

Honda R&D



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Comment on the Cover

This is the cover of the second edition of the Technical Review published under our series theme “The Functional Beauty of Honda Products.”

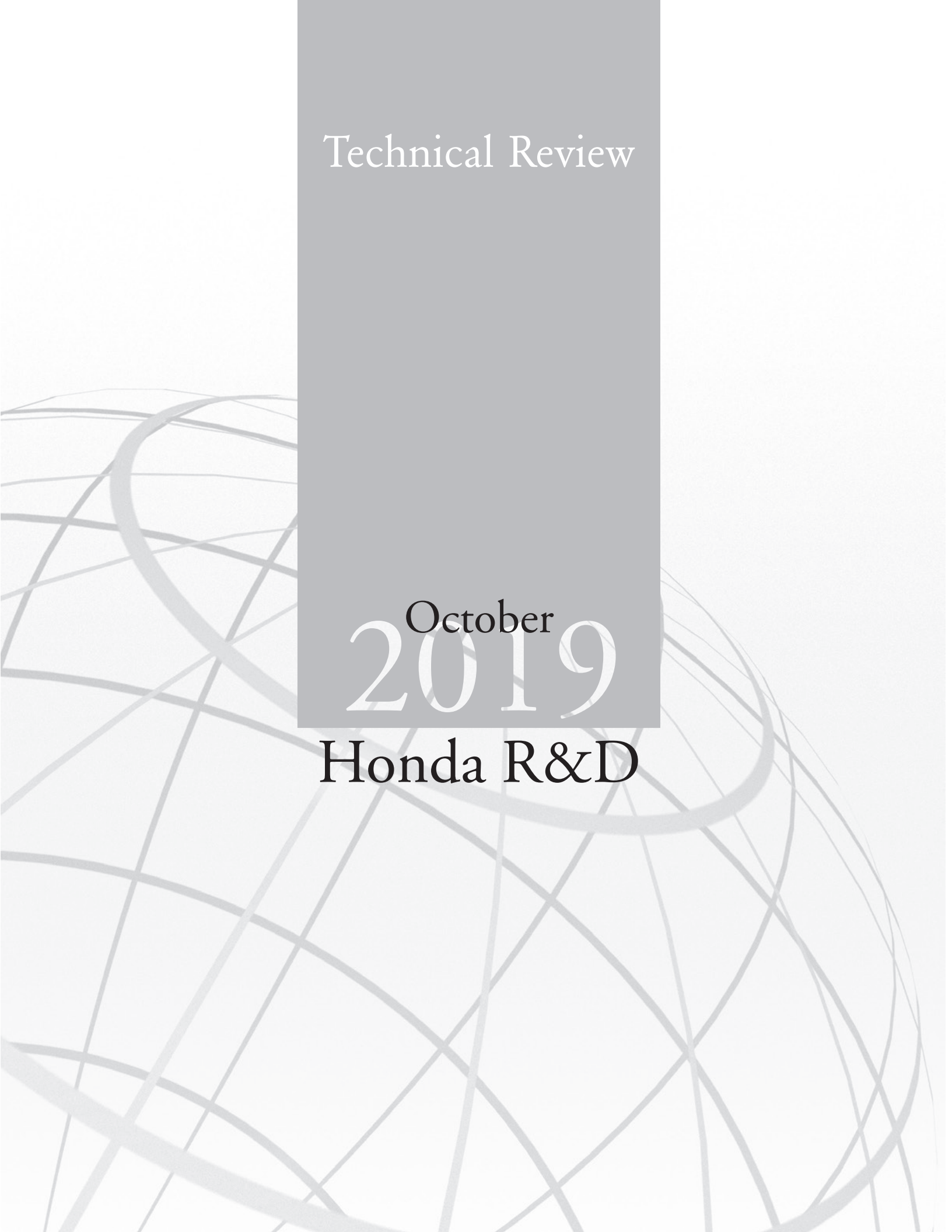
A CVT belt was selected as the motif, in a composition that expresses the way in which the rings circulate while describing a circle.

Given that we demand that CVTs freely control the contradictory qualities of friction and lubrication, the realization of higher performance levels in the units is a multifaceted process.

Against this background, I believe that these three CVT-related themes are a crystallization of our engineers’ ever-deeper quests and minutely focused efforts.



Automobile Center
Atsushi SASAKI

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

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Styling Design of 2018 PCX

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ABSTRACT

Regarding design themes of the PCX, “elegance” was set for the first generation model and “innovation” for the third. The vehicle designs based on these themes were both well received by the customers. When the design direction was examined for the fourth generation model, it was decided to pursue satisfactions of both supporters of previous designs. Based on these considerations, “beauty and power” was determined as the design theme. Also, a requirement sought by the customers of the targeted market was to have a distinctive presence not overlooked even in a crowded traffic running with other motorcycles. Another important requirement was the convenience in daily use. To fulfill that, styling design featuring curved surfaces, front and rear designs appealing its distinctiveness, and the convenience-centered design for the handle bar and its neighboring area were implemented. As a result, the next generation PCX design was completed.

1. Introduction

The first-generation PCX, which was released in 2009, was developed as a premium sit-in type scooter. In the 125 cc and 150 cc scooter segments, there had not been such sit-in type model up to that point. In Japanese market, they, especially 125 cc class scooters, have been recognized as highly cost effective vehicles. This is due to relatively low maintenance costs such as lower tax and the benefits of the insurance system that can be combined with the coverage of four-wheeled vehicle, as well as better fuel efficiency. The PCX was launched as a higher value added model having product appeals equivalent as those of higher class models, bestowed the model a status distinguished from other models in the same class. As a result, the model met practical needs such as commuting. In addition to that, the model was recognized as a highly value added product that satisfies the feeling of ownership. The customers highly appreciated the features distinguished from those of the other vehicles in the same class. One of such features is the comfortable seat that is usually hard to realize on a vehicle body of such compact dimensions. Fine balance of practicality and higher added values was recognized in various countries other than Japan as well (Fig. 1).

This paper describes the exterior design of the fourth generation PCX released in 2018. The final exterior design of the fourth generation PCX is shown in Fig. 2.

For comparison, Fig. 3 shows the external views of the

first and third generations of PCX, both commercialized so far. For the second generation, the functions were reviewed and the frame structure was revised. However, the design had not been changed. Therefore, in this paper, the second generation model will not be picked up for reference.

Acclaimed in many countries in the world, PCXs have been developed inheriting their unique styling design while incorporating advanced equipment of each era. All the previous models have received high praise from customers, and, same as the total vehicles themselves, there is nothing to be picked up as a dissatisfying point in their designs. For that reason, it was rather not easy to determine what kind of design to deliver to customers in the fourth generation PCX. In these circumstances, the design development of the fourth generation PCX was started.

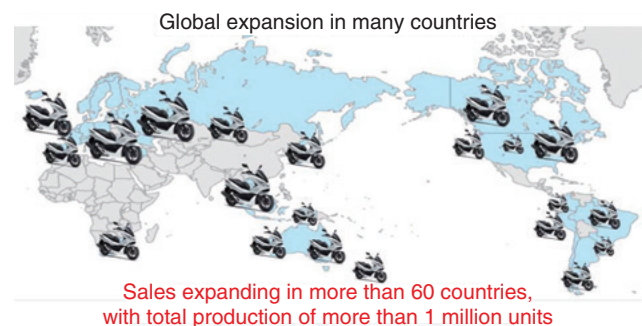


Fig. 1 Global sales of the PCX

¹ Honda Motor Co., Ltd.

² Honda R&D (India) Private Limited



Fig. 2 Fourth generation of the PCX released in 2018



Fig. 3 The PCX of each generation (top: first generation, bottom: third generation)

2. Exterior Design of PCX Relative to Customers' Expectations to Vehicle

When I started the exterior designing of the fourth model of PCX, I thought that the most important thing is to experience the feelings of the customers by being practically involved with the PCX, as the former PCXs have been well received all over the world. Because of this mind-set, I made up my mind to become one of the PCX users when I was assigned to this development project. Through this experience, I could realize the good points the PCX has, such as the vehicle size suitable for riding in a congested traffic environment and the gauge indicators providing good, comprehensive information. Based on what I experienced, I could get a thorough understanding about the importance of usefulness that the PCX offers as a commuter vehicle along with how Japanese users recognize the importance.

Furthermore, we wanted to get more knowledge in Thailand, which is one of the major markets for the PCX, as well, from a closer view in the real conditions to know the customers' acceptance and the environment of the vehicle use. Accordingly, along with a design modeler, I went and stayed in Thailand to take part in the field survey conducted there. This was to know the difference of traffic environment between Japan and Thailand and to understand the kinds of conveniences that Thai customers pay attention to, and to listen to their opinions to this scooter based on these information.

Through the interviews, a significant thing we learned was that the customers in Thailand appeal themselves by showing their motorcycles they ride. The entities they claim have two orientations. One of them is their status and the other is their safety consciousness.

The first thing to be noticed as the background, the traffic situation in Bangkok, where so many four-wheel and two-wheel vehicles are running mixed together, was exhibiting a different status from that of Japanese traffic congestion. The PCXs were recognized immediately even in that kind of traffic circumstance compared to cub type motorcycles or other motorcycles and scooters, surrounding the PCXs. In particular, the vehicle front area shaped with a large-scaled form that comprises the lighting equipment was taking charge of asserting their existence (Fig. 4). This assertion of existence satisfies customers' minds of having status, and at the same time, contributes to their safety by letting the surrounding people be aware of their existence in the traffic. On the other hand, such a feature was not felt in the rear appearance of the PCX passing by. As the photo shows (Fig. 5), the insistence of tail light unit was not strong enough.

As the next step of our survey, we directly met the PCX users to hear opinions from them. Men in their 30s are target users of the PCX in Thai market. We have met one of those users who was an owner of the third generation PCX. Here we introduce an example of such interviews (Fig. 6).

As same as Japanese users, commuting was the user's main purpose of using the PCX. In addition, the point he liked the most about the PCX was its styling design. He liked its dignified form, which is comparable to scooters in upper segment of 250 cc class, and the elegant styling design different from those of cub type motorcycles and other



Fig. 4 View of intersection in Bangkok. Front area design of the PCX is distinctively standing out from other motorcycles.



Fig. 5 View of intersection in Bangkok. Rear area design looks less distinctive.



Fig. 6 User survey in Bangkok, Thailand

scooters. The customers' needs for exterior design of the vehicle also appears in installed customized parts. However, regarding customized parts as well, any significant difference was not found compared with the vehicles that Japanese users own.

There is a customary practice in Thailand that the hierarchy they belong to is exhibited by what they own. That is also applied to parking spaces where the designated parking areas are defined depending on the engine capacities and vehicle types of motorcycles (Fig. 7). In other words, I reaffirmed their unchallenged dignity that does not get melted into surroundings but asserts their existence.



Fig. 7 Parking lot exclusive to high-end scooters such as PCX

3. Exterior Design Development for Fourth Generation PCX

For the exterior designing of the developed PCX, requirements and directions, which will be mentioned below, were defined in order to satisfy the former customers who recognized a great value in the exterior designs of preceding PCXs.

3.1. Exterior Design Satisfying Good Styling and Usability

With the development key word of "Personal Comfort Saloon," the PCX has evolved over the times to provide new values to the customers around the world. The first generation of the PCX has been marketed mainly in Japan and some Asian countries. From the third generation, it has been sold to customers all over the world and that resulted in having good reputations. Moreover to say, the points highly evaluated were common in all countries. The PCX is the only scooter that has combined characteristic of 125 cc class size and weight while offering sitting-in seating. The preceding PCXs have realized a good balance between

usability that fulfills almost all the requirements from users in commuting use and good exterior design that satisfies their desire for ownership. That is to say, the PCXs have followed a good feature that realized both presence and usability exceeding other scooters in the same class. In order to make the developed product good enough as the fourth generation of the PCX, we wanted to create a next-generation version of styling design for the exterior of the PCX, while letting it have a required presence and usability, and moreover, advanced features that have been inherited through the former models as their development key concept. On the other hand, design points that contribute to enhance subtle usability, which includes arrangements of the equipment around the handlebar, displays on the instrument panel, and the storage compartment, were sophisticatedly enhanced. In addition, we kept in mind that the design should be so arranged to provide comfortable operation feeling.

3.2. Inheriting Former Generations' Stylings, Setting Exterior Design Direction for Fourth Generation of Model

In the previous section, basic elements required for the fourth generation model and the things that the preceding PCXs have established were described. However, it took time to determine the direction of styling based on these factors. Each of the styling designs of the first and the third models had a unique personality and had individual fans for each design.

The styling design of the first PCX expressed elegance and gained the support of many people. The styling of the third generation PCX expressed advanced features such as the use of LEDs in lighting equipment, and many people supported this design as well.

Therefore, we wanted to make the exterior design for the fourth generation model of PCX to have both elegance of the first generation and technically advanced sense of the third generation. By doing so, we desired to have the both customers who favorably received either of those exterior designs be satisfied (Fig. 8).

3.3. Exterior Design Concept for Fourth Generation PCX

The exterior design concept of the fourth generation PCX was set as “Beauty and Power” (Fig. 9).

In addition to elegance that the first generation had and technically advanced sense that the third generation had, the styling design trend of the recent luxury four-wheeled vehicles was also reflected to the design. In expressing the contrary directional keywords of beauty and power into one design modeled creation, we aimed a creation of design modeling expression that feels the existence of massive lumps while exhibiting the beauty emanated from smoothly connecting surface shape bridging the front and rear areas of the vehicle body. In this way, we tried to realize elegant beauty and powerfulness. Furthermore, not only the vehicle

body shapes but the same design modeling expression was reflected to all the parts designs including the lighting equipment.

3.4. Co-creation with Overseas Designers

In order to increase the diversity of ideas for the exterior design, we held a styling study meeting with designers from various countries. This was to realize a design used more globally than ever by jamming up the designers' senses all over the world beyond only the involvement of Japanese people. In response to the design concept of “Beauty and Power,” individual design senses they had were expressed in their sketches. After many discussions we had with overseas designers, we came back to proceeding the work to achieve a styling design suitable for the fourth generation PCX model (Fig. 10).

3.5. Pre-crafted Clay Model and Evaluation on It

For the development of the styling design for the fourth generation PCX, different from the steps we usually take, we decided to craft a clay model to start with preceding to the finalizing of the styling sketch. I selected some idea sketches and examined the modeling. PCX is not a product that pursues users' hobby interest, but a commuter vehicle that supports everyday life. Convenience in daily use was a crucial factor in addition to good appearance. On the other hand, in the vehicle development, the redesigning of the body frame configuration was also envisioned in the development items in addition to the change of the engine unit. In the motorcycle exterior designing, the finalization of the styling sketch practically fixes the shapes of the



Fig. 8 Styling design concepts for first and third generation models



Fig. 9 Design concept for fourth generation model

vehicle frames and the way how the frames are arranged within the vehicle body. Therefore, once the styling sketch is finalized and the clay modeling is started after that, it has already become hard to apply detailed arrangements reflecting thoughtful consideration that leads to usability enhancement, and small changes that have something to do with the feeling arose from the design expressions. In other words, the three-dimensional representation that the modeler wants to realize interferes with the frame components that make up the interior of the vehicle body, forcing the freedom of modeling to be compromised. In this PCX exterior design development, I wanted, more than ever, to heighten the modeling quality of the details up to the eminent reaching the ideal level. For that purpose, the clay modeling was conducted prior to the finalization of the styling sketch this time. Considering the area to be added up the lump to have more volume and the area to be scraped out to decrease the volume, the pursuing clay modeling was made up and the shape was measured to get digital data and immediately check its feasibility with frame designers. With this process, we achieved both performance and appearance while securing degree of freedom for the design modeling.

Moreover, in the process of the exterior design creation, we carefully examined the vehicle body surface expressions under the direct sunlight in the open air. The reason why we had special attention to the evaluation results of this out-of-door observation was that, particularly for this model, in the very limited portions with small total surface area usually allocated on a scooter type vehicle body, we were trying

to create curvatures of richly plumped surface and the expressed formation of lights and shades should have been thoroughly examined whether to match what the designer had intended (Fig. 11).

3.6. Determining Styling Design Sketch and Representing into Three-dimensional Objects

In the middle of this styling design procedure, final sketch was selected, and was reflected to the clay modeling work that was already in the process. The selected sketch was challenging to model (Fig. 12). In most of the cases, a sketch drawn by a designer expresses clearly described



Fig. 11 Preceding clay model in photographic views taken indoors and outdoors



Fig. 10 Discussion scene with designers from various countries on sketches, and sketches they prepared



Fig. 12 Final sketch

characteristic lines. However, in the final sketch of this particular PCX, there was almost no characteristic line. Generally, a line created in modeling is a representation of the ridge or valley line made where two surface planes meet. That is, the quality of the constructed surface planes determines the properties of the line. Accordingly, a characteristic line that appears in the sketch is the line created as a result of the surface planes to come across. In the experience expert designers gained so far, they read the properties of each surface from the momentum of the line, and create each solid using it as a guide. However, not any line that could be identified as a characteristic line was represented in the final sketch provided this time, and the usual modeling method could not be applied (Fig. 13). As we could not use a characteristic line as a designing clue, we controlled the surface rises and falls and we continued the design work repeating massing-up and shaping-off relying on the gradations of light and shadow. In each time, we put a silver film and checked the contrast to pursue a beautiful surface. In order to express a sense of vigor and skeletal structure beyond expression of an elegance, we considered the allocation of “where to accumulate power, where to release power” or “where to impose a tension.” Muscles of a human body and various animals were also referenced, while observing the surface representations applied for the latest four-wheeled vehicles. Since the models created from such a process included the possibility of various surfaces that were not expressed by the sketch alone, the interpretation of the viewers varied, and the surface shape could not be determined immediately.

Furthermore, the factors required for the appearance are not only the beauty and newness of the design. Even a slight change in the surface shape will affect aerodynamics and change the penetration path of wind and rainwater. A

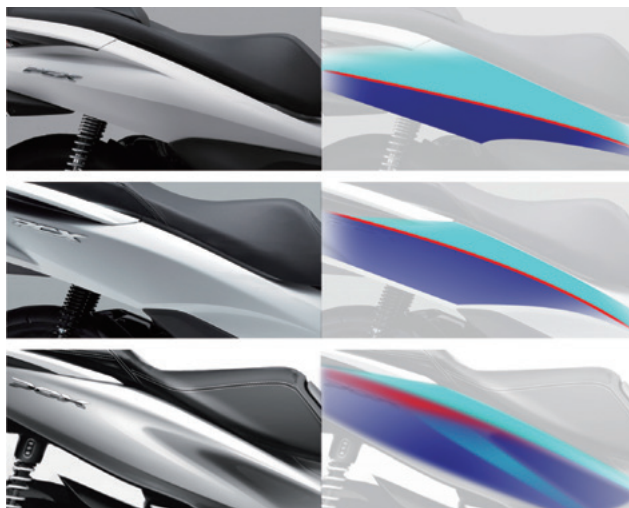


Fig. 13 Previous PCX models and final sketch (top: first generation, middle: third generation, bottom: fourth generation)

two-wheeled vehicle rider is directly exposed to the outer environment because the vehicle does not have a sheltering structure like on a four-wheeled vehicle. Consequently, even a slight change in the surface shape property will affect the comfort of the rider. As the PCX is a motorcycle that asserts itself as a premium sitting-in type scooter, comfort during driving is an especially important factor compared to other models. Therefore, that requirement is more demanding than conventional scooters.

Particularly for the PCX developed this time, it was necessary to represent the newness by the expressions produced by the body surface planes. On the other hand, strong constraints are imposed in its aerodynamics design as the vehicle body exterior surface planes do not have clear characteristic lines. Accordingly, we repeatedly did the trials and investigations to pursue the creation of visual newness while fulfilling the rider's comfort. Variety of opinions arose from the differences in perceptual cognitions produced by a surface plane, and we repeatedly conducted the verification of those opinions by actually crafting the proposed surface planes. We made a lot of efforts in doing this repetitive process and tried to find the direction to be followed. Figure 14 shows the history of changes of the clay model in our efforts. The top left is the one at the initial stage of the development. The transition follows the direction from left to right and the bottom right is the final completed model.

3.7. Further Maturing of Styling Design

Through the above mentioned process, the clay mockup model was completed. When the design process got to this stage, we can, for the first time, move onto the verifications to the quasi product level. When we reach this stage, we felt the need to further strengthen the uniqueness and presence as the fourth generation as well as the freshness compared with the third generation design, which we could not notice at the clay modeling stage.

To achieve that, the designs of front and rear areas of the vehicle body, which are important for assertion of individuality, were revised. Furthermore, the sophistication of the unique design model expression of the side areas of the vehicle body was raised up even more. As a result, while succeeding the characteristics of the first and the third generation models, it was made possible to make a styling that can clearly tell customers that the generation of the model had been changed. The difference before and after the refinement process is shown in Fig. 15.

3.8. Distinctive Characteristics of Developed Styling Design

The characteristics of the styling design developed this time will be described below compared to the styling designs of preceding generations of the PCX.

The design trends of European automobiles in each era have been adopted to the preceding models of the PCX. In the first generation model, the impression of relaxed stretch

with elegance, formed by horizontally based characteristic line flowing through the front and rear areas of the vehicle, was expressed above the swelled shape of well-rounded curvatures. In the third generation model, the front upper and lower bodies, which used to be designed as separate surface planes in the first generation model, were united to engender a sense of unity with the rider. At the same time, as its basis, a surface design expressing high tension was implemented so that the styling was designed to be simple

and modern. As described in the previous chapter, the fourth generation model was designed to express keywords of contradictory properties that were beauty and power. In the model design, on the vehicle body surface connecting the front and rear portions, we tried to create an expression that makes feel the lines weaved above that vehicle body and the body mass expressing the power. For the model design of the side covers, we tried to convey symbolic expression of the convergence of the body masses while



Fig. 14 Design changes of clay model



Fig. 15 Comparison of photographs of clay model designs before and after additional deeper sophistication (top: before change, bottom: after change)

imagining sturdy muscle. Furthermore, the surface plane that gives tense atmosphere was formed in the manner of supporting the center area structure of the side covers from surroundings (Fig. 16).

As for the silhouette of vehicle body sides, the modeling beauty of connecting flowing line with a sense of high quality from the front to rear area of the body and the sense of unity of a rider and the vehicle contribute to generate the presence of the PCX. We thought that should be inherited in the fourth generation model as well (Fig. 17).

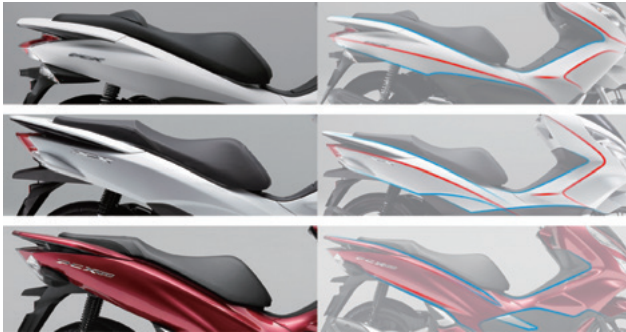


Fig. 16 Comparison of designed surfaces of individual PCX models (top: first generation, middle: third generation, bottom: fourth generation)



Fig. 17 Riders' postures and atmosphere of integration with vehicle body (left: first generation, middle: third generation, right: fourth generation)

4. Design Details

4.1. Front Design

The preceding PCX models have been developed strongly committed to the front designs in order to assert their presences.

The first generation model was equipped with a front combination light that integrated dual headlights, turn signals, and position lights, which occupied the majority of the front area. Due to the size of the area of this integrated lighting system, it was possible to show a presence that no other vehicle had (Fig. 18, top). In the third generation model, the LED lighting system, which was being adopted in European automobiles at that time, was first introduced to a commuter type motorcycle. Characteristic lines using

light guides were represented by white light lines that can be yielded only by LEDs but never by incandescent bulbs. By this application of the device, a new lighting system was realized, and it contributed to the vehicle visibility as well (Fig. 18, middle). In the fourth generation model, the LED substrates were placed diagonally with respect to the headlight that is the main illumination. It is commonly said that diagonal arrangement of an electrical circuit board is hardly feasible but, as this obstacle was overcome, aggressive sharp look of the vehicle face was realized. In addition, the position lights connectedly designed to the turn signal lamps were illuminated in a line manner enhancing the presence. Moreover, the parts in the motif color were arranged so that to bridge over the headlight and the position light. That made the design sharp making a new vehicle face that have never been existed (Fig. 18, bottom).

4.2. Rear Design

In the PCX developed this time, it was desired to realize a tail light with more enhanced distinctiveness and presence compared to the third generation model. For the tail light in the third generation model, some structures were installed within. However, when it was looked from outside, it only exhibited a simple square shape. Furthermore, because the turn signals were placed outer sides, dynamism was missing in its design (Fig. 19, middle). In the fourth generation model, the tail light and stop lamp have got located in the same area to make the best use of the limited space within the rear lighting component. As a result, the degree of freedom in design for impressing the light-emitting part was increased, and a unique and sharp impression could be given by the strip-like shapes located at upper and lower rows extending in the left and right. By placing the turn signals between the upper and the lower rows, it was integrated with the entire design of the rear lighting



Fig. 18 Comparison of headlight designs (top: first generation, middle: third generation, bottom: fourth generation)

component, and was able to show its presence when lit (Fig. 19, bottom). By applying these designs, when the vehicle is viewed from the back as well, it was made possible for the vehicle to be clearly identified as the fourth generation PCX even at a glance.

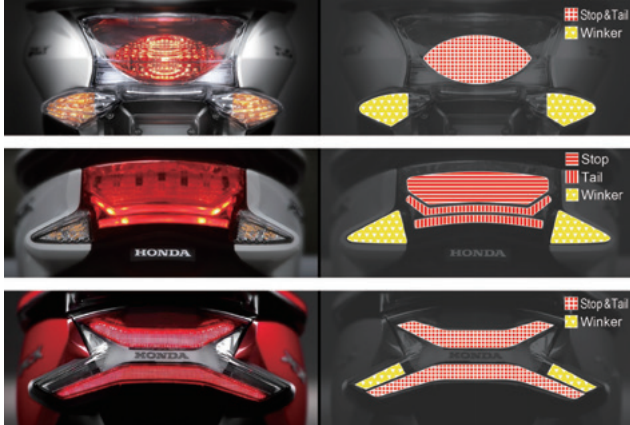


Fig. 19 Comparison of designs of rear combination lamps (top: first generation, middle: third generation, bottom: fourth generation)

4.3. Cockpit Design

As a scooter used for daily commuting, when usability is considered, it is very important for the information presented by the displays and indicators to be enabled to be confirmed instantly. In the fourth generation model, the visibility was enhanced further while expressing the advancement in functionality by applying full-digital indicators. We decided to give priorities of the indicators' visibility in the order of speedometer, clock, fuel gauge, and odo/trip meter. These are arranged in the center of the gauge unit so that the desired information can always be recognized with priority. The turn signal indicators have a characteristic shape that matches the overall design of the instrument panel. The other indicators are arranged on the left and right sides of the gauges to make it easier to grasp the presented information when lit. In the design of gauge indicators' numbers and letters, the letters of numbers for speedometer, which has the highest importance as information, were given the largest font size. Regarding the other information, we changed font sizes appropriately while paying attention to their spacing as well to get better visibility (Fig. 20). Moreover, the digits for numbers used for speedometer have increased to 33 segments per digit in order to obtain better legibility than the commonly used one-digit 7-segment characters. We made the displayed

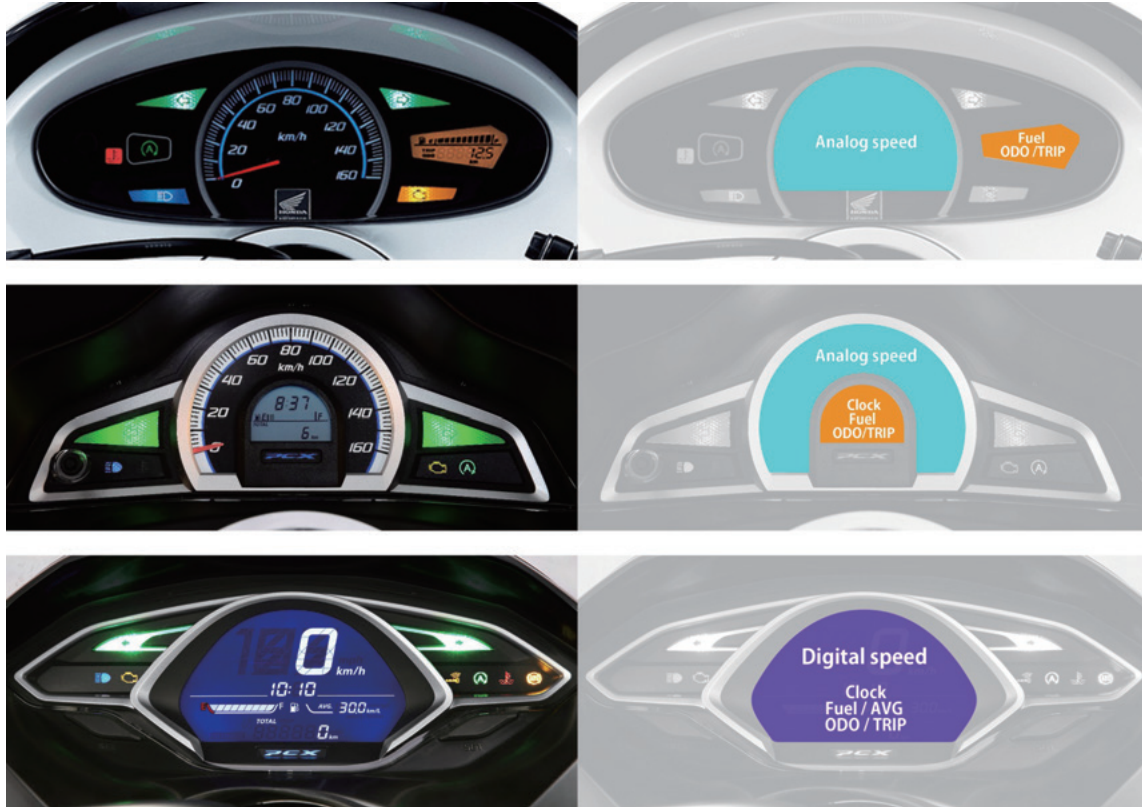


Fig. 20 Designs of meter units of previous and current models (top: first generation, middle: third generation, bottom: fourth generation)

characters stand out by using a different font from those used in other locations (Fig. 21).

For the handlebar, bar-type handle was adopted that is inherited from the first generation model. This is also a result of considering users' convenience such as attaching accessories around the handlebar. In the first and third generation models, the handlebars and its adjacent were chrome plated. In the PCX developed this time, the design was changed to the design featuring black color as well as chrome plate. The shape of the handle weights was the same from the first generation to the third generation models. However, in the fourth generation, special attention was paid to the shape of the handle weight as well, realizing a higher quality appearance (Fig. 22).

4.4. Engine Unit

We had been planning a large scaled advancement in the power unit compared to previous ones such as by having a derivation model of hybrid engine type (hereafter HEV) and the enhancement of drivability with an increased capacity of the air cleaner. Along with that, the engine unit including the air cleaner was entirely redesigned to express its evolution. As a conventional design trend, the belt converter cover and the air cleaner cover are generally separated.



Fig. 21 Comparison of number of segments (left: 7 segments, right: 33 segments)



Fig. 22 Designs of handlebars and their adjacent areas (top left: first generation, middle left: third generation, bottom left: fourth generation), and handle weights (top right: first and third generations, bottom right: fourth generation)

However, in the fourth generation, the belt converter and the air cleaner were united in its styling design as if they are one functional part, and a design with a sense of unity was applied. The belt converter cover has a design that combines a sharp shape reminiscent of a swing arm and a circular shape that feels the pulley built into the cover (Fig. 23). In the future, this engine unit developed this time is expected to be used for many other products than the PCX as well. Therefore, the design of the engine unit, even as a separate component, was designed to have a universal image that can be applied to any product while maintaining the appearance that reminds evolutions.

4.5. Wheels

The developed PCX adopted wider tires. Along with this, the design of the wheels was also renewed. The spokes have a shape that becomes wider from the hub to the rim, and it has a silhouette that makes you feel the movement to the direction of rotation. Through these design effects, we aimed to give a sturdy impression and to make it look attractively beautiful when rotated as well. In addition, the number of spokes was increased from 5 to 8 from the preceding model, thereby producing elegance when rotated (Fig. 24). In the process of this design work, a prototype

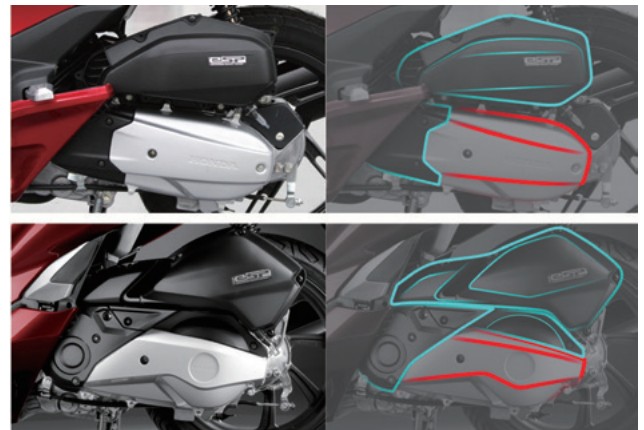


Fig. 23 Design of engine unit (top: first and third generations, bottom: fourth generation)



Fig. 24 Wheel design (left: first and third generations, right: fourth generation)

wheel was actually rotated to see if the representation when rotating looked beautiful and the evaluation results were repeatedly reflected to mature the details of the shape. In addition to the favorable effect on the styling design, the increase of the number of the spokes also enabled to decrease the radial stress on the rims and contributed to make them thinner. The hubs and neighboring area were also lightened, and the wheel could be made lighter by 0.7 kg (front and rear wheels combined) compared to those of the previous generation model⁽¹⁾.

4.6. Coloring Design

In the coloring design, we aimed for a high-quality impression with elegance and calmness. Figure 25 shows the color lineup of each country. We took into consideration that the color lineup in developed exterior design would feel a globally united image. The coloring design was determined considering that the beauty of the styling model created this time would stand out best. Especially, the tone of the red color is the coloring that heightens the impression of elegance and high quality by enhancing the beauty of

modeling design of the fourth generation PCX. The silver is a color tone that beautifully expresses the protrusions and dents of the vehicle body surface planes together with a high quality impression brought by the calm color tone. Black and white colors were adopted as standard colors that are widely loved by any age groups.

In this development, the coloring for the model with a hybrid engine was also designed at the same time as that for the gasoline engine model. In order to express the difference from the coloring of the gasoline engine model, we modified not only the body color but also the color of the functional parts. Details will be described in Chapter 5.

4.7. Completion of Mass Production Model Design

Applying the methods described so far, the exterior design of the fourth generation model was completed. No effort was spared to achieve the distinctive presence and ownership desired for the fourth generation PCX. The distinguished presence, and satisfaction of ownership, sought for the fourth generation PCX had been achieved by implementing all the necessary measures. These measures



Fig. 25 Color variations applied to major markets

range from the whole vehicle body styling design, including the beauty of the modeling expression that connects the front and rear areas of the body, vehicle outer surface with a distinctive surface plane structure, and the coloring design that enhances the impression, to the designs of functional parts created with attentions paid even to the subtle details (Fig. 26).



Fig. 26 Completed design of mass production model

5. Exterior Design of Hybrid Model

The PCX hybrid is a model equipped with the hybrid system firstly introduced for motorcycles. For that reason, we wanted have the design exclusive for the hybrid model giving the different impression from the gasoline engine model. We made it to have the sense of unity as one of the Honda hybrid products by adopting the emblem with the hybrid logo same as in four-wheeled hybrid vehicles of Honda. Therefore, from the sketch stage, we examined how the logo was investigated in the exterior design. Those investigations included the balance of the exterior design

with the PCX logo, the position and size of the emblem to be attached to the vehicle body (Fig. 27).

In addition, the blue color, which symbolizes hybrid vehicles, was applied to the line shape illumination parts of the headlights and the tail light, the seat stitches and the ornament of the smart key. We represented the sense of unity by applying a deep dark blue color to the vehicle body while making the accented color of blue, which belongs to the same color system, stand out. The luggage compartment under the seat, because a part of that space is occupied by lithium-ion batteries, was specially designed to accommodate a full-face helmet. As mentioned above, the quality of the completed vehicle was brushed up by applying intentions of the designers not just to the whole-scale vehicle styling but to the finest details as well (Fig. 28).



Fig. 28 Completed design of mass production hybrid model



Fig. 27 Color sketch and mass production model of hybrid model

6. Conclusion

In the development of the fourth generation PCX, we endeavored to design a product that offers a new satisfaction together with or beyond the advantageous aspects the preceding PCXs had. In this development, the designers' involvement was not limited to the outer covering parts but included all the functional parts. The design of the fourth generation PCX is the culmination of the strong intention the design team had and the achievements engineers realized with the high technologies. This model has been delivered to 355,000 customers worldwide by December of 2018 since the first delivery in Vietnam in December, 2017. It is reported that the model has been accepted with a good impression, even to the users of the first and third generation models regardless whether in Japan or overseas. I feel confident and proud that this PCX has become a product that customers around the world can truly appreciate. I sincerely hope that the PCX brand we have built will continue to be loved as a premium everyday commuting vehicle for customers all over the world.

References

- (1) Omori, J., Shokaku, I., Takayama, K.: Development of 2018 PCX and PCX HYBRID, Honda R&D Technical Review, Vol. 31, No. 1, p. 41-48

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Design of Instrument Cluster and Control Switches for 2018 Gold Wing

Taishi INOUE¹

ABSTRACT

The Gold Wing was developed as a large luxury touring motorcycle. The styling design of the instrument cluster and control switches is therefore important in creating an enjoyable experience for the rider through their appearance and operation. In its first full model change in 17 years, the Gold Wing was updated with numerous advanced technologies. That brought significantly increased information displayed and rider controls. The design goal was set as an integrated human-machine interface (HMI). The integration is made among mutually interrelated information on the display and device operations. That enables the intuitive operation and information presented in the most natural way minimizing riders' stress. For the instrument cluster of this particular vehicle, a beautiful appearance is also a crucial design factor. The design objects range the operation parts, the surface textures and the shapes. For those, further advanced designs were implemented aiming at the relevance to the current generation and luxurious feels. By doing that, we tried to make the vehicle worthy deserving the motorcycle of the highest class. These efforts resulted in achieving both the functionality and beautiful appearance at high level. The design contributed to the owners to have joy of riding while satisfying the pride of ownership of a globally recognized motorcycle. In addition, the design has helped the new Gold Wing has received excellent customers' reviews.

1. Introduction

The Gold Wing is Honda's premiere large touring motorcycle. It began with the launch of the GL1000 in 1975, with its pet name "Gold Wing" becoming an iconic brand name over its history that spans over 40 years⁽¹⁾.

For the first full model change of the Gold Wing in seventeen years since 2001, the development concept underwent a complete redesigning given the changes in the market and the generations who buy motorcycles, while also considering taking the Gold Wing brand far into the future. A drastic change was made: the new model would have a light-weight, high-density package for ultimate enjoyment of riding, while maintaining the luxurious presence that has been a Gold Wing tradition. This was a drastic change from the direction of luxury and grandness that the previous Gold Wings have pursued⁽¹⁾. From this complete overhaul of the vehicle's concept, another requirement was established: the new Gold Wing would offer the maximum enjoyment a rider can experience, from its riding performance and daily usability to comfort during long-distance touring.

As such, it was decided that this luxury motorcycle model deserves a full-suite in-vehicle infotainment (IVI)

system utilizing the latest technologies. "Infotainment" refers to the concept of providing information and entertainment in one system. This technical paper provides introduction on the IVI system and its controls, the human-machine interface (HMI), as well as their exterior designs.

Starting with the 1980 GL1100 model, the Gold Wings of the past have come equipped with the latest audio devices for motorcycle applications (Fig. 1)⁽²⁾. The Gold Wing also became the first motorcycle to apply a GPS system in its 2006 minor model change (for north America; 2008 for Japan), cumulating in the establishment of an IVI system integrated with audio.

In the twelve years following this, however, the IVI technologies in the automotive world have advanced greatly, exemplified in the smartphone connectivity functions. This created the necessity for the new Gold Wing to have function updates as well.

The electronic control technologies in the motorcycle world have also made great strides, which led to electronic controls being used in all the key components of the Gold Wing in this development, including the engine, suspension system, throttle, brakes, and transmission. This allows for the rider to have a wider array of options to choose from to

¹ Honda Motor Co., Ltd.



Fig. 1 History of the Gold Wing from perspective of their infotainment systems

meet personal preferences and scenes, such as riding modes and various settings. The expansion of the IVI system functionality and general advancement of electronic control technologies led to an increase in the volume of information communicated to the rider through the instrument cluster (an integrated unit containing all the various gauge units essential to vehicle operation), as well as an increase in how much a rider must or can do to operate the motorcycle. The overall styling design of the new Gold Wing was also completely refreshed to be in line with the project concept, while maintaining a high-quality feel that has been the hallmark of this iconic motorcycle model. The styling designs of the instrument cluster and handlebar switches/buttons had to be harmonious to be integrated into the overall styling.

Since the styling designs of these components would have a direct impact when the vehicle is being operated on the road, good legibility and ease of use should be prioritized over their shapes and textures.

With that said, large touring motorcycles such as the Gold Wing are also seen as luxury leisure products and their users expect beautiful forms, colors, textures and feel in these products to command the high prices at which they are offered. When parked in a street corner, people will recognize a motorcycle's instrument cluster as part of the vehicle's exterior design. The team took this into consideration and engaged in developing a styling design that leaves a strong impression of beauty and quality even when the motorcycle isn't being operated. Figure 2 shows the area in a complete vehicle.

2. Changes of User-related Information and Expectations for the New Gold Wing

2.1. Up-to-date IVI and Vehicle Body Electronic Control Technology Enhancement

Apple CarPlay (Apple CarPlay is a trademark of Apple Inc.) released in 2014 and Google Android Auto™ (Google Android Auto™ is a trademark of Google LLC), are services



Fig. 2 Instrument cluster, handlebar switches, and center panel of developed Gold Wing

developed to allow people to use their smartphones in their cars that have been applied to products by automotive makers and IVI device makers worldwide^{(4), (5)}. This led to the establishment of an environment in which the functions of smartphones the people have become familiar with, such as communication, audio and GPS navigation, can also be used while driving.

This change in operational environment in automotive caused motorcycle users to want the same. The functionality of such systems were considered especially useful for the Gold Wing, which is primarily for touring purpose, particularly the voice-command operation seemed more beneficial to motorcycle riders than automotive drivers. The Gold Wing being Honda's only motorcycle with IVI, the team saw the potential in customer demand for the latest infotainment system with excellent connectivity befitting this model. As such, it was determined that the new Gold Wing would have the IVI system compatible with Apple CarPlay, the first in motorcycle application.

The new electronic controls added to the model include Walking Mode⁽¹⁾ in which the vehicle can be more easily maneuvered when feet are touching the ground, and four ride modes⁽³⁾ of Tour, Sport, Rain, Econ to best match the scene and rider's riding style.

2.2. New Challenges

(A) Method of displaying increased amount of information

The enhancement of IVI and electronic controls created the need to add a variety of status indications, and that means that necessary information must be immediately recognized without causing misunderstanding.

Enlarging the instrument cluster to accommodate the increased number of items to be displayed would go against the development goal of making the vehicle body more compact. Making the icons and text smaller would not be acceptable for the reason of negatively impacting their legibility. The challenge here became the organization and streamlining of the information and displaying them in the most clearly and easily understandable way.

(B) Redesigning switches to accommodate information increase

With the increase of available options, items needing user input were added; the focus then here is to allow the rider to operate the system as needed quickly and easily without causing any errors to be made.

Up to the previous model, most of the operations related to the audio or GPS navigation systems had one control switch or button per function. This meant that the body panel was full of control switches and buttons but remembering where they were allowed for simple and easy operation. The handlebar switch boxes were also at the largest possible size that the fingers could reach (Fig. 3). Increasing the number of controls to correspond with the additional operation items would contradict the team's goal of making vehicle body more compact. Making the control switches and buttons smaller would also not be acceptable for that would impact the model's maneuverability. Making the control switches and buttons smaller would also not be acceptable for that would affect vehicle operability. It was therefore determined that the operational method should be redesigned from ground up for simplification of operation, as well as decreasing the number of controls and making them smaller.



Fig. 3 Instrument cluster, handlebar switches, and body panels of previous model

3. Solving Challenges with HMI

It is of utmost importance that the riding operation not be hindered by visual recognition of information and operation of devices, therefore the target became an excellent linkage between the information display and the operation controls designed in a way that is clearly understandable and intuitive to use. Which means that there should be a strong awareness of HMI in creating a design that integrates the information display in the instrument cluster and the operation controls. Also, the styling design of the instrument cluster of a motorcycle should be in harmony with the styling design of the overall vehicle as it is also part of the exterior design.

- Realize excellent instrument cluster legibility suitable for a premier motorcycle
- Maintain proportions and voluminousness of overall vehicle
- Create a feel of quality and class in the exterior requisite to keep Gold Wing's legacy

More details will be described below with consideration to these points of legibility, operability, and design.

4. Legibility of Instrument Cluster

4.1. Configuration of the Overall Instrument Cluster

As part of the instrument cluster that already requires only minimum eye-scanning amount, a 7-inch, multi-functional and full-color TFT liquid crystal display (LCD) unit was applied. Its size, at 7 inches, is among the largest in motorcycle application. It also allows the settings to be made for the navigation, audio, telephone and various other functions.

The speedometer and tachometer were applied with an analog styling design using pointers and dial faces and placed on the left and right of the center display respectively, creating a more stretched out, streamlined look. This is a change in direction from the previous model, which had the two gauge units placed above the center display creating a stack in the center. With the new configuration, the functions are placed along a narrow line between the two rear view mirrors, allowing for the rider to check the information effortlessly and efficiently within the eye gaze area to check the rear view mirrors (Fig. 4).

4.2. Targeted Effect of Analog-style Gauge Units from HMI Perspective

As indicated in Section 4.1., much of the information displayed in the instrument cluster including the full-color TFT-LCD screen is in text. To counterbalance that, the traditional "analog" style of gauge display design was chosen on the basis of allowing the rider to understand engine speeds and vehicle speeds intuitively by position of the pointers rather than reading numbers in text.

The "analog" display method also allows to add functions to the direction to which the pointer points. The

tachometer can be set at an angle so the pointer points straight up at peak power output to notify the drivers and riders when to shift the gear during motorsport and races (Fig. 5). The same can be done with speedometers to point up when cruising on highways.

In this development of the new Gold Wing, the system was designed so the pointers of the speedometer and tachometer point close to horizontal and perpendicular angles at 55 mph and 1800 rpm (in main market of USA), respectively.

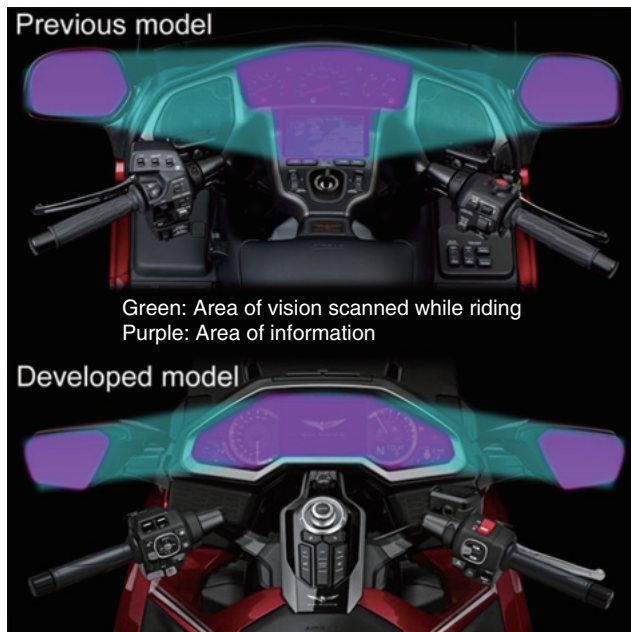


Fig. 4 Comparison of visual areas classified by gazing time of rider's eyes

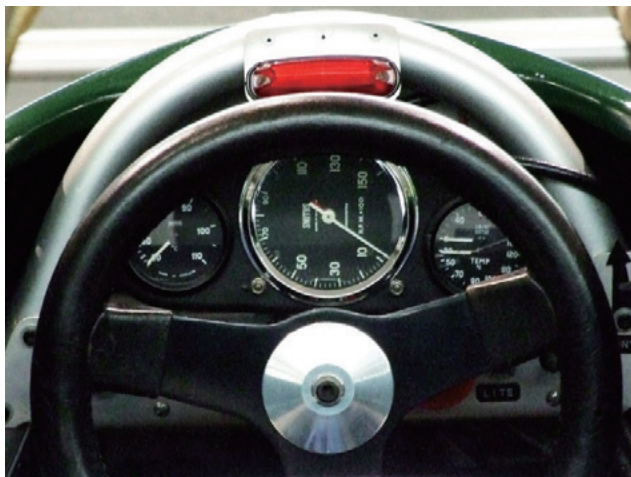


Fig. 5 Rotated tachometer on 1980 Ralt Honda racing car optimized to show upshifting of gears

4.3. Analog Gauge Unit Placements

4.3.1. Smaller size

If good legibility is needed for an analog-style gauge unit, it should be made large. However, a larger instrument cluster would greatly impact the exterior form of the vehicle and go against the development concept. The challenge here was to secure excellent visibility for an instrument cluster with limited area.

Visibility and legibility are also related to eyepoint distances. The gauge units would appear larger if seen from close proximity, so making them smaller while providing good legibility is possible. Here, the sizes in appearance are shown as areas covered by rider's view by using viewing angles. The gauge units of the previous model were approximately 1000 mm away from the eyepoint of a rider with an average build; the same distance for this development model is approximately 800 mm (Fig. 6). This is due to the rider position shifting forward with smaller vehicle body and power unit, as well as the steering axis offset to the rear that occurred as a result of the application of double wishbone front suspension⁽¹⁾. To secure the same level of viewing angle as the previous model with gauge unit diameter of 110 mm, the new Gold Wing with a shorter distance was set at 90 mm diameter (Fig. 6).

4.3.2. Compact arrangement

Simply placing two 90 mm gauge units on the right and left of the 7-inch, full-color TFT-LCD screen would create a center distance of 263 mm. From an appearance styling design perspective, the balance seemed off with the two gauge units being too far off to their respective sides. Also

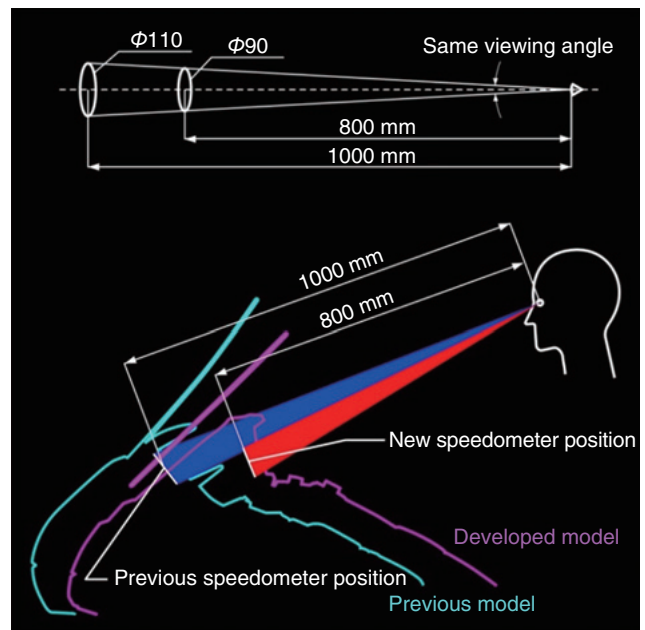


Fig. 6 Comparison of positions and diameters of speedometers in light of viewing angle

from an HMI perspective, this placement overly stretched the viewing angle in the horizontal direction, creating too great a distance between the sets of information seen on the right and left sides of the display to glance during riding. The components had to be brought closer to center. A simulation was carried out with even smaller gauge units at 80 mm diameter. From a styling design perspective, the inner widths remained unchanged even with a smaller diameter, resulting in the two circles on the sides appearing even more separated. The pointer travel distance remained at approximately 5.5 mm in mixed traffic, proving the decreasing of the right-left pitch being not as effective in comparison to the legibility sacrifice made with a smaller gauge unit diameter (Fig. 7).

Unlike the clock, the pointers of “analog” style circular gauge units such as speedometers and tachometers will not go around 360 degrees. It is therefore possible to cut out areas where pointers do not need to reach. The inner areas of the gauge units facing the center display were cut out in an attempt to decrease the necessary amount of eye scanning. The cut out ratio was set at 15% to maintain the impression of the gauge units appearing circular. The center distance was shortened from 263 mm to 235 mm while maintaining the 90 mm diameter for the gauge units, giving good legibility of essential information for riding with an overall width equivalent to the previous model (Fig. 8).

4.4. Details of Instrument Cluster Layout

Figure 9 shows the overall layout of the instrument cluster. A full-color TFT-LCD screen is positioned in the center. Black-and-white LCD screens are placed in the lower left part of the speedometer and from the bottom

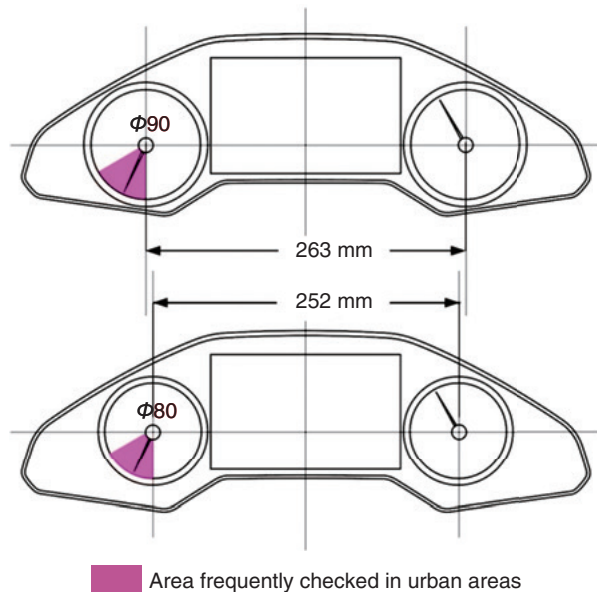


Fig. 7 Study of gauge display arrangement applying composition of full circles

to the lower right part of the tachometer, indicated by red frames numbered 1, 2 and 3. These areas display at all times the information necessary for riding the motorcycle. Their exact placements and the types of information shown will be explained hereafter.

The warning lights that indicate failures of functions are placed toward the center top of the area necessitating minimum eye scanning effort for immediate recognition (4, 5, 6 and 7 in Fig. 9). The low fuel indicator light (8 in Fig. 9) and water temperature warning light (9 in Fig. 9) are positioned above the fuel gauge and water temperature gauge in the black-and-white LCD screens on the left and right, respectively, to allow the rider to understand at first glance what is being warned.

Items related to the speedometer and distance are consolidated on the left side of the instrument cluster. This area contains the odometer, trip meter, cruise control setting, fuel gauge, and remaining fuel indicator related to cruising distance, among others (1 in Fig. 9). The status of cruise control is displayed on the speedometer face (10

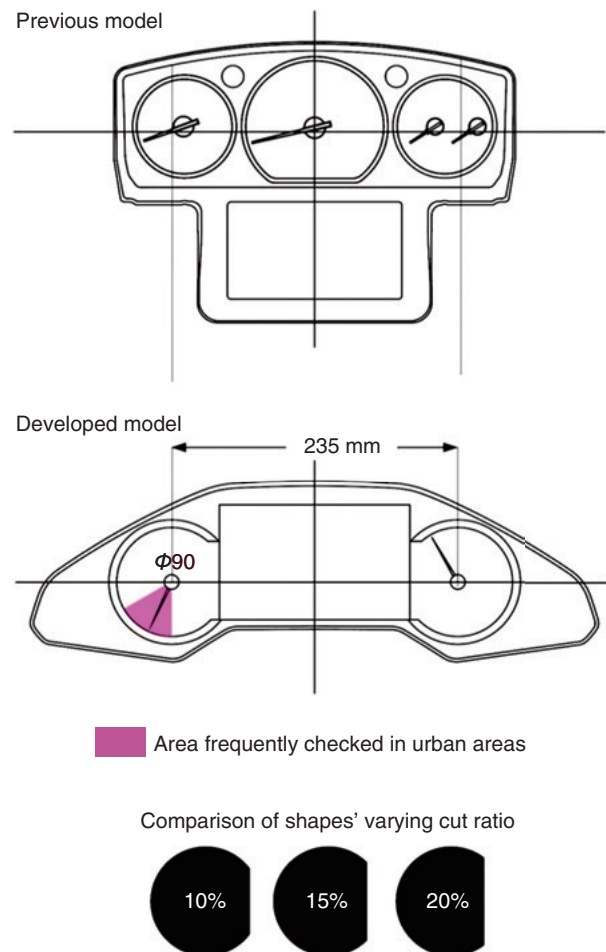


Fig. 8 Examples of space reduction arrangement for rotating pointer type gauge displays maintaining required visual diameters

in Fig. 9), so related information such as the set speed, operating condition, and current speed can be checked in the same area.

Information related to the engine is grouped on the right side of the instrument cluster.

The gear position is shown in letters at a maximum height of 16 mm on the left end of the black-and-white LCD screen (2 in Fig. 9) below the tachometer. Riding mode settings are also displayed in the same area. On the dial face of the tachometer are the neutral indicator and the “idling stop off” indicator (11 in Fig. 9).

The black-and-white LCD screen on the far right (3 in Fig. 9) shows items that are checked and/or set when not in motion or are seldom checked when riding, such as side stand indicator, trunk open indicator, suspension pre-load indicator (set as needed based on number of passengers and amount of luggage), heater indicators, etc., in pictograms.

4.5. Legibility on Multi-functional Information Display

Apple CarPlay and Honda’s proprietary system (for IVI and vehicle settings) are displayed on the same screen, therefore it is important that switching between the two can be clearly recognized and be done smoothly without awkwardness.

The corners on the square icons on Honda’s IVI home

screen were rounded to match the styling of those seen on Apple CarPlay’s home screen. Differentiation was made by giving Honda’s IVI system a different color tone. The text indicating icon names were also made larger and placed next to the icons to clearly differentiate from the Apple CarPlay display. Great care was made in treatment of the text, considering the use of the system while riding a motorcycle. Out of the humanist typefaces known for high legibility, Frutiger was selected for its excellent visibility and legibility⁽⁶⁾, its size on the home screen set at approximately 6.8 mm (Fig. 10). Apple CarPlay uses San Francisco, an Apple Inc. system font, and the size on the home screen is approximately 4.8 mm.

4.6. Settings Display and Limit on Amount of Information Displayed

The setting display is designed so the items appear in a list after reaching a category’s level from the home screen. This shows the options in a vertical manner, simplifying and reducing time of operation of viewing and selecting the items with up/down and select buttons.

In understanding the limit on the number of items on the list that can be recognized in a short time, we referred to American psychologist George A. Miller’s theory of the capacity for human short-term memory being “seven, plus

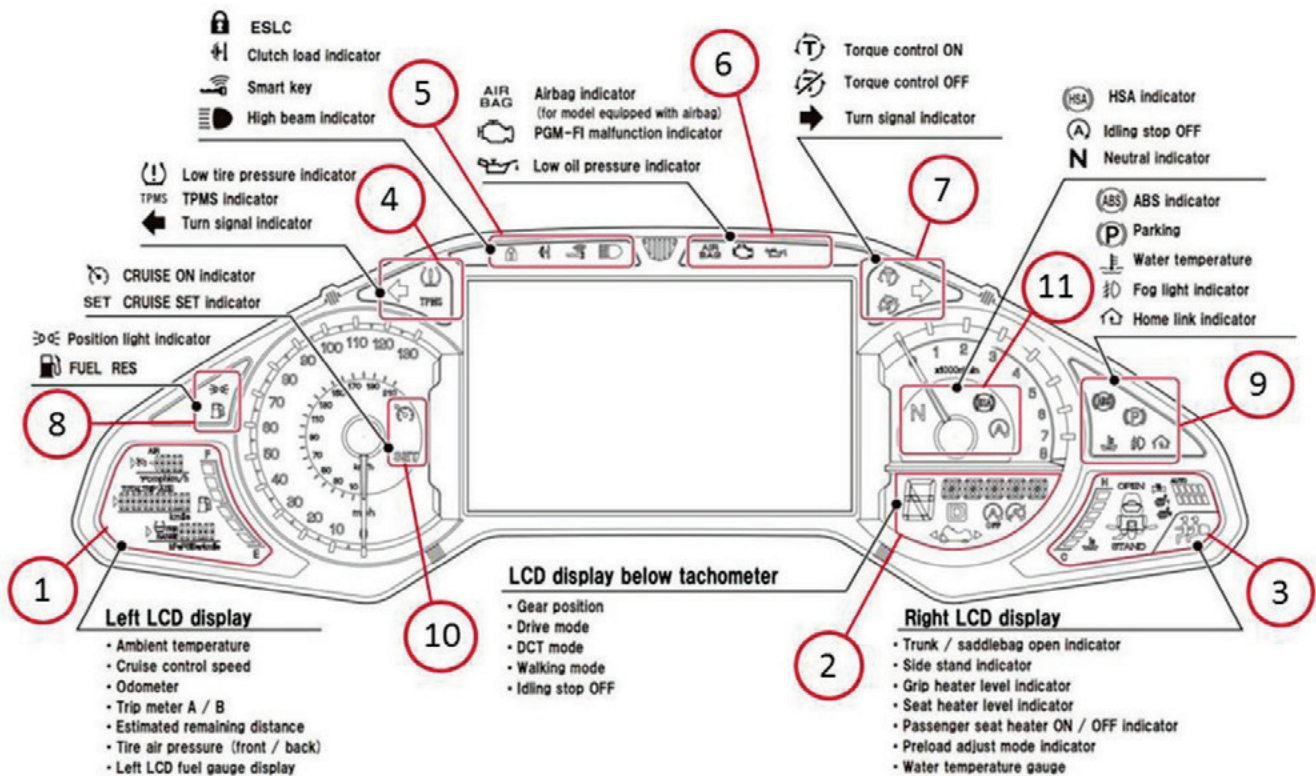


Fig. 9 Arrangement of display components on instrument cluster

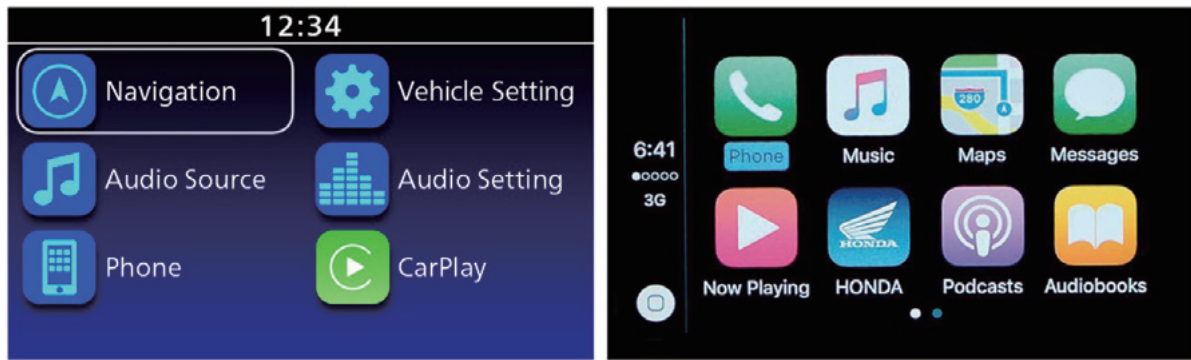


Fig. 10 Honda HMI home screen (left) and Apple CarPlay™ home screen (right)

or minus two” at a time. The “items” here refers to “chunks of information” with meaning, and in this context they are the items on the list including the color tones and settings⁽⁷⁾. In this development of the Gold Wing, we opted to display five (seven minus two) at a time based on aforementioned theory, to allow for recognition of information with a good margin even when riding the motorcycle (Fig. 11).

4.7. Guidance and Feedback Through Colors

In information display, it is important to give feedback as guidance to prompt the rider to make correct inputs, and show response to the inputs made. Those were done through changes in tones of colors on the Gold Wing, taking advantage of the full-color system. The basic colors were white and cyan, with gray added for only when needed.

The following four were applied to the system to allow for the rider to intuitively understand the operations without being explained.

- (1) Selectable options (in white text inside white frame)
- (2) Option user is on to make selection (highly visible cyan with tones reversed inside frame for accentuation)
- (3) Feedback showing selection has been made (migrate to

next operation screen page, or show only the frame in cyan)

- (4) Options that are not selectable (text and frame in gray to decrease the presence)

An example of an actual setting page is shown in Fig. 12.

4.8. Tone and Brightness of Colors on Display

As mentioned above, the new Gold Wing applies three types of display: full-color TFT-LCD, analog-like design of gauges and pointers for speedometer and tachometer, and black-and-white LCDs on the right and left.

First, the difference in their lighting methods and considerations to visibility and legibility will be described. The colored and black-and-white LCDs use the light emittance technology with backlights similar to smartphones and computer displays. With the analog gauge units, indirect lighting from the pointer bases are used to illuminate the gauge unit faces, and light transmitted from the backlight is used on the gauges, letters, and marks, etc.

Motorcycles are used in various lighting conditions such as daytime, nighttime, dusk and dawn, etc., so it is

Menu list items contain group of properties of text, color, and status

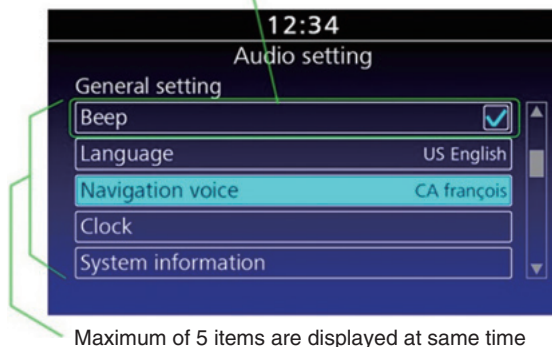


Fig. 11 Example of menu list

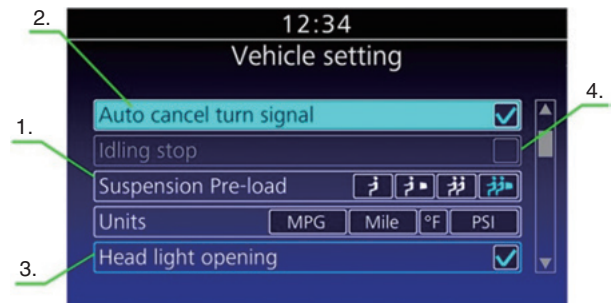


Fig. 12 Example of screen design

essential to give secure good legibility to the gauge units using multiple lighting systems with varying brightness and characteristics. If looking only at the full-color TFT-LCD, giving it a dramatic contrast between the visuals (e.g., in white) and the background (e.g., in black or other dark colors) would allow for good legibility. However, if the rider turns his eyes to the analog-like gauge units after looking at a bright screen, the legibility would be negatively affected by the residual visual effect (Fig. 13). Residual visual effect is a phenomenon in which the light/shade contrast and hues appear reversed. In this development of the Gold Wing, black was selected for the analog-like gauge units for minimum impact from external lighting and a similarly dark color for the background of the full-color TFT-LCD screen to match the analog gauge units, creating an environment in which no dramatic brightness difference will be made. The black-and-white LCD screens on the sides of the instrument panel appear to have white text on black background in daylight, but the background turns dark blue when the backlights come on at night. This is done so the units have the highest possible contrast between the visuals and background for optimum legibility even when the backlights are on. The full-color screen in the center also uses a similar dark shade of blue to give the entire instrument cluster a consistent color scheme (Fig. 14).



Fig. 13 Residual visual effect caused by excessively high contrast on display screen



Fig. 14 Color tone coordination by using unified colors

5. Objective and Configuration of Switches/Buttons for Operability Enhancement

5.1. Coordination Between Display and Operation

It is assumed that riders wear gloves when riding motorcycles, so making the operation switches and buttons smaller would affect vehicle operability. To solve this issue, each switch/button was given multiple functions with the new Gold Wing. To enable this, as was described in the information display section, the information to be displayed on the full-color TFT-LCD in the center was to have multiple levels and the information to be linked with the operational switches/buttons (Fig. 15). The aim here was to have the rider quickly understand visually the system's feedback on what's already selected or what changes were made.

This linkage of display and controls contributed to the streamlining of the switches and buttons by simplifying the basic operations to “select” and “decide.” This method was applied to many other items including various settings changes, music, phonebook, and setting of destination on the navigation, eliminating the complications that could arise from a system having multiple different operation methods.

Operation of the system during riding of the motorcycle must be done without letting go of the handles, so the items are selected using the cross-shaped switch unit on the left handlebar, then decided with the Enter button in its center. The interface dial on the center panel and the Enter key on the dial surface have the same function, but they cannot be actuated when the vehicle is being driven. This dial is used only when detailed settings need to be made when stopped, as well as for complicated operations such as entering text.

Separate buttons were made for the “return” function to take back to the previous page and the “home” function to go all the way back to the home screen for the sake of simplifying rider operation, since these two functions were assumed to be used relatively frequently.

The relation between ease of use and the number of switches/buttons is determined by the balance between operating directly and in stratified menus. In this development, we studied automotive models that are advanced in terms of IVI and identified what functions

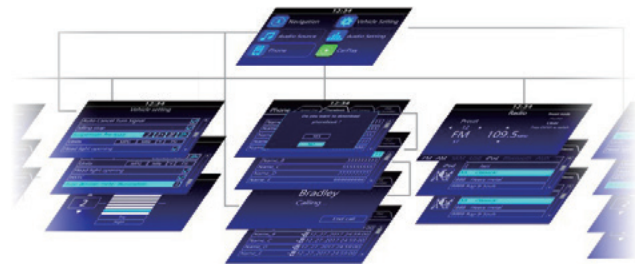


Fig. 15 Layered screen display structure

should be operated directly. Our finding was that those functions that directly impact human senses or cause the driver to feel stress if it cannot be controlled immediately after feeling the need, such as cabin temperature setting and audio volume setting, usually had buttons dedicated to the respective function.

In this development of the Gold Wing, the volume adjustment function for the audio equipment, audio source switching function, and the switches/buttons for opening and closing windscreen, and HVAC functions fall under this category.

The audio source switching button can also mute the sound by pressing it down for a while. This is a convenient feature for when the rider wants to quickly mute the sound in situations such as using the telephone function or having a conversation using the intercom. Attributes such as engine output characteristics, suspension damper attenuation characteristics, DCT models' shifting characteristics also directly impact the rider's feeling. In this development of the Gold Wing, these were integrated to make four riding modes that can be selected with the riding mode selector switch on the right handlebar. The finished right and left handlebar switch units are shown below (Fig. 16).

5.2. Striving to Make a Stress-free Operation System

One element of what makes switch/button operation effortless is to take advantage of what have become the de facto standards after being in use over a long time.

One such example is the on/off dial, where turning clockwise turns the system on and counterclockwise turns it off as does the ignition dial. In terms of using a dial to control the amount or values, a popular method is to turn clockwise to increase and counterclockwise to decrease.

The interface dial applied to the center panel of this Gold Wing can function as a dial-type switch unit as well as to make selections on the display by rotating in eight directions of up, down, right, left and the four points between them (Fig. 17).

Rotating the interface dial allows for a quick scroll-through of the menu listing in a vertical orientation used often in the Gold Wing. This contributes to quick access to lists with a large number of items to choose from, such as music title list and telephone number list imported from a smartphone.

With a list, the general reaction is to start from the top and go down and therefore going down is "going forward" and going up is "going back." The operation of the interface dial of the Gold Wing follows this to "go forward" by turning clockwise and "go back" by turning counterclockwise. The item selected is positioned in the center of the screen at all time, so going down a list would make the list move relatively upward. This means the images on the screen seem to be moving in the reverse direction, but since the rider is going down the list this operation will feel natural and intuitive.

As described above, the system was designed so the

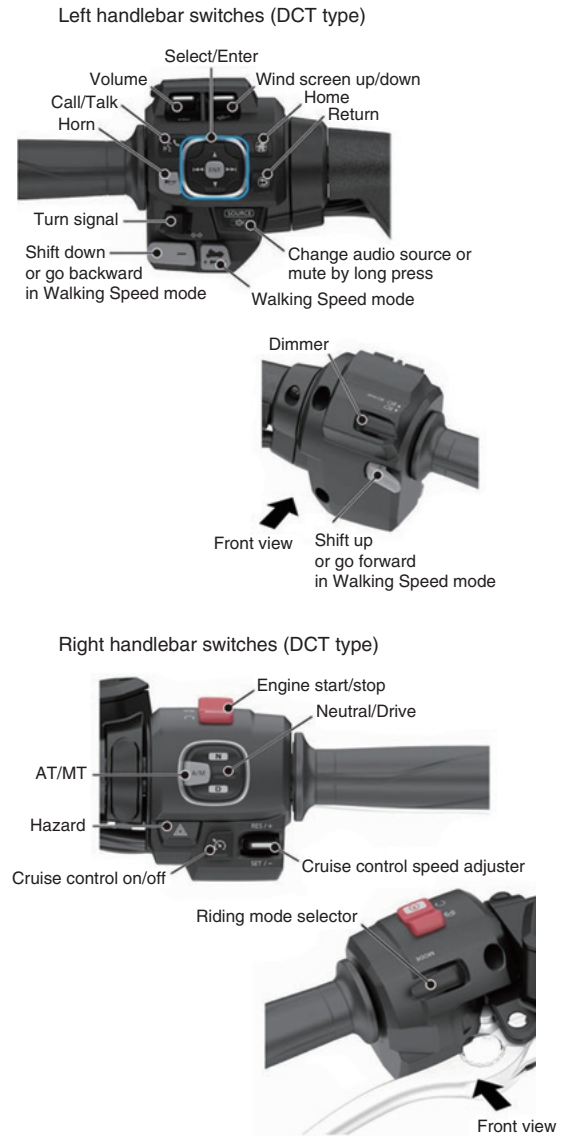


Fig. 16 Handlebar switches

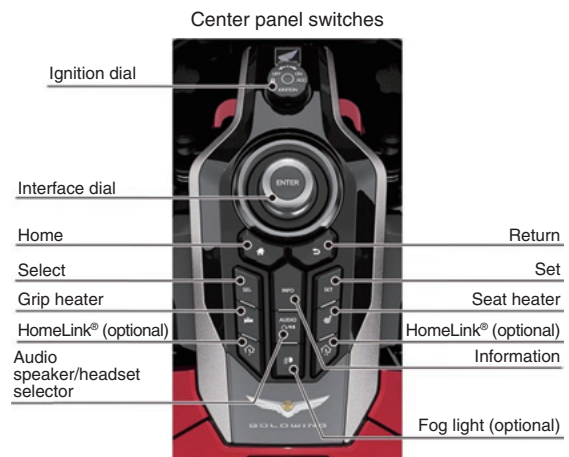


Fig. 17 Switches and dials on center panel

rider will feel that its operation and the information on the displays would feel natural, effortless and intuitive. This did cause the number of items needing rider input to increase, however the operation was streamlined to be only “select” and “decide” for simplicity. This also helped in reducing the number of switches and buttons, contributing to making the vehicle more compact.

6. Creating Attractive Appearance

6.1. Value of Physically Built Analog Gauge Units

The speedometer and tachometer designed in an analog manner are composed of pointers and dial face with an additional metallic decoration ring on their periphery (Fig. 18). This design specifically targets at creating an appropriately luxurious feel for the Gold Wing.

Technically it is possible to display images of gauge units in an analog style using high-resolution computer graphics on the large full-color TFT-LCD screen, as many automotive models already do. The blackness of the surface when the screen is turned off actually adds a feel of powerful presence and depth. The attractiveness of the area can be further improved through the designs of surrounding parts and components (Fig. 19). In case of motorcycles, however, the same approach as automotive cannot be taken as there are much fewer parts around the instrument cluster that can be used to build on beautiful appearance.

The instrument cluster is seen as a part of the styling design of a motorcycle, so it is ideal to have it appear attractive even when the power is turned off. In the case of the Gold Wing, the instrument cluster should become a key feature in creating a feel of luxury by its appearance as well.

There are many interpretations of what “luxury” is. One such definition is that its value withstands the test of time.

In the world of watches, a product segment very similar to instrument cluster design, analog watches remain the most popular^{(8), (9)}.

Supported from this example, we determined that having gauge units actually built with real pointers and dial faces would be essential in creating a universal feel of luxury and high value.

6.1.1. Outer decoration ring on analog gauge units

The outer decoration ring for the gauge units was changed from the chrome plated type used throughout previous Gold Wing models was changed to a circular hairline finish type. The aim was to have the hairline finish evoke a feeling of something that has been machined out of a solid piece of metal, fine lines of diffused reflections and soft reflections of light in the machined direction creating subtle elegance (Fig. 20).

Gauge blocks placed along the decoration ring were made with a transparent material to have a chiseled and mechanical form (Fig. 21). The frame around the instrument cluster was applied with a semi-matte dark silver paint to give the overall tone a well-coordinated look.



Fig. 19 Instruments cluster unit of Honda NSX as example of automotive application



Fig.18 Decoration ring with metallic texture



Fig. 20 Hairline finish on decoration ring

6.1.2. Attractiveness created with light and reflection of light

The outer decoration ring described in the previous section is comprised of a surface that is curved inward in a bowl-like manner, and in combination with the illuminated gauge blocks serves as the receiving end for the pointer tip. This construction created a well-defined three-dimensional form and depth for the overall dial face. This arrangement of components also allows for the light being emitted at the illuminated pointer's end to be reflected at the decoration ring, showcasing the gauge unit in a distinctively alluring "analog" styling (Fig. 22).

The light emitted from the pointer's tip reflects onto the outer ring and moves with the pointer. This gives the impression that the surface being pointed by the pointer is glowing.

The reflection occurs further out from the end of the pointer itself, so this effect makes the gauge units to look larger than their actual dimensions. Doing this with chrome plating would create a single point of strong reflection, but this being hairline-finished, the light diffuses lightly in stripes and the afterglow of the reflection moving along the ring makes it look like a band (Fig. 23).



Fig. 21 Exploded view showing gauge ring construction

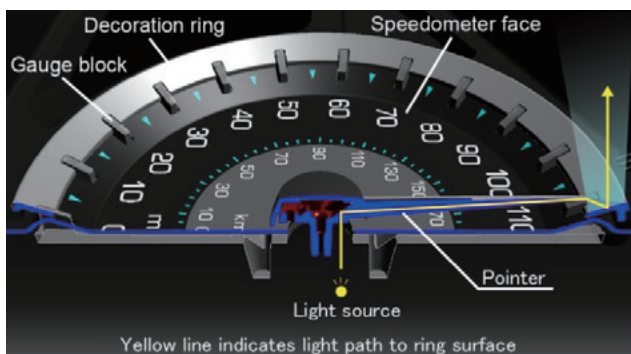


Fig. 22 Cross sectional view of rotating pointer type gauge and light paths of illumination

6.2. Attractiveness of Dynamic Movement

6.2.1. Tachometer pointer

The revving up and down of an engine can be felt through the motorcycle's exhaust sound, vibration and pulsation, as well as seen with the movement on the tachometer. The joy of riding a motorcycle is often likened to having a conversation: riders often feel that there is an exchange between them, and that the changes in engine revolution is more than a referential number on the display. This means that the movement of the tachometer also contributes to making the experience of riding an enjoyable one.

If a dynamic pointer movement is required, for example, the pointer's operational angle can be increased so even a small amount of RPM change can show a greater pointer movement. However, a large area cannot be used for tachometer alone given the limited space given for information display. In an early prototype had the gauges placed at an even interval.

But the engine speed would remain in the 1000-4000 rpm zone if using a big-engined Gold Wing on normal speed roads, making the pointer to appear to be just vibrating near the top of the tachometer. This led to the conclusion that this arrangement would fall short of achieving of "having a conversation with the vehicle."

For this reason, the gauges were marked at uneven intervals and the operation angle of the pointer expanded for frequently used engine speed range. However, having only one area with uneven gauges would make it appear awkward. The solution to this was to apply a gradual change to the gauges on both sides of the said area for a smooth, seamless integration. With this, the angle between 1000 rpm and 4000 rpm were expanded from 45 to 57 degrees to make visible greater movements of the pointer in the normal usage area, which would make the riding experience even more enjoyable (Fig. 24). The pointer movement and the metallic texture, combined with the light reflection design, we were able to create a beautiful, attractive instrument cluster design worthy of a world-class touring motorcycle.

6.2.2. Animations on instrument cluster and full-color TFT-LCD screen

The full-color TFT-LCD screen in the center displays an opening animation when the switch is on and an ending

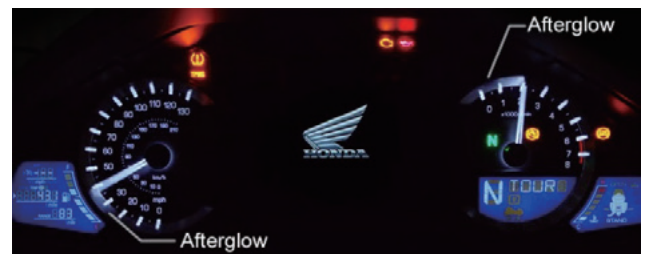


Fig. 23 Afterglow effects following rotation of pointer

animation when the main switch is off. When turning the system on, the black-and-white LCD screens turn on and the pointers on the gauge units move to the end before coming back. A chrome-plated style of the Honda wing emblem then appears on the center of the full-color TFT-LCD screen, then the screen transitions to the function menu. This process has two purposes: making system startup more entertaining, and checking of the instrument cluster display working properly. In the main switch turn-off animation, the Gold Wing emblem appears and fades out. This was done particularly to give the rider a lingering feel after enjoying riding the Gold Wing (Fig. 25).

This application of an “opening” effect using the analog gauge units, black-and-white LCD screens and full-color TFT-LCD screen was Honda’s first.

6.3. Center Panel

In the center of the center panel is an interface dial finished with aluminum-like semi-gloss paint, with matte black buttons behind it. The dial and the panel around the buttons is finished with piano black. The panels covering the sides are painted in the same color as the frame around the instrument cluster of silver, with a lower level of gloss and brightness. The clear contrast between the low-gloss

side covers and interface dial and the glossy black panel helped create a new feel of luxury.

The redesigned Gold Wing emblem was also applied towards the rear end of the black center panel. This was done with the aim of further accentuating the brilliance of the emblem, one of the very few chrome-plated parts used, by placing it in a black background. The entire center panel is shown in Fig. 26.

6.4. Integration with the Overall Styling Design

Much of the attractive appearance of the individual parts described so far have been based on their surface treatments.

Both the instrument cluster and center panel are, as is with the engine, mechanical parts that also impact the vehicle’s appearance. A measure was taken to create a well-integrated look by referring to the engine color management concept⁽¹⁰⁾.

The form, too, was designed to maximize the attractiveness of the appearance of the entire vehicle. First, the outline of the instrument cluster was made to match the silhouette of the upper vehicle body from the rider’s view. The lower center of the instrument cluster and the front end of the center panel must have a way to avoid being contacted by the handlebars when the

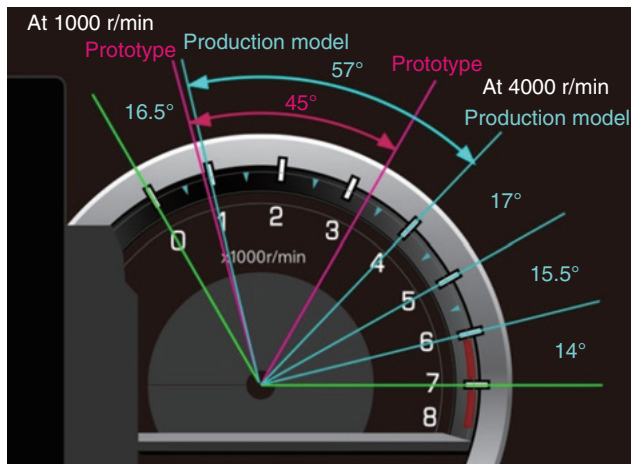


Fig. 24 Comparison of angular intervals of gauge among divisions for tachometer



Fig. 26 Center panel



Fig. 25 Animations displayed at turning on and turning off of vehicle system

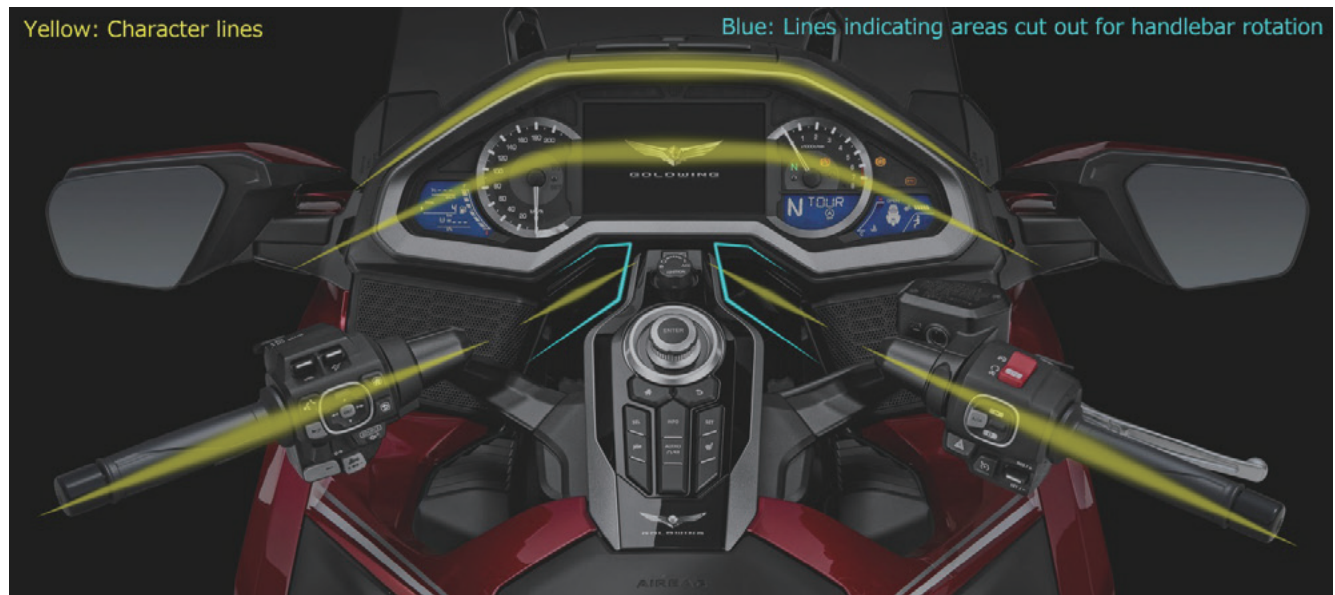


Fig. 27 Designing key lines of instrument cluster and control switches

vehicle is being steered. Two trapezoidal cut outs were made in the lower center of the instrument cluster to be in line with the directions of handlebar movements. The front end of the center panel is tapered so the direction of connection with the wide end is in the same direction. As such, much attention was given to even the directions to which the outlines of individual components point in order to be seamlessly integrated with the overall styling design (Fig. 27).

7. Summary

The instrument cluster and rider operation parts in this development of the Gold Wing were designed with much consideration to their beauty in appearance, since they are also key components in the styling of the entire vehicle. The gauge units mechanically built in a traditional “analog” style give joy to the riders, and the well-thought through information display method and operation method cleverly solved the challenge posed by a great increase in the number of operations. We believe that these greatly contributed to the IVI enhancement worthy of a premier motorcycle model, as well as executing in full the project objective of changing direction to a light-weight, high-density vehicle package.

We also believe that the insight gained on HMI through this development will greatly contribute to the future motorcycle developments at Honda.

References

- (1) Kaida, K., Nakanishi, Y.: Development of 2018 Gold Wing, Honda R&D Technical Review, Vol.30, No.2,

p.1-15

- (2) <http://hondanews.com/channels/gold-wing/releases/honda-gold-wing-timeline-2>
- (3) Kusuda, A., Yokokawa, Y., Nakai, K.: Development of Ride Modes for Various Conditions for 2018 Gold Wing, Honda R&D Technical Review, Vol.30, No.2, p.38-46
- (4) <https://www.apple.com/jp/ios/carplay/>
- (5) Android Auto https://www.android.com/intl/ja_jp/auto/
- (6) <https://www.monotype.com/fonts/fonts-for-cars/>
- (7) Miller, A G.: The Magical Number Seven, Plus or Minus Two Some Limits on Our Capacity for Processing Information, Psychological Review, Vol. 101, No. 2, p.343-352, (1956)
- (8) <https://www.rasin.co.jp/blog/sihh/>
- (9) <https://www.rasin.co.jp/blog/baselworld/>
- (10) Katsuragawa, M.: Styling Design of 2018 Gold Wing - Coloring -, Honda R&D Technical Review, Vol.31, No.1, p.18-27

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Taishi INOUE

Application of Modular Design Technology to Consumer-use General Purpose Engine

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ABSTRACT

A series of engines for the consumer market has been developed that responds to a range of demands by maximizing the sharing of constituent parts, which also realizes a reduction in the total number of parts employed across the series, a reduced cost, and a high level of environmental performance. Three different displacements engines have been realized in the series by changing only the cylinder modules, consisting of uniblock cylinders of differing bore diameters and pistons. Parts, for which a range of specifications are demanded across the series of different displacements depending on the engine framework, have been shared as a crankcase module, reducing the total number of parts employed in the engine block for engine variations by 43% against the previous models. Simplification of the crankcase shape and the realization of an increased cooling performance reduced the mass of materials in the engine block by 25% against the previous engines. These innovations have reduced the cost of the engines by approximately 15% against the previous models. The combustion chamber shape was modified and a long stroke was employed in order to boost the environmental performance of the low-displacement model, and each of the three different displacement models clears environmental regulations in its countries of sale. In addition, crankcase configuration modification and valve mechanism position have made it possible to realize a greater ease of maintenance than the previous models.

1. Introduction

Since Honda's consumer-use GCV engine series⁽¹⁾ went on the market in 1997, they have been fitted in home-use products such as home lawnmowers and high-pressure washers, mainly in the North American and European markets. A cumulative total of 36 million units have been sold. The GCV engine series is a series of three engines with differing displacements, 135, 160, and 190 cc. In addition to their use in Honda lawnmowers, approximately 80% of the unit volume produced is sold as OEM engines to the manufacturers of competitors' lawnmowers, high-pressure washers, and other complete machines. As OEM engines, it is necessary to provide a series lineup and specification variations in order to respond to series development and product requests from complete machine manufacturers.

At the same time, because price competition in the area of consumer-use products is fierce and low cost is also a strong demand on the engines fitted in products, low-cost engine frameworks that offer both engine performance and specification variations are a source of market competitiveness.

The recent tightening of environmental regulations has made it challenging to maintain a balance between exhaust emission reductions and low cost. Responding to this necessity was not easy, especially in the case of the low-displacement GCV135, and so this unit has been discontinued.

Honda has, therefore, developed a new GCV engine series that realizes a high-level balance between environmental performance, reliability, specification variations, and cost as Honda engines. Figure 1 shows the external appearance of the new GCV engines.

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Fig. 1 External view of new GCV engine

2. Development Goals

The development sought to realize a consumer engine possessing a high level of environmental performance, which responded to a range of demands via the engine framework renewal while also maximizing the potential for interchange of constituent parts, and which also reduced the total number of parts and realized low cost. The following main development targets were set:

- (1) A reduction of at least 10% in cost
 - A reduction of at least 30% in the total number of parts used across the series in the engine block section
 - A reduction of at least 20% in the aluminum mass in the engine block section
- (2) Clearance of a range of national environmental regulations
- (3) Realization of equivalent or greater ease of maintenance to the previous models

3. Basic Engine Framework

For the complete machine manufacturers who make up the customers to which we supply engines, variations in the power of consumer work equipment, such as lawnmowers and high-pressure washers are necessary in order to respond to differences in workload. Three displacements, 145, 170, and 200 cc, were set for the new GCV engine series. Table 1 compares the main specifications of the series.

Details of each component will be provided below:

3.1. Engine Block Structure

3.1.1. Cylinder module

The new GCV engines employ a uniblock cylinder that integrates the cylinder head and the cylinder. Together

with the piston system parts, this forms a cylinder module. As Fig. 2 shows, the series of three engines of differing displacements has been formulated using three cylinder modules in which only the bore diameter differs.

3.1.2. Crankcase module

The constituent parts of the crankcase module, the crankcase and the crankshaft, are the parts that connect with the mounting frame, and are, therefore, parts for which numerous variations in specifications are set in response to requests for different frames. Due to this, if dedicated parts were used for engines of each displacement in the series, the total number of parts used in the series would increase.

As Fig. 2 shows, in the new GCV engine series, the crankcase module is shared by each displacement. This has made it possible to reduce the total number of parts used in the series. Figure 3 shows the configuration of the engine blocks for the entire new and old series.

Demand for engine start via a starter motor rather than a recoil starter has recently increased. The previous models employed a configuration in which the starter motor was attached to the cylinder. In the new series, the starter motor is attached to a crankcase cover with a simple shape. This has made it possible to produce variations at low cost.

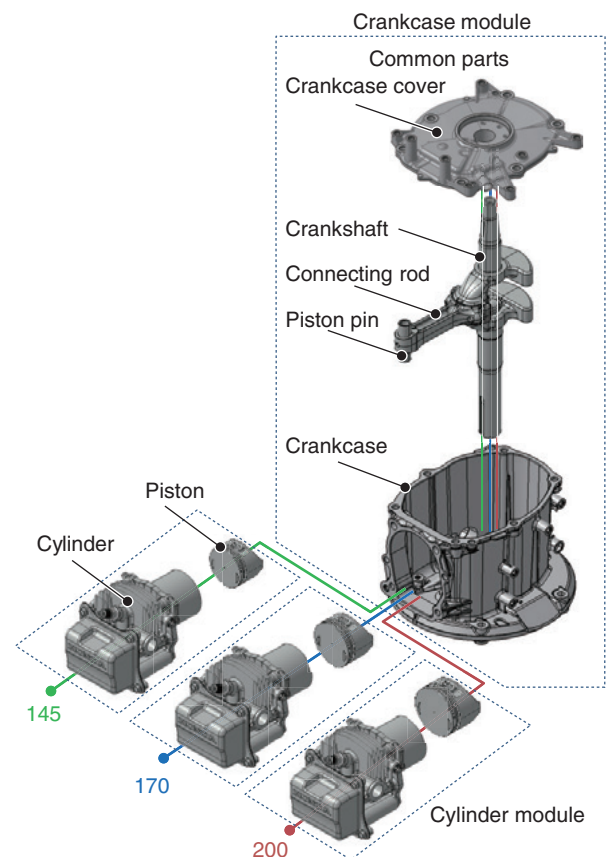


Fig. 2 Structure of engine block module

Table 1 Engine specifications

	Previous model			Developed model		
Model	GCV135 (Discontinued)	GCV160	GCV190	GCV145	GCV170	GCV200
Cam type	OHC			OHV		
Displacement (cm ³)	135	161	187	145	166	201
Bore x Stroke (mm)	64 x 42	64 x 50	69 x 50	56 x 59	60 x 59	66 x 59
Compression ratio	8.5			7.7	8.0	
Maximum power (kW/rpm)*	2.6/3600	3.3/3600	3.8/3600	3.1/3600	3.6/3600	4.2/3600
Maximum torque (N·m/rpm)*	6.9/2500	9.4/2500	11.3/2500	9.1/2500	11.1/2500	12.7/2500
Dry mass (kg)	10.4	10.5	12.4	10.1		

*According to SAE J1349

The dividing line between the crankcase and the crankcase cover was set at the upper section of the crankcase. This made it possible to realize a simple crankcase shape that is connected in straight lines, reducing the amount of materials employed. Figure 4 shows the cross-sectional shape of the crankcase.

3.2. Auxiliary Equipment

In the engine block module made up of the cylinder module and the crankcase module, the intake system, exhaust system, fuel system, and cover have been modularized for their respective functions in order to satisfy the demands of complete equipment manufacturers

for specification and design variations. This has made it possible to create specifications suiting the country of sale. For example, in regions such as North America that have

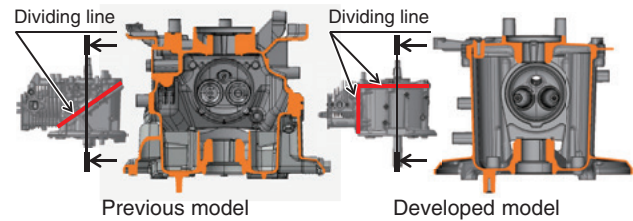


Fig. 4 Shape of crankcase

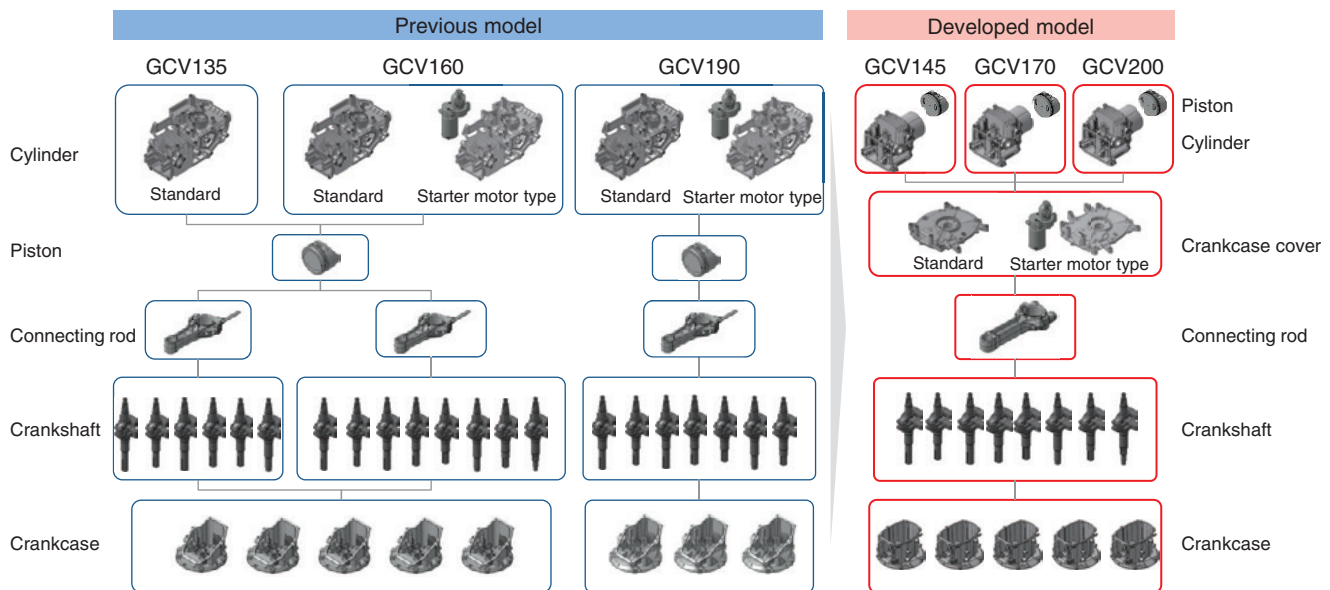


Fig. 3 Comparison of engine block structure

evaporative emissions regulations, it is possible to select a fuel system module made up of a fuel tank cap fitted with activated carbon, a multilayer blow-molded fuel tank, and a low-permeation fuel tube. Figure 5 shows the auxiliary system module configurations.

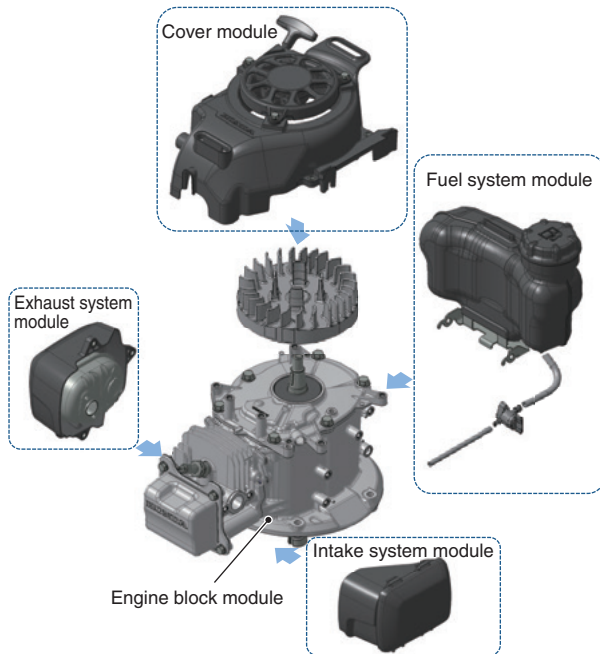


Fig. 5 Engine structure



Fig. 6 Top cover and fan cover of developed model

With regard to the covers, the use of a configuration in which a top cover is fitted over a fan cover that has the function of guiding cooling air has made it possible to respond to the demands of OEM for the use of brand colors and shapes. Figure 6 shows the top cover structure.

3.3. Cooling System

If cooling air can be used efficiently in forced air-cooled engines, it is possible to reduce the size of the cooling fins and this contributes to reducing the amount of materials employed. A numerical analysis was conducted in order to effectively concentrate cooling air on high-temperature sections. The results of this analysis are shown in Figs. 7 and 8. By this means, it was possible to reduce the cooling fin area of the new GCV200 by 28% against the previous GCV190 model, despite the fact that displacement had increased.

In addition, in order to fit the cooling fins in the most effective position for cooling, a configuration was adopted in which a flange bolt was used at the upper section of the cylinder and was fastened from the inside of the crankcase, balancing cooling performance and low cost. Figure 9 shows an engine cross-section of the fastening section.

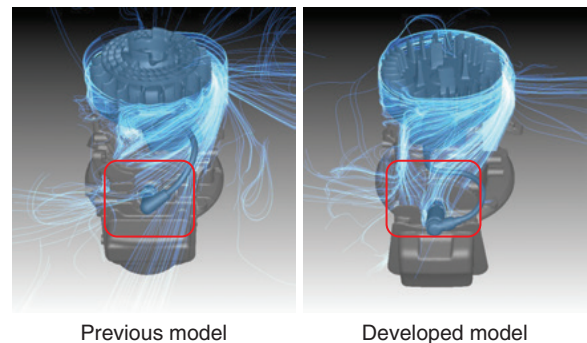


Fig. 7 Flow lines of cooling air

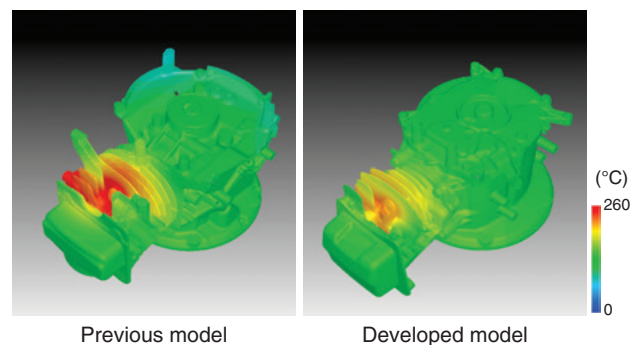


Fig. 8 Temperature of engine surface

3.4. Combustion Chamber

In developing the series towards lower-displacement engines, it was necessary to realize better environmental performance. In the new GCV engines, the combustion chambers are compact pent-roof chambers, and a longer stroke than the previous models has been employed. This has strengthened the in-cylinder flow motion and increased turbulence kinetic energy (TKE), making it possible to realize stable combustion with low emissions and high fuel efficiency. Figure 10 shows the results of the TKE calculations at the ignition timings employed in the engines.

In addition, the intake and exhaust valve diameters and spark plug position are the same for all the models of differing displacements, which has increased productivity by sharing machining.

3.5. Valve System

The valve mechanism was changed from the overhead camshaft (OHC) mechanism with a timing belt built into the engine employed in the previous models to an overhead valve (OHV) mechanism using push rods and valve lifters. In the previous models, the cooling air channels were positioned close to the combustion chamber, and the

timing belt chamber obstructed the flow of cooling air. The use of an OHV mechanism has made it possible to expand the cooling air channels, contributing to increased cooling performance. Figure 11 shows a comparison of the previous and developed shapes.

In addition, cost was reduced by situating the valve mechanism in the oil at the bottom of the cylinders, and adopting a plastic camshaft. Furthermore, the system configuration makes it possible to remove the crankshaft without disassembling the valve mechanism when the engine is undergoing maintenance. Figure 12 shows a cross-section of the engine.

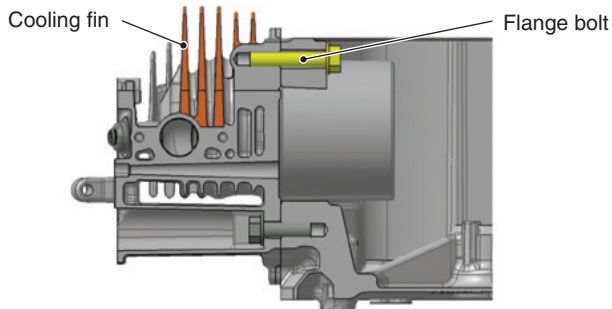


Fig. 9 Cross-section of cooling fins

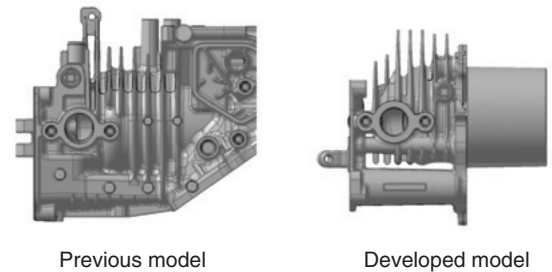


Fig. 11 Side view of cylinder

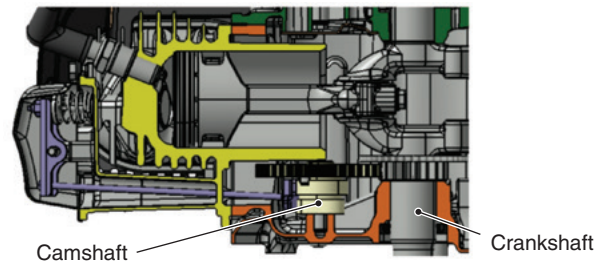


Fig. 12 Layout of camshaft

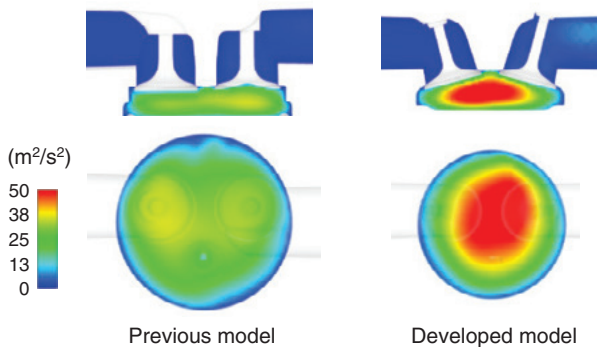


Fig. 10 Results of TKE calculations

4. Effect of Application of New Technologies

4.1. Cost

Reduction of the amount of materials used through the realization of mass savings and the sharing of parts across the differing-displacement engines in the series realized a reduction of approximately 15% in cost against the previous GCV series engines.

4.1.1. Total number of engine block parts across series

The sharing of parts has reduced the total number of

parts used across the series in the engine block section, which is shared by the variations, by 43% against the previous models.

4.1.2. Mass reduction

Minimization of the cooling fin sizes and modification of the engine block shape reduced the mass of the aluminum diecast parts that form the engine block in the new GCV200 by 25% against the previous GCV190. Figure 13 shows the aluminum mass employed in the engine blocks of the previous and developed engines.

As indicated by Table 1, despite the fact that displacement was increased against the previous model, the overall mass of the GCV200 was reduced by 18% against the previous GCV190.

4.2. Environmental Performance

An increase in combustion performance has reduced HC and NOx emissions in the low-displacement GCV145 by 22% against the previous GCV135. Figure 14 compares

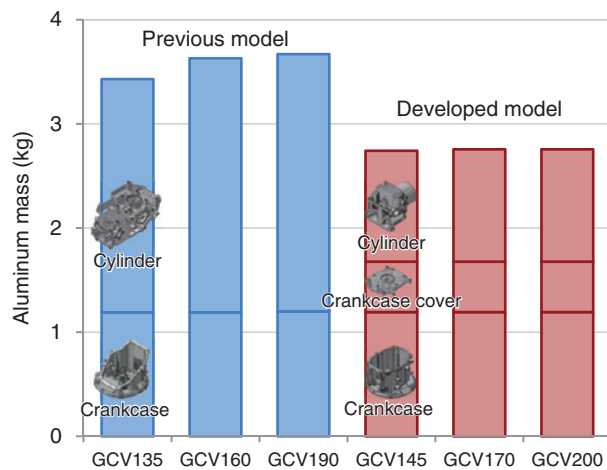


Fig. 13 Aluminum mass of engine block

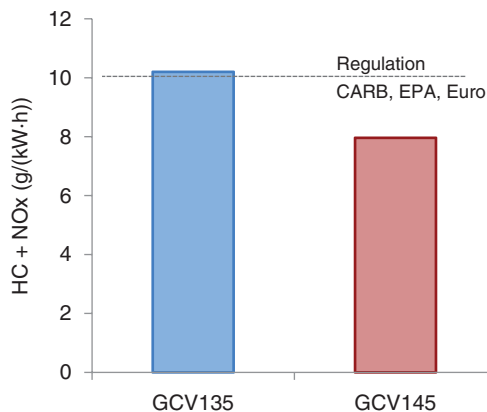


Fig. 14 Mode emissions

emissions in the developed and previous models. These results have made it possible for all models to clear national exhaust gas regulations in their countries of sale.

4.3. Ease of Maintenance

The crankcase configuration, discussed in 3.1.2., has made it possible to disassemble the engine while it is still attached to the frame during work to replace a lawnmower crankshaft. In addition, the use of an OHV mechanism does away with the need to disassemble and reassemble the valve mechanism for adjustment, reducing the man hours involved in maintenance by 60%. Figure 15 shows a diagram of engine parts during replacement of the crankshaft.



Fig. 15 Replacement of crankshaft

5. Conclusion

A series of general purpose engines for the consumer market has been developed that responds to a range of manufacturer demands by maximizing the sharing of constituent parts, and which realizes a reduction in the total number of parts employed across the series, a reduced cost, and a high level of environmental performance. In order to maximize the interchangeability of constituent parts in the design of the basic framework of the engines, the engine block and the auxiliary parts were modularized.

The use of different cylinder modules with the largest module, the crankcase module, as a shared part, made it possible to realize a lineup of three engines.

This development produced the following results:

- (1) The cost of the engines was reduced by approximately 15% against the previous models.
 - The total number of parts employed across the series in the engine block was reduced by 43%
 - The mass of the aluminum employed in the engine block was reduced by 25%
- (2) All the engines of three differing displacements satisfy the environmental regulations of their countries of sale.
- (3) The engines realized greater ease of maintenance than the previous models.

Reference

- (1) Suzuki, T., Hojo, H., Inoue, S., Moriyama, H., Fujinaga, Y.: Development of General Purpose OHC Engine with Built-in Timing Belt, HONDA R&D Technical Review, Vol. 10, p. 122-127

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Technology to Boost CVT Fuel Efficiency through Series Connection of Mechanical and Electric Oil Pumps

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ABSTRACT

A new oil pump system that combines mechanical and electric oil pumps has been developed in order to optimize oil pump work under low-load driving conditions. The use of the new system has reduced energy loss in the oil pumps, increased CVT transmission efficiency, and increased vehicle fuel efficiency.

A distinctive feature of the hydraulic circuit employed in the developed system is the positioning of the electric oil pump downstream from the mechanical oil pump, realizing a configuration in which it is possible to switch between the sources of hydraulic pressure supply to the pulleys and other parts. In addition, the configuration helps prevent any decline in driving performance in the event that the electric oil pump loses function.

The rated output of the electric oil pump employed was selected in order to realize benefits under a variety of driving conditions, and the control applied in the actual vehicle was optimized. As a result, an increase of 0.71% in CVT average transmission efficiency and an increase of 1.23% in fuel efficiency in an actual vehicle were realized.

1. Introduction

CVTs realize shift operation with lower fuel consumption than stepped transmissions⁽¹⁾. Due to this, their application has rapidly become more widespread as a means of boosting fuel efficiency, with a particular focus on city driving⁽²⁾. When the vehicle is cruising however, a driving mode in which shift is not necessary, there are occasions on which the transmission efficiency and fuel efficiency of CVTs are lower than those of stepped transmissions⁽³⁾. The realization of increased transmission efficiency in CVTs will, therefore, be an important factor in realizing further increases in fuel efficiency.

One reason that the transmission efficiency of CVTs falls behind that of stepped transmissions is the fact that oil pump work is high^{(4), (5)}. Figure 1 shows ratios of causes of energy loss in a CVT in JC08 mode. In a CVT, hydraulic pressure produces belt clamping force, and the unit transmits torque by means of friction between the metal pushing V-belt and the pulleys as a result of this belt clamping force. It is necessary for the maximum belt clamping force to be several times greater than the maximum pressure employed in a stepped transmission.

Figure 2 shows a schematic diagram of the allocation of oil pump work in a CVT. The required oil pump work in a CVT can be broadly divided into two types. One of these is the work necessary for the belt clamping force mentioned above and for the engagement of the clutch. High pressure, P_H , and a necessary flow rate, Q_A , including shift, are demanded for torque transmission. The other is the work necessary for supply of oil to the torque converter and for lubrication. In this case, hydraulic pressure P_L is demanded. As this indicates, it is necessary for the CVT oil pump to satisfy two demands. As it is necessary for the oil pump to supply high-pressure working oil to generate belt clamping pressure at high flow rates that maintain a state of lubrication, oil pump work during operation occurs within the maximum scope that incorporates these two demands. The region within this maximum scope in which there is no effective oil pump work represents energy loss.

Against this background, two-rotor-type trochoid pumps^{(4), (5)} and variable-discharge oil pumps⁽⁶⁾ have been put into practical application in order to reduce oil pump energy loss. However, these units switch the discharge rate in response to the CVT operating state in stages, and they therefore do not realize the optimization of oil pump work.

¹ Automobile Center

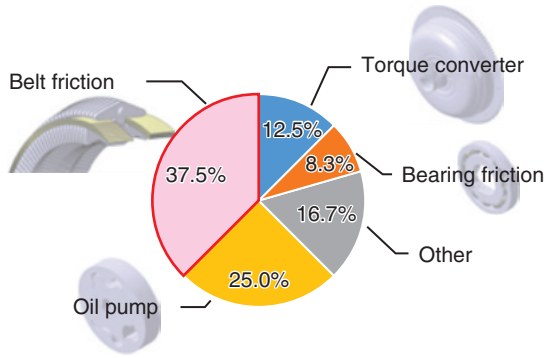


Fig. 1 Breakdown of power consumed in CVT (during operation in JC08 mode)

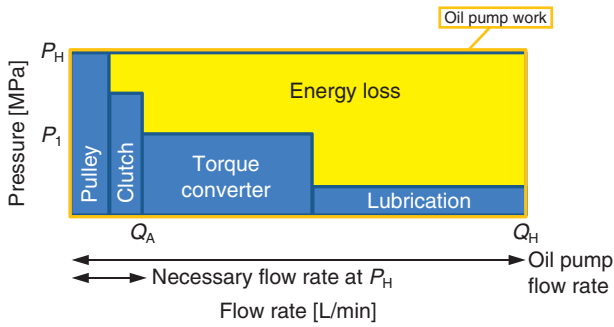


Fig. 2 Oil pump work in CVT

This paper will discuss a technology for the realization of increased fuel efficiency using an oil pump system developed in the quest for a higher level of oil pump work optimization.

2. Development Aims

CVTs realize stepless shift by inducing a stroke in the pulley axial direction and changing the winding diameter of the belt. During shift, changes in the pressure of the working oil and the pulley stroke occur simultaneously. When a pulley stroke is generated, the capacity of the pulley oil chambers fluctuates, and the necessary flow rate for the working oil also changes. As a result, the absence of any reduction in the theoretical discharge rate of the oil pump is a precondition for the ability to respond to sudden shifts.

When the oil pump is driven by the engine crankshaft, the discharge rate of the mechanical oil pump (termed “MOP” below) is proportional to the engine speed⁽⁷⁾. Even at the same discharge pressure, energy loss will be greater at higher engine speeds. In order to optimize oil pump work, it would be necessary to use an oil pump drive method that is independent of engine speed. The research under discussion selected an electric oil pump (termed “EOP” below).

A design concept encompassing the selection of EOP specifications and EOP operation in order to minimize oil pump work was developed. The necessary consideration in applying an EOP was to realize a hydraulic circuit that would help prevent any interruption to hydraulic pressure supply even when the EOP lost function.

Mainly, the following three items were set as the development aims:

- (1) The development of a hydraulic circuit that would help ensure that driving performance was not impeded even if the EOP lost function.
- (2) Selection of a rated output for the motor used in the EOP that would minimize oil pump work.
- (3) Optimal control of EOP in response to driving conditions.

3. Basic Structure

Figure 3 shows a schematic diagram of the operation of the EOP in relation to driving conditions. In sections A and B, when the system’s maximum pressure, P_H , is low, and the demanded flow rate, Q_A , is also low, the two oil pumps are operated together in order to reduce energy loss. In section C, when both the maximum pressure and the demanded flow rate are high, the EOP is stopped and hydraulic pressure is supplied by the MOP alone.

3.1. Hydraulic pressure Circuit with Oil Pumps in Series

Figure 4 shows a schematic diagram of the hydraulic pressure circuit proposed in this research. The two oil pumps are positioned in series, and the source of supply of high hydraulic pressure is switched by the opening and closing of a bypass valve. The hydraulic channel pressure between the EOP intake side and the MOP is designated P_1 , and the hydraulic pressure in the EOP discharge section is designated P_H . In addition, the system is provided with a P_1 sensor to measure pressure P_1 in order to realize hydraulic pressure feedback for EOP control.

Due to the series connection of the oil pumps, the electric

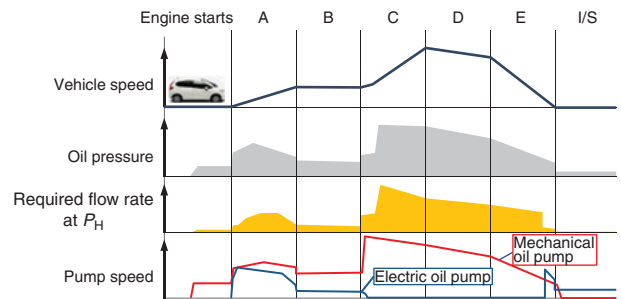


Fig. 3 Schematic image of electric oil pump operation

OP receives discharge pressure from the MOP, and outputs further increased pressure. When the bypass valve is closed, the MOP is responsible for the low-pressure sections, and the EOP is responsible for differential pressure from the low-pressure to the high-pressure sections. Figure 5 shows a breakdown of the work for the oil pump system at this time. Figure 5 indicates that the MOP energy loss area, shown in Fig. 2, has decreased in size. When high pressure is demanded for belt clamping or the driving scenario necessitates sudden shift, it is not possible to maintain the allocation shown in Fig. 5, due to the EOP's limited discharge performance. Therefore, in driving scenarios in which the MOP's discharge performance exceeds that of the EOP, the bypass valve is opened and the MOP is exclusively responsible for hydraulic pressure supply. The system returns to the allocation of the oil pump work shown in Fig. 2.

Due to the series configuration, the EOP is not able to increase the discharge flow rate of the MOP. Since the theoretical discharge flow rate of the MOP remains unchanged from a conventional hydraulic circuit, there is no effect of friction reduction in the MOP unit.

Conventional EOP employed in idling stop systems (termed "I/S" below) do not realize the basic hydraulic characteristic or durability that were necessary

specifications for the development under discussion. An EOP was, therefore, newly developed.

It was necessary to consider the fact that EOP may become unable to operate due to malfunctioning of the power source, the harness employed for communications, or constituent electric parts such as the motor control circuits. Since the developed system adopts a configuration in which the bypass valve is switched by front-rear differential pressure, steady-state operation is possible even in the event of electric malfunctions. Therefore, even if the EOP fails to function, normal driving will not be impeded because the system will switch to supply from the MOP in isolation.

3.2. Selection of EOP Output Specifications

The following equation expresses the oil pump work when hydraulic pressure is supplied by a conventional MOP in isolation, as shown in Fig. 2:

$$W^* = \frac{P_H Q_H}{\eta_p} \quad (1)$$

The following equation expresses the oil pump work when hydraulic pressure is supplied by the EOP and the MOP, as shown in Fig. 5:

$$W = \frac{P_1 Q_H}{\eta_p} + \frac{(P_H - P_1) Q_A}{\eta_{sys} \eta_a} \quad (2)$$

The first term on the right-hand side is MOP work; η_p is pump efficiency. The second term is EOP work; η_{sys} is the system efficiency of the EOP system, and η_a is generation efficiency. Since the power necessary for driving the EOP is supplied by the engine, it is important to consider its generation efficiency. Given this, the degree of reduction in energy loss, ΔW , is expressed by the following equation:

$$\begin{aligned} \Delta W &= W^* - W \\ &= (P_H - P_1) \left(\frac{Q_H}{\eta_p} - \frac{Q_A}{\eta_{sys} \eta_a} \right) \end{aligned} \quad (3)$$

As Eq. (3) indicates, because $P_H - P_1 > 0$, when $Q_A / \eta_{sys} \eta_a \geq Q_H / \eta_p$, ΔW is negative. In other words, system efficiency η_{sys} has an important meaning in terms of making maximal use of the EOP.

System efficiency η_{sys} is given by the following equation:

$$\eta_{sys} = (P_H - P_1) * Q / (V * I) \quad (4)$$

Q: EOP discharge flow rate

V: EOP input voltage

I: Current consumed by EOP

As system efficiency η_{sys} is mainly dependent on the rated output of the EOP motor, the selection of motor output is an important factor.

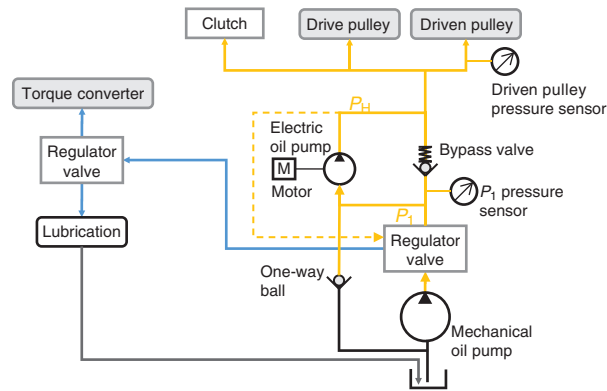


Fig. 4 Hydraulic circuit

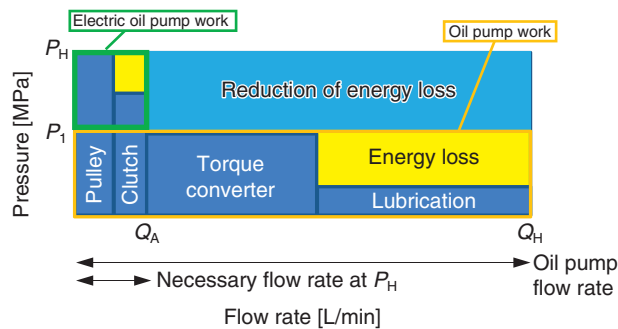


Fig. 5 Developed oil pump system

Figure 6 shows an example of the system efficiency of the EOP employed in the developed system. The horizontal axis shows the EOP discharge flow rate, and the vertical axis shows discharge pressure. The more the area is shaded red, the higher the system efficiency. The broken lines show equivalent output diagrams that multiply the horizontal and vertical axes. As Fig. 6 indicates, system efficiency is higher on the high-output side.

The higher the rated output of the motor employed in the EOP, the higher the pressure and flow rate that the state shown in Fig. 5 can be maintained. This means that energy loss in the MOP can be reduced across a wider range of driving conditions. However, as the motor's rated output increases, its mass increases and it becomes larger, with the result that it becomes unsuitable for vehicle fitting⁽⁸⁾. It was, therefore, desirable to select the lowest motor rated output for the wanted effect.

Figure 7 compares the degree of reduction of energy loss at vehicle takeoff for different EOP motor rated outputs. As this graph shows, motor rated output and the effect of motor rated output are non-linear characteristics.

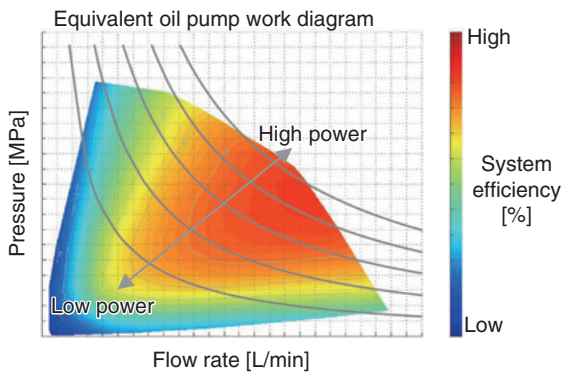


Fig. 6 System efficiency of electric oil pump

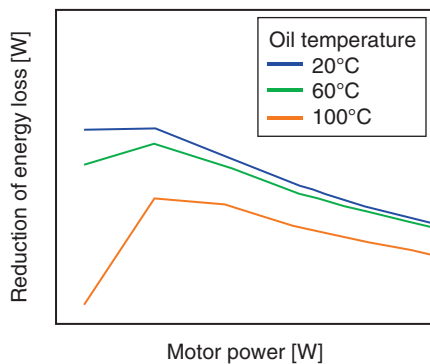


Fig. 7 Comparison of reduction of energy loss at vehicle takeoff for differing electric oil pump rated outputs

Figure 8 shows the results of a comparison, in desktop studies, of the degree of reduction of energy loss under conditions that combine a variety of driving states. At the highest motor rated output of 300 W, energy loss in the MOP was less than 100 W. This means that CVT efficiency is highest at a motor rated output of 300 W. However, because the power loss due to the operation of the EOP is highest at 300 W, the motor rated output that is preferable from the perspectives of low energy loss and good fuel efficiency as considered for the vehicle as a whole is 100 W.

Figure 9 shows the results of a comparison in desktop studies of the degree of energy loss reduction for different motor outputs rated in representative fuel consumption modes. These results show that for each of the fuel economy modes, which involve different driving patterns, a rated output of 100 W offers an adequate effect of energy loss reduction.

3.3. EOP Control

The EOP employed in the developed system receives command values for motor speed from a host CPU via CAN communications. The EOP applies control in accordance with the received motor speed command values.

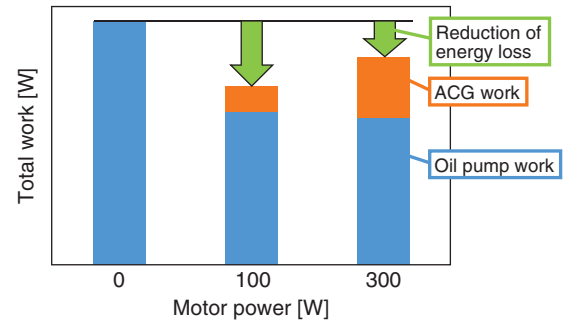


Fig. 8 Effect of reduction of energy loss

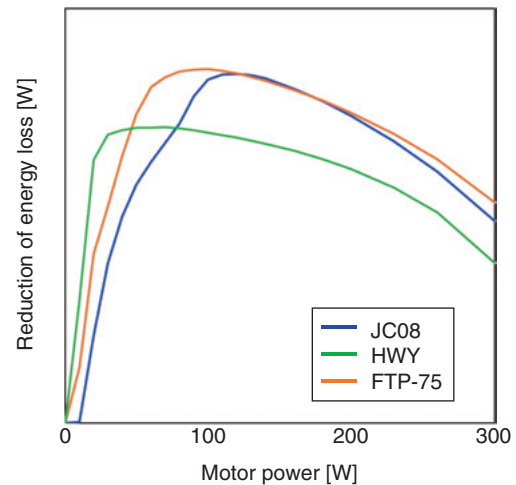


Fig. 9 Fuel economy improvement

Figure 10 shows the state transition diagram that represents the control concept for the developed system. Five states are defined; Driving and I/S are the basic states for EOP operation. In cases in which the system state transitions from Driving to a state in which it is not possible to reduce energy loss in the MOP, or the necessary hydraulic pressure or flow rate exceeds the performance of the EOP (state changes that would cause operational or other issues in different systems), the system changes state by reverting to Standby. The system does not return from Driving to Standby directly, but passes through Stop transition. This continues until the system has transitioned into a state in which the bypass valve is open.

Figure 11 shows the flow of calculation of the EOP motor speed command value in the Driving state. This control flow is positioned as subordinate to the control flow in which the CVT pulley clamping force is calculated, and employs the physical values for hydraulic pressure and the discharge flow rate necessary for shift determined by the governing flow as variables. It can be assumed that due to variations in the parts making up the hydraulic pressure

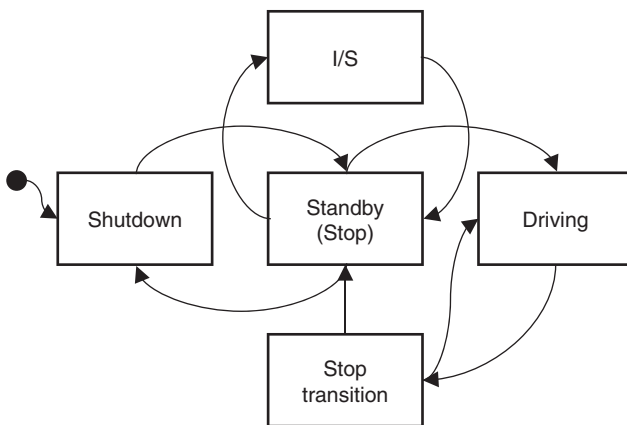


Fig. 10 State transition diagram

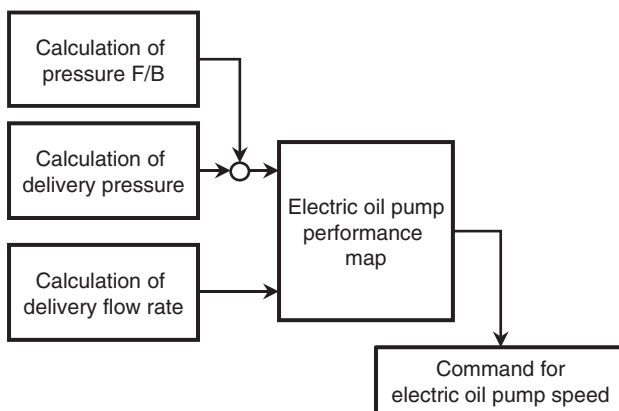


Fig. 11 Calculation of command value

control system, there will be occasions on which it is not possible to reduce pressure to the target MOP discharge pressure based on the command motor speed. In order to make it possible to also respond to these situations, the pressure difference between the P_1 pressure sensor value and the target pressure is employed as feedback pressure. The necessary variables are compared with the EOP performance map, and the motor speed command value is calculated.

In the Stop transition state, in response to the driving status, the EOP motor speed command value that will minimize fluctuations in hydraulic pressure when the bypass valve is open is calculated.

Figure 12 shows changes in hydraulic pressure when the EOP is in operation in the developed system. At point A, the EOP motor speed is increasing, and the discharge flow rate is also increasing. Since the bypass valve is open at this time, there is no change in pressure P_1 , the MOP discharge pressure. When motor speed reaches point B, the flow passing the bypass valve is also entirely passing the EOP side. As there is no front-rear pressure differential, the bypass valve is closed by spring force, and pressure P_1 begins to decline. When point C is reached, pressure P_1 becomes the same as the torque converter circuit pressure, and does not decline any further. In this driving state, the optimal value for the EOP motor speed is the speed at point C, at which the MOP discharge pressure ceases to decline, and any motor speed in excess of this would be undesirable from the perspective of increasing energy loss as generation proceeds.

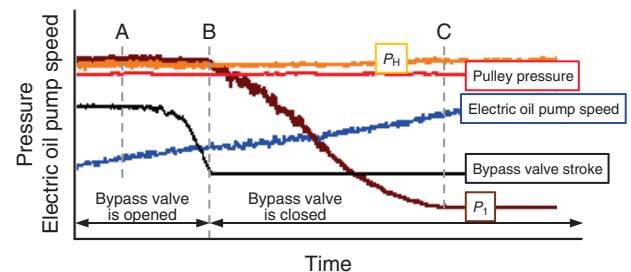


Fig. 12 Reduction in P_1 pressure

4. Effect in Boosting Performance

This section will discuss the performance increases against a conventional system obtained in an actual vehicle fitted with the developed system.

4.1. Transmission Efficiency

Figure 13 shows transmission efficiency by shift ratio during steady-state driving. Transmission efficiency is increased when the EOP is operating, and the average rate

of increase in efficiency under driving conditions within the normal operating range is 0.71%.

4.2. Current Consumption during I/S

The EOP employed for I/S in a conventional vehicle maintains a fixed motor speed using ON-OFF control; the system is configured to exhaust the portion that exceeds the flow rate that is necessary to maintain clutch pressure. While the necessary flow rate displays a positive correlation with oil temperature⁽⁹⁾, because the discharge flow rate of the EOP does not change, the current consumed by the motor has no relation to oil temperature, and is basically constant.

In the developed system, oil pump work has been optimized by employing motor speed commands in accordance with the necessary flow rate for each oil temperature, reducing current consumption. Figure 14 shows discharge flow rates for the EOP during I/S. Despite the fact that the output of the EOP motor employed in the developed system was high compared to existing motors for I/S, optimization of control has reduced the current consumed, contributing to the realization of increased fuel efficiency in an actual vehicle.

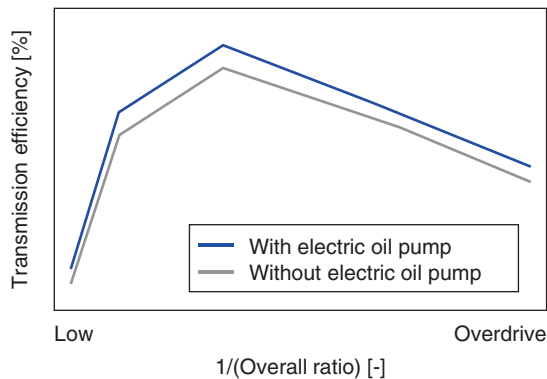


Fig. 13 Transmission efficiency

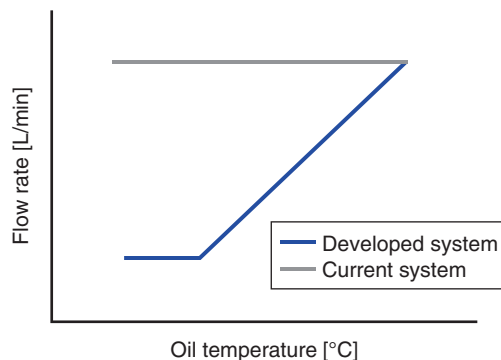


Fig. 14 Flow rate during Idling stop

4.3. Effect in Increasing Fuel Efficiency

Figure 15 shows the increase in fuel efficiency in NEDC mode in a vehicle fitted with the developed system. It was verified that vehicle total fuel efficiency was increased by 1.23% even when the increase in generation work for the operation of the EOP is considered.

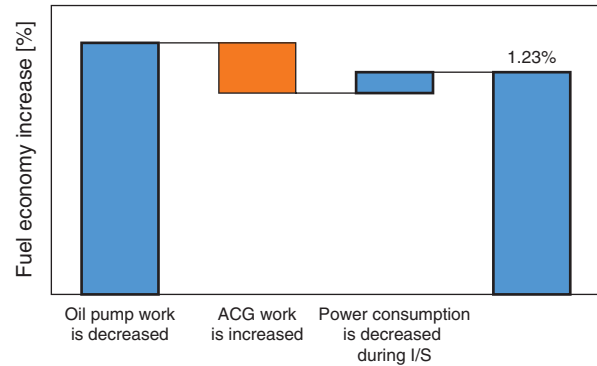


Fig. 15 Fuel efficiency increase (NEDC mode)

5. Conclusion

An oil pump system has been developed as a technology to boost fuel efficiency in CVTs for small vehicles. This development produced the following results:

- (1) A hydraulic circuit structure was developed that helps to ensure that driving performance is not impeded even if the EOP loses function.
- (2) The optimum motor rated output was selected, helping realize an average increase of 0.71% in transmission efficiency and an increase of 1.23% in fuel efficiency in NEDC mode.
- (3) An EOP control was developed that feeds back the MOP discharge pressure and is able to vary the EOP motor speed from suspension to optimum speed irrespective of variations in the constituent parts of the hydraulic pressure control system.

References

- (1) Van der Sluis, F., Van der Noll, E., De Leeuw, H.: The Success of the Pushbelt CVT -Status and new Development-, 2012 JSAE Annual Congress Proceedings, No. 26-12, p. 9-16, (2012)
- (2) Van der Sluis, F., Yildiz, S., Brandsma, A., Veltmans, P., Markus, K.: The CVT Pushbelt Reinvented for Future Compact and Efficient Powertrains, 2017 JSAE Congress (Spring) Proceedings, p. 245-252, (2017)
- (3) Fornari, R. V., Suzuki, A. R., Yamada, D. S.: Prospects

- for Future Non-Manual Transmissions Developments for Vehicle Applications in Brazil, SAE Technical Paper, 2015-36-0348, doi:10.4271/ 2015-36-0348, (2015)
- (4) Ohgata, Y., Ninomiya K., Kumasawa, S., Tamura, N.: Development of CVT Which Improved Fuel Economy, Belt Noise and drivability, Transactions of the Society of Automotive Engineers of Japan, Vol. 44, No. 5, p. 1211-1214, (2013) (in Japanese)
 - (5) Nirasawa, H., Uneyama, T.: New CVT for 2018 Model Year N-BOX, Honda R&D Technical Review, Vol.29, No.2, p.29-35
 - (6) Inukai, K., Shibahara, A., Murakami, Y., Uchino, T., Kojima, M., Nishiyama, K.: Development of New CVT for Midsize Vehicle, Honda R&D Technical Review, Vol.24, No.2, p.47-53
 - (7) Nishida, Y., Ono, H., Terashima, H., Tokoda, F., Numami, K.: Development of Continuously Variable Discharge Oil Pump, 2017 JSAE Congress (Spring) Proceedings, p. 1942-1945, (2017) (in Japanese)
 - (8) Sakata, T., Yokoi, Y.: Development of Electric Pump Unit for Drive Train, KOYO Engineering Journal No. 159, p. 74-77, (2001) (in Japanese)
 - (9) Miyajima, J., Watanabe, T., Kato, Y.: Development of a Thermosensitive Variable Oil Pump, 2017 JSAE Congress (Spring) Proceedings, p. 1938-1941, (2017) (in Japanese)

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Technology for Estimation of Fuel Cell Stack Temperature Using Transient Heat Transfer Simulation in Cell Stacking Direction

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ABSTRACT

A simulation of transient heat transfer in the cell stacking direction was developed that makes it possible to estimate the temperature of the MEA at different arbitrary stacking positions within the stack, including the case components and the end cells, at a high level of accuracy under all operating conditions that can be assumed for the vehicle environment. With regard to factors that have a considerable effect on the temperature of the MEA but necessitate a high volume of calculations, the introduction of a correction term based on fundamental data measured in a stack while generating power simplified the calculations. In addition, by contrast with conventional simulation technology, factors that have little effect on the temperature of the MEA were identified and omitted. These initiatives have made it possible to conduct coupled calculations for the constituent components of the stack without any decline in the accuracy of temperature estimations. It was verified that the simulation of transient heat transfer in the cell stacking direction was able to realize the target figure for temperature estimation accuracy of within $\pm 5\%$ against actual measurements for both steady-state stack generation conditions and stack warm-up mode, including transient states.

1. Introduction

Honda is advancing the development of the fuel cell vehicle (FCV) as a technology able to respond to the twin issues of CO₂-driven global warming and the securing of energy with the depletion of fossil fuel resources⁽¹⁾.

Figure 1 shows the structure of the fuel cell stack fitted in the 2016 model year CLARITY FUEL CELL (termed “the stack” below). The stack is composed of layered fuel cells (termed “cells” below) and stack case components (termed “case components” below). The membrane electrode assembly (MEA) inside the cells generates heat as a result of factors, including energy loss in cell reactions, but the flow of coolant between the cells cools the system and controls it to the correct temperature.

The cells in contact with the case components (termed “the end cells” below) are susceptible to the effect of heat released from the cells to the case components and atmosphere, as shown in Fig. 2. Figure 3 shows the temperature distribution of the cells positioned in the center of the cell stacking direction (termed “the center cells” below) and the end cells when the stack is started with the case components at low temperature.

The temperature of the end cells is lower than that of the center cells. When the MEA temperature differential between the center cells and the end cells (ΔT_{MEA}) becomes high, the volume of condensate in the end cells becomes higher than it is in the center cells. This

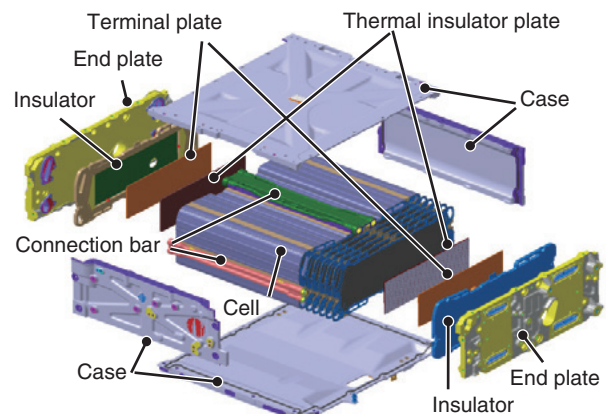


Fig. 1 Structure of fuel cell stack

¹ Automobile Center

phenomenon is one of the causes of instability in cell voltage behavior⁽²⁾.

In order to lower ΔT_{MEA} during stack warm-up, it is necessary to both increase the insulation performance of the case components and supply the appropriate level of heat to the cells by means of heat exchange with the coolant and heat generation by the current-carrying components. As this implies, not only cooling of each cell by the coolant, but also control of the temperature of the end cells to the appropriate level under all operating conditions, are important factors in the thermal design of the stack.

Honda has worked to develop a simulation technology able to estimate the internal environment of the cells during power generation by making original modifications to market software^{(3), (4)}. However, because the purpose of this simulation technology, like standard fuel cell simulation technologies⁽⁵⁾⁻⁽⁸⁾, is to estimate the in-plane environment of the center cells (parameters including pressure, temperature, and MEA water content), it presented the issue of necessitating a high volume of calculations for parameters other than temperature.

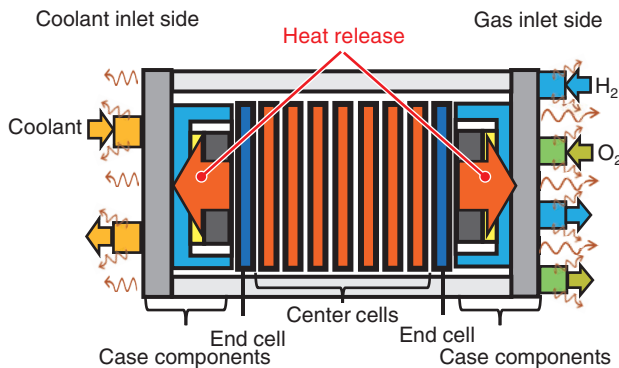


Fig. 2 Schematic image of heat release from case components

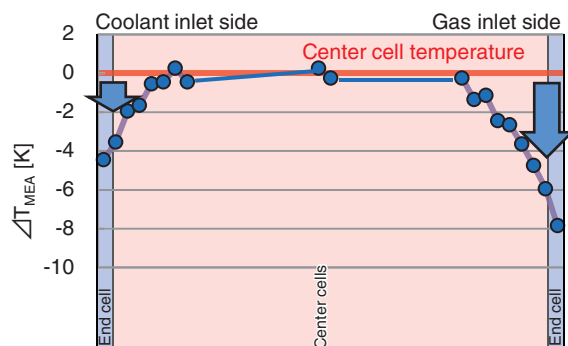


Fig. 3 Relationship between ΔT_{MEA} and position in stacking direction

Two methods can be considered for the estimation of ΔT_{MEA} using conventional simulation technologies. The first of these, for single-cell calculations, is to add a heat release term to the input parameters in the cell stacking direction, thus considering the cell being focused on as an end cell. The second method is to conduct coupled calculations for all the cells forming the stack and the case components (termed “stack coupled calculations” below). Because the first method does not consider reduction in the temperature of contiguous cells, it is not suitable for high-accuracy estimation. The second method was able to estimate reduction in the temperature of the end cells with a high degree of accuracy, but it necessitated large-scale calculations.

The development goal for the simulation technology discussed in this paper was to estimate the temperature of the MEA at different arbitrary stacking positions within the stack to a high level of accuracy without large-scale calculations.

2. Examination of Calculation Methods

2.1. Effect of Each In-cell Heat Exchange Factor on MEA Temperature

Figure 4 shows the flow directions of the various fluids in the stack fitted in the CLARITY FUEL CELL. The structure of the stack sees air and coolant flowing in the same direction, while H_2 flows in the opposite direction.

Anode gas (H_2), cathode gas (air), and coolant are assumed to be the factors that are able to be used for heat exchange within the cells in relation to the heat generated from the MEA during power generation. The dominant heat exchange factors for power generation at current densities of 0.03 A/cm^2 , 0.1 A/cm^2 , 0.3 A/cm^2 , 0.7 A/cm^2 , and 1.5 A/cm^2 , representing generation conditions assumed to occur within the vehicle environment, were identified by calculating the amount of heat exchange from the degree of change in temperature at the inlet and outlet and physical property values (heat capacity and flow rate)

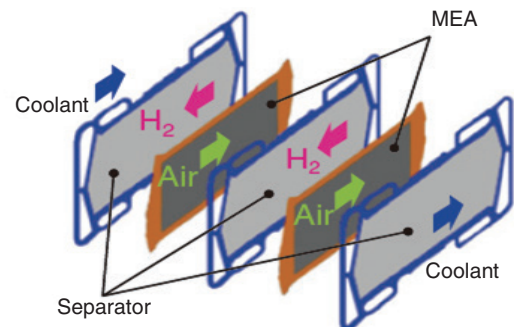


Fig. 4 Flow directions of anode gas, cathode gas, and coolant

for each factor. Figure 5 shows the ratio of amount of heat exchange for each fluid with the amount of heat exchange for the coolant as the benchmark. These results show that the amount of heat exchange for the gases (H_2 and air) is $\pm 3\%$ or less in relation to the amount of heat exchange for the coolant. The ratio of the heat exchange amounts for the gases and the coolant differs depending on the stack current per unit area because the inlet temperature and flow rate of the coolant and the gases differ. In addition, the fact that H_2 heat exchange occurs on the heat release side is due to the structure of the stack, which causes H_2 to flow in the opposite direction to the direction of coolant flow, the dominant heat exchange factor.

2.2. Effect of MEA In-plane Heat Conduction on MEA Temperature

The transfer of MEA heat is assumed to be effected through heat conduction and heat transfer in the cell stacking direction orthogonal to heat conduction. Figure 6 shows an image of heat exchange when in-plane heat

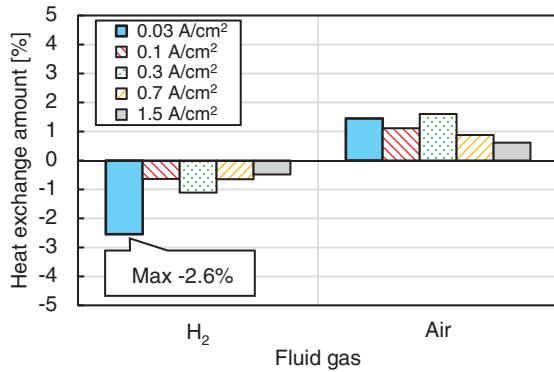


Fig. 5 Heat exchange amount for each fluid with coolant as benchmark

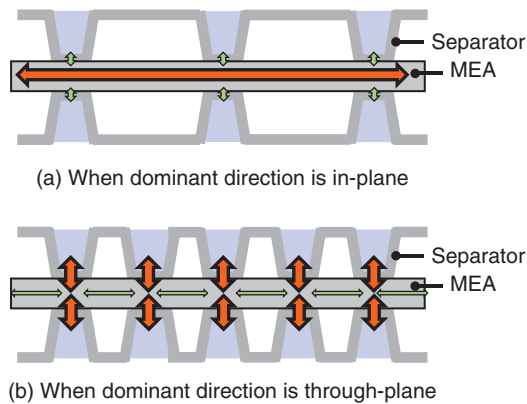


Fig. 6 Schematic image of heat exchange in cell in in-plane direction and through-plane direction

conduction is dominant and when heat transfer in the cell stacking direction is dominant. It is assumed that when the effect of heat conduction in the MEA in-plane direction is greatest, the in-plane MEA temperature distribution is uniform, irrespective of the temperature gradient of the coolant flowing from the stack inlet to the stack outlet. Conversely, it is assumed that when heat transfer in the cell stacking direction is dominant, the MEA in-plane temperature distribution follows the temperature gradient of the coolant flowing from the stack inlet to the stack outlet.

Given this, temperature measurements were conducted at the three center cell MEA in-plane temperature measurement positions shown in Fig. 7 (at the coolant inlet, at the center of the cell, and at the coolant outlet) under power generation conditions of 0.1 A/cm^2 , which are representative of stack warm-up mode. Figure 8 shows a graph of temperature differentials ($\Delta T_{\text{cool in-out}}$) with coolant inlet temperature as the benchmark. These results indicate that the temperature distribution in the MEA in-plane direction is identical to the coolant temperature distribution. In other words, heat exchange between the coolant and the MEA can be considered to be more strongly correlated with MEA temperature than heat conduction in the MEA in-plane direction.

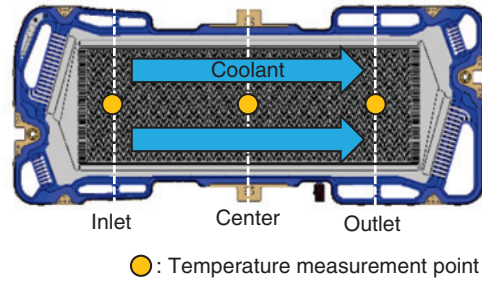


Fig. 7 Temperature measurement points for inlet, outlet, and center cell

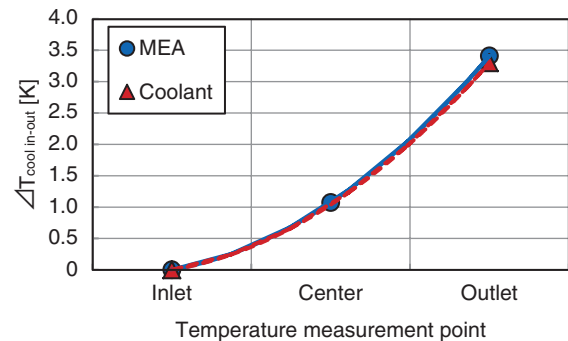


Fig. 8 Temperatures of MEA and coolant

2.3. Concept of Calculation Method

Based on the results discussed above, it was considered that calculation of the heat exchange between H_2 and air and MEA in-plane heat conduction could be omitted because their influence on the temperature of the MEA was minimal. In addition, focusing on the fact that the effect of MEA in-plane heat conduction was minimal made it possible to limit the calculation position for each component to one point in the cell plane. While the conventional simulation technology is a tool for the estimation of cell in-plane temperature distribution, as Fig. 9 shows, the distinctive feature of the simulation technology under consideration in this paper is the ability to estimate temperature distribution in the cell stacking direction, including the case components. The technology makes it possible to arbitrarily set calculation points in the cell plane, and by conducting calculations for multiple in-plane points, it is possible to accurately estimate the temperature distribution for the stack as a whole.

When creating simulations of the cell stacking direction, there are factors on which the three-dimensional structure of the cells has a considerable effect, such as Reynolds number, an indicator of the degree of turbulence. Because of this, estimative accuracy is insufficient when only the cell stacking direction is simulated. However, conducting detailed calculations for factors affected by the three-dimensional structure of the cells would enormously increase the calculation volume. Therefore, it was decided, in addition to general theoretical equations and physical property values, to introduce a correction term (a correlation coefficient) based on fundamental data measured in a stack while generating power. This addition made it possible to reduce the calculation volume in the simulation technology discussed in this paper without any reduction of estimative accuracy.

Based on the range of variation in actual measurements, taking sensor errors and variations between components into consideration, the target value for temperature estimation accuracy was set at within $\pm 5\%$ of actual measurements. This target value represented the same estimation error as the conventional cell in-plane simulation technology.

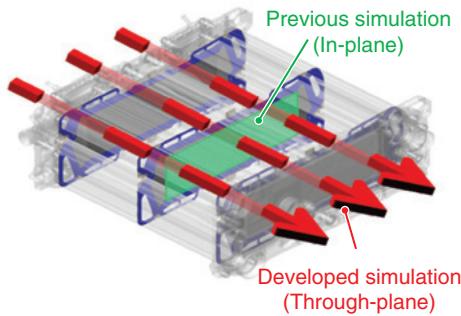


Fig. 9 Roles of developed simulation and conventional simulation

3. Simulation Creation of Transient Heat Transfer in Cell Stacking Direction

3.1. Heat Transfer within Components

The heat transfer within component A [Fig. 10(a)], considered by the simulation of transient heat transfer in the cell stacking direction, is assumed to be made up of heat conduction, heat transfer (heat release to the air, heat exchange with the coolant), and internal heat generation, as shown in Figs. 10(a), (b), and (c) and Eq. (1). The meanings of the symbols in the equation are shown in a list of symbols at the end of this paper.

$$mC \frac{dT_n}{dt} = Q_c - Q_t + Q_h \quad (1)$$

The terms related to heat conduction in Eq. (1) are expressed as Eqs. (2)-(4).

$$Q_c = Q_{n-1} - Q_n \quad (2)$$

$$Q_{n-1} = \frac{\lambda}{l} (T_{n-1} - T_n) A_c \quad (3)$$

$$Q_n = \frac{\lambda}{l} (T_n - T_{n+1}) A_c \quad (4)$$

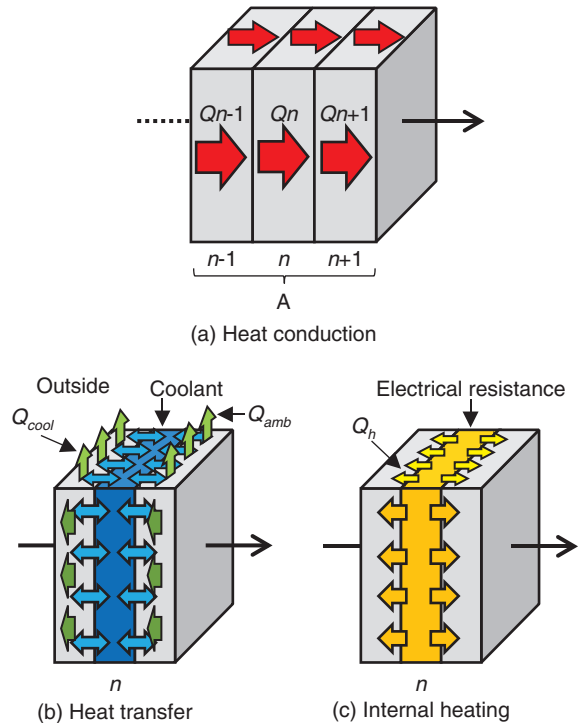


Fig. 10 Schematic image of heat flux in components

The terms related to heat transfer in Eq. (1) are expressed as Eqs. (5)-(8). Here, the heat transfer coefficient of the coolant in the coolant heat exchange expressed by Eq. (7), h_{cool} , is a value that fluctuates due to the coolant flow velocity and the effect of three-dimensional structures, such as the shape of the cell flow channels. Due to this, it is not easy to express h_{cool} using a theoretical equation. Given this, a relational expression for the correlation coefficient α and the coolant flow rate, determined experimentally in the actual cell shown in Fig. 11, was employed as the h_{cool} correction term in the simulation of transient heat transfer in the cell stacking direction. This made it possible to express the fluctuations in h_{cool} due to fluctuations in the coolant flow rate for a specific cell shape without any increase in the volume of calculations.

$$Q_t = Q_{amb} + Q_{cool} \quad (5)$$

$$Q_{amb} = h_{amb} A_{amb} (T_n - T_{amb}) \quad (6)$$

$$Q_{cool} = \alpha h_{cool} A_{cool} (T_n - T_{cool}) \quad (7)$$

$$\alpha = f(v_{cool}) \quad (8)$$

Different equations are employed for internal heat generation in general components and the MEA in Eq. (1). Equation (9), which considers Joule heat, is employed for general components, while Eq. (10), which considers heat loss in the chemical reactions occurring in the fuel cells, is employed for the MEA.

$$Q_{h\ parts} = I^2 R \quad (9)$$

$$Q_{h\ MEA} = I \Delta E \quad (10)$$

3.2. Heat Transfer between Components

The movement of heat at the interface between

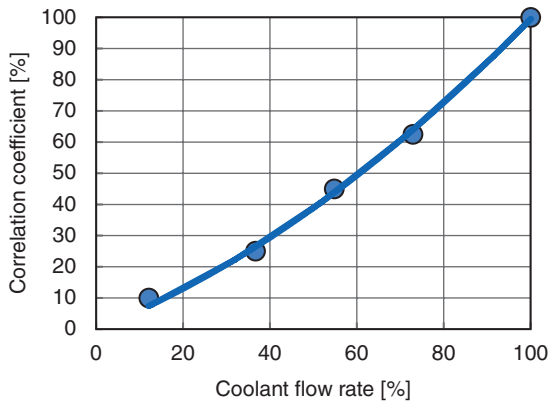


Fig. 11 Relationship between correlation coefficient and coolant flow rate

components A and B (Fig. 12) considered by the simulation of transient heat transfer in the cell stacking direction was set as the heat transfer between components and the heat generated by interface resistance, as expressed by Eqs. (11)-(13).

$$Q_B = Q_{A-B} + Q_{hs} \quad (11)$$

$$Q_{A-B} = h_{A-B} A_c (T_A - T_B) \quad (12)$$

$$Q_{hs} = I^2 R_s \quad (13)$$

By applying the above equations to each of the component parts of the stack and inputting their respective physical property values and dimensions as parameters, it was possible to create a model of the stack components.

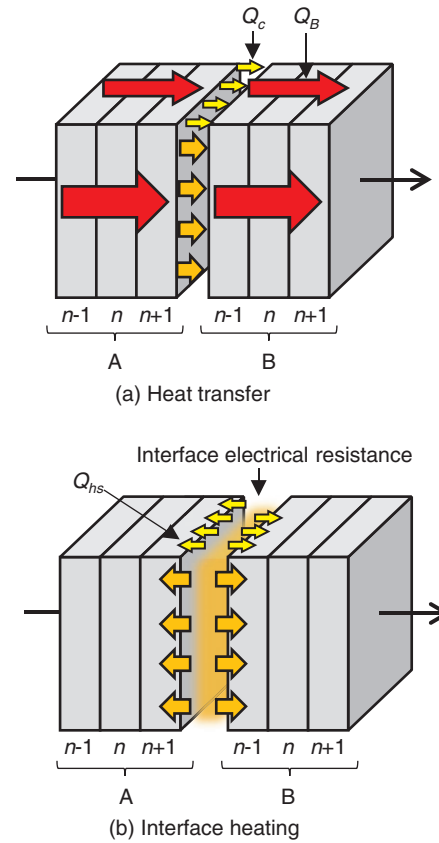


Fig. 12 Schematic image of heat flux between components A and B

4. Simulation Results and Verification of Accuracy

4.1. Verification of Temperature Estimation Accuracy in Different Generation Modes

Calculations were conducted for power generation at

current densities of 0.03 A/cm², 0.3 A/cm², 0.7 A/cm², and 1.5 A/cm², representing generation conditions assumed to occur within the vehicle environment. Actual measurements and calculation results were compared, with the center cells, the end cells, and the end plates, the outermost of the case components, as the targets of the evaluation (Fig. 13). The maximum temperature estimation error during steady state operation for each operating condition was 3.7% against the actual measurements, indicating that the target figure was realized. These results verified that the simulation of transient heat transfer in the cell stacking direction is not restricted to specific modes, but is applicable to stack temperature estimation across a wide range of generation modes, from low loads to high loads.

4.2. Verification of Temperature Estimation Accuracy in Stack Warm-up Mode

Figure 14 shows an image of stack warm-up mode, the mode in which it is assumed that the temperature differential between the MEA in the center and end cells (ΔT_{MEA}) reaches its peak in the vehicle environment. This mode is a sequence of modes, in which the stack is started in a state in which the cell temperature has declined in the external air temperature due to a long soak, and the cell temperature begins to increase due to idling generation (transient phase) until the temperature becomes constant (steady-state phase). In this mode, the case components, which possess a high heat capacity but a limited tendency to follow the temperature of other components, are prone to the development of a heat differential with the end cells. As a result, the heat release from the end cells to the case components increases, increasing ΔT_{MEA} . The accuracy of temperature estimation in transient states was verified by comparing actual measurement data and calculation results for this mode.

Figure 15 shows the temperature transition of actual measurements and calculation results for the center cells,

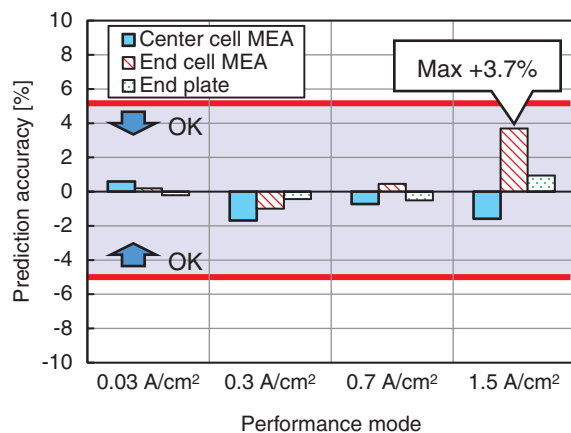


Fig. 13 Simulation and measurement results for fuel cell stack part temperatures during steady-state operation

end cells, and end plates in a transient state in stack warm-up mode. The temperatures in this graph are expressed as ratios, with the external temperature as the benchmark

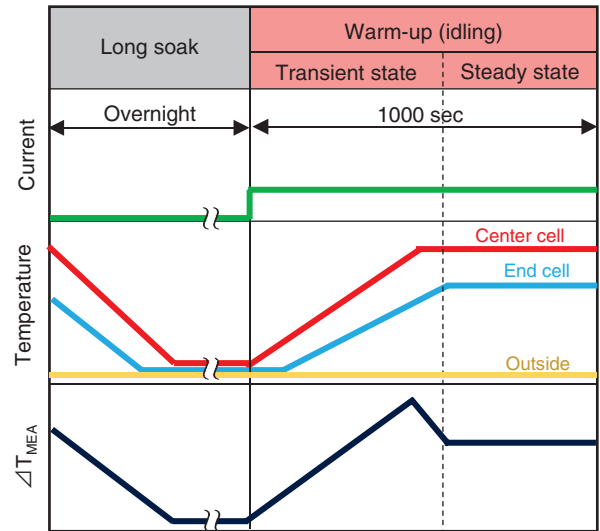


Fig. 14 Schematic image of warm-up mode

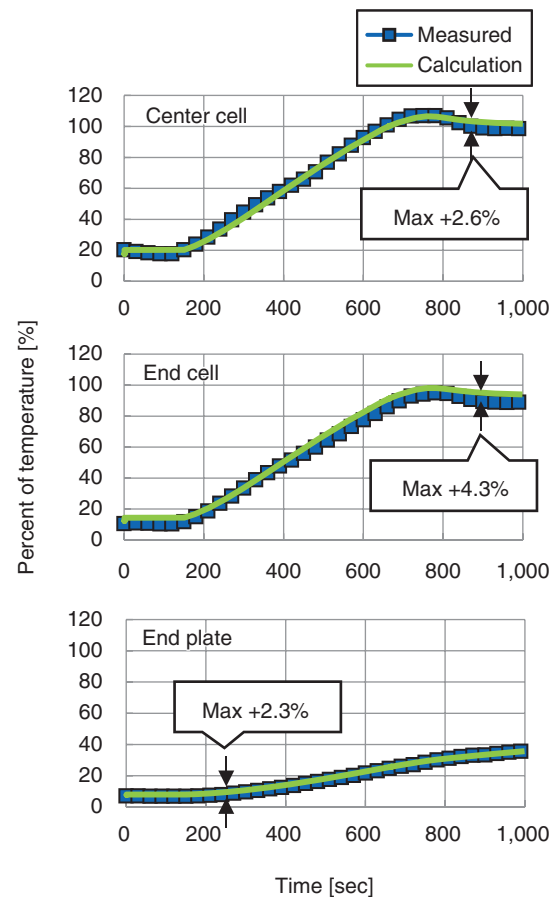


Fig. 15 Simulation and measurement results for fuel cell stack part temperatures in transient state during warm-up mode

and the temperature of the center cells after 1000 sec has elapsed as the maximum temperature. These results verified that the simulation technology is capable of reproducing the process of temperature increase in each component from a state in which the temperature of the cells had declined in the external air temperature after a long soak. In addition, the temperature estimation error was a maximum of 4.3% against measurement results, demonstrating that the simulation also realized the target value in stack warm-up mode, which includes transient states. This verified the fact that the simulation of transient heat transfer in the cell stacking direction could also be applied to the estimation of stack temperature, including transient states.

4.3. Verification of Calculation Speed

Using the conventional simulation technology, when a work station-level calculation resource was employed, calculation time for stack warm-up mode was several days or more for only one center cell. By contrast, using the simulation technology discussed in this paper, when a laptop computer-level calculation resource was employed, calculation time for stack warm-up mode was 60 sec. Even when calculations were conducted for multiple in-plane points, it was possible to complete the calculations in a matter of several minutes. This demonstrates that the development of the simulation of transient heat transfer in the cell stacking direction has made it possible to estimate the temperature distribution for the entire stack with a practical calculation time, even when standard calculation resources are employed.

5. Conclusion

A simulation of transient heat transfer in the cell stacking direction was developed that makes it possible to estimate the temperature of the MEA at different arbitrary stacking positions within the stack at a high level of accuracy and without large-scale calculations. With regard to factors that have a considerable effect on the temperature of the MEA but necessitate a high volume of calculations, the introduction of a correction term based on fundamental data measured in a stack while generating power reduced the calculation volume. At the same time, by contrast with the conventional simulation technology, a method of identifying and omitting factors that have little effect on the temperature of the MEA was developed. Temperature estimation error under steady-state stack generation conditions was a maximum of 3.7% against actual measurement results. In addition, temperature estimation error in stack warm-up mode including transient states was a maximum of 4.3% against actual measurement results. These results demonstrate that the temperature estimation accuracy of the simulation of transient heat transfer in the cell stacking direction achieved the target value for estimation error of within $\pm 5\%$ against actual measurement results.

References

- (1) Kimura, K., Kawasaki, T., Ohmura, T., Atsumi, Y., Shimizu, K.: Development of New Fuel Cell Vehicle CLARITY FUEL CELL, Honda R&D Technical Review, Vol. 28, No. 1, p. 1-8
- (2) Chiba, H., Iwasawa, C.: Optimization of Fuel Cell Generation Environment by Water Observation using Neutron Radiography, Honda R&D Technical Review, Vol. 25, No. 1, p. 122-131
- (3) Yoneda, M., Tago, Y., Suzuki, K., Takimoto, M., Ejiri, E.: Macroscopic Modeling and Simulation for Polymer Electrolyte Membrane Fuel Cells, ECS Transactions, Vol. 3, Issue 1, p. 1105-1114, (2006)
- (4) Nagumo, K., Inoue, M.: Numerical Prediction of Water Content Distribution of Catalyst Coated Membrane of Fuel Cell Stack using Simulation, Honda R&D Technical Review, Vol. 28, No. 1, p. 26-33
- (5) Yoneda, M., Yoshimura, H., Takayama, T., Motegi, H., Takayama, R., Kariya, N.: Prospects of Modeling and Simulation for Polymer Electrolyte Fuel Cell, Mizuhojohosoken Giho, Vol. 6, No. 1, p. 51-58, (2014) (in Japanese)
- (6) Nonoyama, N., Ikogi, Y.: PEMFC model development for designing MEA: Model Validation and a Parameter Study for Low Humidity Operations, ECS Transactions, Vol. 16, Issue 2, PART 1, p.13-21, (2008)
- (7) Yamamoto, K., Fujita, Y., Yasumoto, E., Sugawara, Y.: Multiscale Simulation for Fuel Cell, Panasonic Technical Journal, Vol. 63, No. 1, p. 72-77, (2017) (in Japanese)
- (8) Takayama, T., Yoshimura, H., Tsukamoto, T., Motegi, H., Takayama, R., Yoneda, M.: Kotaikobunshigata Nenryodenchi Shimyureta "P-Stack4.0" Furusutakkuseino Kaisekio Jitsugensuru Senyosofutouea, Kurinenerugi, Vol. 26, No. 1, p. 1-8, (2017) (in Japanese)

List of Symbols

A_{amb}	Heat transfer area of external air [m^2]
A_c	Heat conduction area [m^2]
A_{cool}	Coolant heat transfer area [m^2]
C	Heat capacity [J/(kg·K)]
h_{A-B}	Coefficient of heat transfer between A and B [W/(K· m^2)]
h_{amb}	Heat transfer coefficient of external air [W/(K· m^2)]
h_{cool}	Heat transfer coefficient of coolant [W/(K· m^2)]
I	Current [A]
l	Length [m]
m	Mass [kg]
t	Time [sec]
T_A	Surface temperature of A [$^{\circ}C$]
T_B	Surface temperature of B [$^{\circ}C$]
T_{amb}	External air temperature [$^{\circ}C$]

T_{cool}	Coolant temperature [°C]
T_n	Temperature of n components [°C]
T_{n+1}	Temperature of $n+1$ components [°C]
Q_{A-B}	Amount of heat transfer between A and B [W]
Q_B	Amount of heat conduction in B [W]
Q_{amb}	Amount of heat release to external air [W]
Q_{cool}	Amount of heat exchange with coolant [W]
Q_c	Amount of heat conduction [W]
Q_h	Amount of internal heat generation [W]
$Q_{h\ parts}$	Amount of internal heat generation for general components [W]
$Q_{h\ MEA}$	Amount of internal heat generation for MEA [W]
Q_{hs}	Amount of heat generation at interface [W]
Q_{n-1}	Amount of heat conduction for $n-1$ [W]
Q_n	Amount of heat conduction for n [W]
Q_i	Amount of heat transfer [W]
R	Specific electrical resistance [Ω]
R_s	Interface electrical resistance [Ω]
v_{cool}	Coolant flow rate [L/min]
α	Correlation coefficient [-]
λ	Thermal conductivity [W/(m·K)]
ΔE	Voltage drop [V]

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Prototype-less Development of Honda Smart Key System using Magnetic Field CAE

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ABSTRACT

In order to increase the development efficiency of smart key systems, a technology was developed that makes it possible to verify the mounting positions and key position detection performance of LF antennas in a short period and with no prototype vehicle using magnetic field CAE. A body model employed in collision CAE was simplified and the state of conduction in the vehicle was added in order to formulate body and seat models to study the magnetic field distribution inside and outside the vehicle, which changes in response to the body shape and the state of conduction between connected sections, at stages of vehicle development during which no prototype exists. The results of magnetic field CAE analysis and tests showed that it is possible to analyze the magnetic field produced by an LF antenna, which changes due to the effect of body shape and state of conduction, in a short period. In addition, a small, wireless electromagnetic sensor displaying stable measurement accuracy was developed, making it possible to increase analytic accuracy and boost the efficiency of studies. As a result, it is now possible to conduct smart key developments at the initial stage of new vehicle development, when no prototype vehicle exists.

1. Introduction

The diffusion of smart key systems has made it possible for users to conduct operations such as locking and unlocking the vehicle doors and starting up the power unit simply by carrying a key and operating a switch, with no need to insert a mechanical key into a keyhole. Honda commenced using smart key systems starting from the 2003 model Accord, and now uses the Honda Smart Key System in almost all of its vehicles.

Honda Smart Key detection employs the wireless communications shown in Fig. 1. Multiple low-frequency (LF) antennas are fitted inside and outside the vehicle (for example, in the center console and the door handles) in order to detect the position of the key.

When a high-frequency current is impressed in an LF antenna, a magnetic field is generated in the surrounding area. The strength of this magnetic field becomes attenuated in inverse proportion to the cube of the distance from the antenna. The key measures the strength of the magnetic fields produced by each of the LF antennas, and sends this information to the ECU fitted in the vehicle using radio waves in the ultra-high-frequency (UHF) band. Based on this information regarding the strength of the magnetic field, the ECU judges whether the key is inside or outside the vehicle.

When the ECU judges that the key is outside the vehicle,

it allows the vehicle doors to be locked or unlocked; when it judges that the key is inside the vehicle, it allows the power unit to be started.

Further, because the system is designed to only respond to a key that has been registered with the vehicle in advance, it also boosts vehicle security performance as an immobilizer. In addition, key position information can also be used for other functions that increase the convenience of the vehicle, for example Honda's Smart Welcome Lamp⁽¹⁾, which illuminates the door handles when the user

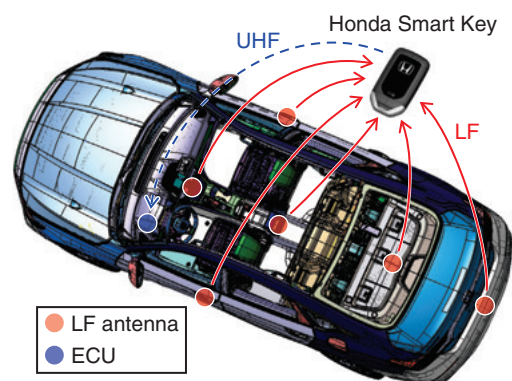


Fig. 1 Wireless communications in Honda Smart Key system

¹ Automobile Center

approaches the vehicle (Fig. 2) and Hands-free Access Power Tail Gate⁽²⁾, which allows the tailgate to be opened and closed via the user's foot for hands-free access (Fig. 3).

At the same time, misdetection of the position of the key could result in the key being locked in the car or the power unit being mistakenly started, and it is, therefore, necessary to design the systems to accurately judge key position.

However, most vehicle bodies are constructed from conductive materials such as ferrous materials and aluminum alloys. The lines of magnetic force generated by the LF antenna have the property of curving under the influence of conductive materials, and the mode of diffusion of the magnetic field inside and outside the vehicle changes due to factors such as the vehicle design and the state of conduction between the conducting parts themselves.

Due to this, when developing smart key systems, it is necessary to determine the mounting positions for the LF antennas and verify key position detection performance in each new vehicle development. In addition, the same vehicle model might have different specifications; minivans, for example, are produced with specifications offering two rows of seats and specifications offering three rows of seats. In cases in which it is considered that the mode of diffusion of the magnetic field inside and outside the vehicle might be affected, it is necessary to conduct verifications for each specification.

Key position detection performance is verified at the initial stage of the new vehicle development process, using an existing mass production vehicle in which part of the structure has been modified. Since the vehicle design and

structure of the vehicle employed in verification and the new vehicle are different, verification is conducted again, but if the target detection performance is not obtained, it is necessary to change the design, including the design of peripheral parts, in order to change the mounting positions of the LF antennas.

Today, the use of prototype-less development^{(3), (4)} in order to shorten the development period has gained ground, and there are demands for the use of this method in the development of smart key systems. In order to make it possible to conduct prototype-less development of smart key systems, the research discussed in this paper, therefore, developed a simulation technology using magnetic field CAE able to evaluate the LF antenna mounting positions and key position detection performance at the initial stage of new vehicle development.

2. Development Aims

The following aims were set for the development of a simulation technology capable of prototype-less evaluation of smart key systems:

- (1) Ability to utilize data that can be obtained at the initial stage of new vehicle development.
- (2) Ability to evaluate LF antenna mounting positions and key position detection performance.
- (3) Ability to conduct calculations within a practical timeframe for developments for mass production.

3. Magnetic Field CAE Model

A model was developed for use in magnetic field CAE in order to analyze the magnetic fields produced inside and outside the vehicle by the LF antennas.

3.1. Body Model

Since the mode of diffusion of the magnetic field inside and outside the vehicle changes with differences in vehicle design, it would be necessary to formulate a body model for each new vehicle development. For the sake of efficient body model development, a body model used in collision CAE⁽⁵⁾ was employed as the base in the development of the magnetic field CAE model.

However, because collision CAE models are designed to reproduce the deformation of the vehicle body in a collision, the shape of each part is modeled in detail, and as a result, analytic calculations take time. In order to reduce the time necessary for calculations, the model was simplified by isolating only the elements necessary for magnetic field CAE analysis. In addition, the state of conduction between the conductive parts was also added to the model.

3.2. Seat Model

As in the case of the body model, the seat model was simplified by isolating only the elements necessary for magnetic field CAE analysis.



Fig. 2 Smart Welcome Lamp



Fig. 3 Power Tail Gate for hands-free access

When lines of magnetic force pass through a conductor with a loop structure, electromagnetic induction generates an induced current in the conductor. The lines of magnetic force are distorted by the magnetic field produced by this induced current, and this affects the magnetic field distribution. Due to this, the seats (which have a large and complex structure by comparison with other parts possessing conductive loop structures) have a considerable effect on the magnetic field distribution inside and outside the vehicle.

In addition, the state of conduction of the seat rails and the movable sections of the seats changes due to the effect of the position of the seats and the oxide films formed on the metal surfaces. The state of conduction between each of the seat frame components that forms the conductive loop structure of the seat was therefore added to the model.

3.3. LF Antenna Model

The LF antennas are approximately 60 mm in total length, and formed by winding coils around ferrite cores. The magnetic field CAE model reproduced the same shapes and electrical characteristics as an actual LF antenna, including the ferrite core dimensions and magnetic permeability, the number of coil windings, and the wire diameter and inductance.

In order to verify the ability of the model to reproduce the magnetic field distribution produced by the LF antennas mounted on the vehicle body, tests were conducted with LF antennas placed on metal plates, and the measurements were compared with the results of analysis using CAE. Assuming a scenario in which the key was on the floor of the vehicle, the results were compared for a position at a height 50 mm above the metal plate.

The market electromagnetic field simulation software CST Studio Suite and a general-purpose work station were employed in the magnetic field CAE analysis. The finite element method (FEM) was employed as the solution method.

Figure 4 shows the relationship between the strength of the magnetic field and distance from the LF antenna. Analytic deviation was a maximum of 8%. Converted into distance, this was smaller than the length of a single key, and it was judged that sufficient accuracy had been achieved to know whether a key is inside or outside the vehicle at the initial stage of development.

Figure 5 shows an example of analysis of the magnetic field distribution for an LF antenna mounted close to the center of the interior of the vehicle. The LF antenna was situated in the body model at the same position and angle as it would be in an actual vehicle, and the magnetic field distribution created by the LF antenna was analyzed.

The results showed that magnetic field CAE made it possible to study the uneven spread of the magnetic field generated by the LF antenna in the interior of the vehicle due to the effect of the seats and other conductive parts, and the shielding of the field by the vehicle body and the doors.

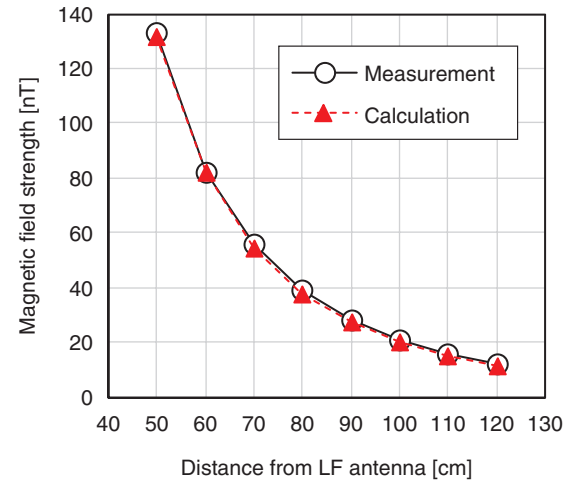


Fig. 4 Relationship between distance and magnetic field strength in measurement and CAE results

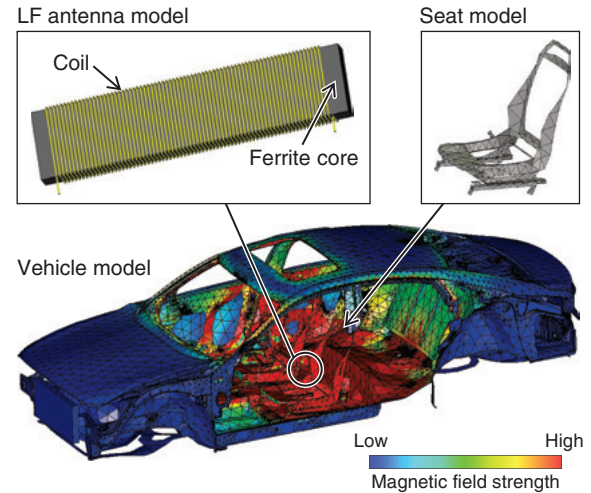


Fig. 5 Magnetic field analysis using CAE

4. Verification of Magnetic Field CAE

The effect of the conduction state of the seat frames on magnetic field distribution was studied using magnetic field CAE. In addition, the ability to reproduce the magnetic field distribution in an actual vehicle using magnetic field CAE was verified by comparing the results of measurements conducted in tests with CAE results.

4.1. Seat Employed in Tests

In order to determine the effect of the state of conduction of the seat frames on the magnetic field distribution, an experimental seat was created that made it possible to freely change the state of conduction between each of the seat

frame components (Fig. 6).

For the experimental seat, the main structure of a vehicle seat was isolated, and metals and resins of differing conductivity were selected as the materials employed in the joints between the seat frame components. A scenario in which the joints were all resin and the framework components were all insulated from each other was defined as completely non-conductive. Additionally, a scenario in which the joints were all metal and the framework components were all conductive was defined as completely conductive.

4.2. Development of Magnetic Field Sensor

It can be considered that the key may be placed in a narrow space, such as a cup holder or console box. Given this, it is necessary during the development of a smart key system to verify that the key position can be detected even when the key is located in a range of narrow areas in the vehicle.

A small wireless magnetic field sensor able to measure nano-tesla-order magnetic fields, of the same magnitude as that of the key, was, therefore, developed (Fig. 7).

In addition, it increased the efficiency of verification of analytic accuracy by clarifying measurement deviation due to factors including the measurement accuracy of the measurement device and the directivity of reception, as included in the measurement results.

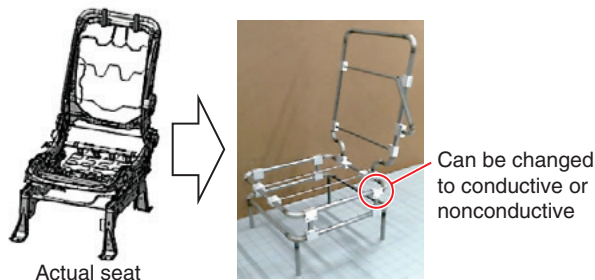


Fig. 6 Experimental seat

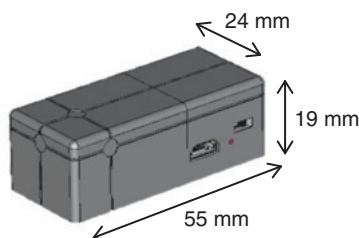


Fig. 7 Magnetic field sensor

4.3. Effect of Seat Frame on Magnetic Field Distribution

In order to determine the effect of the state of conduction of the seat frames on the magnetic field distribution, the magnetic field distribution was studied using magnetic field CAE when the state of conduction of the experimental seat was completely non-conductive and completely conductive.

The experimental seat and the LF antenna were positioned on a metal plate that simulated a vehicle floor panel. The completely non-conductive seat was insulated from the metal panel, while the completely conductive seat was conductive with the metal panel.

Figure 8 compares magnetic field distributions, analyzed using magnetic field CAE, for the completely non-conductive and completely conductive seats. In the case of the completely non-conductive seat, the magnetic field diffuses evenly, while diffusion is obstructed in the case of the completely conductive seat. These results confirmed that the magnetic field distribution changes due to the state of conduction of the seat frame.

In order to apply magnetic field CAE in development for mass production, it would be necessary to be able to detect the key position irrespective of the state of conduction of the seats. In order to consider the effect of the seat frame on the magnetic field distribution to the greatest extent possible, it was, therefore, decided to make the seat model completely conductive, and to also make the seat and the floor panel mutually conductive.

4.4. Reproduction of Magnetic Field Distribution inside Vehicle

The correlation between the analysis results and the measurement results was studied using an actual vehicle in order to verify the analytic accuracy of magnetic field CAE.

In order to determine the contribution of the state of conduction of the seat frame, the driver's seat was replaced by an experimental seat in a completely conductive state. At this time, the seat and the floor panel were also in a conductive state.

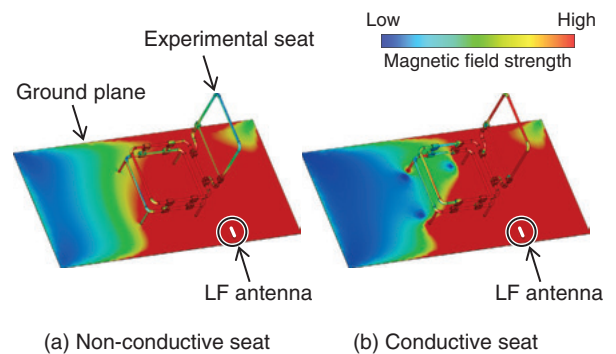


Fig. 8 Comparison of magnetic field distribution with conduction difference in seats

Figure 9 shows the results of measurements and CAE analysis of the magnetic field distribution produced by an LF antenna mounted close to the center of the interior of the vehicle.

Even when the measurement deviation of the magnetic field sensor is taken into consideration, the maximum deviation between the measurement and analysis results was 10%.

Converted into distance, this was smaller than the length of a single key, confirming that it was possible to detect the position of a key in an actual vehicle with a sufficient level of accuracy using magnetic field CAE.

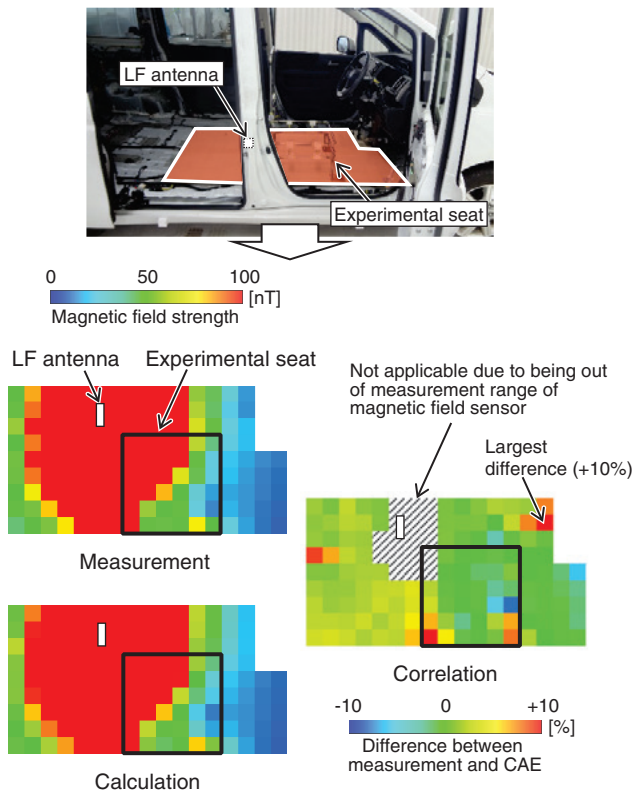


Fig. 9 Comparison of magnetic field distribution in measurement and CAE results for actual vehicle with experimental seat

5. Verification in Actual Vehicle

5.1. Study of Positioning of LF Antenna

This section will discuss an example of a study of the positioning of an LF antenna mounted inside a vehicle using magnetic field CAE.

Inside the vehicle, LF antennas are mounted on the instrument panel, the center console, and the trunk room, but in the case of sedans, because the structures in the vicinity of the trunk room are complex, key position

detection at the rear of the vehicle interior becomes an issue.

Magnetic field CAE was, therefore, used to study the performance of multiple LF antenna positioning proposals in key position detection at the rear of a sedan interior. Figure 10 shows the results of an analysis of the magnetic field distribution generated by an LF antenna positioned at the center console.

The analysis results indicate that only a slight difference in LF antenna mounting position changes the magnetic field distribution in the vicinity of the trunk room.

In this example, Proposal C, in which the magnetic field would diffuse uniformly towards the rear of the interior of the vehicle, was adopted in order to boost key detection performance at the rear of the vehicle interior.

Using magnetic field CAE, it was possible to study the positioning of LF antennas without evaluations employing an actual vehicle.

5.2. Key Position Detection Performance

Since the magnetic field distribution inside and outside the vehicle changes with the state of conduction of the seat frames, the realization of accurate key position detection under any seat frame conduction conditions becomes an issue.

A study was, therefore, conducted using a mass production minivan with three rows of seats in order to determine the effect of the seat frames on the magnetic field. In the interior of the vehicle, LF antennas were mounted in the instrument panel, on the floor close to the center of the vehicle interior, and in the cargo room.

Figure 11 shows measurement and analysis results for the magnetic field distribution inside and outside the vehicle

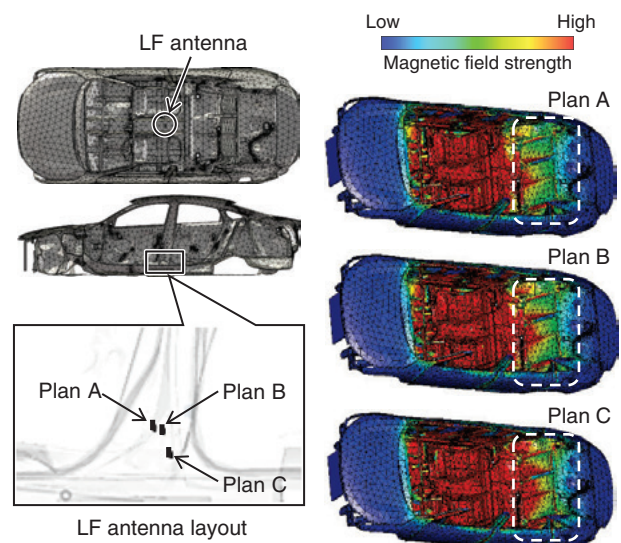


Fig. 10 Determining optimum layout from multiple antenna layouts

produced by an LF antenna positioned on the floor close to the center of the vehicle, where the effect due to the seats is greatest.

Figure 11(a) shows the distribution of the magnetic field that leaks to the outside of the vehicle. The results indicate that leakage is greatest from the gaps between the front and rear doors and the body, which are close to the LF antenna. The status of magnetic field distribution and the position at which leakage was greatest predicted by magnetic field CAE largely corresponded to the measurement results.

In order to realize accurate detection of key position outside the vehicle, the maximum value of leakage of the magnetic field obtained from magnetic field CAE was set as a reference value, and it was judged that a key position within a range in which the magnetic field was smaller than this reference value was outside the vehicle.

Figure 11(b) shows the magnetic field distribution inside the vehicle. A key position within a range in which the magnetic field was greater than this reference value

was judged to be inside the vehicle. The ranges, within which the keys were judged to be inside the vehicle, as predicted by magnetic field CAE largely corresponded to the measurement results.

These results verified that magnetic field CAE is capable of making predictions with a sufficient level of accuracy to conduct evaluations of key position detection performance at the initial stage of vehicle development.

5.3. Model Development Period and Calculation Time

A body model and seat model were formulated for magnetic field CAE using a body model for collision CAE during the development of a mass production four-door sedan. The total period for formulation of the models was approximately one month.

The calculation time for a model that positioned one LF antenna inside the vehicle was approximately five hours.

Since magnetic field CAE can be applied in analyses at the appropriate time following the study of mounting positions of the LF antennas, verifications can be conducted more efficiently than evaluations using an actual vehicle.

The results discussed above demonstrated that it is possible to evaluate smart key systems in a shorter period than evaluations using an actual vehicle by formulating magnetic field CAE models using data that can be obtained at the initial stage of development of new mass production vehicles.

6. Conclusion

A development technology for smart key systems able to be applied at the initial stage of vehicle development has been developed using magnetic field CAE.

The following is an overview of this development technology:

- (1) A technology was developed that allows the formulation of vehicle models from data available at the initial stage of vehicle development; these models make it possible to study the magnetic field distribution in a short period.
- (2) In order to verify the accuracy of the magnetic field CAE analysis, a small magnetic field sensor able to measure the magnetic fields inside and outside the vehicle with a known deviation was developed.
- (3) Studies using an experimental seat demonstrated that the state of conduction of the seat frames was the major factor of deviation in CAE analysis.
- (4) It was judged that it is possible to verify smart key systems using the results of analyses performed using magnetic field CAE.
- (5) It was verified that the new development technology could be applied in developments for mass production.

References

- (1) https://www.honda.co.jp/factbook/auto/LEGEND/201411/LEGEND_1411.pdf (in Japanese)

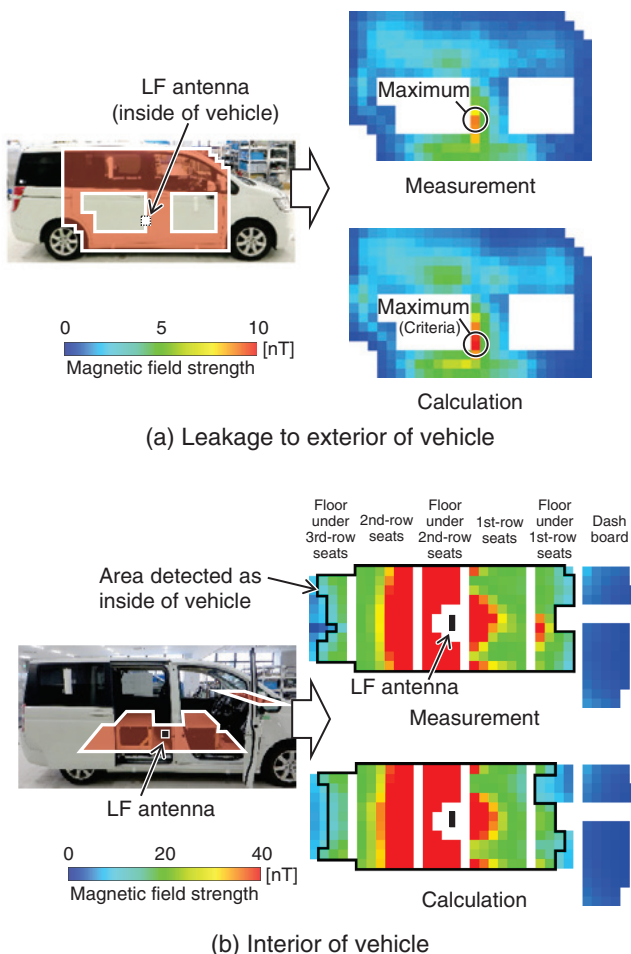


Fig. 11 Comparison of magnetic field distribution inside and outside mass production vehicle in measurement and CAE results

- (2) https://www.honda.co.jp/factbook/auto/CR-V/201808/CRV_201808.pdf (in Japanese)
- (3) Fukazawa, M., Unigame, S.: Method for Predicting Vibration Transmission of Car Air-conditioning Hose in Light of Prototype-less Development, Honda R&D Technical Review, Vol.30, No.2, p.87-92
- (4) Yoshida, T., Hamada, Y., Numoto, M., Inada, T.: Development of Advanced Keyless System Using Magnetic Field CAE, Mazda Technical Review, No. 30, p. 181-185, (2012) (in Japanese)
- (5) <https://www.honda.com/safety/virtual-and-real-world-tests/>

■ Author ■



Yoshio TANDA



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Prediction Technology for Thermal Mechanical Fatigue Strength of Cylinder Head

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ABSTRACT

A method was established for predicting the thermal mechanical fatigue strength of a cylinder head without the need for verification by engine tests. The accuracy of thermal analysis was enhanced by revising the turbulent model and physical property values used in CFD calculations to calculate the thermal boundary conditions. Thermal mechanical fatigue analysis used a material model that takes into account the kinematic hardening rule and hardness overaging, which is dependent on temperature. The analysis accuracy was also increased by changing the bolt connection model from FEM node merging to multi-point constraint connection. Engine durability tests using an actual engine were performed to verify the accuracy of this CAE prediction technology, and the results confirmed that cracks occurred at the durability test cycle predicted by CAE. This research helped eliminate the former need for verification with measured temperatures, and enabled model-based prediction of thermal mechanical fatigue strength in the design phase.

1. Introduction

Automobile engines face increasing demands for higher power, better fuel economy, and lower emissions. This is creating the need to address higher exhaust gas temperatures due to the ending of fuel rich operation in order to protect exhaust parts, and increased thermal load due to higher specific power produced by turbocharged downsizing engines, etc.

The cylinder head consists of a combustion chamber and exhaust port, which are exposed to high temperature, high-pressure flames and combustion gases, a water jacket through which coolant flows, a cam chamber with valve system and oil passages that are exposed to engine oil, and an intake port that is a passage for fresh air. This means that even though the cylinder head is a single part, the thermal distribution in the operating condition exhibits large deviation, which produces thermal strain. The locations where thermal mechanical fatigue (TMF) strength due to high thermal load is an issue are the combustion chamber and exhaust port, which become hot during operation, and

the adjacent water jacket. It is a challenge to measure these locations with strain gauges, so CAE has been used to predict TMF.

TMF^{(1), (2)} accumulates due to repeated thermal strain produced by changes in both the thermal distribution and stiffness distribution stemming from the material properties and constraints. Factors producing thermal strain in the cylinder head can be divided by constraint type into the two types of internal constraint and external constraint, and thermal strain occurs due to the compound action of these constraints. Thermal strain due to internal constraint is produced by differences in thermal expansion due to deviation of the thermal distribution within a single part. For example, thermal strain is produced by relatively cooler surrounding areas constraining the hot combustion chamber. In contrast, thermal strain due to external constraint arises from constraint by other parts. For example, when the turbocharger undergoes thermal expansion, force is transmitted to the connections with the cylinder head via the bolts, which produces strain.

Evaluation of TMF strength calls for accurate prediction

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of thermal strain. However, the previous evaluation method had the following issues. First, it is necessary to perform temperature verification for CAE using the temperature measurement results in an actual engine using a cylinder head with approximately temperature measurement 100 points. This takes a period of around half a year from design to evaluation. Second, the CAE-based strain prediction accuracy was low in high-temperature, high-strain locations, and it is necessary to take into account more detailed material properties. Third, the turbocharger is high-temperature and high-stiffness and exhibits large thermal expansion, so high strain is produced around the bolt connections with the cylinder head. However, the contact model of the bolt connections did not match the actual situation, and the strain prediction accuracy was low. CAE verification previously used the temperature measurement results for the initial prototype, and if the TMF strength evaluation results did not satisfy the evaluation criteria in the condition with these issues remaining, the cylinder head had to be redesigned and durability tests repeated, which resulted in major development reworking. If the TMF strength evaluation results did not satisfy the evaluation criteria, the cylinder head had to be redesigned and durability tests repeated, which resulted in major development reworking.

2. Technological Elements of Evaluation Method

Figure 1 shows the system diagram for generation of thermal strain. Factors producing thermal strain can be divided into thermal distribution and stiffness distribution factors. Figure 2 shows the flow of CAE for TMF strength evaluation. First, the thermal boundary conditions of the exhaust port and water jacket are calculated by CFD, and then thermal analysis is performed. The calculated thermal distribution is used to perform TMF analysis, and the strain is extracted from the calculation results and used to evaluate the mechanical fatigue strength.

In order to address the previously mentioned issues for prediction of TMF strength, it is necessary to enhance the thermal boundary condition, material behavior, and constraint analysis accuracy. The key factors for calculating the thermal boundary conditions are the turbulent model used in CFD and the heat transfer coefficient calculation formula. The key factors for material behavior are the hardening rule and hardness overaging under a high temperature environment, and the key factor for constraint is the bolt connection model. This research performed the following contents with the goals of reducing redesign and durability test repetition and enabling evaluation of TMF strength in the design phase.

- (1) Develop a method for calculating the thermal boundary conditions in order to achieve accurate thermal analysis without the need for verification using an actual engine.
- (2) Develop a material model capable of reproducing

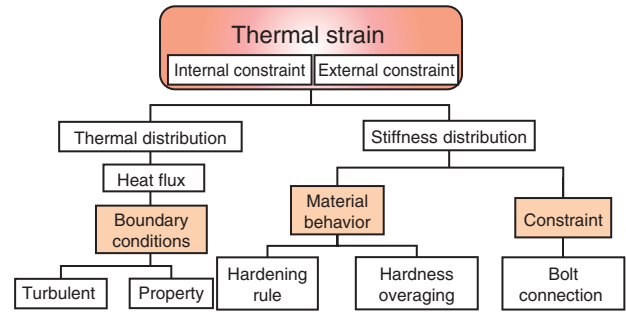


Fig. 1 Diagram of thermal strain factors

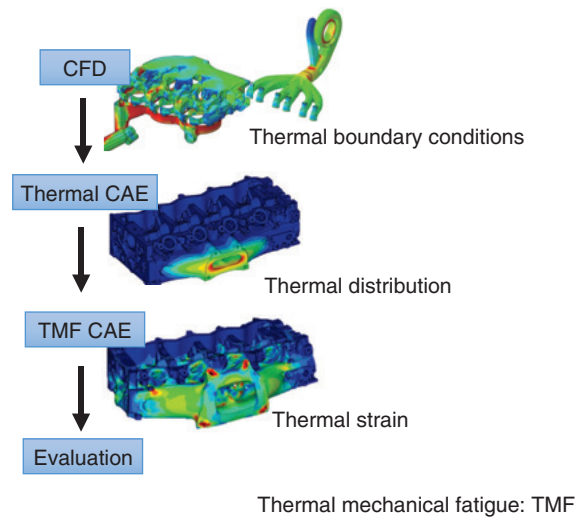


Fig. 2 Analysis flow of TMF evaluation

nonlinear properties of materials under high thermal loads.

- (3) Develop a connection model capable of reproducing the strain around bolt connections.

3. Development of Evaluation Method

3.1. Thermal Boundary Conditions

The methods used to calculate the thermal boundary conditions were revised to achieve a thermal analysis accuracy that would enable TMF strength evaluation in the design phase; that is to say, before verification using an actual engine is possible (Table 1). Here, the thermal boundary conditions are the heat transfer coefficient (HTC) from the fluid and the fluid temperature near the wall surface (hereafter, the boundary temperature). The CFD calculation method for the water jacket and exhaust port was revised, and CFD-based combustion analysis was newly performed for the combustion chamber. The target thermal analysis accuracy was $\pm 10^\circ\text{C}$.

Table 1 Thermal boundary conditions

		Previous method	New method
Water jacket and exhaust port	Turbulent model	Standard $k-\varepsilon$	Realizable $k-\varepsilon$
	Gas/LLC property	Constant	Variable with temperature
Combustion chamber	Boundary condition	Empirical value	Combustion CFD

3.1.1. Thermal boundary conditions of water jacket and exhaust port

The thermal boundary conditions of the water jacket and exhaust port are the HTC and the constant value boundary temperature obtained by CFD. The HTC of the water jacket was calculated by performing steady-flow analysis. The HTC of the exhaust port was obtained by performing unsteady-flow analysis for pulsating flow, using the average of the HTC at each crank angle.

Both the water jacket and the exhaust port are complex passages, so reproduction of separation and reattachment phenomena can be expected to enhance the HTC analysis accuracy. A more appropriate turbulent model was therefore investigated. The reproducibility of separation and reattachment was verified using the back-step flow model⁽³⁾,⁽⁴⁾ often referenced for accuracy verification of turbulent models [Fig. 3(a)]. The turbulent model was changed from the previous standard $k-\varepsilon$ model to a realizable $k-\varepsilon$ model⁽⁵⁾. This increased the separation and reattachment flow analysis accuracy, which enhanced the reproducibility of the pressure coefficient distribution on the wall surface [Fig. 3(b)].

Furthermore, in thermal analysis using the exhaust port HTC calculated by the previous method, the temperatures inside the exhaust port tend to be higher than measurements, so temperature adjustment was performed by multiplying with a coefficient of approximately 0.6 to 0.8. It is thought that because the physical property values of exhaust gas vary greatly at different temperatures, use of constant exhaust gas physical property values when calculating the HTC increased the HTC deviation. Thus, a new HTC calculation method was developed that uses physical property values that are functions of temperature.

Figure 4 shows the verification results of thermal analysis accuracy due to changing the turbulent model and revising the HTC calculation method. For comparison purposes, the exhaust port HTC uses the CFD calculation results without multiplication by a coefficient. Thermal analysis using thermal boundary conditions obtained by the new method achieved a prediction accuracy of $\pm 10^\circ\text{C}$ compared to measurements.

3.1.2. Calculation of thermal boundary conditions of combustion chamber by combustion analysis

The previous method reproduced the thermal distribution by segmenting the thermal boundary conditions of the combustion chamber based on a heuristic, assuming a constant boundary temperature, and weighting the HTC. However, this method calls for adjustment by weighting the HTC compared to measurement results, which presumes that temperature measurement test results are available. In addition, recent adoption of direct injection and turbocharging is resulting in increasingly complex combustion modes and combustion chamber shapes that make temperature verification by the previous method a challenge. Therefore, this research investigated a method capable of accurately calculating combustion chamber temperatures in the design phase.

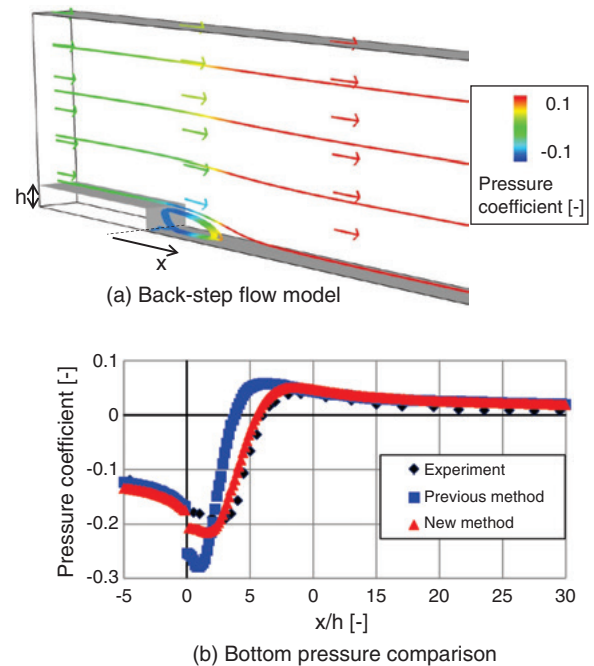


Fig. 3 Turbulent model verification in back-step flow

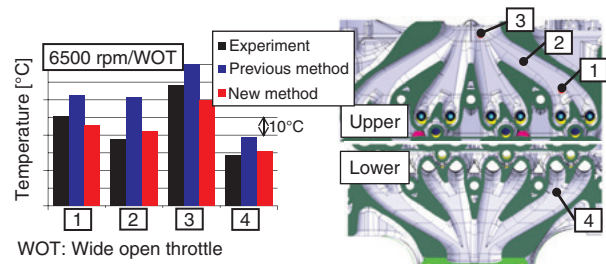


Fig. 4 Temperature comparison on exhaust port surface between experiment and CAE

CFD-based combustion analysis was used to understand the distribution of thermal boundary conditions in the combustion chamber. The boundary temperature and HTC at each crank angle were then extracted from the combustion analysis results and averaged to obtain the thermal boundary conditions. Figure 5 shows the thermal boundary conditions obtained by combustion analysis. The heat flux q from the flame to the combustion chamber is obtained by Eq. (1).

$$q = h \cdot \Delta T \text{ [W/m}^2\text{]} \quad (1)$$

h : HTC (heat transfer ratio) [W/m²K]

ΔT : Difference between combustion chamber wall surface temperature and boundary temperature [K]

The heat flux was previously expressed by weighting the HTC. However, the combustion analysis results show that both the boundary temperature and HTC have distributions, and it was determined that it would be challenging to express the heat flux with one or the other using a fixed value as in the previous method. Therefore, combustion analysis was performed to map both the boundary temperature and HTC onto the thermal analysis model.

The calculated temperatures were compared with engine test results to verify the accuracy of the new method (Fig. 6). Whereas the previous method based on a heuristic prior

to verification with actual engine temperature measurement results had error of up to 20°C compared to measurements, the new method achieved the target accuracy of $\pm 10^\circ\text{C}$.

3.2. Development of Material Model

The material model used in previous TMF analysis was a linear material model that used the Young's modulus, Poisson's ratio, and linear expansion coefficient at each temperature, and did not take into account plasticity or overaging. However, in order to evaluate the strength of high-temperature locations that are constrained by surrounding areas and undergo plastic deformation, such as the combustion chamber, it is necessary to accurately understand the nonlinear behavior of the material. The material model handled by CAE-based TMF evaluation uses a Young's modulus that takes into account temperature dependency, the stress-strain characteristics (S-S curve), and the linear expansion coefficient. Of these, the S-S curve is the key parameter for predicting the nonlinear behavior of the material. This research modeled the kinematic hardening rule⁽⁶⁾ and hardness overaging^{(7), (8)} in order to reproduce the nonlinear behavior of hot materials. In addition, the strength of the aging hardened alloy used in the cylinder head drops due to overaging as a result of constantly being at high temperature. However, rig tests showed that reduced energy loss due to shrinkage of the hysteresis loop resulted in a longer life, so in order to perform stricter strength evaluation, it was decided to not take creep into account.

3.2.1. Application of kinematic hardening rule

The S-S curve obtained by repeating the thermal cycle varies according to the hardening rule. Hardening rules include the isotropic hardening rule and the kinematic hardening rule, and this research compared both rules. A TMF test was used to verify the CAE accuracy with regards to the S-S curve when a thermal load is repeatedly applied. This test uses a mechanical load and thermal cycle to apply a compound load to a material unit test piece, and has been used to evaluate the TMF resistance of various parts. Figure 7 shows an outline of the test. A model was created by carving out only the evaluation location as shown in Fig. 7(b) in order to determine the correlation between CAE and measurements. The test mode [Fig. 7(c)] repeated heating and cooling between 50°C and 250°C under load conditions that produce compressive strain during the heating period. Figure 8 shows the results of comparing the hysteresis loops measured during the test and the hysteresis loops obtained by CAE-based analysis for an input mechanical strain of $\pm 0.3\%$. These results show that compared to the nonlinear behavior calculated by the isotropic hardening rule shown in Fig. 8(a), use of the kinematic hardening rule shown in Fig. 8(b) more closely matches the TMF test results. As a result of taking into account the kinematic hardening rule, the Bauschinger effect⁽⁹⁾ could be simulated from the 1st cycle of the hysteresis loop, and the maximum stress of the 3rd cycle had error of 3% or less compared

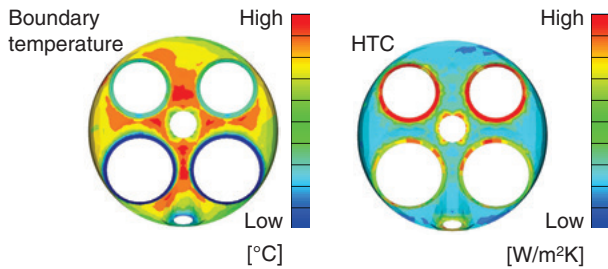


Fig. 5 Boundary conditions calculated by CFD

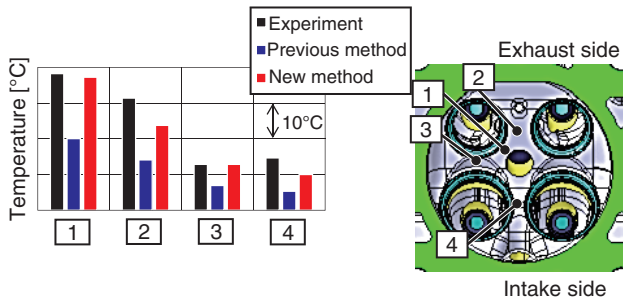


Fig. 6 Temperature comparison on combustion chamber surface between experiment and CAE

to TMF test results. In contrast, when using the isotropic hardening rule, the maximum stress of the 3rd cycle had error of up to 30% compared to TMF test results. Therefore, the kinematic hardening rule was used.

3.2.2. Consideration of hardness overaging

The aluminum alloy used in the cylinder head is strengthened by heat treatment, so extended exposure to a high-temperature environment produces hardness overaging, which reduces the material strength and also changes the S-S curve. Figure 9 shows the hardness overaging characteristics of the cylinder head material depending on the time exposed to each temperature environment. In

particular, as the combustion chamber and other parts are exposed to high temperatures during engine operation, CAE-based strength evaluation should take into account hardness overaging. Hardness overaging is a phenomenon that occurs continuously over time, so it is a challenge to express by elasto-plastic analysis. This research divided the degree of overaging into the four levels of A (HRB 60 or more), B (HRB 40 to 60), C (HRB 20 to 40), and D (HRB 20 or less) for use as the degree of hardness overaging, and created an S-S curve for each degree of overaging as shown in Fig. 10. Figure 10(a) shows the S-S curve for each hardness of HRB 70, 50, 30, and 10. These four hardness levels were respectively assigned as the typical hardness for each degree of overaging A to D. The S-S curve for each hardness was measured at various temperatures, and a temperature-specific S-S curve for each degree of hardness overaging was created as shown in Fig. 10(b) and applied to thermal fatigue analysis.

The degree of overaging changes according to the temperature and time, so the degree of overaging of the analysis object varies according to the engine operating conditions. The temperature is obtained by thermal analysis,

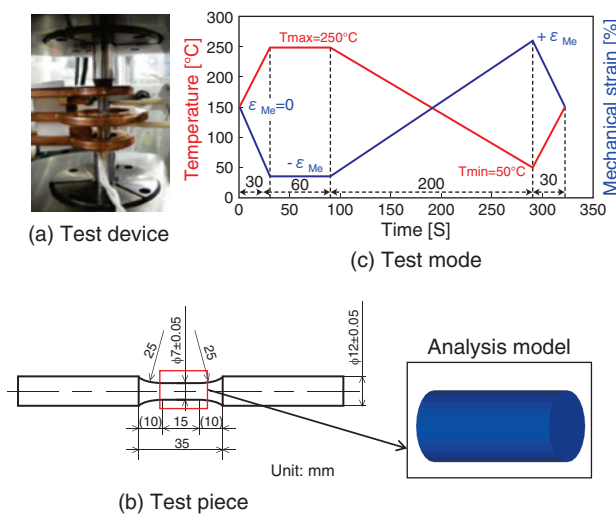


Fig. 7 TMF test outline and analysis model

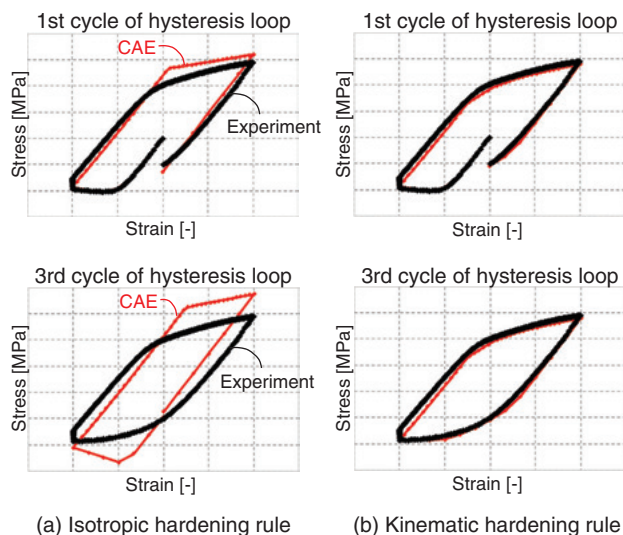


Fig. 8 Hysteresis loop due to hardening rule

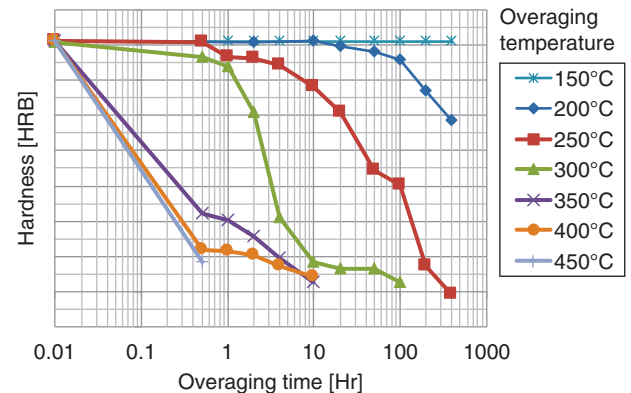


Fig. 9 Hardness overaging characteristics of aluminum alloy

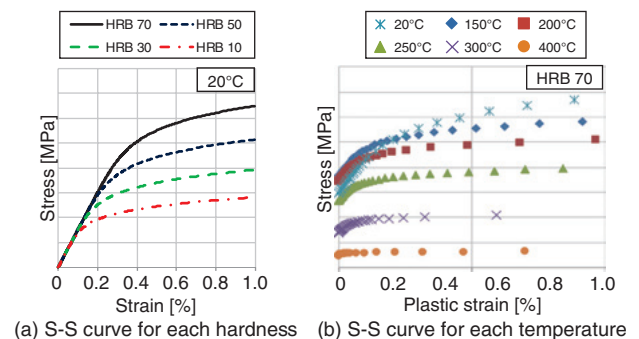


Fig. 10 Creation of S-S curve for each degree of overaging

and the overaging time is determined by the assumed operating mode, so the degree of overaging of each part of the analysis object can be understood from these factors. Strength analysis that takes into account thermal overaging of the material can be realized by performing elasto-plastic analysis separately using the S-S curves for each degree of overaging of each part.

3.3. Connection Model

It has been challenging to predict the strain around the bolts that fix the turbocharger on the cylinder head using conventional CAE. This is thought to be an issue when modeling the bolts that transfer the external constraint by the turbocharger to the cylinder head.

Bolt bending due to thermal expansion of the turbocharger has a large impact on the turbocharger connecting areas. The previous thread connection model expressed the FEM model of the connecting areas between the bolt and the internal threads by merging nodes [Fig. 11(a)]. However, in connecting areas with merged nodes, the model was unable to reproduce the behavior where the bolt separates from the internal thread when bending force is applied to the bolt, so the impact of external constraint could not be properly expressed. Therefore, the multi-point constraint (MPC) shown in Fig. 11(b) was used to track the bolt in the thrust direction, and the force transmission due to the bolt axial force and bending was reproduced by defining contact in the radial direction. A verification test was performed using a test piece that simulates the area around the cylinder head exhaust flange to verify the plausibility of the bolt connection model (Fig. 12). This test fixed an aluminum test piece that simulates the cylinder head to a surface plate, connected a steel plate that simulates the exhaust flange with a bolt, and applied a load to the steel plate. Fifteen strain gauges were installed continuously around the internal threads, and the strain distribution around the bolt connecting area was measured. A load of 20 kN was applied in the vertical direction using a Shimadzu Autograph precision universal tester. The rig test results and CAE results are shown in the lower part of Fig. 12, and confirmed that the analysis accuracy in all areas was enhanced compared to the previous method.

Next, an actual engine was used to verify the analysis accuracy of the thermal strain produced by the external

constraint of the exhaust system around the bolt connecting area. This test used a single-cylinder cylinder head (Fig. 13). The connection model using MPC provided better strain analysis accuracy compared to the previous bolt connection model with merged nodes, and achieved an accuracy of $\pm 10\%$ compared to an actual engine.

These enhancements in analysis accuracy were applied to analysis of cylinder head TMF strength, and the results reconfirmed the combustion chamber as the location with the greatest thermal strain.

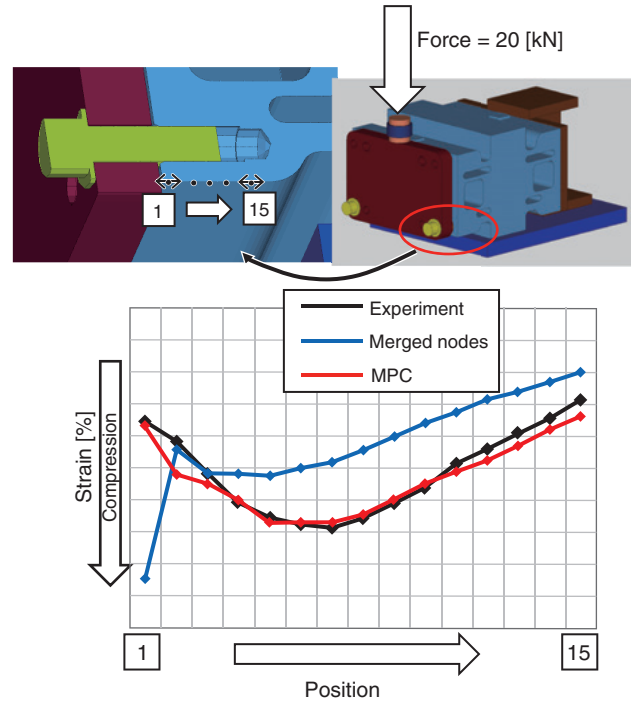


Fig. 12 Advanced rigid body connection model

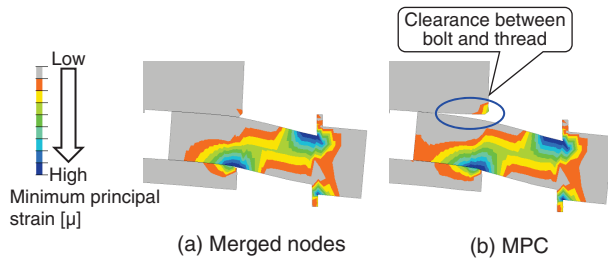


Fig. 11 Rigid body connection model

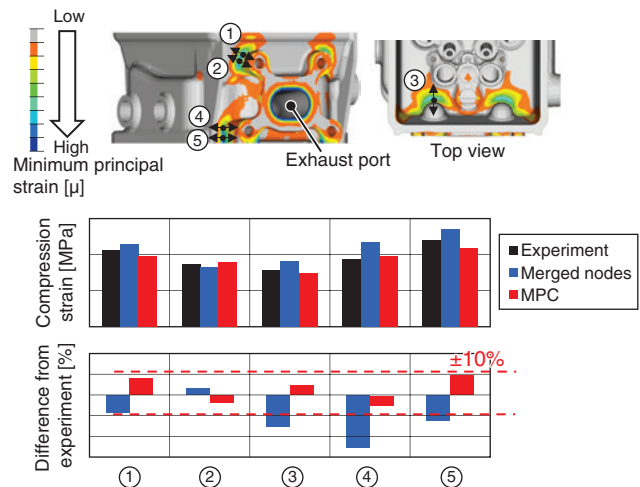


Fig. 13 Verification on thread connecting area with exhaust port

4. Comparison of Prediction Technology and Test Results

A concentrated durability test using an actual engine was performed focusing on the combustion chamber, which is the cylinder head location with the highest thermal strain, in order to confirm the accuracy of TMF evaluation using the new method. Table 2 shows the engine specifications used in this test. This durability test increased the thermal strain by adjusting the boost pressure in the WOT condition to increase the power and by controlling the coolant flow rate. In addition, the durability test mode was set by calculating the holding time during which the thermal strain is generally saturated from the results of temperature measurement tests during mode operation (Fig. 14).

Figure 15 shows the TMF analysis results in this durability test mode plotted on an S-N diagram. The change from the previous linear material model (blue plots in Fig. 15) to a nonlinear material model (red plots in Fig.

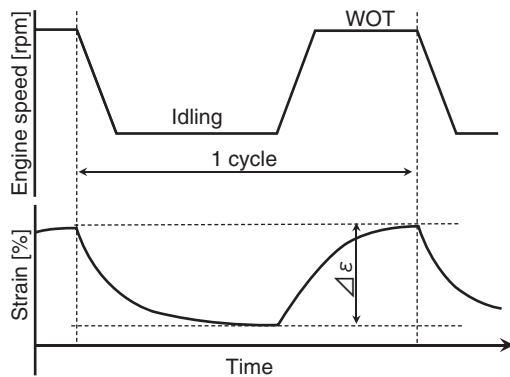


Fig. 14 Durability test cycle

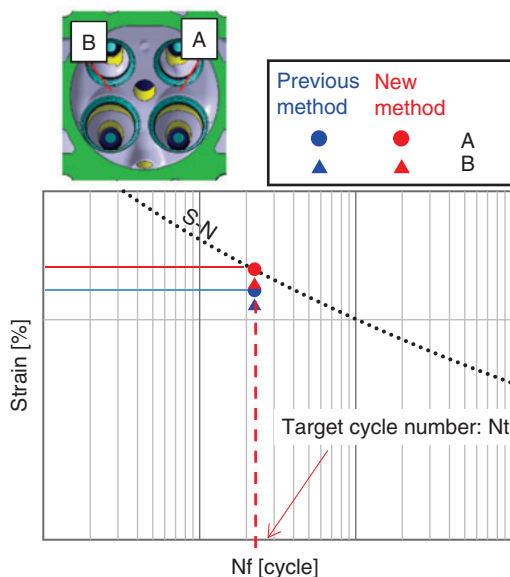


Fig. 15 Effect of new method on S-N diagram

Table 2 Engine specifications

Engine type	L4 DOHC VTEC Turbo
Bore × Stroke [mm]	73.0 × 89.4
Displacement [cm ³]	1496
Compression ratio [-]	10.6
Maximum power [kW/rpm]	110/5500
Maximum torque [Nm/rpm]	203/1600-5000
Fuel type	Gasoline (RON91)

15) resulted in higher calculated thermal strain. This is because the thermal strain analysis accuracy was enhanced by reproducing the kinematic hardening rule and hardness overaging due exposure to high temperature and high strain.

A durability test was performed with the number of durability test cycles N_t determined so that cracks occurred only at location A, which has higher thermal strain according to the S-N diagram. Figure 16 shows the durability test results. The results confirmed that cracks occurred only at location A and not at location B.

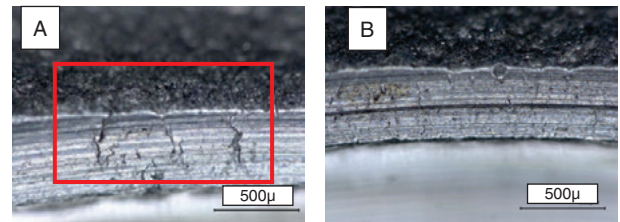


Fig. 16 Combustion chamber after durability test

5. Conclusion

A method was established that enables model-based prediction of the cylinder head TMF strength in the design phase. The developed technologies are summarized below.

- (1) Technology was developed that can predict cylinder head temperatures with an accuracy of $\pm 10^\circ\text{C}$, without verification using an actual engine. This was achieved by revising the CFD analysis method for the water jacket and exhaust port, and using combustion analysis to obtain the thermal boundary conditions.
- (2) A material model was developed that takes into account the hardening rule and hardness overaging, which enabled reproduction of material behavior up to high heat loads.
- (3) The bolt connection model was revised, and a model was developed that accurately reproduces the strain distribution of an actual engine.

References

- (1) Udoguchi, T.: Thermal Cycle Fatigue, Journal of the Japan Welding Society, Vol. 37, No. 6, p. 573-585, (1968) (in Japanese)
- (2) Toda, H., Kobayashi, M.: Thermal fatigue fracture of materials, Journal of Japan Institute of Light Metals, Vol. 59, No. 6, p. 312-319, (2009) (in Japanese)
- (3) https://turbmodels.larc.nasa.gov/backstep_val.html
- (4) Lee, C. Y., Park, S. H., Cho, S., Lee, I.: Delayed-Detached Eddy Simulation for Optical Path Control on Backward-Facing Step Flow, Seventh International Conference on Computational Fluid Dynamics, ICCFD7-2701, (2012)
- (5) Shih, T-H., Liou, W. W., Shabbir, A., Yang, Z., Zhu, J.: A New k- ε Eddy Viscosity Model for High Reynolds Number Turbulent Flows – Model Development and Validation, Computers & Fluids, Vol. 24, Issue 3, p. 227-238, (1995)
- (6) Lee, M. G., Kim, D., Chung, K., Youn, J. R., Kang, T. J.: Combined Isotropic-Kinematic Hardening Laws with Anisotropic Back-stress Evolution for Orthotropic Fiber-Reinforced Composites, Polymers & Polymer Composites, Vol.12, No.3, (2004)
- (7) Toda, H., Fukunaga, T., Kobayashi, M., Uesugi, K., Kobayashi, T., Yamada, T., Ooba, Y., Kakizaki, M.: Thermo-mechanical fatigue property of a surface-hardened AC4CH cast aluminum alloy, Journal of Japan Institute of Light Metals, Vol. 58, No. 6, p. 236-241, (2008) (in Japanese)
- (8) Moizumi, K., Tezuka, H., Sato, T.: Influence of precipitation hardening and eutectic Si microstructures on thermal fatigue life of Al-Si-Mg-Cu based cast alloys, Journal of Japan Institute of Light Metals, Vol. 54, No. 6, p. 205-210, (2004) (in Japanese)
- (9) White, C. S.: A Combined Isotropic-Kinematic Hardening Model for Large Deformation Metal Plasticity, Final Report, MTL TR 88-46, U.S. Army Material Technology Laboratory, (1988)

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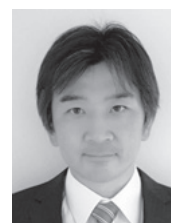
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Establishing Design Method for Mold Temperature Control Piping of Rapid Heat and Cool Molding Used for Fabrication of High Strength Exterior Resin Parts

Daisuke SUGIO¹

ABSTRACT

Assuming the mass production of good appearance and high-strength plastic parts to be used on motorcycles, we studied the Heat and Cool (H&C) molding method to fabricate low-cost glass fiber reinforced polypropylene parts. In order to satisfy both molding cycle time and mold surface temperature uniformity requirements when temperature control piping are laid along a three-dimensional surface of a mold cavity, we used actual measurements made on test pieces to systematically define basic specifications for the piping arrangement. In addition, to deal with portions where an ideal piping design cannot be made, we developed a heat transfer simulation method that can effectively determine the surface temperature distribution of a mold cavity. With this, for a system that uses saturated steam from a boiler as the heating medium, it was indicated that a simple method to assume hot water for the heating medium can in fact be quite useful.

1. Introduction

Motorcycles are widely used in a number of Asian countries, including China, India, Thailand, Indonesia, and Vietnam, where they make up large consumer markets that support real economic growth in these very areas. Partly as a result of public transportation infrastructure not yet fully developed in these regions, motorcycles are widely used as an important means of transportation to carry both people and goods. Alongside the maturity of the market in recent years, a good appearance is becoming more desired than ever, in addition to utility needs. Accordingly, as exterior parts of motorcycles are required to have higher productivity and even greater freedom in terms of shaping, plastic parts produced by injection molding are now widely in use.

For example, rear grab rails and carriers that once used steel piping or casted aluminum to satisfy required strengths are now being produced using glass fiber reinforced plastics (GFRP). GFRP is a plastic material of which strength and elasticity are reinforced by glass fibers embedded in the base resin. While glass fibers are excellent reinforcing materials, they are prone to exposure on the surface of a molded product. Hence, it is generally not an easy task to achieve a good appearance in such products. As a result,

various efforts have been made including changing resin properties and surface coatings as a means to hide the fibers. For example, using glass fiber reinforced polyamide (PAGF) can suppress fiber exposure through altering properties of said resin.

However, optimizing the property of resin leads to higher material costs, while applying coating causes a productivity issue due to additional processes. Therefore, we opted to study the feasibility of using polypropylene as a lower cost base resin material, in order to evaluate the possibility of glass fiber reinforced polypropylene (PPGF). Generally, conventional injection molding with PPGF cannot satisfy the appearance normally required of the product. This is due to fast crystallization of PPGF that quickly solidifies the surface layer with the glass fibers exposed (see Fig. 1). To address the issue, we selected the Heat and Cool (H&C) molding method, with the intention of aiming to establish a mass production process that can satisfy external appearance requirements without the need to apply coating.

With H&C molding, a heating medium such as hot steam is used to heat the mold and then water or another cooling medium is used to cool it down during the same molding cycle. As the molding mold is heated to a desired temperature, the resin is injected in the mold to perform

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the molding. As molding is made at a high temperature of the mold, resin solidification is delayed and subsequently the characteristic of transferring the shape of the cavity is enhanced to produce mold products of excellent appearance that are free from exposed fibers or weld lines. However, by decreasing the heating and cooling cycle time in order to increase productivity, the unevenness of the mold surface temperature increases, thus leading to a poor uniformity in the external appearance, in addition to molding defects such as gas burning and burrs. As such, the issue right here was how to establish a design method that would effectively minimize any unevenness in mold surface temperatures.

In this study, to accurately estimate the mold surface temperature, we established a method using actual measurements from test pieces⁽¹⁾ and a method using simulation⁽²⁾, which are described in Chapters 4 and 5, respectively. In addition, Chapter 6 describes the effectiveness of these methods that were established using an example case of a rear grab rail.

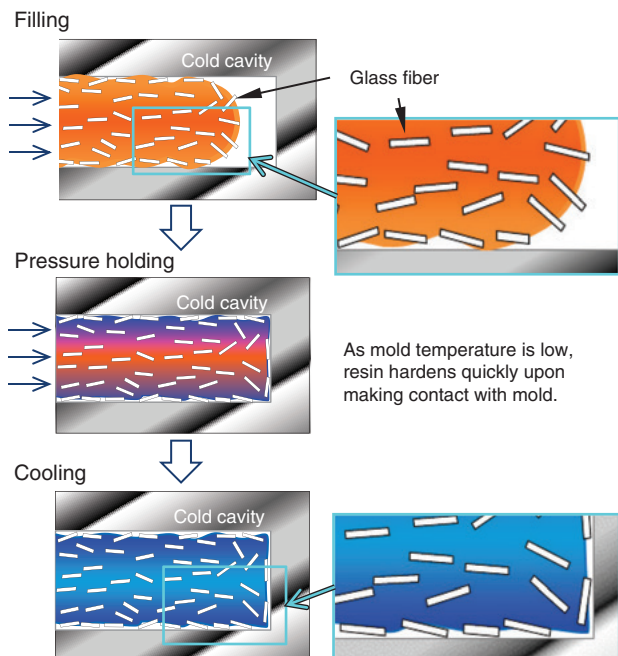


Fig. 1 Exposure of glass fibers on surface of glass fiber reinforced plastic molded by conventional injection molding

2. Setting Target Values for Molding Cycle Time and Mold Surface Temperature

While we could use induction heating, a high pressure boiler or another special mold heating system to build an efficient H&C molding system, we opted to use saturated steam from ordinary boilers that are commonly used in our

factories as the heating medium in consideration of easy deployment at other production sites. Also, according to productivity requirements, we set a molding cycle time of 120 seconds as the target value.

To evaluate the range of cavity surface temperature that can help ensure a good appearance in the product, we built a mold to produce thick test pieces that can simulate rear grab rails. The two graphs in Fig. 2 show how the surface gloss and weight of the molded PPGF part change when the cavity surface temperature is varied. The surface gloss values shown are the values measured at an incident angle of 60 degrees according to ASTM D523. This shows that a sufficient surface gloss cannot be achieved when the surface temperature is excessively low. Also, an excessive surface temperature increases contraction and reduces weight, which cause burrs and insufficient strength in the product. Accordingly, as the molding process for PPGF is required to help prevent exposure of glass fibers and to satisfy other molding quality requirements, we set a target temperature range of 120 to 145°C within which the temperature has to be controlled for the entire cavity surface.

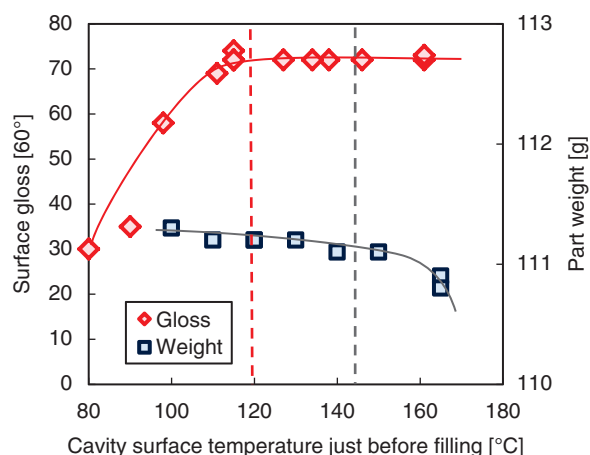


Fig. 2 Surface glossiness and weight of molded part in relation to different cavity temperatures

3. Piping Design for Three-dimensional Shaping

In order to address the issue of different surface temperatures of the mold cavity when the mold is heated, it is necessary to maintain an equal distance between the temperature control piping and the cavity surface (see Fig. 3). To achieve this, we used a “milled groove channel method” in this study to form temperature control piping by joining two blocks of molds instead of using a monolithic mold (see Fig. 4)⁽³⁾. As using this method can form the piping along the cavity surface, while maintaining an equal

distance between the piping and the cavity surface, it was felt we would be able to reduce any time lag in the surface temperature rise when the mold is heated. Also, to better track the cavity surface, we tried to minimize the diameter of the channel piping as far as possible.

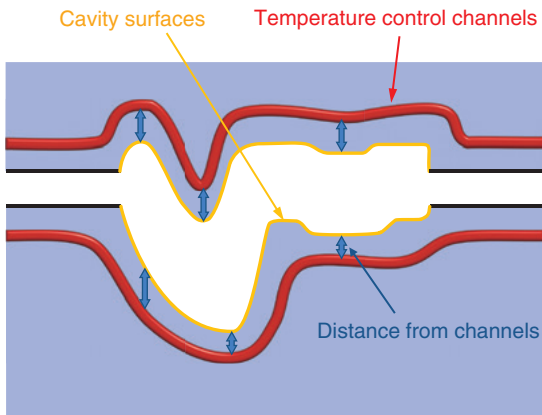


Fig. 3 Cross section of ideal Heat and Cool channels

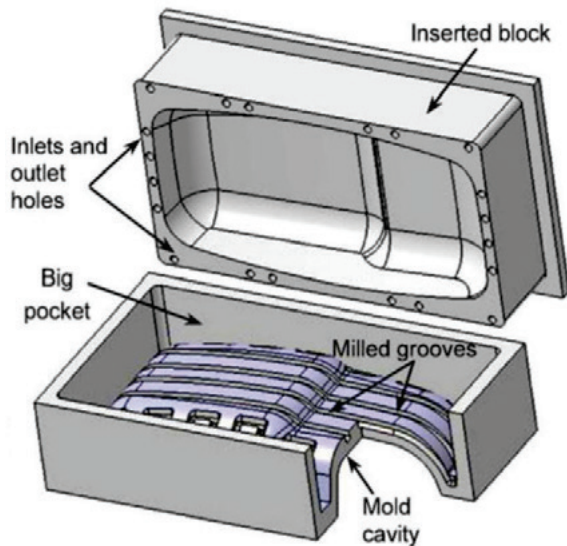


Fig. 4 Example of piping for cooling fabricated by Milled Groove Channel method

4. Defining Specifications for Mold Temperature Control System Based on Test Piece Measurements

To reduce the heating and cooling times, it is necessary to place the temperature control piping close to the cavity surface. However, if the piping is placed too close to said

surface, it may cause some temperature differences between the surface points with and without a nearby heating piping. Therefore, it is necessary to define the piping specifications in such a way that the heating and cooling can be efficiently performed, while keeping the surface temperature difference within an acceptable limit. In this, there were four major parameters – diameter (D), depth (H), pitch (P), and length (L) of the channel (see Fig. 5). By studying the behavior of these four parameters through actual measurements on a structural model that simulates the mold, we were able to establish a method to define mold temperature control specifications. These details are described in the following sections.

4.1. Channel Diameter

Although a smaller channel diameter is ideal for laying the channels along a three-dimensional surface of the mold cavity, an excessively small diameter lowers the temperature of the heating medium. Figure 6 shows how the temperature difference between the inlet and outlet ports of the test piece mold changed according to the pipe (channel) diameter in the temperature rise test when hot steam of 180°C was in use. As can be seen from the graph, the temperature difference between the inlet and outlet ports significantly increases when the channel diameter is less than 5 mm. At the same time, the channel diameter should be as small as possible as previously discussed. Accordingly, we selected a channel diameter of 5 mm for this study.

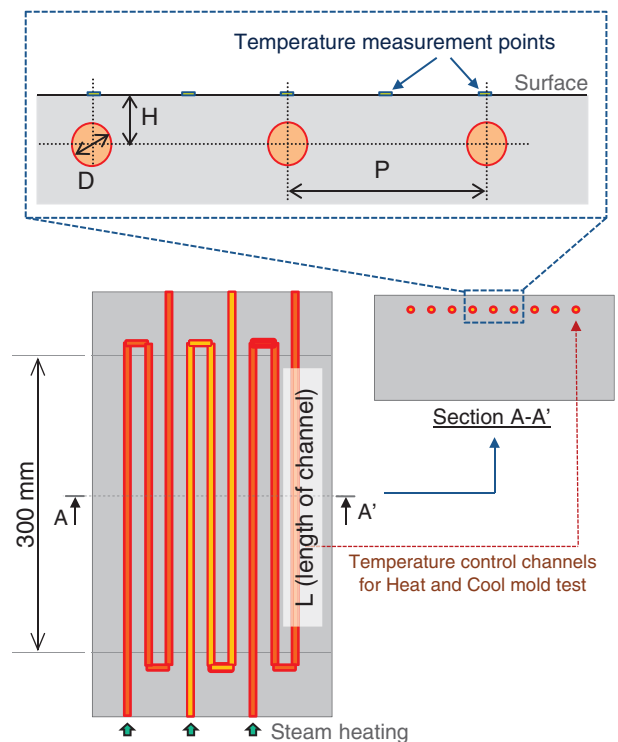


Fig. 5 Structure model of piping system for analytical investigation

4.2. Channel Depth

Figure 7 shows the relationship between the channel depth and the time required to raise the cavity surface temperature of the test piece mold to 120°C with the temperature control piping. It can be clearly seen that the smaller the depth, the shorter the heating time required. However, it should also be noted that an excessively small depth can reduce the durability of the mold. Therefore, aiming for a balance between mold durability and productivity requirements, we determined a channel depth of 10 mm for this study.

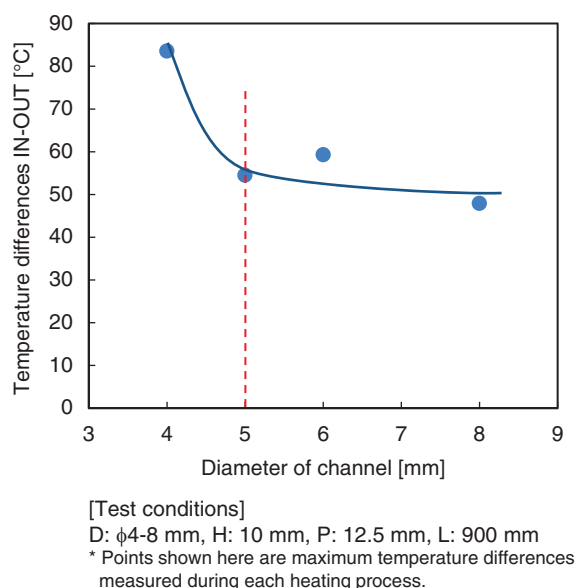


Fig. 6 Temperature differences between inlet and outlet

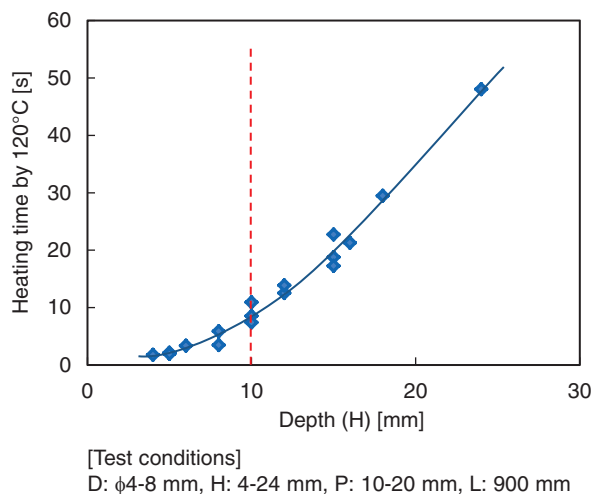


Fig. 7 Heating times to raise temperature from 40 to 120°C with different pipe diameters and pitches

4.3. Channel Length

As in the case of channel depth, we tested the temperature rise of the test piece mold by using hot steam at 180°C to determine how it was affected by channel length. While a longer channel length can ease the piping layout, the resulting temperature drop of the heating medium makes it challenging to sufficiently raise the temperature in areas near the outlet port. To find out the effects of this, we measured the cavity surface temperature of the test piece mold along the piping route. Figure 8 shows the results from this measurement. The temperature plot in this graph shows the temperature measured at each measuring point when the highest temperature among these points has reached 130°C. The horizontal axis shows the length along the channel measured from the inlet port. From the graph, it can be seen that the surface temperature starts to drop after exceeding a channel length of 1,000 mm. This suggests that the temperature of the steam in the channel is lowered in these particular areas. Based on this test result and considering that the channel length should be as long as possible as previously noted, we determined that the channel length should be within 1,000 mm.

4.4. Depth and Pitch of Channel

Following to above, we continued with the study of the depth and pitch of the channel. As shown in Fig. 9, red areas just above the channel can be easily heated up, while it is more challenging to raise the temperature in blue intermediate areas. Also, while reducing the depth can quicken the temperature rise, it increases the temperature difference between the red and blue areas. Furthermore, the resulting small thickness of the mold between the channel and cavity surface reduces the strength of the mold. At the same time, while reducing the pitch can reduce temperature differences between red and blue areas, it also reduces

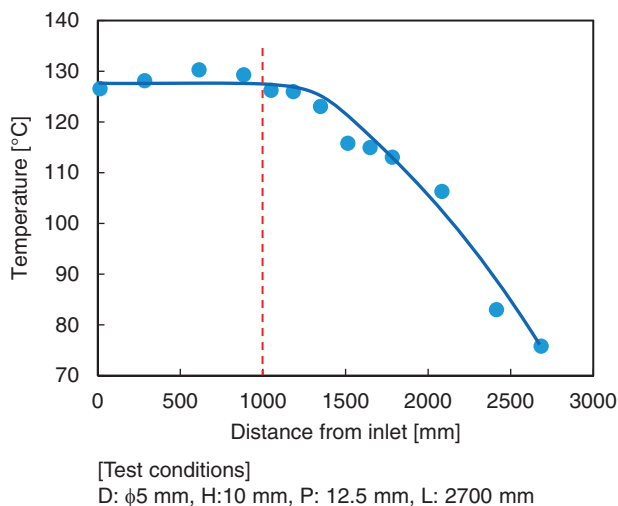


Fig. 8 Temperatures measured along channel on surface of structure model

the strength of the mold, as a direct result of the smaller distance between channels. Hence, the optimum values of depth and pitch are determined relative to each other, and its effect is presented as the temperature difference between red and blue areas, as shown in Fig. 9. We compared the temperature difference with the ratio of depth to pitch (H/P) as shown in Fig. 10. It can be seen from the graph that H/P of less than 0.5 significantly increases the temperature difference. Also, as increasing H/P decreases the temperature difference, we can uniformly raise the surface temperature to a better extent. On the other hand, the depth should be kept as short as possible in order to reduce heating times. According to the above considerations, we decided to set the ratio of depth to pitch in a range of 0.5 to 1.0. Also, as the depth was set to 10 mm based on the durability limitation of the mold as previously noted, we decided to set the pitch in the range of 10 to 20 mm.

4.5. Basic Piping Specifications Determined

As discussed above, after organizing ideas on how to minimize the heating time and temperature differences on the mold cavity surface through proper specifications for the temperature control piping, we determined the following parameters based on actual measurements: $D = 5$ mm, $H = 10$ mm, $L = 1,000$ mm or less and $P = 10$ to 20 mm (H/P of 0.5 to 1.0).

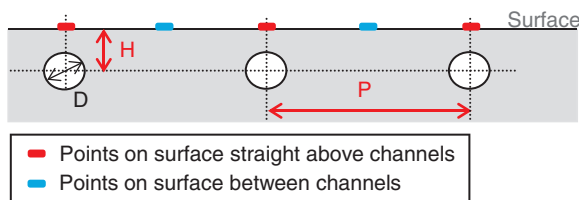
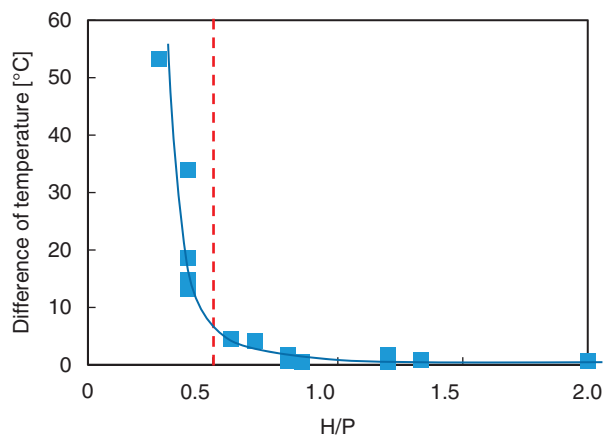


Fig. 9 Temperature measurement points on surface of structure model



[Test conditions]
H: 5-15 mm, P: 7.5-17.5 mm, $\phi 5$ mm, L: 900 mm

Fig. 10 Variation in deviations of cavity temperature in reference to ratio between H and P (H/P)

5. Simulation to Estimate Temperature Distribution on Cavity Surface

5.1. Necessity for Simulation Technology

For an actual mold design, there are portions of the mold where the temperature control piping cannot be ideally placed using the same pitch due to certain limitations caused by the use of fasteners for assembling the mold, in addition to the specific shape of a product. As such portions may not be properly heated to a desired temperature when the mold is heated, it can cause an uneven surface temperature distribution of the cavity. Therefore, unless such portions are specially treated in design, the surface temperature of the cavity could become uneven. Figure 11 shows an example of such a case. As shown by the red line, a significant temperature difference is observed at some of the temperature control points. Figure 12 shows some portion examples where significant temperature differences can be seen. As previously mentioned, the use of fasteners for assembling the mold as well as the specific shape of an actual product makes it impractical to place the temperature control piping in the same pitch as shown in the top view of Fig. 12 (a). Moreover, as shown in the cross section of Fig. 12 (b), even though the temperature control piping seem to be placed in the same pitch according to the parameter specified as viewed from the top, there can be portions where the cavity surface is more distant from the piping than at other areas. In such cases, the desired temperature distribution cannot be obtained. As such, it may become necessary to change the shape of the piping to increase the width of the channel as shown in Fig. 12 (B2) to help increase the temperature rise at these portions.

As discussed above, implementing a special temperature control piping configuration may be required at certain places in an actual mold design. Accordingly, when such a modification has to be applied, it is necessary to evaluate whether the required temperature distribution can

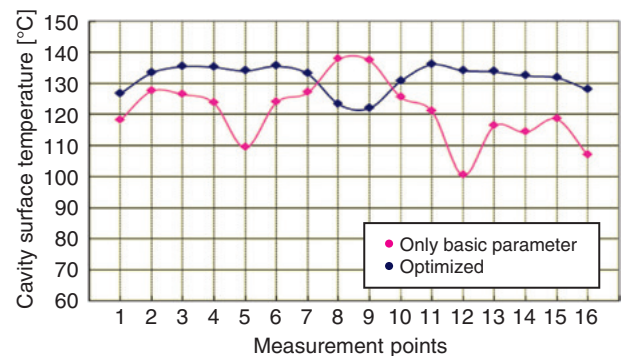


Fig. 11 Temperature distribution of cavity surface of mold for rear grab rail by optimized channel design compared with that by basic parameter channel design

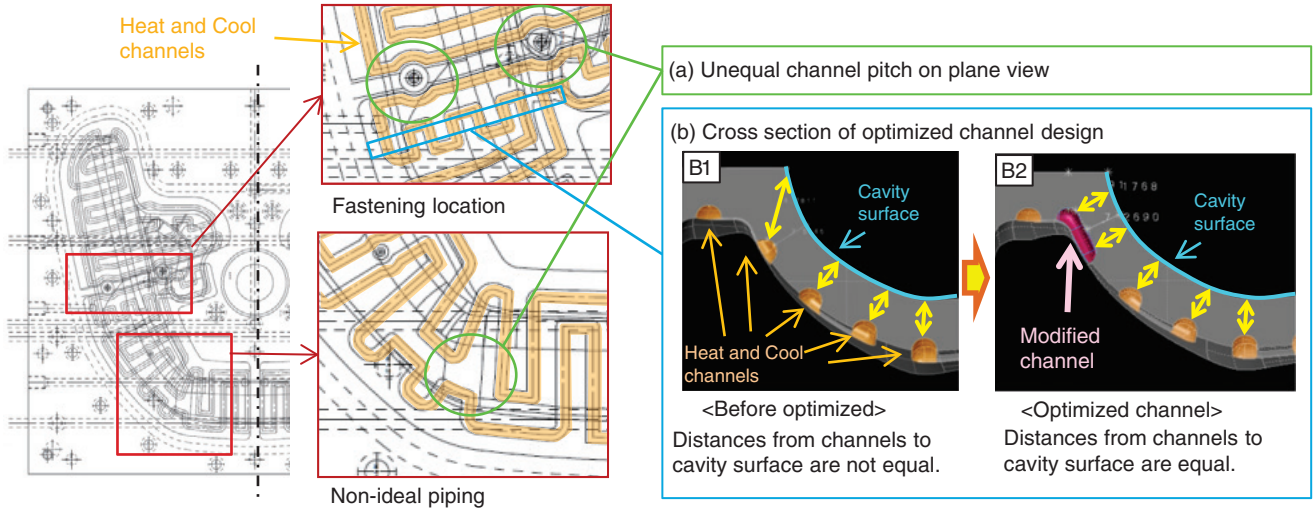


Fig. 12 Examples of non-ideal piping and cross section of optimized channel design of Heat and Cool channels

be obtained at each portion before the mold is actually manufactured. To allow for such an evaluation, we studied simulation methods that can estimate the cavity temperature distribution with the intention to make it a tool that supports a parameter-based piping design.

5.2. Simulation Method

For the temperature distribution simulation, we used 3D TIMON[®] of Toray Engineering, which is a software for plastic injection molding simulation, considering ease of use in terms of component development. We calculated the heat transfer in the temperature control piping by obtaining the Nusselt number Nu using the Gnielinski equation (1)⁽⁴⁾.

$$Nu = \frac{F_2 (Re - 1.0 \times 10^3) Pr}{1 + \frac{12.7(Pr^{2/3} - 1.0)}{1.414213562 \times F_1}}$$

$$F_1 = 3.64 \log_{10} Re - 3.28$$

$$F_2 = \frac{1}{2F_1^2}$$

$$Re = \frac{D \cdot u}{\nu} \quad Pr = \frac{\eta \cdot c_p}{k}$$

Nu : Nusselt number
 Pr : Prandtl number
 η : Viscosity
 k : Thermal conductivity
 c_p : Specific heat
 Re : Reynolds number
 D : Diameter
 u : Flow velocity
 ν : Kinematic viscosity

The hot steam used as the heating medium for the H&C molding method is saturated steam from a boiler, which is regarded as a mixture of liquid phase water and gas phase water (steam). When a heating process is initiated,

hot steam of 180°C is supplied from the boiler to the temperature control piping and is rapidly cooled by the cold water that remains in the channels. As a result, it flows as liquid-phase water in the channel (see Fig. 13).

While continuously supplying steam will replace the liquid-phase water in the channel, as the wall of the temperature control piping has a temperature lower than that of the saturated steam, the steam changes its state to liquid-phase water, as a result of condensation in the channel. Even after the liquid-phase water has been replaced by hot steam in the temperature control piping, both liquid and gas phases of water coexist in the piping, while heat transfer from the steam is made via the liquid-phase water.

From the above discussions, it was thought that the main contributor to heat transfer was water, not steam. Accordingly, we modeled the system on the assumption that hot water of 180°C is the heating medium.

5.3. Simulation Results from Test Pieces

To verify the validity of the simulation, we compared actual measurements with simulation results in relation to the surface temperature rise of the test piece. For this comparison, we modeled a test piece mold shown in Fig. 5 for the simulation. In terms of measurements, we supplied hot steam to the test piece mold as in an actual H&C molding process and measured the temperature change on the surface, which is regarded as the cavity surface of an actual mold. In addition, as the temperature control channel is defined by the four parameters of diameter D , depth H , pitch P , and length L , we produced a number of test pieces with different parameters. To verify the correlation between simulation results and actual measurements, we compared the way in which each parameter affects the heating time and temperature distribution on the cavity surface.

Figure 14 shows simulation results of the heating time

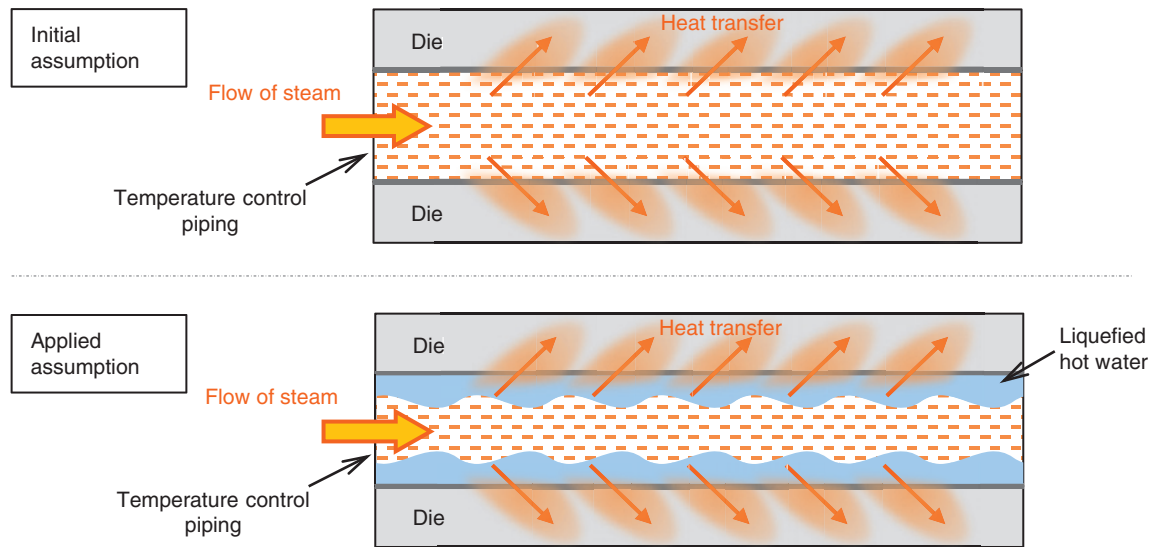


Fig. 13 Consideration on behaviors of heat medium in temperature control piping

needed to raise the surface temperature to 120°C for the values of depth H when hot water is assumed to be the heating medium. Although the heating time difference between that of the simulation and actual measurements increases at a piping depth of more than 15 mm, the simulation results are very close to the measured time at a depth H of 15 mm or less, which is mainly used in the H&C molding method. At the same time, it can be seen that the simulation assuming hot steam to be the heating medium showed a longer heating time due to a reduced amount of heat transferred to the mold, thereby failing to reproduce actual measurements. Figure 15 compares simulation results with actual measurements of surface temperature

differences at a time when the temperature reached 120°C for the depth to pitch ratio (H/P) as previously shown in Fig. 10. From this comparison, it can be seen that the temperature difference curve obtained from simulation deviates by less than 2°C from that of actual measurements for a H/P range of 0.5 to 1.0, which is the range most likely to be used in actual cases, showing that it is sufficiently accurate for temperature difference verification purposes. From these results, we determined that the simulation to assume hot water as the heating medium was practically useful for approximate analysis purposes, as it provides a robust correlation with actual measurements.

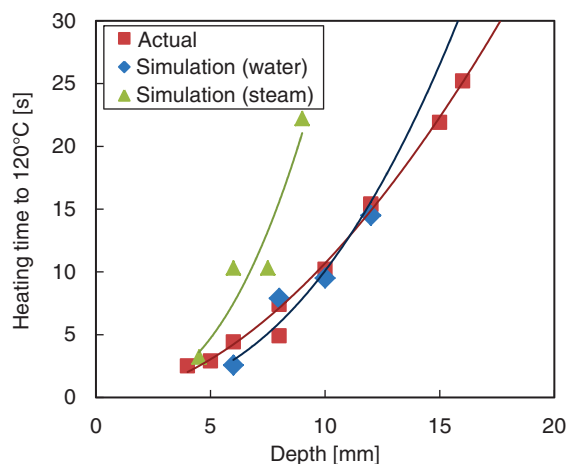


Fig. 14 Heating times to raise temperature from 40 to 120°C compared with different simulation methods

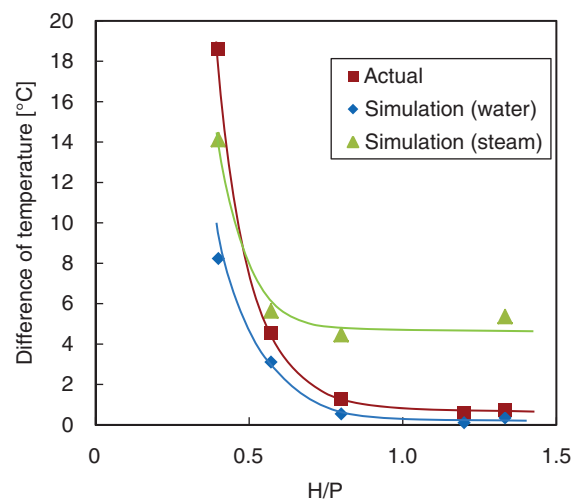


Fig. 15 Variation in deviations of cavity temperature in reference to ratio between H and P (H/P) compared with different simulation methods

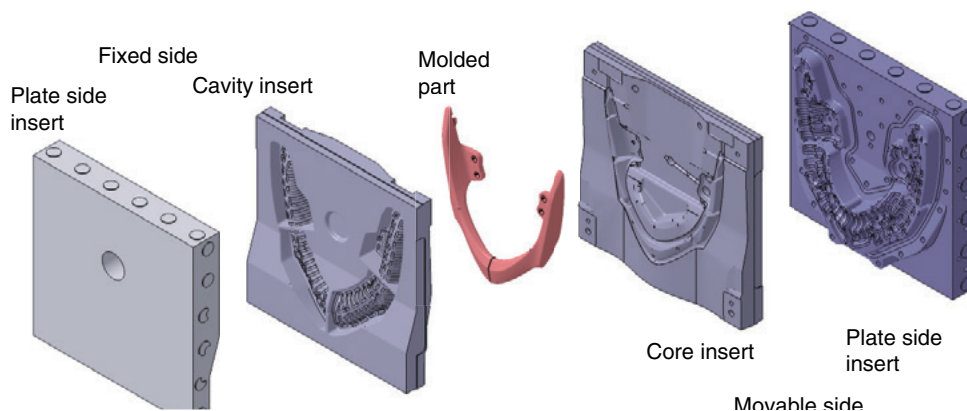


Fig. 16 Overview of mold parts fabricated by Milled Groove Channel method

6. Applying Mold Temperature Estimation Method to Rear Grab Rail

6.1. Outline of Rear Grab Rail Mold

Figures 16 and 17 show the mold structure and temperature control piping, respectively, of the mold used for H&C molding of a rear grab rail, of which piping specifications have been determined based on actual measurements with the test piece. The temperature control piping is formed in between two split mold blocks that are joined together. We provided four channels for both movable and fixed molds.

6.2. Validation

Figure 18 shows the cavity surface of the mold for a rear grab rail. The dots in the picture show measurement points. Figure 19 shows the temperature measured at

these points when the cavity surface temperature exceeds 120°C at all points. From this figure, we confirmed that all measurements were within target range of 120 to 145°C .

Also, as shown in Fig. 20, a molding cycle time of 100 seconds was achieved, which was well below the target cycle time of 120 seconds.

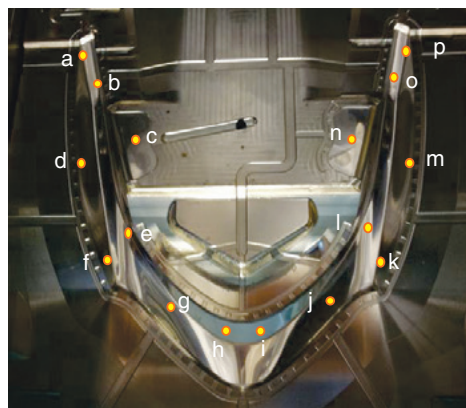


Fig. 18 Measurement points on cavity surface of mold for rear grab rail

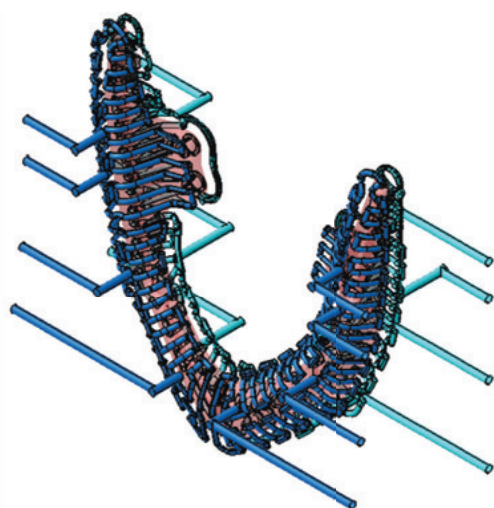


Fig. 17 Thermal control piping arrangement of Heat and Cool molding for rear grab rail mold, whose design is based on investigated specifications

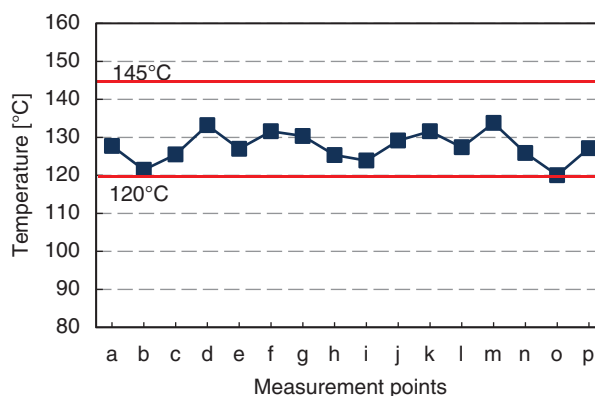


Fig. 19 Temperature distribution of cavity surface of mold for rear grab rail

Figure 21 shows external views of molded product. While exposed glass fibers are seen on the product molded with common injection molding at a mold temperature of 40°C, no such fibers can be seen on a product molded using the H&C molding method.

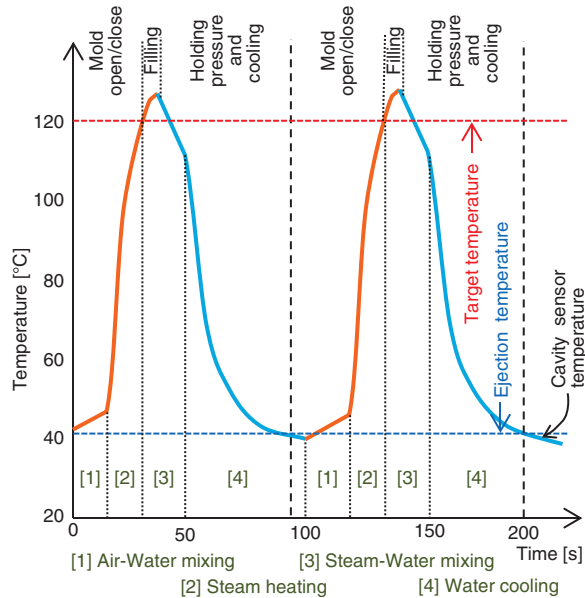
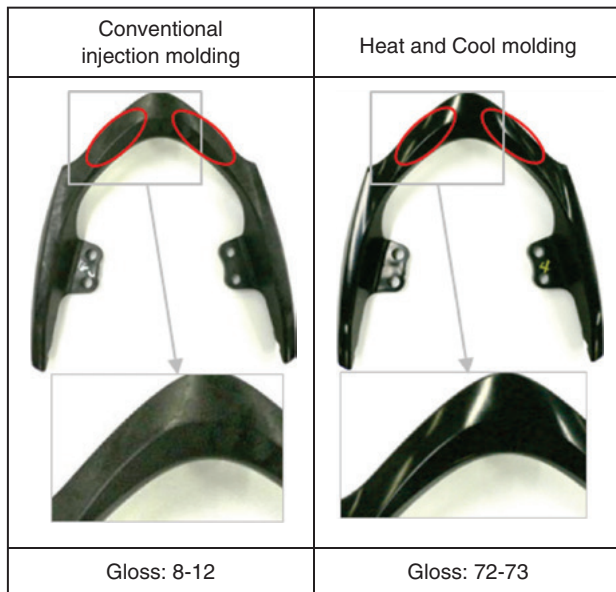


Fig. 20 Cavity temperature changes and molding cycle in actual molding processes using developed molds



○ Gloss measurement area

Fig. 21 Exterior appearances and measured gloss value of part molded by Heat and Cool method compared with part molded by conventional method

6.3. Comparison Between Simulation Results and Actual Measurements

We performed a simulation to obtain the temperature distribution on the cavity surface when a mold is heated with hot water as the heating medium. We compared the simulation results with actual measurements for the temperature distribution using the values obtained when all measurement points were at 120°C or higher. Figure 22 shows the temperature distribution obtained by the simulation together with measurement points that are used for comparison of simulation results with measurements. Figure 23 shows the temperature comparison results. It can be clearly seen from the graph that the simulation result is

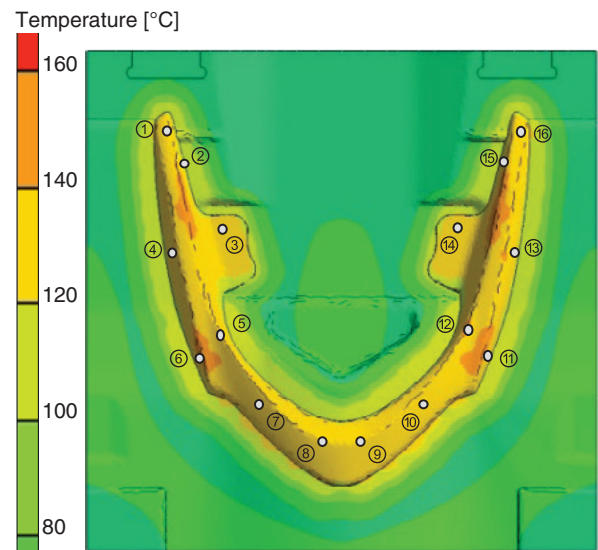


Fig. 22 Measurement points of cavity surface temperature of rear grab rail mold

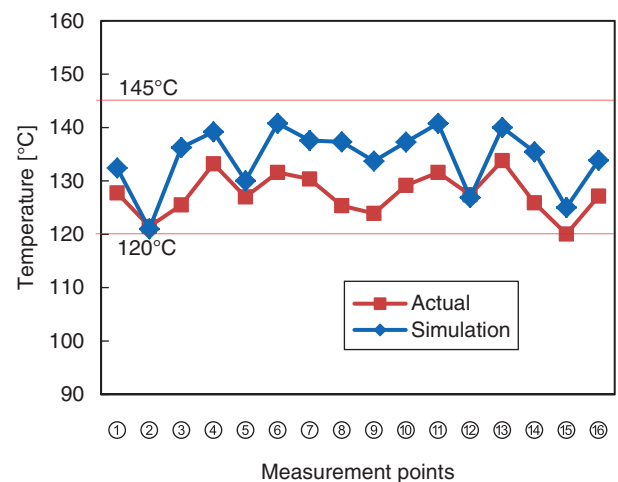


Fig. 23 Temperature distribution of cavity surface of rear grab rail mold

reproducing the trend of the actually measured temperature distribution.

When observing the entire measurement points, we can find that the simulation result has a temperature variation range of 20°C, while the actual measurement has a variation range of 15°C, which is narrower than that of the simulation. Also, separately from the above, the time it actually took to heat the mold up to 120°C was 23 seconds, which was longer than the simulation result of 10 seconds.

It can be assumed that these differences are due to the difference in the amount of heat being supplied. In the case of simulation, an ideal condition is assumed for the calculation in that hot water of 180°C is supplied to each channel at a constant flow rate. On the other hand, for an actual case it can be assumed that the amount of heat supplied is less than an ideal condition as both the temperature and flow rate may be lower than under an ideal condition. When a large size mold is used or if the boiler capacity is insufficient, it can deviate from that of the ideal condition assumed for a simulation. Accordingly, we can reasonably assume that the shorter heating time and quicker thermal diffusion of the simulation cause a trend of wider temperature distribution among measurement points than in an actual case. Therefore, although the simulation may not be useful in estimating the heating time if the capacity of the boiler is insufficient for the mold size, it is still considered to be useful in determining whether the resulting temperature distribution on the cavity surface is acceptable or not, as the simulation will act to provide a more stringent evaluation.

7. Application to Production Model

We applied the new molding method developed based on the above study results to produce rear grab rails for use on an actual motorcycle model being produced in Thailand. Figure 24 shows the actual mold used for mass production. Figure 25 shows an external view of the target motorcycle

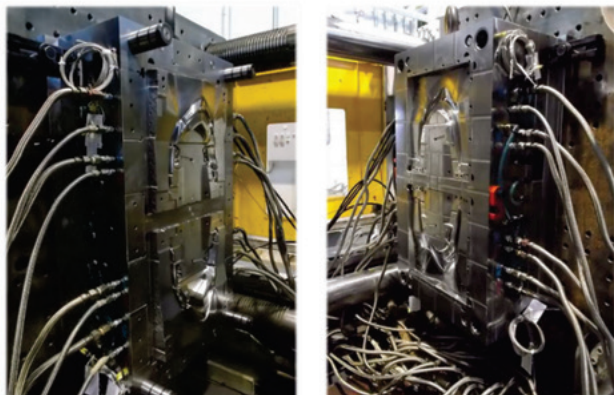


Fig. 24 Grab rail mold of Heat and Cool molding for mass production



Fig. 25 Mass production rear grab rail of Heat and Cool molding applied to Thai model

model and rear grab rail. The mold developed based on this study and used in production has achieved a production record of 400,000 shots, and an updated mold is currently in use for production.

8. Conclusion

In order to establish a technology to accurately estimate the cavity surface temperature of a mold used for H&C molding of PPGF, we established a method to use actual test piece measurements as well as a method that uses simulation. In the method using simulation, for a system that uses saturated steam from a boiler as the heating medium, we indicated that a simple method assuming hot water to be the heating medium is quite useful.

The method using test piece measurements allows for a basic design of mold temperature control piping that can evenly and quickly heat up the cavity surface.

Although the simulation method established in this study may not be useful in estimating heating time, it is still useful in terms of determining whether the resulting temperature distribution on the cavity surface is acceptable or not, as the simulation will act to provide for a more stringent evaluation. We can fully design the mold temperature control piping by additionally using the simulation, as we can estimate and evaluate the surface temperature distribution on the cavity surface even when the method based on test piece measurements cannot be applied due to any restrictions on the piping layout.

References

- (1) Sugio, D., Okazaki, S., Kaneko, M.: Application of Rapid Heat and Cool Molding to High Strength Outer Parts without Painting Treatment, SAE International Journal of Materials and Manufacturing, Vol. 10, No. 1, DOI 10.4271/2016-32-0024, (2017)
- (2) Sugio, D., Okazaki, S., Kaneko, M.: Study on Simulation Method for Evaluation of Mold Design Applicable to Heat & Cool Molding Method for Plastic Parts Used in Motorcycles, Proceedings of 2017 JSAE Annual Congress (Spring), No. 111-17, pp. 2857-2862, (2017)
- (3) Shayfull, Z., Sharif, S., Zain, A. M., Ghazali, M. F., Saad, R. M.: Potential of Conformal Cooling Channels in Rapid Heat Cycle Molding: A Review, Advances in Polymer Technology, Vol. 33, No. 1, DOI 10.1002/adv.21381, (2014)
- (4) Gnielinski, V.: New equations for heat and mass transfer in turbulent pipe and channel flow, International Chemical Engineering, Vol. 16, No. 2, pp. 359-367, (1976)

■ Author ■



Daisuke Sugio

Prediction Technology for Vibration at Takeoff and Acceleration in AWD Vehicle Fitted with CVT

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ABSTRACT

A vibration prediction technology for the early examination of noise and vibration performance at development stages prior to the manufacture of a prototype vehicle has been developed. CVT belt resonance and differential rotation and the fluctuation component in the rear differential clutch were modeled and incorporated in a multi-body analysis model for use in noise and vibration performance design in vehicle development. The torsional vibration transfer characteristic of the CVT belt and the rear differential clutch vary in response to driving conditions in driving scenarios involving vehicle takeoff and acceleration. Parameters to be applied in the model are therefore identified in unit tests of the CVT and all-wheel drive system, which are developed in advance of the vehicle body. The developed technology is effective in performance design at development stages prior to the manufacture of a prototype vehicle, and reduces the man-hours involved in repeating performance validation using actual vehicle tests.

1. Introduction

The number of SUVs sold is increasing worldwide. Many SUVs are fitted with all-wheel drive (AWD) systems that allow them to display a high level of driving performance even on snow-covered roads and other low- μ roads, and unpaved roads. The number of these vehicles fitted with high-fuel-efficiency CVT as an environmental measure is also increasing.

The need to enhance noise and vibration (NV) performance in order to increase comfort is also growing, but at the same time, there are other strong demands on automakers, including the realization of weight reductions in order to reduce fuel consumption and the achievement of reduced cost, making vehicle development an increasingly complex process. Similarly, it is also necessary to shorten the development period and reduce the number of prototypes manufactured. As a response to these demands, the use of simulation technologies at stages of development prior to the manufacture of the vehicle is an effective means of increasing development efficiency.

There has been a recent tendency among automakers to increase the allocation of torque to the rear wheels at takeoff and in driving scenarios involving acceleration as a means of enhancing driving performance. In order to control the consequent displacement of the rear differential unit (RDU) mounts in response to the drive reaction force from the tires, it is necessary to increase the support stiffness of the RDU

mounts. However, when the stiffness of the RDU mounts is increased, the amount of vibration from the RDU mounts isolated to the vehicle body is reduced, resulting in a decline in NV performance. At the same time, because the lockup domain of the torque converter in a vehicle fitted with a CVT is broader than that of one fitted with an AT, the amount of vibration isolated by the lockup damper is lower. As a result, vibration input from the engine to the propeller shaft, known as torque fluctuation input, is higher. Because the specifications of the propeller shaft and RDU mounts are highly dependent on the layout and the support method, it is not easy to vary them after the prototype vehicle has been manufactured.

In addition, because the torsional vibration transfer characteristics of the CVT belt and the RDU clutch vary transiently at vehicle takeoff, reproduction using conventional simulation models that assume steady-state conditions is challenging. A vibration prediction technology was therefore developed that was capable of predicting, prior to the manufacture of a prototype vehicle, the increase in vibration input from the RDU mount to the vehicle body with the realization of increased AWD driving performance, and assisting in the examination of specifications that would enhance NV performance. A multi-body analysis model incorporating a CVT belt and RDU clutch was created, with the FF based AWD vehicle provided with an inline 4-cylinder engine shown in Fig. 1 as the subject of the study.

¹ Automobile Center

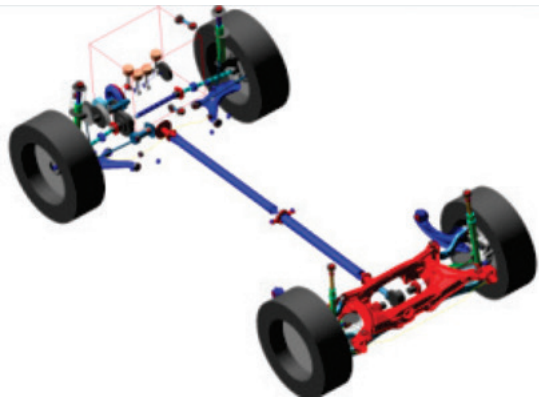


Fig. 1 Vehicle multi-body simulation model

2. Research Issues and Goals

This chapter will consider the issues associated with a technology for the prediction of vibration at takeoff and acceleration in an AWD vehicle fitted with a CVT.

In the conventional AWD NV performance design simulation models employed in vehicle development, CVT are modeled using rigid body elements. In addition, RDU clutches are modeled using static friction characteristics that do not include differential rotation or a fluctuation component.

In the case of the belt CVT forming one of the subjects of the research under discussion, the excitation force and resonant frequency of the CVT belt vary with driving conditions, for example the shift ratio or the input torque from the engine at takeoff and acceleration. CVT belt simulations conducted to date⁽¹⁾⁻⁽³⁾ have not considered energy loss through friction, and have not been able to reproduce the damping coefficient or excitation force in relation to changes in driving conditions such as the shift ratio or the input torque.

In addition, the electronically controlled hydraulic RDU clutch under study in this research produced a speed differential between the front and rear wheels at takeoff and acceleration, resulting in differential rotation. The torsional vibration transfer characteristic of RDU clutches varies in response to differential rotation and the fluctuation component. Because the models employed in RDU clutch simulations conducted to date⁽⁴⁾ have reflected only dynamic friction characteristics measured under conditions without a fluctuation component, they have not been able to reproduce the torsional vibration transfer characteristics of the RDU clutch at takeoff and acceleration. With regard to methods of measuring the friction characteristic, it is not possible to measure the clutch friction characteristic in a state in which differential rotation is occurring or a fluctuation component is acting under actual driving conditions. Because of this, it is not possible to accurately reproduce the torsional vibration transfer characteristic of an AWD

system provided with a propeller shaft using conventional simulation technologies.

These examples of simulation technologies are almost all examples of reports involving single CVT or clutch units. There are almost no examples to date of research on the prediction of input vibration to the vehicle body using combinations of complex systems.

The research discussed in this paper developed a method of identifying the damping coefficient of a CVT belt, which varies with factors such as the shift ratio, and the coefficient of dynamic friction of an RDU clutch, which varies in response to differential rotation and the fluctuation component, based on unit test results for each of the systems, which are developed in advance of the actual vehicle. The project then went on to develop a technology for the prediction of RDU mount vibration in an AWD vehicle fitted with a CVT (Fig. 2).

The target vibration frequency range for the research was 90 Hz to 120 Hz, which encompasses CVT belt, propeller shaft, and RDU mount resonant frequencies that represent a considerable issue in terms of realizing NV performance at vehicle takeoff. The target driving conditions were those at vehicle takeoff and acceleration. The research sought to make it possible to predict the level of vibration of the RDU mounts by creating a model using only data available at development stages prior to the manufacture of a prototype, with CVT belt speed fluctuations employed as input in addition to engine torque fluctuations, and employing it in combination with a vehicle response model.

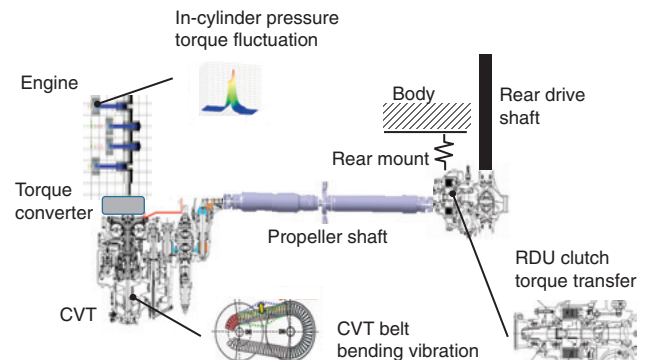


Fig. 2 Transfer mechanism

3. Modeling: Overview and Concept

The simulation model was a multi-body analysis model encompassing the engine, CVT, propeller shaft, rear drive shaft, RDU clutch, RDU mounts, and vehicle body. This chapter will discuss the modeling of the CVT belt and RDU clutch using the technology developed in this research.

3.1. Modeling of CVT Belt

Because the CVT belt undergoes elastic deformation in the target frequency range, it was expressed using mass elements, spring elements and damping elements. Figure 3 shows a schematic diagram of the CVT belt model. In order to reproduce the string vibration of the belt in the 100 Hz range while reducing the calculation load, rather than using a finite element model, the CVT belt was modeled using one spring element and one damping element. The primary and secondary pulleys were modeled using mass elements. When the CVT belt is in motion, it loses energy through contact between the elements, rings, and other metal parts. Equation (1) shows the relationship between the equivalent viscous damping coefficient between the mass elements and energy loss⁽⁵⁾. C' is the equivalent viscous damping coefficient, E is energy loss per cycle, ω is angular velocity, and A is amplitude.

$$C' = \frac{E}{\pi \omega A^2} \quad (1)$$

As Eq. (1) shows, the theoretical viscous damping coefficient can be calculated from the actually measured energy loss and amplitude, but it is no simple matter to directly measure the amplitude of each element, ring, and other part while the rotating belt is resonating. In the research under discussion, torsional torque fluctuation tests of the CVT unit were conducted under conditions simulating actual driving, and the damping coefficient was identified from the measured degree of fluctuation in pulley rotation. The combination of this coefficient with the mass and spring characteristic determined by the design specifications made it possible to create a model that could reproduce elastic deformation in the CVT belt under differing shift ratios and other conditions.

3.2. Modeling of RDU Clutch

In an RDU clutch, a wet-type multi-disc clutch is coupled between the input-side shaft, which is directly connected to the propeller shaft, and the output-side shaft, which is directly connected to the hypoid gear. In the target frequency range, the clutch is not subject to elastic deformation, and can be expressed using mass elements and friction elements. Figure 4 shows a schematic diagram of the AWD system model including the RDU clutch. Because the lubrication

state of the clutch varies with differential rotation and the load acting on the clutch, it is not easy to calculate the clutch friction characteristic for numerical analysis. The clutch friction characteristic was therefore determined experimentally in the project under discussion. The friction characteristic was measured using a low-velocity friction apparatus (LVFA). The LVFA measures the dynamic friction characteristic by applying a thrust load to the clutch unit and imparting a forced rotation that does not include a fluctuation component. Figure 5 shows the dynamic friction characteristic against differential rotation as measured in the LVFA. Equation (2) is a relational equation for the pressing force on the clutch and the transmitted torque. T is transmitted torque, μ_d is the dynamic friction coefficient, n is the number of clutches, R is the dynamic radius of the clutch, and F is the pressing force acting on the clutch.

$$T = \mu_d n R F \quad (2)$$

When the vehicle accelerates at takeoff, the RDU clutch experiences differential rotation, and in addition receives torque fluctuation from the engine and CVT on the input side. It has been reported that the dynamic friction characteristic in relation to the torsional vibration that formed the subject of this research differs from states in which there is no fluctuation component⁽⁶⁾. It is not possible to accurately express the torsional vibration transfer characteristic of an RDU clutch at vehicle takeoff and acceleration via a conventional model expression that uses the dynamic friction coefficient μ_d measured in an LVFA that does not consider the fluctuation component. It is therefore necessary to

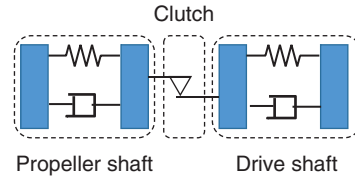


Fig. 4 Modeling of AWD unit

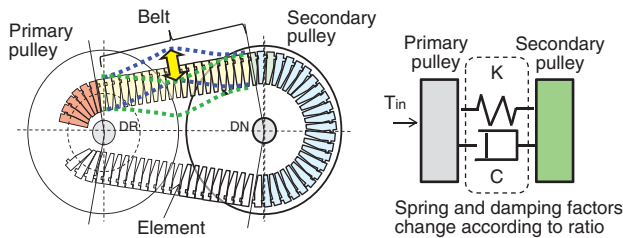


Fig. 3 Modeling of CVT belt

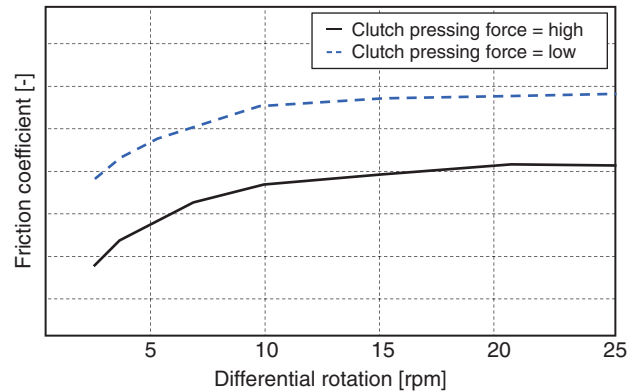


Fig. 5 Clutch friction coefficient in response to differential rotation

consider the dynamic clutch friction characteristic in a state that includes a fluctuation component.

In the research under discussion, torsional torque fluctuation tests were conducted with a unit AWD system as the subject, and the dynamic friction coefficient with consideration of the fluctuation component produced by differential rotation was identified from the measured degree of fluctuation in rotation. Equation (3) is a relational equation proposed by the research. $d\mu_d/dV$ is the dynamic friction coefficient fluctuation characteristic identified from the fluctuation component. By expressing the relationship between the pressing force on the clutch and transmitted torque with consideration of average torque and the fluctuation component, it is possible to express the torsional vibration transfer characteristic of the RDU clutch.

$$T = \left(\mu_d + \frac{d\mu_d}{dV} \cdot dV \right) nRF \quad (3)$$

4. Method of Identification and Results of Verification

4.1. Method of Identification of CVT Parameters

For identification of the CVT belt parameters, the spring coefficient and damping coefficient were identified with the mass as constants and the objective function as the fluctuation in pulley rotation speed measured in a unit test apparatus.

Figure 6 shows a schematic diagram of the CVT unit test apparatus. The CVT was attached to a drive dynamometer, and to an absorption dynamometer by means of a left-right driveshaft. The range for each parameter for the test conditions was set as shown in Table 1 based on the use domain in an actual vehicle and the domain of occurrence of CVT belt resonance. In addition, the fluctuation of the pulley rotational speed was measured under steady-state conditions for each of the conditions shown in Table 1.

Figure 7 shows results for the identification of the damping coefficient against rotational speed and input torque at a low pulley ratio as a representative condition.

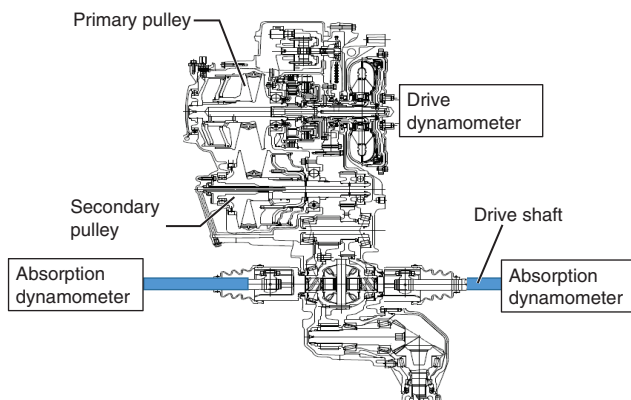


Fig. 6 CVT unit test

Table 1 CVT unit test parameters

Rotational speed	1000 to 4000 rpm
Input torque	0 to 200 Nm
CVT pulley ratio	0.5 to 2.65

Figure 8 shows the results of a simulation of the rotational speed of the secondary pulley at vehicle takeoff and acceleration as results for verification of the reproducibility of the identified parameters. Based on the fact that the change in the resonant frequency of string vibration and the degree of fluctuation in the speed of rotation of the secondary pulley were reproduced, it was judged that the simulation possessed sufficient accuracy.

4.2. Method of Identification of RDU Clutch Parameters

For identification of the RDU clutch parameters, the friction coefficient was identified with the mass as constants and the objective function as the fluctuation in the rotational speed of the propeller shaft measured for each condition in a unit test apparatus.

Figure 9 shows a schematic diagram of the AWD system test apparatus. A drive dynamometer was attached to the end of the propeller shaft, and absorption dynamometers were attached to the left-right driveshaft via the RDU. In order to measure the RDU clutch torsional vibration transfer characteristic during torsional resonance of the propeller shaft, a low-inertia dynamometer capable of fluctuating input in the 100 Hz range, simulating CVT belt vibration, was employed as the drive dynamometer. The range for the test condition parameters was set as shown in Table 2, based on the use domain in an actual vehicle.

Next, the method of identification of parameters and the results of parameter identification will be discussed.

Based on the results of previous tests, it was found that the rotating propeller shaft displays hysteresis in relation to

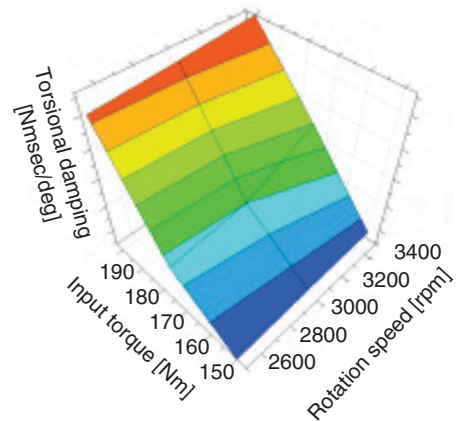


Fig. 7 Verified belt damping parameters at low pulley ratio

Table 2 AWD unit test parameters

Input rotation speed	500 to 2000 rpm
Input torque	50 to 150 Nm
Torque fluctuation amplitude	10 to 80 Nm
Differential rotation speed of clutch	0 to 20 rpm

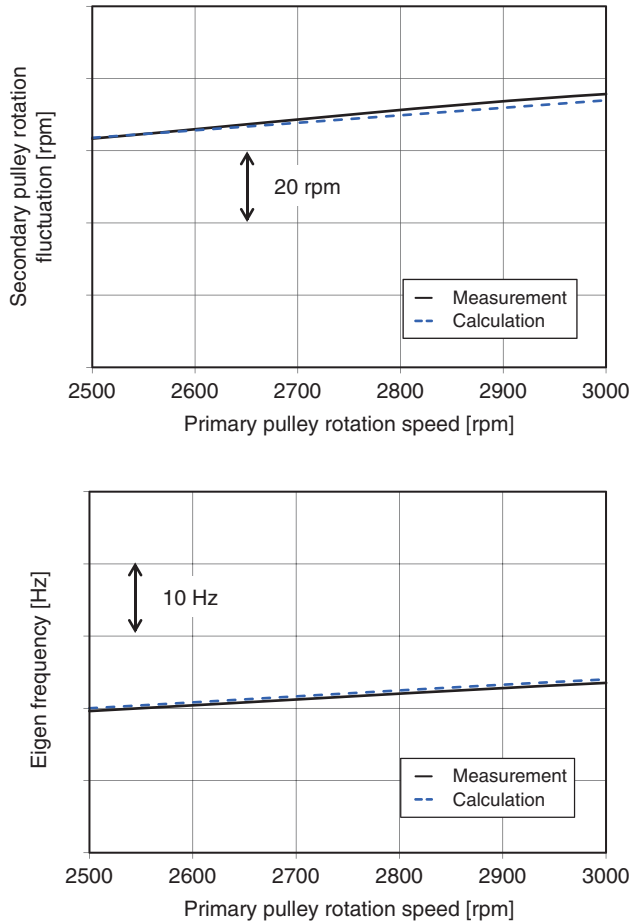


Fig. 8 Simulation results for CVT belt at vehicle takeoff and acceleration

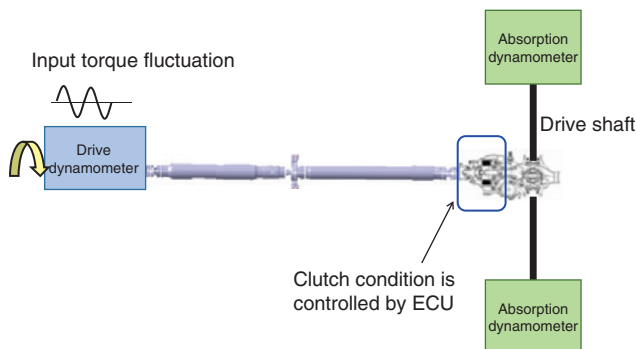


Fig. 9 AWD unit test of torsional vibration

the torsion angle and input torque characteristic. The energy loss produced by hysteresis is considered to be an effect of factors including the sliding resistance and looseness of the joints and bearings, and calculating the damping characteristic for numerical analysis is a challenge. In order to identify the damping coefficient of the propeller shaft, measurements were therefore conducted in the test apparatus shown in Fig. 9 with no differential rotation of the clutch. The damping coefficient was identified with the mass and spring as constants and the rotation speed of the propeller shaft measured in tests as the objective function. Figure 10 shows the results of identification of the damping coefficient. The modeling of the RDU clutch was conducted using Eq. (2), and the model reflected the dynamic friction coefficient μ_d with no fluctuation component.

Next, the measurements were conducted with differential rotation of the RDU clutch using the test apparatus shown in Fig. 9 in order to identify the coefficient of friction of the RDU.

Average torque transfer was expressed as the conventional dynamic friction coefficient μ_d measured using the LVFA, and torque fluctuation transfer was defined as the dynamic friction coefficient fluctuation characteristic $d\mu_d/dV$ that varies with differential rotation speed. Figure 11 shows the results of parameter identification of the dynamic friction coefficient fluctuation characteristic $d\mu_d/dV$ against the fluctuation component, with actually measured fluctuation

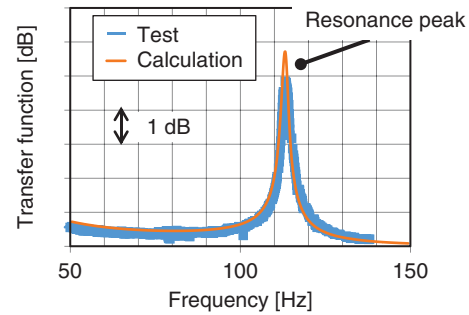
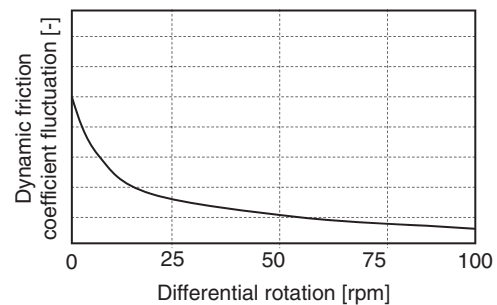


Fig. 10 Simulation results for transfer function of torque fluctuation without clutch differential rotation


 Fig. 11 Verified dynamic friction coefficient fluctuation parameters $d\mu_d/dV$

in the propeller shaft rotation speed as the objective function, and Fig. 12 shows simulation results for the torque fluctuation transfer characteristic. Considering the dynamic friction coefficient fluctuation characteristic made it possible to reproduce the torsional vibration transfer characteristic in the target frequency range (solid line) with sufficient accuracy even with differential rotation of the clutch, by comparison with simulation results that did not consider the dynamic friction coefficient fluctuation characteristic (broken line).

4.3. Results of Verification of Prediction of RDU Mount Vibration

The CVT belt model and RDU clutch model for which parameters had been identified based on the results of unit tests were incorporated in a vehicle model, and the vibration of the RDU mounts at vehicle takeoff and acceleration was predicted. The target accuracy for the prediction technology was set as the ability to calculate the peak level of vibration within the target frequency range to an accuracy of ± 3 dB, which would not hinder studies at development stages prior to the manufacture of a prototype. Figure 13 shows actual

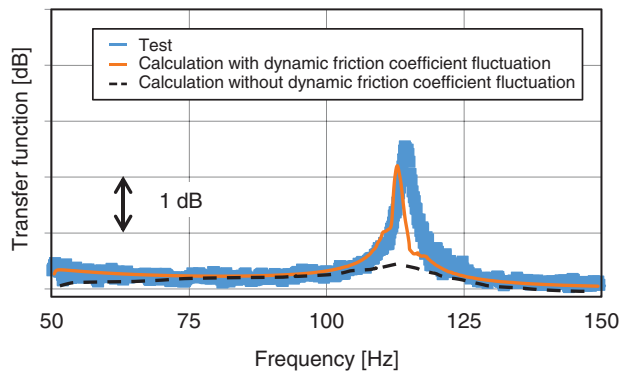


Fig. 12 Simulation results for transfer function of torque fluctuation with clutch differential of 10 rpm

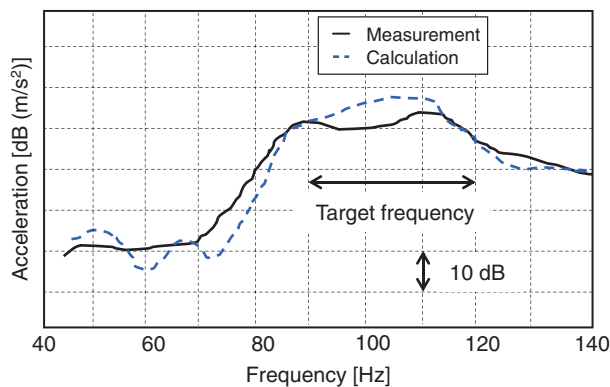


Fig. 13 Spectrum accuracy of rear differential mount vibration

vehicle test results and simulation results for the vibration of the RDU mounts at WOT acceleration from a stationary state. The calculation results realized the target accuracy for RDU mount vibration in the 90 Hz to 120 Hz frequency range under analysis, demonstrating that the method possesses sufficient accuracy for application in vehicle development.

5. Example of Application

This chapter will discuss an example of a study of NV performance in a prototype vehicle assuming a development process utilizing the modeling technique developed in this research.

Increased allocation of drive force to the rear wheels in order to enhance driving performance on snow-covered roads and hill-climbing performance was set as the condition, and NV performance was examined.

Figure 14 shows the results of a simulation of transfer load input to the vehicle body from the RDU when AWD performance is enhanced, in this case by increasing the allocation of drive force to the rear wheels, using the prediction technology developed in this research. The calculation results show that the transfer load input to the vehicle body under this condition (dotted line) increases against the current model (broken line). While methods that respond to the excitation source are also available, the research under discussion studied the reduction of in-cabin sound pressure with the RDU mounts considered as changeable parts. The target for the transfer load input to the vehicle body shown in Fig. 14 (solid line) was achieved by increasing the support span in order to increase support stiffness in relation to drive reaction force, reducing the spring characteristic of the RDU mounts under a drive force load.

Figure 15 shows results obtained by manufacturing a prototype and reflecting the examined specifications in a vehicle. The performance targets for interior noise were realized.

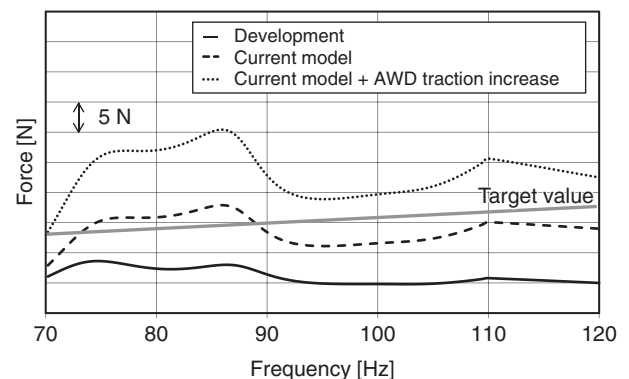


Fig. 14 Simulation results for input force from RDU to vehicle body at vehicle takeoff and acceleration

The results discussed above indicated that it had been possible to develop a prediction technology able to be used in the study of system characteristics that satisfy AWD driving performance and NV performance targets assuming vehicle development demands.

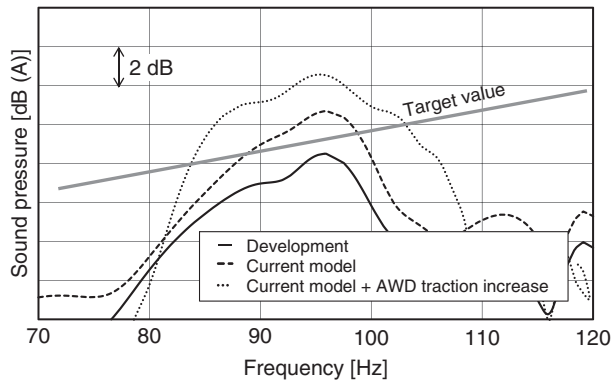


Fig. 15 Vehicle test results for in-cabin sound pressure

6. Conclusion

A simulation model able to be employed in the study of NV performance has been developed based on the conventional simulation model used in AWD NV performance design, in order to expand the range of frequencies and driving conditions analyzed, and balance NV and driving performance in systems combining a CVT and AWD.

- (1) A method of modeling the CVT belt was developed in which the damping coefficient, which varies with driving conditions including the shift ratio and input torque, is identified experimentally in CVT unit tests.
- (2) A method of modeling the RDU clutch was developed in which the dynamic friction coefficient fluctuation characteristic, which varies with differential rotation speed and the fluctuation component, is identified experimentally in AWD system tests.

The reflection of these developments in a simulation model has made it possible to predict RDU mount vibration at development stages prior to the manufacture of a prototype vehicle, and to study specifications for NV and AWD driving performance at the initial stage of development.

References

- (1) Sakagami, K.: Analysis of Slip in Metal V-belt Type CVT, Transactions of Society of Automotive Engineers of Japan, Vol. 49, No. 2, p. 290-295, (2018) (in Japanese)
- (2) Saito, T., Miyamoto, K.: Study of Self-induced Vibration in an Operating Metal Pushing V-belt CVT,

SAE Technical Paper, 2012-01-0309, (2012)

- (3) Tarutani, I., Nagasawa, Y., Harima, K., Tsuchiya, S.: Ring Behavior Analysis of Metal CVT Belt, JSAE Annual Congress (Spring) Proceedings, No. 1-16, p. 5-9, (2016) (in Japanese)
- (4) Koizumi, Y., Fujikawa, S., Morita, K., Marutani, T., Nishikawa, H., Nakajima, H.: Optimal design of fuel consumption and NV for 4WD vehicle under drivetrain non-linearity and powertrain control, JSAE Annual Congress Proceedings, No. 44-12, p. 5-8, (2012) (in Japanese)
- (5) Okamoto, J., Nakayama, K., Sato, M.: Toraiboroji Nyumon - Masatsu, Mamo, Junkatsu no Kiso, SAIWAI SHOBO, p. 112-116, (2014) (in Japanese)
- (6) Nagasawa, Y.: New Method for Measuring $d\mu/dV$ Characteristics Using Fluctuation of Sliding Velocity, R&D review of Toyota CRDL, Vol. 33, No. 2, p. 74, (1998)

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Enhancement of Power Transmission Efficiency in Metal Pushing V-belt CVT with Multi-V-surface Pulley and Modified-pitch Belt

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ABSTRACT

To date, slips in the interior of the belt and pulley misalignment have been allowed as phenomena inherent to the mechanism of metal pushing V-belt CVTs. The research discussed in this paper reduced loss, which had conventionally been allowed, for the first time ever in order to further increase efficiency. A slip in the interior of the belt is generated by a differential between the belt pitch radius and the ring winding radius when the belt is wrapped around the pulleys. Slippage between the elements and the rings was reduced by reducing this radius differential. In addition, changing the shape of the pulley V-surface from 11 deg to 9 deg and adopting a curved multi-V-surface shape made it possible to reduce the tensile force of the rings and thus reduce the contact load in the sliding sections. Furthermore, the developed pulley-belt mechanism reduces the stroke differential between the drive and driven pulleys when shifting and controls the occurrence of misalignment irrespective of the speed ratio. This has increased power transmission efficiency by a maximum of 2.4% and by 0.9% averaged over the speed ratio width.

1. Introduction

There has recently been a rapid shift towards the electrification of automotive powertrains against the background of the need to reduce consumption of fossil fuels. Nevertheless, it is predicted that even in 2030, there will still be 30% or more of conventional vehicles that use internal combustion engines alone as their power source⁽¹⁾. Given this, it is necessary to boost fuel efficiency above current levels in conventional vehicles also. Considering these market trends, the application of metal pushing V-belt CVTs is increasing as a means of boosting fuel efficiency performance. Honda has conducted in-house development of metal pushing V-belts⁽²⁾, and manufactures the units in-house. Figure 1 shows a breakdown of loss in a metal pushing V-belt CVT in JC08 mode. From this, it can be seen that the metal pushing V-belt is the element generating the greatest amount of loss. This results from the fact that the geometric trajectory of the metal pushing V-belt differs from its actual trajectory. Up to date research focusing on metal pushing V-belt loss in order to reduce this loss has studied the optimum V-surface angle⁽³⁾ and configurations that will reduce pulley thrust and control

pulley inclination⁽⁴⁾. However, the slipping that occurs in the interior of the belt and the twisting of the belt itself (termed “misalignment” below) have been accepted as phenomena inherent to the metal pushing V-belt CVT mechanism.

The research discussed in this paper focused on reducing the loss originating in the mechanism of the metal pushing V-belt CVT, and sought to create a theoretical element form and advance production technologies in order to

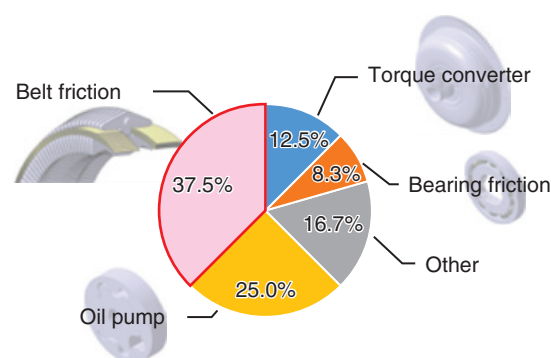


Fig. 1 Breakdown of power consumed in CVT (during operation in JC08 mode)

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realize that form. The research was successful in boosting the performance of metal pushing V-belt CVTs. This paper presents an overview of the theory involved in the research.

2. Development Aims

In a CVT, the belt runs between a fixed and a movable pulley. Power is transmitted from the drive pulley to the driven pulley by the friction force of the belt V-surfaces (Fig. 2). The metal pushing V-belt is composed of two sets of layered rings and several hundred elements (Fig. 3). The belt transmits power by means of the mutual pushing of the elements. The pitch radius of the metal pushing V-belt is determined with the positions at which the elements mutually push each other as the rocking position (Fig. 4). As Fig. 5 shows, because there is a difference in the belt pitch radius, R , and the ring winding radius, $R + \alpha$, a circumferential velocity differential, ΔV , is generated between the elements and the rings. This circumferential velocity differential causes the elements and rings to slip

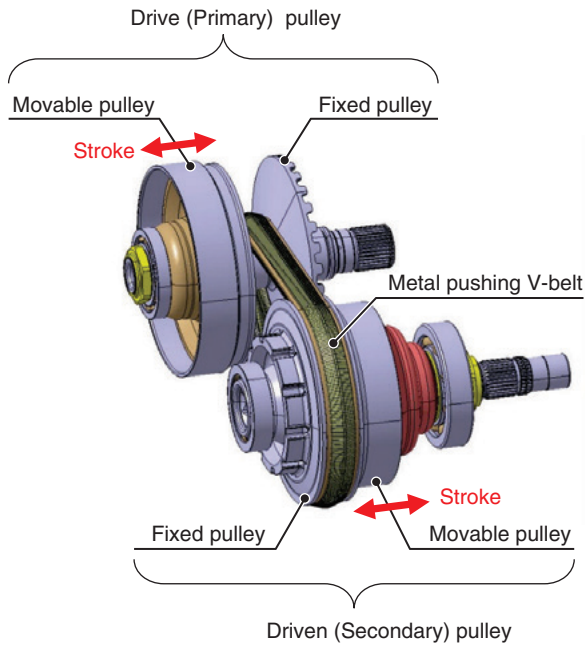


Fig. 2 Configuration of pulleys and metal pushing V-belt

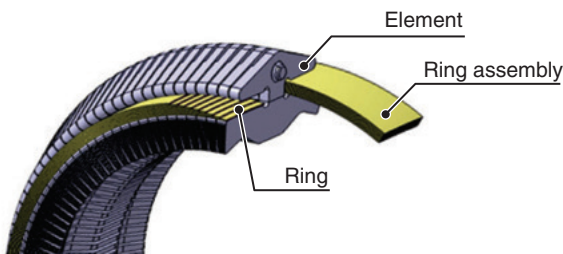


Fig. 3 Configuration of metal pushing V-belt

with the rotation of the pulleys. Similarly, the ring thickness produces a difference in the winding radius between each layered ring, and as a result, the circumferential velocity differential generates slip between the rings.

The friction loss in each section of the metal pushing V-belt resulting from slip in the interior of the belt, P , is given by Eq. (1), using the coefficient of friction of the slipping sections, μ_s , the vertical load on the slipping sections, N , and the slip velocity, ΔV .

$$P = \mu_s N \Delta V \quad (1)$$

In other words, the friction loss due to slip in the interior of a metal pushing V-belt could be reduced by reducing the slippage per unit time between the elements and the rings and between the rings themselves (termed “slippage” below) and the ring force pressing on the elements. The research discussed here ought to increase power transmission efficiency with a focus on this point.

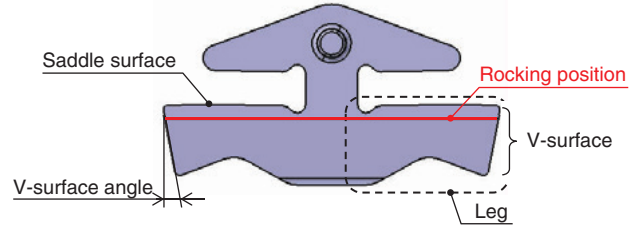


Fig. 4 Configuration of belt element

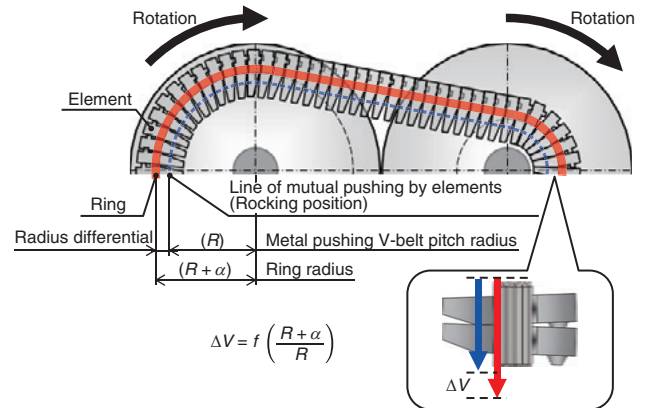


Fig. 5 Velocity differential between elements and rings

3. Method of Increasing Power Transmission Efficiency

The specifications of the pulleys and the belt were changed, as shown in Table 1, in order to increase power transmission efficiency. Broadly speaking, three items were changed. The first of these was the moving of the element rocking position closer to the saddle surface (Fig. 4)

Table 1 Specifications of pulley and belt

Item		Current model	Developed model
Pulley	Distance between pulley axes	156 mm	156 mm
	V-surface	Single angle 11 deg	Multiple angle 9 deg / Round
Belt	Element width	24 mm	24 mm
	Rocking position	1.00 mm	0.55 mm
	V-surface	Single angle	Multiple angle
		11 deg	9 deg
		Grooved	Grooved
	#1 ring diameter	208.8 mm	208.8 mm

in order to reduce the differential of the winding radius between the elements and the ring of the metal pushing V-belt, α , thus reducing slippage, ΔV . In addition, in order to respond to the issue of element durability following the reduction of the winding radius differential, contact surfaces with contiguous elements were set on the backs of each element. The second point changed was the reduction of the pulley V-surface reference angle, in order to lower the tensile force of the metal pushing V-belt and reduce the contact load, N , between the elements and the rings and between the rings themselves. The third change was making the outer circumference of the pulley V-surface curved (Fig. 6), in order to reduce the pulley belt clamping force (termed “pulley thrust” below), which increases with the reduction of the pulley V-surface angle and additionally increases ring durability by reducing belt misalignment. Curvature was also added to the element V-surfaces with the change in the shape of the pulley V-surfaces.

3.1. Change in Element Rocking Position

Since the metal V-belt is a pushing belt, the elements receive power from the pulleys, and the speed of the belt is determined by the position at which the elements mutually push each other (the rocking position). By contrast, the

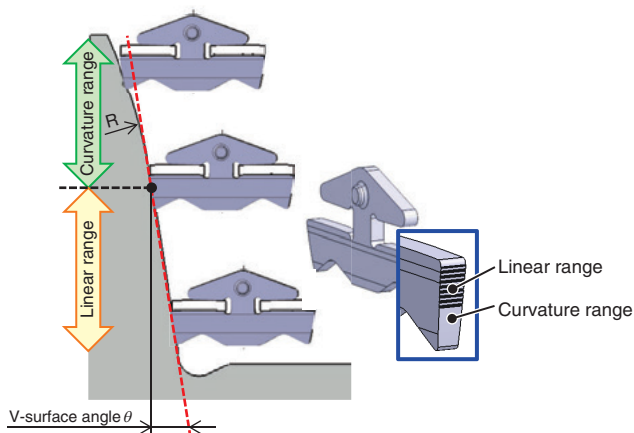


Fig. 6 Pattern diagram of pulley and belt

rings move by being dragged as they rest on the element saddle surfaces. Due to this, the elements and the rings rotate without slipping on the pulley side, on which the winding length of the metal pushing V-belt is greater⁽⁵⁾. Slipping of the elements and rings occurs in the other pulley, where the belt winding length is smaller due to the difference in the winding radius. Similarly, the rings mutually slip because they are layered in the direction of the pulley winding radius. In other words, it would be possible to reduce slippage by attempting to equalize the winding radius of the elements and rings, which are generating slip, to the greatest extent possible.

The slippage occurring between the elements and the rings and between the rings themselves could be changed by using a speed ratio that would change the winding radius differential between the drive pulley and the driven pulley. Slip would not occur using Mid ratio, in which a differential between the winding radius of the metal pushing V-belt around each pulley was not generated, and friction loss would also be minimized using this ratio. On the other hand, slip would become conspicuous at Low or High ratios, in which the winding radius differential would be large, and friction loss would also become high.

In the research under discussion, the rocking position was reduced from 1.00 mm to 0.55 mm in order to reduce the winding radius differential between the elements and the rings (Fig. 7).

Since the slippage between the elements and the rings increases in proportion to the winding radius differential between the elements and the rings, α , a reduction of 0.45 mm in the winding radius differential would result in a 45% reduction in slippage (Fig. 8). However, in the case of the layered rings, slippage would not change because their configuration would not be changed.

Despite the fact that the utility of the measure has long been known⁽⁶⁾, the rocking position has not been moved closer to the shear-machined section (the element profile). This is due to concerns that the creation of gaps between the elements would hinder the maintenance of a state of stable contact between the elements, affecting their durability. Two causes can be considered for this. The first of these is that the leg section (Fig. 4) becomes

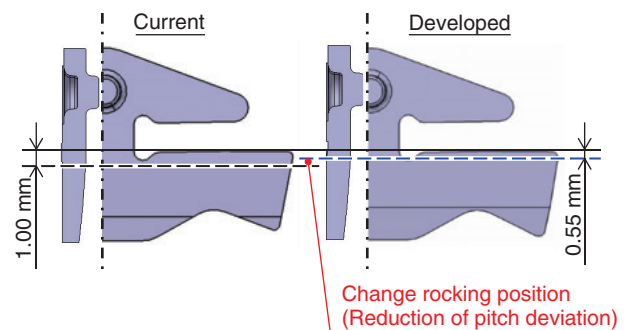


Fig. 7 Comparison of current and developed elements

thinner as a result of changes in the thickness of the plate (droop) when the element is formed. Figure 9 shows a contour diagram that expresses element plate thickness as height data. This diagram shows that the rocking section is lower than the upper section of the saddle surface in the element with a thin leg section. The second cause is the occurrence of mutual displacement between elements. Due to the configuration of the metal pushing V-belt, there is no constraint on parts other than the rings and pulleys, and micro-displacements occur between the elements when the belt is in operation⁽⁷⁾. As the rocking position is 1.00 mm in a conventional element, this displacement can be ignored. However, the displacement cannot be ignored in an element with a lower rocking position. Since the metal pushing V-belt receives power from the pulleys, a load is applied to the V-surface (the edge of the leg). If gaps occur between the elements at this time, excessive bending stress is generated on the elements in comparison to a state in which there are no gaps. As a result, it was important to respond to this issue for the application of the technology discussed here. In order to do so, step differences were engineered on the back surfaces of the elements as shown in Fig. 10, creating a shape that made it possible to maintain the state of contact⁽⁷⁾. Figure 11 shows the state of contact between the elements with and without the step difference. The step difference allows the elements to receive power across their entire width, while also making it possible to maintain durability.

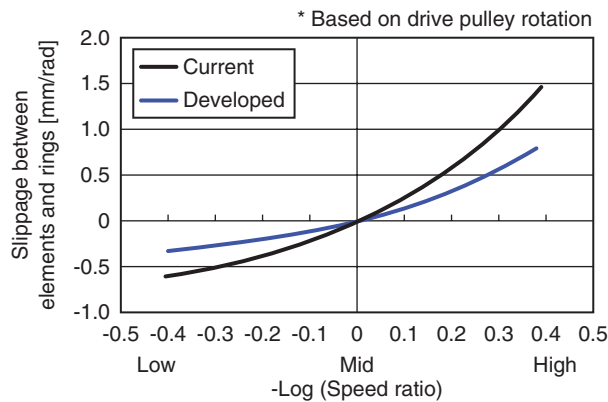


Fig. 8 Reduction of slippage between elements and rings

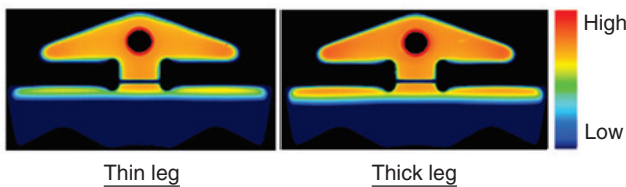


Fig. 9 Element thickness indicated by height

3.2. Application of Multi-V-surface Pulley

When the position of the metal pushing V-belt wound around the pulleys (the speed ratio) has been determined, the force that pushes the elements in the pulley radial direction due to pulley thrust and the force that maintains the elements of the metal pushing V-belt in that position due to the rings are in equilibrium. In other words, the contact load between the elements' saddle surfaces and the rings, N , is determined. As Fig. 12 shows, the contact load N corresponds to the radial component of pulley thrust, Q , and the pulley V-surface angle, θ . The contact load is therefore dependent on pulley thrust and the pulley V-surface angle, and friction loss between the elements and rings could be reduced by reducing their radial components. However, pulley thrust Q plays a role in determining torque

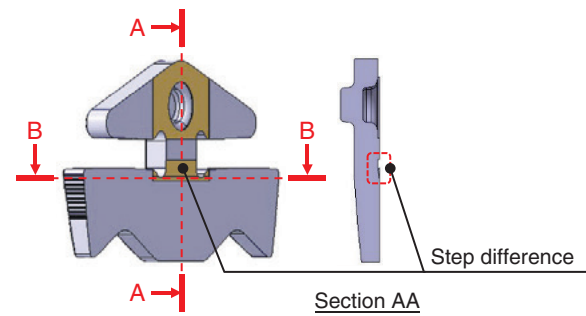


Fig. 10 Step difference on back of element

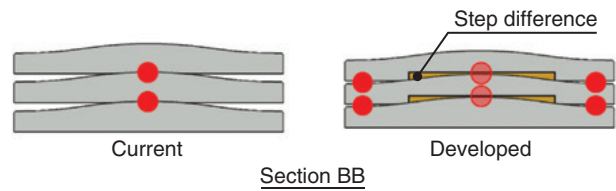


Fig. 11 Contact position of element

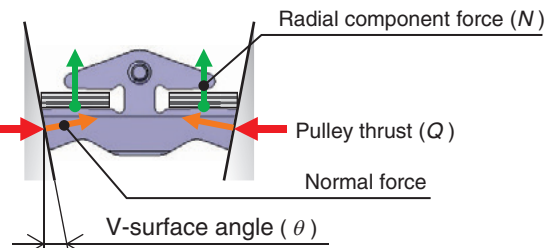
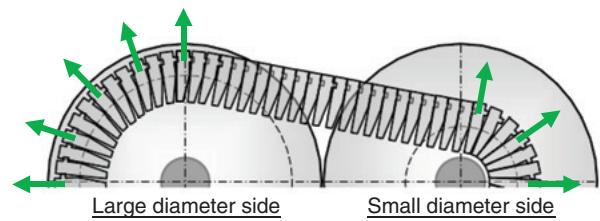


Fig. 12 Force applied to pulley and belt

transmission capacity. The torque transmission capacity of each pulley, T , can be expressed as Eq. (2) using the winding radius, R , and the belt friction coefficient, μ_b , allowing the necessary pulley thrust for torque transfer to be determined.

$$T = \frac{2\mu_b R Q}{\cos \theta} \quad (2)$$

It should be noted that the effect of operating conditions and other factors is omitted here, allowing for unambiguous expression. In addition, the pulley thrust has a low-thrust side that determines the torque transmission capacity given by Eq. (2) and a high-thrust side that determines the speed ratio. Depending on the speed ratio, these can be either the drive pulley or the driven pulley. At this time, because the necessary thrust is added to the winding radius component of the V-surface angle θ at the high-thrust pulley that determines the speed ratio, thrust in excess of the torque transmission capacity is applied. In other words, the system presents antagonistic characteristics: in the case of torque transmission capacity, the smaller the V-surface angle θ , the more pulley thrust can be reduced, while in the case of the high-thrust side that determines the speed ratio, the higher the V-surface angle θ , the more the added thrust can be reduced.

The research under discussion changed the pulley V-surface shape from an 11 deg single shape to a 9 deg curved and multi-V-surface shape in order to reduce the contact load between the element saddle surfaces and the rings. As a result, the pulley V-surface angle θ , which determines the contact load N , was set to transition from a narrow angle to a broad angle in the pulley radial direction (Fig. 13).

This made it possible to reduce the contact angle when the winding radius of the metal pushing V-belt is small, and thus to reduce the contact load between the element saddle surfaces and the rings. In addition, because it was also possible to increase the contact angle when the winding radius is large, it was possible to reduce the pulley thrust on

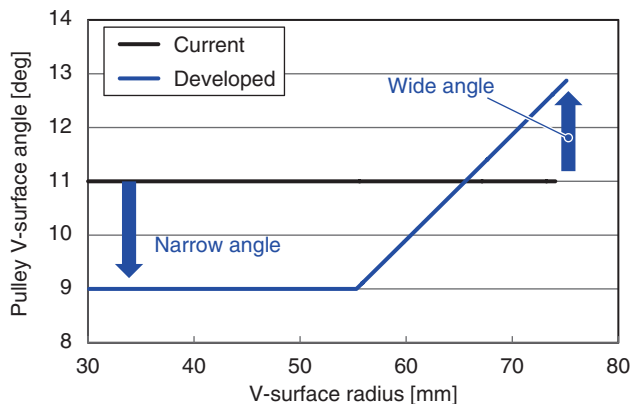


Fig. 13 Radius and angle of pulley V-surface

the high-thrust side, which increases as the angle narrows. Figure 14 shows the effect of reduction in the contact load between the element saddle surfaces and the rings through the use of the multi-V-surface shape as the force acting between the pulley axes (termed “pulley inter-axial force” below). The pulley inter-axial force is the force received by the pulley axes when the metal pushing V-belt is pushed by pulley thrust.

3.3. Reduction of Misalignment

Three points were taken into consideration in the adoption of the multi-V-surface pulley. The first of these was not increasing the stroke of the movable pulley against the previous type (11 deg single V-surface shape). The second was to attempt to help prevent the pulleys and the belt V-surfaces from making point contact on the side with the small metal belt pitch radius. The third point was to attempt to remove the stroke differential with the movable pulley, thus making it possible to do away with misalignment. Metal pushing V-belt CVTs change the pitch radius of the metal belt via the stroke of the movable pulleys on the drive and driven sides, and determine the arbitrary speed ratio by means of the radius differential. The necessary stroke at this time determines the size of the transmission, and the movable pulley stroke when creating each speed ratio is determined by the distance between the pulleys and the circumference of the metal belt. As Fig. 15 shows, there is a difference in each pulley stroke necessary for shift between the drive and driven sides, and half of this difference becomes misalignment.

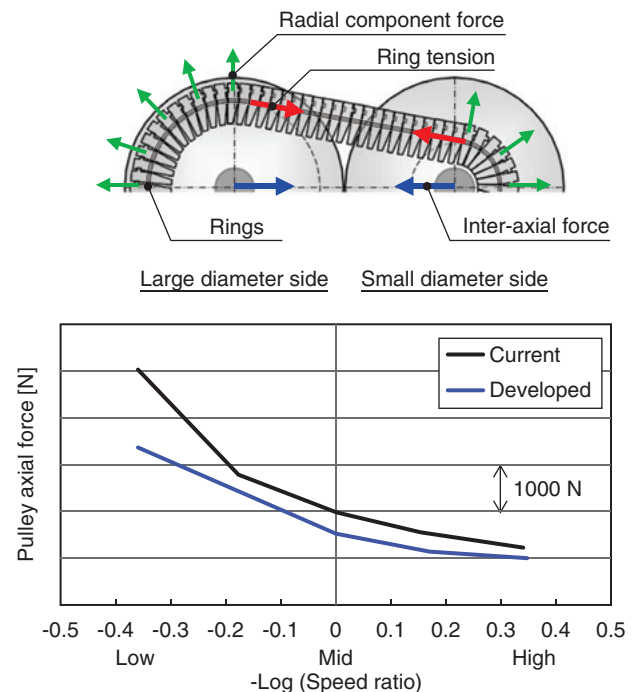


Fig. 14 Pulley inter-axial force

Misalignment, C_{MA} , is given by Eq. (3), using the winding radius, D , and the distance between the drive and driven pulleys, s , when the speed ratio, i , equals 1.0 (Mid ratio).

$$C_{MA} = \frac{D^2(i-1)^2}{\pi s(i+1)^2} \tan \theta \quad (3)$$

In the research under discussion, the generating line function for correction of misalignment via the pulley V-surface shape was defined using Eq. (4), with Y as the pulley shaft axis, and X as the radius axis.

When $X \leq$ Mid ratio radius,
 Pulley shape coordinates = $X \tan \theta$ (4-1)

When $X >$ Mid ratio radius,
 Pulley shape coordinates = $X \tan \theta + C_{MA}$ (4-2)

As it is necessary to add the actual misalignment correction amount to the extent of the curvature engineered into the elements, Eq. (4) is expressed in a simplified form.

By imparting the amount corresponding to misalignment to the pulley generating line in this way, the pulley stroke is rendered almost exactly the same for each pulley (Fig. 16). The ability to control misalignment irrespective of the speed ratio contributes to boosting the alignment of the elements.

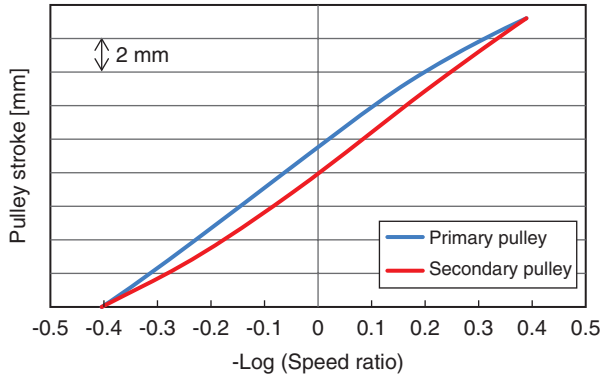


Fig. 15 Pulley stroke in 11 deg pulley

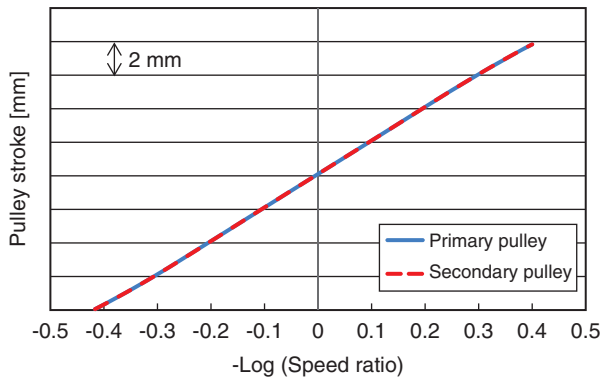


Fig. 16 Pulley stroke in developed pulley

4. Effect in Increasing Power Transmission Efficiency

Power transmission efficiency was tested in unit tests at different speed ratios for each of the two methods of boosting power transmission efficiency under study. Power transmission efficiency varies as a result of factors including the difference in the winding radius at the drive and driven pulleys due to the speed ratio. Due to this, power transmission efficiency was evaluated at five speed ratios: Low, Mid, High, and the ratios 1.5 and Top, which are intermediate between these. The results showed an increase of 2.4% in Low ratio, 1.6% in High ratio, and 0.9% in average power transmission efficiency (Fig. 17). Here, the rate of increase in average power transmission efficiency is expressed as the average of the rate of increase for each speed ratio that was set. It is considered that the power transmission efficiency declined in the Mid ratio because the V-surface angles of the drive and driven pulleys were the same, and the pulley thrust on the high-thrust side increased due to the effect of reducing the angle.

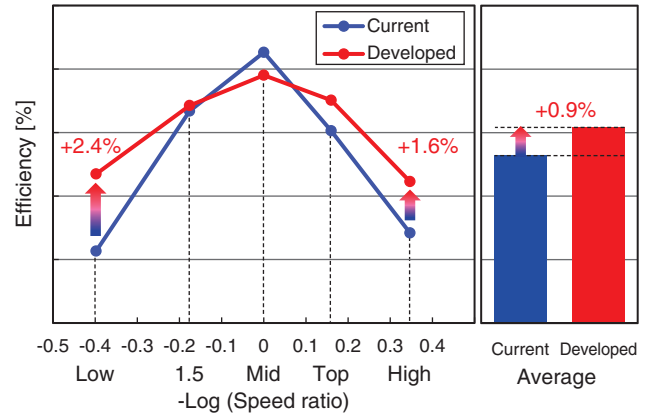


Fig. 17 Power transmission efficiency

5. Conclusion

Friction loss due to slip in the interior of a metal pushing V-belt was reduced in order to increase the power transmission efficiency of CVTs.

- (1) The pulley winding radius differential between the elements and the rings of the metal pushing V-belt was set low by changing the rocking position of the elements, making it possible to reduce friction loss by reducing slippage. In addition, the issue of durability was responded to by stabilizing the mutual contact between the elements.
- (2) The pulley V-surface shape was changed to a multi-V-surface shape using the characteristics of the high-thrust and low-thrust sides, making it possible to reduce the

contact load between the elements and the rings of the metal pushing V-belt, thus reducing friction loss.

- (3) The application of a multi-V-surface shape made it possible to correct misalignment due to the stroke differential between the drive and driven pulleys, boosting the alignment of the metal pushing V-belt.

The application of these changes to the pulley-belt mechanism increased average power transmission efficiency by 0.9%.

References

- (1) Jidosha shin jidai senryaku kaigi chukan seiri (an) hosoku shiryo, Ministry of Economy, Trade and Industry, P. 18, (2018) (in Japanese)
- (2) Yagasaki, T., Aoyama, H., Saito, T., Ishii, K., Okada, Y., Arikawa, M.: Development of Metal Pushing V-Belt for Continuously Variable Transmission, Honda R&D Technical Review, Vol. 14, No. 1, p. 181-190
- (3) Yagasaki, T., Aoyama, H., Ichijo, S., Totsuka, H., Harada, M.: Optimization of V-surface Angle for Metal Pushing V-belt CVT, Honda R&D Technical Review, Vol. 22, No. 1, p. 169-176
- (4) Ichijyo, S., Yagasaki, T., Ando, T., Sakagami, K., Sumida, S.: Study of High-efficiency CVT Shifting Mechanism using Metal Pushing V-belt, Honda R&D Technical Review, Vol. 26, No. 1, p. 34-41
- (5) Saito, T., Miyamoto, K.: Transmission Efficiency Prediction for CVT Metal V-belt with Ratio Feedback Control, CVT-HYBRID 2010 Proceedings, p. 29-34
- (6) Morimoto, Y.: Mudanhensokuki CVT nyumon, Grand Prix BOOK PUBLISHING CO.LTD., p. 53-56, (2004)
- (7) JP, Patent Application No. 6094911, (2017)

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Method of Estimating Instantaneous Safety Factor for Self-optimizing Control of Clamping Force in CVT

Kyohei SAKAGAMI, Ph.D.¹

ABSTRACT

When fluctuations in the drive pulley speed are transmitted to the driven pulley via the metal V-belt, the transmitted fluctuations are attenuated according to the belt slippage. The research discussed in this paper focused on these fluctuations in the pulley speeds and developed a method for estimating the instantaneous safety factor for belt clamping force. First, a vibration model was built by regarding the drive and driven pulleys as a one-dimensional vibrating system connected by elastic bodies. Based on this vibration model, the instantaneous safety factor was then estimated from changes in the pulley speeds. In order to verify the method's validity, the actual safety factor was calculated by measuring the driveshaft torque and belt clamping force in a test vehicle and compared with the value estimated using the proposed method. In verification testing, changes in vehicle body weight and left-right wheel differential were applied to the test vehicle. With the assumption that aging may affect the results, verifications were also performed with two test pulleys with different coefficients of friction.

The results verify that, in the required estimation accuracy range of a safety factor of 1.25 or lower, instantaneous safety factor can be estimated to within an error range of 2%. Using the proposed method, belt clamping force can be adapted even for changes in friction coefficient, a traditionally unknown variable.

1. Introduction

In recent years, with the ever-increasing demands for fuel efficiency improvements, it has become more important to increase the power transmission efficiency for transmissions. While automotive transmissions come in many different formats, the continuously variable transmission (CVT), which can optimize the brake specific fuel consumption (BSFC) of the engine, has rapidly spread in recent years⁽¹⁾. Structurally, as mainstream CVTs use metal V-belts, hereinafter "CVT" will refer to the metal V-belt CVT. In a CVT, the transmission adjusts steplessly to the desired speed by changing the winding radius for belts on a pair of pulleys. A breakdown of power loss in a CVT is shown in Fig. 1. The sample breakdown is with the CVT running in JC08, a mode for measuring fuel efficiency. The majority of CVT power loss is attributable to running the oil pump and internal frictional loss in the metal V-belt.

In a CVT, oil pressure is used to generate clamping force with the belts, which in turn creates friction between the metal V-belt and the pulleys to transmit power. An oil pump is required to generate this oil pressure.

The metal V-belt is comprised of many elements held

together by two pairs of laminated rings (Fig. 2). As the elements and rings can both move freely relative to one another, slip will occur between the elements and rings of a metal V-belt while the vehicle is running. This slip is the main cause of frictional loss within the metal V-belt.

Reducing the belt clamping force is an effective means of enhancing the efficiency of a CVT's power transmission. Reduced belt clamping force will reduce the tensile force

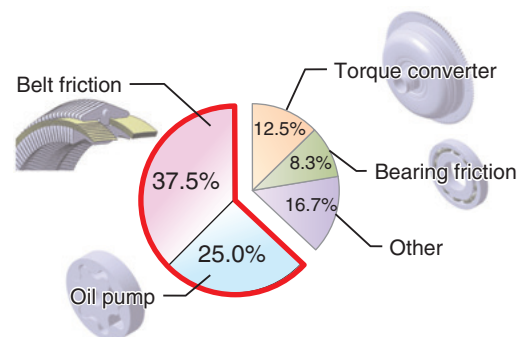


Fig. 1 Breakdown of power consumed in CVT (during operation in JC08 mode)

¹ Automobile Center

acting on the rings, which can attenuate internal frictional loss in the metal V-belt. It can also reduce the amount of work done by the oil pump.

However, it is important to determine the minimum force needed to clamp the belt as excessive reduction will result in greater slippage between belt and pulley. The friction characteristics of metal V-belts are shown in Fig. 3⁽²⁾. While the belt will experience micro slips (elastic slip) due to its own elastic deformation, a region of the belt partially adheres to the pulley [area shown in Fig. 3(A)]. When belt clamping force is insufficient, the belt will slip at all points on the pulley (sliding slip). In this case, there is no adhesion area with the pulley [area shown in Fig. 3(B)]. Sliding slip will actually lower power transmission efficiency and could also damage the belt. Thus, in terms of

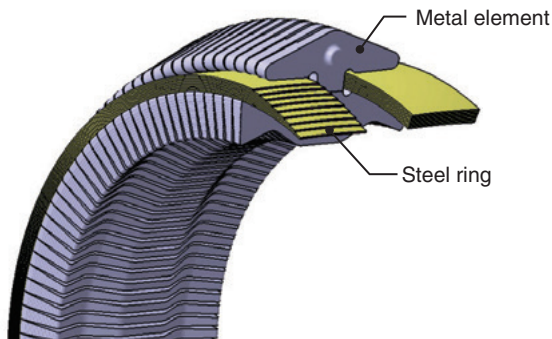


Fig. 2 Structure of metal V-belt

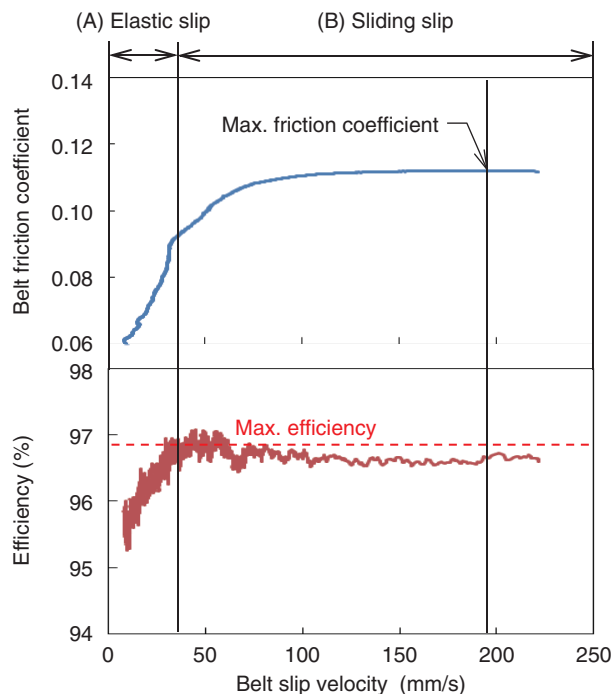


Fig. 3 Characteristics of belt friction

both power transmission efficiency and wear resistance, it is preferable to keep reductions of belt clamping force within the range of elastic slip. This makes the issue to address changes in the coefficient of friction. The friction between metal V-belt and pulley is boundary friction. The coefficient of friction is affected by roughness of the frictional face and the shear strength of the boundary lubrication oil film, which is formed by lubricants⁽³⁾. Also, as these factors change with operating load, changes in the coefficient of friction over time should be factored in. Accordingly, when setting a belt clamping force, a margin is built in, assuming the minimum coefficient of friction of the conceivable usage cases. For a low-mileage CVT, this clamping force margin will be excessive.

Methods have been researched for learning the limits of belt clamping force with a vehicle in motion⁽⁴⁾⁻⁽⁷⁾. In all cases, however, sudden increases in belt slip are used to determine the limits for belt clamping force. While brief, this generates sliding slip, which raises concerns of abrasive wear advancing. Additionally, surging oil pressure and response lag could also result in adhesive belt wear.

Power transmission efficiency is maximized at the elastic slip limit [the boundaries in Fig. 3(A)-(B)]. Given the required tolerances for oil pressure surge and response lag, however, the clamping force margin cannot be completely eliminated in actual controls. Determining the safety factor at the moment the elastic slip limit is reached should allow for this clamping force margin to be minimized. This safety factor, used to reveal the actual clamping force margin, will be called the instantaneous safety factor.

The purpose of this study is to circumvent the sliding slip in estimating this instantaneous safety factor. In a previous study, a method was proposed for predicting the elastic slip limit⁽⁸⁾. This paper explores using predicted elastic slip limit to estimate instantaneous safety factor. Further, the proposed method is also applied on an actual vehicle for validation.

2. Modeling Slip and the Proposed Safety Factor Estimation Formula

2.1. Preconditions and Assumptions

The goal in this study is to increase power transmission efficiency while cruising (in a steady state). The test vehicle is powered by an internal combustion engine, and engine cylinder pressure fluctuation is input to the CVT. The vehicle is assumed to be driving straight ahead with no slip between the tires and road surface. The CVT shift state is assumed to be at a speed ratio of $i \leq 1$ (acceleration side, $i = N_{DR}/N_{DN}$), where i is a steady-state $i = \text{const.}$

Based on these preconditions, the following assumptions were introduced in developing a safety factor formula.

(i) Belt slip occurs on the driven side

Slip from elastic deformation of the belt (elastic slip) is the dominant factor in belt slip⁽²⁾.

Figure 4 represents schematically the change in belt elastic strain energy associated with pulley rotation⁽⁹⁾. When the belt is wound in range $\beta_{\text{DR,DN}}$, frictional force acts in range $\alpha_{\text{DR,DN}}$. The drive pulley spins the driven pulley by pushing the belt out. Compressive force here generates compression strain in the belt. In terms of energy, input torque from the drive pulley is stored in the belt as elastic strain energy E_1 . Meanwhile, when torque is transmitted, compression strain is released in the driven pulley, dissipating elastic strain energy E_2 . Thus, the CVT system as a whole only loses E_2 of energy per belt revolution. This loss is synonymous with frictional loss between belt and pulley. Assuming a flat distribution of frictional force in the pulley, E_2 is proportional to α_{DN} in $\alpha_{\text{DN}} \leq \beta_{\text{DN}}$. As torque input is increased, α_{DN} will be enlarged to reach the required frictional force. When $\alpha_{\text{DN}} = \beta_{\text{DN}}$, sliding slip will occur. From the above, an equivalence conversion is possible, where belt slip on the driven side occurs at the energy-equivalent average slip speed V . This equivalence is assumed as follows:

$$\begin{aligned} V &\propto E_2 \\ &\propto \alpha_{\text{DN}}/\beta_{\text{DN}} = T_{\text{DN}}/T_{\text{max,DN}} \end{aligned} \quad (1)$$

where $T_{\max, \text{DN}}$ is the maximum transmittable torque on the driven side and $T_{\text{DN}}/T_{\max, \text{DN}}$ is the usage rate of torque capacity.

- (ii) The left and right driveshafts can be represented by equivalent rigidity

As the vehicle is driving straight ahead with no slip between the tires and road surface, it is assumed that torsional rigidity k_{2L} , k_{2R} of the left and right driveshafts can be expressed as an equivalent rigidity k_2 in the following equation:

$$k_2 = k_{2L} + k_{2R} \quad (2)$$

- (iii) The wheels are considered fixed as a vibration transfer path

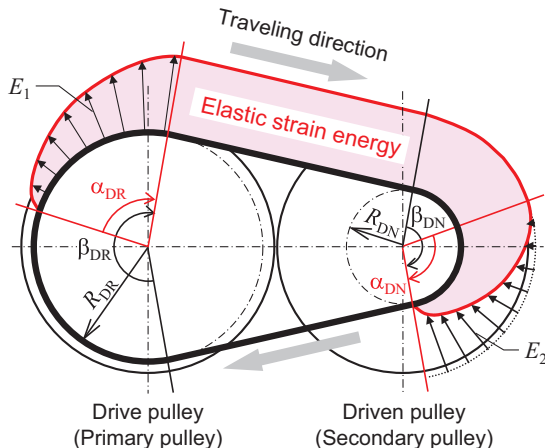


Fig. 4 Schematic diagram of transmitting power

As vehicle body weight is generally much greater than the rotating mass, it is assumed that irregularities in wheel rotation can be ignored.

2.2. Prediction of Elastic Slip Limit

In order to calculate the instantaneous safety factor, the elastic slip limit should be predicted in addition to estimating the slip state. In a previous study, the author proposed a method for predicting the elastic slip limit using only pulley speeds⁽⁸⁾. This method involved preparing a vibration model, taking into account friction attenuation from slip, and estimating attenuation parameters based on the measured pulley speeds. Thus, an appropriate vibration model for plant equipment is required. However, the previous research was only verified on a belt tester; the vibration model should be rethought for the proposed method to be applicable to actual vehicles.

The torsional rigidity of the mechanical parts comprising an actual vehicle powertrain is given in Fig. 5. In the figure, torsional rigidity for each mechanical part is shown in order of engagement and from engine to tires, from left to right. In the proposed method, the modeled range is set as being from drive pulley to the tires. This sets fluctuations in the drive pulley speed as the boundary condition, as will be described below.

In general, elements of torsional rigidity that are lower relative to the moment of inertia tend to contribute more to the natural frequency. In the interest of predicting the elastic slip limit, at the least belt rigidity will need to be considered.

Accordingly, the critical elements in setting a vibration model are as follows (red frame in Fig. 5):

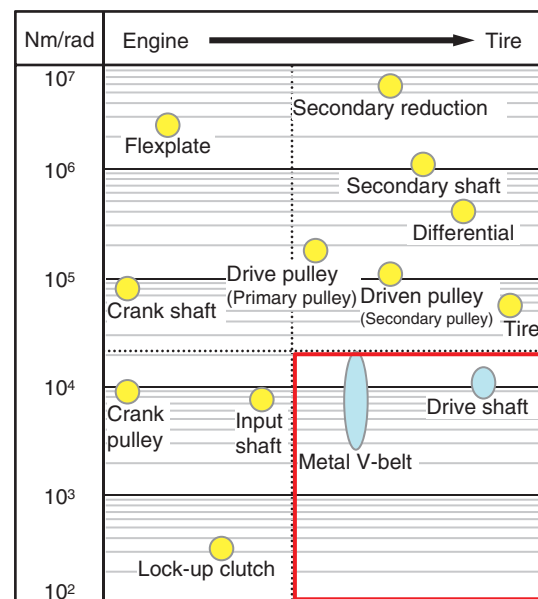


Fig. 5 Torsional rigidity of powertrain

- (i) Elements from drive pulley to tires
- (ii) Elements where torsional rigidity is equivalent to or less than belt rigidity

The belt and driveshaft were treated as torsion spring elements, and other elements were treated as rigid bodies. The vibration model as set is shown in Fig. 6.

With the assumptions outlined in the previous section, and a rotational perturbation θ around the operating point, the variational equations are as follows:

$$\frac{d}{dt} \begin{bmatrix} \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_3 \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -\rho_1^2 \omega_1^2 & -2v_1 \rho_1^2 \omega_1 \zeta_1 & 0 & 2v_1 \rho_1^2 \omega_1 \zeta_1 \\ 0 & 0 & 0 & 1 \\ 0 & 2v_1 \omega_1 \zeta_1 & -\omega_2^2 & -2v_1 \omega_1 \zeta_1 \end{bmatrix} \begin{bmatrix} x_2 \\ x_3 \\ x_3 \\ \dot{x}_3 \end{bmatrix} + \frac{\rho_1^2 \omega_1^2}{\Omega} \begin{bmatrix} 0 \\ a_1 \sin \Omega t \\ 0 \\ 0 \end{bmatrix} \quad (3)$$

Here,

$$x_2 = i\theta_2, x_3 = i\theta_3, i = R_{DN}/R_{DR}, i_a = i_{sec} i_{fin}$$

$$I_2 = WR_{DN}^3 \beta_{DN}, I_3 = I_{DN} + \frac{1}{i_{sec}^2} \left(I_{sec} + \frac{I_{fin}}{i_{fin}^2} \right)$$

$$v_1 = r_{DN}/R_{DN}, \rho_1 = \sqrt{I_3/I_2}$$

$$\omega_1 = R_{DN} \sqrt{\frac{k_1}{I_3}}, \omega_2 = \frac{1}{i_a} \sqrt{\frac{k_2}{I_3}}, \zeta_1 = \frac{c_1 Q_{DN} R_{DN}}{\sqrt{I_3 k_1} \cos \psi}$$

The operating point refers to the average value (expected value) of periodic solution $x = (x_2, \dot{x}_2, x_3, \dot{x}_3)$ for all frequency components (See Ref. ⁽⁸⁾ for exact derivation).

In Eq. (3), there are two unknowns: (ω_1, ζ_1) . All other parameters are either constants or calculable known entities. As the unknown ζ_1 expresses the impact of belt slip, slip

state can be evaluated by estimating ζ_1 . Thus, engine cylinder pressure fluctuations can cause periodic fluctuation in the drive pulley speed. Assuming this speed fluctuation as amplitude a_1 and angular frequency Ω , the speed fluctuation can be expressed as $\dot{x}_1 = \dot{\theta}_1 = a_1 \cos \Omega t$. This speed fluctuation input is expressed by the second term on the right in Eq. (3). Now, a speed fluctuation of $\dot{x}_3 = a_3 \cos(\Omega t - \phi)$ is transmitted to the driven pulley, yielding $a_3/a_1 = m$. Substituting $\dot{x}_3 = a_3 \cos(\Omega t - \phi)$ for Eq. (3) and solving for (ω_1, ζ_1) , (ω_1, ζ_1) can be expressed as (m, ϕ) .

$$\omega_1(m, \phi) = \Omega \frac{\left[\left(\frac{\Omega}{\omega_2} \right)^2 \left(1 + \frac{1}{\rho_1^2} \right) - 1 \right] \left[\left(\frac{\Omega}{\omega_2} \right)^2 - 1 \right]}{\sqrt{\left(\frac{\Omega}{\omega_2} \right)^2 \left[\left(\frac{\Omega}{\omega_2} \right)^2 - 1 \right] \left(1 - \frac{\cos \phi}{m} \right)}} \quad (4)$$

$$\zeta_1(m, \phi) = \frac{1}{2v_1} \cdot \frac{\omega_2}{\omega_1} \cdot \frac{\left[\left(\frac{\Omega}{\omega_2} \right)^2 - 1 \right] \left[1 - \frac{1}{\rho_1^2} \left(\frac{\Omega}{\omega_1} \right)^2 \right]}{\frac{\sin \phi}{m} \cdot \frac{\Omega}{\omega_2}} \quad (5)$$

Thus, the current slip state ζ_1 can be estimated using Eq. (5) to evaluate amplitude ratio m and phase lag ϕ from the observed fluctuation in the drive and driven pulley speed. In addition, Eq. (4) indicates that the unknown belt rigidity k_1 can be estimated as well.

The above method illustrates how to estimate current slip state ζ_1 . The value for ζ_1 at the elastic slip limit, which is ζ_c , can be shown unambiguously.

$$\zeta_c = \frac{\rho_1}{v_1} \left[\frac{\rho_1^4 + 20\rho_1^2 - 8 + \rho_1(\rho_1^2 - 8)^{\frac{3}{2}}}{32(\rho_1^2 + 1)^3} \right]^{\frac{1}{2}} \quad (6)$$

This can be achieved by solving the Jacobian matrix in Eq. (3) for its maximum real number ζ_1 . From Eq. (6), elastic slip limit threshold ζ_c can only be solved for known values. Therefore, if current slip state ζ_1 can be estimated, the margin levels can be predicted up to the elastic slip limit.

2.3. Estimation of Instantaneous Safety Factor

Transmitting power with no belt slip demands a belt clamping force capable of transmitting the input torque. In an actual vehicle, oil is supplied to the pulley piston chamber to clamp down on the belt. With oil pressure providing the control power, oil pressure surging and response lag should be accounted for in the control system design. With that, controls should anticipate a clamping force margin, and safety factor s_f is defined by the following equation:

$$s_f = \frac{T_{\max}}{T} = \frac{Q}{Q_{\min}} \quad (7)$$

Here, $T_{\max}$ is the maximum transmittable torque for currently applied belt clamping force Q (axial thrust of the pulley), and $Q_{\min}$ is the minimum transmittable belt

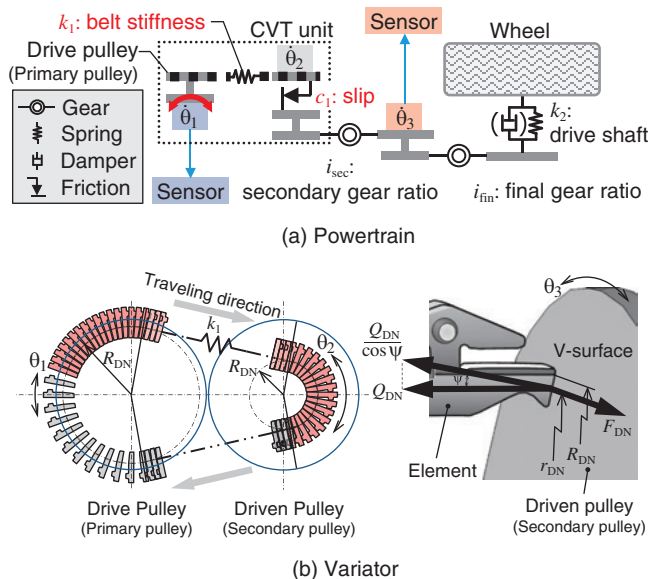


Fig. 6 Simple model of transmitting power

clamping force for current input torque T . “Transmittable” here refers to power transmitted within the elastic slip range. With this paper’s criteria, safety factor s_f was solved for the driven side as sliding slip occurs at the driven pulley.

From the previous section, the elastic slip limit at the point of slip velocity can be determined by comparing ζ_1 and ζ_c . However, the clamping force margin for the elastic slip limit cannot be calculated directly. As such, we will attempt to express safety factor s_f as a function of ζ_1 using the known characteristics, which are:

- (i) At $\zeta_1 = \zeta_c$, $s_f = 1$ (the elastic slip limit)
- (ii) $\zeta_1 \propto T_{DN}/T_{max,DN} = 1/s_f$ (From the assumptions in Eq. (1) and that ζ_1 correlates to slip velocity)

Graphing the above relationships yields Fig. 7. Sample test results are also plotted on Fig. 7. As ζ_c is known from Eq. (6), the relationship in Fig. 7 can be represented using ζ_1 in the following form:

$$s_f(\zeta_1) = \frac{1}{K_\tau(\zeta_1 - \zeta_c) + 1} \quad (8)$$

Here, K_τ is a coefficient. As the unloaded slip velocity will depend on the belt structure, the safety factor was estimated by first calculating K_τ with Eq. (8) for the belt used (with different specifications).

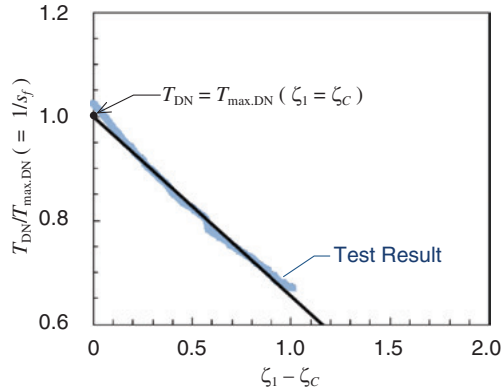


Fig. 7 Relationship between torque capacity and ζ_1

3. Testing Method

The previous section discussed a method for estimating instantaneous safety factor using the fluctuation in the drive and driven pulley speed. In order to verify this method as valid, instantaneous safety factor as measured in a moving test vehicle was compared with estimates. The test vehicle was configured as given in Fig. 8. The test vehicle used was a front-engine, front-wheel-drive vehicle. The power transmission path from engine to CVT consisted of a torque converter and planetary gear mechanism, through which power could be cut or switched to drive either forward or backward. Power output from the CVT was passed through

a two-stage deceleration mechanism with secondary and final reduction, then transmitted to the wheels.

CVT operation is represented by the maximum driven transmittable torque $T_{max,DN}$ with belt clamping force Q_{DN} applied to the driven pulley, which is given by the following equation:

$$T_{max,DN} = \frac{2\mu_c Q_{DN} r_{DN}}{\cos\psi} \quad (9)$$

Here, μ_c is the coefficient of friction at the elastic slip limit, r_{DN} is the effective radius of friction at the belt winding position, and ψ is the semi-vertical angle of the V-groove. From its definition and substituting Eq. (9) in Eq. (7), the safety factor is:

$$s_f = \frac{2\mu_c Q_{DN} r_{DN}}{T_{DN} \cos\psi} \quad (10)$$

Current load torque T_{DN} and belt clamping force Q_{DN} for the driven pulley were measured and substituted into Eq. (10) to solve for s_f , which was then treated as the measured instantaneous safety factor. The effective radius of friction r_{DN} can be solved geometrically from current speed ratio i ($= N_{DR}/N_{DN}$). Given the challenges in directly measuring driven pulley load torque T_{DN} , torque $T_{L,R}$ acting on the left and right driveshafts was measured. The torque, as measured with a strain gauge, was converted to driven pulley load torque T_{DN} using the following equation:

$$T_{DN} = \frac{T_L + T_R}{i_{sec,fin} \eta_{sec,fin}} \quad (11)$$

$i_{sec,fin}$ is the reduction mechanism gear ratio, and $\eta_{sec,fin}$ is the gear transmission efficiency. $\eta_{sec,fin}$ was 0.99 for both. Belt clamping force Q_{DN} was calculated by measuring the oil pressure as supplied to the piston chamber and adding the centrifugal oil pressure component. The coefficient of friction μ_c was pre-measured for the belts and pulleys before installing them in the test vehicle.

The test conditions are given in Table 1. Tests were conducted while cruising at speeds of 70-100 km/h. Thus,

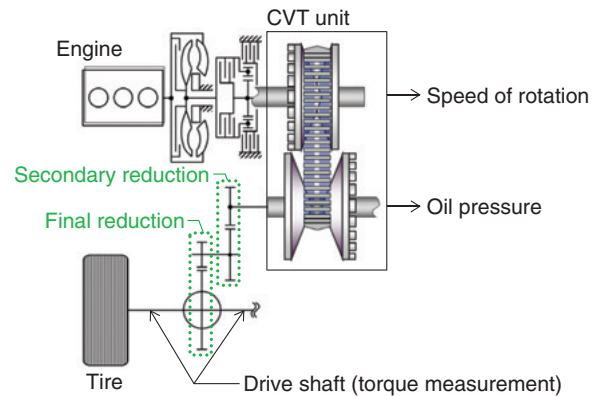


Fig. 8 Schematic view of test vehicle

Table 1 Test conditions

Vehicle speed	70, 80, 90, 100 km/h
Speed ratio, i	0.468
Vehicle body weight	1440, 1651 kg
Differential diameter rate, ε_{LR}	0.9-1.1 (used chassis dynamometer)
Oil temperature	85-105 degC

speed ratio on the acceleration side was nearly constant. Vehicle body weight was 1,440 kg when empty and 1,651 kg at seating capacity. The vehicle was warmed up before test driving.

Also, while straight driving is assumed for this study, note that a differential between left and right wheels could occur even while driving straight if the rolling radius of the left and right tires differs. As this differential could impact estimation accuracy, the test vehicle driving wheels were secured on a chassis dynamometer, and the left and right wheel rotation speed was changed. With a rolling radius $R_{L,R}$ for the left and right wheels, the differential diameter rate ε_{LR} between left and right is defined by the following equation:

$$\varepsilon_{LR} = \frac{R_R}{R_L} \quad (12)$$

Here, rotating speed $N_{L,R}$ for the left and right wheels changes according to differential diameter rate ε_{LR} as follows:

$$\begin{aligned} N_L &= \frac{2\varepsilon_{LR}}{1 + \varepsilon_{LR}} \cdot \frac{N_{DN}}{i_{sec}i_{fin}} \\ N_R &= \frac{2}{1 + \varepsilon_{LR}} \cdot \frac{N_{DN}}{i_{sec}i_{fin}} \end{aligned} \quad (13)$$

Where N_{DN} is the driven pulley rotating speed. For cases where a spare tire is equipped, such as with a flat tire, a left-right wheel differential of $\varepsilon_{LR} = 0.9-1.1$ was simulated.

4. Test Results and Discussion

With the vehicle cruising at a speed of 70-100 km/h, slip state ζ_1 was calculated based on Eq. (5). The relationship between ζ_1 and the measured safety factor at ζ_1 is shown in Fig. 9. The standard test conditions (Empty vehicle at 1440 kg, no left-right wheel differential: $\varepsilon_{LR} = 1$) from Table 1 were used. From the results of Fig. 9, K_τ in Eq. (8) was determined with the least squares method. As approximated with Eq. (8), the results were superimposed with a solid line in Fig. 9. Instantaneous safety factor was estimated based on this K_τ and ζ_1 , which was calculated from the speed fluctuation while driving.

Estimation error for safety factor as estimated with the proposed method under the above standard conditions is shown in Fig. 10. Safety factor estimation error ε_{sf} was defined as follows:

$$\varepsilon_{sf} = \frac{s_{fest} - s_f}{s_f} \quad (14)$$

where s_{fest} is the estimated safety factor value and s_f is the measured value. Estimation error was broken down into error from vehicle speed [Fig. 10(a)] and error from the safety factor range as measured [Fig. 10(b)]. Looking at Fig. 10(a), while the maximum observed estimation error was roughly 30%, no correlation was found between estimation error and vehicle speed. Meanwhile, in Fig. 10(b), estimation error did tend to decrease as safety factor approached 1. Presumably, this suggests that estimation is easier as the safety factor decreases because belt slip (i.e. variation in pulley speed) increases. Estimation error was 15% or lower for the $s_f \leq 1.67$ range ($1.50 \leq s_{fest} \leq 1.93$) and 2% or lower for the $s_f \leq 1.25$ range ($1.22 \leq s_{fest} \leq 1.24$). At the least, this confirms that, in the $s_f \leq 1.67$ range, safety factor can be estimated up to the elastic slip limit without involving sliding slip.

However, note that the above estimation accuracy is for identifying K_τ for the standard conditions. In terms of method verification, it remains to be confirmed whether the above estimation accuracy can withstand changes in vehicle conditions. Once again, the assumed vehicle conditions used in deriving the proposed method are:

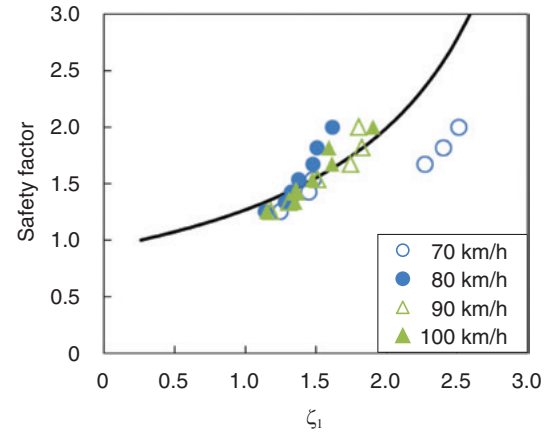
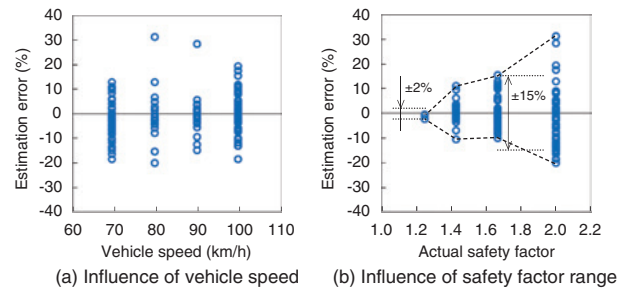
Fig. 9 Relationship between safety factor and ζ_1 

Fig. 10 Accuracy of estimating instantaneous safety factor

- (i) The left and right driveshafts can be represented by equivalent rigidity
 - (ii) The wheels are considered fixed as a vibration transfer path
- (i) is satisfied with no difference between left and right wheels, and (ii) is satisfied at sufficiently large vehicle body weights.

For (i), driveshaft rigidity for front-engine, front-wheel-drive vehicles generally differs from left to right (left and right are of unequal lengths).

With a left-right wheel differential, equivalent stiffness k_2 will change (Fig. 11). In Fig. 11, the top chart shows the rate of change for equivalent stiffness k_2 against differential diameter rate ε_{LR} , and the bottom chart shows the predicted values for estimation error as calculated from error propagation. In the range of $\varepsilon_{LR} = 0.9$ -1.1, estimation error is expected to be 4% or lower, which is negligible. Estimation error was similarly predicted for assumption (ii) (Fig. 12). The predicted values for estimation error when vehicle body weight is changed are shown in Fig. 12. At vehicle body weights of 400 kg and higher, the projected estimation error was similar to (i) at 4% or lower. Body weights for the test vehicle were 1440 kg (when empty) and higher. As such, the impact of changes in vehicle body

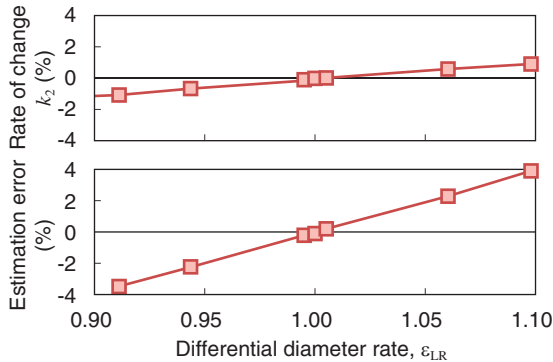


Fig. 11 Predicted influence of equivalent torsional rigidity of drive shaft

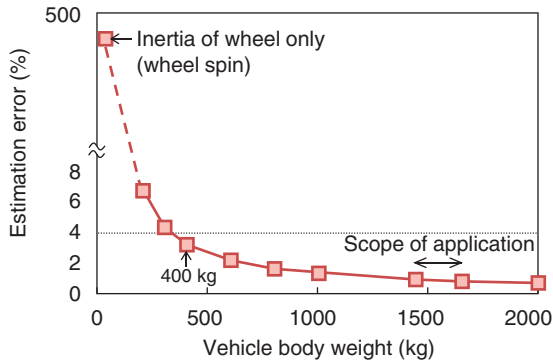


Fig. 12 Predicted influence of vehicle body weight

weight on estimation error can presumably be ignored. From the perspective of the rotating and vibrating systems, reduction in vehicle body weight is synonymous with wheel spin, where the wheels slip on the road. Figure 12 illustrates that estimation error increases sharply as vehicle body weight approaches the wheel unit mass from 400 kg. While pointless to reduce belt clamping force during wheel spin, care should be taken when constructing the controls to help prevent malfunctions.

The test results for assumptions (i) and (ii) are given in Fig. 13. Figure 13(a) shows the estimation error with the weight of the vehicle at nominal riding capacity (1651 kg). Figure 13(b) shows the estimation error with a differential diameter rate of $\varepsilon_{LR} = 0.9, 1.1$. In the figures, the dashed lines represent the maximum and minimum values (linear interpolated) for estimation error with the standard conditions [from Fig. 10(b)]. For both vehicle body weight and differential diameter rate, the estimation error falls within the dashed lines, confirming that estimation accuracy remains equivalent to the standard conditions. Focusing on the $s_f \leq 1.25$ range, where the risk of sliding slip is increased, the estimation accuracy was quite high, with an error range of 2% or less. Also, with left and right wheels of equal rolling radius, the lateral acceleration was 23.5 m/s^2 (2.4 g), corresponding to a differential of $\varepsilon_{LR} = 0.9, 1.1$. From this, instantaneous safety factor can presumably be estimated for cornering, assuming normal cornering under standard conditions.

To this point, the discussion on validation has concerned the aspect of changes in the vehicle. Another aspect is validation for change in coefficient of friction. In order to simulate how coefficient of friction μ_C declines over time, a test V-belt pulley with mirror-finished groove surfaces was prepared. The test vehicle was mounted with the test pulley and then subjected to the same tests again. The test results are given in Fig. 14. Figure 14(a) shows coefficient of friction μ_C for the test pulley. The mirror-finished surfaces reduced coefficient of friction μ_C by approximately 8%. Figure 14(b) shows the estimation error with the reduced coefficient of friction μ_C . In the graphs, the dashed line

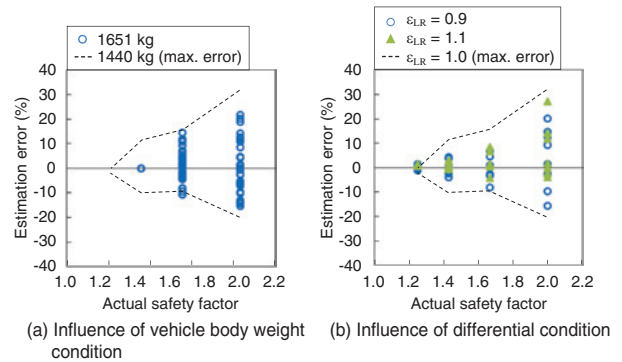


Fig. 13 Verification test results of estimating instantaneous safety factor

represents the estimated error range for the standard conditions. At $s_f = 1.67$, while the resulting estimation error does increase past that for the standard conditions at some points, there is sufficient marginal clamping force at these points. At $s_f = 1.43$, where the risk of sliding slip is elevated, estimation accuracy was confirmed to be equivalent to that under standard conditions. There were also no positive or negative offsets for the estimation error range.

As shown above, estimation accuracy was confirmed to remain constant, independent of changes in vehicle state or coefficient of friction. In the required estimation accuracy range of $s_f \leq 1.25$, this verifies that instantaneous safety factor can be estimated to within an error range of 2%.

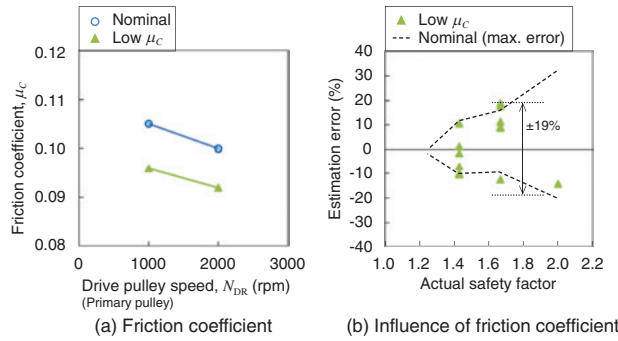


Fig. 14 Changes in estimated instantaneous safety factor over time

5. Conclusion

We have proposed a method for estimating instantaneous safety factor using only the fluctuation in the drive and driven pulley speed. With the proposed method verified as valid for cruising, we obtained the following results:

- (1) Instantaneous safety factor can be estimated independently of changes in vehicle status (vehicle body weight and left-right wheel differential) and coefficient of friction.
- (2) In the required estimation accuracy range of $s_f \leq 1.25$, estimations were within an error range of 2%.

In future studies, we aim to estimate instantaneous safety factor under sudden acceleration and other transient states. It will be important to validate responsiveness for these cases as we continue to expand on our estimation theory.

References

- (1) van der Sluis, F., Yildiz, S., Brandsma, A., Veltmans, P., Kunze, M.: The CVT Pushbelt Reinvented for Future Compact and Efficient Powertrains, 2017 JSAE Annual Congress Proceedings, p. 245-252, (2017)
- (2) Sakagami, K.: Analysis of Slip in Metal V-belt Type

CVT, Transactions of Society of Automotive Engineers of Japan, Vol. 49, No. 2, p. 290-295, (2018) (in Japanese)

- (3) Luengo, G., Israelachivili, J., Granick, S.: Generalized effects in confined fluids: new friction map for boundary lubrication, Wear 200(1-2): p. 328-335, (1996)
- (4) van der Sluis, F., van Dongen, T., van Spijk, G., van der Velde, A., van Heeswijk, A.: Efficiency Optimization of the Pushbelt CVT, SAE Technical Paper, 2007-01-1457, (2007)
- (5) van der Noll, E., van der Sluis, F., van Dongen, T., van der Velde, A.: Innovative Self-optimising Clamping Force Strategy for the Pushbelt CVT, SAE Int. J. Engines 2(1), p. 1489-1498, (2009)
- (6) Oshiumi, Y., Iwatsuki, K., Nishizawa, H., Yamaguchi, H.: Optimal Control of Clamping Force for Metal V-belt CVT, 2005 JSAE Annual Congress Proceedings, No. 8-05, p. 11-14, (2005) (in Japanese)
- (7) Nishizawa, H., Yamaguchi, H., Suzuki, H., Osawa, M., Iwatsuki, K., Oshiumi, Y.: Friction Characteristics Analysis for Clamping Force Setup in Metal V-Belt Type CVT, SAE Technical Paper, 2005-01-1462, (2005)
- (8) Sakagami, K.: Prediction of Friction Drive Limit of Metal V-Belt, SAE Int. J. Engines 8(3), p. 1408-1416, (2015)
- (9) Sakagami, K.: Kinzoku V Beruto Shiki CVT no Masatsu Dendo Genkai ni kansuru Kenkyu, Doshisha University, Doctoral dissertation, p. 70-73, (2018) (in Japanese)

List of Symbols

$a_{1,3}$	Amplitude of speed fluctuation (1: drive pulley; 3: driven pulley)
c_1	Damping factor with belt slippage
$F_{DR,DN}$	Tangential friction force on the V-surface (DR: drive pulley; DN: driven pulley)
i	Speed ratio, N_{DR}/N_{DN}
i_{fin}	Gear ratio of final reduction
i_{sec}	Gear ratio of secondary reduction
$I_{2,3}$	Moment of inertia
I_{DN}	Moment of inertia of driven pulley
I_{fin}	Moment of inertia of differential
I_{sec}	Moment of inertia of secondary shaft
k_1	Compression stiffness of belt
k_2	Equivalent torsional rigidity of drive shaft
$k_{2L,R}$	Torsional rigidity of drive shaft (L: left side; R: right side)
m	Amplitude ratio, a_3/a_1
$N_{DR,DN}$	Speed of rotation of pulley (DR: drive pulley; DN: driven pulley)
$N_{L,R}$	Speed of rotation of wheel (L: left wheel; R: right wheel)
$P_{DR,DN}$	Applied oil pressure for pulley thrust (DR: drive pulley; DN: driven pulley)
$Q_{DR,DN}$	Pulley thrust (DR: drive pulley; DN: driven pulley)

$Q_{\min,DR,DN}$	Minimum pulley thrust for transmitting load torque on pulley (DR: drive pulley; DN: driven pulley)
$r_{DR,DN}$	Effective radius of friction (DR: drive side; DN: driven side)
$R_{DR,DN}$	Belt pitch radius (DR: drive side; DN: driven side)
$R_{L,R}$	Effective rolling radius (L: left wheel; R: right wheel)
$T_{DR,DN}$	Load torque on pulley (DR: drive pulley; DN: driven pulley)
$T_{\max,DR,DN}$	Maximum transmittable torque on pulley (DR: drive pulley; DN: driven pulley)
s_f	Safety factor
V	Mean slip velocity between belt and pulley
W	Linear density of belt
$\alpha_{DR,DN}$	Active arc (DR: drive pulley; DN: driven pulley)
$\beta_{DR,DN}$	Belt wrap angle (DR: drive pulley; DN: driven pulley)
ε_{LR}	Differential diameter rate
ε_{sf}	Relative error of estimating safety factor
ϕ	Phase lag
η	Power transmission efficiency
μ_c	Mean friction coefficient on the V-surface of pulley
θ_1	Degree of perturbation of angle of rotation of drive pulley
θ_2	Degree of perturbation of angle of rotation of elements wrapped around driven pulley
θ_3	Degree of perturbation of angle of rotation of driven pulley
$\omega_{1,2}$	Natural angular frequency
Ω	Excitation angular frequency
Ψ	1/2 the V angle
ζ_1	Quantity of slip state
ζ_c	Critical state quantity (elastic slip limit)

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Editorial Afterword

This year, Japan made the transition from the Heisei era to the Reiwa era. The Honda R&D Technical Review commenced publication in the first year of the Heisei era as a journal that would provide vindication of Honda's unmatched ingenuity. At first, our ambition outpaced ability, but the Technical Review has grown since those days, and the publication now boasts 33,328 registered subscribers, and is also distributed to approximately 600 institutions and organizations. I would like to take this opportunity to thank everyone who has been involved with the Technical Review up to this point for their tireless efforts.

Today, however, the number of applications for publication in the Technical Review is unfortunately tending to decline. In their daily activities, many people are satisfied with presentation materials that are easily intuitively understandable, and I feel that interest is declining in research papers and dissertations. At the same time, though, we are in a period of transition both globally and locally, and we often encounter situations that fundamentally overturn our commonsense attitudes. This is an era in which we are required to produce concepts from scratch. It is surprisingly challenging to communicate such new concepts and the processes by which they will be implemented, but I strongly feel that publication in the form of technical papers is an excellent means of accurately transmitting a large amount of related information in a short period. Nevertheless, mastery of this method necessitates a good deal of training, for example in reading, writing, and thinking philosophically. We may consider this an ongoing task for the future in our quest for unmatched ingenuity.

As in the case of the change from the physical publication to publication as an e-book in the Heisei era, the form in which the Technical Review will be transmitted will change in the Reiwa era in ways that we cannot imagine. I am resolved, therefore, to enjoy the changes that the new era will bring to both the content of our papers and the media by means of which that content is transmitted.

Yuta URUSHIYAMA

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