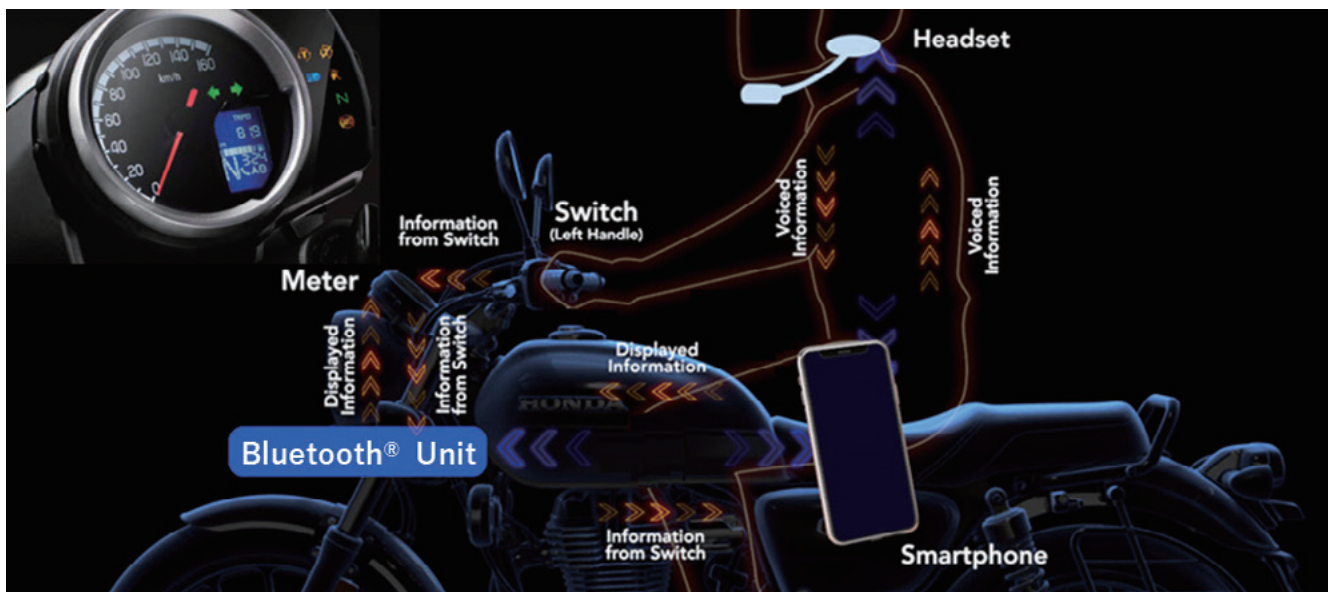


Fig. 23 Schematic diagram of system

number of levels is limited to 4 as Fig. 24 shows because when operating only by voice feedback, you become puzzled where in the structure you are in if the hierarchy is composed of a large number of levels. To determine the hierarchy structure, we conducted actual running tests using the test riders of 20-50 years of age who had undergone an advance assessment by the inhouse bioethics committee. With 3-to-5 levels of selection options provided for each function, we tested which level of the hierarchy is recognizable only by the voice feedback. Accordingly the levels are set at 4 because the operating hierarchy level becomes hard to recognize when there are 5 levels or more.

6.4. Common Switches for Control of Smartphone

To allow control of smartphone without taking a hand off the handlebar, we choose to commonly use the multi-function meter control switch being widely used in recent

motorcycles. Figure 25 shows the controller on the handlebar.

Pressing and holding the right button of the four buttons laid in a cross shape turn the mode to control of multi-function meter. Pressing and holding the left button turn the mode to control of smartphone. Thus, the system permits connected operation of the smartphone without increasing number of switches and compromising user-friendliness.

Thus, the up-to-date connected system has been realized by the coupling of the motorcycle and the smartphone so that the rider can use the smartphone without stress when riding.

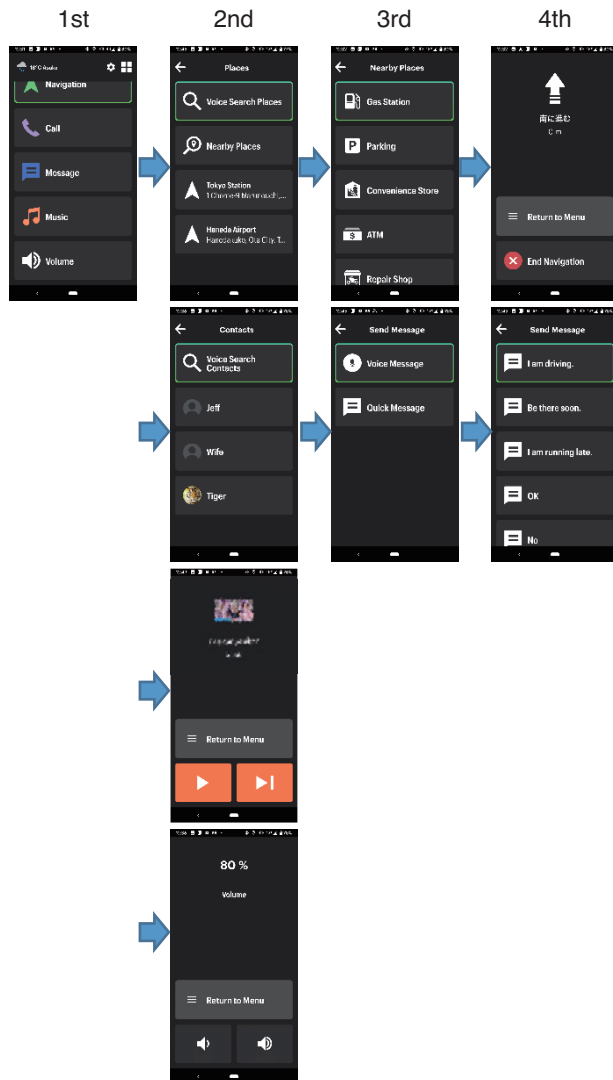


Fig. 24 Application hierarchy tree

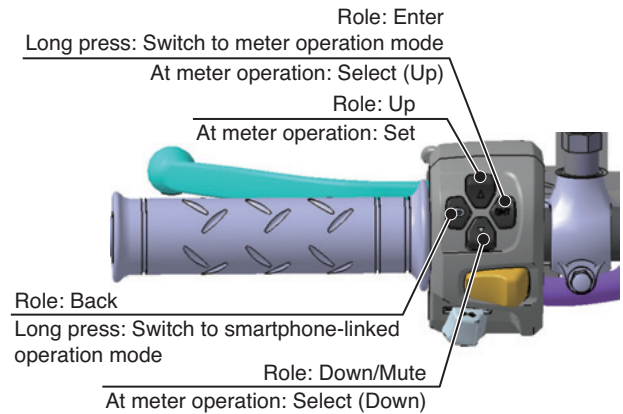


Fig. 25 Handle controller

7. Conclusion

In this development, the target was set at realization of such a motorcycle that satisfies a wide variety of user needs such as daily use and recreational activities. The chassis is designed on the basis of CAE analysis to achieve the relaxed upright riding position, the moderate steering response, and the stability while keeping the motorcycle easy to handle in city areas. To attain strong and massive thumping effect, the exhaust sound is tuned and the matching engine vibrations of the same frequency as the exhaust sound are emphasized. For convenience, the connectivity is added for easy use of a smartphone. The high-tech identity of global Honda and the beauty of classical motorcycle are thus realized in H'ness CB350.

With a theme "From daily use to long tours – The Honda Basic Roadster," this model matches global users' sense of value. Derivative models are imported into Japan. We hope a large number of customers enjoy this motorcycle.

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Development of Honda Techmatic System for 2020 Model Year FIT

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ABSTRACT

The Honda Techmatic System, a hand-operated driving assistance device that supports the independence of users with physical disabilities, has been developed as an official Honda product for the 2020 model year FIT.

Starting with the 2020 model, a Qi wireless charger for smartphones has been installed in the area where the existing Techmatic control lever is installed. By providing Qi functionality in a Techmatic system vehicle, Honda aimed to reduce the amount of effort required by drivers. Honda was able to continue use of the 2016 model year FIT control lever mechanism by moving it further forward in the vehicle. This compensated for the loss of horizontal space due to the Qi charger and for the installation point loss directly below the control position. In this way, Honda made the control lever mechanism more compact without impacting usability.

1. Introduction

Barrier-free thinking has evolved over time into the idea of diversity. In the field of mobility, work is progressing on building environments in which everyone can get around more easily⁽¹⁾. However, in order for people with physical disabilities to access public transportation, they still often need the help of others, such as for setting up ramps when entering/exiting vehicles, or helping to push a wheelchair up a slope. Therefore, a vehicle that allows people with disabilities to get around freely without the help of others is a growing necessity.

Typically, the drive controls of a vehicle are arranged on the assumption of full use of all limbs, i.e., arms and legs. The accelerator and brake pedals are operated with the right foot and the steering wheel requires both hands. In addition, there are various switches such as the light switches and the gear-shifting controls. Many people who have a limb disability can thus find it challenging to drive a vehicle⁽²⁾. Driving assistance devices developed to address this issue are mechanisms that are added to a vehicle. These vary depending on the specific limb disability of the driver so as to enable that person to drive the vehicle⁽³⁾.

This paper will provide an overview of the Honda Techmatic hand-operated driving assistance device, and present a development case in which the assist device was installed together with a wireless recharger (a Qi charger) for smartphones in the 2020 model year FIT.

2. Driving Assistance Devices

In Japan, a 1960 revision to the Road Traffic Law made it possible for people with physical disabilities to obtain a license. In the 1980s, devices such as those shown in Fig. 1⁽⁴⁾ were studied. In 2019, approximately 190,000 people had a conditional driver's license that limited them to "vehicles for people with physical disabilities"⁽⁵⁾.

In terms of sales, the Japanese market consists of 4,000 units/year. Two specialized retrofit manufacturers account for 80% of these sales. A number of automobile manufacturers have participated in the market in the past, but as far as the authors can determine, this number has fallen.

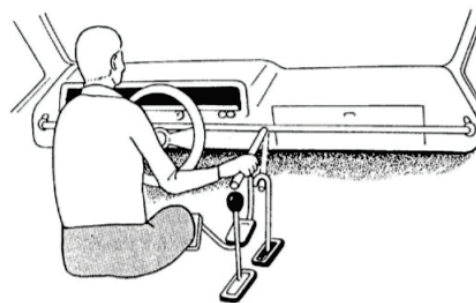


Fig. 1 Driving assistance device developed in 1980s

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By offering an official Honda product, Honda provides customers with the convenience of a vehicle and accessories that can be delivered together, rather than having to modify a vehicle post-purchase, and thus helps provide people with physical disabilities more access and opportunities to engage in activities without constraints.

2.1. History of Driving Assistance Devices

The history of Honda driving assistance devices began in 1968 when Tech Production began developing and installing hand-operated driving assistance devices under the “Honda Techmatic System” trademark⁽⁶⁾. In 1982, Honda released the Honda Franz System⁽⁸⁾, Japan’s first foot-operated driving assistance device. This was in conjunction with a revision of the Road Traffic Law for people with upper limb disabilities, who until that time could not receive a driving license⁽⁷⁾.

2.2. Types of Driving Assistance Devices

As shown in Fig. 2, various driving assistance devices are available, depending on the disability of the driver. These are explained in detail below.

2.2.1. Hand-operated driving assistance devices

These are devices that enable people with disabilities in their lower limbs to drive using only their hands. They are designed so that the accelerator and brake can be mechanically operated by hand. For example, Fig. 3 shows an example of a lever type, in which the accelerator is

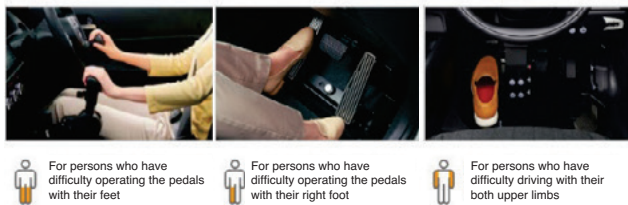


Fig. 2 Types of driving assistance devices

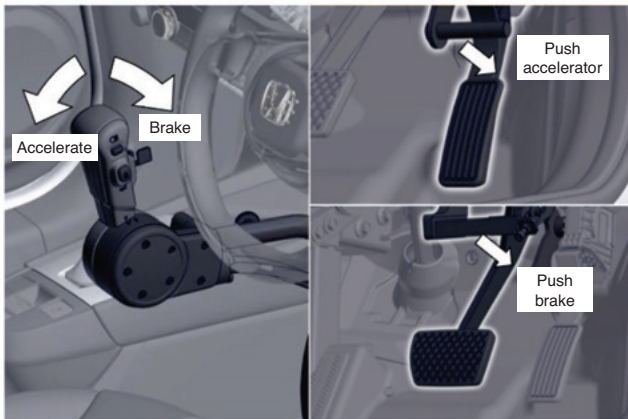


Fig. 3 System of driving assistance device for hand use only

applied when the lever is pulled, and the brakes are applied when the lever is pushed.

Honda offers a D-type lineup as shown in Fig. 4, which is a combination of devices in which the left hand operates the accelerator and brake pedals as shown in Fig. 3, and the right hand turns the steering wheel. The hand-operated driving assistance devices have a roughly 50% share of the driving assistance device market in Japan, and various lever-type and steering-wheel-type devices are offered.

2.2.2. Left-footed accelerator

This is a device that enables people with a disabled right leg to operate the accelerator and brake with their left foot. As shown in Fig. 5, an accelerator pedal is installed on the left-foot side. When this is depressed, the connecting mechanism causes the vehicle accelerator to also be depressed. To help prevent the right foot from stepping on the vehicle accelerator, the right-foot pedal is replaced with a footrest.

This is Honda’s B-type lineup, as shown in Fig. 6. Sales of this product have been increasing in recent years, and it is believed that market demand is growing.



Fig. 4 Honda Techmatic D-type

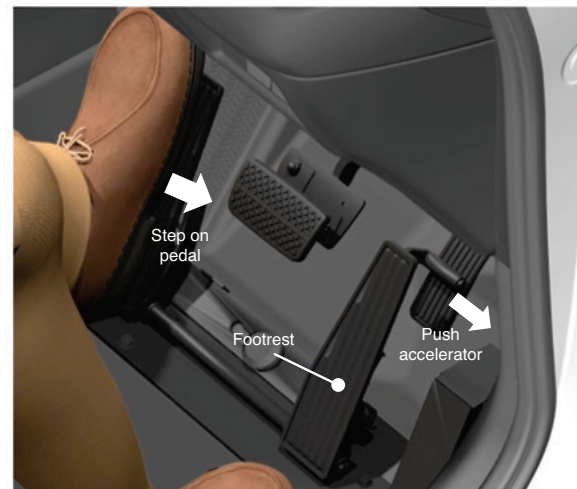


Fig. 5 System of driving assistance device for left foot

2.2.3. Foot-operated driving assistance devices

These are devices for enabling people with disabilities in both upper limbs to embark, drive, and disembark from the vehicle using only their legs. Honda offers the Honda Franz System lineup, as shown in Fig. 7.

The Franz System was developed by Germany's Eberhard Franz. Figure 8 shows the system. The left foot operates the device like a bicycle pedal, with the rotation transmitted to the steering wheel to provide steering input.



Fig. 6 Honda Techmatic B-type



Fig. 7 Honda Franz System for 2017 model year FIT

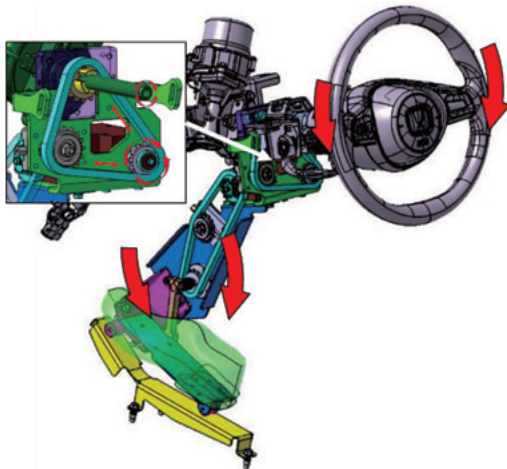


Fig. 8 Franz System driving assistance device

3. Development of the Hand-operated Driving Assistance Device for the 2020 Model Year FIT

3.1. Development Goals

In the lever-type hand-operated driving assistance device described in Clause 2.2.1., the left hand controls the accelerator and brakes. Therefore, for the purpose of usability, the lever is positioned to the right of the center console. To help ensure sufficient strength and durability, the lever is fixed directly to the body of the vehicle below the center console⁽⁶⁾.

In the 2020 model year FIT, the Qi charger, which in recent years has become a common automobile accessory, is installed in the forward part of the center console. With smartphones, people with physical disabilities can check for information about services at their destination, such as accessible toilets or parking with sufficient space for entering/exiting a vehicle. Smartphones are also useful at the destination, to search for new information when plans suddenly change. Smartphones are thus an important tool, and this makes the Qi charger, which can recharge devices that are simply placed on top of it, an important device for people with disabilities, as it does away with the hassle of connecting cables.

Therefore, in this development, Honda aimed to configure the Techmatic lever layout while still enabling use of the Qi charger.

3.2. Moving the Installation Position and Reassessing the Structure

In developing Techmatic for the 2020 model year FIT, Honda looked to continue using the Techmatic D-type while deploying the latest functionality and equipment. As shown in Fig. 9, the drink holder in the 2016 model year that was used as a fixing point directly under the body of the control lever was replaced by the Qi charger in the 2020 model year. At the same time, the gear-select lever was positioned right behind the Qi charger, and the space that was previously available to secure the Techmatic control lever to the body, as shown in Fig. 10, was no longer available. From the point of view of left-hand operability, the control lever had to be in the same position as previously, near the right rear of the gear-select lever, and a structure had to be



(a) 2016 model year



(b) 2020 model year

Fig. 9 Comparing drink holder of 2016 model year FIT and Qi of 2020 model year FIT

present that could provide sufficient support for the driver's body during braking. Therefore, in this development, the position of the control lever attachment relative to the vehicle body was moved 54.5 mm forward and 72.5 mm laterally toward the outside of the vehicle, compared to the 2016 model year. It was set on the side surface of the center console so as not to interfere with the seat when the latter was moved backward and forward. Furthermore, a structure was studied that could withstand loads from that side surface while fitting within the limited interior width of 25 mm.

3.3. 2020 Model Year Load Support Structure

For the purpose of this development, Honda decided to produce a load-bearing structure and the minimum necessary layout based on the new control lever mounting position. The control lever needed a support structure that could withstand longitudinal loads under braking, as well as loads imparted in all possible directions that could occur during embarking or disembarking or in unintended circumstances. With these considerations, for this structure Honda utilized two gear-select lever fixing points on the driver side and the two forward fixing points that were used in the 2016 model. With the change in control lever position, the two passenger-side fixing positions would no longer be available to bear loads, so additional structures were required to support the remaining four available fixing positions from the front, back, left, and right. Here, Honda considered a structure in which load paths were controlled only on the right side of the center console for longitudinal loads, while a T-shaped structure on the right side provided support for lateral loads.

Inside the center console, space between the Qi charger

and the gear-select lever and between the gear-select lever and center console side surface was utilized. Figure 11 shows how the structure that receives the longitudinal loads inside the center console and the structure that receives the lateral loads were installed between the Qi charger and the gear-select lever. Longitudinal loads are received by the two fixing points indicated by the triangles in the figure on the right side of the gear-select lever. The two fixing points indicated by the triangles in the figure near the center of the center console provide both horizontal balance when longitudinal loads are applied, and support for lateral loads.

3.4. Examining the Structure

In consideration of driver usability, the control lever was installed in the same location as the 2016 model year. The load-bearing requirement was based on an assumed maximum load in the longitudinal and lateral directions representing the full weight of the user being applied to the control lever. Figure 12 shows the actual structural layout resulting from this conception. The structure at the side of the console is better able to bear longitudinal loads with

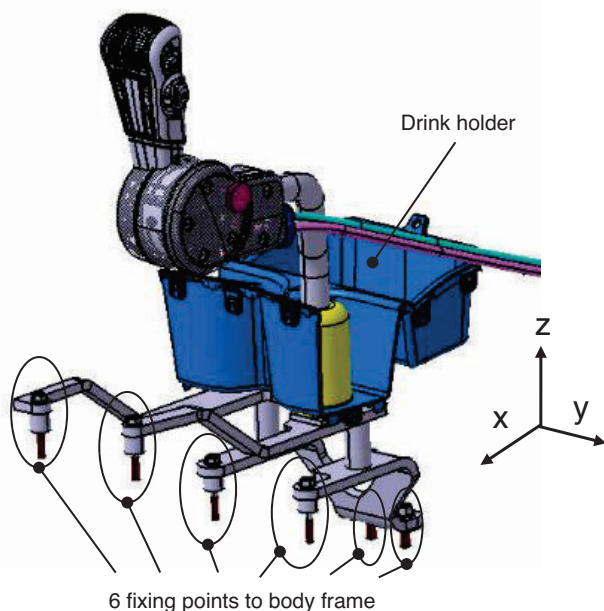


Fig. 10 Fixing image of 2016 model year FIT

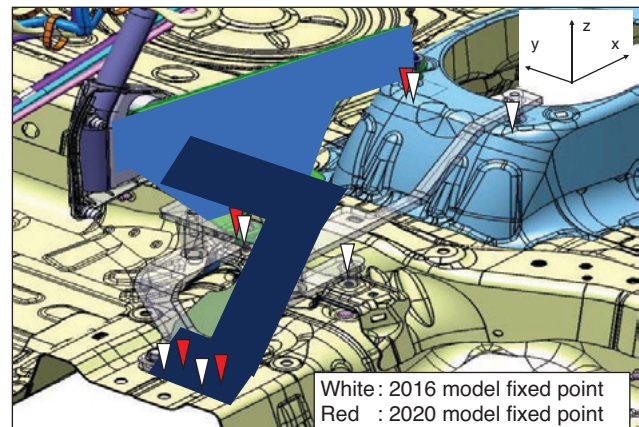


Fig. 11 Concept of structure for Techmatic

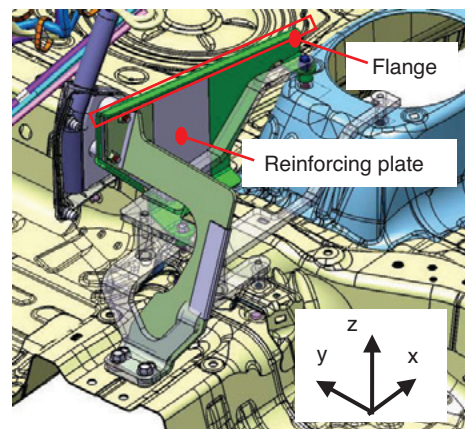


Fig. 12 Layout of load resistance structure

the two fixing points and, additionally, a flange that was added to the upper surface of the bracket. For lateral loads, a reinforcing plate was added to the aforementioned bracket to mitigate deformation, while an L-shaped load-bearing structure was fixed to the floor, being routed around other vehicle parts.

In constructing this layout, the longitudinal and lateral load paths were studied at the same time. These load paths are shown in Figs. 13 and 14. The red arrows in the figures indicate loads applied from the control lever, while the loads distributed within the bracket and applied to the floor fixing sections are shown by the blue arrows.

The basic thinking with regard to these load paths is as follows. The longitudinal loads applied to the control lever are evenly distributed from the pipe to the flat surface where deformation is suppressed by the attached flange. The two fixing points at the side of the console resist forward loads with pulling resistance of the fixing bolts and rearward loads with compression resistance. For lateral loads, of the four fixing points in the floor, the front three bear loads to the left with pulling resistance of the fixing bolts. The bolts bear loads toward the right with compression resistance. Although the structure provides less momentary support compared to the 2016 model year, it was made so that it could withstand the anticipated loads with the minimum available four fixing points, instead of the six fixing points of the previous model. It was then verified that the structure had the strength to resist the generated stresses.

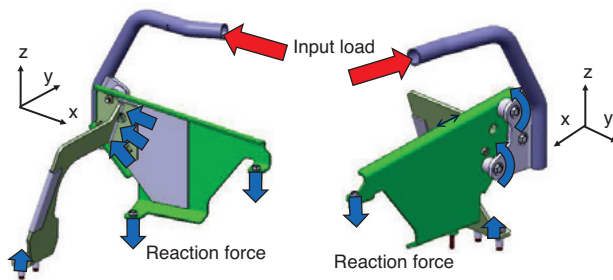


Fig. 13 Load path of back to front direction pressure

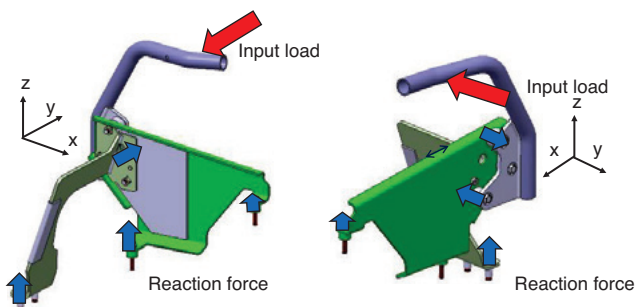


Fig. 14 Load path of outside to inside direction pressure

3.5. Verification Results

A strength analysis of this layout was performed for longitudinal and lateral loads. The strength analysis results are shown in Figs. 15 and 16.

According to the results of the analysis, although the longitudinal and lateral loads are concentrated on the pipe, the longitudinal loads are distributed to the two fixing points at the side of the console and the lateral loads are distributed to the three forward fixing points. This was as expected.

With the picture provided by these results, a test using actual equipment was conducted, which showed residual displacement within tolerances when the specified loads were applied. The physical test results are shown in Fig. 17. The vertical axis of the graph shows the change in load, where the required load is 100%, and the horizontal axis shows the ratio of displacement of the control lever, where maximum displacement is 100%.

As shown above, by utilizing a three-dimensional structure, it was possible to maintain the same structural strength in a more limited space and with only four fixing points, compared to the six fixing points previously available. Also, the new structure could be installed more quickly than the previous model. Further, the structure made it possible to maintain the same control lever position as before, which was a consideration in terms of ease of use. Figure 18 shows a comparison between structures of the 2016 model year and the final 2020 model year.

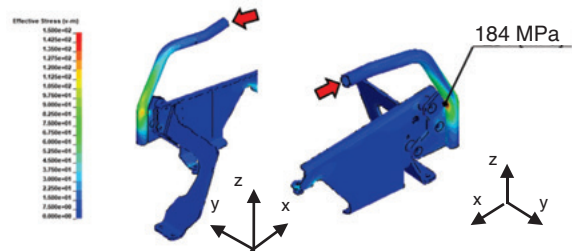


Fig. 15 Analysis of longitudinal direction

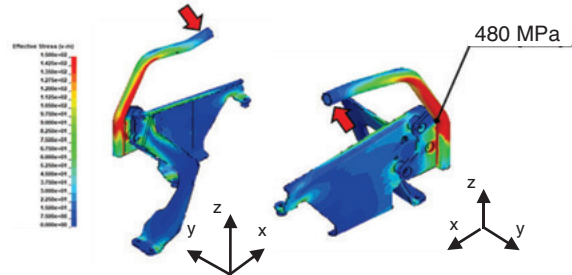


Fig. 16 Analysis of lateral direction

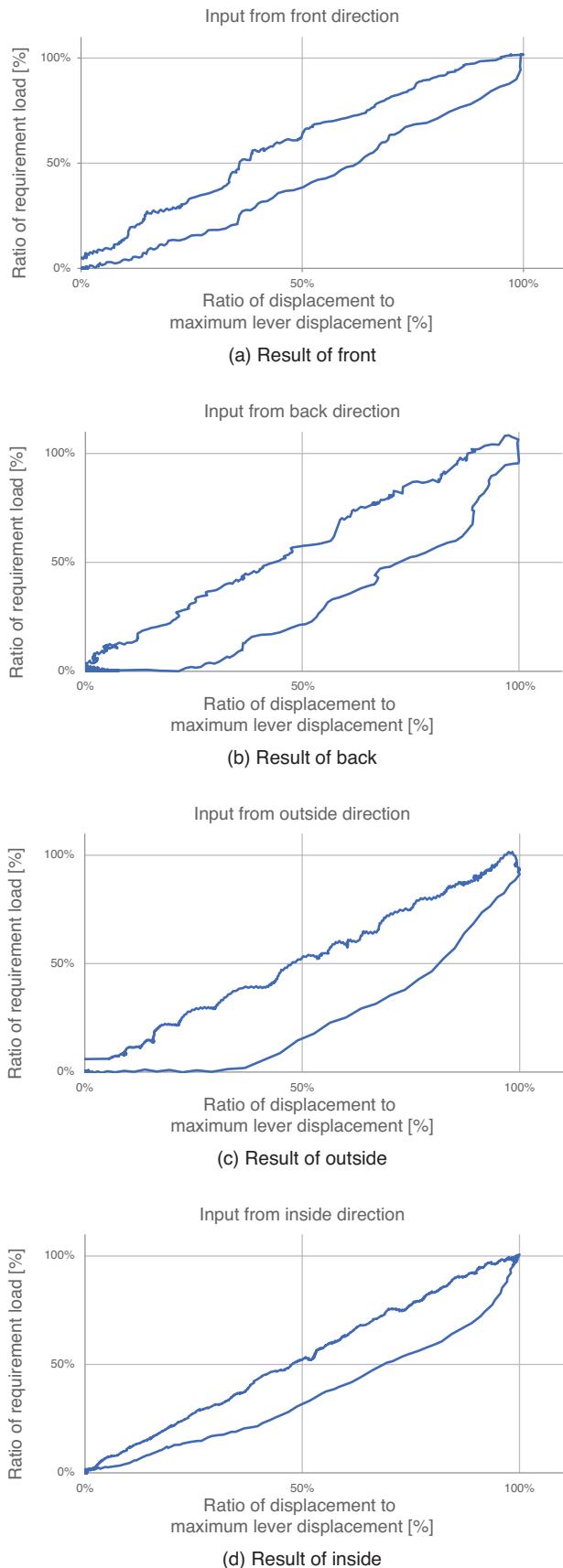


Fig. 17 Result of strength test

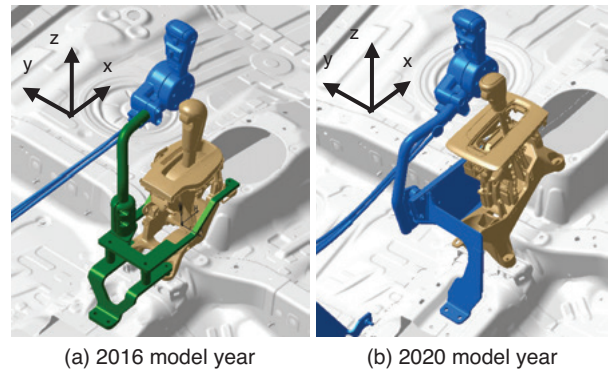


Fig. 18 Comparison of Techmatic for each model year

4. Conclusion

As a result of this development, Honda was able to achieve a different mounting structure on the side of the console, while still making the best use of the existing core Techmatic system.

A Techmatic system compatible with the 2020 model year FIT has now been released to market. From this initiative, Honda will accelerate and expand the company's "Fun for Everyone. Honda"⁽⁹⁾ concept throughout the industry, and work to further develop products to bring the joy of mobility to even more customers.

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Development of EU3200i Handy Portable Generator

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ABSTRACT

The newly developed EU3200i inverter generator of the 3-kVA class provides portability together with the ability to operate a variety of household appliances with leisure applications. The generator adds three key technologies to the 2-kVA class EU2200i to yield higher power output and enhanced usability while limiting increases in size and mass. The first was to mount a dedicated engine and alternator for compactness and high power output, increasing the specific power. The second was the initial adoption of a battery-less fuel injection system in the power product field, realizing simple startup capability while reducing fuel consumption. Finally, the new resin monocoque structure achieves noise levels on a par with previous products while limiting increases in mass. The power-weight ratio has been enhanced 15% over the preceding EU2200i, and with volume of 79 L and mass of 26.8 kg, the generator delivers the highest level of compactness and lightest level of weight in the 3-kVA class.

1. Introduction

Against a background of abundant demand for use as a power source for RV camping and other such applications in the leisure sector, the North America market for inverter generators is steadily expanding (Fig. 1). By power output, the 2-kVA class has a larger market, but in recent years, the 3-kVA class market has also been increasing. Where conventional use for generators in RV camping was mainly for air conditioning, the equipment used has grown more diverse (refrigerators, coffee makers, cooking appliances, and so on) so that higher power output became necessary, which is believed to be the reason for the rise in 3-kVA class unit sales.

In the leisure sector, generators are typically stored in the household then placed in the vehicle to be transported to the location where it is used. After arriving at the destination, there would also be many occasions when the generator is carried from the vehicle to where it is used, so the market considers portability to be important. For the 2-kVA class, therefore, a suitcase-style product that can be carried with one hand, like the EU2200i [Fig. 2(a)], is the mainstream. It was also determined that the need for portability is also high in the growing 3-kVA class. The 3-kVA class EU3200i generator (Fig. 3) was thus developed as an evolution of the suitcase-style EU2200i. The present paper introduces the technologies applied to the EU3200i to realize higher power output and enhanced usability.

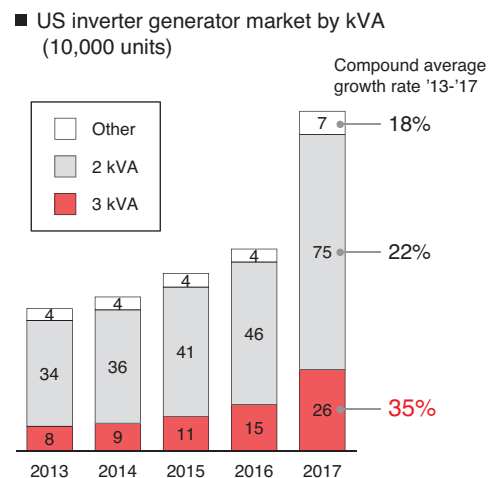


Fig. 1 US inverter generator market by kVA



Fig. 2 External view of previous Honda generators

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Fig. 3 External view of EU3200i

2. Model Overview and Main Specifications

The EU3200i product concept is a 3-kVA class generator that is lightweight, compact, and easy to use in leisure settings. Table 1 shows the main specifications of this generator in comparison with those of the base model EU2200i. The output to mass ratio has been increased from 104 VA/kg to 119 VA/kg, a 15% enhancement over the previous EU2200i. The new model has a volume of 79 L and mass of 26.8 kg. As shown in Fig. 4, this achieves greater compactness and lighter weight than existing products in the 3-kVA class from other companies or the EU3000i Handi [Fig. 2(b)]⁽¹⁾.

The exterior appearance was styled as “premium intelligence design” expressing the following three values.

- Change and evolution that are perceptible at a glance
- A small number of elements to give an appearance of simplicity and intelligence
- Senses of satisfaction, fulfillment, and superiority from possessing the product

In order to make the structure more attractive, a polyhedral shape utilizes the effects of light and shadow for an appearance of lightweight compactness, giving the product a uniquely distinctive appeal. The polyhedral shape also provides the product with surfaces that reflect light and others that are shadowed, further highlighting the product’s attractiveness. Removing the four corners also reduced the risk of damage and lowered the potential of bumping into a person, car, or other object during transport. This has also contributed to the product’s distinctive shape that conveys an impression of originality not found in any other company’s product.

Development used the existing EU2200i model with its high degree of portability as a base to which three key technologies were applied.

- (1) Dedicated engine and alternator that are compact and provide high power output
- (2) Battery-less fuel injection (FI) system, a first for power products, that realizes simple startup while reducing fuel consumption
- (3) A resin monocoque structure that achieves noise levels on a par with preceding products while limiting increases in mass.

The technologies that realize high power output and enhanced usability are described in detail below.

Table 1 Comparison of specifications

Model name	EU2200i	EU3200i
Max power [VA]	2200	3200
Rated power [VA]	1800	2600
Mass [kg]	21.2	26.8
Output to mass ratio [VA/kg]	104	119
Width [mm]	290	306
Enclosure volume [L]	63	79
Remote monitoring	Bluetooth	Bluetooth
Fuel supply system	Carburetor	Fuel injection
Run time (1/4 rated) [h]	8.1	8.6
Run time (rated) [h]	3.2	3.3
Noise level (rated) [dB(A)]	90	91

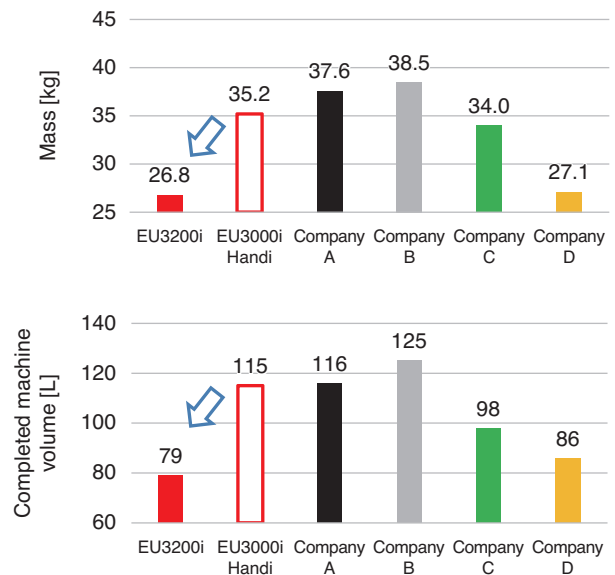


Fig. 4 Comparison of 3-kVA class generator mass and volume

3. Engine and Alternator

3.1. Compact, High-power GX130 Engine

The GX130 engine that is mounted in the EU3200i retains the width dimension of the GXR120 engine mounted in the EU2200i in order to realize the suitcase style. At the same time, the following three technological measures were also adopted to balance between engine power supporting increased power generation output and greater compactness and lighter weight.

- (1) High speed, high compression ratio, and increased intake air volume for higher power output

timing belt on the exterior of the bearing, reducing the distance between crankshaft bearings from 97.5 mm to 70 mm. This measure helped ensure support rigidity for the increased load from the higher compression ratio while also concentrating the crankshaft bearings on the cylinder barrel side. This allowed simplifying the structure of the crankcase cover, reducing engine weight by 0.6 kg over the GXR120.

3.1.3. Heightened cooling efficiency by separation of combustion chamber and valve chamber

The increased power output described above and support for stoichiometric operation described below resulted in a higher engine temperature for the GX130 than in conventional models. Installing it in a suitcase-style generator raised issues in establishing the engine's cooling performance. In order to address these issues, the timing belt was positioned on the exterior (the shaft-end side), and as shown in Fig. 6(B), the combustion chamber and valve chamber were separated. This made it possible, as shown in Fig. 6(C), to create a new cooling air course between the cylinder and timing belt. The resulting increased heat-radiating surface area on the cylinder head enabled higher speed while retaining the same-size cooling fan. Figure 7 shows a graph comparing trends in inverter generator power output and cooling fan outer diameter. The ratio of generated power output and fan outer diameter in the EU3200i has changed, indicating an increase in cooling efficiency.

3.2. Greater Compactness and Higher Power of Alternator

At the same time as in the engine, a compact, high-power alternator was also introduced in the power generating unit. Table 3 shows the specifications of the alternator.

In order for the alternator to achieve higher power output while also maintaining the width dimension of

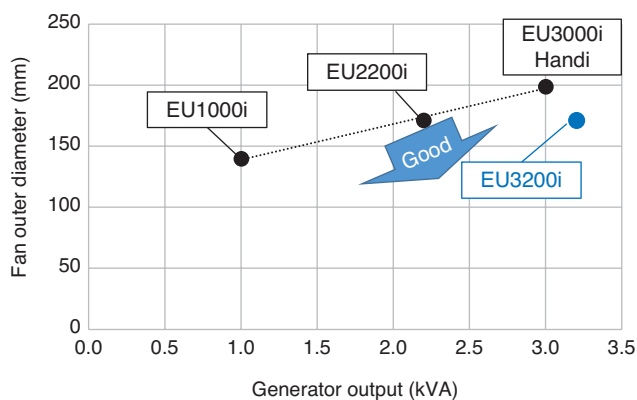


Fig. 7 Trends of fan outer diameter and generator power

Table 3 Comparison of alternator specifications

Model name	EU2200i	EU3200i
Outer diameter of rotor [mm]	150	
Continuous rated output of alternator [kVA]	2.4	3.7
Efficiency of alternator (rated) [%]	86<	
Thickness of stacking [mm]	24.15	21.00
Mass [kg]	3.2	3.0
Output weight ratio [kVA/kg]	0.75	1.16
Magnet type	Ferrite	Neodymium-based bond magnet
BHmax [MGOe]	5.5	10.0
Material of stator core	35A360	
Material of yoke	JSC270E	SPHC

the completed machine, it needed to have the same outer diameter as the EU2200i (rotor outer diameter of 150 mm) and be capable of producing 3.2 kVA of power. The measure to achieve this was to increase the alternator power output by adopting a neodymium bonded magnet with an energy product BHmax that at maximum is 1.82 times that of the ferrite magnet material in the EU2200i. Increasing the magnetic force in order to optimize every part also made it possible to reduce the stator thickness by 14%, reducing total mass of the alternator by 6.3%.

Using the above methods, this compact alternator retains the same outer rotor diameter as in the EU2200i while increasing maximum power output by 55%, as shown in Fig. 8.

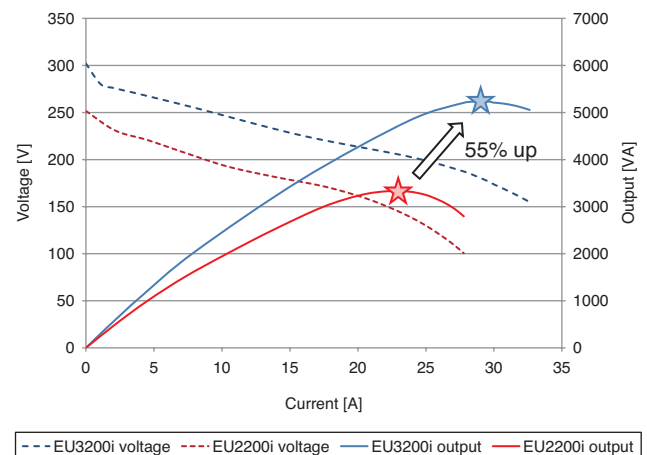


Fig. 8 Comparison of EU3200i and EU2200i alternator performance at 5500 r/min

4. Battery-less FI System

In order for the EU3200i to realize enhanced usability through simple startup while reducing fuel consumption by stoichiometric operation, it adopted the first battery-less FI system in power products.

4.1. Stoichiometric Operation for Reduced Fuel Consumption

In other than wide-open throttle usage, the EU3200i runs in stoichiometric operation mode by feedback control that utilizes the FI system and an exhaust O_2 sensor. As shown in Fig. 9, the engine maintains fuel consumption at the same level as conventional engines in terms of the power air-fuel ratio when running at maximum power. At or below the rated load of 4800 r/min, lower fuel consumption was actualized by stoichiometric operation. With this approach, as shown earlier in Table 1, the generator achieved a 44% increase in rated power over the EU2200i. The lower fuel consumption due to stoichiometric operation, however, made it possible to limit its increase in fuel tank capacity (from 3.6 to 4.6 L) to just 28% while maintaining continuous operating time at the rated power.

4.2. Simple Startup System

An FI system normally uses a battery-powered starter motor for startup. In the present development, however, an FI startup system was created that can start using a recoil starter operation just as in existing generators.

Portable generators are generally not started with the kick-starters used in motorcycles, instead using a hand-operated recoil starter. For this reason, it was necessary to generate electric power for the starter device at low rpm. This was accomplished by having the user pull on the recoil starter to turn the crankshaft, generating three-phase AC power with the alternator. The generated electric power is converted to DC power by the power source circuit inside the engine control unit (ECU), the DC power then being supplied to the FI device. The ECU also controls and drives the FI device,

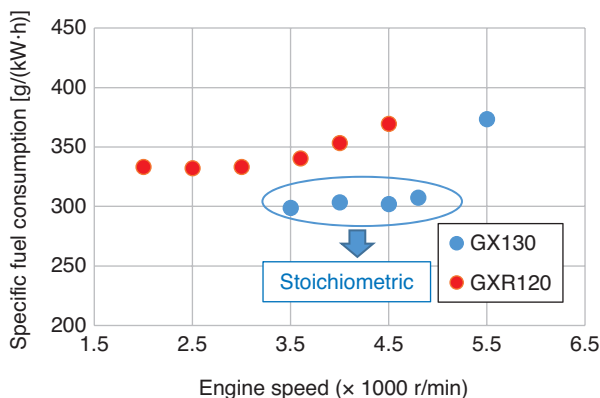


Fig. 9 Comparison of specific fuel consumption

consisting of fuel pump, injector, ignition coil, and throttle motor, thereby handling the starter control function.

The parts that require electric power to operate at startup are shown in Fig. 10, and a comparison with the existing EU2200i system is shown in Fig. 11. The differences in power required during startup because of the adoption of FI and other such reasons are that where the EU2200i has a “3 W” system (three power consuming parts), the EU3200i is “21 W” (seven power consuming parts). The parts that consume power during startup are shown below.

- EU2200i (three parts)
Ignition unit, ignition coil, throttle
- EU3200i (seven parts)
ECU, ignition unit, fuel pump, injector, engine temperature sensor (TE sensor), intake pipe negative pressure sensor (MAP sensor), throttle

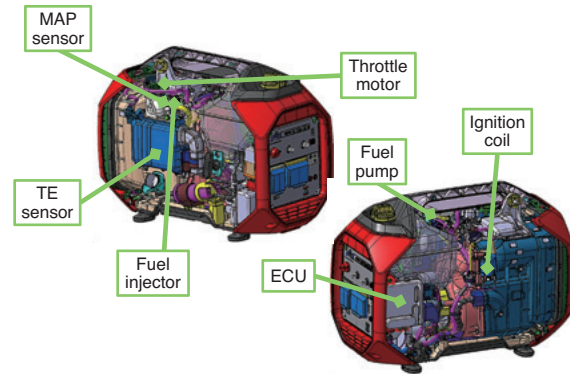


Fig. 10 Power consuming parts at startup

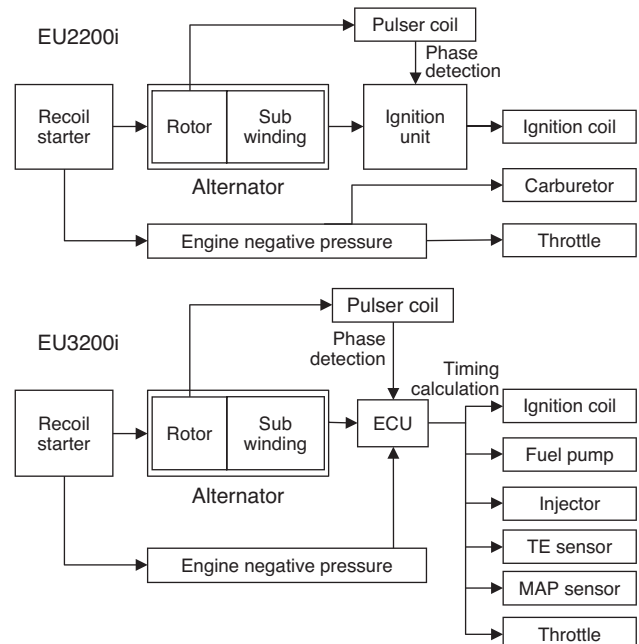


Fig. 11 Comparison of system with EU2200i

Table 4 Comparison of recoil specifications

Model name	EU2200i	EU3200i
Recoil reel diameter [mm]	100	65
Number of recoil rope turns	2.6	3.9

The following three technological measures were adopted to address the issues involved in generating electric power by a low-speed recoil starter without using a battery.

- Alternator specifications that meet the power requirement at low rpm.
- Control to supply a pilot injection of fuel immediately after the crankshaft turns to enable the engine to start at low rpm with fewer chances for ignition
- Recoil starter specifications (Table 4) with smaller reel diameter than in the EU2200i, and with the number of rope turns increased to obtain rotation speed when starting

Through these measures, a battery-less FI system was realized with enhanced usability that does not need choke operation at startup but that still provides the recoil starter operating feeling, number of starter operation times, and environmental temperature that allows equivalent starting with the existing EU2200i.

5. Packaging Technology

For the structure of the EU3200i housing, a single-handle suitcase-style shape was adopted to realize the convenient portability of a 2-kVA class inverter generator. In addition, a frame concept that would produce less noise than other products in the EU series was taken into consideration, revealing that the following four main measures would be required to address the issues.

- (1) Lower-noise housing by adoption of low-displacement, high-speed engine
- (2) Housing shape to realize greater compactness, lighter weight, and good portability
- (3) Structure that enables assembly while visually inspecting fuel pipe and electric parts
- (4) Support increased exhaust and exhaust air temperature resulting from stoichiometric operation

The housing structure of the EU3200i with these four measures implemented will be explained below.

5.1. Lower-noise Housing

The technological method used to downsize the EU3200i housing to the size of a 2-kVA class generator was adoption of a compact, lightweight, high-speed engine. With the installation of this engine in mind, a study was made of a housing structure that would be compact and lightweight, that would have cooling performance capable of handling continuous operation, and that would achieve a noise value of 91 dB(A) at the rated output of 2.6 kVA. The

frame structure of the EU3200i is shown in Fig. 12.

First of all, in order to decide on the packaging, a study was made of the layout of the main engine accessories in addition to a noise-reducing structure for the housing.

Sound-absorbing material was positioned in the resin shrouds that are nearest the high-speed engine and that direct cooling wind to raise engine cooling performance. This constituted a structure that fully covers the engine.

In further combination with the resin cover of the housing, this forms a full-cover resin structure in two layers that also has sound absorbing material to reduce mechanical noise from the engine.

5.2. Frameless Monocoque Structure and Housing Side Surface Shape

While the housing panels in a single-handle suitcase style provide for great compactness, lighter weight, and reduced noise, they also need to serve as a framework that protects interior parts in addition to providing a shape that realizes good portability.

When the suitcase style is used, the side surfaces of the housing are of a shape that may sometimes contact the user's leg during transport. The side surface was therefore given a rounded shape in order to reduce any unpleasant sensation. Figure 13 shows the shape of the housing side surfaces in cross-section.

If the side surfaces are too round, however, they will add dead space to the storage space for the engine, electric parts, and so on, potentially enlarging the housing.

In order to realize good portability together with reduction of dead space as much as possible, the curvature of the round side surfaces was studied. A continuous change was applied to the curvature of the middle part of the housing side panel as well as the upper part, where there is the greatest likelihood of contact when carrying the generator.

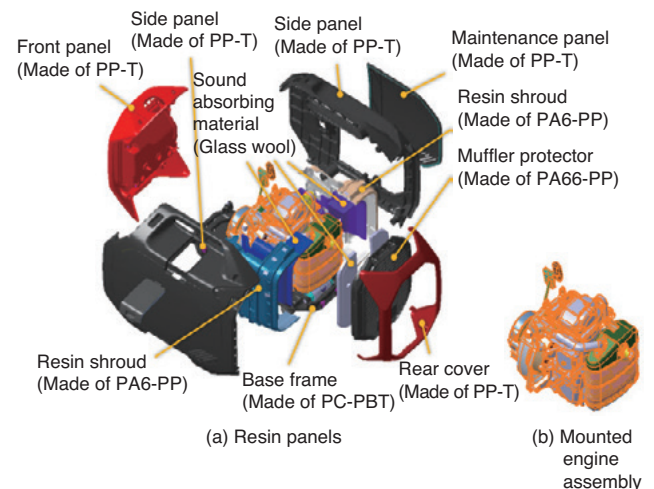


Fig. 12 Frame structure of EU3200i

Giving the side surface a curved shape and making it a shell structure heightens the strength and rigidity of the housing panels that comprise the monocoque frame. In this way, the housing was successfully shaped to realize greater compactness of the housing panels with good portability.

5.3. Fuel Tank Mounting Structure

Assembly of parts in conditions that do not allow visual inspection can cause fuel leaks, startup malfunction, or other serious quality imperfections. It is necessary that parts related to fuel leakage and generator startup and output, in particular, be housed in a structure that can be assembled while being visually inspected. Figure 14 shows a comparison with the structure of the previous model (the EU2200i) using a monocoque structure.

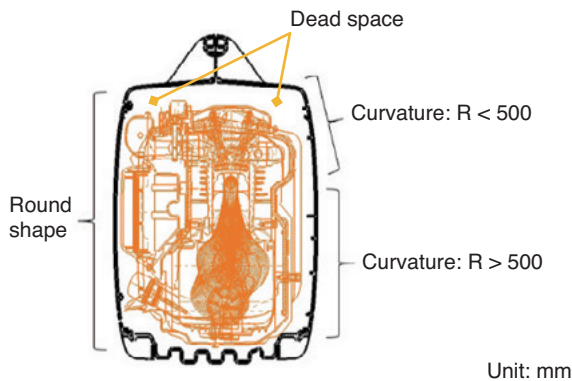


Fig. 13 EU3200i housing side shape (sectional view)

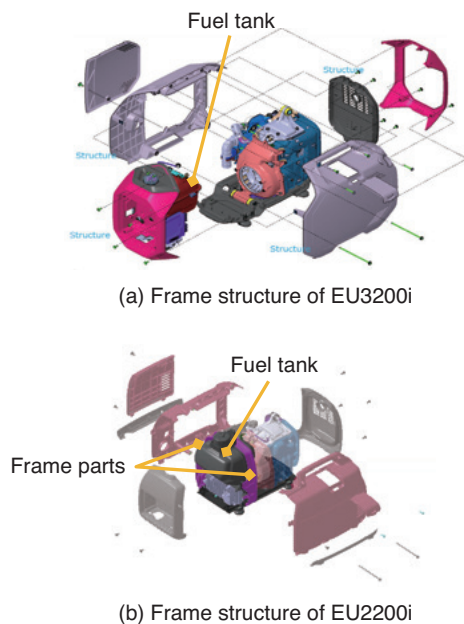


Fig. 14 Comparing frame structure of EU3200i and EU2200i

In the EU3200i, the fuel tank was designed to be assembled in a structure with a stand-alone front panel. As a result, the housing can be assembled with the side panels removed, allowing assembly while visually inspecting the fuel pipe and harness with electric parts.

In the EU3200i, the brackets for mounting the engine, the head cover, and the fan cover that also has an engine mount function are made of diecast aluminum. These all provide support for the engine main body, and are in turn supported by attachment through rubber mounts to the rear portion of the handle part on both side panels of the housing and to the base frame part.

In addition, the metal fuel tank and inverter unit are attached through rubber mounts to the front panel, and are assembled into a structure that can stand alone. Fastening this to the housing to form an integrated unit provides it and the engine mount structure with the strength and rigidity needed when the generator is being handled. An overview of the structure is shown in Fig. 15.

The following two items describe unique characteristics of this structure.

First, in addition to using the housing panels to comprise a monocoque structure, the panels were also given the function of holding the fuel tank in place, allowing the structure to stand free without an internal frame.

Second, this structure provides the internal constituent parts of those housing panels, the engine, and so on, with the strength and rigidity needed when the generator is being handled while also holding the mass of those parts directly in place by means of the housing handle.

The adoption of this structure reduced the number of

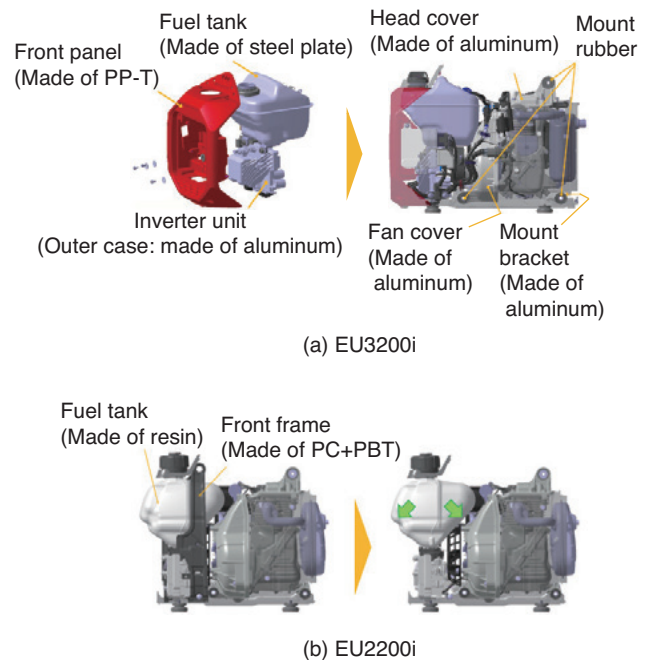


Fig. 15 Fuel tank supporting structure

parts and enhanced the packageability, contributing to a more compact size and lighter weight.

5.4. Support Increased Exhaust and Exhaust Air Temperature That Result from a Shift to Stoichiometric Operation

In conjunction with the rise in exhaust temperature during operation that results from the shift to stoichiometric operation, the positions of the muffler protection exhaust air outlets in the rear part of the generator were concentrated in the upper middle of the generator with the aim of enhancing the cooling performance of the muffler. The flow of cooling air around the muffler is shown in Fig. 16.

Arranging this structure so that cooling air flows over the surface of the muffler enhanced its cooling performance.

This structure also provides a labyrinthine structure that suppresses the emission of mechanical noise from the engine. Sound absorbing material realizes a mechanical noise reduction of -1 dB(A), contributing to achievement of the target noise value for the completed machine.

On the other hand, the amount of heat that the cooling air carries away from the muffler has increased, also raising the exhaust air temperature. The shape of the exhaust air louvers therefore takes into account the effects of the exhaust air on the environment where the generator is used.

In studying that shape, the environments in which the generator is used during operation were surveyed and needed target values were established. As a result of the survey, many of the environments in which the generators are used were found to be lawns. Figure 17 shows a location that is typical in terms of usability.

The target temperature for the exhaust air that is produced by operation of the generator was set at a value such that the grass would not be withered by contact with the exhaust air.

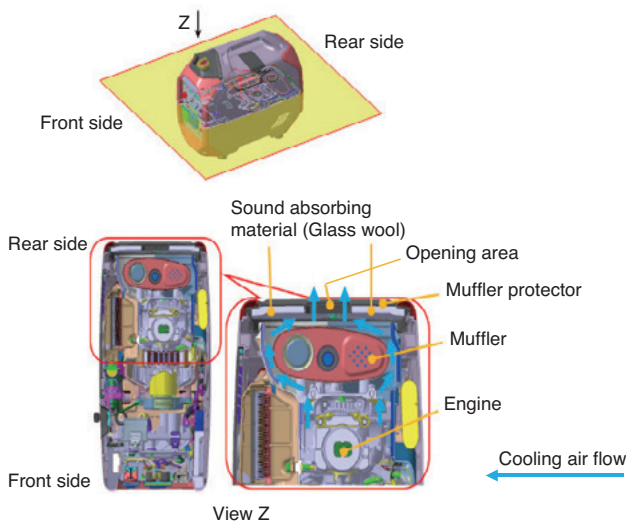


Fig. 16 Cooling air flow in muffler labyrinth structure

The shape for the exhaust air louvers was studied in light of their influence not only on the cooling performance of the generator but also on its noise characteristics. The angle of the louvers in the rear part of the housing was therefore set by determining the temperature requirements for each part of the generator while also confirming their influence on noise values. Figure 18 shows the area where the exhaust air louvers are located. Figure 19 shows the exhaust air contact surface temperature and the noise value at rated output that were achieved by the adoption of this structure. Setting the housing shape to take into account



Fig. 17 Example of typical placement location of single-handle portable generator

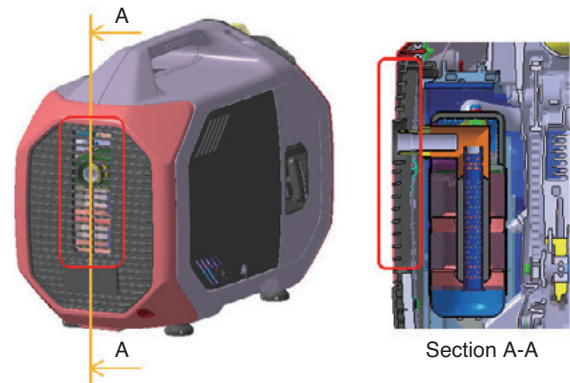


Fig. 18 Exhaust louver of EU3200i

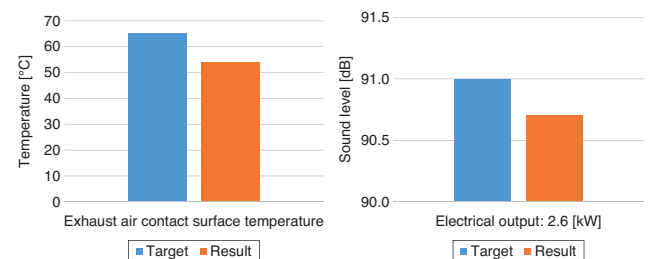


Fig. 19 Exhaust air contact surface temperature and sound level at rated output

the above factors of actual usability and influence on the environment where used achieved housing packaging that is unprecedented in its increased compactness, lighter weight, and lower noise as well as in its good portability, and that is also in conformity with its environment of use.

6. Conclusion

The EU3200i was developed on the concept of a lightweight, compact 3-kVA class generator that is easy to use in leisure scenarios. The power-weight ratio was enhanced 15% over the preceding EU2200i with its suitcase style. With a volume of 79 L and mass of 26.8 kg, it has achieved the top level of the 3-kVA class as a compact, lightweight generator. The following three main technologies applied to this model achieved a balance between higher power output and enhanced usability.

- (1) The dedicated GX130 engine with low displacement, high speed, and high compression ratio and the alternator that adopts a neodymium bonded magnet provide compactness and high power output.
- (2) While removing the need for a battery, which increases mass, the generator has stoichiometric feedback operation to hold down fuel consumption and at the same time realize simple startup as the first power product to adopt a battery-less FI system.
- (3) Use of a resin monocoque structure removes reinforcing frame members, achieving noise levels on a par with previous products while limiting increases in mass.

Through adoption of the above main technologies together with the immediately recognizable new “premium intelligence design,” the EU3200i is expected to become a product that customers will thoroughly enjoy not only in North America but worldwide as well.

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Development of 2 kW Class Electric Power Unit eGX

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ABSTRACT

The general-purpose GX series engines are installed in a broad range of work machines, serving as power sources for small construction machinery such as tamping rammers and plate compactors. Now, Honda has developed the eGX, a 2-kW-class zero-emission (zero emissions of HC/NO_x air pollutants and CO₂ greenhouse gas) electric power unit that can replace these engines. The eGX comes in integrated types and separated types, configured for differing work machine mounting conditions and usage environments. Of these, the integrated type was made for replaceability with engines of the same power band. The body size, the shape of the output shaft connected to the work machine, and other aspects are designed for replaceability with the GX engines. Since durability is important for the motor, vibration resistance of 150 m/s² or more was attained through the rotor shaft support structure and stator connecting structure. Fretting abrasion countermeasures were also implemented. In addition, a sealed motor case was used to provide the water and dust protection required on construction sites, and cooling sufficient for continuous operation at the rated output of 1.6 kW was attained with the use of inside and outside fans.

1. Introduction

Thanks to their high reliability and ease-of-mounting, general-purpose GX series engines⁽¹⁾ are used as power sources for various kinds of work machines and installed in a broad range of work machines, including those in the small construction machinery market. At the same time, in recent years the European Tier 5 Non-road Engine Emissions Regulations (otherwise known as EU Stage V)⁽²⁾ have started to go into effect, requiring even lower emissions from work machine engines. Also, for construction work inside tunnels and night-time work near residential areas, there is a demand for compact work machines and low-noise equipment. Some of this market demand has been met with work machine electrification that provides reductions in emissions and noise. However, it has become apparent that there is a need to electrify existing engine-powered work machines with minimal changes to the work machine itself while maintaining the same workability and durability.

From the point of view of workability, power supply systems for electrically driven work machines can be largely divided into power cord and battery types. Power cord-type work machines have electric motors that receive electrical power via cords plugged into an installed power source near the worksite. Power source, cord routing, and other requirements mean that such types cannot be used for all working environments and work angles/positions, and are

not suited for self-propelled work machines. With battery types, batteries may be installed inside the work machine, or attached so that they can be removed and replaced. Both methods have the advantage that, with a charged battery, they can be used at sites where there is no power supply, at unrestricted work angles/positions, and for self-propelled work machines. However, there are issues with achieving sufficient battery capacity for the required continuous run times, dealing with power shortfalls, and other challenges.

In terms of the durability of machines used in road construction and similar work, ground compaction before paving creates an environment with a lot of vibration and dust. Therefore, the motor, battery, and electrical devices need to be protected. Among existing battery-powered electrically driven work machines, there are those with exclusively designed devices and protective structures for the batteries and electrical devices. Also, output and other performance metrics are deliberately restricted compared to existing engine-powered work machines, in order to protect the electrical devices.

To respond to these needs, Honda has developed the eGX, a 2-kW-class zero-emission (zero emissions of HC/NO_x air pollutants and CO₂ greenhouse gas) electric power unit. Figure 1 shows an external view. This paper describes the technologies applied to increase the durability of the motor, which presented many technical issues in the mounting compatibility structure and inside the eGX itself.

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Fig. 1 External view of eGX

2. Development Goals

The development of the eGX involved two product concepts: mounting compatibility with engine-powered work machines, and performance and reliability that can withstand harsh usage environments. In concrete terms, the goals were as follows.

- (1) Packaging that would allow installation in work machines equipped with existing GX120⁽³⁾ and GX100⁽⁴⁾ engines, for compatibility with, and the same workability as, engine-powered work machines
- (2) Vibration resistance and environmental resistance to achieve durability in the expected usage environment of construction work machinery

Figure 2 shows an external view of the GX120 and Fig. 3 shows an external view of the GX100.



Fig. 2 External view of GX120



Fig. 3 External view of GX100

In addition, the following goals were set in order for the motor to have the same durability, work performance, and environmental resistance of engines.

- (a) Vibration resistance of 150 m/s²
- (b) Dust resistance (JIS Class 8 Z 8901) for levels expected to occur in ground compaction work environments at a construction site
- (c) Ability to withstand anticipated high-pressure cleaning
- (d) Cooling to allow a continuous rated output of 1.6 kW/3600 rpm or more

The above continuous rated output was a target value set to cover about 70% of work machines, based on a survey of required output for small-size construction machinery with mounted 2-kW-class engines (as of October 2021; survey by Honda).

3. Applied Technologies

3.1. Mounting Compatibility Structure

Two models of the eGX were established, according to the expected installation and running conditions of the machines in which they would be mounted. One model would be an integrated type, with the motor, battery, and power distribution unit (PDU) in a single package. The other would be a separated type, in which the control assembly, consisting of the battery and PDU, would be separate from the motor. Table 1 provides the main specifications of the eGX integrated and separated types, and the GX120 and GX100. Figure 1(a) shows the integrated type, and (b) shows the separated type. Both types would use the same removable lithium-ion battery pack and have the same workability as engine-powered work machines. Batteries would be replaceable. By having spare batteries on hand, operators could change batteries to keep the work machines operating continuously for long periods of time.

In order for the position of the motor output shaft (rotor shaft) to be compatible with installed engines, the same dimensions were used. Figure 4 shows the position dimensions of (a) the eGX and (b) the GX120.

Figure 5 shows the structure of the eGX integrated-type power unit. In consideration of the need to replace the battery during use, the battery was positioned on top of the motor⁽⁵⁾. This corresponds to the location of the fuel tank in engine-mounted work machines. To minimize the maximum width, the PDU is located on the side of the motor. Around the exterior, a plastic cover provides protection for the internal parts from external forces, dust, water, and so on. With regard to vibration resistance, the battery and PDU are composed of battery cells and a large number of electronic components. Therefore, the package included an L-shaped main frame installed to the motor and a vibration-damping mounting structure with vibration-damping rubber on the various mounting points⁽⁶⁾.

Next, Fig. 6 shows the eGX motor structure. In the motor, the stator and rotor are protected by a sealed structure consisting of aluminum front and rear cases. The stator was

Table 1 Specifications of eGX, GX120, and GX100

Model name	eGX		GX120	GX100
Type	Integrated type	Separated type	Integrated type	Separated type
Category	Permanent magnet DC motor		Single cylinder, OHV 4-stroke engine	Single cylinder, OHC 4-stroke engine
Cooling system	Forced air cooling		Forced air cooling	Forced air cooling
Shaft layout	Horizontal		Horizontal	Horizontal
Continuous rated power (kW/rpm)	1.6 / 3600		2.1 / 3600	1.7 / 3600
Maximum power (kW/rpm)	1.8 / 3600		2.6 / 3600	2.1 / 3600
Maximum torque (N·m/rpm)	5.6 / 3600 (3 s rating)		7.3 / 2500	5.7 / 3600
Set speed (rpm)	3600	3000, 3300, 3600	1400 - 3900	1850 - 4100
Noise level (L_{WA} (dB))	78	81	100	100
Overall mass (kg)	25.0	22.2	15.5	11.1
Battery mass (kg)	6.4		-	-
Operation temperature range (°C)	-15 to 40		-15 to 40	-15 to 40

fixed to the front case with bolts. The rotor has a shaft for power transmission in the center and magnets arrayed on the inner surface of the outer shell. The shaft is supported by bearings built into the rear and front cases. An inside fan is fixed to the yoke surface of the rotor, and an outside fan is fixed to the shaft. These cool the motor by creating airflow inside and outside the case, respectively. A fan cover and a top cover are positioned over the outside fan to guide the cooling airflow. These are fixed to the rear case.

For the eGX, the idea was to deploy the same parts used for generator alternators and use the same manufacturing

processes. Therefore, for the layout structure of the alternator parts, development began with the outer rotor structure, which is used in generators. In addition, the

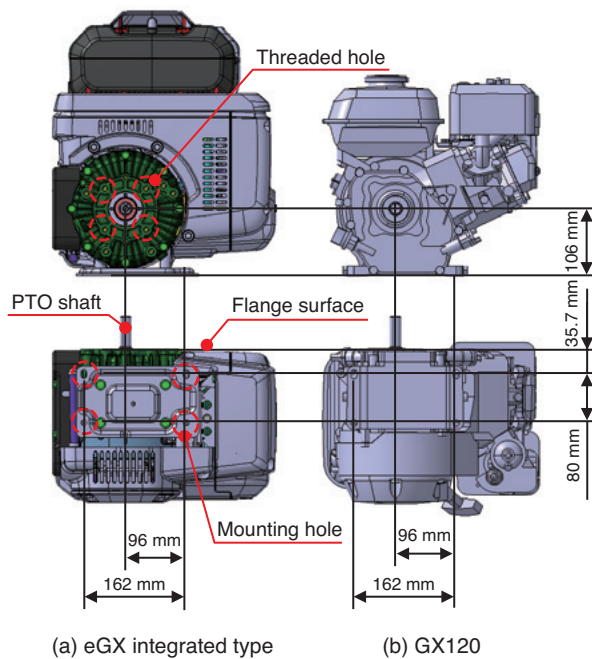


Fig. 4 Fitting dimension comparison of eGX integrated type and GX120

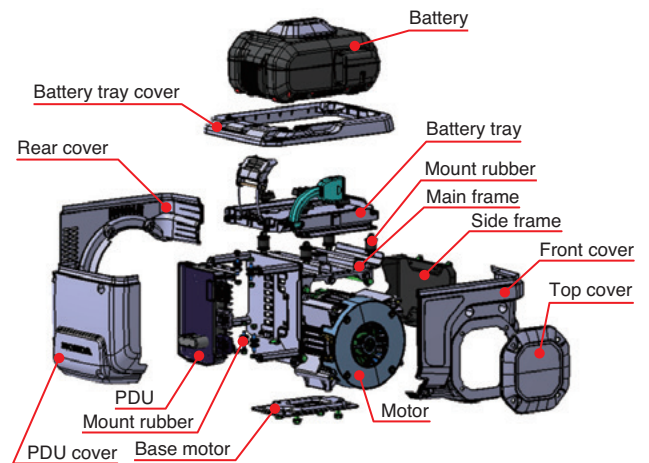


Fig. 5 Structure of eGX integrated type

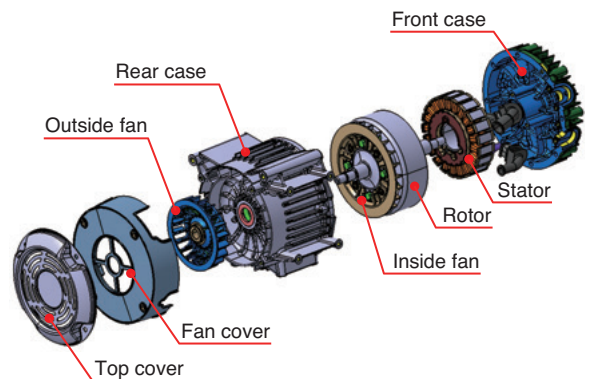


Fig. 6 Structure of eGX motor

following enhancements were made to meet the harsh environmental requirements for construction machinery, such as strength and vibration, dust, and water resistance.

3.2. Vibration Resistance

For the motor, the vibration resistance specification was set to 150 m/s² or better. This is because it was expected to be mounted on compaction machines such as tamping rammers and plate compactors, which present the harshest conditions in terms of vibration. Compaction machines tend to transmit vibration even to the power unit section, and existing general-purpose engines are also specially designed to handle strong vibrations. When assuming the mounting of an electric power unit on a compaction machine, the battery and PDU can be protected from vibrations with rubber mounts. However, the relative position of the motor's power transmission shaft and the drive unit of the machine is fixed, and rubber mounts cannot be used here. Therefore, the motor itself has to be highly vibration resistant. Accordingly, in order to create a motor with this vibration resistance, the results of assessments that determined the specifications are shown below.

3.2.1. Rotor support structure

To enhance rotor vibration resistance, the placement of the ball bearings that hold the rotor is very important. To maximize output and performance within the scope of production considerations, the air gap between the rotor and stator was set to the minimum of 0.5 mm. This air gap should be maintained even when vibrations occur in the motor section. The results of the rotor support structure assessment based on this are given here.

As described in Section 3.1., a design condition was to use the same form as generator alternators. Therefore, with the intention of using all the same yoke and magnet parts, a design was proposed in which the rotor shaft would have cantilever support that could withstand the rigors of a compaction machine, as in Fig. 7(a). At the same time, the double-sided layout that provided support with bearings on both ends of the case, which was considered to provide better vibration resistance, was also examined, although in this case there would be fewer shared parts with the generator alternator. This layout is shown in Fig. 7(b).

For the above two specifications, the rotor displacement

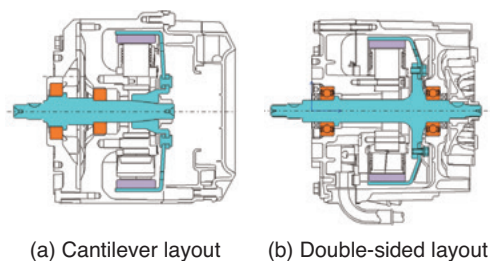


Fig. 7 Bearing layout

was calculated using a static analysis that modeled vibration excitation forces applied to the motor case in the radial direction. Figure 8 shows the results. Although the initial cantilever support structure idea met the static displacement target levels, when mounted in an actual tamping rammer, the rotor and stator came into contact. On the other hand, static analyses of the double-sided layout showed a 30% reduction of displacement compared to the cantilever support structure. This was then prototyped, and in durability tests with a tamping rammer, motor part contact was reduced. This led to the adoption of the double-sided layout.

3.2.2. Rotor shaft lubrication

The bearings were press-fitted into the front and rear cases. As the clearance between the rotor shaft and bearing was set loosely, there was concern that small vibrations could occur that would cause wear on the rotor shaft. If this wear were to progress, it would be possible that the clearance would increase, leading to increased noise, reduced motor efficiency, excessive vibration, and damage. Therefore, it would be necessary to reduce the amount of wear.

The amount of wear was assessed for tamping rammer mounting. Figure 9(a) shows an image of the rotor shaft after durability testing without additional protection. The brown areas show sections of bearing contact, and wear with pitching has occurred.

Here, an enhanced specification was considered. Figures 9(b), (c), and (d) show images of bearing contact on the shaft after applying various specifications and conducting durability tests.

Figure 9(b) shows a shaft with molybdenum disulfide coating. The molybdenum disulfide coating reduced initial wear, but as operating times increased, the amount of wear also increased. The molybdenum coating itself lacked durability, so while it provided good initial lubrication, this performance could not be maintained.

Figure 9(c) shows a shaft treated with a hard chrome plating. The aim was to reduce the amount of wear by increasing surface hardness by 17%, and the intended result was obtained.

Figure 9(d) shows the application of general-purpose industrial grease between the shaft and bearing to help

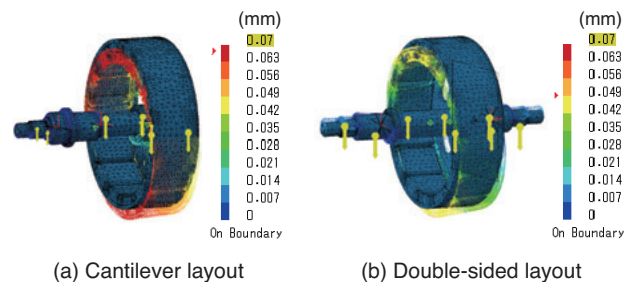


Fig. 8 Rotor displacement

avoid direct contact between metals. Durability testing resulted in a reduction of wear to less than 1/10. Also, the amount of wear was reduced, without maintenance, even when operation exceeded the expected life of the eGX.

With two good results to choose from, in the form of hard chrome plating or grease, grease application was selected for its good manufacturing and engineering characteristics.

3.2.3. Rotor shaft/yoke fastening structure

To provide the eGX with general-purpose versatility, Honda prepared the necessary range of shapes of output-shaft-type rotor shafts. By separating the yoke/magnet section and shaft sections of the rotor, the shaft only could be provided in multiple specifications. The results of assessments of the fastening structure for the separated sections are given here.

Figure 10(a) shows a structure in which, in the same way as generator alternators, a hub is placed between the yoke and shaft, and the yoke and hub are fastened together with rivets. When the motor with this specification was mounted in a tamping rammer, as shown in Fig. 11, fretting abrasion developed and advanced on the contact surfaces of the yoke and rivets. This wear could not be mitigated.

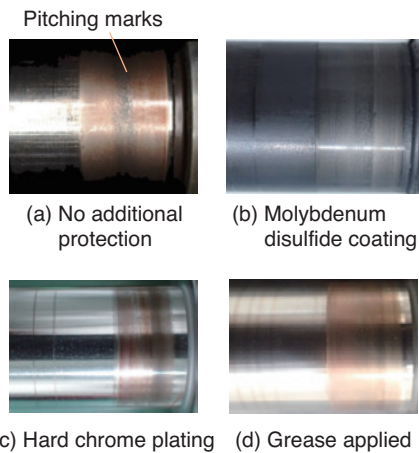


Fig. 9 Shaft condition inside of bearing after durability test

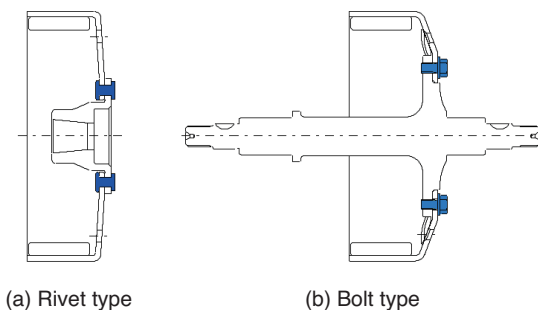


Fig. 10 Rotor shaft fastening

The rotor shaft was therefore connected to the yoke with bolts⁽⁷⁾. This is shown in Fig. 10(b). This change addressed the issue described above, and was adopted as a specification.

3.2.4. Stator fastening structure

The stator was fastened to the front case with a bolt. The fastening point was limited to a position between the slot and the shaft due to its structure, but it was necessary to suppress stator slot vibration and protect the electrical winding.

On generators, two M5 bolts were used for this fastening, but on a tamping rammer, this damaged the stator. It was believed that the cause of this was resonance of the stator. Figure 12 shows the results when acceleration of the stator was checked by applying acceleration of each frequency to the front case. When the stator was fastened with two bolts, the point of resonance was around 360 Hz. Additionally, analysis of the tamping rammer input vibration components showed that there was a main input caused by the rolling compaction cycle at around 11 Hz, but there was also a relatively large input at around 400 Hz. It was inferred that the characteristics of this input and stator resonance generated excessive vibrations that impacted the strength of the stator.

As a response, a fastening was assessed that used three M6 bolts to increase the point of resonance to 500 Hz or more. The result, as shown in Fig. 12, confirmed that the acceleration near 360 Hz was reduced to about 1/10 of the previous figure, and the resonance point became 600 Hz or higher. This change addressed the issue described above, and was adopted as a specification.

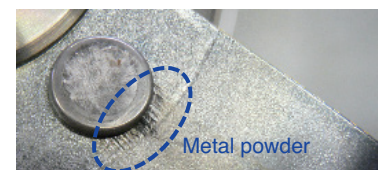


Fig. 11 Fretting abrasion on plane of rivet

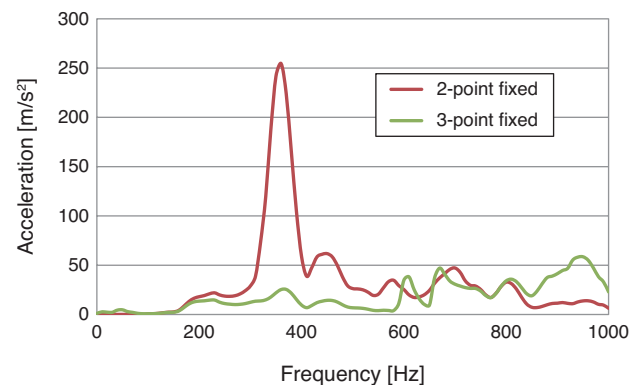


Fig. 12 Acceleration of stator

3.3. Environmental Resistance

3.3.1. Dust and water resistance

Figure 13 shows the internal structure of the eGX. As shown in Fig. 14, the motor section was protected with a dust seal to stop dust and water from entering through the shaft area. Figure 15 shows the drain holes and vent hole in the motor case. The drain holes located at the bottom of the motor section help ensure that water will be discharged in cases when water contaminates the motor due to an unanticipated event. The vent hole located higher up on the motor is part of the water protection specification, and enables the water to drain from the drain holes.

3.3.2. Cooling

When considering motor cooling, it is important to understand stator loss level, as this is a source of heat generation. Figure 16 shows a total loss analysis during eGX rated operation. The breakdown of the total loss during rated operation was about 20% for the motor controller and about 80% for the motor itself. With regard to this motor loss, copper loss that occurs in electrical winding and iron loss that occurs in the core together account for 47% of loss. Friction loss in the bearings accounts for about 23%. Airflow loss from the outside and inside fans used in this design accounts for about 12% of loss. Some of the iron loss is externally cooled by direct heat conduction from the

core to the case or shaft, but part of the iron loss and all the copper loss is discharged into the inside of the motor, and this leads to an increase in the localized temperature. This means that temperature protection provided by the internal sensor for pulse detection affects the continuous rated operating time of the power unit.

Therefore, the following cooling methods were considered.

Figure 17 shows the detailed structure of the motor.

The 21 poles of the stator core consisted of electrical winding on insulators, and each winding is connected by a bus ring. The sensor for pulse detection is fastened to the stator core, and the stator core is fastened to the case by bolts. Fourteen pole magnets are attached to the inner circumference of the rotor yoke, and these rotate through the air gap at the outer circumference of the stator. For the motor type, an outer rotor structure is used. The benefit of the outer rotor structure is that it is possible to array the magnets of the rotor on the outer circumference, thus allowing for larger magnets. In particular, for manufacturing purposes, ferrite magnets are used in the motor and in the manufacture of power generators. Compared to the more commonly used neodymium-type rare earth magnets, the maximum energy product is as low as 1/5 or less.

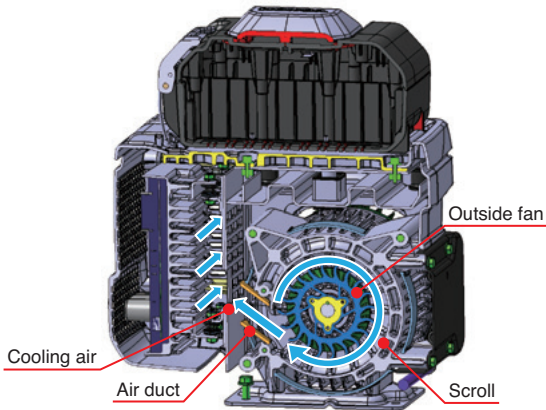


Fig. 13 Internal view of eGX

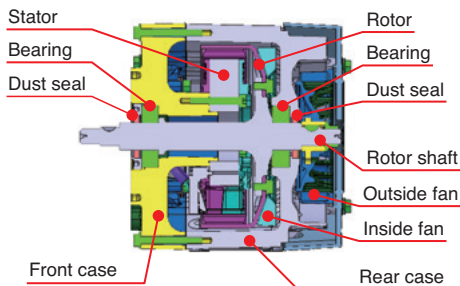


Fig. 14 Internal view of eGX motor

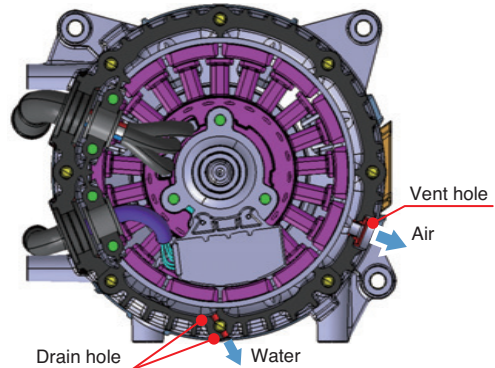


Fig. 15 Water hole layout

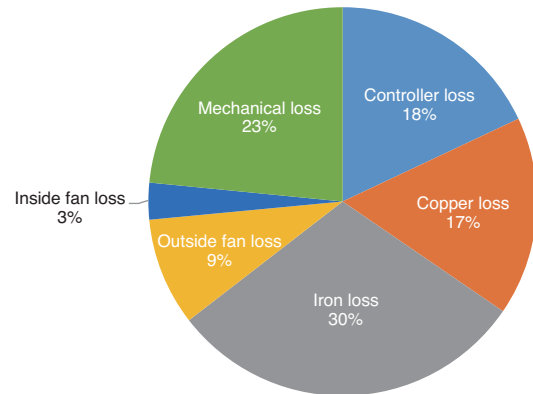


Fig. 16 Loss analysis of electric power unit

By arraying more magnets on the outer circumference and increasing the equivalent cross-sectional area of the magnets, the amount of air gap flux between the stator and the rotor could be increased.

On the other hand, the electrical winding of the stator is located on the innermost side, passing through the air gap with the rotor core as well as the air gap with the case. This presents an issue for the outer rotor structure; namely, having to cool the surrounding sections that are heated due to the current flowing through the electrical winding.

In many general-purpose industrial motors, a cooling method is used in which an outside fan is located above the shaft, and the airflow from this fan is circulated by forced convection to the outer part of the motor case. However, the eGX has an enclosed motor case for the purpose of dust and water protection when used in construction machinery. Therefore, the cooling airflow from the outside fan alone was not sufficient to reduce the internal temperature of the motor. Due to this, the installation of an inside fan was investigated as a means to create cooling airflow inside the motor. Figure 18 shows the results of fluid analysis of cooling airflow generated by an inside fan, using a simple model.

Since it was a challenge to reproduce the air gaps between 21 adjacent electrical windings in this analysis, the fluid analysis was performed using a simple model with a wider air gap between the stator and rotor.

The inside fan mounted on the upper part of the rotor rotates and pushes the air inside the stator to the outside of the fan. This air passes through the gap between the outer surface of the rotor and the inner surface of the case and flows to the lower side of the stator. At the same time, the air at the lower side of the stator flows to the rear upper surface over the electrical winding of the stator. This internal air circulates between the electrical winding, where copper loss heat generation occurs, and the stator core, where iron loss heat generation occurs, and thus carries the heat from inside the motor to the outer case.

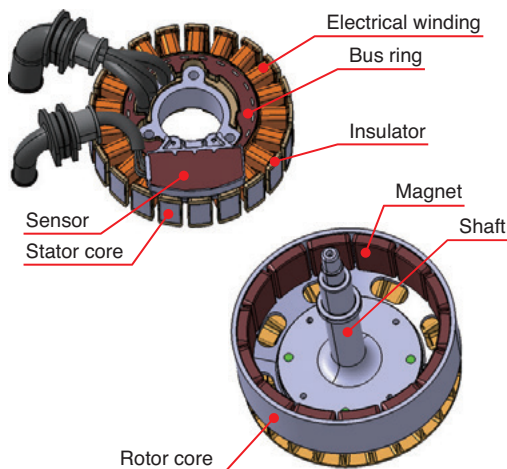


Fig. 17 Detailed structure of motor

Figure 19 shows the result of the flow rate analysis of the airflow through the gap between the outer surface of the rotor and the inner surface of the case. Based on the calculations, four times greater internal airflow can be expected in a motor with an inside fan compared to one without such a fan.

As a result of this investigation, the eGX motor was equipped with both an outside and inside fan for cooling purposes. Figure 20 shows a cross-sectional view of the cooling system. To summarize the cooling airflow,

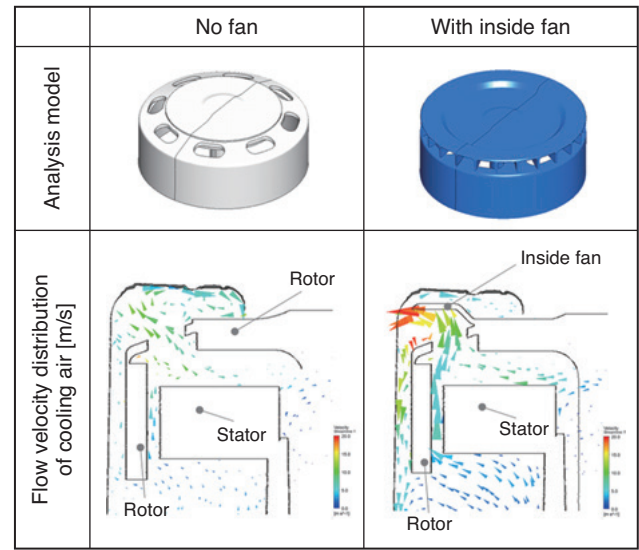


Fig. 18 Fluid analysis result inside motor

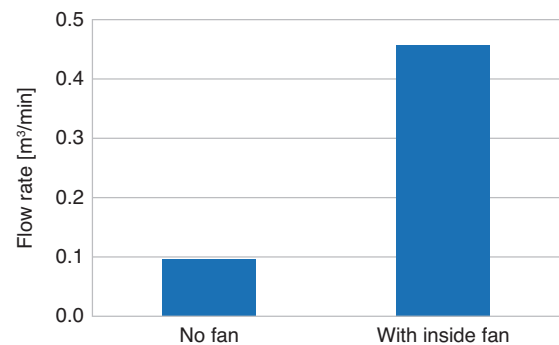


Fig. 19 Flow rate passing through gap between rotor and case

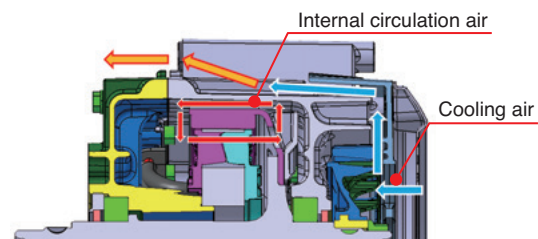


Fig. 20 Image of motor cooling system

Table 2 Temperature rising test result at rated point after 50 min

Measurement place	40°C conversion		
	No fan (°C)	With outside fan (°C)	With inside and outside fans (°C)
Motor internal atmosphere near sensor	91	74	67
Internal sensor	96	78	69
Electrical winding (Rear side)	150	130	96
Electrical winding (Front side)	141	122	91

the inside fan draws the internal air around the electrical winding from a hole to the rotor core and the air inside the motor is circulated by convection. This raises the heat transfer rate from the electrical winding to the air inside the motor and from the internal air to the housing. The outside fan dissipates heat from the housing into the atmosphere by blowing cooling air on the outer surface of the housing⁽⁸⁾.

Next, Table 2 shows the test results comparing temperature rises after 50 minutes for different conditions during rated motor operation. The test compared cooling performance under three conditions: no fans at all, outside fan only installed, and inside and outside fans installed. With no fans, the temperature of the electrical winding rose rapidly, the motor's internal temperature also rose, operating temperatures were not maintained, and continuous rated operation could not be attained.

When only the external fan was installed, the internal temperature of the motor was about 20°C lower than with no fans at all, but a localized temperature rise occurred around the electrical winding and operating temperatures were not maintained.

In a setup with both inside and outside fans, operating temperatures were maintained and continuous operation was possible.

4. Conclusion

Honda has developed the eGX, a 2-kW-class zero-emission electric power unit that is able to withstand the harsh usage environments faced by small construction machinery market. The following technologies were used to achieve mounting compatibility and motor durability.

- (1) In consideration of the work machine mounting conditions and usage environment, an integrated type and separated type were established. The integrated type was made for mounting compatibility with engines of the same power band. The packaging was constructed so that the external dimensions, layout of electrical devices, and position and installation surface of the output shaft had the same dimensions.
- (2) With regard to motor durability, by optimizing the rotor shaft support structure and implementing fretting abrasion countermeasures on the bearings support parts,

using a bolt-fastening structure between the yoke and rotor shaft, and optimizing the stator fastening, Honda built a motor that could withstand vibrations of 150 m/s² or more.

- (3) In terms of environmental resistance, mainly by using a largely sealed-structure motor case, Honda built a motor that was protected against both dust and water. Also, by using a design that had inside fans in the motor as well as outside fans, Honda achieved sufficient cooling for a continuous rated output of 1.6 kW/3600 rpm.

These have made it possible to meet the need to electrify existing engine-driven work machines in the face of ever-tightening environmental regulations.

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Development of Anti-vibration Type Brush Cutter for European Commercial Market

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ABSTRACT

Honda's first anti-vibration type brush cutter UMK450E XEET was developed for the European commercial hand-held market. This development established vibration reduction technology by focusing on the shape of the mount rubbers equipped to the anti-vibration frame and the driveshaft bearing placement. The mount rubber shape was newly designed to simultaneously satisfy demands for vibration reduction, swing stiffness during brush cutting work, and actual machine own-weight support. A new bearing placement design method was established that uses the driveshaft as a dynamic damper. This bearing placement design suppresses handle vibration variation due to the static unbalance amounts of the cutting head and the engine flywheel. The newly designed mount rubber and bearing placement realized a low vibration brush cutter that satisfies the user demands in the European commercial market.

1. Introduction

Honda has deployed and expanded sales of brush cutters equipped with 1-horsepower and 1.5-horsepower engines in the European consumer hand-held market. In this development, a brush cutter equipped with a 2-horsepower engine targeting the European commercial market has been newly developed.

Brush cutters are hand-held work machines that are used for grass cutting and other work by farmers and landscapers in the 2-horsepower class commercial market. Many users work for long times a day, especially in this market. According to the results of market research (Honda survey), the working time of home users in the consumer market averages about an hour a day, whereas users in the commercial market who are the target demographic of this development were found to work for up to about six hours a day. The excitation force due to the rotation of the cutting head and the engine constantly acts on the handle during the work, so low handle vibration is important for users who work for long times. In addition, users work while adjusting the engine speed with the throttle lever according to the work environment (load), so low handle vibration over a wide engine speed range is also important. To meet these demands, Honda developed the UMK450E XEET, which is the company's first brush cutter equipped with an anti-vibration frame structure. Figure 1 shows an overall image

of the developed anti-vibration type brush cutter.

This development established two main vibration reduction technologies in order to satisfy user demands in the European commercial market. First, the shape of the mount rubber, which is the main component of the anti-vibration frame structure, was newly designed. This reduced vibration over a wide engine speed range. Next, focusing on the bearing placement of the driveshaft that transmits the engine drive force to the cutting head, a new bearing placement design method was established. This reduced the variation of the handle vibration value relative to the increased excitation force due to the imbalance of the rotor. This paper introduces these vibration reduction technologies.



Fig. 1 Image view of UMK450E XEET

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2. Development Goals

2.1. EU Physical Agents (Vibration) Directive and Machinery Directive

The EU Physical Agents (Vibration) Directive 2002/44/EC⁽¹⁾, which is a key index when establishing target values for vibration reduction, is explained below. This directive determines the standard values for vibration exposure level as a minimum requirement for protecting workers from safety and health risks. The vibration exposure action value (EAV) and vibration exposure limit value (ELV) are set as the standard values for vibration exposure level. When the daily vibration exposure level exceeds the EAV, employers are obligated to take steps such as suppressing the vibration exposure time and selecting brush cutters with lower vibration. However, the daily vibration exposure level should not exceed the ELV under any circumstances.

In order to satisfy the EAV and ELV standard values, the allowable daily working time (vibration exposure time) is indicated as a guideline according to the handle vibration value. Figure 2 shows this relationship⁽²⁾. The horizontal axis of the graph is the vibration exposure time per day, and the vertical axis is the handle vibration value. The handle vibration value is expressed by the composite value of the handle vibration values at idling speed and racing speed (133% of engine speed at engine maximum power). The two curves represent the times until the vibration exposure level reaches the EAV and ELV, respectively, according to the handle vibration value. It is necessary to set a development target value for handle vibration that satisfies the working times of the target users based on the graph in Fig. 2.

In addition, the EU Machinery Directive 2006/42/EC⁽³⁾ requires development manufacturers to describe uncertainty along with the measured handle vibration value in the brush cutter operating instructions. Uncertainty here refers to the range of variation in the vibration acceleration that occurs in the measurement procedure and in manufacturing. Anti-vibration type brush cutters in particular have a small handle vibration value compared to brush cutters without an anti-vibration frame. For this reason, users more easily feel the magnitude of the vibration acceleration variation through the handle. This makes it important to reduce uncertainty as much as possible when developing anti-vibration type brush cutters.

2.2. Development Targets

Targets (1) and (2) below were set in this development.

(1) Handle vibration

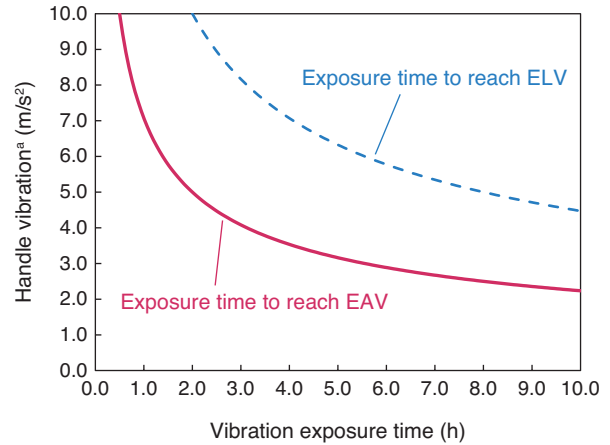
It is necessary to set a target value that can protect the target users, who according to market research work for up to about six hours a day, from safety and health risks based on the EU Physical Agents (Vibration) Directive 2002/44/EC⁽¹⁾. Therefore, a handle vibration value of 2.5 m/s² that satisfies the EAV even when working for eight hours a day was set as the development target for this development. (The handle vibration measurement method complies with

ISO22867-2011⁽⁴⁾.) An additional development target that resonance exceeding 2.5 m/s² does not occur over the entire practical engine speed range was also set.

(2) Uncertainty of handle vibration value

Users can freely replace the cutting head according to the work application. This means that cutting head replacement may unexpectedly increase the excitation force and excite resonance of the pipes and other parts. Such resonance is a factor increasing the uncertainty of the handle vibration value. A key to reducing uncertainty is to reduce the variation of the handle vibration value relative to the variation in the static unbalance amount of the equipped cutting head. In addition, it is also desirable to reduce the variation of the handle vibration value relative to the variation in the static unbalance amount of the engine flywheel, which is the main rotating component. In consideration of the respective variation in the static unbalance amounts of the cutting head and the engine flywheel, the target for this development was set to keeping the variation (uncertainty) of the handle vibration value to less than 2.0 m/s². (The handle vibration measurement method complies with ISO22867-2011⁽⁴⁾.)

In addition to the above-mentioned development targets (1) and (2), this development also took into account the swing stiffness described below from the viewpoint of enhancing merchantability. To reduce handle vibration, it is necessary to design small spring constants for the mount rubber equipped to the anti-vibration frame. However, when the spring constants of the mount rubber are small,



$$a: \text{Handle vibration (m/s}^2\text{)} = \left[\frac{1}{2} (I_d^2 + R_a^2) \right]^{1/2}$$

I_d : vibration value at idling speed (m/s²)

R_a : vibration value at racing speed (m/s²)

Note: I_d and R_a are three-axis composite values of each filtered acceleration response (r.m.s. value). Racing speed is engine speed of 133% of speed at maximum engine power.

Fig. 2 Relationship between handle vibration and allowable vibration exposure time

the swing stiffness during brush cutting work decreases. Decreased swing stiffness here refers to an increase in the angular displacement delay of the working part (cutting head) relative to the angular displacement of the handles during the swing operation of brush cutting work. Figure 3 shows an image of angular displacement delay. The mount rubber was designed bearing in mind that such angular displacement delay of the working part may lead to reduced merchantability, such as reduced work efficiency and an increased feeling of fatigue due to an inability to operate as intended.

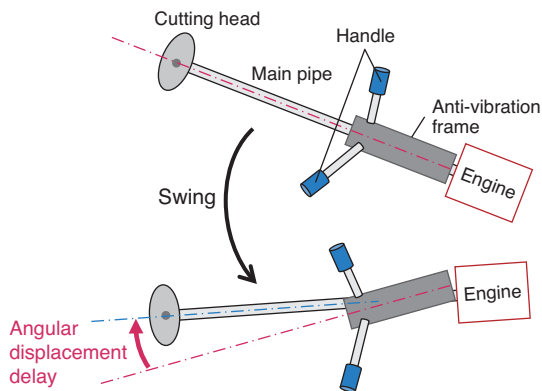


Fig. 3 Angular displacement delay of cutting head from handle swing motion

3. Vibration Reduction Technology

This development established a new mount rubber design and a new bearing placement design method to achieve the development targets described in Section 2.2.

3.1. Mount Rubber

3.1.1. Anti-vibration frame structure

Figure 4 shows an outline of the anti-vibration frame structure from the engine to the handle attachment part in the developed anti-vibration type brush cutter⁽⁵⁾. The anti-vibration frame has a structure that holds a resin box frame with two mount rubbers each in the front and rear for a total of four mount rubbers. The mount holder and clutch housing fastened to the main pipe support each mount rubber, and the handle pipe is fastened to the resin box frame via the handle holder. This type of anti-vibration frame structure enables reduction of the vibration transmitted to the handle from the cutting head and engine, which are sources of vibration.

Next, the features of the mount rubber shape and mount rubber support structure are described. Figure 5 shows an example of a mount rubber shape in the examination process, and Fig. 6 shows the mount rubber support structure. The mount rubber shape has an outer wall part

and a center hole-shaped part connected by multiple ribs. As shown in Fig. 6(a), the resin box frame has support parts on the front and rear ends into which the outer wall parts of the mount rubbers are inserted. Meanwhile, the mount holder and clutch housing have structures with pin-shaped protrusions in two places each, as shown in Fig. 6(b). The structure supports the mount rubbers from the inner wall side by inserting these pin-shaped protrusions into the hole-shaped parts in the center of the mount rubbers. Figure 6(c) shows an image of the condition with each part assembled. As a result, this structure supports the mount rubbers by the outer wall part and the center hole-shaped part, and reduces the vibration transmission rate with the rib shapes in-between.

3.1.2. Mount rubber design requirements

(1) Spring constant of vibration displacement region

The region where the hole-shaped part in the center of the mount rubber is displaced by an input load due to vibration in the condition with the user holding the handles was defined as the vibration displacement region. Based on

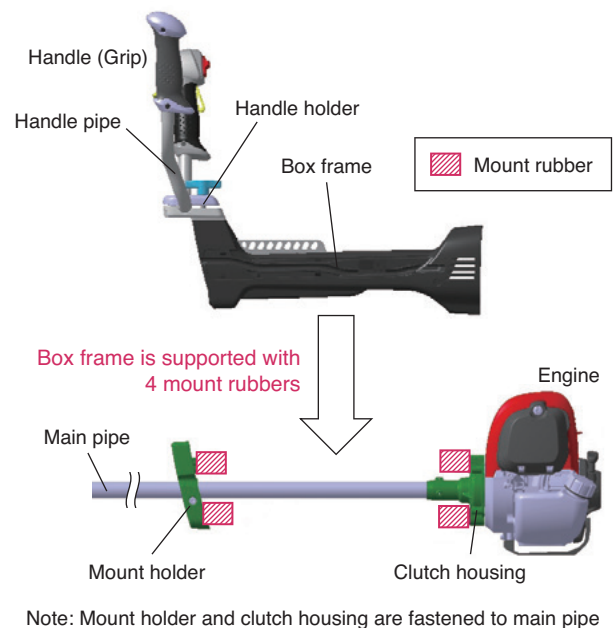


Fig. 4 Anti-vibration frame structure

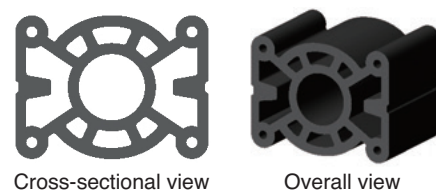


Fig. 5 Mount rubber

preliminary testing results, the expected displacement in this region was set to less than around 3 mm. In such a vibration displacement region, reduction of the spring constant is demanded. To achieve the target handle vibration value, this development examined the shape on the premise of a mount rubber shape with many gaps, as shown in Fig. 5. Here, from the viewpoint of “(3) Actual machine own-weight support” described hereinafter in this chapter, there is a concern of large deformation and buckling of the mount rubber if the spring constant in the actual machine own-weight direction is too small. Therefore, in the vibration displacement region, the basic guideline was to reduce the spring constant of the swing direction displacement.

(2) Spring constant of swing displacement region

The region where the hole-shaped part in the center of the mount rubber is displaced by the input load of swing operation as shown in Fig. 3 was defined as the swing displacement region. Based on preliminary testing results, the expected displacement in this region when the mount rubber shape has many gaps, such as shown in Fig. 5, was set to around 3 mm or more. In such a swing displacement region, it is necessary to increase the spring constant to secure sufficient swing stiffness. To achieve the contrary

spring constant demands of both the vibration displacement and swing displacement regions, this development designed the mount rubber shape so that the spring constant in the swing displacement direction is nonlinear according to the displacement.

(3) Actual machine own-weight support (low vibration, durability)

The developed anti-vibration type brush cutter has a structure that supports the actual machine own-weight of approximately 10 kg with mount rubbers in four locations. The mount rubbers use a shape with many gaps to reduce vibration, so there is a concern that if the mount rubbers deform greatly or buckle due to the actual machine own-weight, the increased spring constant may lead to increased vibration and reduced durability. Therefore, from the viewpoint of durability and low vibration, consideration was given to avoid the occurrence of large deformation and buckling due to the actual machine own-weight.

3.1.3. Examination of mount rubber specifications

The specification examination processes for satisfying each design requirement of the mount rubber are described below.

(1) Spring constant of vibration displacement region

In examining the spring constant in the vibration displacement region, multiple levels of mount rubbers with different spring constants were prototyped in the initial development stage. Based on the correlation between the spring constant and the measured handle vibration value, the spring constant (spring constant in the swing direction: 5.2 N/mm) that achieved the target handle vibration value was set as the design target value.

Next, Computer Aided Engineering (CAE) analysis was used to conduct a mount rubber shape parameter study. Specifically, enforced displacement analysis was performed on various mount rubber shapes with different thickness distributions and numbers and arrangement of ribs, as shown in Fig. 7, and the spring constant of each

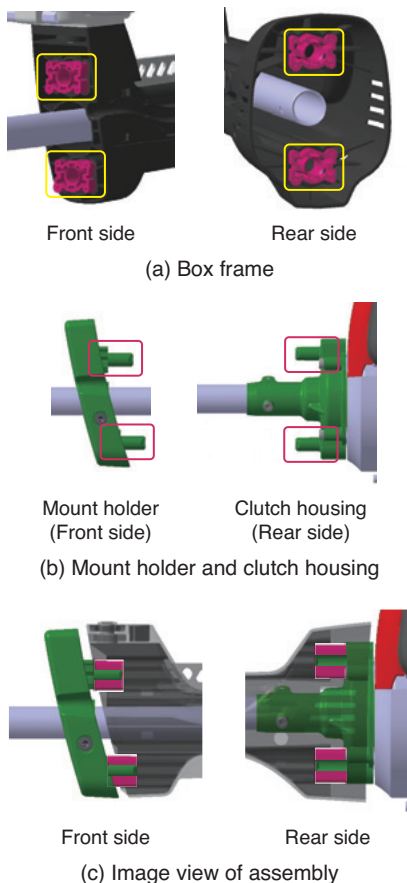


Fig. 6 Support structure of mount rubbers

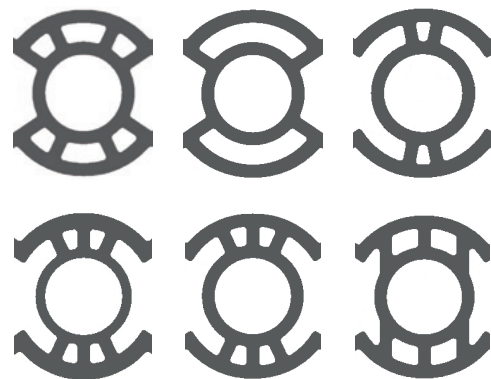


Fig. 7 Examples of mount rubber shape in parameter study

shape was calculated. Figure 8 shows an image of enforced displacement analysis. The two parts of a simple box frame shape simulating the mount rubber insertion part and the mount rubber were modeled. This model was used to calculate the spring constant by outputting the reaction force value when the hole-shaped part in the center of the mount rubber is forcibly displaced. Based on the tendencies understood through the parameter study, a shape was found that satisfies the target spring constant (Fig. 9).

(2) Spring constant of swing displacement region

Comparative verification was performed with a competing anti-vibration type brush cutter, and the spring constant target values were set to secure sufficient swing stiffness. To achieve the spring constant target values of both the vibration displacement and swing displacement regions, spring constant characteristics that are nonlinear relative to swing direction displacement, as shown in Fig. 10, were set as the design target. Note that the slope of

the graph corresponds to the spring constant.

A structure with protrusion shapes on both ends relative to swing direction displacement was adopted to satisfy the nonlinear spring constant characteristics shown in Fig. 10. CAE enforced displacement analysis was performed to adjust the protrusion shape. Figure 11 shows the protrusion shapes of the initial and final specifications, and Fig. 12 shows the CAE analysis results of the spring constants corresponding to each shape. The protrusion shape of the final specification achieved high spring characteristics with a spring constant (graph gradient) that is the target gradient or more in the swing displacement region, while also achieving low spring characteristics with a graph gradient that is the target gradient or less in the vibration displacement region.

(3) Actual machine own-weight support (low vibration, durability)

First, the shapes that realize low vibration in the state

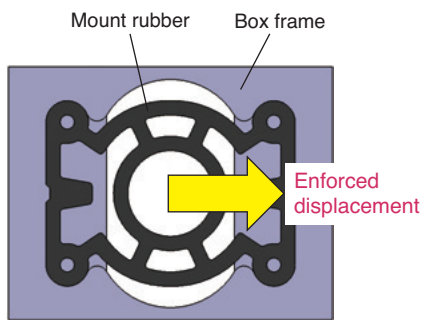


Fig. 8 Image view of enforced displacement CAE analysis

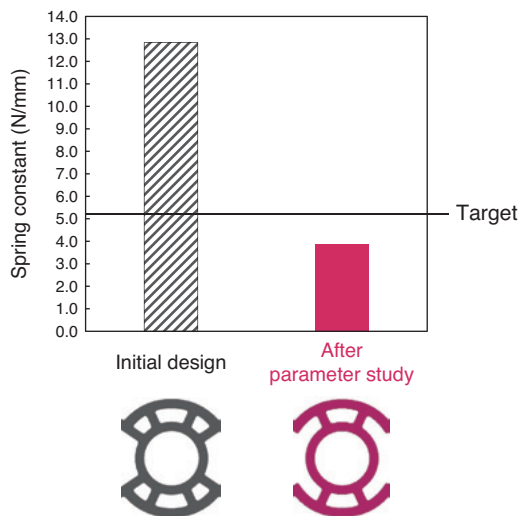


Fig. 9 CAE results of each mount-rubber spring constant

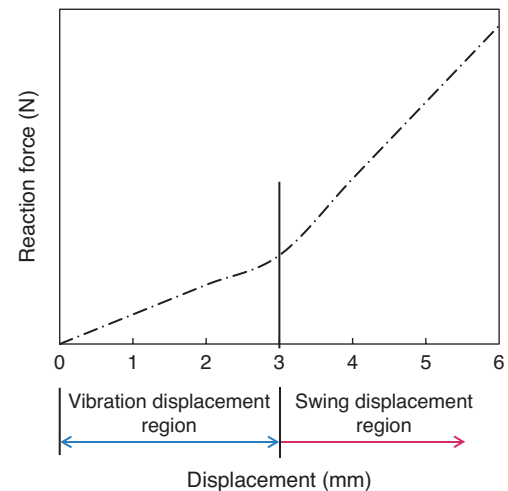


Fig. 10 Target nonlinear spring constant characteristic

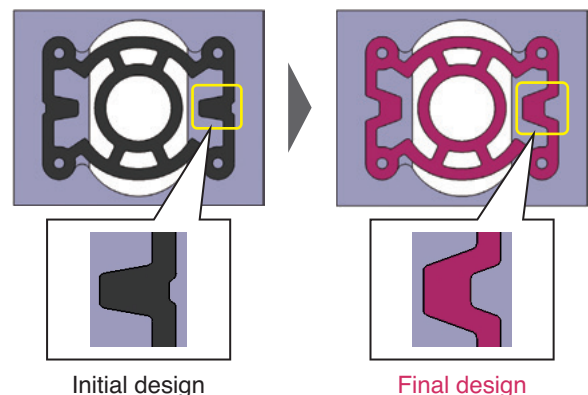


Fig. 11 Comparison of protrusion shapes

with the actual machine own-weight supported were examined. This development focused on the clearance at the bottom of the mount rubber insertion part, as shown in Fig. 13. In the state with the actual machine own-weight supported, the clearance setting affects the deformation state of the mount rubber, and the spring constants and handle vibration value change. Therefore, the clearance setting of the insertion part also becomes a key design element in addition to the mount rubber shape. This development conducted a parameter study of the clearance by CAE enforced displacement analysis and actual measurement of handle vibration, taking into account the displacement due to the actual machine own-weight. This CAE enforced displacement analysis modeled the two parts of a simple box frame shape simulating the mount rubber insertion part and the mount rubber as in Fig. 8. In this model, the reaction force value during enforced displacement of the hole-shaped part in the center of the mount rubber was output in two analysis steps as follows. In the first analysis step, enforced displacement was produced in the direction that the actual machine own-weight acts. The enforced displacement here referred to the mount rubber deformation state when mounted on an actual

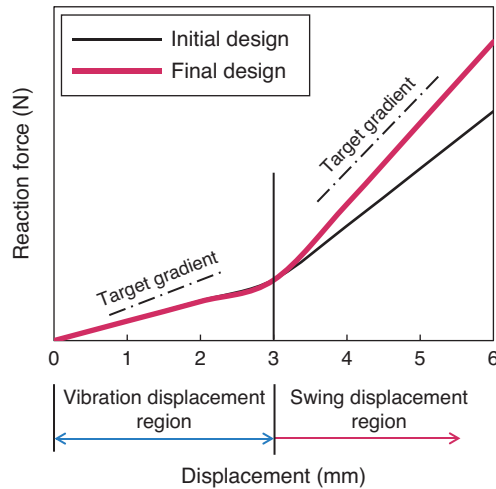


Fig. 12 Nonlinear spring constant characteristics

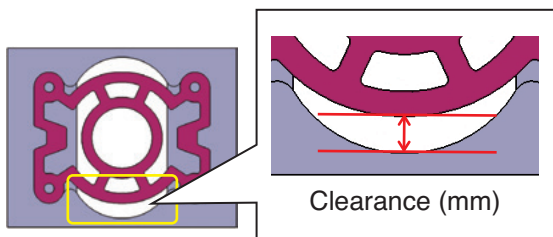
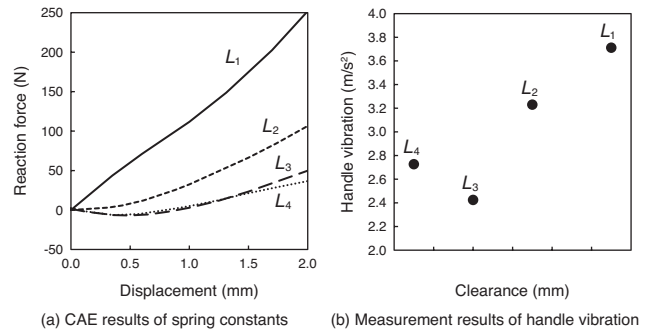


Fig. 13 Clearance at bottom of mount rubber

brush cutter, and was set to produce the same deformation state. In the next analysis step, enforced displacement was produced in the swing displacement direction as in Fig. 8. Figure 14(a) shows the results of enforced displacement analysis of each clearance. The displacement on the horizontal axis indicates the enforced displacement in the swing direction, and the gradient of the graph corresponds to the spring constant. In addition, Fig. 14(b) shows the actual measurement results of the handle vibration value for each clearance. Based on the results in Figs. 14(a) and (b), the clearance was set to the clearance (L_3 in the figure) at which both the spring constant of the vibration displacement region and the handle vibration value are the lowest.

Next, the reasonableness of the shape was confirmed from the viewpoint of durability. First, the mount rubber deformation state was visually confirmed in the condition with the actual machine own-weight supported. The results confirmed that at the set clearance, there was no site with large local elongation or self-contact between inner walls. In addition, durability tests on a test bench and during actual work under the condition of continuous engine operation, and durability tests in various assumed work modes were also conducted. These tests confirmed that there was no damage to the mount rubber.

Through the above examination process, the final mount rubber specification was found (Fig. 15).



Note: (a) L_1, L_2, L_3, L_4 are clearance values ($L_1 > L_2 > L_3 > L_4$)
(b) Clearance amount is 1.0 mm on scale

Fig. 14 Spring constants and handle vibration depending on clearance values

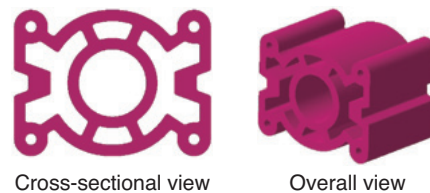


Fig. 15 Mount rubber of UMK450E XEET

3.2. Driveshaft Bearing Placement Technology

3.2.1. Basic drivetrain component structure

Figure 16 shows the basic drivetrain component structure. The rotational force of the engine is transmitted to the cutting head through the centrifugal clutch, driveshaft, and reduction gear. The driveshaft is supported by multiple rubber bearings positioned inside the main pipe. In addition to rubber bearings, resin liner materials are also sometimes used as bearing parts for typical brush cutters. The developed anti-vibration type brush cutter UMK450E XEET uses a support structure with multiple rubber bearings (hereinafter, bearings).

3.2.2. Dynamic damper concept

A new bearing placement design method was established to suppress resonance of the main pipe relative to the increased excitation force due to the unbalance amounts of the cutting head and engine flywheel. Although the number of bearing placements differs for each brush cutter, the bearings are generally placed at equal (or nearly equal) spans, as shown in Fig. 16⁽⁶⁾. By contrast, this development set a bearing placement with unequal spans and used the driveshaft as a dynamic damper^{(5), (7)} (Fig. 17). As shown in the figure, an area (the wide span area in the figure) with an extremely wide placement span between two adjacent bearings is provided. The first bending mode vibration of the driveshaft portion supported at both ends in this area is used as a dynamic damper. The natural frequency of the first bending mode vibration of the driveshaft can be set to any frequency by adjusting the placement span of the two bearings at both ends of the wide span area (the distance between #5 and #6 in the figure). This enables the generation of anti-resonance in the main pipe (mount holder attachment position) for any frequency. The transmission rate of resonance vibration to the handle is suppressed by generating anti-resonance at the mount holder attachment position.

As an applied design technique, the remaining bearing placement spans (#1 to #4 in the figure) can also be changed to add dynamic damper areas. This enables simultaneous suppression of resonance for multiple excitation frequencies.

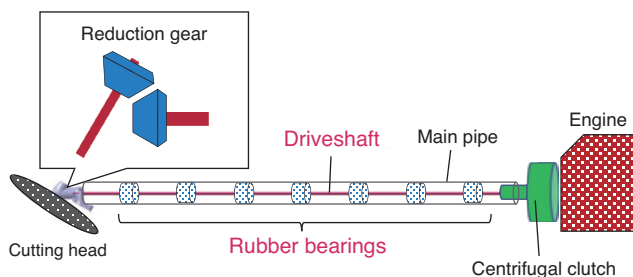


Fig. 16 Drivetrain component structure

This design method only changes the bearing placement, and does not call for installation of additional devices or changes to part shapes (e.g., changing the main pipe diameter, etc.). This enables implementation of resonance measures without impacting the total weight or swing stiffness, which are important from the viewpoint of merchantability.

3.2.3. Bearing placement of UMK450E XEET

This development set the bearing placement shown in Fig. 18. Dynamic damper areas were provided in two locations to simultaneously suppress resonance relative to the excitation force of both the cutting head and the engine, which have different frequency components due to the reduction gear. The target frequencies for resonance suppression were the respective frequencies of the first-order components of cutting head and engine rotation corresponding to racing speed. The natural frequencies of the two dynamic dampers were set by the respective bearing placement spans (between #2 and #3 and between #5 and #6 in the figure) according to these two target frequencies.

3.2.4. Confirmation of effects

The handle vibration for the bearing placement of the dynamic damper concept (Fig. 18) and the equal-span bearing placement (Fig. 16) were compared by measurement during actual operation. The handle vibration measurement method complied with ISO22867-2011⁽⁴⁾. The vibration acceleration of the left handle was measured when the engine speed was varied from idling to wide open throttle. Figure 19⁽⁷⁾ shows the results. Figure 19(a) and (b) show the respective handle vibration tracking data for the first-order components of cutting head and engine vibration. Note that the rotational speed ratio of the reduction gear between the cutting head and the engine is

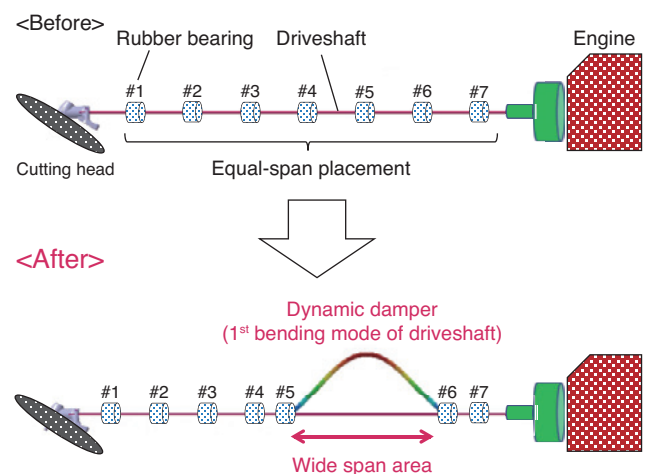


Fig. 17 Dynamic damper outline

0.77:1. As shown in the figure, the vibration levels at the target frequencies of the first-order components of both cutting head and engine vibration were suppressed by 72% and 58%, respectively, compared with the equal-span bearing placement.

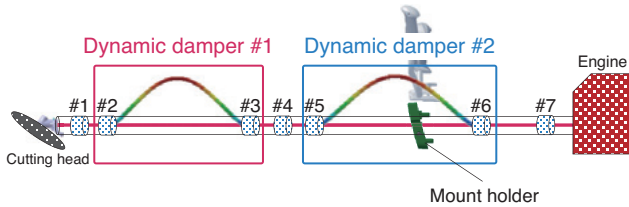
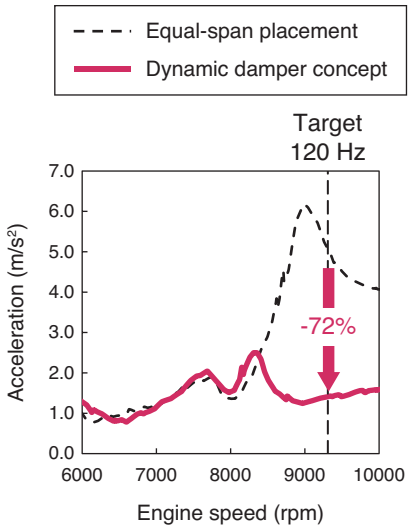
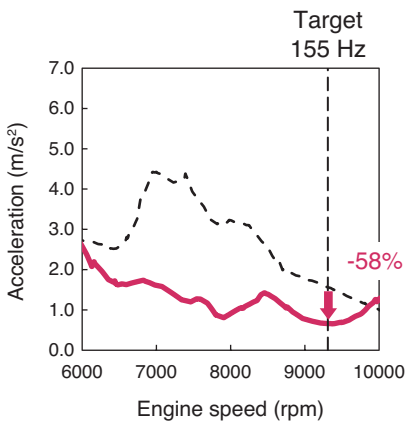


Fig. 18 Bearing placement of UMK450E XEET



(a) Vibration of first-order component of cutting head



(b) Vibration of first-order component of engine

Fig. 19 Measurement results of handle vibration

4. Confirmation of Degree of Achievement

The degree of achievement of the development targets (1) and (2) described in Section 2.2. is shown below.

4.1. Handle Vibration

Figure 20 shows the measurement results of handle vibration⁽⁸⁾. The handle vibration value on the vertical axis is the composite value of the handle vibration values at idling speed and racing speed as described in Section 2.1. A steel blade and a nylon blade, which are typical cutting heads, were used for measurement. Steel and nylon cutting heads with average unbalance amounts were mounted and the handle vibration was measured. Figure 20 confirmed that the target handle vibration value of less than 2.5 m/s² was achieved for both the left and right handles.

In addition, Fig. 21 shows the measurement results of handle vibration when the engine speed is varied. The vibration level was confirmed to be less than 2.5 m/s² over the entire engine speed range, and there were no resonance peaks that exceeded 2.5 m/s².

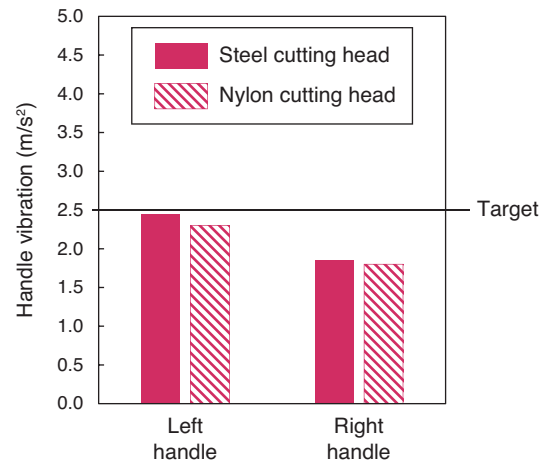


Fig. 20 Measurement results of handle vibration

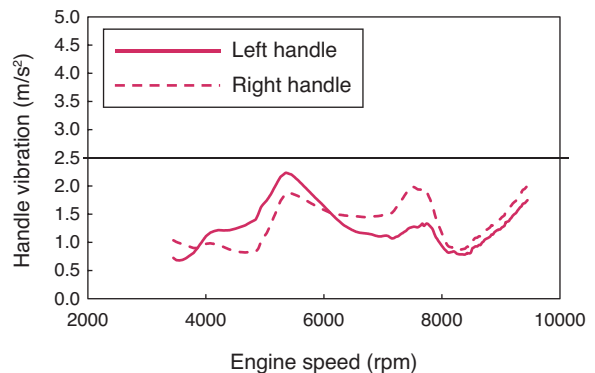


Fig. 21 Measurement result of handle vibration

4.2. Uncertainty of Handle Vibration Value

Figure 22 shows the measurement results of handle vibration when the respective static unbalance amounts of the cutting head and engine flywheel are changed in the bearing placement of the dynamic damper concept (Fig. 18). For reference, Fig. 22(a) also shows the measurement results for the equal-span bearing placement (Fig. 16). The handle vibration value on the vertical axis is the composite value of the handle vibration values at idling speed and racing speed. Figure 22(a) and (b) confirmed that the bearing placement of the dynamic damper concept suppresses the increase in the handle vibration value relative to increases in the respective static unbalance amounts of the cutting head and the engine flywheel. In addition, Fig. 22(a) confirmed that the change in bearing placement contributes to reducing the variation of the handle vibration value relative to the increase in the static unbalance amount.

Figure 23 shows the measurement results of the handle vibration value uncertainty⁽⁸⁾. The handle vibration is the

composite value of the handle vibration values at idling speed and racing speed. The uncertainty conforms to the contents described in Sections 2.1. and 2.2. This figure confirmed that the target uncertainty of less than 2.0 m/s^2 is achieved for both the left and right handles when equipped with either a steel or nylon cutting head.

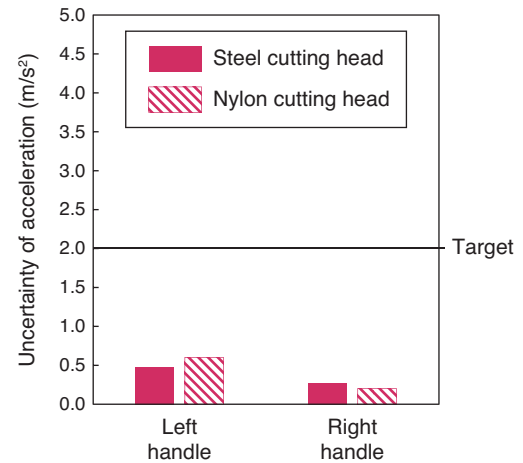


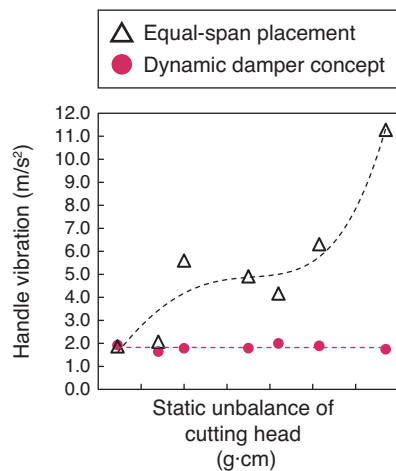
Fig. 23 Measuring uncertainty of handle vibration

5. Conclusion

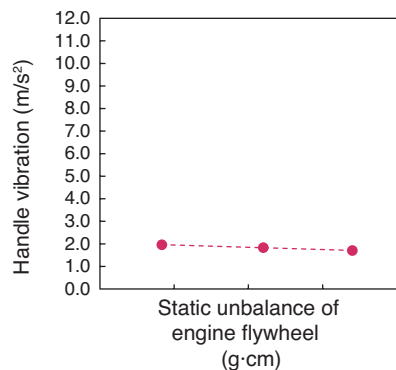
Vibration reduction technology focusing on the mount rubber shape and the driveshaft bearing placement was established to satisfy user demands in the European commercial market. The key points are described below.

- (1) A mount rubber shape with nonlinear spring constant characteristics relative to swing direction displacement was designed in order to both secure swing stiffness and reduce handle vibration. Effective use was made of a CAE parameter study in the design process to optimize the mount rubber rib shape and insertion part clearance. This achieved a handle vibration of 2.5 m/s^2 or less while supporting the actual machine own-weight.
- (2) A bearing placement design method was established that can suppress resonance at any frequency by using the driveshaft as a dynamic damper. This reduced the first-order component of cutting head vibration by 72% and the first-order component of engine vibration by 58% at the respective target frequencies compared to the equal-span bearing placement. It also suppressed the increase in handle vibration relative to increases in the respective static unbalance amounts of the cutting head and the engine flywheel, and achieved the target value for uncertainty of the handle vibration value of less than 2.0 m/s^2 .

These vibration reduction technologies realized a low vibration brush cutter that can satisfy the demands of target users in the European commercial market.



(a) Left handle vibration to static unbalance of nylon cutting head



(b) Left handle vibration to static unbalance of engine flywheel

Note: (a) Static unbalance amount is 2.0 g·cm on scale
(b) Static unbalance amount is 0.3 g·cm on scale

Fig. 22 Measurement results of handle vibration to static unbalance amount

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Development of General Purpose Drive Module and Drive Platform

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ABSTRACT

A general purpose drive module that consolidates the functions needed for driving, such as drive, steering, and impact absorption, and a drive platform equipped with a cooperative control function that governs the driving functions and is easily connectable to an autonomous drive function were developed. This paper describes the configuration of the drive module, the system configuration of a drive platform consisting of four drive modules, and the steerability achieved by cooperative control. An autonomous delivery robot based on this drive platform was developed with Rakuten Group, Inc. and the effectiveness of the drive platform was confirmed in driving demonstration tests conducted on the University of Tsukuba campus with the support of the New Energy and Industrial Technology Development Organization. Use of this general purpose drive module and cooperative control technology enables the efficient creation of various forms of electric mobility.

1. Introduction

Labor shortages due to a declining birthrate and aging population and response to the COVID-19 pandemic are increasingly becoming social issues. As a result and against the backdrop of technological progress, society is starting to see the implementation of autonomous electric mobility that provides services and performs work in place of people, such as delivery, guidance, cleaning, and security, and the field of electric mobility is experiencing remarkable changes.

Honda has developed various electric mobility products that are useful in daily life, such as the four wheel scooter Monpal⁽¹⁾ and the robotic lawnmower Miimo⁽²⁾. Various initiatives have also been pursued with the aim of further enhancing customer lifestyles, such as development of the new four wheel scooter WANDER WALKER CONCEPT⁽³⁾ and an airport guide robot that drives autonomously⁽⁴⁾. However, the timely development of diverse electric mobility is considered necessary in order to support these remarkable changes.

Therefore, this paper introduces a general purpose drive module that consolidates the functions necessary for mobility driving into a single package (Fig. 1), control technology, and a drive platform that is extendable into diverse formats (Fig. 2).

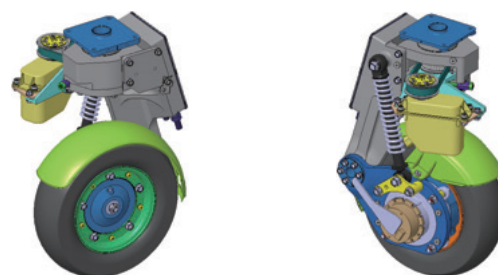


Fig. 1 General purpose drive module

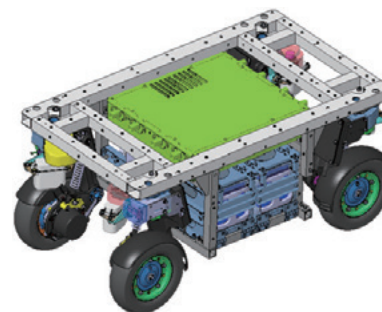


Fig. 2 Drive platform

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2. Concept and Configuration

2.1. Concept

An example of autonomous electric mobility is an autonomous delivery robot. When designing such a robot, first the requirements are defined such as the driving environment and the amount and size of the load. However, the requirements are often undetermined in the initial stage of development, and it is conceivable to go through a process of reconsidering the requirements and reviewing the basic design of the robot. For example, an increase from the initially required load capacity may result in insufficient load capacity, and a change in the robot size can be expected to result in a change in specifications such as wheelbase and tread. The driving environment requirements may also change, resulting in demands for greater driving force or small turn performance than expected, and it may be necessary to respond to new needs not anticipated in the initial stage, such as the addition of monitoring functions.

For these reasons, it is considered important that development of new types of mobility be able to respond immediately to specification changes and extension of functions. In particular, specification changes have a large impact on the drive and steering parts, and it is anticipated that separate design for each application would call for many development resources.

These circumstances led to the concept of a general purpose drive module that consolidates the driving functions such as drive and steering into a single package and a drive platform that performs integrated control of the drive modules. Modularization of the driving functions increases the wheelbase and tread setting flexibility, and different numbers of modules can be used to vary the driving performance. In addition, the drive platform has extendability, which enables the addition of various work and autonomous drive functions. This drive module and drive platform will form a base for various forms of autonomous electric mobility.

2.2. Configuration of General Purpose Drive Module

The general purpose drive module consolidates the functional parts (tires, drive motor, steering motor, etc.) needed for driving, and is operated by providing a power source and input signals. The operating angle of the steering motor is set to $\pm 180^\circ$ in order to obtain any steering angle. The combination of at least a single drive module with driven wheels enables the creation of mobility that can move in any direction, and functionality can be enhanced by increasing the number of drive modules.

For example, mobility that can move freely including turn on a spot and horizontal move is achieved by placing two drive modules between four free driven wheels such as swivel casters or omni wheels [Fig. 3(a)]. In addition, use of four drive modules achieves mobility with both rough

road driving performance and maneuverability [Fig. 3(b)].

2.3. Drive Platform Configuration

The drive platform configuration is explained here using the example of the contents investigated for the autonomous delivery robot described hereinafter. In order to complete the work tasks, autonomous mobility should have a wide drivable range and should be able to climb over steps and drive even on narrow and rough roads, etc. Therefore, the configuration has four drive modules located on the vehicle body, and uses four wheel independent drive and four wheel independent steering.

Four wheel independent drive is effective for driving on slopes and uneven ground and for climbing up steps, and four wheel independent steering enables turning on a spot and horizontal movement, and provides the ability to pass through narrow roads, avoid obstacles, and store the vehicle in the manner of parallel parking.

In this research, the size was set to a width of 700 mm, length of 1200 mm or less, and weight of 200 kg or less, which is equivalent to a four wheel scooter such as Monpal. It is thought that an autonomous delivery robot with these dimensions will be able to drive without greatly obstructing the passage of people even in walking spaces, and move indoors, including elevators.

The drive modules are placed in the four corners of the main frame to take advantage of packaging merits. The power source system and electrical components are consolidated between the drive modules, and all functional parts needed for driving are housed below the main frame. The sensors for autonomous drive and a delivery box are mounted above the main frame.

The main frame has a structure that enables easy replacement according to the application, so changes in the mobility size and format can be supported. For example, extension to security robots with monitor functions equipped on top of the main frame and autonomous work robots with working parts such as cleaning brushes or lawnmower cutter decks installed on the front or rear of the vehicle body can be expected (Fig. 4).

The drive platform system consists of a drive function and a cooperative control function and safety function that govern the drive function. Commands are given to the drive platform system from a separately configured autonomous drive system (Fig. 5).

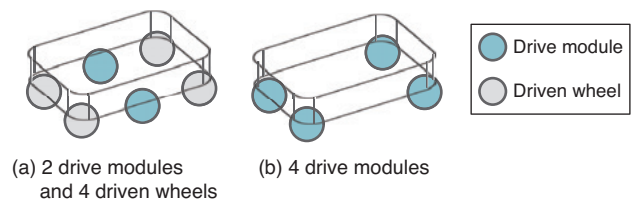


Fig. 3 Example of module layout variations

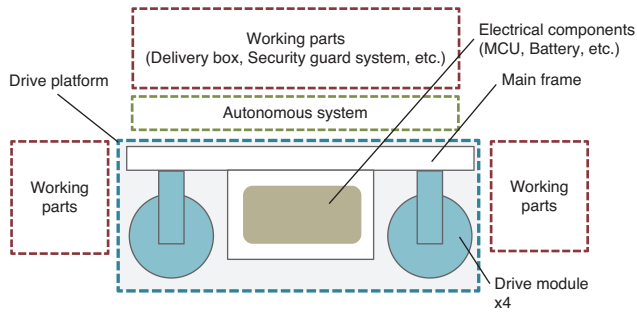


Fig. 4 Layout concept of drive platform

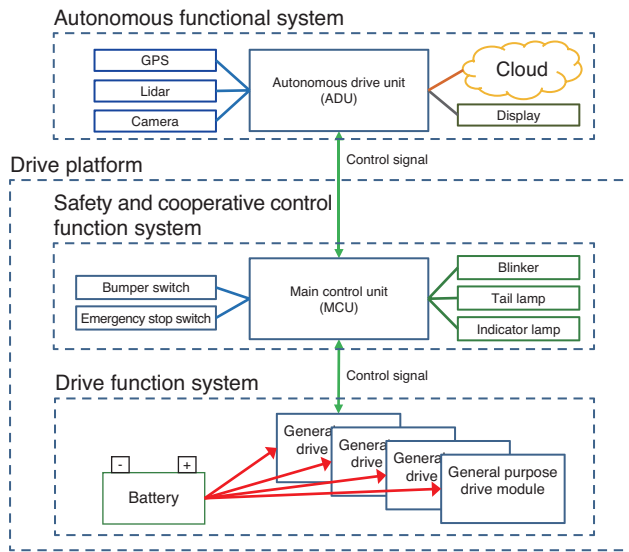


Fig. 5 Control outline diagram

3. Design and Features

3.1. General Purpose Drive Module

Table 1 shows the specifications and Fig. 6 shows the parts layout of the general purpose drive module used to realize the above-mentioned concept.

The general purpose drive module consists of a (e) tire,

Table 1 Specifications of drive module

Drive motor rated output	150 W
Steering motor maximum torque	38 N·m
Tire (Diameter x Width)	280 x 80 mm
Suspension	Bottom link
Wheel stroke	50 mm
Parking brake	Power off brake

a (f) drive motor, a (m) power off brake for stop holding, a (l) swing arm that fixes those parts, a (k) shock absorber for impact absorption and enhancement of tire contact patch characteristics, a (j) steering motor, a (h) belt and (i) pulley for power transmission, a (b) spindle with fixing flange for fixing the module to the vehicle body, and a (c) frame that supports all of the aforementioned parts.

The (e) tire is a urethane-filled airless tire with an outer diameter of 280 mm and a width of 80 mm in consideration of rough road driving performance and maintainability. The (f) drive motor is an in-wheel type three-phase brushless motor with a built-in reduction gear and rated output of 150 W. A (m) power off brake with a release lever is installed to the motor axis as a parking brake.

The (e) tire, (f) drive motor, and (m) power off brake are fixed to the (l) swing arm and installed together with the (k) shock absorber to the (c) frame. The wheel stroke is set to 50 mm. The suspension format is a bottom link type suspension, and the (k) shock absorber expands and contracts according to the drive and braking reaction force. The objective as described hereinafter was to use this mechanism to suppress changes in vehicle body attitude when accelerating and decelerating.

In order to simplify installation to the vehicle body, the mechanical connection to the vehicle body uses a structure that can fix the (b) spindle with fixing flange provided at the top of the module using only bolts. The (o) driven pulley is fixed to this (b) spindle with fixing flange, which serves as the (a) steering axis (Fig. 7). The (i) drive pulley is installed to the (j) steering motor fixed to the (c) frame side, and the motor torque is transmitted to the (o) driven

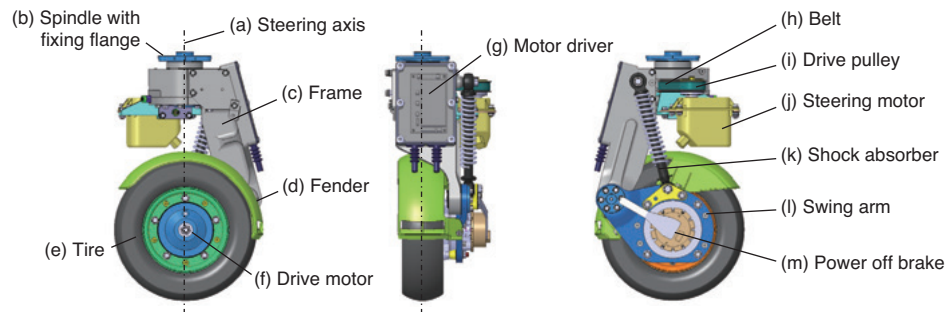


Fig. 6 Design of general purpose drive module

pulley by the (h) belt. As a result, the entire module rotates around this (a) steering axis.

In order to realize compact packaging, the (a) steering axis passes through the longitudinal and lateral center of the (e) tire to reduce the rotation path. This eliminates the caster trail and does not generate self-aligning torque, enabling suppression of the steering force.

The (a) steering axis is hollow and the wire harness of the motor is led through it to the vehicle body. The motor driver for the (g) drive motor is installed to the (c) frame and is operated by providing a power source and signals.

As for the layout of each component, in order to realize compactness and ease of installation, the steering motor mounting structure and the layout of the suspension pivot axis were arranged so that all the components are contained in a cylinder with a diameter of approximately 300 mm, which is nearly equivalent to the tire outer diameter. The drive, steering, impact absorption, and stop holding functions are consolidated in the cylinders as shown in Fig. 8, providing good space efficiency. Furthermore, when developing mobility using this module, designers can basically arrange the frame and other items while leaving this cylindrical space, which enables efficient design promotion.

In order to adjust the toe angle of the tire after the module is installed to the vehicle body, electrical adjustment by steering motor position control is used and a (p) link that enables mechanical adjustment is also provided in the steering motor fixing part in consideration of maintainability.

3.2. Drive Platform

3.2.1. Configuration

Table 2 shows the main specifications and Fig. 9 shows the appearance of the drive platform. The size corresponds to a four wheel scooter such as Monpal, which assumes use in human walking spaces.

As the drive platform combines rough road driving performance and maneuverability, four general purpose drive modules are mounted and four wheel independent drive and four wheel independent steering are used. The maximum speed is 6 km/h in consideration of movement

Table 2 Specifications of drive platform

Length x Width x Height	1180 x 620 x 530 mm
Rated output	600 W
Maximum speed	6 km/h
Minimum turning radius	700 mm
Step height which can be climbed over	90 mm

in walking spaces, and the drive platform can climb up steps as high as 90 mm without a run-up. Each drive module can steer independently in an area of $\pm 180^\circ$, which

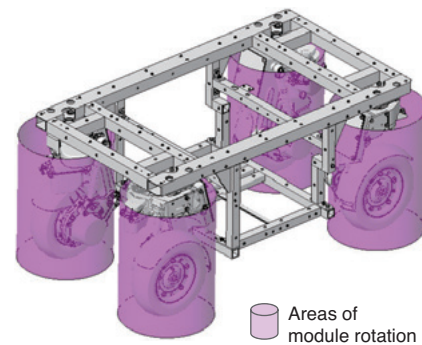


Fig. 8 Areas of module rotation

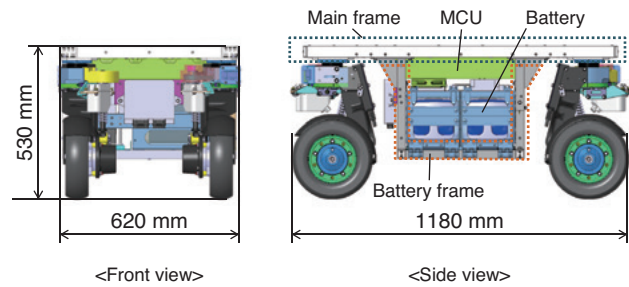


Fig. 9 Layout of drive platform

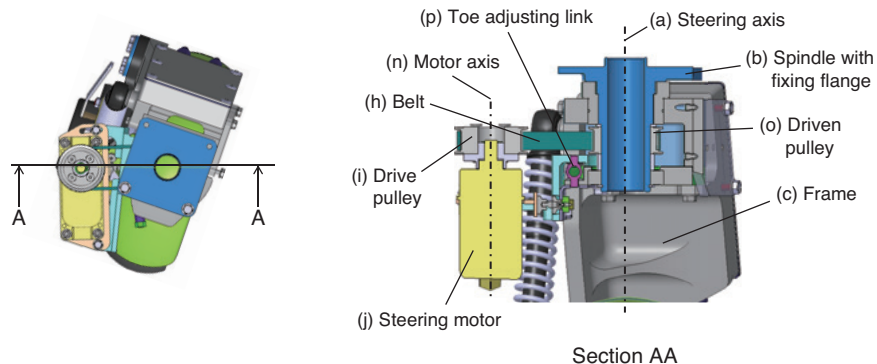


Fig. 7 Cross section of steering axis

enables turning on a spot with a turning radius of 700 mm [Fig. 10(a)] and parallel move including directly sideways [Fig. 10(b)] using the cooperative control function described hereinafter.

Drive modules were placed in the four corners of the main frame to make efficient use of space. In addition, the pivot axis of the swing arm is positioned toward the vehicle body center for the anti dive mechanism described hereinafter. The functional parts needed for driving are consolidated in the drive module, which enables securing a wide space for the batteries, MCU, sensors, etc. Two main batteries are placed sideways below the vehicle body center to secure separate power sources for driving and the autonomous drive function. This consolidated the power source system components including the sub battery below the vehicle body, which lowered the center of gravity.

The power source system components are housed in a box-shaped battery frame and fixed to the main frame. This box-shaped frame forms an independent power source module that can also be placed above the main frame, making it easy to change the drive platform format. The driving system MCU is positioned in the space between the power source system and the main frame. No parts protrude above the main frame and the upper surface of the vehicle body is flat. This provides extendability with a structure that enables easy installation of sensors for autonomous drive and working parts such as a cargo compartment. The main frame also has a replaceable unit structure, so expansion of the wheelbase and tread can be supported.

3.2.2. Anti dive mechanism

In a bottom link type suspension, the shock absorber expands and contracts according to the drive and braking reaction force from the road surface. This mechanism was used with the objective of suppressing longitudinal vehicle body inclination when accelerating and decelerating. Figure 11 shows a schematic of the forces acting on the vehicle body when braking. When the drive motors generate braking torque, the tires receive the braking reaction force from the road surface. This braking reaction force slows down the vehicle body and at the same time tries to rotate

the swing arm around the pivot axis. In the drive modules on the front side, this becomes a force that presses the tires against the road surface, and that reaction force extends the shock absorbers, becoming a force that pushes up the front of the vehicle body. In the drive modules on the rear side, the braking reaction force contracts the shock absorbers such that the rear of the vehicle body sinks. This movement mitigates nose dive due to inertial force, which helps to stabilize the vehicle body attitude. The opposite movement results during acceleration, which likewise helps to stabilize the vehicle body attitude.

The spring rate and pivot axis position of the drive module, the mounting interval of each module, and other parameters were set while confirming how the attitude changes using multibody dynamics simulation software (Adams). Figure 12 shows the analysis models. Model (A) is a bottom link type suspension, and Model (B) is a model without links in which it is imagined that the shock absorbers are installed directly to the tires. The two models differ only in the suspension format, and the wheel rate, weight, center of gravity position, and other parameters are the same.

Figure 13 shows the analysis result of the vehicle body pitch angle when braking force is applied while driving at constant speed on flat land. The horizontal axis plots the time and the vertical axis plots the pitch angle with the nose dive direction as negative. Model (A) exhibits a smaller pitch angle than Model (B), which shows that the desired effect is demonstrated.

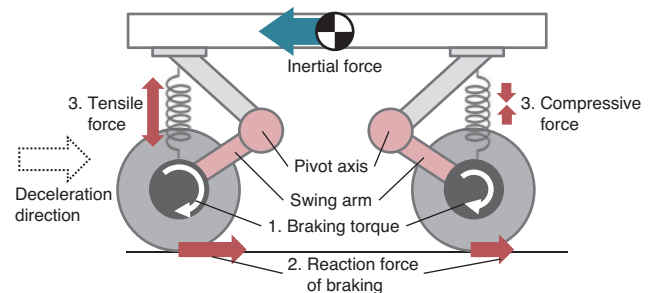


Fig. 11 Force acting on vehicle during deceleration

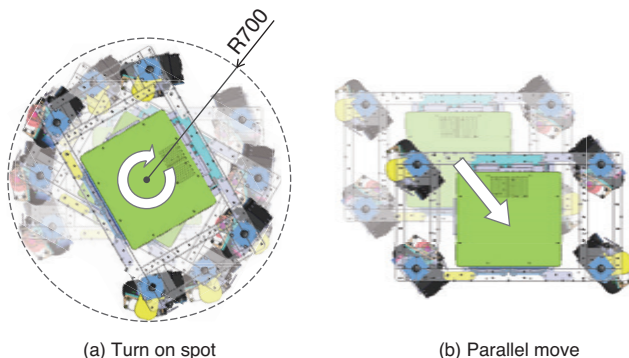


Fig. 10 Maneuverability of drive platform

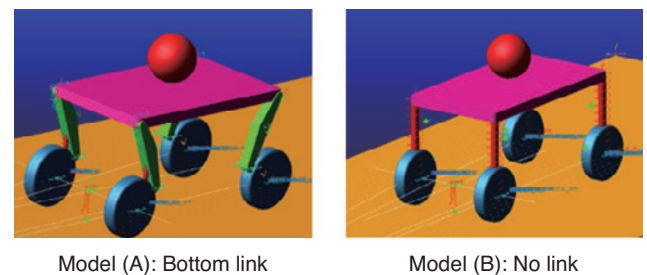


Fig. 12 Simulation models of Adams

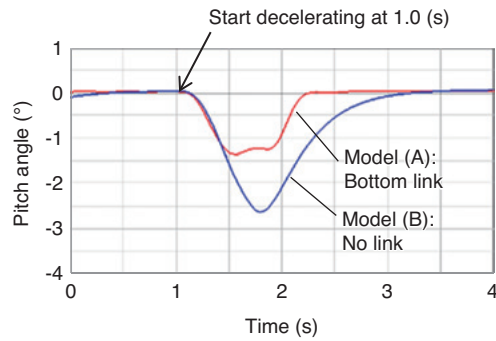


Fig. 13 Simulation result of pitch angle during braking

4. System Configuration and Features

Figure 14 shows the block diagram of the general purpose drive module.

The drive module is equipped with only the drive function and steering function and does not handle complex control, so it has only the necessary functions. The drive function and steering function are independent, and each can be operated separately by supplying power and providing commands.

Figure 15 shows the system block diagram of the

drive platform using this drive module. The drive platform consists of drive modules, an MCU that governs the drive modules, and a power source. Mobility capable of autonomous drive can be configured by providing commands from the ADU with autonomous drive function to the drive platform.

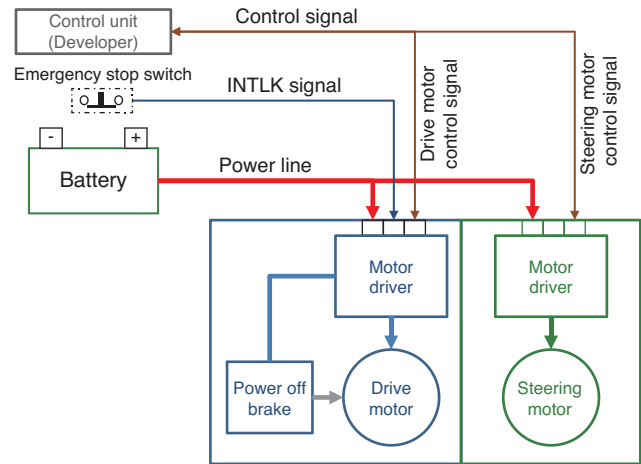


Fig. 14 Block diagram of general purpose drive module

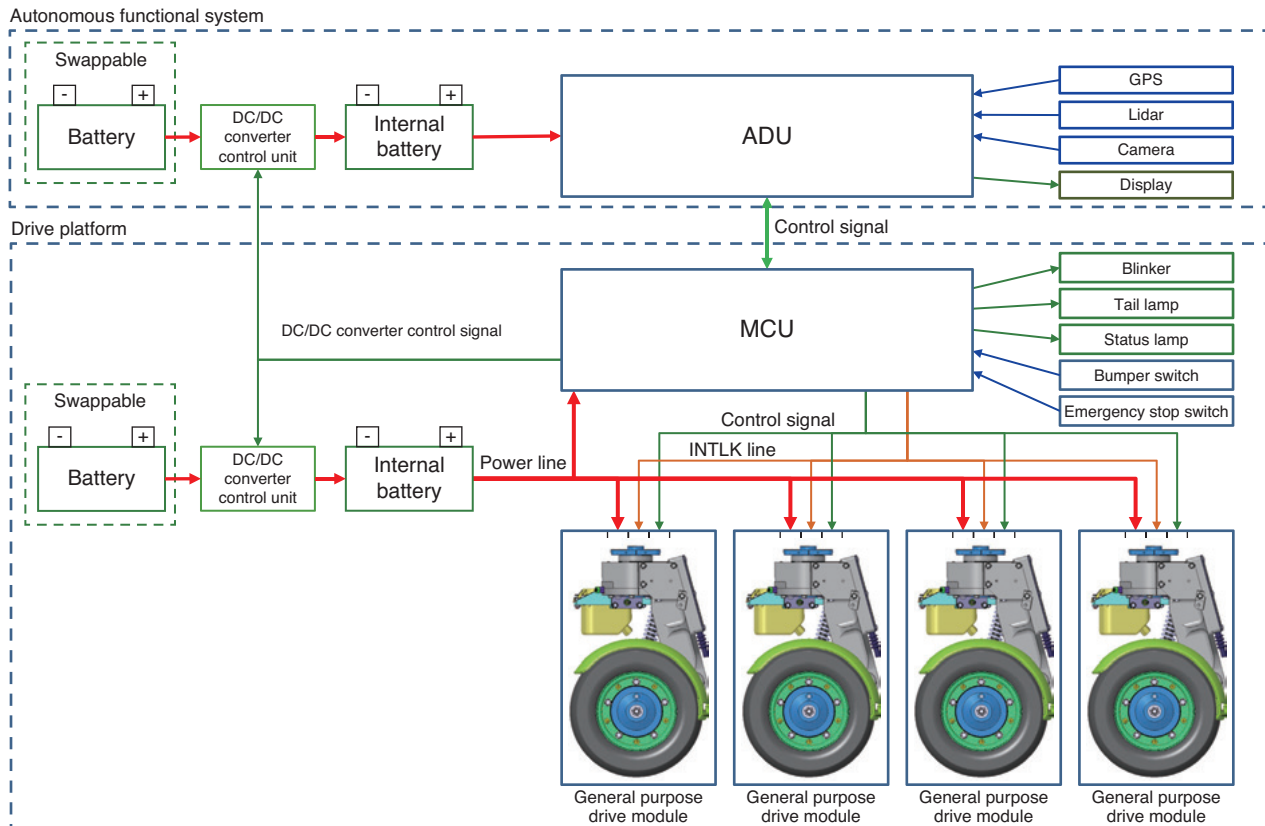


Fig. 15 Block diagram of drive platform

4.1. General Purpose Drive Module

4.1.1. Drive function

The drive function consists of the three blocks of the motor used for drive, the motor driver, and the power off brake. The power source is connected to the motor driver, and the motor speed is controlled simply by indicating the rotation speed via signal communication from the control unit. The motor driver autonomously turns the power off brake ON and OFF. The drive status is fed back to the control unit side via signal communication.

Apart from drive commands by signal communication, there is also an interlock function that stops the drive function, and the drive motor can be stopped by activating that interlock function. Table 3 shows the specifications of the drive function.

Table 3 Specifications of drive function

Rated output	150 W
Maximum continuous speed	182 rpm
Interface	CAN
Control method	Rotation speed control

Table 4 Specifications of steering function

Steering angle	$\pm 180^\circ$
Steering speed	$90^\circ/\text{s}$
Interface	PPM/CAN

4.1.2. Steering function

The steering function consists of a motor that provides steering force and a motor driver, and the steering function of the drive module can be operated by connecting a power source and providing steering commands. The area of the steering angle is set to $\pm 180^\circ$. The unit structure enables rotation in excess of 180° , but the wire harness of the power source and signals is connected through the hollow shaft of the steering axis, so the steering angle is limited to protect the wire harness.

The steering status is fed back to the controller side via signal communication in the same manner as the drive function. Table 4 shows the specifications of the steering function.

4.2. Drive Platform

The system configuration of the drive platform consists of the drive modules described above, the MCU, and a power source.

The MCU can configure mobility capable of autonomous drive by receiving commands from the ADU with autonomous drive function, converting those commands into commands to the drive modules, and providing the commands to each drive module.

4.2.1. MCU monitor function

The MCU monitors the drive platform status and the operating status of each drive module. It also handles operations such as the boot sequence after power on, and monitors the power source status.

The MCU also performs signal communication with the ADU, sends the monitored contents to the ADU, and receives commands to the drive platform from the ADU. The MCU constantly monitors the contents and transmission cycle of this signal communication, and stops the drive platform when an irregularity occurs.

The drive platform is equipped with an emergency stop switch for stopping safely and bumper switches around the entire circumference of the vehicle body for stopping the vehicle when physical contact occurs. The MCU monitors

these switches and stops the vehicle in the event of switch activation.

4.2.2. MCU cooperative control function

The MCU converts driving commands sent from the ADU to the drive platform into commands for arbitrarily placed drive modules based on the Ackermann-Jeantaud theory. Figure 16 shows schematic charts related to this conversion for the case with four drive modules located on the vehicle body.

The charts show top view images of the drive platform. The X-axis and Y-axis are respectively the longitudinal and lateral directions of the drive platform and the XY intersection is the center of the drive platform. The imaginary control criteria A is an image of an imaginary tire on the X coordinates, and B, C, D, and E are images of the drive module tires.

Figure 16(a) shows the steering image of each tire for a normal turn. The ADU sends the steering angle of the imaginary tire at A to the MCU as a steering command. The MCU calculates the turning center point shown in the figure based on the steering command at A from the ADU, and then calculates the steering angle and the indicated rotational speed to the drive motor, which is proportional to the distance from the turning center, for each drive module. By indicating these calculated values to each drive module, the drive platform turns.

Figure 16(b) shows the image for turn on a spot. By steering A by 90° relative to the X-axis, the turning center becomes the drive platform center.

Figure 16(c) shows the image for parallel move. Parallel move is performed by facing each drive module in the same direction.

These operations can also be combined to steer during parallel move, and the drive platform can be operated freely by having the MCU perform the calculations at that time.

By having the MCU handle the commands to each drive module in this manner, the ADU can freely operate the drive platform without the need to take into account the number and placement of drive modules.

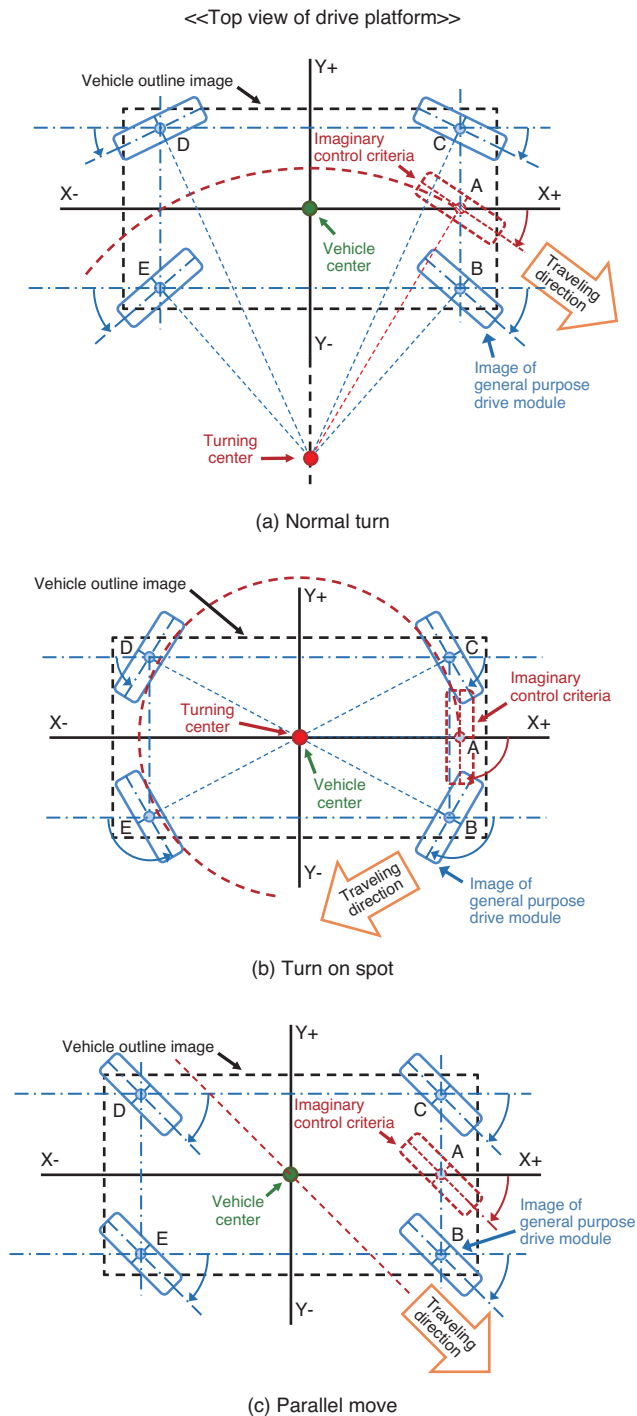


Fig. 16 Schematic chart of cooperative control

5. Confirmation of Effects in Driving Demonstration Tests

5.1. Autonomous Delivery Robot

Figure 17 shows the autonomous delivery robot developed with Rakuten Group, Inc. Honda is in charge of developing the vehicle body portion with autonomous drive

function, which is based on the above-mentioned drive platform, other than the delivery box portion of the robot.

Cameras, ranging sensors, monitors, and other devices for autonomous drive are installed on the front and rear of the drive platform. An emergency stop switch is located on top of the vehicle body, bumper switches are installed around the entire circumference of the vehicle body, and the robot stops when any of these switches is activated.

The exterior is box-shaped with rounded corners, and items such as tail lamps, blinkers, status lamps, and battery indicators are arranged in consideration of visibility and balance. In addition, the areas around the two front cameras were made into circles and ring-shaped reflective material was attached to provide a friendlier expression. The objective here was to not present an intimidating feeling when driving in a walking space with people, and to make it easy to understand the front and back of the robot.

A delivery box developed by Rakuten was fixed to the flat surface on top of the main frame of the drive platform. As stated in the above-mentioned design concept, there was no need for large-scale modification or complex coordination with Rakuten in order to mount the delivery box, and the autonomous delivery robot could be created within a short time.

5.2. Confirmation of Effects in an Actual Driving Environment

Honda and Rakuten, with the support of New Energy and Industrial Technology Development Organization (NEDO), conducted driving demonstration tests of the above-mentioned jointly developed autonomous delivery robot on the University of Tsukuba campus and some public roads from July 19 to August 31, 2021.

The driving route on the university campus included locations with braille blocks and rough pavement, but the rough road driving ability provided by four wheel drive and the suspension enabled stable driving (Fig. 18). In the course of the tests it was also necessary to climb over curbs approximately 40 mm in height between paved and unpaved road, but the robot did not become stuck there.

It was necessary to emergency stop the robot a number of times while driving during the tests in order to secure safety. However, the anti dive mechanism enabled stopping in a stable manner without disrupting the attitude.

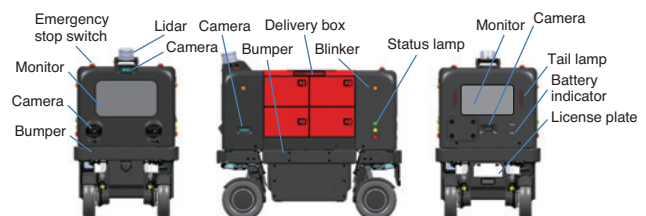


Fig. 17 Autonomous delivery robot developed with Rakuten Group, Inc.

As the robot was able to turn on a spot (turning radius approximately 700 mm), advanced control to turn back was not needed. When changing direction on a road approximately 2.6 m wide (Fig. 19), bicycles parked on the edge of the road and pedestrians coming and going limited the space actually available for turning to only about 2 m, so turning on a spot was effective. There were also an elevator and narrow corners on the path to the indoor robot storage location, but the size equivalent to a four wheel scooter and four wheel independent steering provided good small turnability, which enabled putting the robot in and out smoothly.

In this manner, the autonomous delivery robot could be efficiently operated in an actual driving environment that included uneven and narrow roads, which confirmed the effectiveness of the drive platform.



Fig. 18 Driving environment on University of Tsukuba campus



Fig. 19 Turning on spot on narrow road (Minimum turning radius: approximately 700 mm)

6. Conclusion

A general purpose drive module that consolidates the functions needed for driving, such as drive, steering, and impact absorption, and a drive platform equipped with a cooperative control function that governs the driving

functions and is easily connectable to an autonomous drive function were developed.

This development of autonomous delivery robot used four drive modules to configure a drive platform that combines rough road driving performance and maneuverability. The drive platform has space for adding functionality on the top surface and the front and rear, and has the extendability to create various forms of electric mobility.

An autonomous delivery robot based on this drive platform was developed with Rakuten, and driving demonstration tests were conducted at the University of Tsukuba with the support of NEDO. These tests confirmed the effectiveness of the extendability of the drive platform, the connectivity with the autonomous drive function, and the rough road driving performance and maneuverability.

It is thought that use of the newly developed general purpose drive module and cooperative control technology will enable the efficient creation of various forms of electric mobility, including not just autonomous delivery robots but also autonomous lawnmower robots and autonomous cleaning robots. It is hoped that the timely development of diverse and practical autonomous products will be useful and expand joy in people's lives.

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High Loading Si Anode with 3D Collector and Inorganic Binder

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ABSTRACT

The application of three-dimensional collector and inorganic binder for silicon anodes was investigated with a view to raising the energy density of lithium-ion secondary batteries to a high level. In connection with testing, simulations with compact cells were conducted and confirmed the advantage over conventional electrodes in terms of energy density.

The effectiveness of inorganic binder was confirmed using electrodes (density 0.05 g/cm³) fabricated with a loading mass of about 6.0 mg/cm². It was found that the capacity retention of items without inorganic binder diminished to 25% or less up to the second cycle of the aging process, but the capacity retention of items that included inorganic binder was 78% at 50 cycles. Next, electrode pressing was carried out in order to raise the energy density to a high level. The electrodes obtained in this way (density 0.17 g/cm³ and 0.59 g/cm³) were used to perform the same kind of cycle testing, with the result that while the 0.17 g/cm³ item had the same capacity retention as the unpressed item, the capacity retention of the 0.59 g/cm³ item diminished to 46%. The results from cross-sectional observation of the electrode after testing showed that while the expansion and shrinking of the electrode had been successfully suppressed as anticipated, there were cracks between the interfaces of the composite layer and the collector foil, and the amount of that cracking was confirmed to be greater in the 0.59 g/cm³ item.

1. Introduction

As initiatives involving the government and private sector go forward toward a carbon-free society for the purpose of realizing a sustainable society, Honda is moving ahead with the electrification of power products, motorcycles, and automobiles⁽¹⁾.

Various kinds of secondary batteries can be found on the market, and for automobiles, in particular, lithium-ion secondary batteries (LIB) are widely used for their energy density. Compared to the energy density of gasoline (up to 12000 Wh/kg), however, LIB energy density reaches about 200 Wh/kg at most, so electric automobiles with these cells installed face the issue of a cruising range that is shorter than that of gasoline-powered vehicles^{(2), (3)}. One way to address this issue is to raise the LIB energy density to a

high level. There is energy density that is per unit volume, which is volumetric energy density (Wh/L), and specific energy (Wh/kg), which is energy density per unit weight. When discussing the energy density of the cells that are the subject here, the term used will be volumetric energy density, which is crucial in terms of installation in vehicles.

The volumetric energy density of a battery can be expressed, for example, by the following equation:

$$\text{Volumetric energy density (Wh/L)} = \text{Mean drive voltage (V)} \times \text{Active material capacity density (Ah/g)} \times \text{Electrode density (g/L)}$$

Of these terms, electrode density can be broken down into the amount of active material that exists per unit area, expressed by the loading mass (mg/cm²), and the thickness (μm). In other words, an effective way to raise the energy

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density to a high level is to raise the mean drive voltage, the capacity density, and the electrode density to a high level (realizing a high load of active material and a thinner electrode film).

Up to now, it has been confirmed that applying a three-dimensional mesh structure (hereinafter 3D collector) to the electrode (hereinafter 3D electrode) to increase its loading mass can increase the energy density of the cell by approximately 10%, and that even with the decrease in cycle performance and the increase in internal resistance that can be an issue when increasing the loading mass, this is more advantageous than with conventional electrodes. It has been made clear that this technology is effective for raising the energy density to a high level^{(4), (5)}. The application of high capacity active material to this 3D electrode technology can be expected to further enhance energy density.

An investigation was therefore made into the application to a 3D collector of silicon (Si), which was being studied for a next-generation high capacity anode.

The Si under discussion here for application as active material for the anode has a theoretical capacity density of 4200 mAh/g, and the effective capacity has been confirmed up to about 3600 mAh/g⁽⁶⁾, making this an active material with a capacity approximately 10 times that of the 372 mAh/g of graphite, which is commonly applied to the LIB anode. On the other hand, this same active material expands up to four times in volume during charging and discharging, which means that with each occurrence of another cycle, the expansion and shrinking are repeated so that the composite material that includes the Si active material slips off from the collector foil. Thus the extremely low cycle durability presents issues⁽⁷⁾.

2. Electrode Design Guidelines

Figures 1 and 2 show cross-section scanning electron microscope (SEM) images of the Si electrode with loading mass of 3.0 mg/cm² before and after aging. After aging, it can be confirmed that the expansion of the active material has expanded the electrode thickness by a factor of 1.8, from 25 μ m to 45 μ m. Collapse of the electrode can also be confirmed, and it is apparent that electron transfer has been disrupted. To assure the cycle durability of Si-related electrodes, therefore, it is necessary to reduce the loading mass. Enhancement of each component (active material, binder, conductive agent, and electrolyte) and optimization of electrode specifications and charge-discharge conditions have been recommended as ways to address this issue⁽⁸⁾⁻⁽¹⁰⁾. It is still the case, however, that cycle durability of 100 cycles or more has been confirmed at 3.0 mg/cm² and below, which is one-half or less than the common graphite electrode⁽¹¹⁾⁻⁽¹⁴⁾.

For the present research, therefore, Si was used as the active material and the 3D collector and silicate binder (hereinafter inorganic binder) were combined and an

investigation was made into Si anodes that can show adequate energy density as a cell. An effort was made to mitigate the issue of cycle durability, which arises when loading mass is enhanced, by combining, as shown in Fig. 3, the preservation of micrometer-size structure by means of the lattice structure of the 3D collector with preservation of nanometer-size structure by means of the strong siloxane bond from the inorganic binder, suppressing the splitting apart of the Si particles and of the composite material.

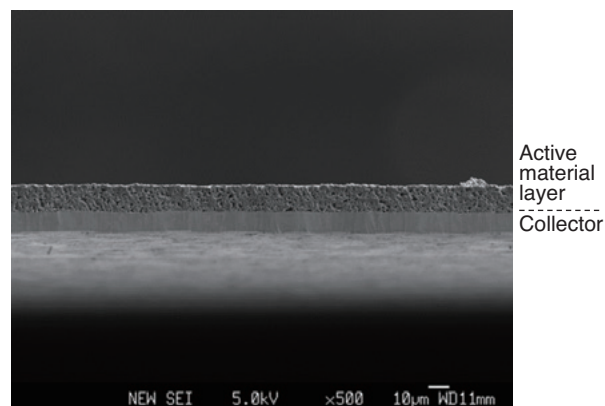


Fig. 1 Cross-section SEM image of Si electrode before aging (magnification: $\times 500$)

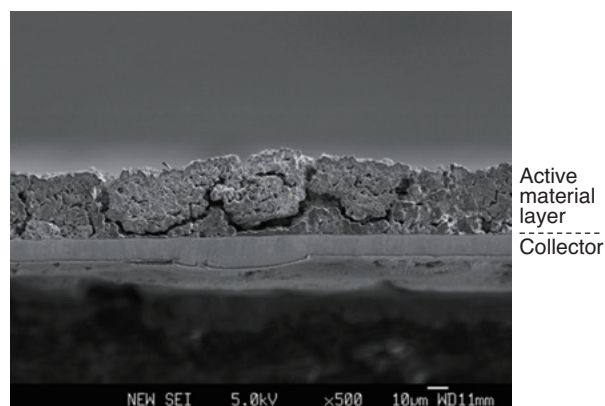


Fig. 2 Cross-section SEM image of Si electrode after aging (magnification: $\times 500$)

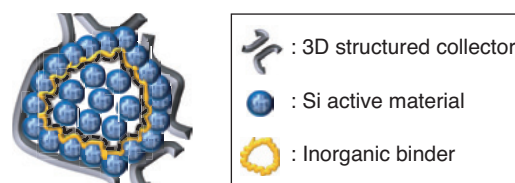


Fig. 3 Image of 3D anode with Si applied

Table 1 Electrode calculation conditions

	Foil coated electrode	3D electrode	3D-Si electrode
Cathode active material	$\text{LiNi}_{1/3}\text{Mn}_{1/3}\text{Co}_{1/3}\text{O}_2$		
Cathode active material content	94 wt%		
Cathode collector foil	Al 15 μm	Aluminum Celmet	
Cathode 3D-structured collector porosity	-	97%	
Anode active material	Graphite		Silicon
Anode active material content	96.5 wt%		91%
Anode 3D-structured collector porosity	-	97%	
Element ratio	83 vol%		
Anode collector foil	Cu 8 μm	Nickel Celmet	
Separator	15 μm		
Negative/Positive ratio	1.05		
Cell volume	360 mL		

In addition to the combining of these components, electrode density is also an important parameter. With a low density electrode, for example, it is possible to leave a certain amount of opening in the pole plate so that the strain on the electrode caused by expansion and shrinking can be suppressed. The cycle durability is thereby enhanced, while on the other hand, since there is capacity that volume does not contribute to, the volumetric energy density is decreased. The present investigation, with a view to raising the energy density to a high level, has also been conducted together with investigation of pressing to obtain higher density.

2.1. Calculation of Energy Density

Assuming a cell size of 360 mL and the calculation conditions in Table 1, the volumetric energy density of the cell for varying loading mass was calculated with electrodes using the conventional collector foil (hereinafter foil coated electrode), electrodes using the collector with a three-dimensional structure (hereinafter 3D electrode), and electrodes that use Si-related active material for the collector with a three-dimensional structure (hereinafter 3D-Si electrode). The calculation results are shown in Fig. 4.

The calculation results for the foil coated electrode are shown by a dotted line, those for the 3D electrode by a double line, and those for the 3D-Si electrode by a solid line. The calculation results shown for the 3D-Si electrode were obtained under conditions of electrode density 0.3 and 0.5 g/cm^3 .

With the foil coated electrode, taking into consideration that a loading mass of about 90 mg/cm^2 was the coating limit according to the past Honda evaluation because of diminished binding capacity largely due to uneven distribution of the binder, the volumetric energy density of the cell reaches 581 Wh/L at the highest, while with the 3D electrode, the preservation of macrostructure due to the 3D collector raises the upper limit of the loading mass so that the volumetric energy density of the cell rises as high as 641 Wh/L. This indicates that the increase in volumetric energy density is the result of the reduced amount of

components used, such as the separator and the collector, which do not contribute to capacity, because the high loading of the electrode has reduced the number of layers of electrodes necessary.

In addition to this, the 3D-Si electrode with Si active material applied is found, in a comparison with cathode loading mass at 90 mg/cm^2 , to have 629 Wh/L at low density (0.3 g/cm^3) and 732 Wh/L at high density (0.5 g/cm^3). At the cathode loading mass of 150 mg/cm^2 , the figures are 695 Wh/L at low density (0.3 g/cm^3) and 817 Wh/L at high density (0.5 g/cm^3). According to the above results, cell volumetric energy density can be enhanced by a factor of 1.4 at most relative to the conventional foil coated electrode in calculation.

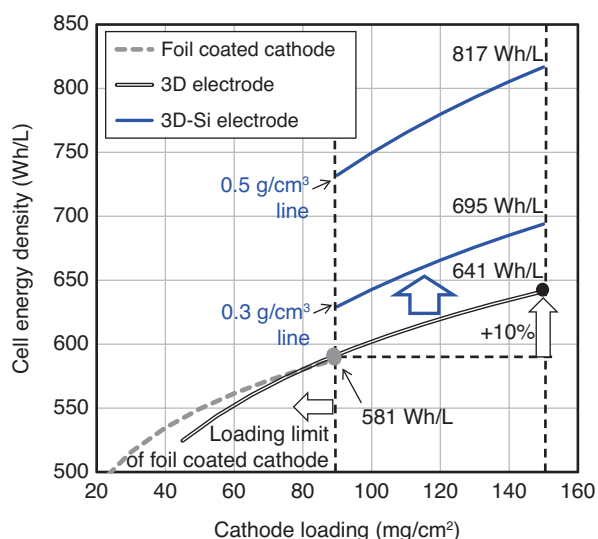


Fig. 4 Relationship between electrode type and cell energy density

3. Testing Method

3.1. Cell Components

3.1.1. Collector with three-dimensional structure

Celmet manufactured by Sumitomo Electric Industries, Ltd., was used for the collector. Aluminum Celmet was used for the cathode and nickel Celmet for the Si-related anode. Ordinarily, copper collector foil is used for a LIB anode, but here the stronger nickel was adopted for the material with the aim of heightening the mechanical strength of the collector as a measure to counter the expansion and shrinking of Si active material.

3.1.2. Cathode

Ternary layered oxide $\text{LiNi}_{1/3}\text{Co}_{1/3}\text{Mn}_{1/3}\text{O}_2$ (NCM) was used for the active material, acetylene black (AB) for the

conductive agent, and polyvinylidene difluoride (PVDF) for the binder, combined with an element ratio of NCM/AB/PVDF = 94/4/2 wt%. After that, the flow shown in Fig. 5 was followed to make the electrode so that after (a) the slurry has been prepared, (b) the electrode coating is applied with a loading mass of 100 mg/cm² on the electrode, after which that electrode is (c) roll pressed to an electrode density of 3.2 g/cm³.

3.1.3. Anode

Table 2 shows the specifications of each electrode. Si powder is used for the active material, AB for the conductive agent, with vapor grown carbon fiber (VGCF), and PVDF as binder with an element ratio of Si/AB/VGCF/PVDF = 91/4.2/0.8/4 wt%. Using this composition, a similar procedure is followed with the cathode to yield an electrode with the loading mass and density shown in Table 2 before coating with the inorganic binder.

The electrode obtained in this way is coated and impregnated with inorganic binder solution (K₂O•3SiO₂) to a loading mass of 1.0 mg/cm² by a spray method with reference to a past document⁽¹⁰⁾. After that, it is heat treated to a temperature of 120°C and the intended electrode is obtained.

This electrode is roll pressed (rolling temperature of 60°C) and electrodes of varying densities, as shown in Table 2, were obtained.

3.1.4. Separator

A microporous polyethylene film of 15 μm thickness was used.

3.1.5. Electrolyte

A mixture of ethylene carbonate (EC), dimethyl carbonate (DMC), and ethyl methyl carbonate (EMC) was decided on for the solvent, at a ratio of EC/DMC/EMC = 30/40/30 wt%. The electrolyte was made by dissolving the supporting salt lithium hexafluorophosphate (LiPF₆) in a 1.2 mol concentration in the solvent mixture.

3.1.6. Cell exterior body

The CR2032 type coin cell was used for the exterior body here.

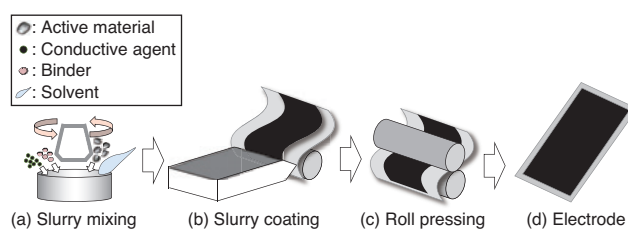


Fig. 5 Lithium-ion battery electrode manufacturing processes

Table 2 Silicon electrode

	Reference	Si-0.05 g/cm ³ (Si-1)	Si-0.17 g/cm ³ (Si-2)	Si-0.59 g/cm ³ (Si-3)
Active material	Silicon			
Active material content	90 wt%			
Electrode loading	6.0 mg/cm ²		5.5 mg/cm ²	5.3 mg/cm ²
Electrode density	0.05 g/cm ³		0.17 g/cm ³	0.59 g/cm ³
Inorganic binder	Not applied	Applied		
Collector	Nickel Celmet			

3.2. Analysis of Electrode

3.2.1. Scanning electron microscope

Cross sections prepared using a cross-section polisher were observed using an ULTRA 55 SEM from Carl Zeiss AG for the purpose of confirming the state of collector contact with active material in the electrode.

3.2.2. Energy dispersive X-ray spectrometry analysis

Energy dispersive X-ray spectroscopy (EDX) with equipment from JEOL Ltd. was used to perform element mapping (C, F, K, Si, and Ni) of the above cross-sections for the purpose of confirming the dispersibility of the active material, conductive agent, and binder in the electrode. Of the elements, C indicates the presence of conductive agent and binder, F the presence of binder, K the presence of inorganic binder, Si the presence of active material, and Ni the presence of collector, respectively.

3.3. Cell Evaluation Method

The cells were placed in a thermostatic chamber at 45°C and each subjected to charge-discharge cycle testing at a constant current of 0.1 C in the range from 4.2 to 2.5 V. Measurements were made with a rest time of one hour. For this testing, five charge-discharge cycles were implemented at a constant current of 0.1 C as aging treatment, after which a total of 50 cycles were implemented.

4. Results and Discussion

Figures 6 to 8 show the cross-section SEM results for each three-dimensional electrode with Si applied at Si-0.05 g/cm³, Si-0.17 g/cm³, and Si-0.59 g/cm³ (hereinafter Si-1, Si-2, and Si-3) before cycle testing. Figures 9 to 11 show the respective results for element mapping using EDX.

Si-1, which is a post-coating electrode, has a film thickness and opening structure (outlined in red in the figures) as the pre-coating collector. In contrast to Si-1, however, Si-2 and Si-3 are samples that have been put through pressing to adjust the density to 0.17 g/cm³ and 0.59 g/cm³, respectively, and it was confirmed that the increase in their density is accompanied by a decrease in large openings that are due to the structure itself, whereas in Si-3 there are none as far as can be determined from cross-section images.

The EDX results for each electrode also confirm that conductive agent (C image) and binder (F image: PVDF; K image: inorganic binder) are present in every sample in

the active material shown in the Si image. In Si-3, however, there is contrast in the K image inside and outside the electrode, suggesting that the amount of inorganic binder inside the electrode is not adequate.

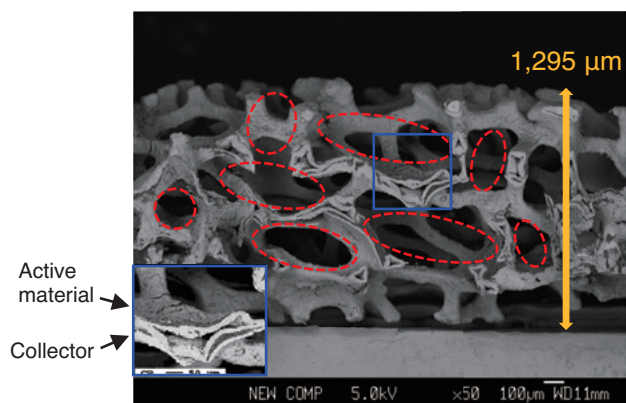


Fig. 6 Cross-section SEM image of Si-1 before test (magnification: $\times 50$)

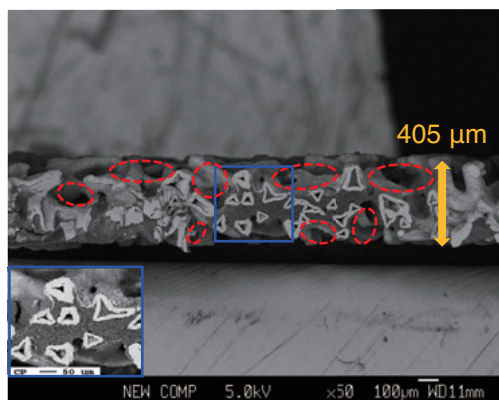


Fig. 7 Cross-section SEM image of Si-2 before test (magnification: $\times 50$)

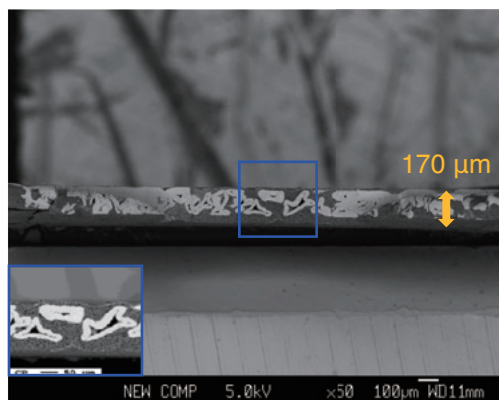


Fig. 8 Cross-section SEM image of Si-3 before test (magnification: $\times 50$)

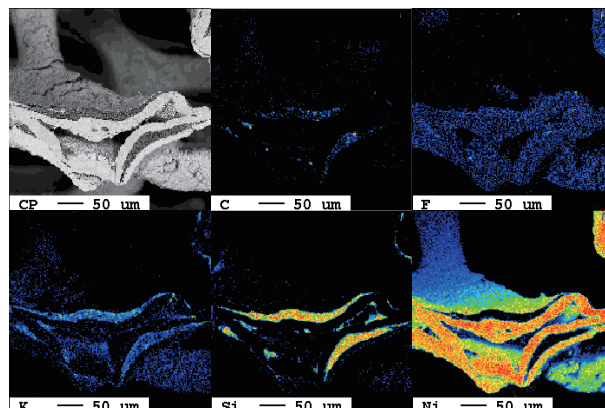


Fig. 9 Cross-section EDX images of Si-1

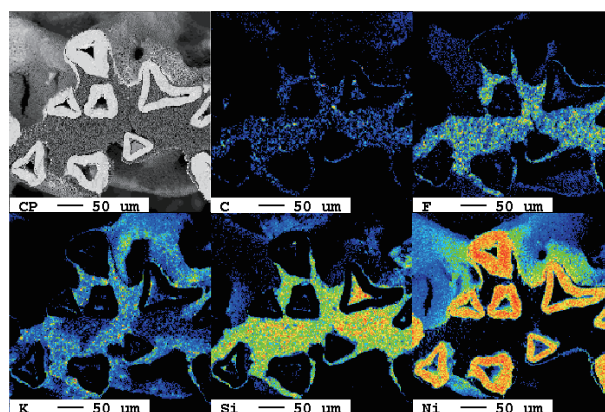


Fig. 10 Cross-section EDX images of Si-2

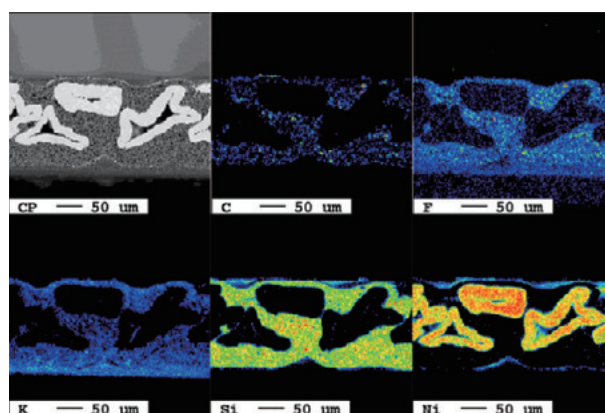


Fig. 11 Cross-section EDX images of Si-3

Figure 12 shows changes in charging capacity with the reference item and with Si-1 during aging (five cycles). In the reference item, capacity retention relative to initial capacity had decreased as far as 25% at the second aging cycle. In Si-1, however, it was at 78%. It was confirmed from these results that the preservation of micro structure by the inorganic binder was effective even in the series to which the three-dimensional collector was applied.

Figure 13 shows the discharge capacity of each electrode with different density (Si-1, Si-2, and Si-3) before and after cycle testing. Figure 14 shows the relationship

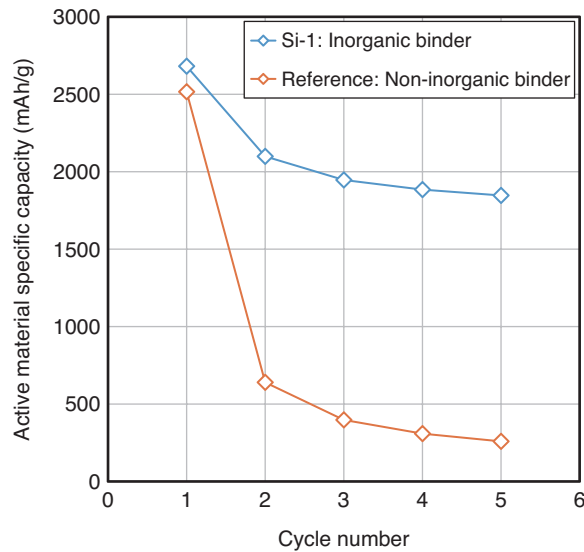


Fig. 12 Charge capacity in aging process of reference item and Si-1

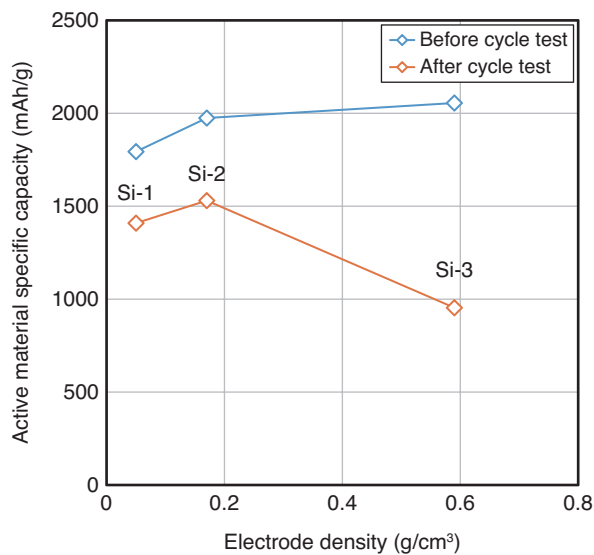


Fig. 13 Discharge capacity of each electrode before and after cycle test

between cycle number and capacity retention. The initial discharge capacities for Si-1, Si-2, and Si-3, in that order, were 1794, 1975, and 2055 mAh/g. This enhancement of capacity is thought to be due to the higher density, which enhanced the contact characteristics of the conductive agent, the active material, and the collector. On the other hand, the discharge capacities after 50 cycles had passed were, in the above order, 1409, 1530, and 954 mAh/g. Looking also in terms of capacity retention, these were 78%, 77%, and 46%, respectively. In terms of cycle durability, this suggested that lower density items had an advantage. The main factors in this are thought to be the openings in the electrode, which functioned as a layer to buffer the expansion and shrinking of the Si active material, and the dispersibility of the inorganic binder. The above results indicate that with the specifications applied here, 78% capacity retention can be achieved at the 50 cycle point even with high loading. They also indicate that discharge capacity and capacity retention are dependent on electrode density.

Figures 15 to 17 show low-magnification (50 power magnification) cross-section SEM images of each electrode removed after cycle testing. Figures 18 to 20 show their high-magnification (2000 power magnification) cross-section SEM images.

From the low-magnification SEM images, the film thicknesses of electrodes Si-1, Si-2, and Si-3, in that order, were found to be 1057, 487, and 200 μm . The cross-section SEM images in Figs. 6 to 8 indicated film thicknesses before cycle testing on 1295, 405, and 170 μm , respectively. In this light, the highest rate of electrode expansion before and after cycle testing is in Si-2, by a factor of about 1.2 at most. From the electrode configurations investigated here, a reduction to about 25% by comparison with the coated

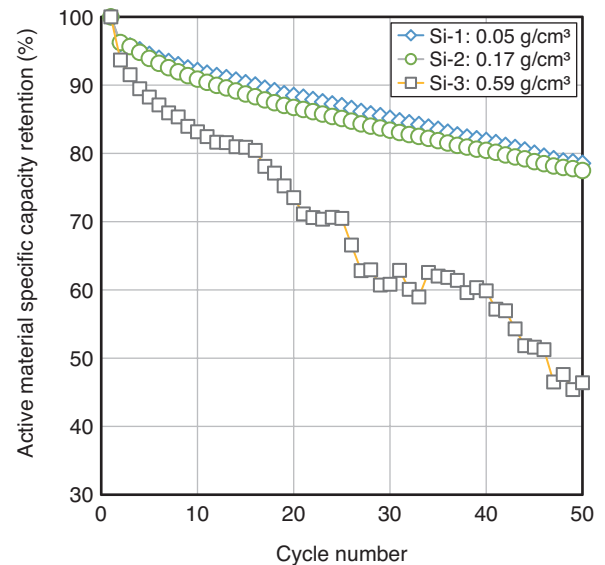


Fig. 14 Capacity retention of Si-1, Si-2, and Si-3

electrodes was confirmed. This is thought to result from the suppression of splitting apart and slipping off in the active material. This suppression occurs because of micro

structure preservation by inorganic binder in the active material and also because this is combined with application of the 3D collector, which can be expected to provide three-

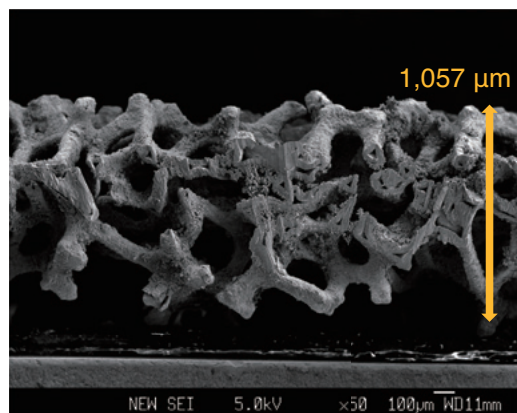


Fig. 15 Cross-section SEM image of Si-1 after test (magnification: $\times 50$)

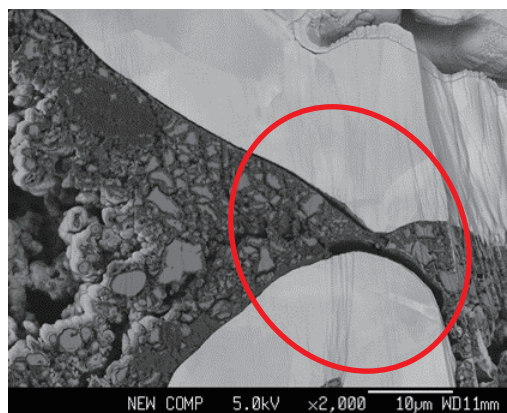


Fig. 18 Cross-section SEM image of Si-1 after test (magnification: $\times 2000$)

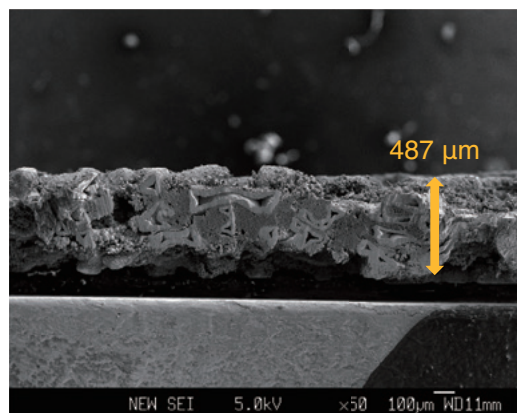


Fig. 16 Cross-section SEM image of Si-2 after test (magnification: $\times 50$)

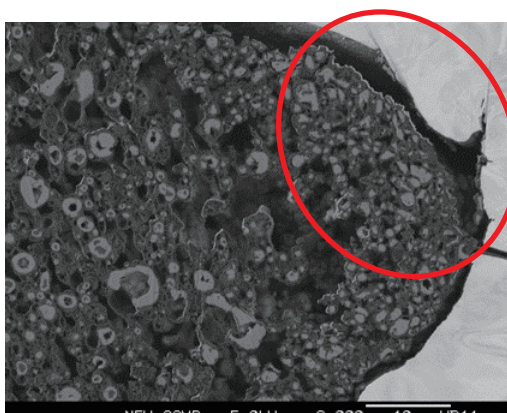


Fig. 19 Cross-section SEM image of Si-2 after test (magnification: $\times 2000$)

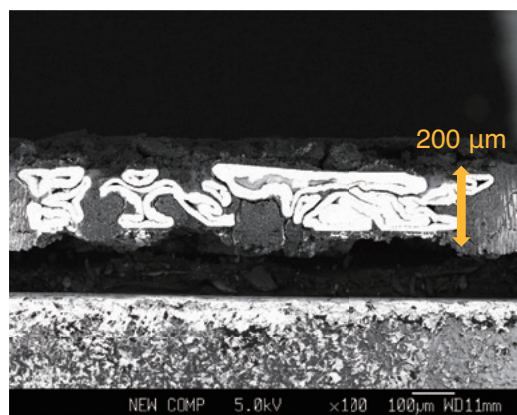


Fig. 17 Cross-section SEM image of Si-3 after test (magnification: $\times 50$)

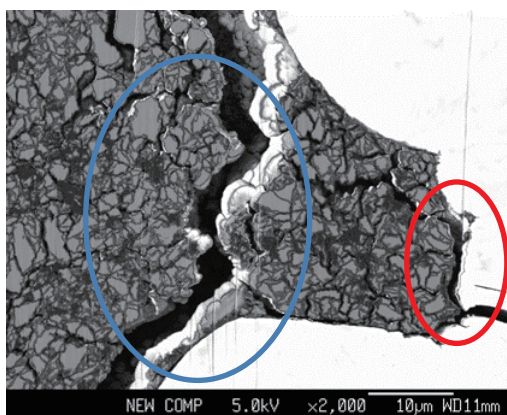


Fig. 20 Cross-section SEM image of Si-3 after test (magnification: $\times 2000$)

dimensional structure preservation from the expansion and shrinking of the active material. It is also the case that in Si-1, the electrode film rather shrinks in thickness, but since these electrodes are unpressed items, this is thought to occur because of the influence of shrinkage under the constraining load of the coin cell. According to the above results, these specifications are confirmed to be an effective means for achieving a balance between high loading with Si-related anodes and high durability.

On the other hand, high-magnification cross-section SEM images confirmed that cracks occur, in the cases of Si-1 and Si-2, between the collector foil and the active material composite layer (outlined in red in Figs. 18 and 19), and in the case of Si-3, between the collector foil and the active material composite layer (outlined in red in Fig. 20) as well as in the active material composite layer (outlined in blue in Fig. 20). It is thought that the cracks in the composite layer lead to isolation of the active material and an increase in resistance that accompanies the interruption of conductive paths, such that capacity retention decreases as a result. It is also thought that, in the cases of Si-1 and Si-2, openings exist in the electrode that are due to the structure itself, and these function as a layer to buffer the expansion and shrinking of the composite layer. In the case of Si-3, however, capacity retention is thought to have been lower because almost no openings exist, and as a result there is a large dynamic load on the interface of the collector foil and the active material composite layer so that cracks also occur in the interface between the collector foil and active material as well as in the composite layer.

The above results confirmed that with the specifications applied here, collapse of the electrode early in the cycles was suppressed even with high loading, but minute cracks in the vicinity of the collector interface caused the interruption of conductive paths, which led to diminished capacity retention. Going forward, it is thought that, in order to suppress the minute cracking, optimization of materials and electrode specifications as well as of charge-discharge conditions will be necessary for the purpose of further enhancing cycle durability with a view to practical application.

5. Conclusion

An investigation was made into the application of a three-dimensional collector and inorganic binder to high loading Si anodes with a view to raising the energy density of lithium-ion secondary batteries to a high level.

- (1) The application of a three-dimensional collector as well as of inorganic binder with a loading mass of about 6.0 mg/cm² yielded a capacity retention of 78% at the 50-cycle point.
- (2) The results from disassembly analysis of the electrode show, on the one hand, that every electrode expands by a factor of no more than about 1.2 at most, and the specifications presented here contributed to suppression

of the expansion and shrinking that is an issue for Si-related anodes. On the other hand, however, the presence of cracks was confirmed between the collector foil and the composite layers as well as in the composite layers.

Going forward, further optimization of specifications of components and electrodes will be pursued with a view to suppression of the cracking that is considered a degradation factor.

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Investigation of CO₂ and Steam Co-electrolysis Conditions for Synthesizing Carbon Recycled Methane

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ABSTRACT

An investigation of methane synthesis by co-electrolysis using renewable energy was conducted. The aim was to achieve highly efficient energy conversion in carbon recycling. Specifically, the investigation focused on the reaction conditions that apply when carbon dioxide and high-temperature steam are used to obtain syngas, which is a mixture of carbon monoxide and hydrogen made by co-electrolysis. First, the co-electrolysis performance of the electrolyte supported cell and of the fuel electrode supported cell were compared in order to determine the characteristics of a solid oxide electrolysis cell that are suited to syngas production. It was found from the results of that comparison that both the electrolysis current density and the syngas conversion rate are higher with the fuel electrode supported cell. Next, the relationship between the electrolysis current density and the source gas throughput quantity, which is a design parameter for the solid oxide electrolysis cell reactor, was obtained using a fuel electrode supported cell. Confirmation was also obtained of the source gas composition needed to match the syngas composition to the theoretical molar ratio for producing methane gas, which is an operation criterion for such reactors.

1. Introduction

Countries around the world have declared ambitious targets and initiatives oriented to achieving carbon neutrality by the year 2050 as a measure to counter the climate change caused by carbon dioxide (CO₂) and other greenhouse gases⁽¹⁾.

One of the measures being investigated is carbon dioxide recycling (carbon recycling) using renewable energy, and attention is being directed to technology for methane production (methanation) by the power to gas (P2G) approach. P2G is a technology for converting the electricity from renewable energy into gaseous fuel. The source gas for methanation is CO₂, and exhaust gases from steelworks⁽²⁾, garbage incinerating facilities⁽³⁾, thermal power plants, and other such facilities are considered valuable sources for that CO₂. Methane (CH₄) that has been produced using electricity from 100% renewable energy is defined as carbon neutral green methane, which has prospects for conversion to city gas or use as an alternative fuel⁽⁴⁾. Since the composition of city gas is approximately 90% CH₄, green methane can be injected into existing natural gas and city gas pipelines for transport and storage.

The aim of the present research, therefore, is the kind

of highly efficient carbon recycled CH₄ synthesis shown in Fig. 1. In co-electrolysis, simultaneous electrolytic reactions in CO₂ and high-temperature steam (H₂O) yield syngas. In other words, this is a system to synthesize CH₄ by means of the mixture of carbon monoxide (CO) and hydrogen (H₂) that is syngas.

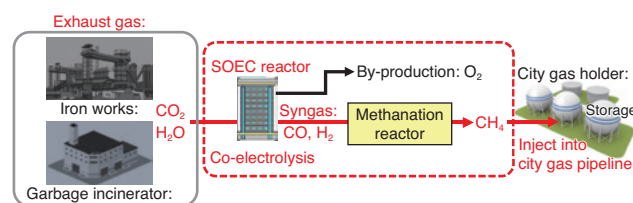


Fig. 1 Co-electrolysis methanation system via syngas

2. Achieving High-efficiency Methane Production

2.1. Methane Synthesizing Process

There are two main methods of CH₄ production by P2G, as shown in Fig. 2, that are being investigated⁽⁴⁾.

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In the method in Fig. 2(a), H₂ is produced by steam electrolysis or alkaline water electrolysis carried out upstream. Next, the produced H₂ and CO₂ are mixed together and supplied to a downstream thermochemistry reactor where CH₄ synthesis is carried out by the Sabatier reaction.

In the method in Fig. 2(b)⁽⁵⁾⁻⁽⁷⁾, the co-electrolysis reaction of CO₂ and H₂O upstream produces syngas that is supplied to a downstream methanation reactor. There CH₄ synthesis is carried out by thermochemical reaction.

Comparison of these two methods with attention to the chemical species involved in the reactions shows that for the method in Fig. 2(a), it is necessary to temporarily store the H₂ produced upstream by electrolysis and mix it with CO₂ so that it matches the mol number needed for CH₄ synthesis downstream. It can then be introduced into the thermochemistry reactor. When large-scale production of CH₄ is envisioned, therefore, it will be necessary to install new storage tanks and pipelines for the electrolyzed H₂. Preparation of the related statutes and technical regulations associated with that activity will also be an issue.

In Europe, the possibility of injecting electrolyzed H₂ into city gas pipelines and using it as a part of city gas is also being considered. However, as H₂ has a lower heat value per volume than CH₄, there are limits on the amount of H₂ that can be mixed in, and its use has not been expanding⁽⁸⁾.

With the approach in Fig. 2(b), on the other hand, if the composition of the syngas produced upstream by co-electrolysis can have a mol number that matches what is needed for CH₄ synthesis downstream, then that total amount can be introduced directly into the methanation reactor and there will be no need for intermediate gas storage. It is also the case that the source gases in Fig. 2(b) have lower mol numbers so that the energy needed to raise the pressure and temperature of the source gas supply can be limited proportionately. The H₂O that is produced as a by-product simultaneously with CH₄ following the synthesis reaction also has a lower mol number, so the dehydration energy can be reduced.

For the present research, therefore, it was decided to use the method in Fig. 2(b), which can be expected to consume less energy outside the reaction path and to involve more

highly efficient methanation.

2.2. Syngas Production by Co-electrolysis

In order to more efficiently obtain syngas to serve as source gas for methane synthesis, a solid oxide electrolysis cell (SOEC) was used as the co-electrolysis reactor for the present research.

Figure 3 shows a conceptual diagram of co-electrolysis by an SOEC reactor. The CO₂ and H₂O that serve as source gases are simultaneously subjected to electrolysis reduction and the syngas that will serve as a source for the CH₄ synthesis reaction is obtained.

In co-electrolysis, the reactions shown in Eqs. (1) and (2) occur simultaneously in the fuel electrode. CO, H₂, and oxygen ions (O²⁻) are produced from the source gas and electrons (e⁻).



The aqueous shift reaction shown in Eq. (3) also occurs in parallel as by-production.



At the oxygen electrode, oxygen (O₂) is generated from O²⁻ as shown in Eq. (4).



Syngas production by co-electrolysis occurs as an endothermic reaction, as shown in Fig. 4. The energy required for this co-electrolysis reaction is shown in Eq. (5).

$$\Delta H = \Delta G + T\Delta S \quad (5)$$

ΔH : Enthalpy [kJ]

ΔG : Gibbs energy [kJ]

T : Operating temperature [K]

ΔS : Entropy [kJ/K]

Here ΔH is defined as the total energy needed for the reaction, ΔG as the energy supplied as electricity, and $T\Delta S$ as the energy supplied as heat. As a portion of the reaction energy can also be supplied as heat in this way, the use, for example, of waste heat from steelworks, garbage incinerating facilities, thermal power plants, and other such

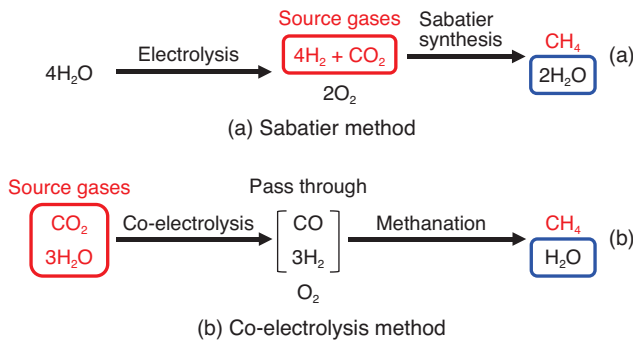


Fig. 2 Comparison of methane synthesis processes

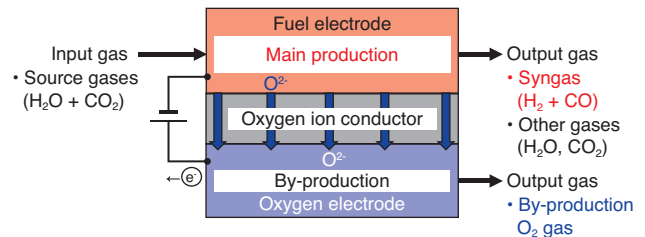


Fig. 3 Structure and material flow in SOEC co-electrolysis reactor

facilities could reduce the input energy. This is anticipated as an advantage of SOEC over other methods⁽⁹⁾. This may result in the apparent P2G energy conversion efficiency exceeding 100% when SOEC is used, so the heat balance needs to be taken into consideration when comparing with other methods.

The SOEC requires an operating temperature of at least 650°C or more in order to lower the resistance for O^{2-} transport in the ion conductor⁽¹⁰⁾. On the other hand, the heat generated by the overvoltage that accompanies this lowering of O^{2-} transport resistance can be used for endothermic reaction. The potential at which the heat absorbed by a high-temperature electrolytic reaction is in balance with the heat generated by electrolytic cells is called the thermo-neutral voltage. The electrolytic voltage is around 1.3 V/cell. When the voltage exceeds this, therefore, self-sustaining operation becomes possible and a supply of heat from outside becomes unnecessary.

2.3. Design Parameters for SOEC Reactors Using Co-electrolysis

The size of an SOEC reactor used for syngas production is decided on the basis of the CO_2 target throughput quantity ($Ta\ CO_2\ tqr$), as shown in Eq. (6)⁽¹¹⁾. To make the SOEC reactor more compact and reduce its manufacturing cost, it is effective to reduce the cell stacking number ($No\ cr$). For that purpose, it is necessary to raise the electrolysis current density and increase the CO_2 throughput quantity ($CO_2\ tra$) per unit active area.

$$Ta\ CO_2\ tqr = CO_2\ tra \cdot S\ ac \cdot No\ cr \quad (6)$$

$Ta\ CO_2\ tqr$: CO_2 target throughput quantity per minute per SOEC reactor [t - CO_2 /min/Reactor]

$CO_2\ tra$: CO_2 throughput quantity per minute per unit active area [t - CO_2 /min/ m^2]

$S\ ac$: Active area of cell [m^2 /Cell]

$No\ cr$: Cell stacking number per SOEC reactor [Cells/Reactor]

On the other hand, increasing the flow rate of source

gas input to the SOEC reactor in order to increase the CO_2 throughput quantity may lower the rate of conversion to syngas⁽¹²⁾. It has been conjectured that this occurs because when source gas exceeding the throughput capacity of the SOEC reactor is input, blow-by ends up occurring. This is why it is important to ascertain the source gas throughput quantity from the electrolysis current density and to ascertain the blow-by source gas quantity from the syngas conversion rate in order to determine the electrode area per cell and cell stacking numbers.

2.4. Operation Criteria for SOEC Reactors Using Co-electrolysis

With syngas production using an SOEC reactor, the aqueous shift reaction in Eq. (3) occurs in parallel with the electrolytic reactions of CO_2 and H_2O shown in Eqs. (1) and (2). For this reason, it is possible that the amount of H_2O in the source gas input to the SOEC reactor that is converted to H_2 , and the amount of CO_2 that is converted to CO , may not be equal to each other. The theoretical molar ratio required to carry out CH_4 synthesis in the downstream methanation reactor is $H_2/CO:3$. In order to have the gas output from the SOEC reactor match this ratio, therefore, it is important to adjust the H_2O/CO_2 molar ratio in the source gas that is input upstream to the SOEC reactor as well as to ascertain the temperature dependence of the aqueous shift reaction.

3. Research Goals

With regard to methanation carried out through syngas production by co-electrolysis of CO_2 and H_2O using an SOEC reactor, the following matters constituting SOEC reactor design parameters and operation criteria were investigated with the aim of increasing the efficiency of methanation systems.

- (1) Selection of SOEC type suited to syngas production and grasp of that SOEC type's structure
- (2) Confirmation of electrolysis current density characteristics, source gas throughput quantity, and syngas conversion rate
- (3) Confirmation of source gas H_2O/CO_2 molar ratio for syngas production that provides the theoretical ratio for CH_4 synthesis and confirmation of reaction conditions

4. SOEC and Test Apparatus

4.1. Selection of SOEC Type Suited to Syngas Production

The basic structure of an SOEC is as shown in Fig. 5. It is made up of three main layers, which are the fuel electrode layer, the electrolyte layer, and the oxygen electrode layer.

At present, no SOEC designed for dedicated co-electrolysis use exists. Therefore a solid oxide fuel cell (SOFC) was put to use as an SOEC. The fuel electrode layer is a combination of the nickel (Ni) catalyst, which has e^- conductivity, and a porous structure of oxide ceramic

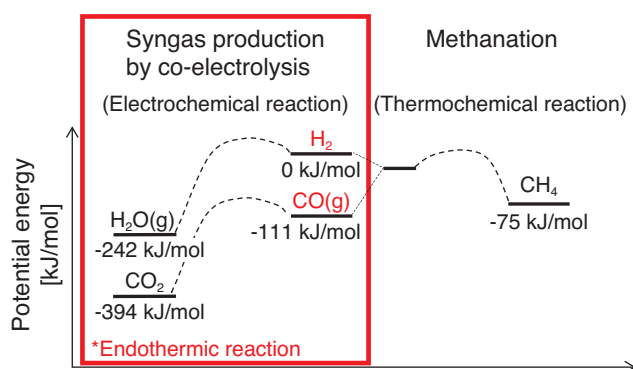


Fig. 4 Potential energy diagram in syngas synthesis and methanation

(YSZ, GDC) with O²⁻ conductivity. Here the e⁻ are received, the CO₂ and H₂O source gases go through reduction by co-electrolysis reaction, and syngas is produced. The electrolyte layer (ScSZ, YSZ) is the layer that transports the O²⁻ produced in the fuel electrode to the oxygen electrode that is the positive electrode by electrical field. It also functions as a barrier that blocks the permeation of gas between the fuel electrode and the oxygen electrode. The oxygen electrode layer is an oxide ceramic (LSM, GDC, LSC) that includes lanthanum (La) and strontium (Sr), forming a porous structure that has O²⁻ conductivity and e⁻ conductivity. This is where O²⁻ transported from the fuel electrode is oxidized and produces O₂ and e⁻.

When the structures of the two types of SOEC shown in Fig. 5 are compared, the electrolyte supported cell (ESC) is a structure in which the electrolyte layer is thick and the fuel electrode layer is thin. The fuel electrode supported cell (FSC) is a structure in which the electrolyte layer is thin and the fuel electrode layer is thick. As the FSC has a thinner electrolyte layer, it can also be considered to have lower O²⁻ transport resistance, but when used as an SOEC, it is not known which of the ESC and FSC has the advantageous structure. For the present research, therefore, the ESC and FSC were first compared and a selection made.

4.2. SOEC Reactor for Co-electrolysis Testing

The structure of the SOEC reactor used for the present research is shown in Fig. 6.

A tubular type of SOEC also exists⁽¹¹⁾. For the present research, however, a planar SOEC was selected in order to simplify the comparison of different SOEC materials and structures, the expansion of electrode area (active area) by stacking, and so on.

In exterior shape and size, the SOEC is 50 mm square and the electrode is 40 mm square. The SOEC is sandwiched between collectors (mesh of 100 0.7-mm diameter Au wires), and outside that is an inter connector made of alumina, with piping attached, in a structure that holds the contents in place. The inter connector is formed with 10 straight gas channels that are rectangular in cross-section, 2 mm wide and 0.4 mm deep. Their flow paths are arranged so that the fuel electrode gas channels and oxygen electrode gas channels form counter flows with each other.

Type	ESC	FSC
Cross-sectional view		
Plan view		

Fig. 5 Layer structure and materials in SOEC

4.3. Test Apparatus

Figure 7 shows an overview of the co-electrolysis test apparatus. The CO₂ for use as source gas is supplied at 99.99% purity from a gas cylinder. After having the supply flow rate of the CO₂ adjusted by a mass flow controller (MFC), it is humidified by being passed through a pure water bubbler that is heated to 90°C or higher. The H₂O/CO₂ molar ratio is adjusted using the dew point setting of the bubbler, and is then input as source gas to the fuel electrode gas channel intake of the SOEC reactor. The oxygen electrode gas channel intake of the SOEC reactor was supplied with a carrier gas consisting of inert argon (Ar) gas with 5% of O₂ by volume. Gas was input to the SOEC reactor with the reactor outlets of both electrodes open to atmospheric pressure.

Electrolysis current was applied by the parallel operation of intelligent bipolar power supplies (PBZ20-20 from Kikusui Electronics Corp.). The volume flow of the gas output from the SOEC reactor after co-electrolysis reaction was measured using high-accuracy film flow meters (SF-U Series VP-3U, VP-4U from HORIBA STEC Co., Ltd.). After dehumidification, the gas was injected into

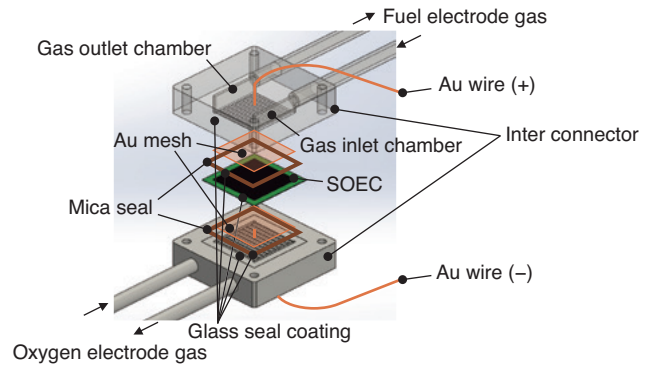


Fig. 6 SOEC reactor test rig structure

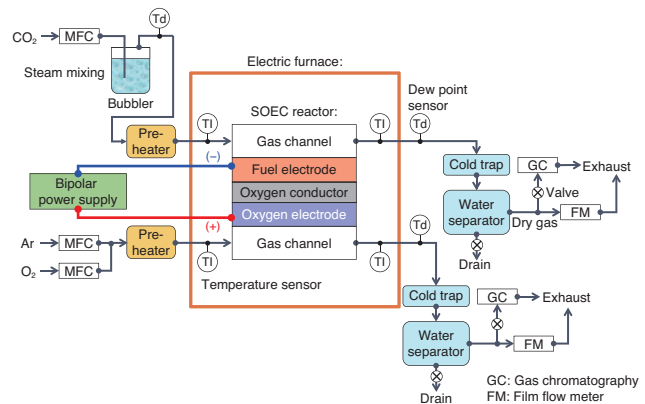


Fig. 7 Co-electrolysis test apparatus

a micro gas chromatograph (990 Micro GC from Agilent Technologies) and its composition was analyzed. Gas analysis of H₂, O₂, N₂, CH₄, and CO was carried out with Ar gas as the carrier gas, with molecular sieve 5A in the separation column, using a thermal conductivity detector (TCD) for measurement. For CO₂ analysis, measurement was made with a TCD using helium (He) gas as the carrier gas and Pora PLOT Q in the separation column.

5. Test Conditions

The present research was carried out as described in Sections 5.1. to 5.3., in that order.

5.1. Selection of SOEC Type

To select an SOEC type that is suited to syngas production, the ESC and FSC were compared by the syngas conversion rate, which is expressed as the proportion of output syngas quantity by co-electrolysis reaction per input source gas quantity. The test conditions are shown in Table 1.

The H₂O/CO₂ molar ratio for the source gas input to the SOEC reactor was set at 3, the same value as the theoretical molar ratio for CH₄ production. The operating temperature was set at 800°C, taking O²⁻ transport resistance in the ion conductor into consideration. The flow rate of source gas input to the SOEC reactor was set at 1.12 NLM, which is 10% of source gas utilization at an electrolysis current density of 1.0 A/cm². The electrolysis current was adjusted so that the cell voltage reached a thermo-neutral voltage of 1.3 V/cell and maintained this level under these conditions. Next, the source gas flow rate was gradually reduced from this point and the conditions under which electrolysis current density and syngas conversion rate reach their highest level at thermo-neutral voltage were confirmed.

5.2. Evaluation of Electrolysis Current Density Characteristics

Electrolysis current density characteristics were evaluated in order to obtain design parameters for the SOEC reactor. The test conditions are shown in Table 2.

The H₂O/CO₂ molar ratio for the source gas input to the SOEC reactor was set at 3, the same value as the theoretical molar ratio for CH₄ production. The operating temperature was set at 850°C, taking the heat resistance temperature of the cell into consideration. The cell voltage was adjusted to a thermo-neutral voltage of 1.3 V/cell.

5.3. Evaluation of Source Gas H₂O/CO₂ Molar Ratio Characteristics

The source gas H₂O/CO₂ molar ratio characteristics were evaluated in order to obtain criteria for the SOEC reactor operating conditions. The test conditions are shown in Table 3.

The operating temperature was set at 700, 800, and 850°C in order to determine the temperature dependence of the aqueous shift reaction. Here again the cell voltage was

Table 1 Evaluation conditions for selection of SOEC type

Parameters	Values	Units
Steam/CO ₂ molar ratio	3	N.A.
Operating temperature	800	°C
Source gas flow rate	0.16-1.12	NLM*
Operating pressure	No pressurization	N.A.
Cell voltage	1.3	V

*NLM: Normal liters per minute

Table 2 Evaluation conditions for characteristics of electrolysis current density

Parameters	Values	Units
H ₂ O/CO ₂ molar ratio	3	N.A.
Operating temperature	850	°C
Source gas flow rate	0.16-0.56	NLM
Operating pressure	No pressurization	N.A.
Cell voltage	1.3	V
Current density	1.3-2.2	A/cm ²

Table 3 Evaluation conditions for characteristics of source gas H₂O/CO₂ molar ratio

Parameters	Values	Units
H ₂ O/CO ₂ molar ratio	0.1-5	N.A.
Operating temperature	700, 800, 850	°C
Source gas flow rate	0.16	NLM
Operating pressure	No pressurization	N.A.
Cell voltage	1.3	V

Table 4 Indices for evaluating co-electrolysis performance

Evaluation index	Definition
Current density	Current per unit active area
H ₂ conversion rate	Molar number of production H ₂ / Molar number of input H ₂ O
CO conversion rate	Molar number of production CO / Molar number of input CO ₂
Syngas conversion rate	Molar number of production H ₂ and CO / Molar number of input H ₂ O + CO ₂
Faraday efficiency	Reaction electrons of production gas / Supplied electrons
Conversion energy efficiency	Combustion heat value of production gas (HHV) / Supplied electricity

adjusted to the thermo-neutral voltage of 1.3 V/cell.

The indices used to evaluate co-electrolysis performance are shown in Table 4⁽¹³⁾.

Electrolysis current density is the electrolysis current per unit active area. The greater the current density, the higher the source gas (mixture of CO₂ and H₂O) throughput quantity will be.

The rate of H₂ conversion by co-electrolysis expresses the conversion rate of H₂O in the source gas input to the SOEC reactor into H₂. Similarly, the CO conversion rate expresses the conversion rate of CO₂ into CO. The syngas conversion rate is the sum of the H₂ conversion rate and CO conversion rate, and it expresses the conversion rate of source gas into syngas. Higher conversion rates can be said to indicate higher performance as a reactor.

The Faraday efficiency expresses the reaction electron number as converted from the amount of syngas production per input electron number, and the directly produced CH₄ was also included in the reaction electron number. The conversion energy efficiency expresses the rate of energy conversion to the high heating value (HHV) of the gas that was produced from the amount of electricity input to electrolysis. Since the heat energy indicated in Eq. (5) is also used in the co-electrolysis reactions for the present research, the energy in the produced gas may at times exceed the input electricity.

6. Results and Discussion

6.1. Selection of SOEC Type

The results from comparison of co-electrolysis performance by ESC and FSC are shown in Fig. 8. This shows the relationship between the syngas conversion rate and the electrolysis current density with respect to the flow rate of source gas input to the SOEC reactor.

When the source gas input flow rate was increased, the electrolysis current density increased regardless of differences in SOEC structure. Conversely, the syngas conversion rate went down. In the results of this test, the electrolysis current density reached its highest level under the condition of a source gas flow rate of 1.12 NLM. With the FSC type, the figure was 1.4 A/cm², and with the ESC type, it was 0.2 A/cm², indicating that the former can be expected to achieve throughput quantities that are

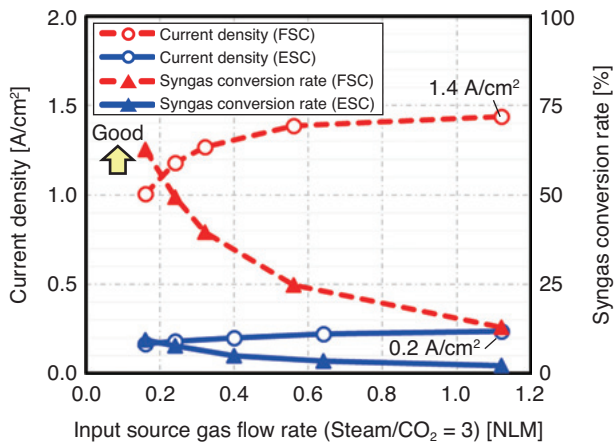


Fig. 8 Comparison of co-electrolysis performance ESC vs FSC at 800°C

Table 5 Comparison of co-electrolysis performance ESC vs FSC at 0.16 NLM

Parameters	ESC	FSC	Units
Syngas conversion rate	10	63	%
Current density	0.2	1.0	A/cm ²

seven times greater. The FSC is thought to have a greater throughput quantity than the ESC because, as shown by the cross-sectional view in Fig. 5, the electrolyte layer thickness of 3 μm in the FSC is thinner than the 150 μm in the ESC, so that this layer in the FSC has lower transport resistance of O²⁻.

On the other hand, the source gas input flow rate that yielded the highest syngas conversion rate was 0.16 NLM. The electrolysis current densities in ESC and FSC under this condition are shown in Table 5.

As for the syngas conversion rate, the ESC figure was 10% while the FSC figure was six times greater at 63%. Under this condition, the FSC also achieved an electrolysis current density of 1.0 A/cm², for five times larger throughput quantity than the ESC, at 0.2 A/cm². This difference in the conversion rate is thought to be due to the fuel electrode layer thickness, which is 50 μm in the ESC but 400 μm in the FSC, eight times thicker because it includes a support structure function, so that the amount of catalyst is correspondingly greater, and this promoted electrochemical reaction.

In light of the above results, the FSC type of SOEC was selected as suited to syngas production. Subsequent investigation of syngas production conditions was also carried out using the FSC type.

6.2. Evaluation of Electrolysis Current Density Characteristics

Figure 9 shows the changes occurring in electrolysis current density and in the amounts of H₂, CO, and CH₄ synthesized when the source gas input flow rate was gradually increased from 0.16 NLM. When the source gas

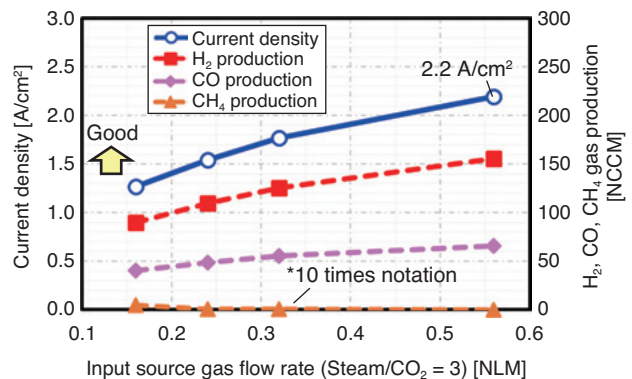


Fig. 9 Current density and produced gases at various input source gas flows at 850°C

flow rate was increased, the electrolysis current density increased so that the amounts of H_2 and CO synthesized also increased. However, it was found that the increase in amounts slowed down on the higher flow rate side so that it did not increase proportionately. Traces of CH_4 production were also detected, but a decrease in the amount produced was observed on the high flow rate side.

Next, Fig. 10 shows the results for calculation of the syngas conversion rate with respect to electrolysis current density from the test results in Fig. 9.

It was found that when the electrolysis current density increases, the syngas conversion rate decreases. This is thought to occur because the unreacted source gas that the SOEC reactor was unable to process completely was subject to blow-by. Source gas blow-by wastes the heating and humidifying energy for that blow-by amount, which lowers system efficiency. It is for this reason that SOEC reactor design gives priority to the syngas conversion rate.

The characteristics of source gas throughput quantity and syngas conversion rate, which constitute SOEC reactor design parameters, were successfully determined from the above. In order to increase the source gas throughput quantity, it is necessary to investigate SOEC reactor structure and related matters that will promote the supply of source gas to the catalyst in a way that reduces the amount of unreacted source gas.

6.3. Evaluation of Source Gas H_2O/CO_2 Molar Ratio Characteristics

Figure 11 shows the results when the H_2O/CO_2 molar ratio of the source gas input to the SOEC reactor is changed so that the composition of the output syngas will be $H_2/CO:3$, which is the theoretical molar ratio for CH_4 synthesis.

It was found that in order to make the output gas have a molar ratio of $H_2/CO:3$, it was necessary to increase the H_2O/CO_2 molar ratio of the input source gas to excess H_2O as the operating temperature rose higher. It has been conjectured that because the aqueous shift reaction from the right side to the left side of Eq. (3) takes place in parallel

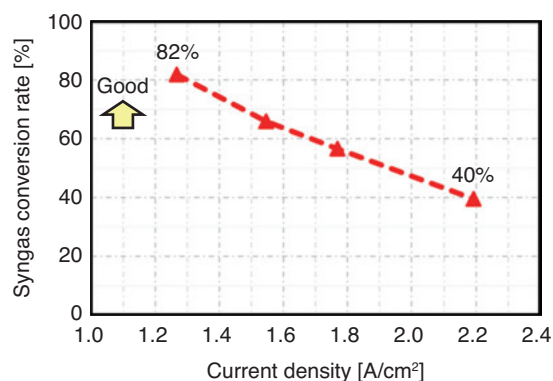


Fig. 10 Syngas conversion rate at various current densities at 850°C

with the co-electrolysis reaction. It is thought that the higher the temperature rises, the more the reactions that produce H_2O and CO from the H_2 produced by co-electrolysis and from the CO_2 in the unreacted source gas are promoted. The H_2O/CO_2 molar ratio that the input source gas should have in order for the output gas to have the theoretical molar ratio for CH_4 synthesis was 3.0 at 700°C, 3.3 at 800°C, and 4.2 at 850°C.

Figure 12 shows the Faraday efficiency and conversion energy efficiency at the various temperatures when this is done.

The Faraday efficiency was from 90% to 92% at the various temperatures, indicating high electrolysis performance. It was also found that the conversion energy efficiency exceeded 100%, at from 102% to 103%. This was because syngas was produced by an endothermic reaction that, in addition to the electricity directly involved

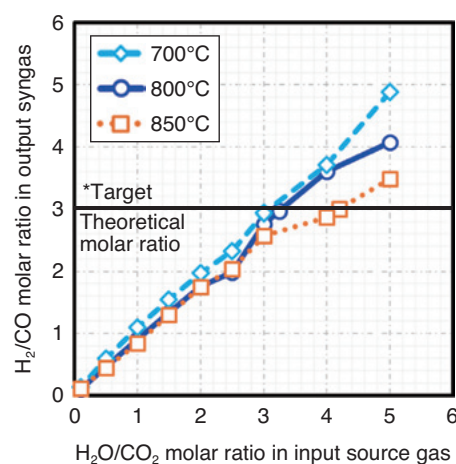


Fig. 11 H_2/CO output molar ratio vs H_2O/CO_2 input molar ratio at various temperatures

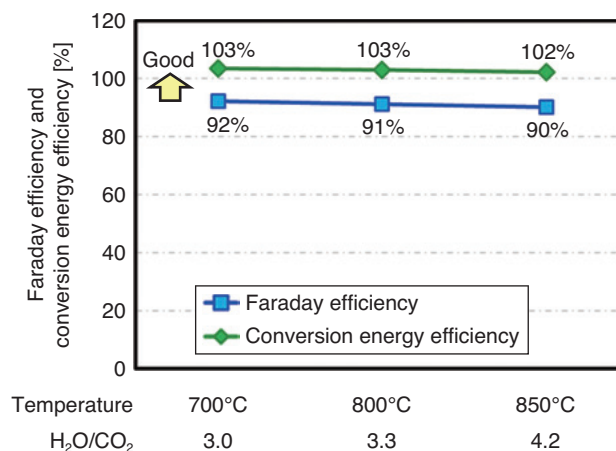


Fig. 12 Faraday efficiency and conversion energy efficiency at various temperatures

in the reaction, was also able to use the heat energy added to the SOEC.

Next, the highest electrolysis current density reached at the various temperatures is shown in Fig. 13. The highest electrolysis current density signifies the source gas throughput quantity, and it was found to reach its highest point at 800°C.

Similarly, the syngas conversion rate at the various temperatures is shown in Fig. 14. The syngas conversion rate shows the same tendency as the electrolysis current density with regard to temperature characteristics, and here also it was found to reach its highest point at 800°C.

It is conceivable that the syngas conversion rate was higher at 800°C than at 700°C because the rise in temperature caused the O²⁻ transport resistance to diminish so that the electrolysis current density increased and the source gas throughput quantity increased. At 850°C, however, the syngas conversion rate decreased, and it is conceivable that this was because the aqueous shift reaction in Eq. (3) caused the amount of H₂O produced to increase, causing an excess of H₂O in the source gas, leading to the decrease in the conversion rate.

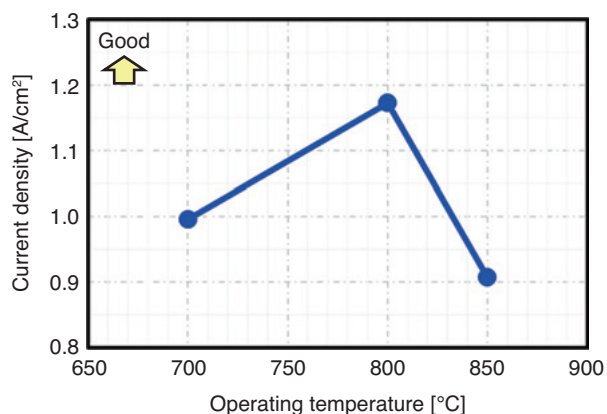


Fig. 13 Current density at various temperatures

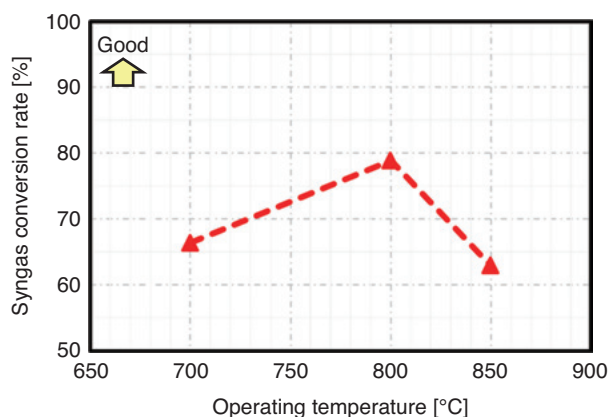


Fig. 14 Syngas conversion rate at various temperatures

7. Conclusion

With the aim of achieving highly efficient carbon recycled CH₄ synthesis, an investigation of the operation criteria and reaction conditions was conducted for an SOEC reactor that yields syngas as a mixture of CO and H₂ produced by co-electrolysis of CO₂ and H₂O. The following results were obtained.

- (1) The results from comparison of ESC and FSC co-electrolysis performance showed that FSC achieved higher CO₂ throughput quantity per unit active area, as expressed by electrolysis current density and syngas conversion rate, and FSC was found to be suited to syngas production.
- (2) Although increasing the input source gas quantity caused electrolysis current density to increase, the syngas conversion rate went down. This suggested that, in order to aim for a highly efficient CH₄ synthesizing system, it would be more advantageous to operate at a lower flow rate even if the size of the SOEC reactor is increased.
- (3) It was found that in order to produce an output gas composition of H₂/CO:3, which is the theoretical molar ratio for CH₄ synthesis, it was necessary to increase the H₂O/CO₂ molar ratio of the source gas to excess H₂O. It was also found that as the electrolysis current density and the syngas conversion rate were both higher at an operating temperature around 800°C, there is a temperature suited to SOEC co-electrolysis operation.

It was found from the above that in order to achieve highly efficient CH₄ synthesis, an SOEC reactor with FSC installed should be operated at around 800°C at a low flow rate. For greater compactness of the SOEC reactor, the desired aim would be a high source gas flow rate, or in other words, high electrolysis current density operation. Under present conditions, however, this conflicts with the syngas conversion rate, and it is hoped that research with a view to optimization can be continued in future.

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Developing Method Using CFD to Design Hydrogen Isolation Structure with Waterproof and Dustproof Filters

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ABSTRACT

With a view to still greater compactness and lighter weight in fuel cell systems, a method using computational fluid dynamics was developed to obtain the filter properties necessary to design a hydrogen isolation structure with waterproof and dustproof filters. The filter properties were identified by measuring gas pressure and concentration on both sides of the filter and then using computational fluid dynamics to simulate the measurements to reproduce their content.

The identified properties were confirmed to be versatile and applicable for other initial conditions of gas type, flow, and concentration.

The design method was also validated by comparing the gas concentrations obtained from measurement by a simple testing apparatus to the results of computational fluid dynamics using identified properties. It was confirmed that satisfactory prediction was possible even under conditions of extremely low gas flow.

As a result, a design method using computational fluid dynamics was developed to realize a safer, more compact, and lighter-weight hydrogen isolation structure with filters.

1. Introduction

In order to enlarge the interior space of the Honda CLARITY FUEL CELL, the fuel cell powertrain was installed under the front hood⁽¹⁾ and hydrogen (H₂) related parts were enclosed in the hydrogen isolation structure shown in Fig. 1, thereby realizing hydrogen safety. The hydrogen isolation structure was made up of three elements: a barrier, hydrogen sensors, and ducts. If the hydrogen-related parts were damaged due to a vehicle collision or other such reason, then hydrogen could leak inside the isolation structure. That leakage would be detected by the hydrogen sensors and the supply of hydrogen would be shut off.

The hydrogen is discharged outside the vehicle through the ducts to reduce its concentration below the flammable threshold, thereby helping ensure the safety of the vehicle.

Even during normal operation, an extremely low flow of around several cm³/min of hydrogen can pass from the stack through the stack sealing material and permeate

the hydrogen isolation structure. This happens because hydrogen molecules are very small. When this hydrogen exceeds the sensor's threshold concentration value for detection of irregularities, then the system will determine that hydrogen leakage has occurred and will shut down even in normal operation. Therefore, the hydrogen isolation structure needs to discharge that minute amount of hydrogen so as not to exceed the sensor threshold. Consequently, computational fluid dynamics (CFD) was employed to confirm for design reasons that the hydrogen concentration would remain below the threshold even if hydrogen permeation were to occur.

Hydrogen safety requirements have to be met, after which it is desirable for the hydrogen isolation structure to be made more compact and lighter in weight. This can be expected to be achieved by attaching multiple waterproof and dustproof filters in place of the ducts to discharge hydrogen. This report will describe the creation of the CFD method for designing the hydrogen isolation structure with filters and the validity of the method.

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2. Development Goals

Figure 2 shows a conceptual diagram of the hydrogen isolation structure with filters. Air-permeable waterproof and dustproof filters are placed in part of the barrier, and hydrogen that is below the flammable threshold is discharged through the filters into the space under the front hood. In this way, the ducts are discontinued and the isolation structure can be made more compact and lighter in weight. The amount of hydrogen discharged through the filters is determined by the following three factors.

- (1) Installed location: Hydrogen is lighter than air, so it accumulates in the upper interior of the isolation structure. Therefore the amount of hydrogen discharged varies according to the installed locations of the filters.
- (2) Area: The greater the area, the more hydrogen will be discharged.
- (3) Material properties: The permeability of the material itself will change the amount of hydrogen that is discharged.

Since the hydrogen permeation amount varies according to the vehicle driving circumstances and ambient

temperature, the properties used for CFD are required to be flexibly applicable under a variety of initial conditions.

When designing with CFD, a three-dimensional model of the inside and outside of the hydrogen isolation structure with filter space that reproduces its location and area should be made first. Next, the filter properties are defined, then the concentration of hydrogen is calculated relative to the amount of hydrogen as initial condition. Then the area and location of the filters as well as the shape of the hydrogen isolation structure are decided to achieve the hydrogen concentration inside it that keeps below the threshold. Therefore, the filter properties should be obtained before CFD.

However, using dustproof and waterproof filters for hydrogen ventilation is not normal usage, so that the phenomenon of low-flow of hydrogen permeating filters is not generally envisioned. Consequently, the properties needed for the present design do not exist and a method for obtaining them has not been established.

For this research, therefore, a new method for obtaining the properties was to be created and a low-flow gas CFD method was to be established.

The aims of this research are defined as follows:

- (1) To establish a method for obtaining the properties for extremely low-flow gas that permeates filters that are provided as input to CFD, and to obtain the values
- (2) To be versatile and usable under different permeation amount and temperature conditions
- (3) To have the CFD predicted values be a quantitative match with measured values

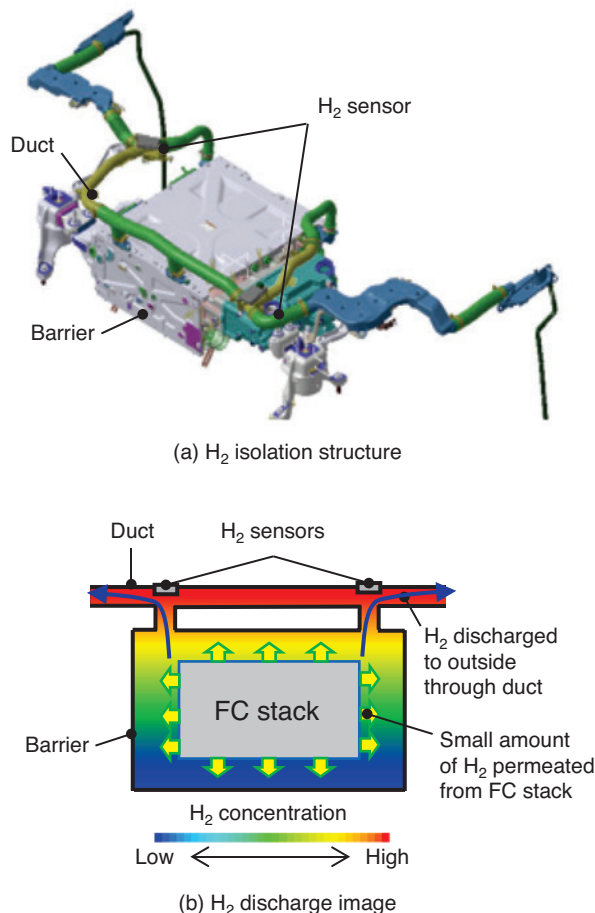


Fig. 1 H₂ isolation structure in Honda CLARITY FUEL CELL

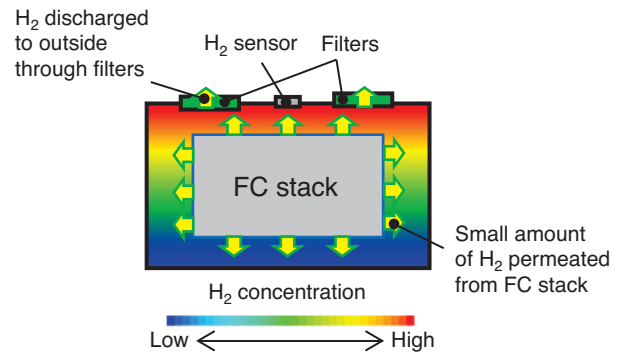


Fig. 2 H₂ discharge image of H₂ isolation structure with filters

3. Creating a Method for Obtaining Extremely Low-flow Gas Properties Through Filters

3.1. Fundamental Equations

The fundamental equations expressing the correlation between the various parameters and the phenomenon of gas transport are shown below. Equation (1) is the advection equation (the Darcy equation), and Eq. (2) is the diffusion equation.

$$\frac{dP}{dx} = -A_1 \mu v + A_2 \frac{1}{2} \rho v^2 \quad (1)$$

$$\frac{\partial C}{\partial t} = D \frac{\partial^2 C}{\partial x^2} \quad (2)$$

A_1 : Coefficient of viscous resistance [$1/m^2$]

A_2 : Coefficient of inertial resistance [$1/m$]

C : Gas concentration [mol/mol]

D : Diffusion coefficient [m^2/s]

P : Pressure [Pa]

t : Time [s]

v : Gas flow velocity [m/s]

x : Distance [m]

ρ : Mixed gas density [kg/m^3]

μ : Mixed gas viscosity [$Pa \cdot s$]

Here the filter properties are A_1 , A_2 , and D . These three properties are determined from the pressure gradient (advection) and the concentration gradient (diffusion) shown in Eqs. (1) and (2). The filter properties can be expressed by advection and diffusion just as in a regular flow field.

The influence of advection is expressed by an equation correlating pressure and flow velocity, and it is typically shown in a quadratic function. The coefficients of that quadratic function are the coefficient of viscous resistance A_1 and the coefficient of inertial resistance A_2 .

The influence of diffusion, as shown in the diffusion equation, arises from the concentration difference. In a medium or larger flow field, advection dominates this flow field with large v on the right side of Eq. (1). Influence of advection decreases and the influence of diffusion increases in a low flow field. Therefore it becomes important to clarify the value of D in distinction to A_1 and A_2 in a minute flow field. For that purpose, a measuring apparatus was devised and measurements of A_1 and A_2 were made.

3.2. Pressure and Concentration Measuring Apparatus and Measurement Method

To isolate the values of A_1 , A_2 , and D , a filter was installed between two spaces and a measuring apparatus was created with the capability to obtain time series data on pressure and concentration simultaneously on both sides. A method was further devised to reproduce the measurements by CFD and obtain the CFD parameters that are needed to develop the design.

Figure 3 shows the measuring apparatus that makes pressure and concentration measurements on both sides of the filter. A straight channel of 20-mm inner diameter was made the flow path. To obtain the properties of the mixture gas, gas inlets are placed on the left and right. Two different types of gas, Gas A and Gas B, are sealed into the flow channel with a closed valve separating them. Then steps are taken to simultaneously supply Gas A, close the Gas A outlet and the Gas B inlet, and open the valve. This creates

a unidirectional flow from the A side to the B side through the filter and all the gas is discharged to the Gas B outlet.

The filters are fixed with aluminum plates as shown in Fig. 4. Both types of filter, sheet filter shown in (a) and button filter shown in (b) can be attached. The diameter of the mounting hole of the filter was 10 mm so as not to distribute horizontally to the filter surface. The plates with filters assembled are sealed between flanges with O-rings on both sides. Concentration sensors and pressure sensors are installed and measurements are made on both sides of the filter. When making measurements, the room temperature and inlet Gas A temperature should be the same so as not to incur measurement error due to the temperature difference.

Concentration sensors (C_2 and C_3) and pressure sensors (P_2 and P_3) are both located on both sides of the filter. Time series measurements were made starting immediately after the valve was opened and until all sensor values reached a steady state.

At this time, the relationship between the pressure drop $P_2 - P_3$ measured under multiple gas flow conditions and the gas flow velocity, which was gas flow divided by cross-sectional area, is expressed in the form of the quadratic function shown in Eq. (3) and Fig. 5. At this time, a and b are coefficients.

$$\frac{P_2 - P_3}{x} = av + bv^2 \quad (3)$$

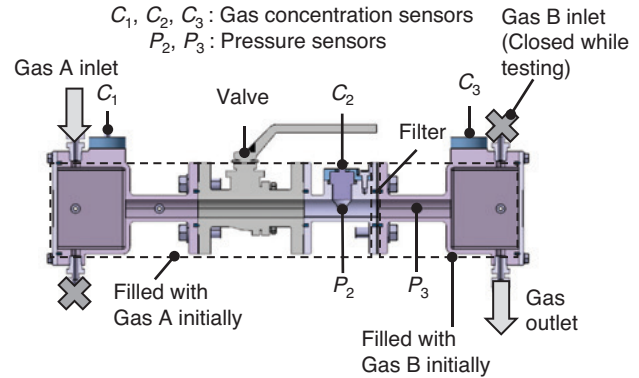


Fig. 3 Filter pressure drop and gas concentration measuring apparatus

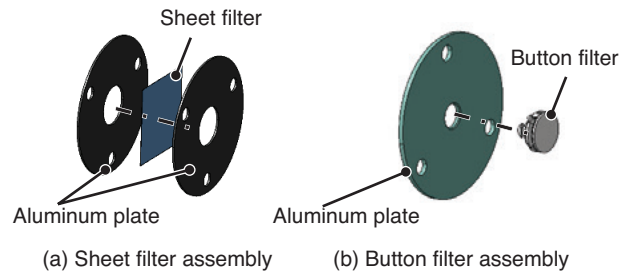


Fig. 4 Filter assembly

Since Eq. (3) is identical to Eq. (1), it follows that these relational expressions apply:

$$-A_1\mu = a \quad (4)$$

$$A_2 \frac{1}{2} \rho = b \quad (5)$$

From these, A_1 and A_2 can be derived.

3.3. Identification of Diffusion Coefficient D by CFD Analysis

Concentrations on both sides of the filter are not subject to difference like pressure, but the same value. C_2 and C_3 approach the same value over time. For this reason, a different method is needed to obtain D .

Therefore, the measurement above was reproduced in CFD to identify D that could obtain the same time series of C_2 and C_3 . A mesh model of the space inside the measuring apparatus and the filter shown in Fig. 3 was created. The Gas A space was filled with Gas A and Gas B space was filled with Gas B as the initial condition. From that state, Gas A was introduced at the specified flow and concentration, and transient calculations were performed to simulate the Gas A flow to the Gas B outlet through the filter (Fig. 6).

By assigning the values of A_1 , A_2 , and D to the filter portion of the CFD, the kinds of phenomena measured in testing are also expressed in CFD. That is, the pressure difference between P_2 and P_3 and the C_3 concentration rise delay due to the filter inhibiting the gas transport (Fig. 7). This delay varies according to A_1 , A_2 , and D . Because A_1

and A_2 are identified from measurement, the remaining D is identified by CFD. This method can be applicable even under low-flow gas conditions because a larger difference between C_2 and C_3 can be obtained under the conditions.

In identifying D , multiple CFD results with different values of D are needed, as in Fig. 8. The square of the difference between measured values and CFD values in C_2 and C_3 was calculated every second and each Δ_{sum}^2 was obtained to integrate them by the specified time t_e . Δ_{sum}^2 is as shown in Eq. (6).

$$\Delta_{\text{sum}}^2 = \sum_{t=0}^{t_e} (C_{\text{test}} - C_{D_i})_{C_2}^2 + \sum_{t=0}^{t_e} (C_{\text{test}} - C_{D_i})_{C_3}^2 \quad (6)$$

C_{D_i} : Gas concentration [mol/mol] in CFD result using diffusion coefficient D_i at time t

C_{test} : Gas concentration [mol/mol] in measurement result at time t

t_e : Specified time [s]

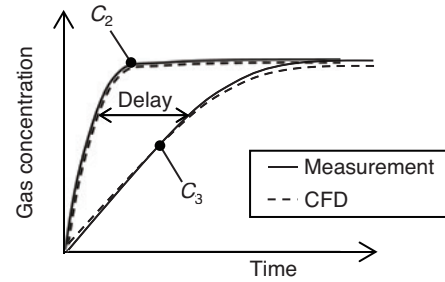


Fig. 7 Image of C_3 gas concentration rise delay against C_2

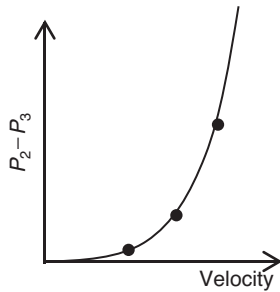


Fig. 5 Gas velocity and filter pressure drop

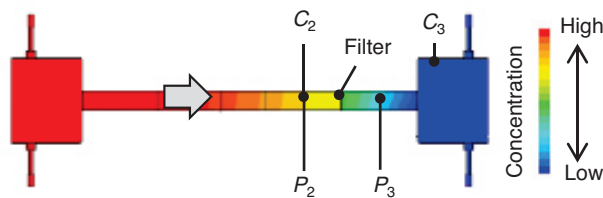
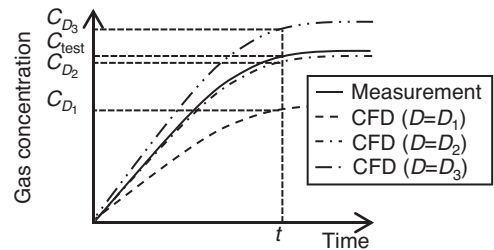
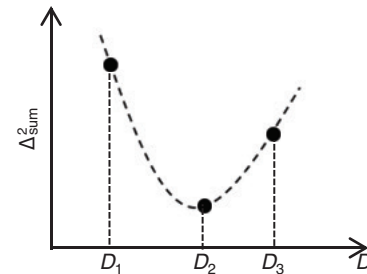


Fig. 6 Image of Gas A concentration distribution



(a) Image of gas concentration profiles with different D_i s



(b) Image of Δ_{sum}^2 against D_i

Fig. 8 Diffusion coefficient identification method

Taking the diffusion coefficient D_i used in each calculation as the horizontal axis and Δ_{sum}^2 from each calculation as the vertical axis, the D that yields the lowest Δ_{sum}^2 was determined.

4. Filter Properties Identification Results and Discussion

4.1. Properties Identification Measurement and CFD Conditions

Table 1 lists the measurement and CFD conditions. The filter used was the TEMISH⁽²⁾ Z3-NTF210SE manufactured by Nitto Denko Corporation. Gas A was a mixture gas of helium (He) and nitrogen (N₂), and Gas B was nitrogen alone to obtain the properties of the mixture gas with helium and nitrogen with respect to the filters.

Gas properties of mixture gas of hydrogen and air were needed for designing the fuel cell system. However, helium instead of hydrogen and nitrogen instead of air were used for safety in this measurement because the flammable concentration threshold of hydrogen was extremely low.

In this measurement, the helium concentration was fixed at 4% and the filter properties were identified from three different Gas A flow results of 5, 10, and 20 cm³/min. The temperature of the room as well as of Gas A and Gas B was fixed at 25°C.

4.2. Solver

FLUENT was used as the solver and transient calculations were carried out. This CFD model has two parts, flow channel and filter, which have their own gas permeabilities. Thus user-defined functions were created to define the mutual diffusion coefficient of mixture gas D_0 to the channel and A_1 , A_2 , and D to the filter. The Chapman-Enskog equation in Eq. (7) was used to calculate D_0 . A_1 and A_2 were given from the measurement and D was given as an assumed value.

The formula for the calculation of D_0 and a list of the variables are shown below.

$$D_0 = \frac{266T^{3/2}}{PM^{1/2}\sigma^2\Omega_D} \quad (7)$$

D_0 : Mutual diffusion coefficient of the mixture gas in the flow channel [m²/s]

M : Molecular weight of the mixture gas [kg/kmol]

σ : Intermolecular potential parameter

Ω_D : Collision integral

T : Temperature [K]

4.3. Identification Results

Figure 9 shows the measurement results of the filter pressure drop $P_2 - P_3$ under three different Gas A flow rates. Dots are the measurement, while the curve was the CFD using identified A_1 and A_2 from the measurement using Eqs.

(4) and (5). The values for A_1 and A_2 were identified through this operation.

Figure 10 shows the results of Δ_{sum}^2 around the identified D . The value for D was identified through this operation.

Using the results of this identification, a comparison was made of the CFD results with the results from measurement of the helium concentrations C_1 , C_2 , and C_3 . Figure 11 shows the results from comparison of concentrations under the three conditions shown in Table 1. The comparison indicated that the CFD results using identified values could satisfactorily reproduce the measured helium concentration profiles under any flow conditions.

4.4. Versatility Check of Identified Results

Versatility of identified filter properties was checked to determine availability to CFD for prediction of filter pressure drop and concentration even under other initial conditions.

Table 2 shows measurements and CFD conditions for verification. Conditions 1 through 3 are for verification related to helium concentration differences. Conditions 4 through 9 are for verification related to gas type using oxygen instead of nitrogen. The identified properties can be checked for their versatility of various flows, concentrations, and gas types.

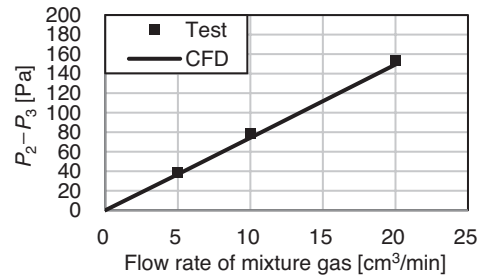


Fig. 9 Comparison of measurement (test) and CFD filter pressure drop with identified A_1 and A_2

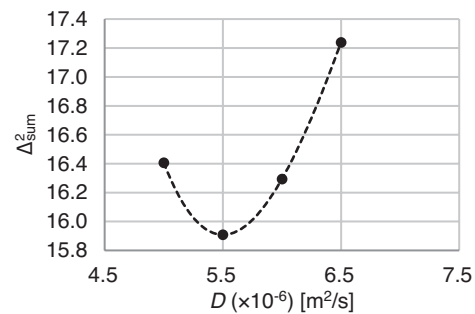


Fig. 10 Δ_{sum}^2 against D_i

Table 1 Measurement and CFD conditions for filter characteristics identification

Gas A				Gas B	Temperature [°C]
He [cm ³ /min]	N ₂ [cm ³ /min]	He+N ₂ [cm ³ /min]	He concentration [%]		
4.8	0.2	5	4	N ₂	25
9.4	0.4	10			
19.2	0.8	20			

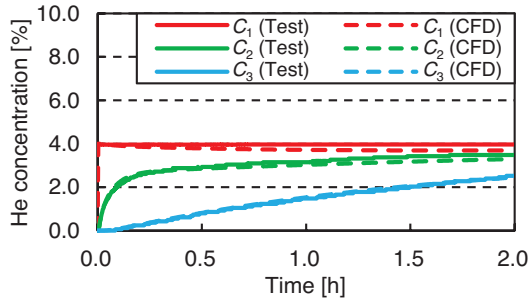
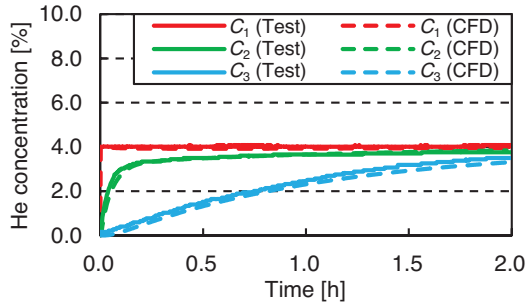
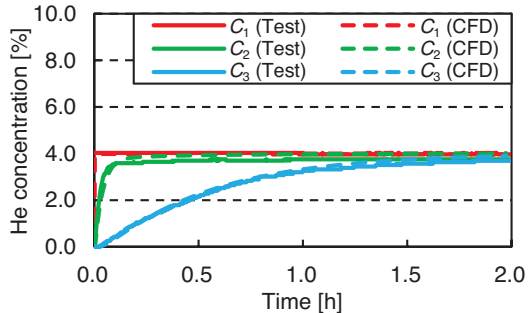
(a) 5 cm³/min(b) 10 cm³/min(c) 20 cm³/minFig. 11 Comparison of measurement (test) and CFD He concentration profiles with identified D at three sensors

Table 2 Measurement and CFD conditions for validation

No.	Gas A				Gas B	Temperature [°C]
	He [cm³/min]	N ₂ [cm³/min]	He+N ₂ [cm³/min]	He concentration [%]		
1	4.6	0.4	5	8	N ₂	25
2	9.2	0.8	10			
3	18.4	1.6	20			
	He [cm³/min]	O ₂ [cm³/min]	He+O ₂ [cm³/min]	He concentration [%]	O ₂	
4	4.8	0.2	5	4		
5	9.4	0.4	10			
6	19.2	0.8	20			
7	4.6	0.2	5	8		
8	9.2	0.4	10			
9	18.4	0.8	20			

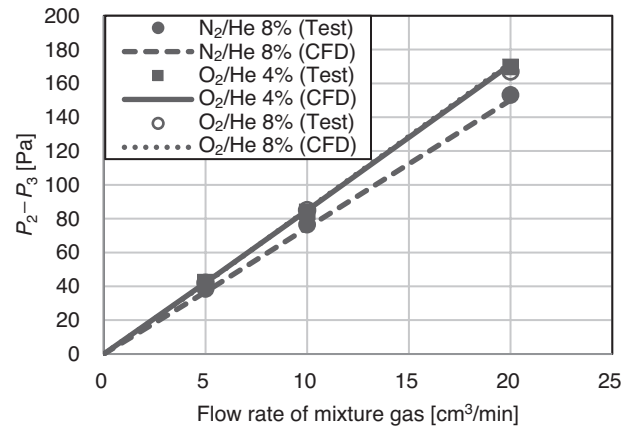
Fig. 12 Comparison of measurement (test) and CFD filter pressure drop with identified A_1 , A_2 , and D

Figure 12 shows the results from comparison of the filter pressure drop measurement values under these nine conditions with the CFD results using the identified values. As shown by these results, it has been confirmed that the identified properties can be used with different concentrations and gas types and still predict the filter pressure drop accurately.

Next, Fig. 13 shows the results from comparison of measurement results and CFD results for concentrations C_1 , C_2 , and C_3 . The CFD results using the identified properties satisfactorily reproduce the measurement results under all conditions.

It was successfully verified that the properties obtained by this method were highly versatile and reliable for use with other concentrations and flow conditions.

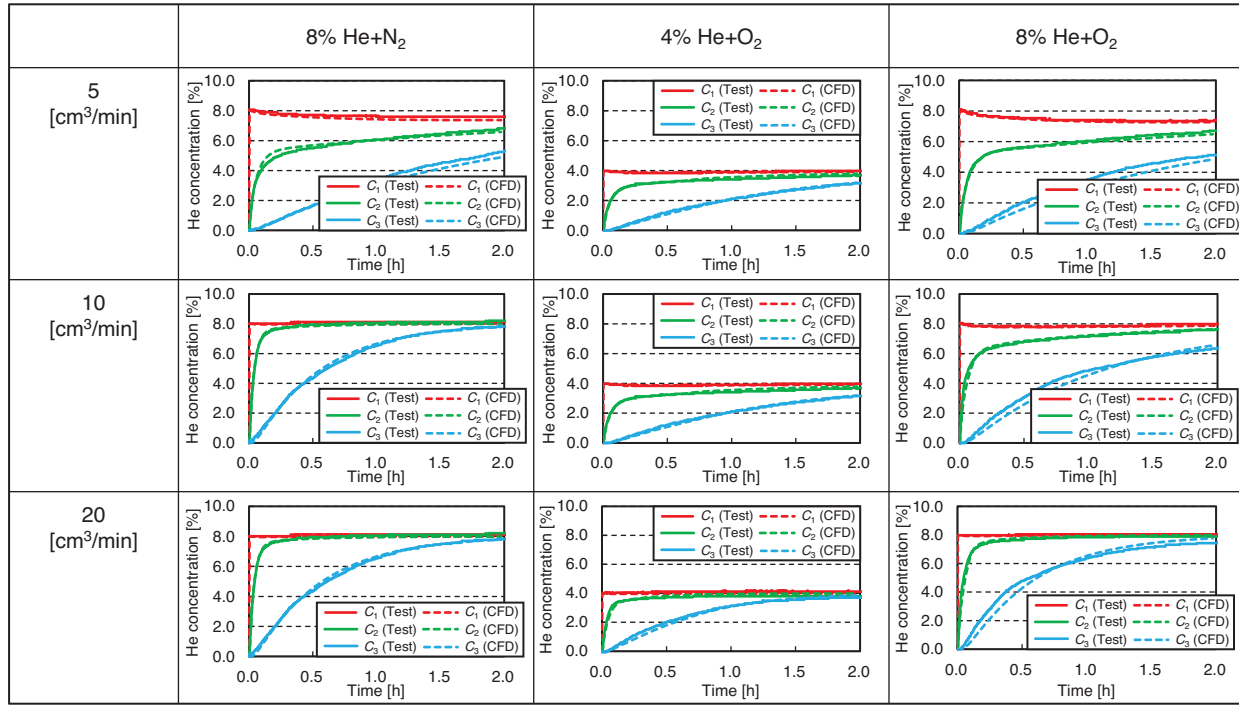


Fig. 13 Comparison of measurement (test) and CFD He concentration profiles with identified D at three sensors for validation

5. Results from Validation of the Design Method

5.1. Overview of Validation

Figure 14 shows a testing apparatus that provides a simple simulation of the hydrogen isolation structure with filters together with a conceptual image of the concentration distribution according to CFD.

The testing apparatus consists of piping for gas inflow, filters with properties obtained in the previous chapter, a box of cuboid shape that simulates the hydrogen isolation structure for the FC stack, and concentration sensors. The inflow of gas is limited to piping at one location only, and helium gas of 100% concentration is made to flow into the box interior from that location. Any other piping and any passages other than the filters are all sealed off. The helium gas that diffuses into the box interior mixes with air as it spreads throughout the whole interior space, then flows out of the box through the filters installed in the sides of the box. When this happens, air also flows into and out of the box through the filters according to the concentration differential between the interior and exterior outside of the box until finally a constant concentration field is created. The helium concentrations at the five locations from Sensor 1 to Sensor 5 in this steady state were measured and validation was performed to determine if these concentrations could be predicted by CFD using the filter properties acquired in the present research.

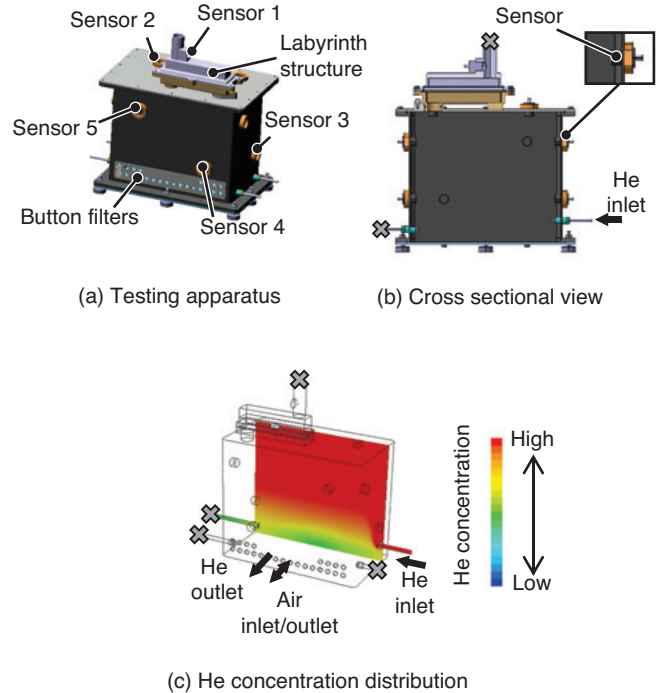


Fig. 14 Testing apparatus and He concentration distribution

Validation was performed under the three conditions shown in Table 3. Condition 1 had no filters so that gas was free to move in and out through 20 filter mounting holes to evaluate CFD performance with extremely low flows. In Conditions 2 and 3, the number of filters and the amount of inlet helium gas were varied from Condition 1 to evaluate the validity of the acquired filter properties and the CFD method. The holes in the portion where filters are not attached under Condition 2 were sealed with aluminum tape. Temperatures were made a uniform 25°C.

5.2. Validation Results

Figure 15 shows a comparison of the results of sensor concentration measurements under every condition with the CFD concentration predicted under the same conditions.

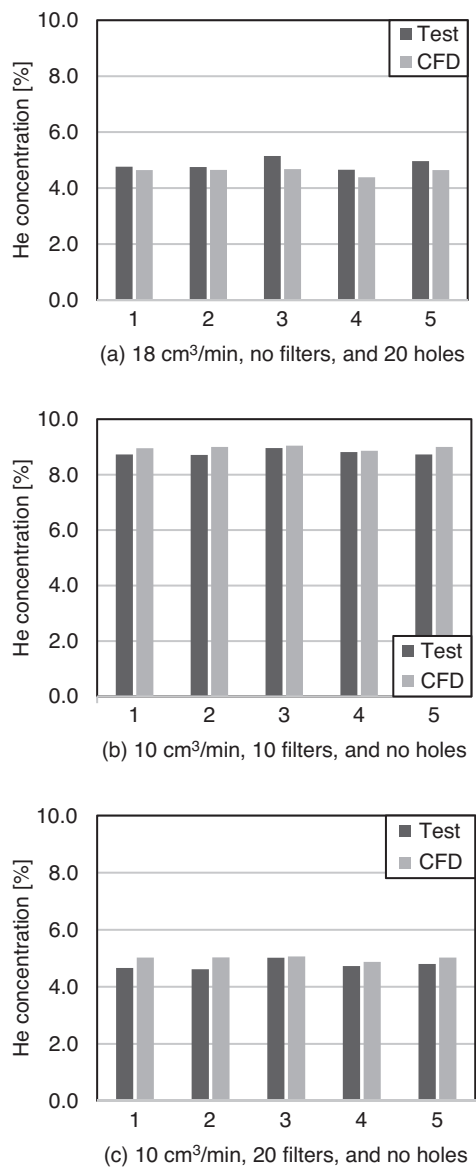


Fig. 15 Comparison of measurement (test) and CFD He concentration at five sensors

Table 3 Measurement and CFD conditions for validation

No.	He [cm ³ /min]	Number of filters	Number of holes
1	18	0	20
2	10	10	0
3	10	20	0

The comparisons in (a) and (c) should have been made at the same flow rate. However, in the test without filters at 10 cm³/min, concentration values below the lower limit guaranteeing sensor accuracy were measured so that the flow in (a) was changed to 18 cm³/min.

The result was that concentrations at high flow without filters were lower than at low flow with filters, and this successfully confirmed that the permeation of air and helium was inhibited by the filters. It was also successfully confirmed in the comparison of (b) and (c) that the fewer the number of filters, the higher the concentration of helium. Although the differences were small in the results for different sensor placement, the simulation successfully reproduced the tendency toward a higher concentration at Sensor 3, near the inlet.

As shown by these results, the CFD results were confirmed to satisfactorily predict the test results under any conditions or sensor locations. This confirmed the validity of the filter properties acquired in the present research and of the CFD method.

6. Conclusion

The results using the filter properties identified by CFD were used to compare the measurements and CFD results under different conditions for validation. Through this research, the design method using CFD for a hydrogen isolation structure with waterproof and dustproof filters was successfully validated.

- (1) The filter properties were identified by measuring gas pressure and concentration on both sides of the filter and then using CFD to reproduce the measurements.
- (2) The design method was confirmed to produce satisfactory prediction even under extremely low gas flow conditions by comparing the gas concentrations from the measurement by a simple testing apparatus to the CFD using identified properties.
- (3) The identified properties were confirmed to be applicable for other initial conditions of gas type, flow, and concentration.

Hydrogen is a flammable gas that is invisible and so it is subject to constraints on location and conditions when testing. A simple method of predicting hydrogen concentrations such as in this research is considered useful in designing a hydrogen isolation structure that is safe as well as compact and light in weight.

This CFD method is also applicable for predicting hydrogen concentrations in an entire fuel cell vehicle, and it can be expected to be an important technology for the realization of a safe hydrogen society.

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B-pillar Load Transfer Mechanism in Side Collision

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ABSTRACT

Based on side collision testing that the Insurance Institute for Highway Safety in the United States has been conducting, a technology was created that can reduce the amount of B-pillar deformation and also generate a high load. In a side collision, it is necessary for the greater part of the load received from the barrier to be supported by the bending of the B-pillar. This was addressed by creating a new load path in the B-pillar. The new load path is able to support the lateral load input from the barrier as an axial load in a downward direction in the vehicle body. This increases the load from the initial time of collision and reduces bending deformation. A load transfer mechanism in the B-pillar cross-section was added for the axial load and this was realized by a load path to the side sill. The structure of the load transfer mechanism was decided through theoretical calculation and simulation. A prototype was fabricated with a load transfer mechanism in a mass-produced B-pillar. Drop tests with the prototype proved that the load generated on the B-pillar was doubled so that the amount of deformation could be reduced to one-fifth.

1. Introduction

Demand for automobile safety performance continues to intensify, and preventative safety technology that helps stop accidents before they happen is making remarkable progress. Meanwhile, there is also the constant enhancement of collision safety standards that mitigate damage when accidents occur, and the Insurance Institute for Highway Safety (IIHS) in the United States continues to conduct research focusing on correlations between safety performance rating points and death rates⁽¹⁾. Future application of new testing criteria (hereinafter, the New IIHS MDB) is planned in order to enhance vehicle safety in side collisions⁽²⁾.

Figure 1 shows an overview of the New IIHS MDB. The weight of the deformable barrier (hereinafter, barrier) is increased from 1500 kg to 1900 kg, and the velocity at which the barrier collides with the vehicle is also increased from 50 km/h to 60 km/h⁽²⁾. This is equivalent to a 1.8-fold increase in collision energy.

Toward the realization of a carbon-free society, it is expected that future automobiles will have equipment such as large-capacity batteries and hydrogen tanks. This will

increase the vehicle body weight, calling for even more energy absorption⁽³⁾.

The authors therefore anticipated future themes for side collisions, and focused on the New IIHS MDB, which is presently the highest energy testing criteria.

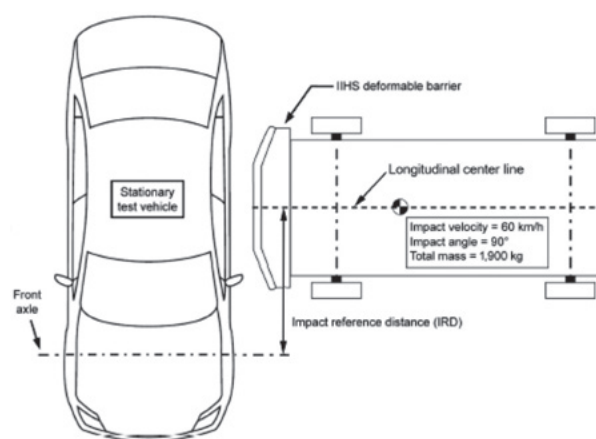


Fig. 1 Deformable barrier alignment with test vehicle

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In the present vehicle body structure, the input load from a side impact originates in the B-pillar (the pillar between the front and rear seats) and is transferred to the side sill, roof side rail, floor cross member, and roof cross member⁽⁴⁾. The barrier collides with the lower part of the vehicle body, so bending deformation occurs with the largest deformation on the lower side of the B-pillar. Safety performance is rated by the two items of distance from the location with the largest deformation to the seat center, referred to as structure, and the dummy injury criterion. Structure is classified into four levels by size, and smaller amounts of deformation receive better ratings. From these perspectives, the B-pillar is a key structure for side collisions.

Research on the B-pillar so far has included control of the deformation mode by deformation mode prediction⁽⁵⁾ and the partial softening method⁽⁶⁾, and weight reduction by application of high-tensile steel⁽⁷⁾⁻⁽¹⁰⁾, etc. However, none involved major advances in the B-pillar structure, and methods such as addition of reinforcing parts and increased thickness were used to generate a large load with small deformation. These lead to increased vehicle body weight, and there are also limits to reinforcement in consideration of limitations on welded plate thickness.

This research therefore took as a representative case the New IIHS MDB, which is the highest energy criteria in side collision testing, and created a new load transfer mechanism that enables the B-pillar to generate a high load with small deformation.

2. B-pillar Load Enhancement Concept

2.1. Load Generation Method of Conventional Structure

Figure 2 shows a diagram of the deformation when input is received from the barrier. The barrier is composed of a protruding part indicated in red that simulates a front bumper beam and a honeycomb structure general part indicated in blue. It deforms according to the shape of the vehicle body. When the B-pillar receives the distributed

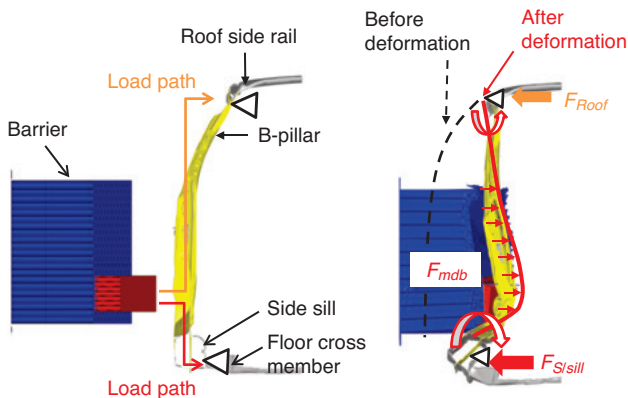


Fig. 2 Deformation mode of B-pillar in conventional structure

load from the barrier (hereinafter, F_{mdb}), the lower side functions as a load path to the roof side rail and the side sill while undergoing the largest bending deformation.

Force applied to a part in a bending direction is disadvantageous in terms of load generation compared to force in an axial direction⁽⁴⁾. In a high-energy collision such as the New IIHS MDB, the B-pillar bending deformation increases further and the tensile inputs to the roof side rail and side sill support parts also increase. The conventional structure is mostly subject to bending deformation, so a large amount of deformation is needed before the highest load is reached. For this reason, the general method is to increase the B-pillar load and reduce the amount of B-pillar deformation and the load on the support parts. This is done by increasing the size and thickness of the B-pillar cross section, which contribute greatly to bending, and by adding reinforcing parts.

2.2. New Conceptual Approach to Load Generation

The basic principle for increasing the load generated by the B-pillar is to apply the balance of moments to transfer the lateral load received from the barrier to an axial load in the vertical direction. Figure 3 shows the schematic diagram of this principle. A clockwise moment is generated when a lateral load F_y is applied to the part fixed to the side sill. This moment is balanced with the counterclockwise moment generated by the vertical load F_z received from the side sill. That is to say, the lateral load can be increased by increasing the vertical axial load.

Figure 4(a) shows a structural diagram of application of this basic principle to the B-pillar. A two-part mechanism (hereinafter, load transfer mechanism) composed of a loading part (hereinafter, L/Part) and a support part (hereinafter, S/Part) was added in the cross section in the lower part of the B-pillar. The L/Part and S/Part contact each other at the slope, as indicated by the blue arrow, and are joined only halfway around the circumference in the

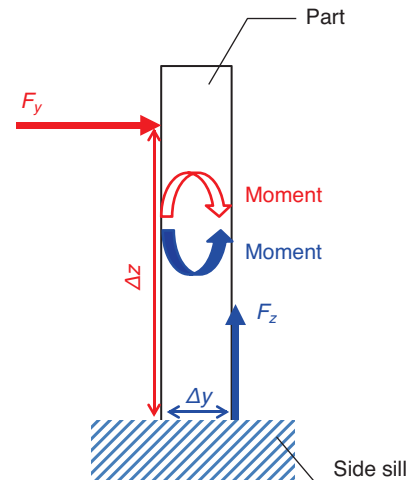


Fig. 3 New basic principles of load transfer

range indicated by the red line starting from junction O in the center of the slope. The S/Part is placed on the upper surface of the side sill via a base plate.

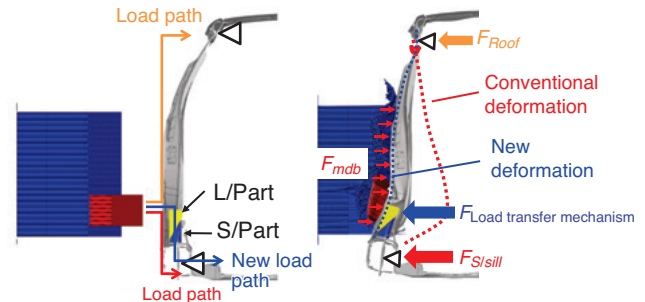
When the lateral load from the barrier is applied, the B-pillar contacts the L/Part and the load transfer mechanism deforms, as shown in Fig. 4(b). The L/Part rotates around junction O, converting the lateral load to the downward direction and transferring it to the S/Part. The semicircular range of the S/Part on the cabin interior side is pressed from the L/Part, generating an axial load in the vertical direction. As a result, the load transfer mechanism creates a new load path that transfers the load to the side sill and floor cross member. Here, F_{y-LP} indicates the lateral load applied from the B-pillar to the L/Part and F_{z-SP} indicates the axial load generated by the S/Part. That is to say, F_{y-LP} and F_{z-SP} have a balanced relationship between the moments originating from junction O, thereby F_{y-LP} can be determined by setting F_{z-SP} and the respective lengths Δy and Δz from junction O over which F_{y-LP} and F_{z-SP} act.

Figure 5(a) shows a conceptual drawing of the load and deformation when the load transfer mechanism is installed in the B-pillar. The blue arrow in the left figure shows the new path by which the load from the barrier is transferred via the load transfer mechanism to the side sill. The axial load of the load transfer mechanism increases the load generated by the B-pillar, which greatly crushes the barrier. Compared with the red dotted line that indicates the conventional structure with large bending deformation of the B-pillar, the amount of deformation is greatly reduced

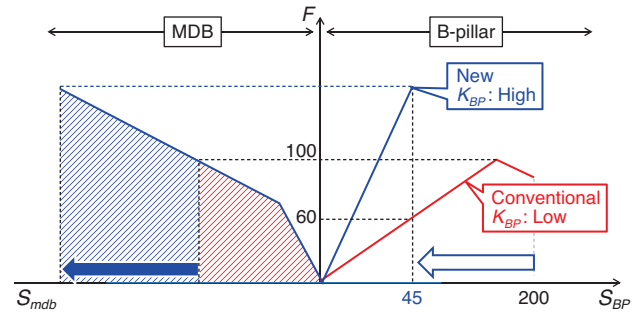
in the new structure indicated by the blue dotted line.

Figure 5(b) shows a conceptual diagram of the effects of the load and amount of deformation. The vertical axis represents the load and the horizontal axis represents the amounts of barrier deformation S_{mdb} and B-pillar deformation S_{BP} . The new structure indicated by the blue line reaches the highest load with a small amount of deformation. As a result, the load gradient (hereinafter, K_{BP}) obtained by dividing the highest load by the amount of deformation becomes larger than that of the conventional structure. The amount of energy absorbed by the barrier increases as indicated by the blue shaded area, so the burden on the B-pillar decreases.

As shown above, the B-pillar load enhancement concept is to use a load transfer mechanism to create a transfer path that supports the load in the axial direction and to increase the load gradient K_{BP} .

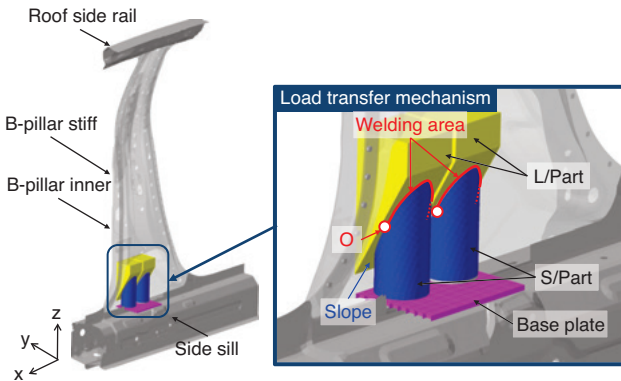


(a) Conceptual drawing of load paths

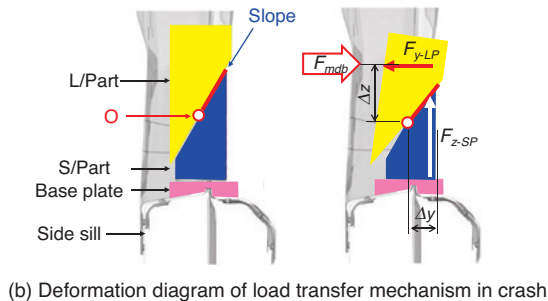


(b) Relationship between force and stroke

Fig. 5 Deformation mode of B-pillar in new structure



(a) Structure of load transfer mechanism



(b) Deformation diagram of load transfer mechanism in crash

Fig. 4 Diagram of new structure

3. Load Transfer Mechanism Performance Requirements and Creation of Design Method

Section 3.1. establishes the target load for realizing the concept described in Chapter 2. Subsequent sections formulate the slope angle, the lateral clearance with the B-pillar, and the S/Part collapse load, which are the three main design variables of the load transfer mechanism, and create a design method. The combined load with the load

Table 1 Numerical values used to derive targets

Variable	Value	Variable	Value	Variable	Value
$P_{mdb,1}$	140 kPa	$A_{mdb,1}$	$2.8 \times 10^5 \text{ mm}^2$	K_{air}	1.3
$P_{mdb,2}$	325 kPa	$A_{mdb,2}$	$1.4 \times 10^5 \text{ mm}^2$	M_{mdb}	1900 kg
$P_{mdb,3}$	325 kPa	$A_{mdb,3}$	$1.4 \times 10^5 \text{ mm}^2$	M_{CAR}	1500 kg
$P_{mdb,4}$	325 kPa	$A_{mdb,4}$	$1.4 \times 10^5 \text{ mm}^2$	V_0	60 km/h
$P_{mdb,5}$	1100 kPa	$A_{mdb,5}$	$5.0 \times 10^4 \text{ mm}^2$	K_{BP}	2.5 kN/mm
				K_{DR}	2.0 kN/mm

black line in Fig. 8. In order to examine only the B-pillar component, this research did not input a load from the door. Therefore, the load of 51 kN received from the barrier in area V was excluded, and the B-pillar targets were $S_{BP} = 46 \text{ mm}$ and $F_{BP} = 115 \text{ kN}$, as shown by the red line in Fig. 8. In addition, Fig. 5(b) shows that the B-pillar with the conventional structure generates a load of around 60 kN when the target amount of deformation of 46 mm is reached. Therefore, the F_{y-LP} demanded of the load transfer mechanism was determined to be $115 - 60 = 55 \text{ kN}$.

3.2. Angle of Load Transfer Mechanism Slope

Next, an examination was conducted to determine the angle of the load transfer mechanism slope. Figure 9 shows the relative positions of the load transfer mechanism and the barrier protruding part. Assuming that the load transfer mechanism receives a uniformly distributed load from the barrier, the load input point is the center of the height between the lower edge of the barrier and the upper edge of the L/Part (hereinafter, O'). Here, expressing the S/Part radius as Δy , the distance from the junction O to O' as Δz , the distance from O' to the lower edge of the S/Part as H , and the slope angle of the contact surface between the L/Part and S/Part as θ , F_{y-LP} , and F_{z-SP} can be expressed according to the balance of moments, as shown in Eqs. (5) and (6).

$$F_{y-LP} \cdot \Delta z = F_{z-SP} \cdot \Delta y \quad (5)$$

$$\Delta z = H - (h + \Delta y \tan \theta) \quad (6)$$

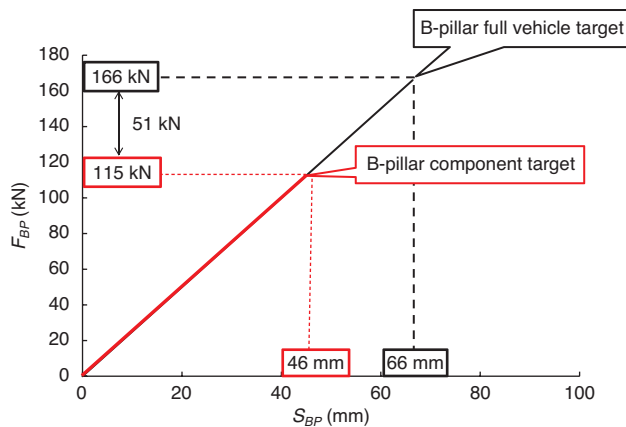


Fig. 8 B-pillar target displacement and load

The size of the load transfer mechanism was set at $Z = 150 \text{ mm}$ and $Y = 50 \text{ mm}$ in consideration of placement in the lower part of the B-pillar. In order for the cylindrical part of the S/Part on the cabin interior side to generate a stable axial load, the height h from the lower surface was set at 20 mm to achieve the shortest dimension on the cabin exterior side of the S/Part. It can be estimated from Eq. (6) that F_{z-SP} will be 175 kN when $\theta = 30 \text{ deg}$ and 112 kN when $\theta = 60 \text{ deg}$. Theoretical examination indicated that increasing the slope angle θ reduces Δz , enabling the generation of a large F_{y-LP} with a small F_{z-SP} and an efficient increase in the B-pillar load.

The combined performance with the load transfer mechanism installed in the B-pillar was confirmed by simulation using LS-DYNA, and the slope angle that results in generation of the highest load at the target amount of deformation was examined.

Figure 10(a) shows the analytical model. The B-pillar is joined to the roof side rail and the side sill. The load transfer mechanism is placed in the B-pillar opening with the lateral position set in the center part of the side sill. The boundary conditions of the front and rear end faces of the roof side rail and side sill are defined as completely fixed to exclude the influence of rotation or deformation of the boundary parts. The B-pillar model is defined by shell elements with a mesh size of 5 mm. By contrast, the barrier model is defined by a rigid body element with the same shape as the protruding part of the New IIHS MDB, and forced displacement of 1 m/s is applied to obtain the relationship between the quasi-static load and displacement of the B-pillar. Evaluation was performed for the three cases of a slope angle θ of 30 deg, 45 deg, and 60 deg, as shown in Table 2. Figure 10(b) shows the shape of each load transfer mechanism. The L/Part is defined by a rigid body element, and the S/Part is defined by a shell

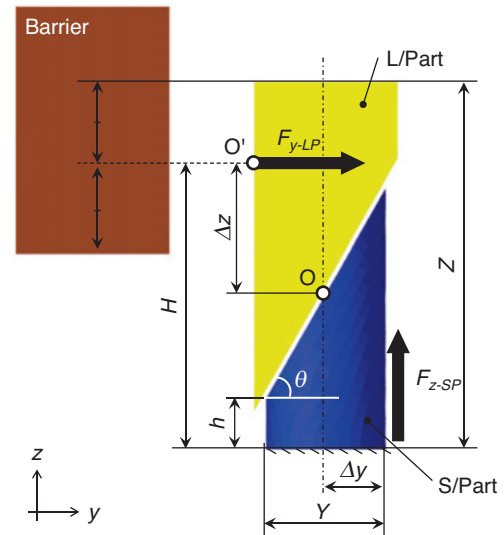


Fig. 9 Schematic diagram of load transfer mechanism

Table 2 Parameter used for investigation of different θ

	Model 1	Model 2	Model 3
θ	30 deg	45 deg	60 deg
$\Delta y/\Delta z$	0.31	0.36	0.49

element simulating 590 MPa-grade high-tensile steel with a thickness of 1.5 mm.

Figure 11 shows the relationship between the B-pillar load F_{BP} and amount of deformation S_{BP} . In the models with the load transfer mechanism installed, the load began to increase at the timing of the amount of deformation indicated by the arrow. This roughly coincided with the lateral clearance of 20 mm between the B-pillar and the load transfer mechanism shown in Fig. 10. As in the theoretical calculations, the load for the case with $\theta = 60$ deg increased to around double that of the conventional model and exceeded the target load of 115 kN at the target amount of deformation of $S_{BP} = 46$ mm indicated by the black dotted line.

It was found from the above that when the S/Part buckling load F_{z-SP} is constant, F_{BP} can be changed according to the angle of the load transfer mechanism slope. In subsequent sections, the examination proceeded with the slope angle $\theta = 60$ deg.

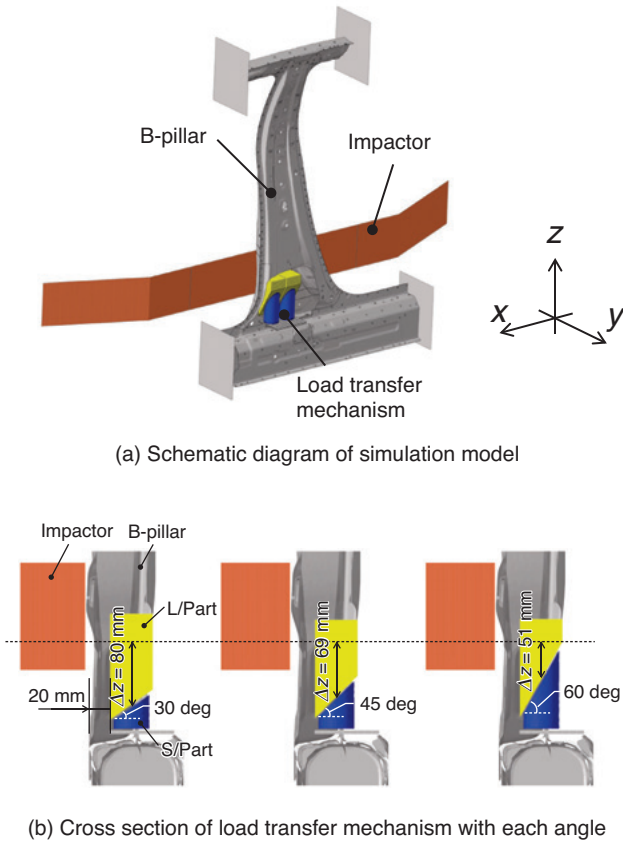


Fig. 10 Appearance of CAE model

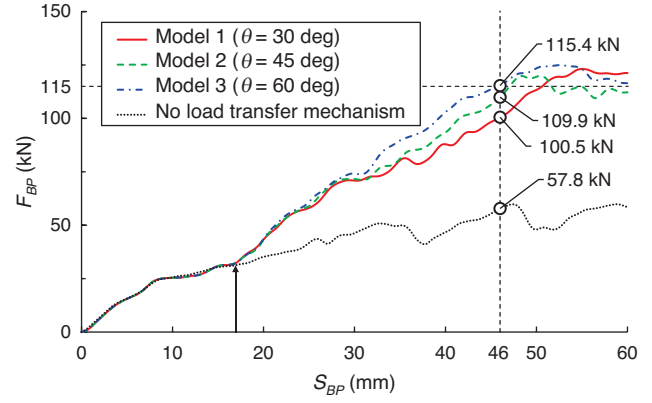


Fig. 11 Load displacement curves at different angles

3.3. Lateral Clearance between Load Transfer Mechanism and B-pillar

It was suggested in the previous section that the timing at which the load increases is related to the lateral clearance between the B-pillar and the load transfer mechanism. It is thought that the load path to the side sill activated at the timing when the B-pillar deformed to the extent that the clearance disappeared and the B-pillar contacted the load transfer mechanism.

Therefore, a clearance that enables the generation of a higher load at the target amount of deformation was obtained by simulation. Model 3 with the slope angle determined in the previous section was used to conduct simulations for a total of four cases with clearance CL of 5, 20, 30, and 50 mm between the load transfer mechanism and the B-pillar, as shown in Fig. 12.

Figure 13 shows the relationship between the B-pillar load F_{BP} and amount of deformation S_{BP} . F_{BP} was found to rise the earliest for the case with the small clearance of 5 mm. However, the case with $CL = 20$ mm exhibited the highest F_{BP} at the target amount of deformation. Figure 14 shows the deformation of the load transfer mechanism and B-pillar at the target amount of deformation. The B-pillar and the load transfer mechanism make contact when CL is from 5 mm to 30 mm, but do not make contact at $CL = 50$ mm.

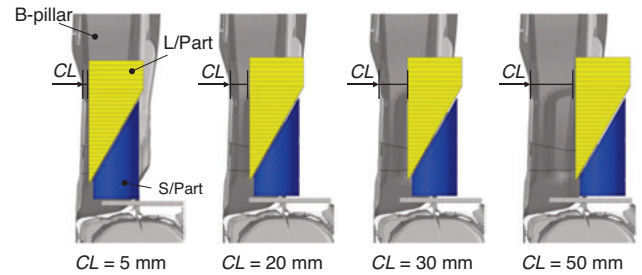


Fig. 12 Schematic diagram of different clearances

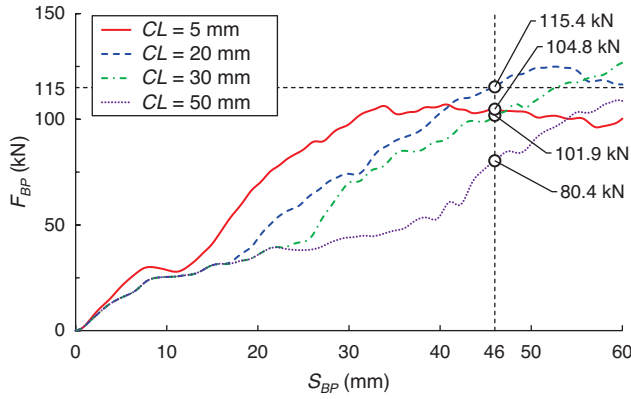


Fig. 13 Load displacement curves at different clearances

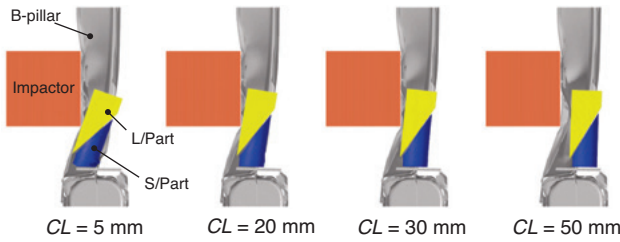


Fig. 14 Cross section of B-pillar and load transfer mechanism at $S_{BP} = 46$ mm

mm. As the clearance increases, the load path to the load transfer mechanism is delayed, and the load rising effect on the stroke of the B-pillar load will be reduced. However, if there is too small clearance, contact with the load transfer mechanism occurs earlier, and the S/Part buckles early and is unable to sustain the force to support the B-pillar.

From the above results, the clearance condition that produces the highest F_{BP} at the timing of the target amount of deformation is 20 mm.

3.4. S/Part Thickness

Taking the slope angle θ of the load transfer mechanism and the clearance determined in preceding sections as fixed values, when the input position Δz from the barrier changes, the F_{z-SP} needed to generate the necessary F_{y-LP} also changes, as shown by Eq. (5). The necessary S/Part thickness was therefore examined under the respective conditions of the New IIHS MDB and former IIHS MDB, which have different barrier collision heights.

The lower edge of the barrier in the New IIHS MDB is positioned 25 mm lower than that in the former IIHS MDB, as shown in Fig. 15. According to Eq. (5), $\Delta y/\Delta z = 0.49$ for the New IIHS MDB and 0.33 for the former IIHS MDB. Therefore, the S/Part load needed to generate F_{y-LP} equivalent to that of the New IIHS MDB under the former IIHS MDB conditions can be estimated as $0.49/0.33 \approx 1.5$

Table 3 Parameters used for investigation of impact point and S/Part thickness

Parameters	Former	New
θ	60 deg	
CL	20 mm	
$\Delta y/\Delta z$	0.33	0.49
S/Part thickness t	0.5 mm, 1.0 mm, 1.5 mm, 2.0 mm, 2.5 mm, 3.0 mm, 3.5 mm, 4.0 mm	0.5 mm, 1.0 mm, 1.5 mm, 2.0 mm, 2.5 mm

times the former value, or 167 kN.

Here, the reasonableness of the F_{z-SP} derived by theoretical calculation was confirmed by simulation using Model 3 with $CL = 20$ mm, as determined in Section 3.3. A total of 13 cases with S/Part thickness t as the parameter, as shown in Table 3, were evaluated with respect to the New IIHS MDB and former IIHS MDB barrier positions.

Figure 16 shows the relationship between F_{BP} and F_{z-SP} at $S_{BP} = 46$ mm. The black dotted line represents the F_{BP} target value, and the blue and red dotted lines respectively represent the F_{z-SP} target values derived by theoretical calculation at the New IIHS MDB and former IIHS MDB collision positions. The value of F_{z-SP} when the F_{BP} target

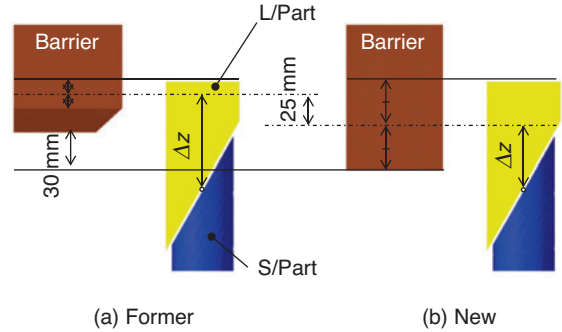


Fig. 15 Schematic diagram of relationship between positions of former and new IIHS MDB

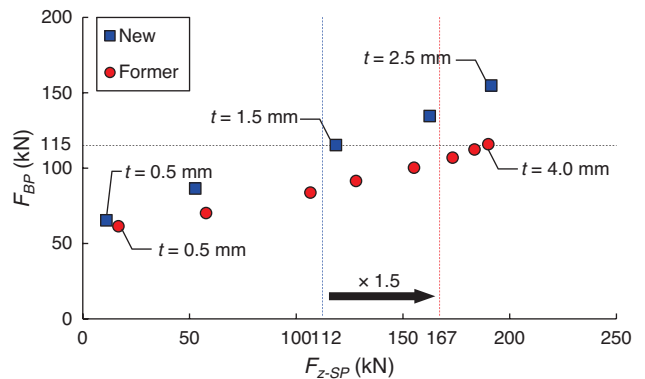


Fig. 16 Relationship between F_{BP} and F_{z-SP} at $S_{BP} = 46$ mm

value was reached was 119 kN ($t = 1.5$ mm) for the New IIHS MDB and 190 kN ($t = 4.0$ mm) for the former IIHS MDB, which roughly matched the theoretical calculation values.

As a result, the condition for achieving the B-pillar target load and amount of deformation is $t = 1.5$ mm or more for the New IIHS MDB and $t = 4.0$ mm or more for the former IIHS MDB.

4. Drop Test Results and Discussion

4.1. Prototyping of Load Transfer Mechanism

A prototype of the load transfer mechanism described in Chapter 3 was fabricated in order to prove the reasonableness of the examination results thus far. Mass-produced C category vehicle parts were used for the B-pillar and surrounding parts. The load transfer mechanism size was set to 150 mm long $\times$ 60 mm wide $\times$ 180 mm high to match the size of the B-pillar opening, as shown in Fig. 17(a) and (b). The L/Part had a lattice block cross-section shape, as shown in Fig. 17(c), and was strong enough to not deform in the drop test. The angle of the junction between the L/Part and the S/Part was set to 60 deg, as obtained in Section 3.2., and the clearance between the B-pillar and the load transfer mechanism was set to 20 mm, as per Section 3.3. The prototype S/Parts used $\phi 50$ general steel pipe made of STKM13C material with strength equivalent to the 590 MPa-grade high-tensile steel, and were fabricated in the thickness range of 0.6 to 2.6 mm generally used in vehicle body parts, as examined in Section 3.4. The fixed base plate was shaped with a sloped surface and a groove for placement on the upper part of the side sill. The mechanism was installed by fastening with bolts to the upper surface of the side sill via the fixed base plate.

4.2. Drop Test Method

The combined performance of the B-pillar and load transfer mechanism to generate a high load with a small amount of deformation as designed was verified with drop

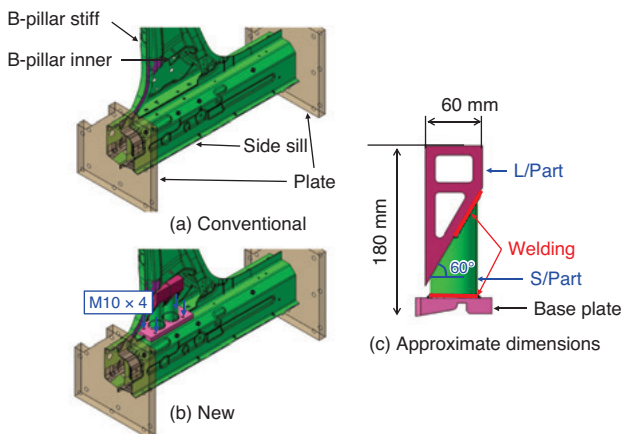
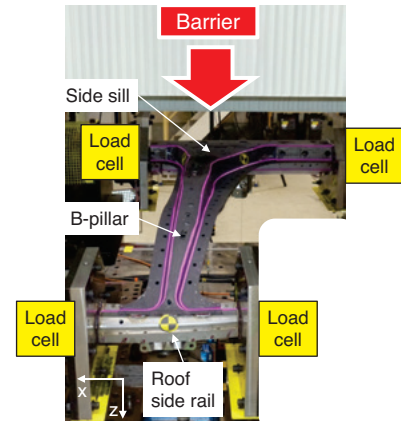


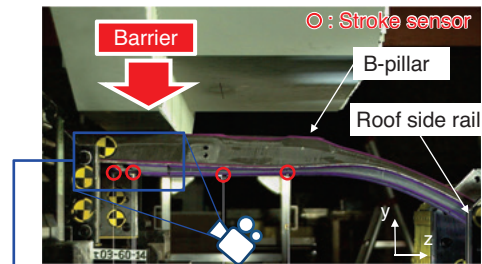
Fig. 17 Overview of load transfer mechanism

tests. Testing used the drop weight test machine at Nippon Steel Corporation's Hasaki Research and Development Center, which is capable of measuring the load and amount of deformation and can also record the deformation with images⁽⁶⁾. The barrier was attached to a weight and allowed to fall freely, and a high-speed camera was used to record the deformation of the test piece and the barrier.

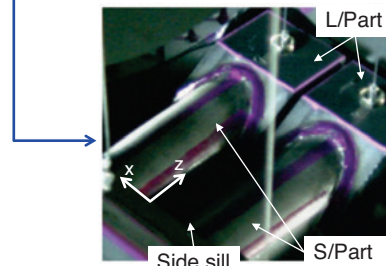
Figure 18(a) and (b) show how the test piece was set up. The test piece is composed of the B-pillar, side sill, roof side rail, and square end plates. The end plates were welded to both ends of the side sill and roof side rail, and were fixed to base plates via load cells to exclude the influence of rotation of the boundary parts. The test piece load was measured by load cells on the barrier side. Stroke sensors were attached in the locations indicated by the red circles in Fig. 18(b) and the amount of test piece deformation was measured. The deformation status of the load transfer mechanism was recorded from the video angle shown in Fig. 18(c).



(a) Test configuration diagram



(b) Moment of collision



(c) Video angle of L/Part and S/Part

Fig. 18 Set up for drop test

Table 4 Drop test conditions

	Test 01	Test 02	Test 03
Barrier	New IHS MDB		
Energy	24 kJ		
θ	60 deg		
CL	20 mm		
S/Part material	-	STKM13C	STKM13C
S/Part thickness	-	0.6 mm	1.4 mm

4.3. Drop Test Conditions

The drop test was performed with the three test pieces, Test 01 to Test 03, shown in Table 4. Test 01 with the conventional structure and Test 02 and Test 03 equipped with the load transfer mechanism were compared to confirm whether a high load can be generated within the target amount of deformation. In addition, Test 02 and Test 03 used a different S/Part thickness to verify that the B-pillar load F_{y-BP} can be changed. The energy of the B-pillar and the barrier when Test 03 reaches the target amount of deformation was calculated by simulation, and the input energy was set to 24 kJ.

4.4. Drop Test Results

Figure 19 shows the relationship between the load and amount of deformation of the B-pillar and the barrier. The red, green, and blue lines respectively indicate the results for Test 01, 02, and 03. The graph covers the range up to where the B-pillar reaches the target amount of deformation. The blue shaded area represents the B-pillar target load and amount of deformation, and the waveform inflection points are indicated by arrows. In Test 01 with the conventional structure, the inflection point appeared at an amount of deformation of 6 mm. The increase in load slowed following the inflection point, and the final F_{y-BP} at the target amount of deformation remained at 58

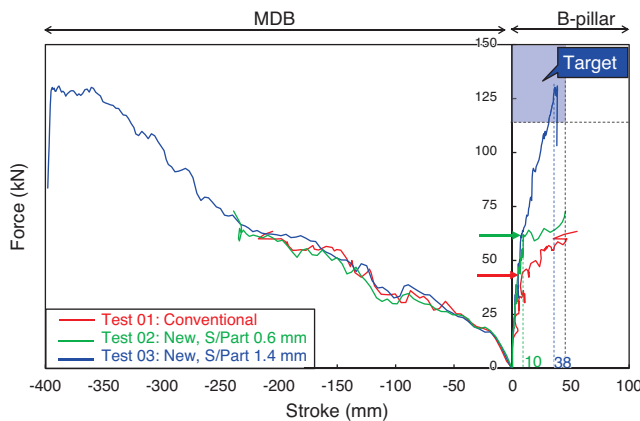


Fig. 19 Comparison of Force-Stroke curve with conventional and new structure

kN. In Test 02, the inflection point appeared at an amount of deformation of 10 mm, with F_{y-BP} reaching 60 kN at that time. There was small increase in load following the inflection point, indicating that the B-pillar target could not be reached with the strength of the 0.6-mm-thick S/Part. In Test 03, there was no inflection point, and the highest load of 130 kN was reached before arriving at the target amount of deformation. The load gradient at this time was $K_{BP} = 3.6$ kN/mm and the B-pillar target performance was achieved.

Figure 20 shows the course of deformation of the load transfer mechanism. In Test 02, S/Part buckling began from 60 ms, when the load reached 65 kN, and the buckling amount increased at 90 ms. In Test 03, only slight signs of buckling appeared around 90 ms, when the load reached 120 kN.

4.5. Consideration of Range of Load Increase by Load Transfer Mechanism

It was predicted from Section 3.3. that as the S/Part buckling increases, the axial load drops and the load supporting the B-pillar decreases. Therefore, focus was placed on the relationship between the S/Part buckling timing and the inflection point, and the effective range over which the load can be increased by the load transfer mechanism was considered.

In Test 02, the S/Part deformation and inflection point timings coincide, and it is thought that the range of deformation that increases the load is up to 10 mm. In Test 03, clear buckling does not occur in the S/Part, so there is no inflection point. That is to say, it can be inferred that the load was continuously generated and the S/Part continued to support the B-pillar up to the highest amount of deformation of 38 mm.

It was concluded from this that the range of deformation within which the load transfer mechanism increases the load is up to the bifurcation point for Test 02 and up to the highest amount of deformation for Test 03. This is shown in Fig. 21(a). In addition, Fig. 21(b) compares the amount of

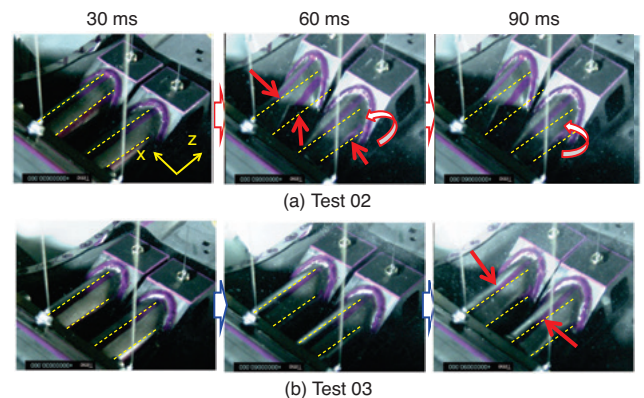


Fig. 20 Comparison of deformation modes of L/Part and S/Part

increase in the load at the respective timings of the amounts of deformation shown in Fig. 21(a). This increase was 20 kN for Test 02 and approximately 87 kN for Test 03.

The above indicates that a high load can be continuously generated by setting an S/Part thickness at which the S/Part can continuously generate an axial load without buckling.

4.6. Consideration of Amount of Energy Absorption and Effect to Reduce the Amount of Deformation

Figure 22 shows the amount of energy absorption at the timing of the target amount of deformation. The amount of energy absorbed by the barrier increased by 2.5 times to 20 kJ for Test 03 compared to 8 kJ for Test 01. In addition, Test 03 absorbed the entire input energy of 24 kJ, indicated by the black dotted line, but Test 02 absorbed 11 kJ less and Test 01 absorbed 14 kJ less. This deficit energy amount is approximately equal to the difference in the amount absorbed by the barrier.

Here, the estimated Force-Stroke at which the load increases along the extension of the Test 01 waveform to reach the same amount of energy absorption as Test 03 is shown in the red shaded areas in Fig. 23. The B-pillar load

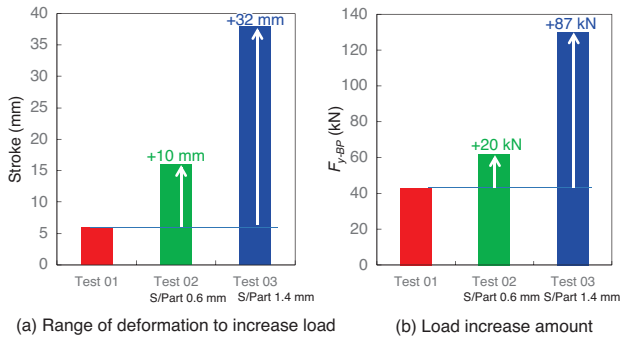


Fig. 21 Comparison of conventional structure and new structure effects

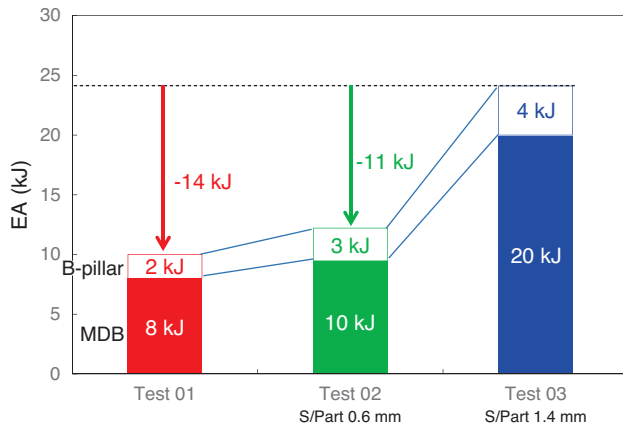


Fig. 22 Comparison of energy absorption with conventional and new structure

is expected to increase by up to 80 kN and the amount of deformation by up to approximately 180 mm. Comparison of the amount of B-pillar deformation using the drop test value for Test 03 and this estimated value for Test 01 indicates that installation of the load transfer mechanism can be expected to reduce the amount of B-pillar deformation by approximately 80%.

From the above, use of the load transfer mechanism to increase the B-pillar load enables increasing the amount of energy absorbed by the barrier and reducing the amount of B-pillar deformation as planned.

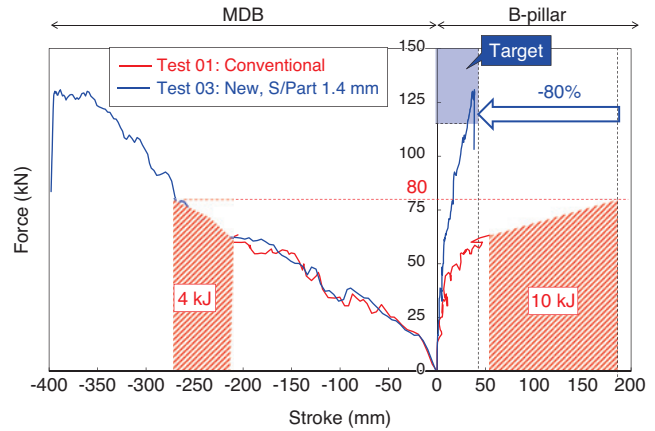


Fig. 23 Comparison of energy absorption with conventional and new structure

5. Conclusion

A concept that leads to further enhancement of vehicle body performance in side collisions was devised with a focus on the IIHS collision test conditions. A prototype model based on that principle was fabricated, and the ability to achieve the target performance was validated by drop tests. These results are summarized below.

- (1) The new concept was to support the lateral input from the barrier as an axial load in the downward direction in the vehicle body and create a load path to the side sill.
- (2) A two-part load transfer mechanism composed of an L/Part and an S/Part was added in the cross section in the lower part of the B-pillar.
- (3) The slope angle of the S/Part and L/Part, the clearance with the B-pillar, and the S/Part thickness were defined as parameters.
- (4) It was proved that in a prototype with the load transfer mechanism installed in the B-pillar, the load was doubled and it was estimated that the amount of deformation could be reduced to approximately one-fifth that of the conventional structure.

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Human Motion Estimation Using Only Inertial Sensors

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ABSTRACT

When human motion is estimated using inertial sensors, measurement over long periods of time results in the accumulation of errors in yaw angle direction. There is a method of compensating for this using magnetometers, but this method is not effective in environments where the magnetism is unstable. A method of compensating for errors in yaw angle direction without using magnetometers was therefore investigated. A method was devised in which the measurement object is defined as motion in which every link of the body has the same yaw angle when averaged over time. Compensation is then applied by reducing the differences in yaw angle between the links. When estimating walking motion, the difference in joint angle estimated by the method using inertial sensors and magnetometers and that estimated by optical motion capture has a root mean square of 8.4 deg, while the corresponding figure with the present method using only inertial sensors is 8.9 deg. This indicates equal levels of accuracy. It was also confirmed that the integral calculation error in yaw angle when measuring walking motion over a long period of time was equally reduced.

1. Introduction

Making correct measurements of human motion is the first step toward a variety of biomechanics analyses⁽¹⁾. If it were possible to obtain a quantitative grasp of bodily motion, then it would be possible to examine appropriate methods of guidance, and evaluate their effectiveness, in the areas of rehabilitation⁽²⁾ and sports training⁽³⁾. This could also be applied to the prediction of risks of injury and disorders resulting from body burden on athletes and laborers engaging in heavy work⁽⁴⁾. In recent years, wearable robots that assist with human movement have been used to reduce body burden as well as in rehabilitation. Since human motion is the basis for control of assistive force⁽⁵⁾, motion estimation is a key functionality for a wearable robot.

Optical motion capture is one of the previously existing methods of motion measurement and it is widely used⁽¹⁾. This method involves attaching reflective markers to the body and capturing images with multiple infrared cameras to calculate three-dimensional positions and then estimate the body orientation. This method enables measurement of position with a high degree of accuracy, but it requires

surrounding the measurement object with multiple cameras, which limits the range of possible measurement. Sometimes the presence of an obstruction between the camera and the measurement object may also make it a challenge to identify the positions of reflective markers. A method has been used in recent years to estimate joint position using machine learning with RGB images captured with ordinary cameras⁽⁶⁾. Since this requires images to be captured externally, however, it has limitations similar to those of optical motion capture.

As a method that does not have the above limitations, there is the use of a compact inertial sensor that can be attached to the body. This method uses this inertial measurement unit (IMU) to estimate three-dimensional orientation⁽⁷⁾. An IMU can measure acceleration and angular velocity, and numerical integration of angular velocity enables estimation of orientation. By comparison with the optical motion capture method and the machine learning-based method that estimates the skeletal frame from imagery, the method here involves less computation and it allows processing at high speed with lower-cost processors. There are issues, however, in that angular velocity measurement errors and numerical integration

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errors accumulate, and the estimated orientation and actual orientation end up diverging over time. In order to address these issues, methods have been proposed that use the direction of gravity estimated from the measured acceleration as a guide in determining compensation for orientation in the roll and pitch directions^{(8), (9)}. In order to further determine compensation for orientation in the yaw direction, methods have been proposed for estimating orientation by means of a magnetic, angular rate, and gravity (MARG) sensor incorporating a magnetometer^{(10), (11)}. However, yaw angle compensation using geomagnetism is only effective in an environment where stable measurement of geomagnetism is possible. Magnetic fields are greatly changed in the vicinity of the steel frames of buildings, metallic fixtures and furniture, and other such magnetic bodies, and this frequently makes it so geomagnetism cannot be measured correctly. When measuring the motion of a human body equipped with a wearable robot, this can also mean that the accuracy of geomagnetism measurement ends up being reduced by the influence of the metallic frame and motor of the device.

As indicated by the above, although the estimation of orientation using the IMU is not affected by the camera's angle of view or obstructions, there are issues with yaw angle compensation that uses geomagnetism as a guide. For the present research, therefore, a method of compensation for yaw angle was examined that does not rely on magnetometers but instead imposes restrictions on the motion of the measurement object and uses those restrictions to determine the compensation. First, the body was divided into segments with boundaries at the joints and the human body was considered as a rigid link system. When the measurement object is limited to motion in which the yaw angle of each link ends up in the same direction when averaged over time, the relationships between the orientations of every link can be used as a guide for determining yaw angle compensation. A method of compensation for the yaw angle in body coordinates determined solely from the IMU was successfully devised based on this conceptual approach, and that method is therefore reported here.

2. Method for Estimating Body Orientation

2.1. Sensor Placement and the Body Model

In order to estimate motion with the IMU, the human body is first modeled as a rigid link system. Figure 1 shows an example of a body with the torso, pelvis, left and right thighs, lower legs, and feet each modeled as rigid links. The links are joined by joints that have three orthogonal rotation axes in each space between the links. An IMU is attached to each link, and the system of coordinates of the IMU and of the link are considered to be coincident. In other words, if the orientation of the IMU is estimated, then the orientation and joint angle of the link will become clear. Every link is allocated a unique number, i , and in the example of Fig. 1,

$i \in \{0, 1, 2, \dots, 6, 7\}$. Link 0 is the link that will serve as a reference for calculation of the yaw angle compensation amount, which will be explained later, and it will be referred to as the base link.

As shown in Fig. 2, each IMU has its own sensor coordinate system. The acceleration and angular velocity obtained from an IMU are as observed in that sensor coordinate system. Here the acceleration in the X, Y, and Z directions in the sensor coordinate system observed using the IMU for Link i are designated $a_{i,x}$, $a_{i,y}$, $a_{i,z}$, respectively. Similarly, the angular velocity around the X, Y, and Z axes in the sensor coordinate system are designated $\omega_{i,x}$, $\omega_{i,y}$, and $\omega_{i,z}$, respectively.

2.2. Representation of Three-dimensional Orientation

Orientation is commonly represented using an Euler angle, which is made up of three-dimensional elements, and a unit quaternion, which is made up of four-dimensional elements. An Euler angle can represent orientation with a small number of elements, but there is an orientation (singularity) that becomes numerically unstable and cannot be represented. In the present research, therefore, orientation is represented using the unit quaternion, which does not have a singularity.

The rotation from coordinate system A to coordinate

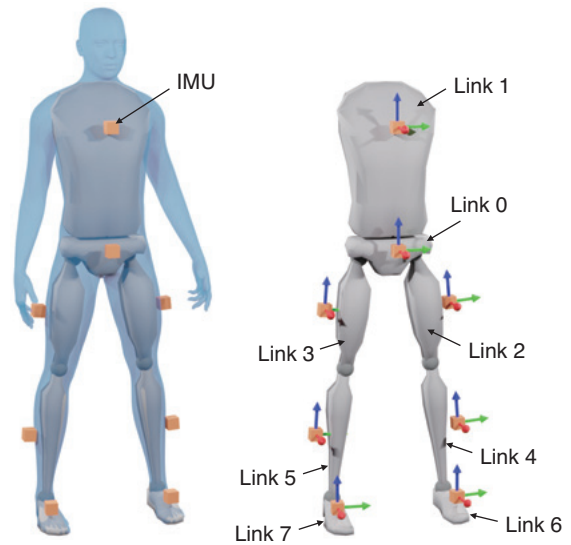


Fig. 1 Articulated human body and IMU allocation

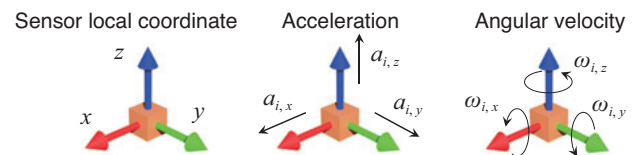


Fig. 2 Sensor coordinate systems

system B is expressed as shown in Eq. (1) by the unit quaternion ${}^A_B\hat{e}$. Here, coordinate system B is taken to rotate only by θ with respect to the normalized rotation axis $r = [r_x \ r_y \ r_z]$ of coordinate system A. Orientation will be a normalized unit quaternion as in Eq. (2).

$$\begin{aligned} {}^A_B\hat{e} &= [e_1 \ e_2 \ e_3 \ e_4] \\ &= [\cos \frac{\theta}{2} \ -r_x \sin \frac{\theta}{2} \ -r_y \sin \frac{\theta}{2} \ -r_z \sin \frac{\theta}{2}] \end{aligned} \quad (1)$$

$$\hat{e} = \frac{e}{\|e\|} \quad (2)$$

As shown in Eq. (3), a quaternion product $\otimes$ can also be used to translate vector ${}^A_V$ in coordinate system A to ${}^B_V$ represented in coordinate system B^{(10), (11)}.

$${}^B_V = {}^A_B\hat{e} \otimes {}^A_V \otimes {}^A_B\hat{e}^* \quad (3)$$

However, ${}^A_B\hat{e}^*$ is a conjugation of ${}^A_B\hat{e}$, and is defined as shown in Eq. (4).

$${}^A_B\hat{e}^* = [e_1 \ -e_2 \ -e_3 \ -e_4] \quad (4)$$

2.3. Roll and Pitch Angle Compensation

The orientation error that is obtained by integrating the angular velocity measured by means of the IMU can be compensated for by taking the direction of gravity estimated from acceleration as a guide. Madgwick, et al.⁽¹¹⁾ proposed a method that takes this as an optimization problem that minimizes the roll and pitch angle error and determines the compensation direction from the gradient of the objective function. For the present research, a similar method of roll and pitch angle compensation has been adopted. First, the difference between the direction of gravity calculated from the estimated orientation of Link i and the measured acceleration is represented by the function f_g , which is defined as follows.

$$f_g({}^S\hat{e}_i, {}^S\hat{a}_i) = {}^S\hat{e}_i^* \otimes {}^S\hat{d}_g \otimes {}^S\hat{e}_i - {}^S\hat{a}_i \quad (5)$$

$${}^S\hat{d}_g = [0 \ 0 \ 0 \ 1] \quad (6)$$

$${}^S\hat{a}_i = [0 \ a_{i,x} \ a_{i,y} \ a_{i,z}] \quad (7)$$

${}^S\hat{d}_g$ is the direction of gravity in an absolute coordinate system while ${}^S\hat{a}_i$ is the acceleration vector measured by means of the IMU. The optimization problem that, using Eq. (5), takes the orientation ${}^S\hat{e}_i$ of Link i as an objective variable is defined as in Eq. (8). This is the least-squares problem that determines the orientation ${}^S\hat{e}_i$ where the measured acceleration and the direction of gravity coincide. The gradient of the objective function in Eq. (8) is defined as in Eq. (9). However, J_g is the Jacobian matrix of f_g .

$$\min_{{}^S\hat{e}_i} \frac{1}{2} \|f_g({}^S\hat{e}_i, {}^S\hat{a}_i)\|^2 \quad (8)$$

$$\begin{aligned} \text{grad}_g({}^S\hat{e}_i, {}^S\hat{a}_i) &= \nabla \left\{ \frac{1}{2} \|f_g({}^S\hat{e}_i, {}^S\hat{a}_i)\|^2 \right\} \\ &= J_g^T({}^S\hat{e}_i) f_g({}^S\hat{e}_i, {}^S\hat{a}_i) \end{aligned} \quad (9)$$

From the gradient in Eq. (9), the angular velocity can be compensated⁽¹¹⁾ as shown below. Here the angular velocity of Link i at time t is ${}^S\dot{e}_{i,t}$ and the acceleration is ${}^S\hat{a}_{i,t}$.

$${}^S\dot{e}_{i,t} \leftarrow {}^S\dot{e}_{i,t} - \beta_g \frac{\text{grad}_g({}^S\hat{e}_{i,t}, {}^S\hat{a}_{i,t})}{\|\text{grad}_g({}^S\hat{e}_{i,t}, {}^S\hat{a}_{i,t})\|} \quad (10)$$

2.4. Objective Function of Yaw Angle Compensation

The objective function of yaw angle compensation is defined by making use of the fact that the measurement object is a motion for which the direction of each link (yaw angle direction) is identical when averaged over time. First, the vector ${}^G\hat{v}_i$ that indicates the yaw angle direction of Link i is defined. ${}^G\hat{v}_i$ is calculated according to Eq. (12) and is a vector that is orthogonal to each unit vector ${}^S\hat{n}_i = [0 \ 0 \ 1 \ 0]$ in the Y axis direction of Link i , and unit vector ${}^G\hat{d}_g$ in the Z axis direction of the global coordinate system. As shown in Fig. 3, ${}^G\hat{v}_i$ is a unit vector on the X-Y plane (horizontal plane).

$${}^G\hat{v}_i = ({}^S\hat{e}_i \otimes {}^S\hat{n}_i \otimes {}^S\hat{e}_i^*) \times {}^G\hat{d}_g \quad (11)$$

$${}^G\hat{v}_i = \frac{{}^G\hat{v}_i}{\|{}^G\hat{v}_i\|} \quad (12)$$

Next, as was done with the roll and pitch angles, the compensation direction is derived by means of gradient descent. Using ${}^G\hat{v}_i$ and the Y axis direction ${}^S\hat{n}_i$ in the base link coordinate system, the objective function for yaw angle compensation is defined as in Eq. (13). This objective function is defined for links other than the base link ($i \neq 0$).

$$f_{yaw}({}^S\hat{e}_i) = ({}^S\hat{e}_i \otimes {}^S\hat{n}_i \otimes {}^S\hat{e}_i^*) \cdot {}^G\hat{v}_i \quad (13)$$

The objective function f_{yaw} is a scalar value. As shown in Fig. 4, $|f_{yaw}|$ is zero when the X-Z plane (sagittal plane) as viewed in the local coordinate system for Link 0 is parallel with yaw angle direction vector ${}^G\hat{v}_i$ for Link i and it is one when they are orthogonal. This is an indicator of the extent to which the yaw angle direction of Link i and the yaw angle direction of the base link coincide.

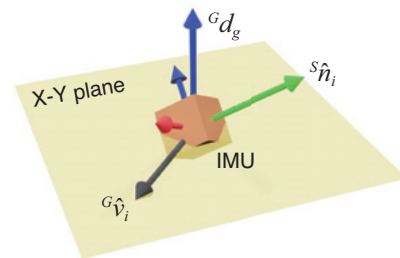


Fig. 3 Yaw angle direction of Link i

2.5. Yaw Angle Compensation

The objective function in Eq. (13) is defined from the orientations of two links, Link $i \neq 0$ and the base link where $i = 0$. When considering yaw angle compensation for the whole body's links, every link other than the base link influences and is influenced by the other links through the base link. Consequently, the yaw direction compensation amount has to be calculated with all the links taken into consideration. Consider unifying the objective functions for yaw angle compensation for all the links and calculating the compensation amount from the gradient. As in the next equation, the orientation of all the links is combined and expressed as orientation Q .

$$Q = [\begin{smallmatrix} {}^S_G\hat{e}_0 & {}^S_G\hat{e}_1 & {}^S_G\hat{e}_2 & \cdots & {}^S_G\hat{e}_{N-2} & {}^S_G\hat{e}_{N-1} \end{smallmatrix}] \in \mathbb{R}^{4N} \quad (14)$$

When determining the yaw angle compensation amount for the whole body, the objective function is defined as follows.

$$\min_Q \frac{1}{2} \|F_{yaw}(Q)\|^2 \quad (15)$$

$$F_{yaw}(Q) = \begin{bmatrix} f_{yaw}({}^S_G\hat{e}_1) \\ f_{yaw}({}^S_G\hat{e}_2) \\ \vdots \\ f_{yaw}({}^S_G\hat{e}_{N-2}) \\ f_{yaw}({}^S_G\hat{e}_{N-1}) \end{bmatrix} \in \mathbb{R}^{N-1} \quad (16)$$

F_{yaw} is an N -dimensional vector and it expresses the extent to which the sagittal plane of the base link and the yaw angle direction ${}^G\hat{v}_i$ of each link are orthogonal. When the yaw angle directions ${}^G\hat{v}_i$ of all the links become parallel with the sagittal plane of the base link, F_{yaw} becomes a vector in which all the elements are zero. As in Eq. (9), the gradient of F_{yaw} can be calculated as follows using the

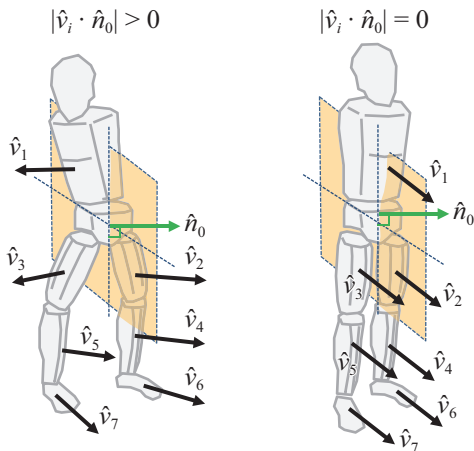


Fig. 4 Objective function for yaw compensation

Jacobian matrix.

$$\text{grad}_{yaw}(Q) = J_{yaw}^T(Q) F_{yaw}(Q) \quad (17)$$

Using the orientation of the whole body Q_t at time t and Eq. (17), the yaw angle compensation amount ΔQ_t can be calculated as follows.

$$\Delta Q_t = \frac{\text{grad}_{yaw}(Q_t)}{\|\text{grad}_{yaw}(Q_t)\|} \quad (18)$$

ΔQ_t is a vector that combines the yaw angle compensation amounts for the IMU of every link, and it is composed as follows with the compensation amount $\Delta e_{i,t}$ of Link i as an element.

$$\Delta Q_t = [\Delta e_{0,t} \ \Delta e_{1,t} \ \Delta e_{2,t} \ \cdots \ \Delta e_{N-2,t} \ \Delta e_{N-1,t}] \quad (19)$$

Factoring Eq. (19) into the elements of each link, compensation for the yaw angle of Link i is obtained by the following equation, as it was for roll and pitch compensation in Eq. (10). However, $\beta_{yaw} > 0$ is the feedback gain for the compensation amount.

$${}^S_G\dot{e}_{i,t} \leftarrow {}^S_G\dot{e}_{i,t} - \beta_{yaw} \Delta e_{i,t} \quad (20)$$

2.6. Filter Configuration

The method proposed in the present research (proposed method) can be divided generally into the operations of performing numerical integration of the angular velocities from the individual IMU sensors and calculating the orientation, and the operations of compensating for orientation by means of Eqs. (10) and (20). The former will be explained first. The next equation is used to translate the angular velocities measured using the IMU to a time differential form of quaternion⁽¹¹⁾.

$${}^S_G\dot{e}_{i,t} = \frac{1}{2} {}^S_G\hat{e}_{i,t-1} \otimes {}^S\omega_{i,t} \quad (21)$$

$${}^S\omega_{i,t} = [0 \ \omega_{i,t}^x \ \omega_{i,t}^y \ \omega_{i,t}^z] \quad (22)$$

However, ${}^S\omega_{i,t}$ in Eq. (22) is the angular velocity of Link i at time t and it is a four-dimensional vector that has the first element as zero. Using ${}^S_G\dot{e}_{i,t}$ and the previous value for orientation ${}^S_G\hat{e}_{i,t-1}$, the orientation is updated as follows. Here, Δt is the sampling time. Since ${}^S_G\hat{e}_{i,t}$ is not a unit quaternion, attention is paid to the necessity for normalization.

$${}^S_G\hat{e}_{i,t} = {}^S_G\hat{e}_{i,t-1} + {}^S_G\dot{e}_{i,t} \Delta t \quad (23)$$

$${}^S_G\hat{e}_{i,t} = \frac{{}^S_G\hat{e}_{i,t}}{\|{}^S_G\hat{e}_{i,t}\|} \quad (24)$$

The orientation ${}^S_G\hat{e}_{i,t}$ that has been estimated by the

above calculations is a simple numerical integration of the angular velocities measured by IMU. This estimated orientation ends up diverging over time from the actual orientation because of error in the measurement values. In order to reduce this error, the compensation amounts obtained from Eqs. (10) and (20) are used as shown below as compensation for the angular velocity ${}^S\dot{e}_{i,t}$.

$$\begin{aligned} {}^S\dot{e}_{i,t} = & \frac{1}{2} {}^S\hat{e}_{i,t-1} \otimes {}^S\omega_{i,t} \\ & - \beta_g \frac{\text{grad}_g({}^S\hat{e}_{i,t-1}, {}^S\hat{a}_{i,t})}{\|\text{grad}_g({}^S\hat{e}_{i,t-1}, {}^S\hat{a}_{i,t})\|} \\ & - \beta_{yaw} \Delta e_{i,t-1} \end{aligned} \quad (25)$$

The configuration of the orientation estimation filter that uses the compensation from Eq. (25) is shown in Figs. 5 and 6. Figure 5 corresponds to the operation of integrating the angular velocity of Link i and applying compensation for acceleration to the roll and pitch angles. Figure 6 also shows the operation of determining the yaw angle compensation amount. Here, Q_{t-1} and ΔQ_{t-1} are the previous value for estimated orientation of every link and the yaw angle compensation amount, respectively. The previous value for estimated orientation of every link ${}^S\hat{e}_{i,t-1}$ emerges at ① in Fig. 5 and it is supplied as input to ① in Fig. 6. The estimated orientation ${}^S\hat{e}_{i,t-1}$ of every link is integrated in Q_{t-1} , which is used to calculate the compensation amount ΔQ_{t-1} by means of Eq. (18). This is factored into the yaw angle compensation amount $\Delta e_{i,t-1}$ for every link and supplied as input from ② in Fig. 6 to ② in Fig. 5. In Fig. 5, $\Delta e_{i,t-1}$ is applied as feedback to angular velocity and is reflected as the compensation amount for yaw angle direction.

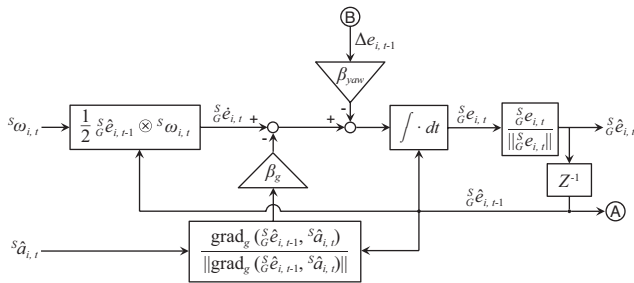


Fig. 5 Block diagram of orientation estimation

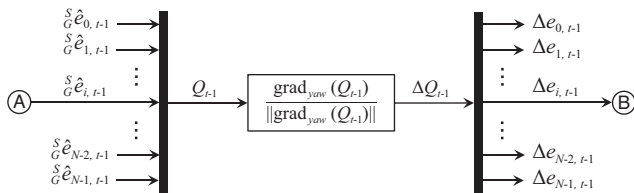


Fig. 6 Block diagram of yaw compensation

3. Experiment

A comparison was made of the proposed method and the previously existing measurement method applied to walking movement. Optical motion capture, which is widely used for motion measurement, was taken as the point of comparison. The IMU and reflective markers for optical motion capture use were attached as shown in Fig. 7 and measurements were performed simultaneously. The MTw Awinda from Xsens was used for the IMU. Angular velocity and acceleration data were acquired through the software development kit provided by the company and the operations for the proposed method were implemented. For optical motion capture, the MAC3D System from nac Image Technology, Inc., was used. Walking measurements were made of two test subjects on a treadmill. Starting from a static standing state, the treadmill speed was raised until the test subjects reached a comfortable walking speed, and measurement of walking was continued for 30 seconds.

It should be noted that this experiment was conducted with the approval of Honda's Bioethics Committee. The experimental subjects participated of their own free will. They had received complete oral and written explanations in advance from the principal investigator regarding the purpose of the present research, its content, and the rights of the participants, and their consent to participate was obtained in writing.

Figure 8 shows the extension and flexion angles at the hip joints and knee joints and the plantarflexion and dorsiflexion angles at the ankle joints for 10 seconds from when walking was started as estimated from the IMU and optical motion capture. In addition, the root mean square (RMS) differences in joint angle between the estimates obtained by the two methods are shown in Table 1. This table shows the RMS of differences in angle calculated for extension and flexion (plantarflexion and dorsiflexion), abduction and adduction, and external and internal rotation for each axis in the hip joints, knee joints, and ankle joints,

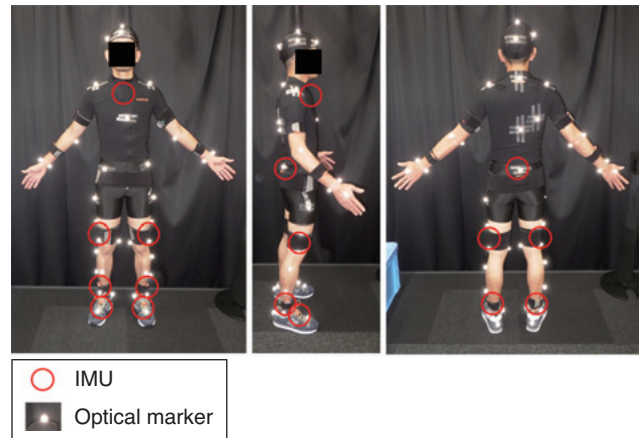


Fig. 7 IMU and optical marker allocation

Table 1 RMS differences between IMU and optical motion capture

Joint		RMS differences [deg]	
		MARG ⁽¹¹⁾	Proposed method
Hip	Extension/ Flexion	4.8	3.8
	Abduction/ Adduction	3.7	4.9
	External/ Internal	5.7	4.4
Knee	Extension/ Flexion	3.7	4.8
	Abduction/ Adduction	4.4	5.1
	External/ Internal	8.4	5.7
Ankle	Plantarflexion/ Dorsiflexion	4.7	5.8
	Abduction/ Adduction	7.2	5.3
	External/ Internal	5.7	8.9

respectively. For the purpose of comparison, the results obtained with measurable geomagnetism in addition to angular velocity and acceleration using a MARG sensor with the Xsens method⁽⁷⁾ are given.

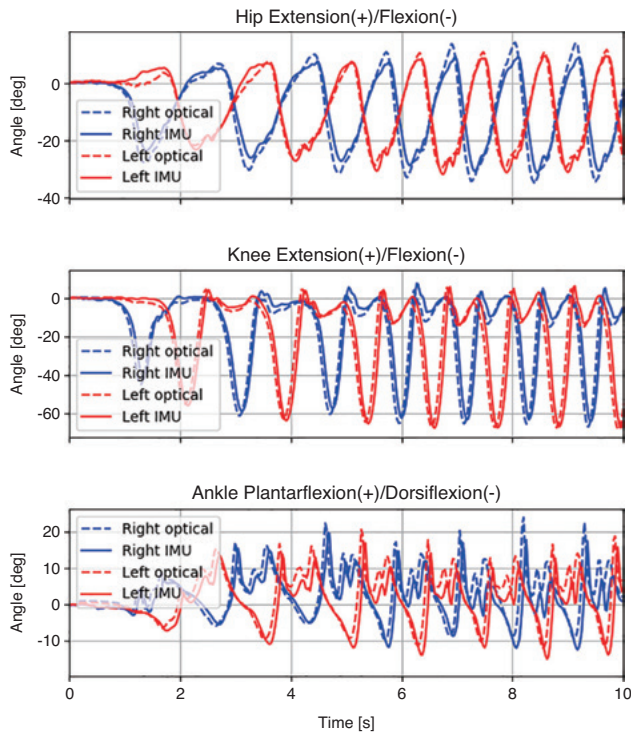


Fig. 8 Comparison of joint angles

Next the effect of reduction in yaw angle error occurring over a long period of measurement was evaluated. This was done by measuring walking that involved a turning movement for a 5-minute period. This movement involved greater angular velocity in the yaw direction than walking straight forward, and the influence of estimation errors would tend to emerge more. The comparison was made between the proposed method and the case in which yaw angle compensation was nullified by assigning a zero value to the feedback gain β_{yaw} for the yaw angle compensation amount described in Section 2.5. Figure 9 shows the internal and external rotation angles for the cases with and without yaw angle compensation.

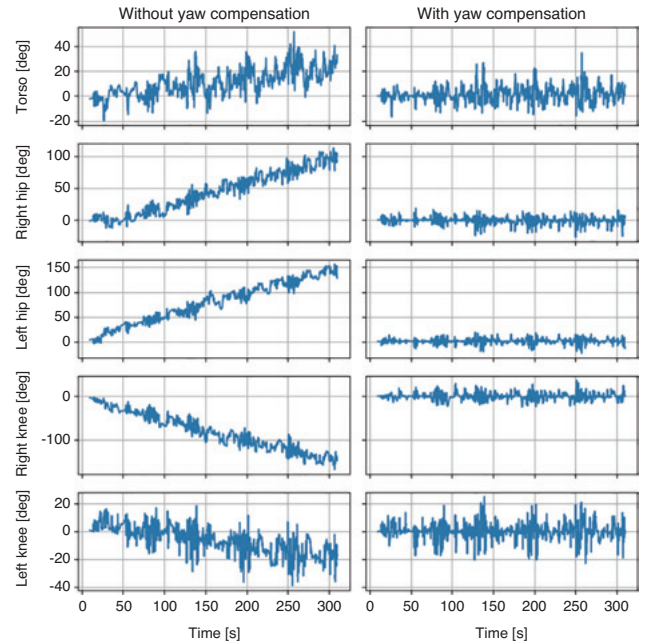


Fig. 9 Comparison of internal/external angles

4. Discussion

The comparison of estimated joint angles in walking movement with the optical motion capture approach (Table 1) showed that the RMS difference in joint angle for the proposed method was from 3.8 to 8.9 deg, which was a result on a par with the RMS from 3.7 to 8.4 deg for the Xsens method using MARG sensors. When the movement is one in which the yaw angles of every link become identical when averaged over time, as is the case with walking movement, the proposed method can be used in the same way as the motion capture systems that are available on the market.

It is also apparent from the results for measurement of walking movement that involved turning (Fig. 9) that when $\beta_{yaw} = 0$ and yaw angle compensation is not implemented,

the amount of drift in the external and internal rotation angles of each joint is large. In the left hip joint, in particular, a drift of 150 deg has occurred in the external rotation direction at a point 300 seconds from the start of measurement. Error in the external and internal rotation angles in the walking movement could not be lessened to a significant extent by roll and pitch angle compensation using the direction of gravity. This extent of drift is thought to result because the bias error in yaw angle direction is reflected so strongly. By contrast, when the yaw angle compensation is activated in the proposed method, this error was resolved to some extent. When walking involves a turning movement, the change in the yaw angle is greater and the influence of inconclusive bias errors becomes greater. The proposed method can reduce the yaw angle drift in the base link coordinate system even under such circumstances, and therefore the external and internal rotation angles show an average transition of 0 deg even after 300 seconds.

5. Conclusion

In order to create a method for estimating motion using only an IMU attached to every link in the body, the measurement object was limited to movement in which the yaw angle of each link ends up in the same direction when averaged over time. A method of yaw angle compensation that takes that limitation as a guide was examined.

An objective function that expresses the extent to which the yaw angles on all the links with an IMU attached coincide was defined, and a compensation direction that reduces that objective function to the lowest level was sought. In this way, a method of compensation for the yaw angle in the base link coordinate system was devised. With regard to walking movement, the RMS of difference between joint angles estimated using conventional IMU and magnetometer and joint angles estimated using optical motion capture is 8.4 deg. The figure for the present method using only an IMU was 8.9 deg, indicating that it is equal in accuracy.

The present research has realized yaw angle compensation in the base link coordinate system for the above measurement object movement that does not depend on a magnetometer, and it is thought that estimation of motion over a long period of time using only the IMU will become possible.

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Haruo AOKI

Correction of errors

No.	Vol. No.	pdf No.	Title of the paper	Page	Corrected part	Details	
1	Vol.33 No.2	33-2_03e	Short Lead Time Development of Customized Vehicles with Preventive Measures for COVID-19 Infection - Social Contribution through Swift Provision of Vehicles -	p. 29	Section 3.4., model	Error	DBA-PR1
						Correct	DBA-RP1
2	Vol.33 No.2	33-2_07e	Comprehensive Study of Solvents for Wet Process Applied to Sulfide- based All Solid State Battery	p. 69	Eq. (3)	Error	$\begin{array}{c} \text{S} \\ \\ \text{S}^- - \text{P} - \text{S}^- \\ \\ \text{S}^- \end{array} + 3\text{H}_2\text{O} \longrightarrow \begin{array}{c} \text{O} \\ \\ \text{O}^- - \text{P} - \text{O}^- \\ \\ \text{O}^- \end{array} + 3\text{H}_2\text{S}$
						Correct	$\begin{array}{c} \text{S} \\ \\ \text{S}^- - \text{P} - \text{S}^- \\ \\ \text{S}^- \end{array} + 4\text{H}_2\text{O} \longrightarrow \begin{array}{c} \text{O} \\ \\ \text{O}^- - \text{P} - \text{O}^- \\ \\ \text{O}^- \end{array} + 4\text{H}_2\text{S}$

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Editorial Afterword

Thank you for reading all the way to the end of the Honda R&D Technical Review. As one of those involved in its editing, I want to express my gratitude to our readers, the authors who submitted their manuscripts for this issue, and everyone who put their efforts into editing and publishing it.

Nearly two years have passed already since COVID-19 became a global pandemic. Last autumn, when the effect of the vaccines had started to bring down the numbers of new infections in Japan, a special event for employees here gave me the opportunity to hear a lecture on Leonardo da Vinci by the author Walter Isaacson.

Leonardo is famous as a painter, but as depicted by this author, he was not only an artist. Behind the *Mona Lisa*, there is a knowledge of anatomy and penetration into the mysteries of the human body. Behind *The Last Supper*, there is a knowledge of geometry. It is very exciting what has been produced in various fields of technology out of this person's accomplishments and explorations.

One of the things I found particularly interesting was that Mr. Isaacson came to understand Leonardo's activities by deciphering and reading his 7,200 pages of handwritten notes. The notes were written on paper, which was precious at that time, so in order not to waste it, each page was completely filled up with miscellaneous notations that have conveyed Leonardo's true accomplishments and the breadth of his inquiring mind to the present day.

This issue also presents a wide range of content, from the introduction of product technologies from the Motorcycle and Life Creation units to highly specialized papers, exemplifying the breadth of Honda technology. This document, available on the Honda website for people around the world to read, is our "notes." It is our hope that the content of this issue will stimulate curiosity in the people who read it, and thereby become an occasion for exchanges on technology between Honda and people around the world.

Hibiki KOGA

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