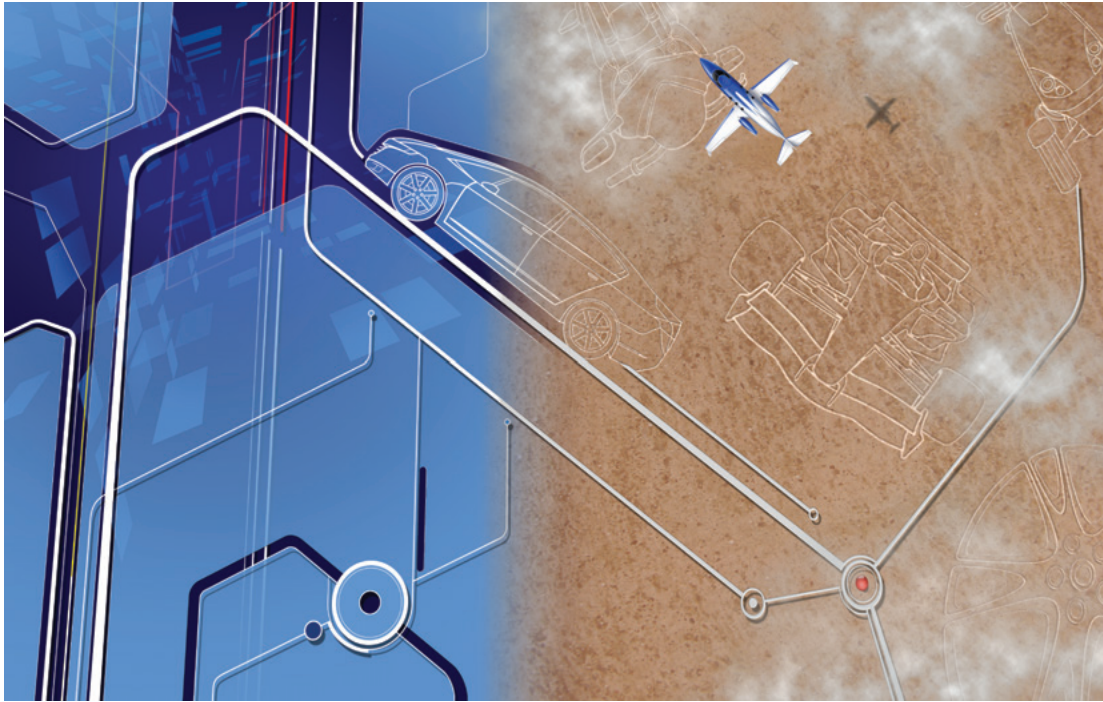


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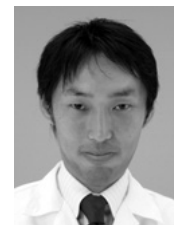
Technical Review

October 2008

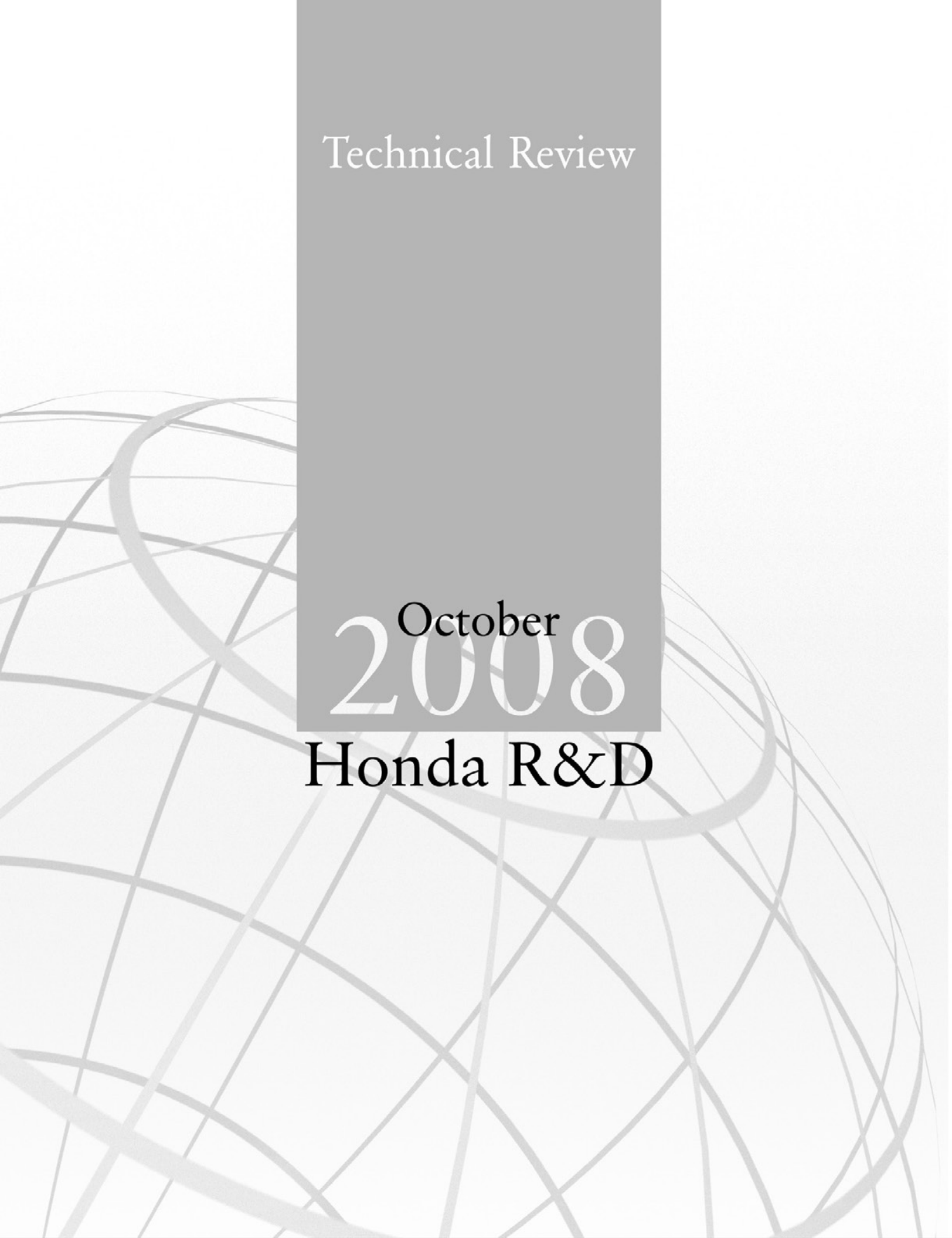


Comment on the Cover

Manufacturers have the satisfaction of leaving a mark on the world by sharing their dreams and achievements with posterity. The better one's achievements, the more they will be appreciated, and the longer people will remember them. Pioneers at Honda have passed on our automotive heritage. Now it's our turn to chart a course for the future. The fanciful geoglyphs on the cover symbolize this ongoing evolution at Honda, in an aerial view extending from the past to the future.



Motorcycle R&D Center
Hiroshi TANAKA



Technical Review

October
2008
Honda R&D

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Style and Design of New Accord for U.S. Market

Daisuke SAWAI* Taichi HAGIWARA*
Hideki TAGAMI* Keiichi ARAKI*

ABSTRACT

Honda sedans have always looked “Advanced” and “Sporty.” Continuing to build upon this theme, the new Accord was developed with the concept keywords of “Advanced & Powerful” to fit with the time. We sought to set a new standard in midsize sedans, combining the reliability and social nature required for a sedan with an unmistakable individuality.

For the exterior, the Accord was developed to assert a sense of presence that is stately, as well as being dynamic and powerful. For the interior, it is an invitation to enjoy a comfortable and refreshing environment infused with premium luxury.

1. Introduction

Currently in its eighth generation in the North American market and gaining stronger presence in Asia every year, the Accord is a global model with annual sales of 700 000 units. It is also marketed in Japan as a luxury sedan with modified specifications, under the name “INSPIRE.”

A longstanding tradition of solid reliability has won widespread support and earned high customer satisfaction. But with intensified competition in this segment and emergence of new categories such as SUVs and large minivans, we saw a need for the Accord’s clear evolution and a greater market presence.

In the U.S. market, the Accord is a benchmark model in the midsize sedan category. It is perceived as a family car, but analysis of the main customer segment revealed a surprising fact that less than 30% of Accord owners were actually “Family” (husband, wife, and kids), showing a change in the user demographics.

Meeting with many owners and carefully investigating usage scenarios and preferences led us to redesign the package from a concept level.

It has evolved from a family car to a personal sedan, with social presence like that of a stylish business suit. In addition to improved economic efficiency and amenities, the Accord has evolved into a more premium vehicle.

2. Package Layout

A personal sedan is mainly used for commuting. Many owners commute on freeways over one hour each way. This consideration motivated us to put together a human-centered package, an environment where people can spend many hours in comfort and drive long distances without undue stress. The A-pillar was shifted backward and its section was made thinner to enhance visibility. Having the hood in sight also gives the driver a better spatial sense of the vehicle. Also, drivers of this sedan class tend to

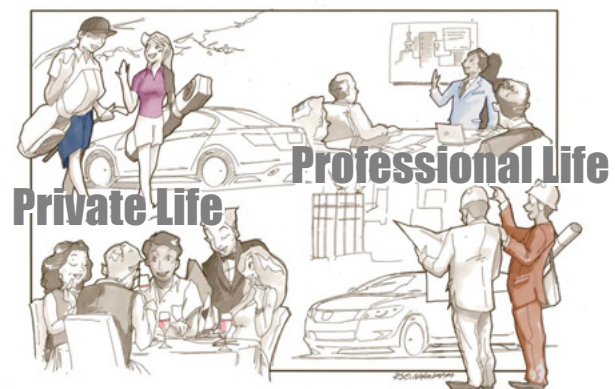


Fig. 1 Usage scenarios

* Automobile R&D Center

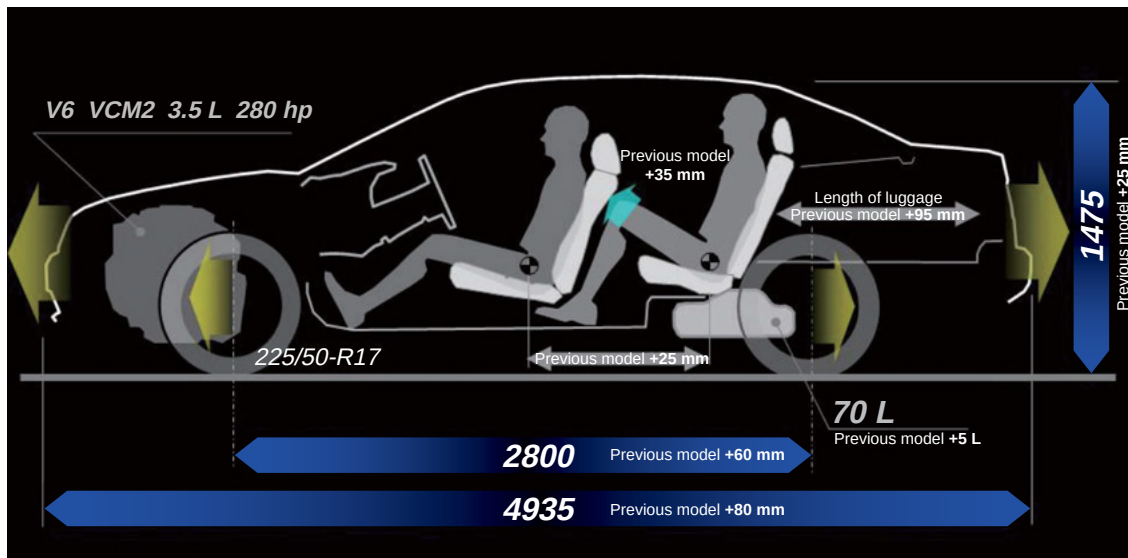


Fig. 2 Package layout

look beyond their own comfort. A pleasant environment for the front passenger and rear seat passengers is also important, because they frequently dine out with another couple or take clients around (Fig. 1).

We thought a cabin should offer an ample hospitality space. 17-inch tires and bigger fuel tank capacity (65 L to 70 L) are featured, yet a spacious cabin that is typical of an FF is further enlarged with a lower floor layout.

The front couple distance (space between the right and left passengers) is enhanced by 40 mm, while the center console was enlarged by 75 mm, leading to a notch above vehicle class with upscale feeling. The rear seat's knee space was increased by 35 mm, making it comparable to minivans and SUVs. The Accord is an invitation to enjoy a sedan in new ways (Fig. 2).

3. Exterior Design

The exterior design, inspired by the “Advanced & Powerful” theme, signals the Accord’s evolution.

We insisted the overall framework to have a sedan-inherent frame, with an independent nose, cabin, and trunk. We strove to create a ram-rod straight posture as if the spine is pierced through the vehicle, from front to rear.

The CIVIC’s front pillar is moved forward to



Fig. 3 Side view

emphasize its original form, but the Accord’s front pillar is moved backward to emphasize a solid sedan frame. The different roles of the CIVIC and Accord are reflected in their frames (Fig. 3).

We intentionally kept the tip of the nose lower in the past generations of Accord, but the new model features a more substantial, aggressive nose and boldly chiseled corners, projecting the image of a solid sedan that is also sporty.

A bigger cabin may make the vehicle appear somewhat boxy, but we actualized a three-dimensional and nimble appearance. The roofline was rounded and tapered to create a three-dimensional shape, with an added benefit of reduced air resistance. Additionally, deep-drawn fenders and outer panels accentuate the width and create good stance, as all four tires grip the ground (Fig. 4).

The luxurious body panels were developed under the outside light. We aimed to make the vehicle appear more three-dimensional with reflections of the surrounding

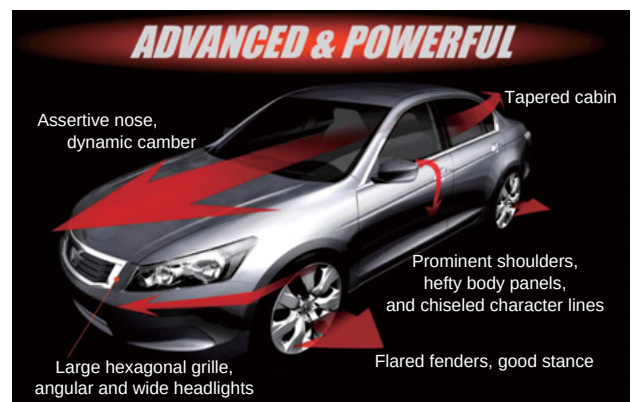


Fig. 4 Advanced, powerful exterior design

scenery. Especially, the hood surface was inflected upward to reflect the gradations of beautiful sky to give a more expressive appearance. Likewise, the shoulders also reflect colors of the sky this way, reinforcing its strong flare shape. Bold, chiseled-looking character lines are featured to express its sharpness in any scenes, be it under a strong sunlight or cloudy and gloomy sky.

The dynamic hexagonal grille and headlights with horizontal motif make the front of the vehicle appear wider; by doing so, we aimed to create a fearless expression (Fig. 5).



Fig. 5 Accord sedan exterior

4. Interior Design

The interior design concept is “Advanced & Comfort.” The aim is to have a combination of an advanced driver’s cockpit, as well as a spacious, exhilarating cabin space in a notch above, premium space. First, to make the improved cabin space even bigger and more comfortable, we ensured the instrument panel’s shape to accentuate a horizontal motif. Along the instrument panel, decoration panels were added from the center, spreading outward, to enhance spaciousness. Further, we constructed the upper dash with bold negative surfaces, while the middle pad is lowered down to reduce a sense of heaviness, thereby actualizing unprecedented exhilarating space (Fig. 6).

The front couple distance was widened, enabling a wider center console and an arm rest. Besides having a sense of spaciousness, it offers a more upscale image. A wider armrest provides enhanced functions with cup-holders and storage space. More luxurious and comfortable cabin space is therefore attained (Fig. 7).

The interface concept was based on instant recognition and intuitive operation, so the displays are placed as high as possible, and operation controls are arranged as close as possible to the driver.

The Progressive Commander is used to enable centralized controls at hand. The display location is higher and away from the driver so as to reduce the eye movement while driving. The advanced cockpit with enhanced visibility and operability is thus actualized (Fig. 8).

Indicators are placed in large, easy-to-see dual clusters, creating a neatly organized appearance yet showing our commitment to quality material and authenticity. Extra lines around the center panel and console are minimized to show a bigger surface area, which in turn creates good

presence. We made a round, three-dimensional shaped seat design. This creates a snug feeling as if one is comfortably wrapped, at the same time giving a refreshing impression from the rear seats (Fig. 9)

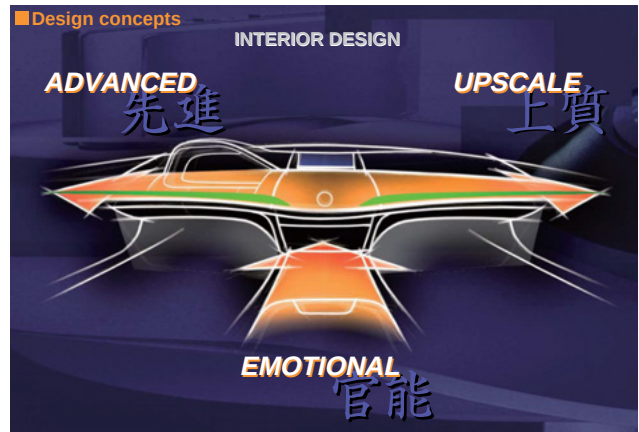


Fig. 6 Advanced, comfortable interior design



Fig. 7 Instrument panel design



Fig. 8 Accord sedan instrument panel



Fig. 9 Accord sedan interior

5. Color Design

5.1. Exterior

The vehicle exterior has good presence, but to further enhance this, a bold and aggressive color line-up and new color development were set out.

In the new color development, under the “Bold Metal & Character” theme, we expressed strength with tough metallic tones. All colors have the common texture theme of bright and intensely shining particles, as if the vehicle were carved out of a chunk of metal and polished. Under this color line-up proposal, the entire variations offer collective strength.

The newly developed gray metallic color, Polished Metal Metallic, was created to show contrasting shades as in metal, to make the great body shape more three-dimensional. Bordeaux Beige Metallic, an unconventional beige color, was created with full-bodied gold shine.

Basque Red Pearl with mellow richness and shading was also newly developed to match the all-new Accord (Fig. 10).

5.2. Interior

To express the interior design concept of “Advanced & Comfort,” we placed an emphasis on an upscale and high-quality image.

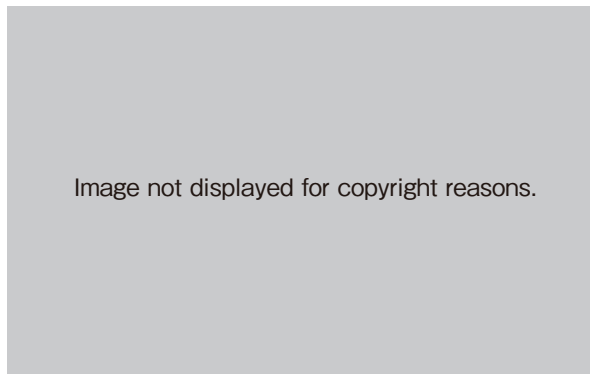


Fig. 10 Exterior colors



Fig. 11 Interior colors

The color line-up offers newly developed modern gray in addition to black and beige. By offering each an individual theme and character, we are able to respond to a wide range of customer preferences. The black interior is combined with bright silver decorations to give a sporty and cool impression, while a warm beige tone sets a rich ambience with luxurious wood grain decorations. The gray interior color is finished in a brighter hue to enhance an exhilarating and upscale image (Fig. 11).

6. Coupe Styling and Design

The Accord has a two-door derivative model (a U.S. focused model, some of which is exported to Middle East). The new coupe reinforces the role of being the image leader for the Accord. To survive in the shrinking coupe market, we thought a more attractive model than ever was needed, thus we reduced the number of common parts for the exterior, and evolved the coupe design into a more original and distinctive direction. Difference from the sedan is emphasized, yet at the same time, we tried to strike a good balance by keeping the Accord family linkage (Fig. 12).

The wheelbase is 60 mm shorter than the sedan to show case its dynamic performance frame, and to highlight the characteristic of this coupe being a personal car. Lower driving position and redesigned rear seat space embody a dynamic form with sleek cabin. The exterior highlight of this coupe is its flared shoulder in contrast with an intensely tapered cabin. It was a designer's chance to demonstrate his ability in creating a surface construction that lithely changes, like well developed muscles supporting the legs of a wild animal. Additionally, the manufacturing division's superb press forming technology enabled the creation of thick fenders and outer panels. Its thick, integrated bumper gives the

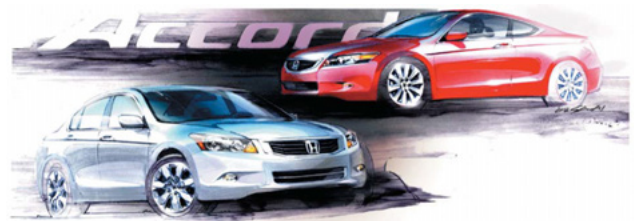


Fig. 12 Sedan and coupe



Fig. 13 Accord coupe

front-end a stronger presence, and the projector headlight creates a sharp and virile expression while keeping its “family connection” with the sedan. Compared to a usability-focused sedan, the coupe’s rear-end focuses more on styling with sporty design cues, such as the bumper shape that reinforces a sense of integration (Fig. 13).

7. Accord around the World

The Accord is becoming an important model in Asia as well. Asia has different values towards automobiles, and this model is positioned as a luxury vehicle. In some regions, the Accord may even be priced similarly as a house. The larger body size leading to a more upscale image was in part due to the Accord’s role in Asia. For this reason, excessive and decorative changes were not added, and only small modifications were applied to comply with the regional regulations. However, the vehicle accessories were enhanced with items including clear combination lamp lens covers and an additional rear fog light garnish.

The interior, with an aim to be more spacious and exhilarating, has the basic frame, but the materials are changed and amenities are added to create characteristics with enhanced upscale ambience.

In the Japanese market, it is sold under the name “INSPIRE” same as its predecessor.

For the interior, exclusive color coordination is introduced. The black interior uses materials with some colors to create more dimensions. For the exterior, a special black with some colored highlights (Premium Blackish Pearl) is added to the line-up to express a sophisticated glow (Fig. 14).



Fig. 14 INSPIRE

8. Conclusion

The new Accord is developed to raise the appeal of a sedan to its highest level. There are some regional differences in specifications and naming, but we believe the finished model offers luxury to its customers around the world.

The cabin space and exterior body are made bigger, yet the fuel economy is improved due to reduced air resistance and application of technology such as the VCM⁽¹⁾.

It may require some time to judge whether the direction of this new Accord conforms to the ever-changing global environment and economic conditions. We hope that the relationship of trust Honda has built with its customers over many generations will further deepen through this new Accord.

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■ Author ■



Daisuke SAWAI



Taichi HAGIWARA



Hideki TAGAMI



Keiichi ARAKI

Style and Design of FREED

Akihiko OHHASHI*

Ohsuke KOIKE*

Akihiro HATTORI*

Eiko HASHIMOTO*

ABSTRACT

Developed in pursuit of “Free life creation”, the FREED might be described as the world’s smallest full-scale minivan. While keeping the vehicle both convenient for city drivers and relatively affordable, we have succeeded in combining the roominess of larger utility minivans with the styling of sporty wagons.

The dynamic exterior fuses the frames of a utility minivan (such as the STEPWGN) and a multi-passenger vehicle (such as the STREAM) in a shape combining a triangle and square enveloped in graceful curving surfaces.

Inside, visual cues around the cabin create the illusion of greater space affording a sense of comfort, convenience, and expansiveness. The stimulating ambiance of an open-air café is captured in an enclosed space for a pleasant and enjoyable interior environment.

1. Introduction

When the MOBILIO was introduced in 2001, it offered consumers the benefits of a minivan with three rows of seats in a short body only 4.07 m long. Beyond the groundbreaking concept, the MOBILIO represented revolutionary technology and design, and this winning combination of features was soon emulated by other automakers.

But our commitment to a shorter overall length also

limited the vehicle’s spaciousness which seemed less generous than that of conventional minivans. This limited our opportunity to provide passengers all the advantages of a compact minivan. Our desire to take this concept to the next level inspired the idea of creating the smallest full-scale minivan that comfortably accommodates three rows of seating for adults (about 185-cm tall) in a compact space (Fig.1).

Figure 2 shows the development concept and design of the FREED.

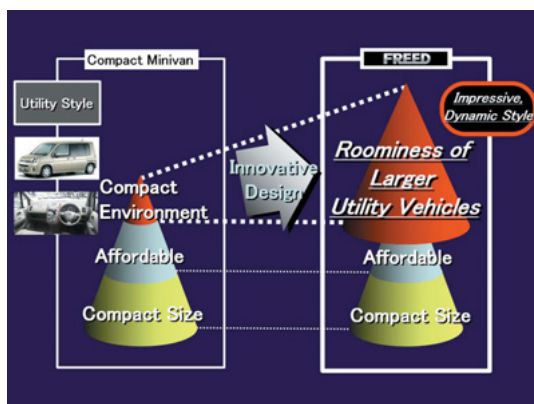


Fig. 1 Approach to development



Fig. 2 Development concept

* Automobile R&D Center

Many compact minivans on the road are convenient and affordable; advantages deriving from their size. At the same time, some seat seven passengers and enable a highly customizable seating arrangement. Despite their small size, the vehicles have therefore earned a reputation for spaciousness and usability.

Yet research reveals that passengers feel cramped riding at full capacity, and owners rarely use some seating arrangements. In short, existing models have not fully met owners' expectations. This conclusion prompted us to go back to the drawing board and reevaluate the benefits people seek from compact minivans. Our solution: to set our sights on maximum spaciousness and convenience in a package of minimal size.

Making it happen meant offering a vehicle that seats seven adults comfortably in a compact body, and allowing free access to any seat from any door simply by walking through the cabin. These conditions formed the framework for design development.

2. Package Layout

For maximum spaciousness and convenience (quintessential minivan advantages) in a compact body, we designed a new, lower floor (Fig. 3).

Combined with other revisions such as a slim fuel tank, this allows a convenient height for children getting in or out by themselves. Inside, it is roomy enough for seven passengers, and captain's seats are arranged for a walk-through layout.

Lowering the floor in front a little more provides just the right elevation for all three rows of passengers. This affords a good view from any seat, and a liberating, pleasant ride for everyone.

For easier navigation through city streets, blind spots are reduced by carefully positioned front pillars. A clearer view in all directions was our priority, and forward-sloping side windows, angled rear pillars, and a downward-curving rear window give a reassuring

command of the road for all drivers (Fig. 4).

We also aimed for perfect utility in the cargo area; indispensable in a minivan. Here, owners will find the FREED a very handy vehicle. It is versatile and spacious for easy loading and unloading of everything from daily groceries to bicycles. Low floors and sufficient opening height demonstrate our attention to detail for efficient loading [Figs. 5 (a), (b)].

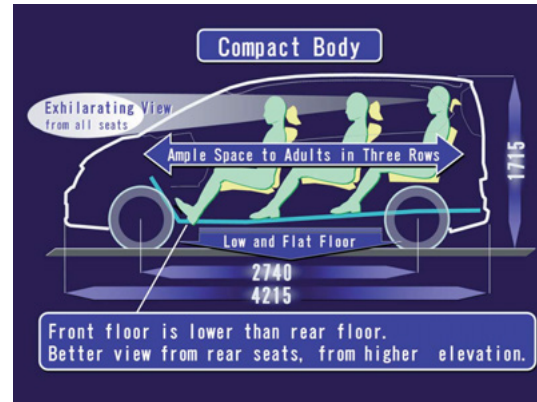


Fig. 4 Vehicle package highlights

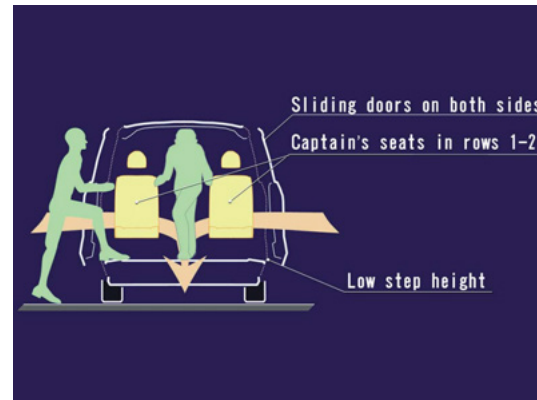


Fig. 5 (a) Easily access to rows 1-3



Fig. 3 Approach to vehicle package



Fig. 5 (b) Cargo area accommodates bicycles

3. Exterior Design

To make the “Free life creation” a reality, we sought a frame unlike any other compact minivan on the road.

In essence, the FREED looks somewhat like a triangle fused to a square. The triangle is formed by the front of the FREED from the first row of seats forward. This section of the frame is reminiscent of a wagon or MPV. The square is formed by the latter section; from the second row back. For optimal utility in the main passenger space and cargo area, we chose a cubic frame (Fig. 6).

This fresh triangular and square frame was just the beginning, however, and to animate the appearance we wrapped it in organic, flowing surfaces everywhere. Even when the vehicle is still, its dynamic appearance makes

it look as if it were in motion.

Sharp accent lines cut into the body, and richly expressive surfaces that make these lines more prominent set the FREED apart from other cars, and impart quite a distinctive look.

At first glance, these carefully drawn character lines stand out from surfaces, appear to deepen, and then gradually fade into the surfaces. In effect, the gradations of shading they produce make the vehicle seem like a sculpture. The design team poured much effort into these panels, and just a touch reveals how richly they embellish the FREED with a sense of workmanship (Figs. 7, 8).

Ultimately, we have distilled satisfying and reassuring driving performance into a compact body with impressive styling that holds its own even against the most refined cityscapes (Fig. 9).

Figures 10 and 11 show the final exterior design.

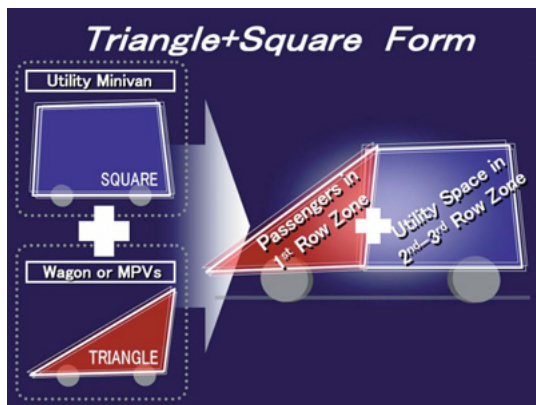


Fig. 6 Exterior design highlights (1)



Fig. 7 Final rendering



Fig. 8 Final rendering



Fig. 9 Exterior design highlights (2)



Fig. 10 Final exterior



Fig. 11 Final exterior

4. Interior Design

In a comfortable interior environment where visual cues and other subtle touches make the FREED seem more spacious than it actually is, we have added features reinforcing usability. Overall, the interior design was developed to call to mind a stimulating and enjoyable open-air café despite the enclosed space (Fig. 12).

The dashboard forms a visually welcoming background. From an open shape with the top half curving around as if to embrace passengers, and the bottom half curving the opposite way to suggest an expansive space. Instrument panel controls are arranged for easy and intuitive operation; audio controls are neatly grouped on top and climate controls on bottom. Additionally, the



Fig. 12 Approach to interior design



Fig. 13 Instrument panel



Fig. 14 Door lining

speedometer and other gauges lie outside the steering wheel. In this position, the instrument panel contributes to a spacious environment around the driver. Gauges can be read instantly while drivers keep their line of sight near the road (Fig. 13).

By dividing the door lining into two separate zones — a forward section visually connected to the dash, and a back section with arm rests, large, low pockets, and other features passengers access — we have focused on a sense of roominess and clearly organized the layout into functional zones (Fig. 14).

Passengers will find the seats as comfortable as those in upscale vehicles. In fact, the seats are built on frames of models in a higher class. More legroom has been provided by making the backs of the first and second rows of seats convex (as if extra material had been carved away), and the headrests appear to float above the seats. Through many details such as these, the interior is designed to feel more spacious.

Conveniences around the cabin include trays, cup holders (carefully sized to hold typical take-out beverage cups), and pockets in handy positions. Large pockets on the bottom of the front doors can hold folding umbrellas and small bags, which can easily be slipped into the wide opening.

Above passengers, the roof is as flat as possible, shaped to let light in from each window and colored in light tones, making it seem brighter and more cheerful as a whole while appearing expansive.

5. Color Design

5.1. Exterior

The first consideration when proposing exterior colors was which colors would enhance its dynamic styling. This led to a range of luxurious, refined colors that assert the FREED's presence even in the most sophisticated cityscapes of Europe, for example (Fig. 15).

Image not displayed for copyright reasons.

Fig. 15 Exterior color options

Premium Night Blue Pearl was chosen as a standard color for its beauty by day under sunny skies and by night under city lights. Flakes of glass in the paint create a rich gleam, and because a characteristic of this coating is to take on subtly different hues of the same color, it

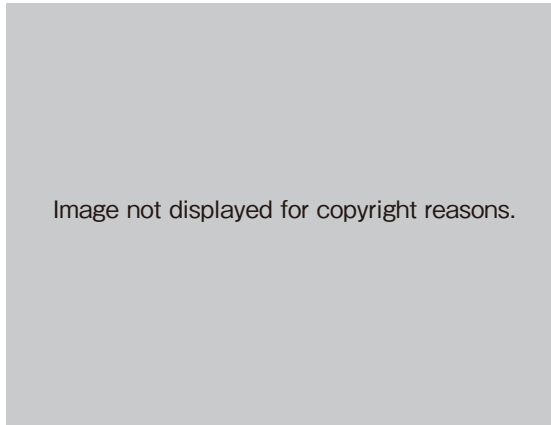


Fig. 16 Inspirations for exterior colors



Fig. 17 Interior inspirations

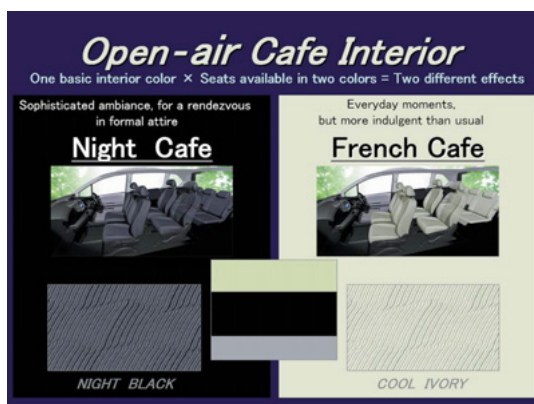


Fig. 18 Coordinated interior color scheme

highlights the graceful body contours of the FREED well by producing a transparent radiance (Fig. 16).

Other color options include an upscale yet brilliant Basque Red, a special black called Blackish Pearl that seems tinged with violet under some lighting conditions, bright and vivacious Luminous Blue Pearl, White Pearl, a light silver, Alabaster Silver, and Polished Metallic Black resembling gunmetal.

5.2. Interior

In choosing interior colors, we followed our general interior design concept of creating a welcoming environment like one's favorite café. For coordinated color schemes with a subtle sense of style, we used a delicate touch in color selection while focusing on basic colors. This aesthetic is reflected throughout the cabin, in the layered pattern on the front of seats, in roof tones, and elsewhere (Fig. 17).

Interior color

Black and gray set the scene inside as the base colors. Other elements are cool ivory for the pillars and roof, and gray for the floor. The vertical design in the door lining called for an alternating black and gray color scheme up to the rear side lining. This approach brings a nice sense of contrast to the spacious cabin area extending from the front to the back, creating a contemporary ambiance (Fig. 18).

Roof color

New French Cool Ivory reflects incident light to brighten the cabin and enhancing the sense of roominess.

Seat fabric

Seats are available in two colors - Cool Ivory or Bluish Black (both introduced especially for the FREED). They contrast with the single color serving as the basic interior color. Cool Ivory is reminiscent of a French café while Bluish Black is quite the opposite; more like a café for an evening rendezvous. Again, a layered pattern in this suede-like tricot material suggests spaciousness.

6. Conclusion

In the FREED, we introduce an unprecedented full-scale minivan that is compact yet spacious and user-friendly. Passengers can drive away their stress, so to speak, in a vehicle that feels liberating in all respects. Its advanced styling is captivating, and our attention to detail extends to superior driving performance. The advantages of compact vehicles in greater convenience and affordability are now more apparent.

The FREED is the first demonstration of the Honda minivan ideal — offering all passengers a pleasant,

exceptionally convenient environment from the beginning to the end of their drive — in a smaller package.

It represents a new beginning in this category as the world's smallest full-scale minivan that captures the advantages of traditional minivans in a more compact format.

■ Author ■



Akihiko OHHASHI



Ohsuke KOIKE



Akihiro HATTORI



Eiko HASHIMOTO

Development of CVT for 2008 Model Year FIT

Masaaki YAMAGUCHI*

Mitsurou OOTAKI*

Keisuke ITO*

Hideo NIRASAWA*

Hirohiko TOTSUKA*

ABSTRACT

A newly developed 1.3 L, 1.5 L class CVT with a torque converter was mounted in the second generation FIT, which is aimed at realizing both higher fuel economy and exhilarating acceleration performance. The main technologies used in the new CVT are the changeover of the start-off method from the conventional start-off clutch to a torque converter, the increase of the pulley distance from 143 mm to 156 mm, the adoption of a high gear ratio, and the adoption of an oil pump driven with a small radius rotor via a chain, a creep control system, and a start-off lockup clutch (L/C) control system.

As a result of these technologies, the transmission efficiency was 1.7% higher than that of the previous FIT CVT, and the torque amplification effect of the torque converter during launch enabled the achievement of higher fuel economy and exhilarating acceleration performance.

1. Introduction

In recent years, amid the attention being paid to countermeasures for global environmental issues, the enhancement of fuel economy is becoming increasingly important from the viewpoint of reducing CO₂ emissions. At the same time, exhilarating acceleration performance, a sought-after car feature, is being demanded. In order to satisfy these needs, a CVT^{(1), (2)}, which realizes highly efficient fuel economy, has been installed in Honda's 1996 and subsequent models. Along with the full model change of the FIT, a CVT with a torque converter designed for the FIT which simultaneously realizes high fuel economy and exhilarating launch performance was developed. This technology is described below.

2. Development Goals

The following three items were established as development goals in order to realize both top level fuel economy and exhilarating launch performance.

- (1) High transmission efficiency due to friction reduction, adoption of high gear ratio, creep control, and L/C control
- (2) Smooth and powerful launch performance due to the use of a torque converter
- (3) A compact CVT suitable for FIT-class vehicles

3. Basic Layout and Main Specifications

Figure 1 shows a main cross-sectional diagram of the CVT fitted with a torque converter for the new model FIT (hereafter called "the new CVT").

The main points in which the new CVT differs from the 2002 model "Fit" (hereafter called "the previous CVT") are the adoption of a torque converter for the start-off mechanism, the use of a large capacity, high-efficiency oil pump driven with a small radius rotor via a chain, the relocation of the pulley bearing, and the increased distance between the pulley shafts in order to reduce metal V-Belt friction. Figure 2 shows an image of the CVT, and Table 1 shows the main specifications.

* Automobile R&D Center

Table 1 CVT specifications

		2008 model FIT	2002 model Fit
Engine specifications	Type	1.3L i-VTEC	1.3L i-DSI
	Max. power [kW/rpm]	73/6000	63/5700
	Max. torque [N·m/rpm]	127/4800	119/2800
Ratio	LOW OD	2.419 0.421	2.367 0.407
	Secondary	1.266	1.333
	Final	3.875	4.333
	RVS planetary	1.024	1.785
Dimension	Overall length [mm]	361	349
	Pulley axis length [mm]	156	143
Weight (Wet) [kg]		77.6	74.5

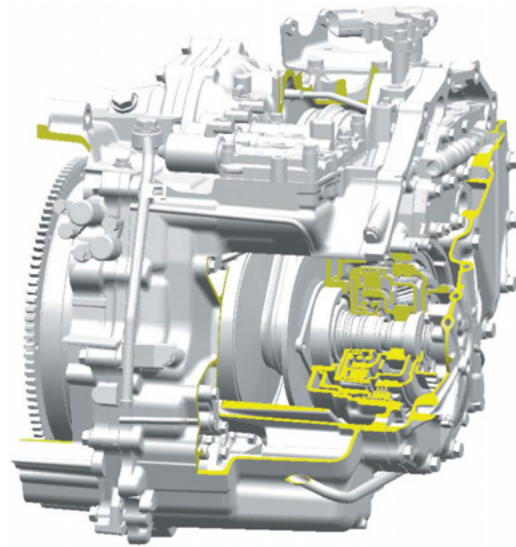


Fig. 2 Image of CVT for new FIT

4. Main Structure

4.1. Packaging Philosophy

The new CVT, which was fitted with a torque converter, has the same case configuration, shaft layout, and hydraulic system layout⁽¹⁾ as those of the previous CVT.

The increase in length due to the torque converter was compensated for by adopting the small oil pump and technology of the drive pulley cover bearing support mechanism, which have enabled a compact construction.

4.2. Torque Converter

4.2.1. Goal of using a torque converter

The previous CVT employed a multi-plate wet clutch for the start-off method. Because a torque amplification mechanism was not applied, it was necessary to set a low gear ratio to enable the necessary drive force for launch. For this reason, the gear ratio while cruising would also decrease. The engine speed would then rise, providing a margin for an increase in fuel economy.

Because the new CVT uses a torque converter, the gear ratio was increased by 14% compared to that of the previous CVT, to be able to realize both higher launch drive force, and, greater fuel economy, which was achieved by decreasing the cruising engine speed.

4.2.2. Reduced diameter of the torque converter

In contrast to the conventional automatic transmission torque converter for a 1.3 L engine, the blades of pump, turbine, and stator in the new CVT were optimally designed, resulting in a 15% reduction of the flow path diameter ($\phi 190$), and an 18% reduction of weight.

Also, the lock up clutch diameter was increased to $\phi 208$ in order to permit lock up from low speed range.

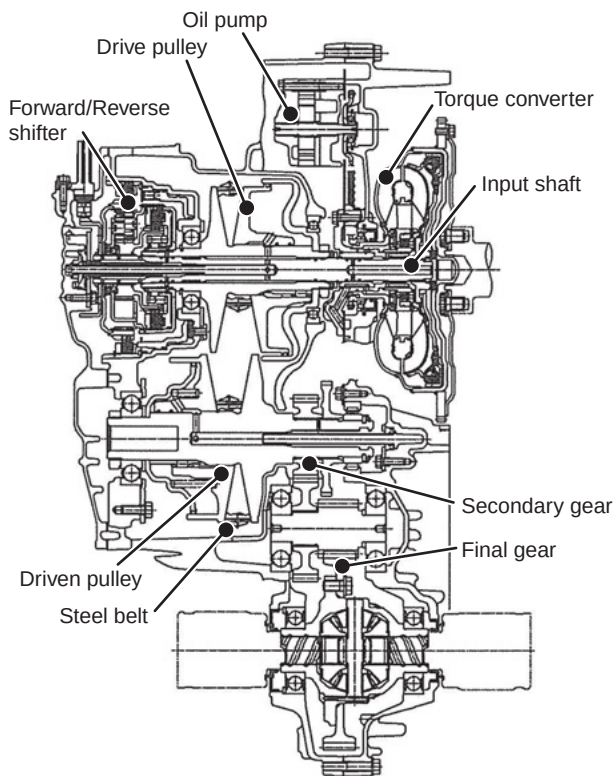


Fig. 1 Main section of CVT

Figure 3 shows the cross-section of the torque converter.

4.3. Oil Pump

4.3.1. Basic layout of the oil pump

In the previous CVT, layout of the oil pump was such that it was driven directly from the input shaft⁽¹⁾. In contrast, the oil pump of the new CVT was driven by the sprocket via a silent chain. This sprocket is connected to the input shaft. Structure of the oil pump is shown in Fig. 4.

With this new layout, the transmission width was reduced, and the diameter of the oil pump rotors was also reduced, thereby reducing friction.

4.3.2. Management of high pressure

The use of a torque converter triggers an increase in the discharge flow and the discharge pressure, which necessitates oil pump increase.

The necessary oil pressure increased from 3.5 MPa for

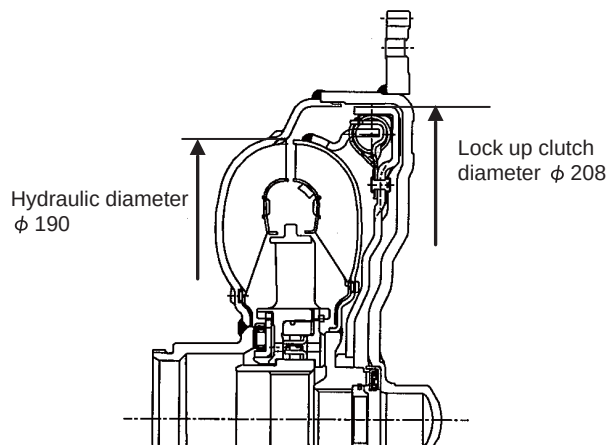


Fig. 3 Structure of torque converter

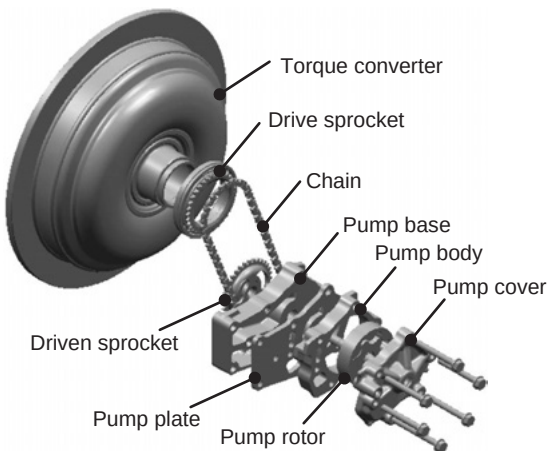


Fig. 4 Structure of oil pump

the previous CVT to 5.5 MPa. In order to help prevent reduced discharge performance due to deformation of the housing, FEM analysis was used to optimize the stiffness of the aluminum die-cast pump cover.

4.4. Bearing Support Structure of Drive Pulley Cover

The bearing to support the drive pulley was mounted on the pulley cover instead of being fitted directly to the shaft. As a result, the entire length of the drive pulley was reduced by 18 mm. Figure 5 shows the layout of the drive pulley bearing.

Whereas the pulley cover of the previous CVT was subjected to the load from the oil pressure alone, the pulley cover of the new CVT is subjected to the bearing support load as well. The shape of the pulley cover was designed to account for the bearing support load, thus enabling the bearing to be mounted on the pulley cover.

4.5. Pulley Strength

Regarding the pulley, stress analysis and deformation analysis using the FEM were carried out in each of the drive and driven assembly conditions in order to establish the optimum shape.

Previously, the support part of the driven pulley was located between the V-surface and the secondary gear. In the new CVT, the bearings were located on outside of the secondary gear. Therefore, the stress at bottom of the V-surface on the driven pulley shaft was reduced by 15%, resulting in optimization of the strength and deformation on the driven pulley. Figure 6 shows the bearing arrangement and the results of stress analysis.

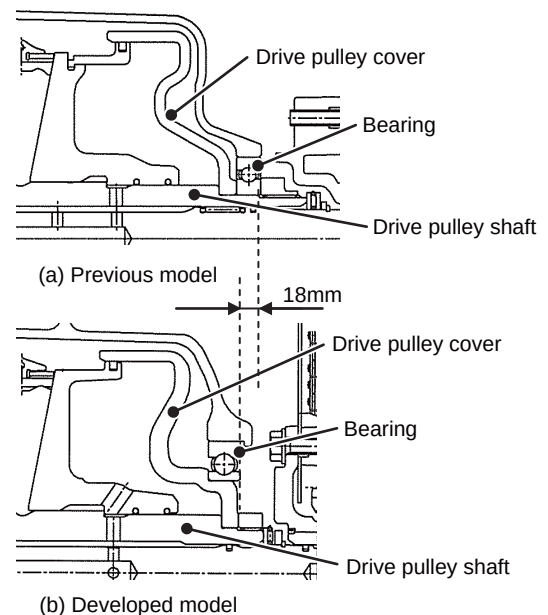


Fig. 5 Comparison of drive pulley bearing layout

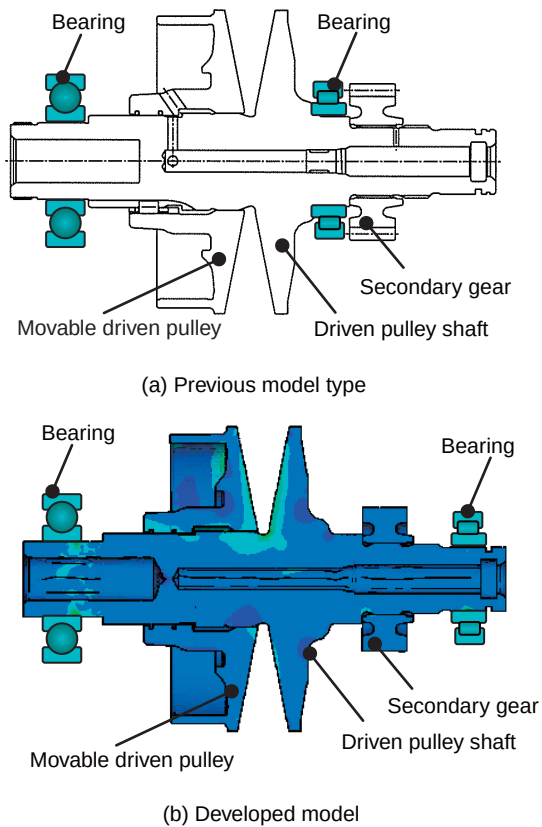


Fig. 6 Comparison of driven pulley bearing layout

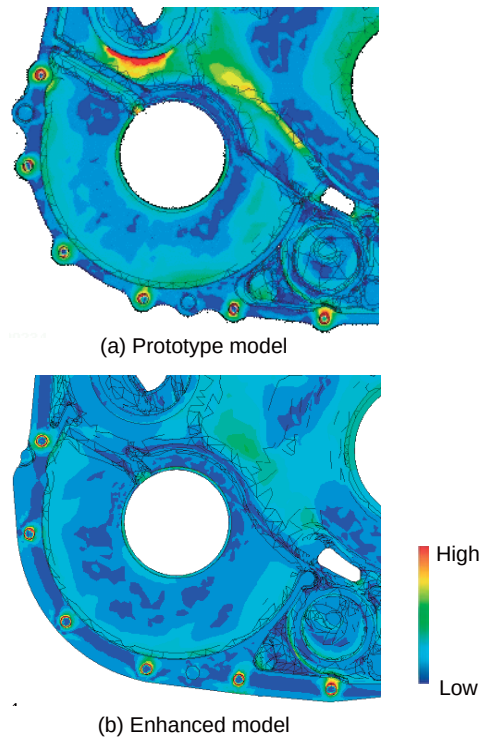


Fig. 7 Transmission case stress distribution

4.6. Metal V-Belt Strength

To enable a 24 mm V-Belt to cope with the high torque resulting from the use of a torque converter, the new CVT adopted torque harmonization control employing an electronically controlled throttle (DBW) to help prevent too large input torque from being applied to the V-Belt during the in-gear transient.

As a result, the V-Belt load was able to be reduced by 50% compared to that generated in the case of conventional control, enabling adequate durability to be obtained with a 24 mm V-Belt.

4.7. Case Strength

In order to cope with the high torque, stress analysis and alignment face sealing analysis using FEM were carried out, and shape of each part of the case was optimized.

Figure 7 shows the results of the analysis for stress in the vicinity of the final gear. It can be seen that adequate case strength was obtained by increasing thickness of the case in vicinity of the final gear and optimizing layout of the ribs.

5. Technology for Enhancing Transmission Efficiency

5.1. Friction Technology

Three main factors that affect CVT friction are metal V-Belt, oil pump, and oil agitation. Technologies used to reduce CVT friction resulting from each of these factors, are described below.

5.1.1. Metal V-Belt

The V-Belt friction tends to decrease when the distance between the pulley shafts is increased. Figure 8 shows the change in friction along with the change in the distance between the drive and the driven pulley shafts. If the distance between the shafts is increased, however, the size of the transmission will also increase. The distance between the pulley shafts in the new CVT was set to 156 mm as the optimum setting for friction

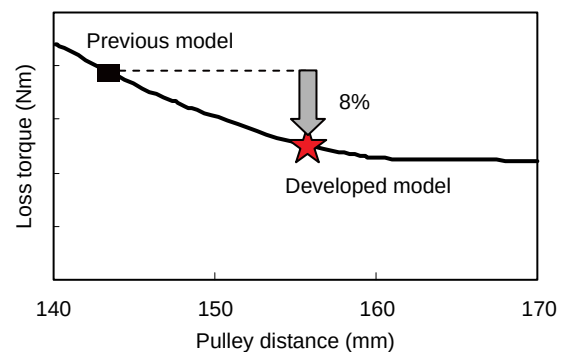


Fig. 8 Loss torque of CVT belt

reduction and transmission size. As a result, an 8% reduction of friction compared to the previous CVT was achieved.

5.1.2. Oil pump

The diameter of the outer rotor compared to the previous model was reduced by 13%, and the discharge per revolution of the rotor was increased by 26%, enabling roughly the same drive torque as that of the previous model.

5.1.3. Oil agitation

Reducing the total amount of transmission oil so as to lower oil level is an effective way to reduce the agitation of the oil. However, lubrication performance is an issue. In order to help ensure that the differential section has adequate lubricating performance, the new CVT was provided with ribs and grooves to feed oil that was lifted up (indicated by A) by the gear, as shown in Fig. 9. This construction helps ensure adequate lubricating performance even if the oil level is reduced, thus enabling friction to be reduced.

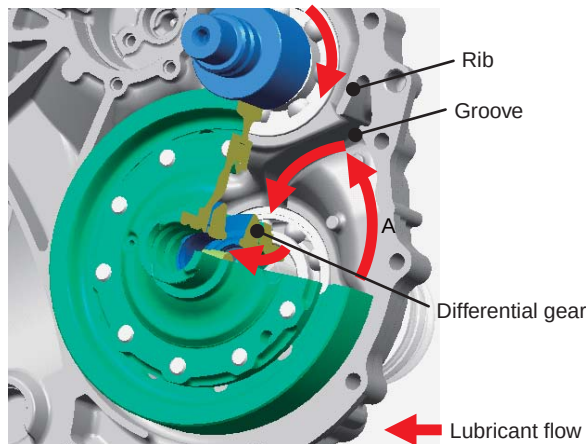


Fig. 9 Lubricant flow at differential gear

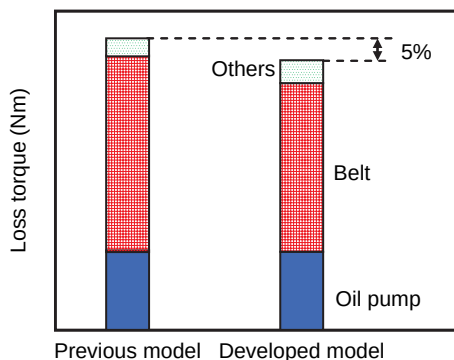


Fig. 10 Comparison of loss torque

5.2. Transmission Friction

As a result of employing enhancement technology for each of the factors mentioned in the previous sub-section, the friction of metal V-Belt was reduced 8% compared to that of the previous CVT, and the friction in the oil pump remained roughly the same. In addition, the friction of each gear due to oil agitation, increased along with the increase in gear size due to the use of a torque converter, but increase of friction was kept to minimum by reducing agitation resistance.

The transmission friction was 5% less than that of the previous CVT. Figure 10 shows a breakdown of this friction.

5.3. Electronic Control Technology

5.3.1. Creep control

By reducing the forward clutch capacity in order to reduce creep torque during idling, friction during idling was reduced. Figure 11 shows the creep motive power

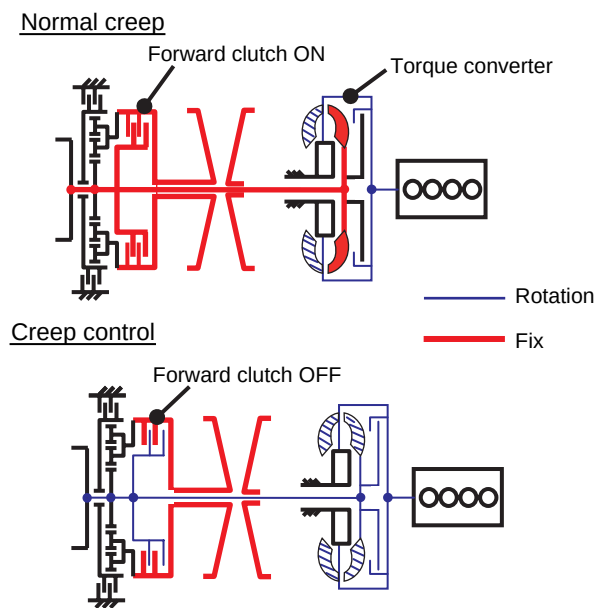


Fig. 11 Power transfer at creep control

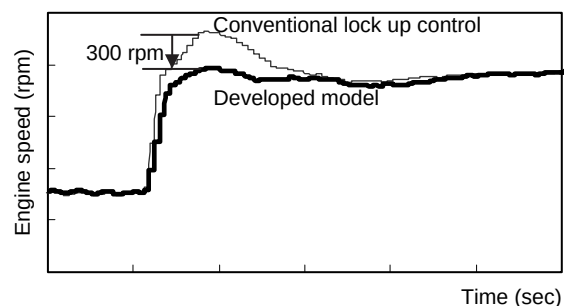


Fig. 12 Comparison of engine speed at launch

transmission condition. Under an idling condition in which creep occurs, the rotating elements downstream of the turbine are completely stopped. However, when a creep control condition occurs, the forward clutch capacity is reduced, after which the turbine begins rotating. As a result, the relative rotational speed between the pump of the torque converter and the turbine falls, and as a result, friction decreases. In this way, the friction during idling was reduced by 70%.

When the vehicle is on slope, the inclination sensor detects the gradient, stops creep control, and applies the normal creep force, thus achieving a drive feeling equivalent to that of the conventional automatic transmission.

5.3.2. Start-off LC control

In the new CVT, the efficiency at launch was enhanced by expanding the lock up region in the low speed region.

By increasing size of the lock up clutch and expanding the lock up region, slip loss of the torque converter during launch was reduced, and engine speed at launch also fell by about 300 rpm. Figure 12 shows the engine speed characteristics during launch.

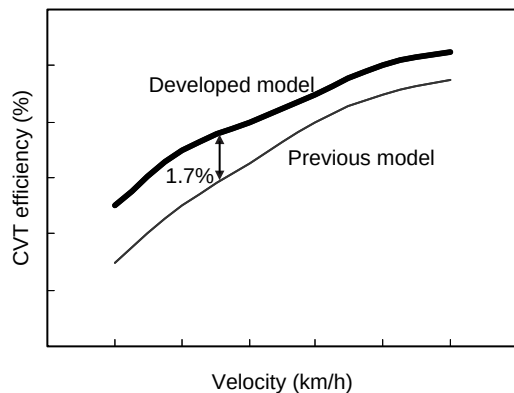


Fig. 13 Comparison of CVT efficiency

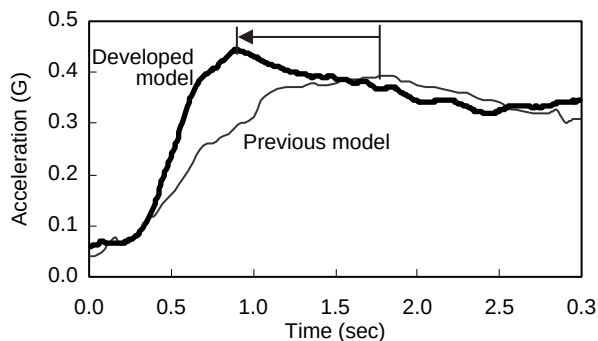


Fig. 14 Comparison of launch acceleration characteristics

6. Conclusion

The CVT with a torque converter for the FIT was developed, and the following results were obtained.

- (1) The efficiency was 1.7% higher than that of the previous CVT due to the reduction of friction, use of a high gear ratio (Fig. 13).
- (2) As a result of adoption of a torque converter, rise time of peak G was reduced compared to the previous CVT (Fig. 14), thus realizing powerful launch performance.
- (3) As a result of adoption of compactness technology, the CVT with good mounting ability with minimum size increase was produced, while utilizing a torque converter.

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■ Author ■



Masaaki YAMAGUCHI



Mitsuru OOTAKI



Keisuke ITO



Hideo NIRASAWA



Hirohiko TOTSUKA

New Analysis Technology for Automotive Chassis Sensitivity of Steering Wheel Vibrations Induced by Tire-Wheel Imbalance or Brake Torque Fluctuation

Jinghong Yu* Brian Nutwell* Brian Brickner*

ABSTRACT

A simulation model for analyzing vehicle chassis sensitivity for the vibrations caused by tire-wheel imbalance and brake torque fluctuation has been developed. Nonlinear elements, such as detailed steering power assist system, frictions and nonlinear bushings are taken into consideration. The simulation model is extensively correlated with laboratory excitation tests in both frequency and time domain. The model is then used to analyze the mechanism of chassis sensitivity, and as a result, contributing chassis vibration modes and influential chassis factors are identified. The analysis results are further correlated with vehicle on-road tests with specification changes in critical factors. Reliable correlation between simulated and measured trends is achieved.

1. Introduction

Chassis vibrations induced by tire-wheel imbalance and brake torque fluctuation are phenomena that affect overall vehicle driving comfort. For purposes of this paper, the vibration of the steering wheel and the chassis system caused by tire-wheel mass imbalance is called “steering shimmy” or “shimmy”, while the vibration induced by braking torque variation during a brake maneuver is referred as “brake judder” or “judder”. Steering shimmy and brake judder are the combined effect of the transfer path characteristics of the chassis system and the excitation force input from the tire-wheel or brake system.

In the typical transfer path for steering shimmy and brake judder, the excitation forces are initially transmitted into the suspension through the front wheel spindles, exciting the vibrations of the suspension system. As the primary path from the suspension to the steering system, a majority of the vibration energy is transferred to the steering rack through rigid body motion. As the secondary path, vibration energy is also transmitted to the steering gearbox through structural flexible body motion. The vibration is then transferred to the steering wheel

through the steering system. For a certain level of excitation input, the vibration can be either amplified or attenuated along the transfer path from the road wheel to the steering wheel, depending on the characteristics of the chassis system.

In the automotive industry, some efforts have been made in analyzing the transfer path of steering shimmy and brake judder^{(1)-(4), (7)}. However, many of the existing analysis tools or methods can not produce satisfying results, in terms of nonlinear characteristics, due to oversimplified modeling. Some of the factors which are often overlooked include detailed power steering system, system frictions, and the nonlinearity of bushings, as well as detailed correlation with experimental measurement.

To better understand the transmission mechanism of the chassis system and to help develop design strategies for reducing steering and chassis vibrations, we set out to develop an effective analysis technology. For analytical purposes, the excitation forces can be decoupled and evaluated separately from the chassis response. Excitation can be measured directly on the vehicle, i.e. using an instrumented road wheel, or on a component level using a brake dynamometer or similar equipment. The measured excitation forces can then be used as an

* HONDA R&D AMERICAS, INC.

excitation input to the CAE models of chassis response.

In this research, the following specific activities are undertaken:

- (1) A multi-body simulation model is developed for a nonlinear, coupled front chassis system which uses a MacPherson strut suspension and hydraulic power steering. Detailed steering and suspension systems, bushing-isolated sub-frame, frictions of joints, the displacement dependence of bushings, and a simple model for the tire and wheel are included in the modeling. The primary path, that is, the rigid body transfer path, is mainly considered.
- (2) The model is validated by laboratory excitation tests under various levels of excitation force in both frequency and time domain.
- (3) The simulation model is used to analyze the transmission mechanism of vibrations in the chassis transfer path. Contributing chassis vibration modes are discovered and important chassis factors are identified.
- (4) The analysis results are further correlated with vehicle on-road tests with specification changes in some critical factors. Comparisons of simulated and measured trends of steering shimmy and brake judder are illustrated.

2. Modeling Approach

A multi-body simulation environment is used to build the simulation model, due to the low-frequency nature of steering shimmy and brake judder phenomena and the software capability of assembling different sub-systems. Custom model templates and sub-systems are created to capture critical elements affecting vibration behavior⁽⁸⁾.

2.1. Model of Front Chassis System

A multi-body-simulation model for the nonlinear, coupled vibration characteristics of the full front chassis is created by the inclusion of the sub-system models of steering, suspension, and a virtual test rig, as illustrated in Fig. 1.

A custom model template is built for the power steering system. Mechanisms such as rack-pinion gear, steering column, universal joints, a rubber coupler and steering wheel, are modeled first. The model is then made to represent hydraulic power-assisting system (HPS) or electric power-assisting system (EPS), as needed. As an example, HPS is only referred hereafter. The power-assisting system is modeled as a flow-source-valve-cylinder system^{(5), (6)}. The control valve consists of two pairs of orifices, of which the flow cross-sectional areas are functions of the angular displacement of the control valve. Fluid inertia and compressibility are considered in the system connection pipes. Compressibility of the fluid in the steering cylinders is taken into consideration. Possible backlash in the steering rack-pinion gear is also modeled.

During the modeling of the suspension system, a

MacPherson strut type of suspension is considered. The stabilizer bar, sub-frame and powertrain mass and inertia properties are also included. The sub-frame attachment can be selected between bushing-isolated and rigidly-mounted on the vehicle body by a switching parameter.

A virtual test rig is created which can accommodate suspension preload from braking or drive torque, the mass and stiffness of the tire-wheel assemblies, as well as multiple degree-of-freedom excitation forces to the spindles. Friction-free bearings are modeled and included to support the tires on the ground. Measured excitation forces are used to define the excitation forces.

2.2. Model Parameters

As one of the nonlinear elements of the chassis system, friction is important for low-frequency vibration phenomena. During this study, frictions in the steering and suspension systems are included in the simulation model, such as:

- (1) Friction between the rack and the gearbox housing.
- (2) Friction torque between the gearbox input shaft and the gearbox housing.
- (3) Friction torque between the pinion and the gearbox housing.
- (4) Friction torque between the input shaft and the pinion.
- (5) Friction torque of steering column universal joints.
- (6) Friction torque at the ball joint between the knuckle and the lower control arm.
- (7) Friction torque at the ball joint between the knuckle and the spindle.
- (8) Friction torque at the ball joint between the knuckle and the tie rod.
- (9) Friction torque at the ball joint between the tie rod and the rack.
- (10) Friction torque between the drive shaft and the spindle.

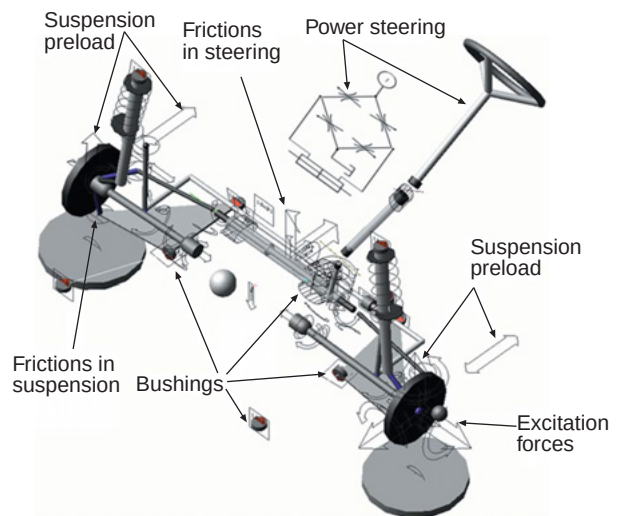


Fig. 1 Multi-body-simulation model of front chassis system

In order to ensure numerical simulation stability and replicate a realistic friction profile, a particular velocity-dependent friction law is used to describe the frictions. Figure 2 illustrates a comparison of the velocity-dependent friction and Coulomb friction. Obviously, Coulomb friction preserves the discontinuity at null relative velocity, and the discontinuity will cause unstable solution in simulation. Meanwhile, Coulomb friction is against physical phenomena where kinetic friction is usually lower than static friction. The proposed velocity-dependent friction profile allows the static friction to exponentially decay to the kinetic one, while guarantying continuity at zero relative velocity. The kinetic friction is defined as 70% of the static friction (a percentage commonly accepted for engineering friction assumption), and the static friction is determined as the measured break-through level.

Steering gearbox mounting bushings and suspension bushings are characterized under appropriate displacement. Stiffness and damping coefficients for the bushings are determined by experimental measurement for various frequencies and displacements. For a suspension hydro-bushing, both displacement and frequency dependence must be considered. However, for a conventional rubber bushing, within the normal frequency range of shimmy and judder phenomena, the stiffness and damping coefficients vary significantly with displacement but do not change greatly with frequency. In such a case, displacement-dependent bushing model is appropriate.

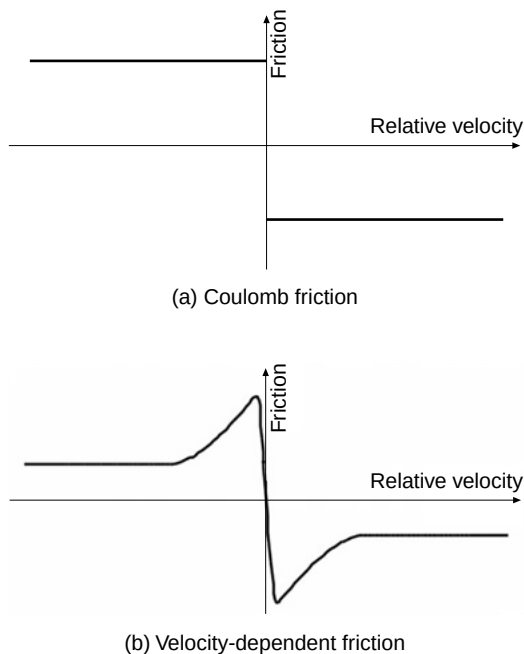


Fig. 2 Velocity-dependent friction profile used in simulation and its comparison with Coulomb friction

In order to understand the major contributions of excitation forces and quantify vertical and longitudinal components of preload for steering shimmy and brake judder, vehicle tests are performed on a SUV to measure the six-degree-of-freedom forces acting on the front wheels. During the tests, steering shimmy is reproduced by driving the vehicle at a speed at which maximum steering wheel vibration is detected with a known unbalanced wheel, while brake judder is duplicated by decelerating the vehicle from high speed at a certain level of deceleration with an induced front brake torque variation.

Let us define the coordinate system of the forces (F_x , F_y and F_z) and moments (M_x , M_y and M_z) acting on the front wheels as that X -components are in the vehicle longitudinal direction with positive toward front for the left wheel and toward rear for the right wheel, Y -components in the lateral direction with positive toward outbound of the vehicle, and Z -components in the vertical direction with positive toward up.

Figure 3 shows the force measurement results for steering shimmy. It is found that the peak of steering shimmy appears at around 14Hz-16Hz for the vehicle under test. It is revealed that F_x , M_y , M_z and M_x are the major contributions for the excitation input of steering shimmy. Similar results are obtained for brake judder phenomena. In addition, suspension preload forces and

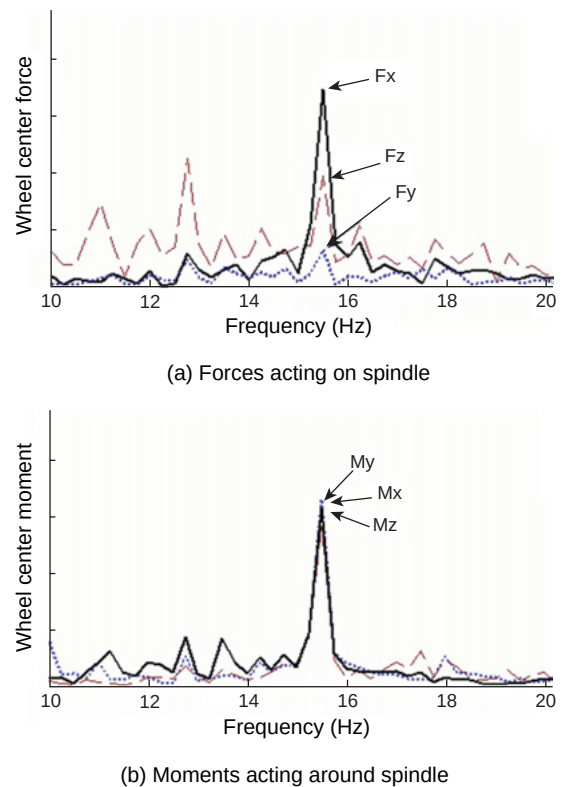


Fig. 3 Excitation forces for steering shimmy

vehicle weight transfer during brake judder measurement are detected by examining the measurement data in time domain.

The experimentally determined excitation forces, such as F_x , M_y , M_z and M_x for shimmy mode, are then inputted into the simulation model through the definition of a six-component general force at the spindles. Since the purpose of this study is to analyze the chassis sensitivity of steering shimmy and brake judder from the spindles to steering wheel, it is appropriate to excite the chassis system at the spindles during simulation. The generation mechanism of the excitation forces and the transmission characteristics of the brake system and tires are not required and discussed here.

3. Validation by Excitation Test

To validate the front chassis model, a laboratory excitation test is needed to reproduce the vibration behavior of the steering wheel and chassis systems during shimmy and judder modes.

There are several existing excitation test methods used by the automotive industry. In these existing methods, the vehicle under test is commonly excited through tire contact patch, which results in limitations such as the difficulty in applying suspension preload, the limitation to single-directional excitation, the need for understanding tire characteristics to derive the transmissibility of chassis system from test, and the variation of excitation input.

A new excitation test method is designed here to address these issues. The basic test capabilities with the new method, as shown in Fig. 4, are as follows:

- (1) Vertical and longitudinal preload to front suspension.
- (2) Multi-directional excitation input at spindle.
- (3) Direct measurement of chassis transmissibility from spindle to steering wheel.
- (4) Stable excitation and repeatable results.

During the excitation testing, the brake is locked for brake judder mode testing, while unlocked for steering shimmy mode testing. The validation work is mainly

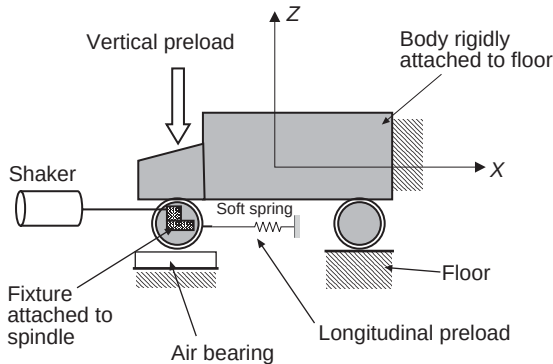
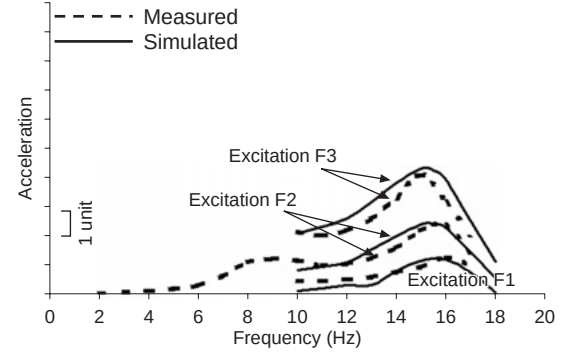


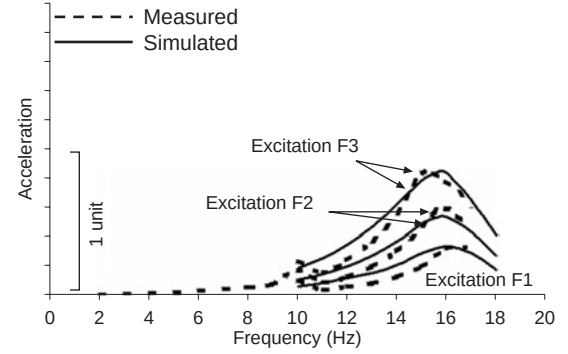
Fig. 4 Laboratory excitation method

focused on the rigidly-mounted sub-frame condition.

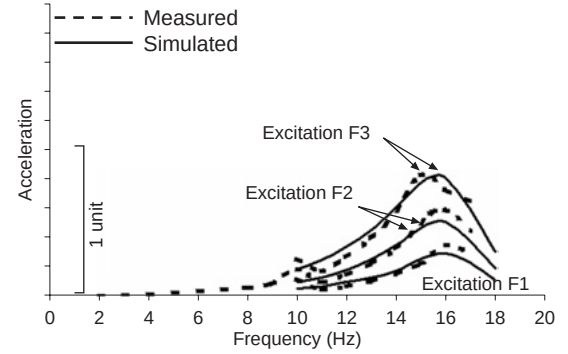
The simulation model is validated by frequency sweep tests under various levels of excitation amplitude. Figure 5 shows the simulated and measured accelerations at steering wheel and other different locations under three different levels of excitation input, for steering shimmy mode in frequency domain. A very good agreement is shown between the simulation and measurement of chassis vibrations under various excitation forces, in terms of both peak amplitude and peak frequency. Similar agreement



(a) Steering wheel



(b) Steering rack, lateral



(c) Knuckle, lateral

Fig. 5 Simulated and measured accelerations for shimmy mode under different excitations (frequency domain)

is revealed for brake judder mode, as illustrated in Fig. 6. Those results confirmed that the simulation model can predict the peak amplitudes and peak frequencies for both steering shimmy and brake judder with high accuracy.

Time waveform data at judder peak frequencies for three different excitation input forces are shown in series in Fig. 7. It is seen that the simulation can yield waveforms very similar to the measured ones. Meanwhile, the nonlinear vibration characteristics of the chassis system are revealed on the non-sine-waveform

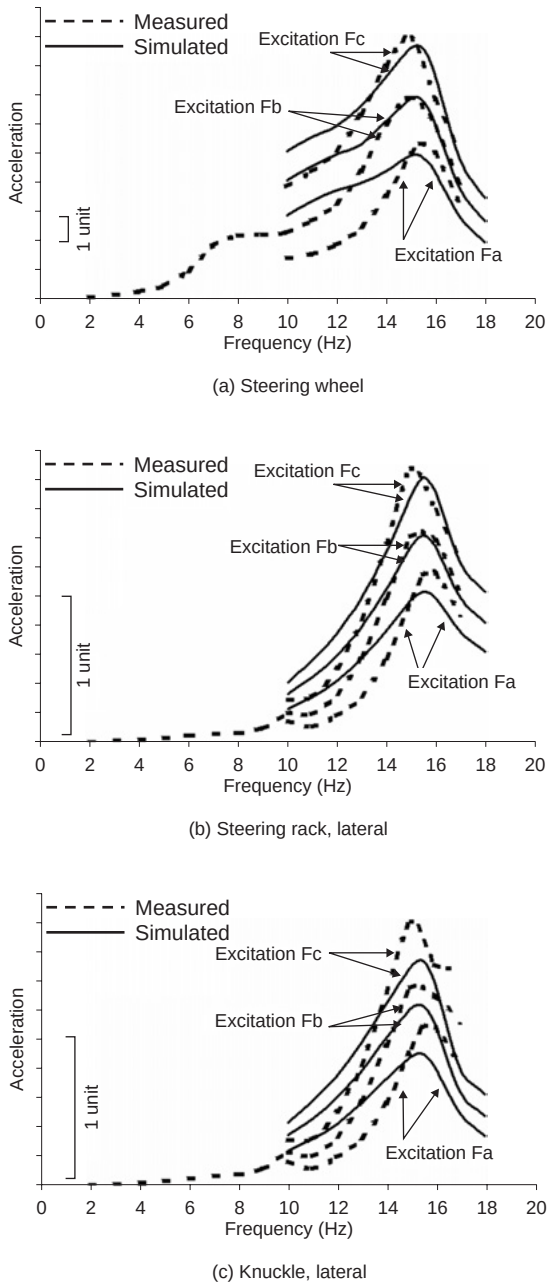


Fig. 6 Simulated and measured accelerations for judder mode under different excitations (frequency domain)

response of the steering rack and knuckle to a pure sine waveform force input. As shown in Fig. 7 (c) and (d), these nonlinear characteristics are also predicted by simulation very well. Similar results are achieved for steering shimmy mode.

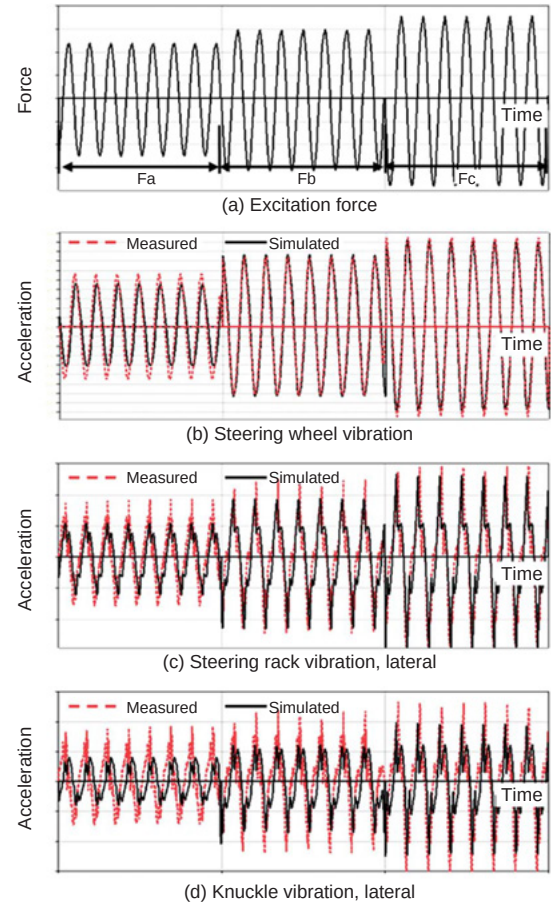


Fig. 7 Simulated and measured acceleration time waveforms for judder mode under different excitations

4. Mechanism Study

4.1. Influential Chassis Modes

The simulation model is used to identify the dominant chassis vibration modes and how they are coupled. This is done by a linear analysis around the equilibrium operating point. As the result, the following two critical chassis modes are discovered in the frequency range of interest for both steering shimmy and brake judder modes⁽⁹⁾:

- (1) Suspension fore-aft out-of-phase mode.
- (2) Steering wheel rotational mode.

The overall chassis sensitivity is the combined effect of these two dominant vibration modes. Modal coupling

phenomenon may occur between those two modes, if they are not separated appropriately. That is, if the frequencies of the modes are too close, the vibration will be amplified twice due to the resonances in both suspension and steering systems at the similar frequencies. This kind of modal coupling mechanism will eventually cause very high level of vibration at the steering wheel. Figure 8 shows the simulation results for the following three different conditions, respectively:

- (1) Independent response of the steering mode.
- (2) Independent response of the suspension mode.
- (3) Overall response of the two chassis modes.

It can be seen that in the case of the independent response of either the steering mode or the suspension mode (this can be achieved by moving the other mode far away from the frequency range of steering shimmy/brake judder), the steering wheel vibration is very low. However, when both of the chassis modes exist, the level of steering wheel vibration becomes much higher, due to the modal coupling mechanism. Building on this understanding to lower the chassis sensitivity of steering shimmy and brake judder, design emphasis can be put on the reduction of the peak amplitude of suspension fore-aft out-of-phase mode, the increase of the peak frequency of the suspension mode, or the reduction of the peak frequency of steering wheel rotational mode.

4.2. Influential Chassis Factors

Now that the chassis model has been validated, it can be used to perform a virtual design of experiment (DOE) to efficiently identify what factors have the greatest influence on the peak amplitudes and frequencies of steering shimmy and brake judder.

Two objective functions for the DOE process are defined to capture the effect of factors on steering shimmy and brake judder. Objective 1 is defined as the maximum amplitude of steering wheel vibration, while Objective 2 is referred as the frequency at which steering wheel vibration peak occurs, during a frequency sweep run. The frequency range is set wide enough to cover frequencies

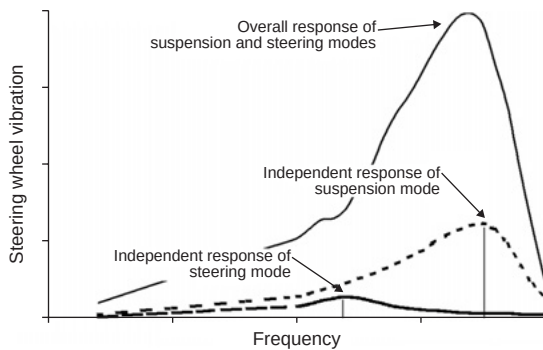


Fig. 8 Coupling mechanism of chassis modes to generate final vibration at steering wheel

of possible factor combinations. Quadratic response surface method is applied to create the design matrix for the simulation runs and to fit polynomials to the results of the simulation.

Based on the virtual DOE analysis, influential factors are identified. Among the influential factors are the locations of suspension geometric points, the mass and inertia of the suspension and steering components, characteristics of the mounting bushings, frictions in the chassis system, pressurized area of the steering cylinder, and steering ratio.

Figure 9 shows the ranked effect, or relative importance, of top influential factors for the peak value of steering wheel acceleration for shimmy mode and judder mode, respectively, while Fig. 10 illustrates the most influential factors for the peak frequency of steering wheel acceleration. It is observed that due to the similarity of chassis transfer path for both steering shimmy and brake judder, the factors exhibit similar importance rankings for both modes.

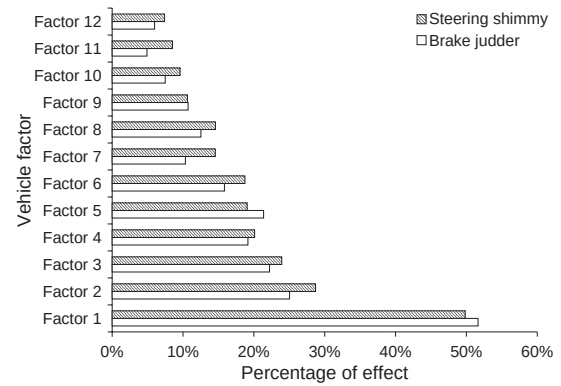


Fig. 9 Top influential factors and their effect on maximum amplitude of steering wheel acceleration when their specifications change from -10% to 10%

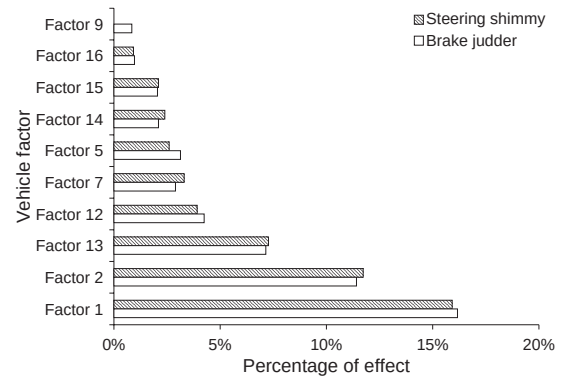


Fig. 10 Top influential factors and their effect on peak frequency of steering wheel acceleration when their specifications change from -10% to 10%

5. Correlation with Vehicle Tests

The simulation results are further confirmed with vehicle on-road shimmy and judder tests. Several top influential factors are selected, and test parts are accordingly designed to deviate from nominal specifications. The relative changes of steering shimmy and brake judder with the change of specifications of those factors are measured by vehicle on-road tests and then compared with simulation results. Based on the repeatability of on-road tests, the target for the correlation between test and simulation is set to 80%.

Figure 11 shows a correlation between measured and simulated percentage change of shimmy and judder due to the change in part specifications. It is shown that the correlation between the simulated and measured results is within 80% for almost all factors considered, for both steering shimmy and brake judder modes. Therefore, it

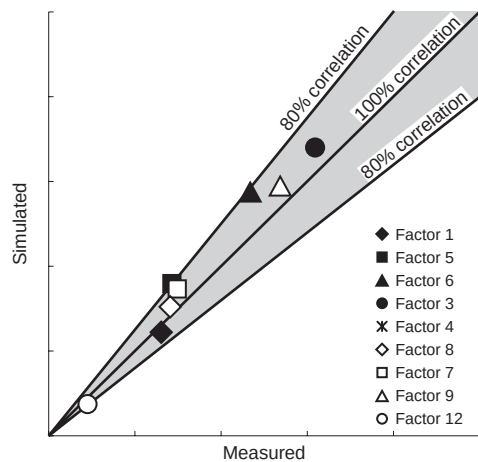
is demonstrated that the simulation can predict the effect of specification changes with good accuracy.

6. Conclusions

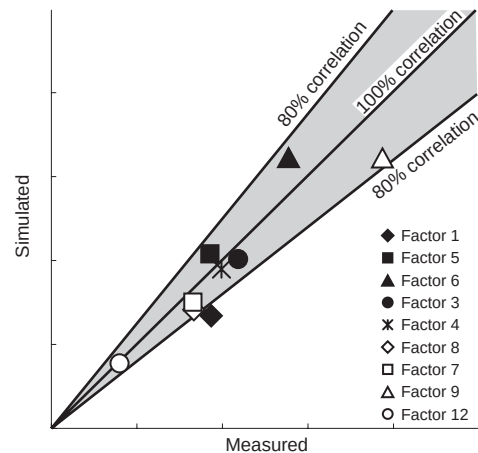
A new analysis technology for analyzing vehicle chassis sensitivity for the vibrations induced by tire-wheel imbalance and brake torque fluctuation has been established. A multi-body simulation model was created for a coupled front chassis system consisting of power steering, suspension, bushing-isolated sub-frame, tire-wheel, and a virtual excitation rig. Nonlinear elements, such as detailed power-assisting system, frictions and nonlinear bushings have been taken into consideration. The model has been extensively validated by laboratory excitation tests.

The simulation model was then used to analyze the transmissibility mechanism and quantify the relative importance of chassis factors. Contributing chassis vibration modes and influential chassis factors were efficiently identified. To further verify the simulation results, vehicle on-road tests were conducted to measure the effect of specification changes of key factors on shimmy and judder. Reliable correlation has been demonstrated between simulated and measured results.

The simulation tool developed here can be used to help quantify the effect of proposed specification changes through simulation during vehicle development, and has made it possible to enhance development efficiency and vehicle performance. In addition, the knowledge gained here about chassis modal coupling and relative importance of chassis factors are useful to establish design enhancement strategies for steering shimmy and brake judder in future models.



(a) Percentage of steering shimmy change



(b) Percentage of brake judder change

Fig. 11 Correlation of simulated and measured change in steering shimmy and brake judder with change of factor specifications

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■ Author ■



Jinghong Yu



Brian Nutwell



Brian Brickner

In-vehicle Super-directive Microphone System for Speech Recognition Using Nonlinear Signal Processing with Two Microphones

Toshiya SHIKANO*

Junichi WATANABE**

Naoshi MATSUO***

ABSTRACT

A super-directive microphone system using digital signal processing has been developed for in-vehicle speech recognition systems for the North American market, in order to increase the recognition accuracy of these systems in high noise environments.

The developed microphone system uses a new super-directive algorithm with a high directional noise reduction rate that functions to reduce noise, such as passenger utterances. To enable adaptation of the algorithm to different in-vehicle noise environments, such as while operating the air conditioner and opening/closing the windows, the system also incorporates an omni-directional microphone and an analog high pass filter. The use of the new system increases recognition accuracy in a variety of noise environments.

1. Introduction

Honda has been gradually using North American navigation systems with speech recognition since the 2003 model year ACCORD. Questionnaire surveys conducted by Honda indicate that customers use speech recognition systems frequently and give them high marks for ease of operation. On the other hand, although tests conducted by Honda showed that these systems yield adequate recognition accuracy under usual road noise conditions, many requests have also been received indicating a preference for the further enhancement of recognition accuracy as sensed by the user. The way such systems have actually been used also indicated a need for modifications. Among the factors interfering with recognition, therefore, attention was focused on noises assumed to be particularly loud in the cabin, and thus the super-directive microphone using digital signal processing was developed for in-vehicle speech recognitions.

* Automobile R&D Center

** FUJITSU LIMITED

*** FUJITSU LABORATORIES LTD.

2. In-vehicle Noise and Its Countermeasures

2.1. In-vehicle Noise

The sources of cabin noise include the following:

- Road noise
- Wind noise that occurs when windows or sunroof are open
- Air conditioner noise and noise from air conditioner blower air hitting the microphone
- Sound of the person in the front passenger seat or other occupants talking

Classified by their physical properties, road noise, wind noise from open windows or sunroof, air conditioning noise, and the like, belong to the category of stationary noise. The sound of the person in the front passenger seat or other occupants talking belongs to the category of non-stationary (sporadic) noise. The sound of air conditioner blower air hitting the microphone is the noise of a clipping signal resulting when wind pressure

makes the diaphragm stick to the wall. This is a structural characteristic of the directional microphone. To summarize, these noises can be categorized as:

- Stationary noise
- Non-stationary noise
- Clipping signal that occurs under wind pressure

Microphone systems that would be effective against these noises were studied.

2.2. Non-stationary Noise and Super-directive Digital Signal Processing

A typical non-stationary noise in the cabin is the sound of passengers talking. Occasions when multiple passengers are in the car could include carrying children to and from school, going shopping at a mall, going to a restaurant for a meal, going on a family trip, and the like. On occasions like these, car navigation systems and other information equipment experience a higher frequency of operation.

Another factor is that the computations in speech recognition processing are performed on the basis of voice characteristics. Therefore, the voices of passengers have characteristics that seriously affect recognition when there is noise. A suppression ratio of 20–30 dB is needed for passenger voices in order to counter interference with recognition of the driver's voice. Since a signal-to-noise ratio of slightly less than 10 dB is the limit for recognition in white noise or similar noise, this is a challenging issue.

One method for suppressing non-stationary noise is to

use multiple microphones and distinguish the needed sound from the incoming direction of the sound source. This includes delay and sum beam forming, adaptive beam forming, and other such methods of directional suppression⁽¹⁾. Another approach is to distinguish the needed sound using independent component analysis, which assumes independent output from multiple noise sources⁽²⁾. With directional suppression, however, a high suppression ratio often cannot be achieved unless there is a sufficient number of microphones. The independent component analysis imposes a heavy computational load that makes it impractical for on-board applications. Therefore, a new super-directive digital signal processing method was selected⁽³⁾. This yields the targeted high noise suppression ratio for using two microphones.

Figure 1 shows the principle of the new super-directive algorithm. As indicated in Fig. 1 (a), when the sound waves of the driver's voice reach the two microphones, there will be a certain time difference between them according to the directions they come from. When this time difference is subjected to frequency axis conversion in a graph, the driver's utterance will be plotted as a nearly straight line sloping upward to the right, as shown in Fig. 1 (b), and it will be distinctly separate from the utterance coming from the passenger seat.

The effect of the new super-directive algorithm is shown in Fig. 2. Figure 2 (a) shows the sound pressure level of passenger utterances in a time series where the driver and passenger utterances occur alternately. When this is processed using the new super-directive algorithm, there is powerful selective suppression of just the

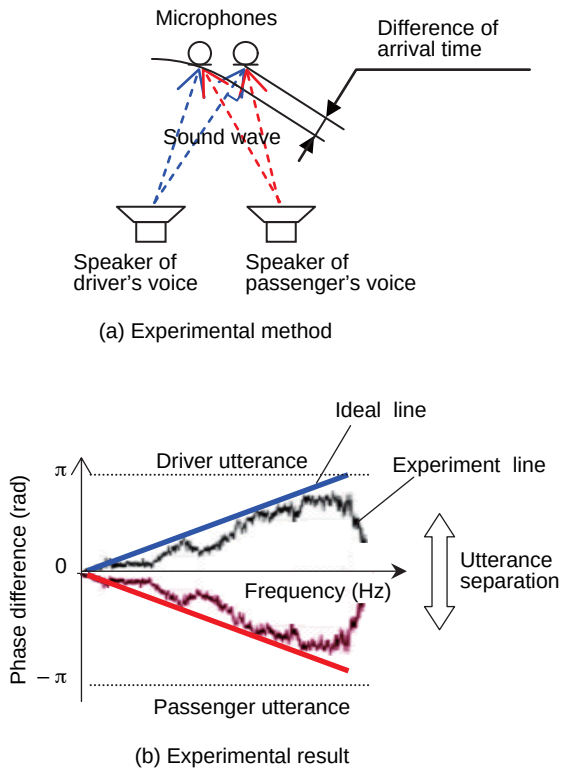


Fig. 1 Principle of channel separation between driver utterance and passenger utterance

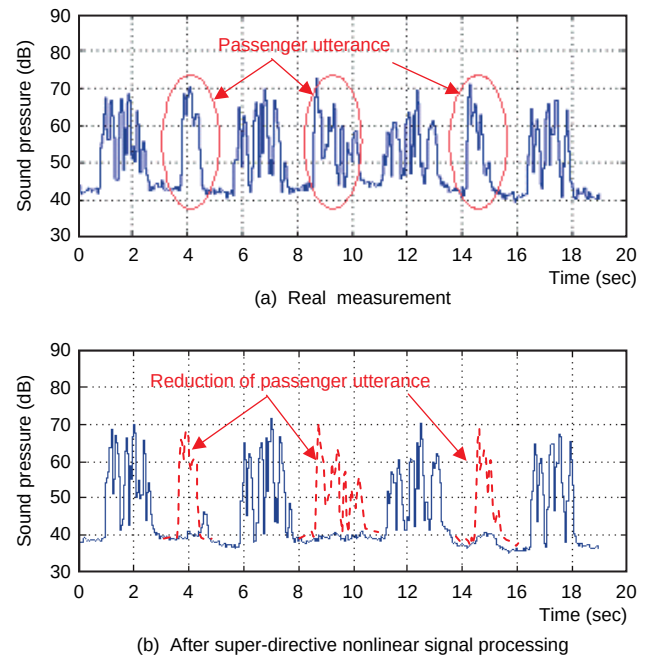


Fig. 2 Effect of channel separation

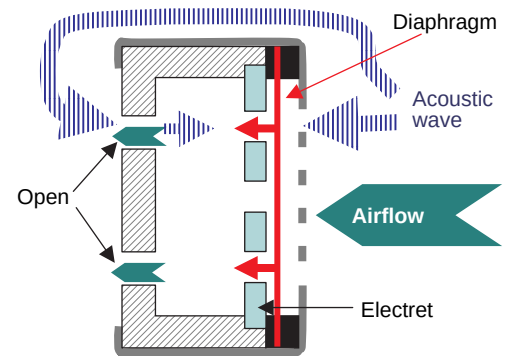
passenger utterance, as shown in Fig. 2 (b). The system is capable of picking out just the driver's utterance. Although done by directional suppression, this enables utterances by the driver to be distinguished from the crowded environment as though by the cocktail party effect in human hearing⁽⁴⁾.

2.3. Clipping Signals from Wind Pressure and the Omni-directional Microphone Element

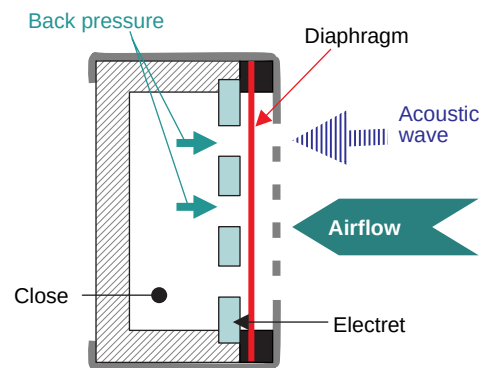
Directional microphone elements are ordinarily used as the microphone elements for on-board speech recognition and for hands-free telephones in order to selectively pick up the driver's voice. As shown in Fig. 3 (a), however, when the wind from an air conditioner is directed toward the microphone, the microphone will output a clipping signal with the voice signal in the case of a directional microphone element^{(5), (6)}. As shown in Fig. 4 (a), this happens because of the microphone structure, which is designed for directionality. The sound pressure against the front surface of the diaphragm and the sound pressure that leaks in from behind, against the back surface of the diaphragm, interfere with each other. In other words, the diaphragm in a directional microphone element is free of back pressure, and the diaphragm is readily displaced by wind pressure from in front, and by vibration or other factors apart from sound pressure that impinge on the microphone element. As a result, the diaphragm sticks to the front or back wall, generating a clipping noise. In contrast to this, as shown in Fig. 4 (b), is the omni-directional microphone⁽⁷⁾. This has back pressure to limit large displacements of the diaphragm, so that wind pressure and other such factors will not readily make the diaphragm stick to the wall. Therefore, as shown in Fig. 3 (b), no clipping signal is output.

The microphone in the cabin is attached at the front part of the roof, as shown in Fig. 5. This layout places

the microphone where the airflow from the dashboard vent will blow on it. In this situation, an omni-directional microphone of the closed cavity type that is not influenced by wind pressure (strictly speaking, a microphone with vents to bleed off pressure differentials so as to cancel out atmospheric pressure) is very resistant to wind pressure. Digital signal processing using two microphone elements provides directionality toward the driver and provides sound quality as good as or better than a directional microphone element. This made it possible to use omni-directional microphones, which perform well under wind pressure.



(a) Directional microphone



(b) Omni-directional microphone

Fig. 4 Comparison of microphone structures

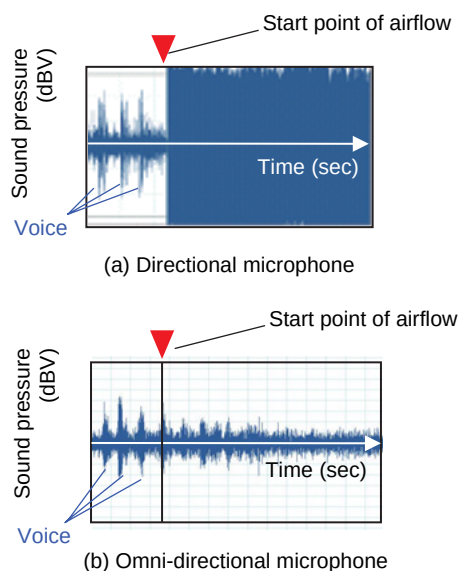


Fig. 3 Comparison of acoustic waves



Fig. 5 Direct airflow from air conditioner

2.4. Stationary Noise and Analog High Pass Filters

Various methods have been devised for signal processing to suppress stationary noise, such as spectral subtraction. When the noise reduction rate is raised, however, the processed result tends to have greater sound distortion relative to the original sound. When the sound distortion increases, unmatching occurs with the acoustic model that is designed for phoneme selection in the speech recognition computation described in a later section. This involves an issue of diminished recognition accuracy.

The omni-directional microphone with the frequency response characteristics shown in Fig. 6 (a) cannot attenuate the distinctive stationary noise in the cabin, which is concentrated at the lower frequency range. In this it differs from the directional microphone element with the low frequency attenuation response shown in Fig. 6 (b).

Therefore, a method of suppressing stationary noise was sought that would not necessitate the strong suppression performed by digital signal processing while also addressing the issues of the omni-directional microphone. The method adopted was to add an analog high pass filter with low frequency attenuation response that has an even steeper slope than the directional microphone element, as shown in Fig. 6 (c), and place it

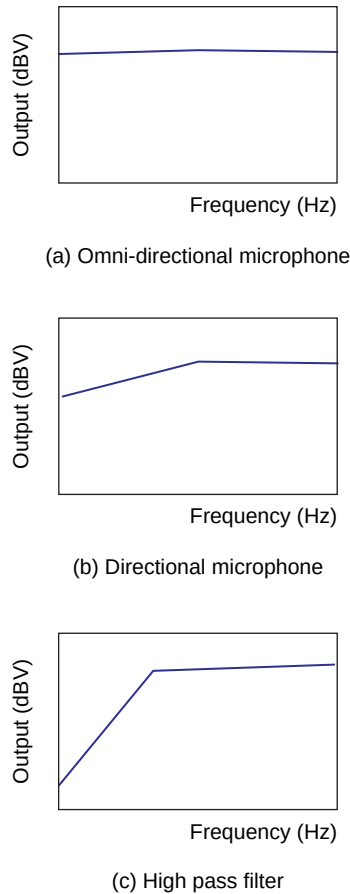


Fig. 6 Frequency response of microphone and high pass filter

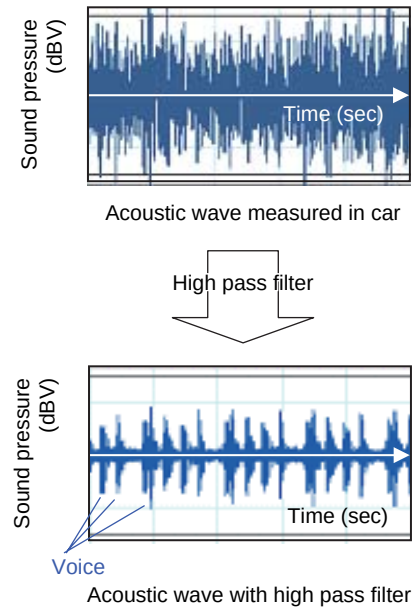


Fig. 7 Effect of high pass filter with omni-directional microphone

before the analog-digital conversion process. The crossover frequency of the high pass filter was set at or below the frequency needed by the speech recognition application in use.

Filtering is done at several tens of frequency bands during speech recognition computation. Since this is done before conversion from analog to digital, however, the information entropy (self-information) of the digital signal can be increased and the quality of the signal after processing enhanced. The effect is shown in Fig. 7. The background noise component at low frequencies is suppressed and the generated signal has a better signal-to-noise ratio.

3. In-vehicle Noise Suppression System

Sections 2.2 to 2.4 raise various issues with the in-vehicle noise environment. In order to address these issues, a method was developed to utilize the new super-directive algorithm in combination with the omni-directional microphone and the analog high pass filter to provide higher background noise suppression performance.

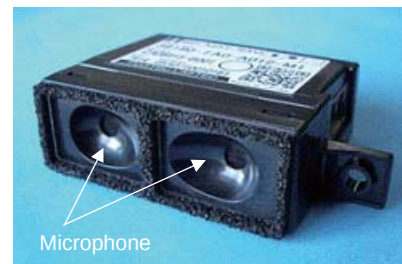


Fig. 8 Digital signal processing microphone unit

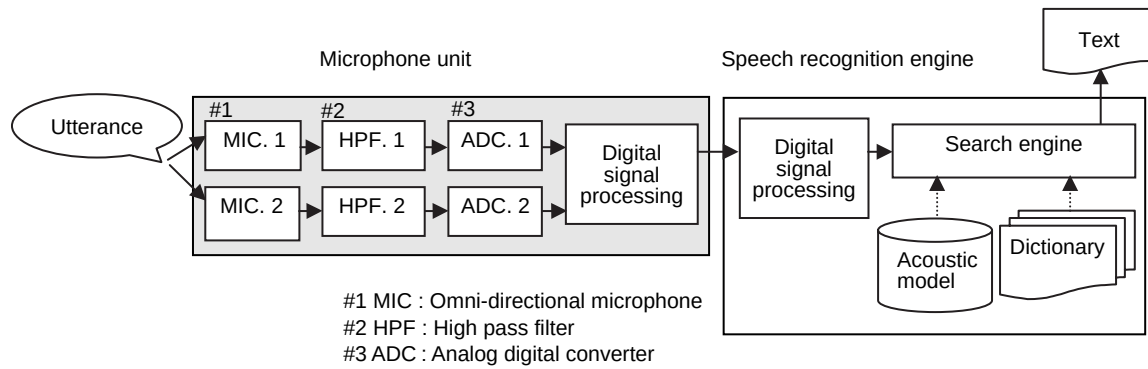


Fig. 9 Block diagram of voice recognition system

A summary view of the developed background noise suppression microphone system is shown in Fig. 8.

A flow chart of the process is shown in Fig. 9. Driver utterances are converted into electrical signals by the two microphone elements. The two signals go through their high pass filters and then undergo analog-to-digital conversion. These two signals are input to digital signal processing and subjected to noise suppression processing using the super-directive algorithm. The above describes the processing performed within the microphone unit. The flow calls for the signals output by this unit to be handed over to the "Speech recognition engine" in the car navigation unit. This enables current car navigation units to be incorporated into the system and used without modification.

4. Results

4.1. Non-stationary Noise Suppression Effect

Tests were performed while driving in an actual vehicle to compare speech recognition using the new and old microphones with utterances by the driver alone, and with simultaneous utterances by the driver and passenger. The sound pressure of the passenger utterances was equal to that of the driver utterances. Also, the periods of passenger utterance were made to overlap with and last longer than the driver utterances.

The results are shown in Fig. 10. The system provided a high level of recognition accuracy even when there were passenger utterances.

4.2. Air Conditioner Airflow Clipping Signal Suppression Effect

Tests were conducted to compare speech recognition using the new and old microphones while idling with the air conditioner turned off, with the fan turned to the highest setting and the airflow directed entirely to about shoulder level, and likewise with the airflow directed overhead (toward the microphones). Figure 11 shows the results.

The use of closed-back-type omni-directional microphones yielded high levels of recognition accuracy even when the air conditioner airflow was directly hitting the microphone elements. Having the air conditioner fan

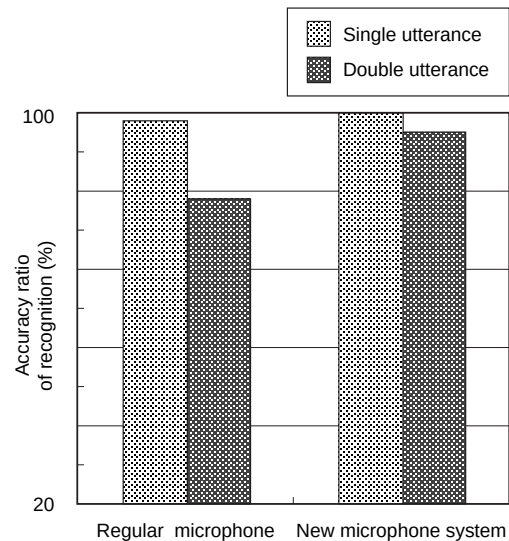


Fig. 10 Recognition accuracy for single utterance and double utterance

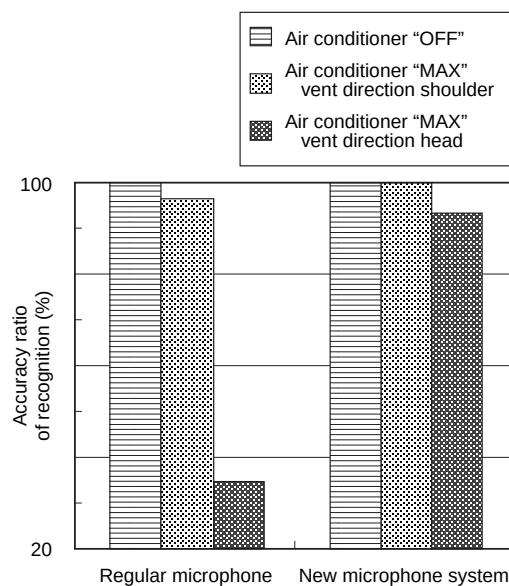


Fig. 11 Recognition accuracy with airflow from air conditioner

turned to the highest setting generated a large amount of noise in itself. This happens because the sound of turbulent flow from the air conditioner blower vent generates a resonance sound determined by the cabin volume. As will be described later, this is a type of wind-throb phenomenon⁽⁸⁾. The fact that the system provided a high level of recognition accuracy, even though the noise itself was very loud, indicates that the analog high pass filter described in section 2.4 was operating effectively, as well.

4.3. Stationary Noise Suppression Effect

Speech recognition tests were conducted with windows closed and with windows open. When cruising with open windows, the cabin noise reached 70 dBA or more. Utterances with ordinary sound pressure are covered over by the noise at that level, and a usable signal-to-noise ratio cannot be obtained. Therefore, this testing was conducted with utterances at a higher sound pressure in order to obtain a signal-to-noise ratio of several decibels.

The noise was very loud and conversation was complex when the windows were open, but, as shown in Fig. 12, the system provided a level of recognition accuracy more or less equal to when the windows were closed. The wind noise that occurs when windows or sunroof are open is a sound produced by a wind-throb phenomenon. It is generated by the sound of turbulent flow coupled with the Helmholtz resonance with the cabin volume. Consequently the central frequencies of the sound are constant regardless of the vehicle speed, and the on-board microphones are influenced at approximately 300 Hz or lower, including the harmonic overtones of the resonance frequencies. The noise resulting from this resonance sound has been suppressed by steep analog high pass filters, making it less likely that recognition accuracy would be decreased.

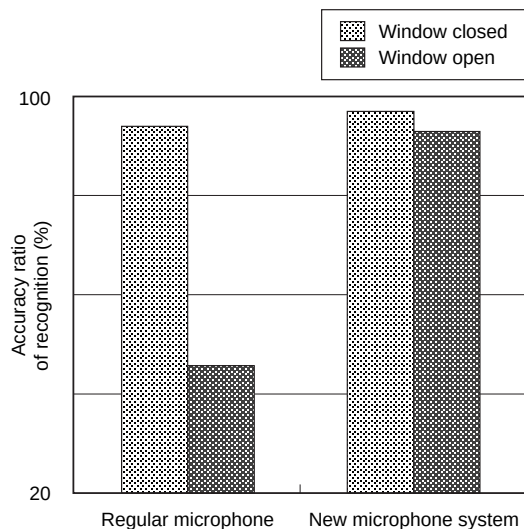


Fig. 12 Recognition accuracy with window closed and window open

5. Conclusion

In order to enhance the speech recognition accuracy of the system under the conditions of actual vehicle use, a microphone system was developed that combines a new super-directive algorithm with a high directional noise suppression ratio, omni-directional microphones, and analog high pass filters. These elements in combination complement each other for enhanced system feasibility and produce high levels of speech recognition accuracy, not only with ordinary road noise, but also with the wind throbb phenomenon that accompanies wind noise when windows or sunroof are open, with air conditioner noise, with air conditioner airflow hitting the microphones directly, and even with the sound of the person in the front passenger seat or other occupants talking.

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■ Author ■



Toshiya SHIKANO



Junichi WATANABE



Naoshi MATSUO

Development of Laminated Windshield Glass Antenna for Digital Terrestrial TV

Tomoyuki FUKUMARU*
Hajime SAITO*

Ken ASAMI*
Nobuyuki INAMI*

ABSTRACT

A front laminated windshield glass antenna for on-board digital terrestrial TV reception was developed. In order to reduce the visual unpleasantness of conventional silver-printed antenna lines, the new antenna structure consists of a copper thin film antenna with a line width of 0.25 mm clasped inside laminated windshield glass. In addition, the amplifier is mounted directly to terminals on the glass surface using capacity coupling with the glass as the dielectrics. As a result, this front laminated windshield glass antenna obtained flat directivity over the front 180° range, and ease of maintenance has also been secured. In addition, the thermal stress when soldering the base plate of the amplifier to the glass surface was confirmed to be sufficiently less than the permissible stress of the glass.

1. Introduction

Conventional on-board analog TV antennas configure the diversity by forming multiple antennas inside the rear windshield glass. In this first development of a digital terrestrial TV system, the diversity is configured using both front and rear windshield glass antennas to realize better receiving performance as a moving object.

This is the first time the antenna has been mounted to the front windshield glass, so the development goals were to secure adequate receiving performance and address the following issues.

- (1) The antenna lines are to be located within the field of view of persons in the front seat, so there should be no visual unpleasantness.
- (2) In the conventional method for connecting an analog TV antenna and coaxial cable, the lead wire produces a large drop in receiving performance, so the structure should form the amplifier directly below the antenna.
- (3) A method for connecting the antenna lines clasped inside the windshield glass with the amplifier should be established, and reliability secured.

2. Structure

2.1. Analog TV Antenna Structure

Figure 1 shows the structure of a conventional windshield glass antenna for analog TV reception. The antenna is formed by printing silver on the glass surface,

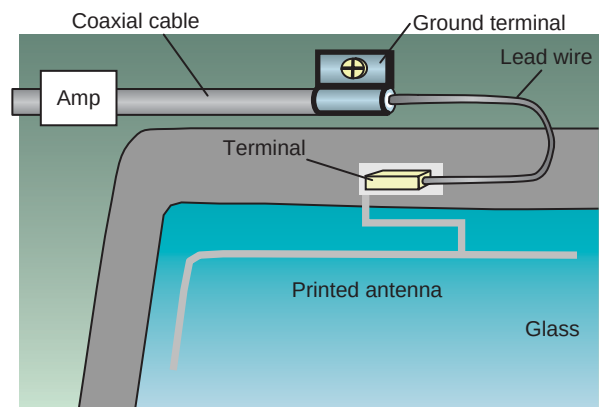


Fig. 1 Current analog TV antenna structure

* Automobile R&D Center

and has a structure that connects to a coaxial cable consisting of a lead wire and a ground terminal. Analog TV broadcast waves have a wavelength of approximately 42 to 280 cm in VHF and UHF band. When a 10 to 20 cm lead wire is connected to the antenna, the lead wire becomes part of the antenna and offsets the resonance frequency, creating the issue of unstable receiving performance. In addition, an amplifier is installed to compensate for signal loss along the coaxial cable.

2.2. Digital Terrestrial TV Antenna Structure

Digital terrestrial TV broadcasts have a wavelength of approximately 42 to 64 cm in UHF band. This is a short wavelength compared to the VHF band, so the resonance frequency offset due to the lead wire increases further, resulting in a drop in receiving performance. This issue was addressed by using a dipole type antenna that can form the input and ground terminals on the glass, and mounting the amplifier directly to the glass surface. This structure enabled the lead wire that had affected performance to be removed. Figure 2 shows a photograph of the antenna, and Fig. 3 shows the internal structure.

To reduce visual unpleasantness, the antenna element used copper thin film punched into the antenna shape, which reduced the line width to 0.25 mm compared to 0.7 mm for conventional silver-printed antenna lines. The structure clasps this antenna element inside laminated windshield glass, and fixes the amplifier on the inner glass surface. The antenna is laminated between the inner glass and the middle film of the laminated windshield glass during the manufacturing process, and the amplifier and antenna are connected by a non-contact structure that uses capacity coupling with the glass as the dielectrics.

A base plate for holding the amplifier is fixed to the inner glass surface by urethane adhesive, and the base plate and silver-printed terminal are connected by solder. The amplifier is secured to the base plate by screws, and a screw-lock structure is used to help prevent the screws from loosening and falling out due to vibration or other

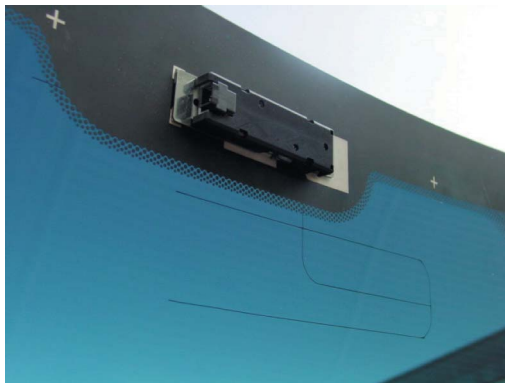


Fig. 2 Laminated windshield glass antenna and amp module

factors. This also enables replacement should the amplifier malfunction, in consideration of maintainability.

2.3. Antenna Structure and Principle

Figure 4 shows the copper thin film antenna structure and principle. This copper thin film antenna has a dipole terminal, and the element is bent in a “U” shape with a branch element and a loop element that create multiple resonance points and expand the bandwidth. A wide bandwidth of 300 MHz is realized by using an antenna element length of $3/4$ wavelength on the low-frequency side, and approximately $1/4$ wavelength on the high-frequency side.

2.4. Capacity Coupling Type Power Supply Terminal

One possible method of leading a terminal from the antenna clasped inside the windshield glass is to lead the copper thin film line that comprises the antenna from the

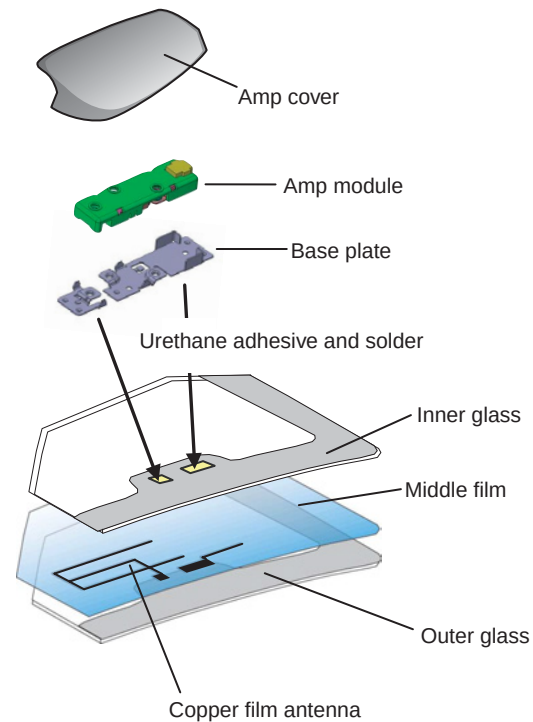


Fig. 3 Digital TV antenna structure

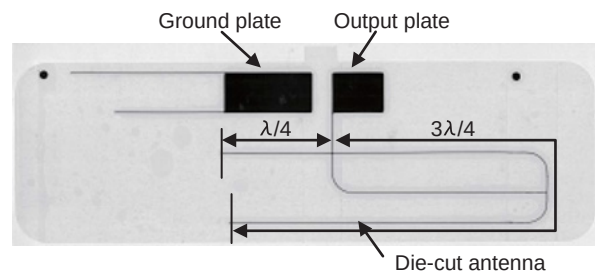


Fig. 4 Photograph of die-cut antenna

edge of the laminated windshield glass for connection. However, this method has the following issues:

- (1) A complex structure is needed to lead the antenna line from inside the laminated windshield glass, which not only adds a glass cutting process, but also raises concerns of disconnection.
- (2) The lead wire portion becomes part of the antenna and brings about a drop in performance due to interference with the vehicle body, electric disclosure, and the like.

These issues were addressed by printing the silver-printed terminal on the inner glass in a position directly opposite the antenna terminal to create a non-contact terminal structure that uses the capacity coupling method with the glass as the dielectrics. Figure 5 shows the relationship between the electrode area and electrostatic capacitance, and Fig. 6 shows the relationship between the electrode area and antenna loss, for glass with a dielectric constant of 7 and thickness of 2 mm. The electrode area was set at 500 mm², which provides stable characteristics at a UHF bandwidth of 470 MHz or more, and the electrostatic capacitance in this case is approximately 15 pF. This configuration does away with the need to lead a line from the copper thin film antenna to the terminal, and creates a simple power supply terminal with low loss.

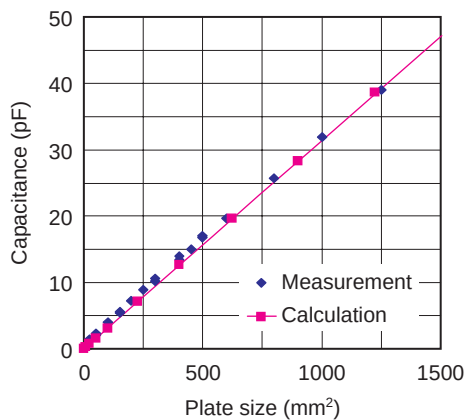


Fig. 5 Relationship of plate size and capacitance

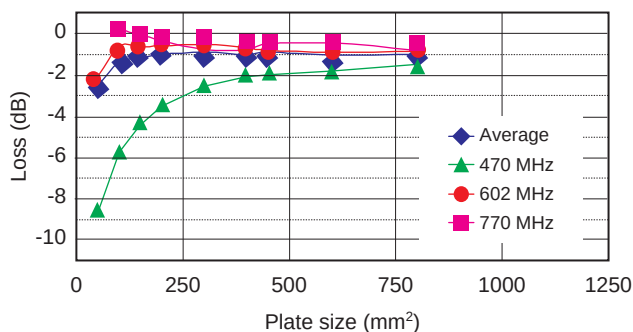


Fig. 6 Relationship of plate size and gain loss

3. Experiments and Results

3.1. Antenna Standalone Simulation

Standalone antenna design was performed by electromagnetic field simulation using the moment method. Figure 7 shows the return loss calculation results. By providing two resonance points at 515 MHz and 710 MHz, a return loss of -5 dB or less was obtained over the digital terrestrial TV band (470 to 770 MHz).

Figure 8 shows the directivity calculation results obtained by simulation. These results show that uniform directivity was obtained with no depression over the front 180° range.

Next, the electromagnetic field of the overall vehicle body with the antenna installed at the top of the front windshield glass was simulated. Figure 9 shows the

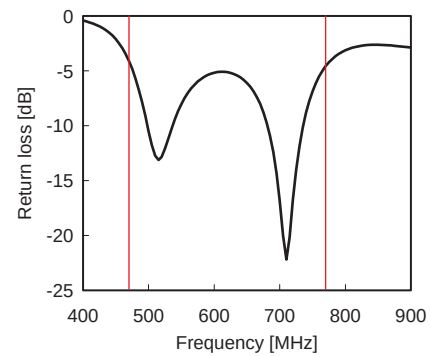


Fig. 7 Return loss

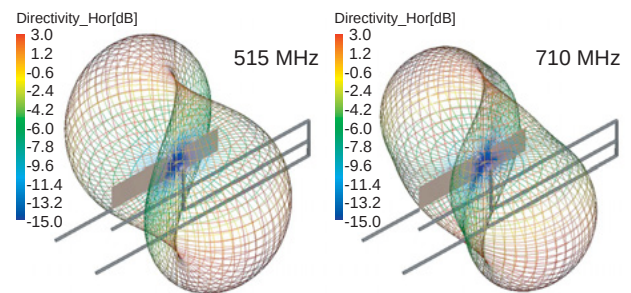


Fig. 8 Antenna directivity of horizontal polarization

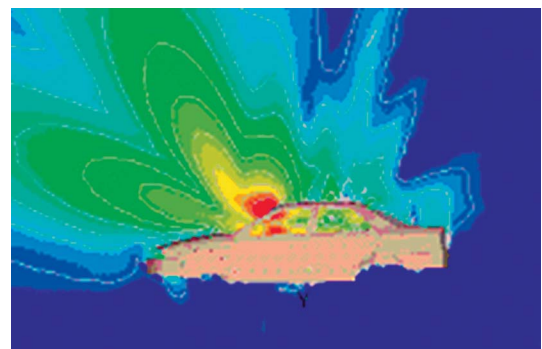


Fig. 9 Electric field at 575 MHz

electric field at 575 MHz. This shows that good radiation properties were obtained in the vehicle front direction.

3.2. On-vehicle Antenna Measurement Results

The developed antenna has diversity from the windshield glass antennas at both the front and back of the vehicle, and the front windshield glass antenna has a reception area of 180° in the vehicle front direction. Figure 10 shows directivity of the on-vehicle copper thin film antenna at 617 MHz. Uniform directivity with no depression could be obtained over the front 180° range. In addition, the measurement results closely matched the simulation.

Figure 11 shows the frequency - gain characteristics of the antenna. A fairly flat frequency response was obtained over the bandwidth from 470 to 770 MHz.

3.3. Reliability when Mounting the Amplifier on the Windshield Glass

The front windshield glass deicer terminal has conventionally been soldered using a terminal on the inner surface of the outer glass. This development is the first to use a method that fixes the connection terminal and the base plate to the terminal on the inner glass surface by soldering, so the thermal stress generated on the glass

surface at this time was simulated. Figure 12 shows the temperature and thermal stress when a heat spot 6 mm in diameter, which is the soldering standard, is heated for 5 s at 300°C , with an initial glass surface temperature of 10°C . The maximum generated thermal stress was confirmed to be 17 MPa, which is sufficiently less than the permissible stress of the glass of 30 MPa. In addition, Fig. 13 shows a simulation of how the glass temperature at the center point of the heat spot and the generated thermal stress change with the heating time. This shows that the glass temperature rises gradually and reaches 235°C after approximately 30 s, but the maximum thermal stress peaks around 5 s and then decreases thereafter. Actual soldering tests were also performed, confirming that there are also no soldering issues present.

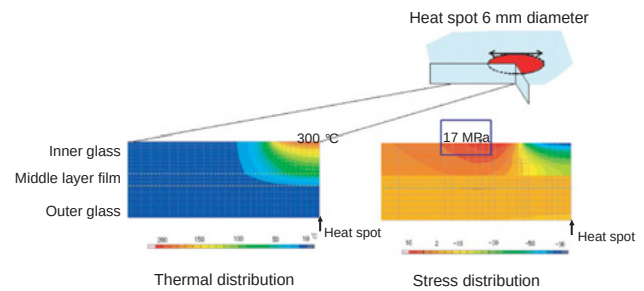


Fig. 12 Simulation of thermal stress in windshield glass

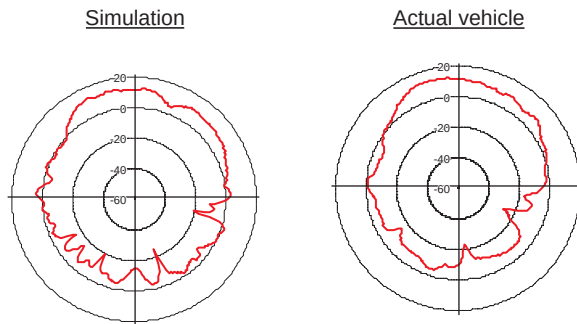


Fig. 10 Antenna gain directivity at 617 MHz

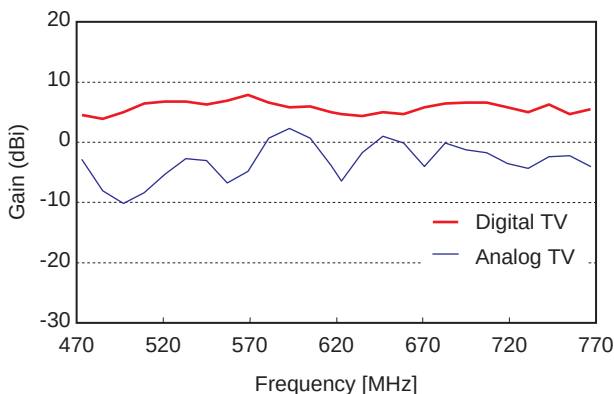


Fig. 11 Average antenna gain in front 180 deg direction

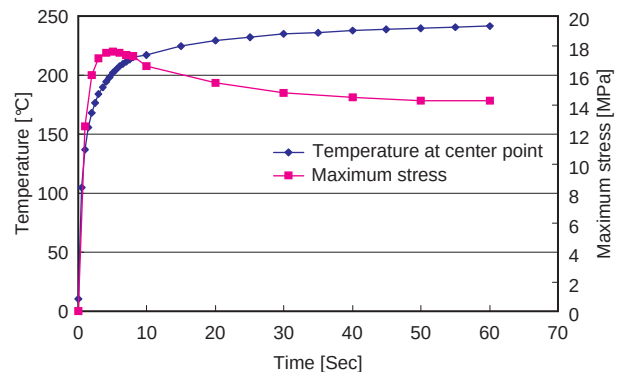


Fig. 13 Relationship of glass temperature and stress force

4. Conclusion

A front laminated windshield glass antenna for digital terrestrial TV reception was developed that provides good receiving performance and reliability.

- (1) A 300 MHz bandwidth was realized by using a $1/4 \lambda$ to $3/4 \lambda$ element.
- (2) Visual unpleasantness in the forward field of view was reduced by clasping a copper thin film antenna with a

- line width of 0.25 mm inside laminated windshield glass.
- (3) A non-contact terminal that uses the capacity coupling method with the glass as the dielectrics enabled the amplifier to be mounted directly below the antenna.
 - (4) The maximum thermal stress when soldering to the terminal on the inner glass surface was verified to be within the permissible range in both simulations and experiments.

■ Author ■



Tomoyuki FUKUMARU



Ken ASAMI



Hajime SAITO



Nobuyuki INAMI

Introduction of Honda Driving Support System – Field Operation Test (FOT) with Road Infrastructure Systems (Driving Safety Support System: DSSS) –

Yuriko INO* Tomo FUJIMAKI* Nobuyoshi ASANUMA*

ABSTRACT

Honda has been taking part in the Driving Safety Support System (DSSS) of the Universal Traffic Management Society of Japan (UTMS Society). Based on an analysis of traffic collisions, Honda has engaged in the two issues below by developing an operation test vehicle and promoting research.

- Dealing with conditions where the counterpart vehicle is not readily visible (due to structures, trucks, buses, or other such shielding obstructions)
- Dealing with motorcycles as vulnerable parties in traffic

The research has involved the application of road infrastructure technology (use of information from sensors installed at the roadside) and effect verification on public roadways. The results indicate that this approach can be expected to effectively reduce collisions.

1. Introduction

The traffic collision situation in Japan continues to be treated as a major issue for society. It was in this context that the government announced the “New IT Reform Strategy” in January 2006 that called for traffic collision deaths to be reduced to 5000 or fewer by the end of 2012, with the goal of making Japan’s road traffic the safest in the world. Under such premise, the ministries and agencies concerned have collaborated with private sector enterprises on a Driving Safety Support System that makes use of IT. This system is to be constructed, subjected to operation tests and effect verification, and evaluated quantitatively for its contribution to collision reduction. Under the supervision of the National Police Agency, the UTMS Society has promoted research on the Driving Safety Support System incorporating road-to-vehicle communications technology. Field operation tests have been conducted in four prefectures (Tochigi, Aichi, Kanagawa, and Hiroshima)^{(1), (2)}.

Honda conducted field operation tests (FOT) in Utsunomiya City, Tochigi Prefecture, in FY2007. These were particularly concerned with formulating system specifications to enhance the visibility of motorcycles.

Honda’s research sought to make the system compatible with actual vehicles, and aimed for practical application.

This paper will introduce background information on the traffic collision situation and describe the system configuration and road-to-vehicle communication technology. It will also describe the public demonstration that was conducted and the course of evaluation of the system’s effectiveness in traffic collision reduction.

Joint field test is planned to take place in Tokyo in FY2008, under the auspices of the Cabinet Secretariat. This operation testing is slated to take place with the collaboration of the National Police Agency, the Ministry of Internal Affairs and Communications, the Ministry of Economy, Trade and Industry, the Ministry of Land, Infrastructure and Transport, and private enterprise.

2. Background

As indicated by the composition of traffic collisions by type based on traffic collision statistics (ITARDA)(Fig. 1), the greatest number of fatal collisions between vehicles occur as crossing incidents, at 17%, while rear-end incidents are the type with the greatest number of collisions resulting in either injury or death, at 32%. Left-

* Automobile R&D Center

turn collisions, right-turn and straight collisions (incidents involving a vehicle making a right turn and a vehicle proceeding straight), and other such intersection system collisions are also numerous. Figure 2 shows the composition of traffic collisions between passenger cars and collisions in which the damaged party is a motorcycle. The most numerous type in both fatal collisions and collisions resulting in injury or death is the crossing collision. This type, together with the left-turn, right-turn collisions, makes up 60% or more of the total. Figure 3 shows the human factors involved in right-turn collisions by counterpart vehicle. In collisions where the counterpart vehicle is a motorcycle, the human factors of “no confirmation of safety” and “less confirmation of safety”

were found to account for 80% of the total. In other words, delay in recognition by the drivers of passenger cars was the main causal factor. It can be inferred that the motorcycle riders on the damaged side also showed some “inadequacy of prediction” or other mistakes in judgment.

It can be inferred from collision background of this kind that providing each of the two sides, both passenger cars and motorcycles, with information about the presence of the other could inhibit the occurrence of delays in recognition. This appears likely to help prevent such collisions before they can happen.

The delay in recognition on the part of the passenger car drivers that is a cause of these collisions may occur

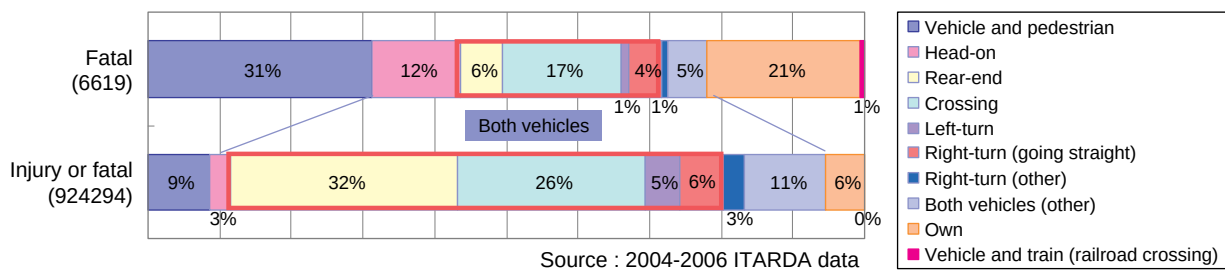


Fig. 1 Traffic collision occurrence rate classified by collision type

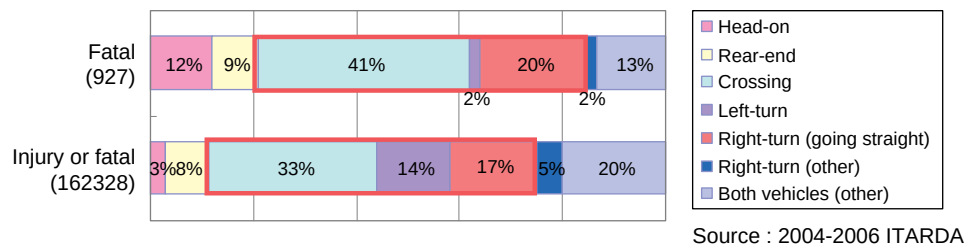


Fig. 2 Traffic collision occurrence rate classified by collision type (vehicle and motorcycle)

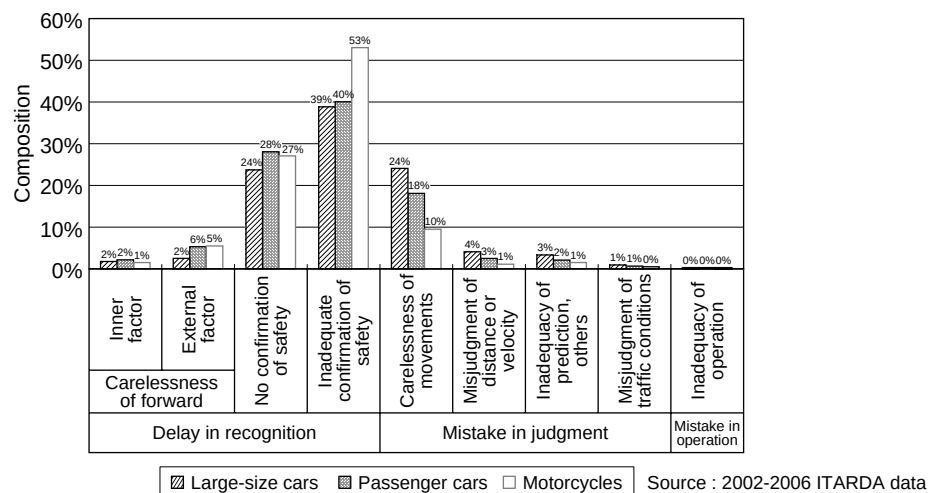


Fig. 3 Human factors in right-turn collision

because of the factors below⁽³⁾.

- (1) The motorcycle profile is so small that the driver misjudges the distance from the motorcycle and the vehicles' approach velocity.
- (2) The motorcycle profile is so small that it is buried among surrounding vehicles and background, and the driver overlooks it.

Implementation of measures that address these causal factors can be expected to enable detection of motorcycles so that fewer misjudgments of vehicle distances and approach velocity are likely to occur.

The Honda DSSS was formulated in light of the above collision background information, allowing it to differentiate the system from the DSSSs in other regions. Honda accordingly implemented a rear-end collision avoidance system as well as left-turn and right-turn collision avoidance systems that provide information about the presence of approaching motorcycles.

3. System Construction

A road-to-vehicle communication system that coordinates the use of infrared beacons and dedicated short range communication (DSRC) was built.

3.1. Road-to-Vehicle Communication System

Figure 4 shows a diagram of the system configuration. Figure 5 shows the FOT vehicles.

3.1.1. Roadside processing

An imaging sensor is used to detect the presence of vehicles in the coverage area. Motorcycles and passenger

cars are differentiated, and that information is transmitted to the DSSS central unit, together with position and vehicle velocity information. The vehicle information is passed through the DSSS central unit and transmitted from the DSRC roadside unit to vehicle DSRC antennas at 100 msec intervals. At the same time, information on the current status of traffic signals acquired in real time from the traffic signals is also transmitted.

The infrared beacon roadside unit is used to transmit relevant vehicle lane information to passing vehicles equipped with infrared beacon receivers.

3.1.2. On-board unit

The on-board unit receives the vehicle lane information sent from the infrared beacon roadside unit. Using that information, the on-board unit performs location position estimation. The information received from the DSRC roadside unit together with the vehicle's own status



Fig. 5 Honda DSSS FOT cars

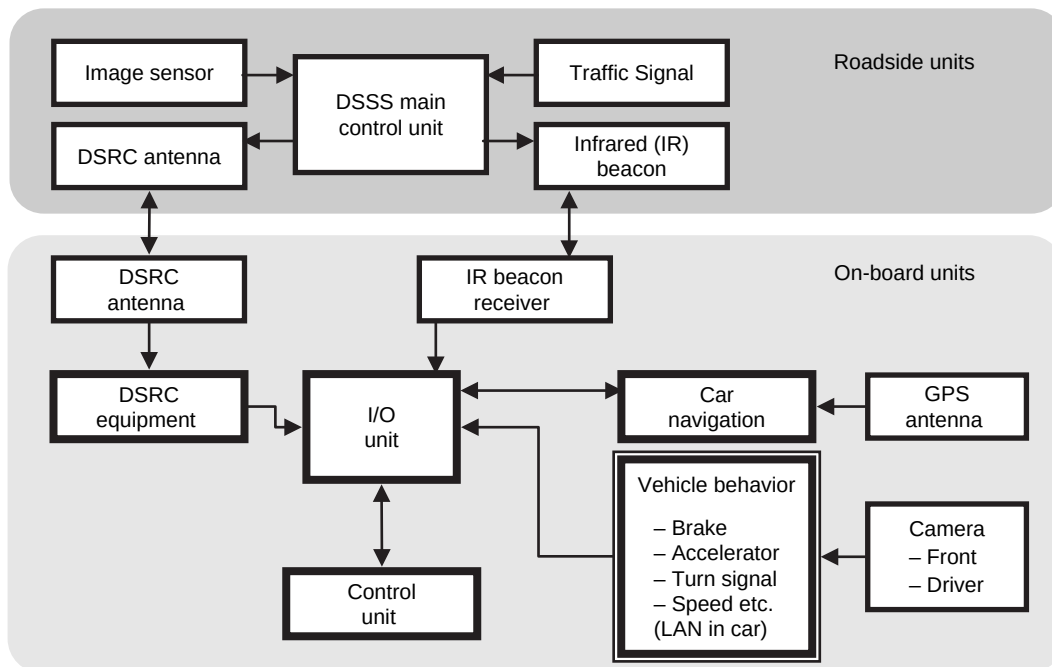


Fig. 4 System configuration

information is then used to judge the likelihood of a collision. The system provides the driver with information on the presence of a counterpart vehicle only when traffic signal current status information indicates a green light.

3.2. Equipment Specifications

Specifications of the equipment used in this system are as shown below.

(1) Infrared beacon

- Communication method: Near-infrared communication
- Wavelength: 850 ± 50 nm
- Transfer speed: Downlink 1024 kbps, uplink 64 kbps
- Communication range: Approximately 3.7 m longitudinally, 3.5 m laterally

(2) DSRC

- Carrier frequency: 5.8 GHz
- Transfer speed: 1024 kbps or 4096 kbps
- Modulation method: QPSK
- Communication cycle: 100 msec

For the imaging sensor, the system employs a sensor developed using spatio-temporal Markov Random Field (MRF) model technology⁽⁴⁾. A spatio-temporal MRF model enables stable, high precision tracking of a moving object to be conducted,

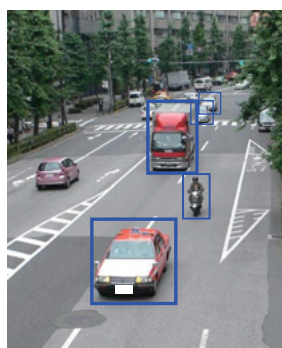


Fig. 6 Image of sensing by MRF model

independent of the type of object, its orientation, movement, or overlap (occlusion). It can also differentiate motorcycles from passenger cars. Figure 6 shows an image of sensing by this method. Rectangular frames indicate that a vehicle is recognized as individual.

3.3. Rear-end Collision Avoidance System

3.3.1. System overview

Figure 7 shows a rear-end collision avoidance system on a road with reduced visibility. The system uses infrastructure equipment to transmit information on the presence of low-speed vehicles and help prevent collisions.

Installation location: Crests and curved locations with reduced visibility

Goal: To help prevent traffic collisions caused by careless driving (compensate for deviations in recognition and judgment)

The rear-end collision avoidance system uses the imaging sensor installed on the roadside to detect the rear end of traffic congestion. Information on the presence or absence of traffic congestion is then transmitted through the DSSS central unit and the infrared beacon to the on-board unit. When the rear end of traffic congestion is present, the system cautions the driver by means of the car navigation display screen and voice announcements. The voice announcement alerts the driver with the words, "Traffic congestion ahead, watch out for rear-end collisions."

3.3.2. System installation conditions

Installation location L_1 (m) for the infrared beacon in the rear-end collision avoidance system is the road distance from the infrared beacon to the rear end of the traffic congestion. This is calculated by Eq. (1).

$$L_1 = V^2/2\alpha + (T_p + T_d) V \quad (1)$$

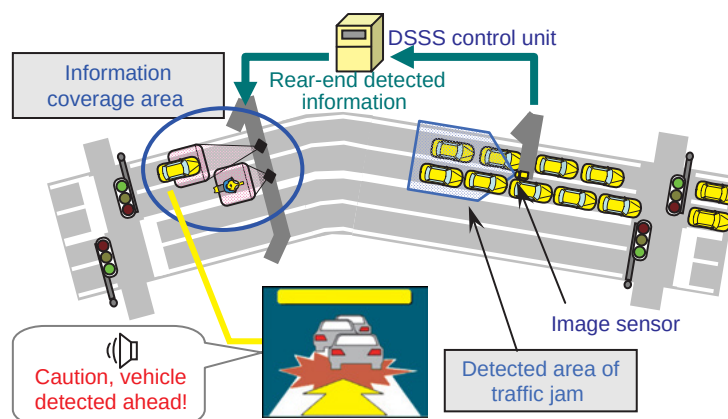


Fig. 7 Rear-end collision avoidance information service

The parameters shown below are preconditions for the function of this system.

- Vehicle velocity V : 13.9 m/s (50 km/h)
- Deceleration α : 0.3 G (G: Acceleration of gravity, 1.0 G=9.8 m/s²)
- Processing time for infrastructure equipment and on-board unit T_p : 2.0 sec
- Driver reaction time T_d : 2.0 sec

3.4. Left-turn Collision Avoidance System and Right-turn Collision Avoidance System

3.4.1. Overview of Left-turn Collision Avoidance System

Figure 8 shows a left-turn collision avoidance system in an intersection with a traffic signal. It helps prevent

collisions by transmitting vehicle information to vehicles making left turns where traffic coming from behind (blind spot) is difficult to see.

Installation location: Outside (roadside) lane at intersections where vehicles approaching from the rear are not readily visible

Goal: Help prevent driving collisions by providing blind spot information (compensate for presence of motorcycle)

Figure 9 shows the flow of logic in the left-turn collision avoidance system. An imaging sensor installed roadside detects a motorcycle approaching from behind, and information on the presence of the motorcycle is transmitted to the on-board unit through the DSSS central

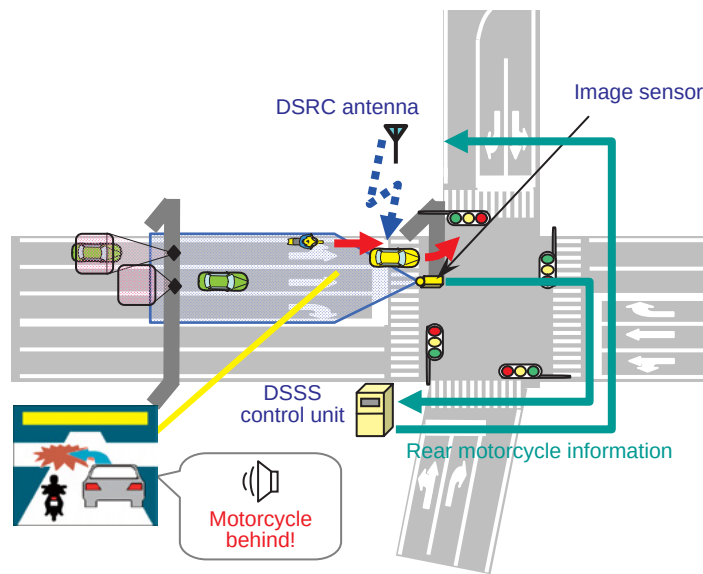


Fig. 8 Left-turn collision avoidance information service

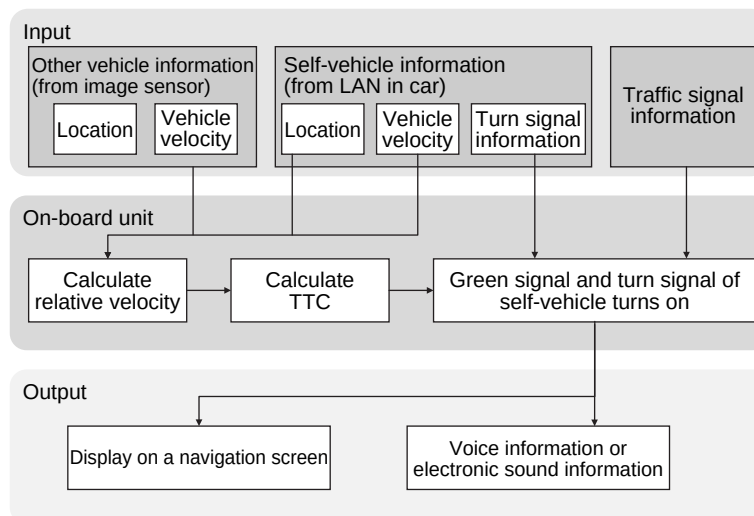


Fig. 9 Algorithm for determination (left-turn and right-turn collision)

unit and DSRC. Vehicle lane information is sent by the infrared beacon and used for location position estimation. The on-board unit computes the time to collision (TTC) using the transmitted information on the counterpart vehicle, together with information on the vehicle's own position and velocity acquired from the on-board LAN. The system cautions the driver using the navigation display screen and voice announcements when traffic signal current status information indicates a green light and the vehicle's own left turn signal is activated. The voice announcement alerts the driver with the words, "There is a motorcycle. Watch out when making a left turn."

3.4.2. Overview of right-turn collision avoidance system

Figure 10 shows the right-turn collision avoidance system in an intersection with a traffic signal. It helps prevent collisions by transmitting information to vehicles making right turns about oncoming vehicles in difficult to see locations.

Installation location: Right-turn lanes in intersections where oncoming vehicles proceeding straight are not readily visible

Goal: Help prevent driving collisions resulting from miscalculation (compensate for misperception of motorcycle velocity and distance)

The same road processing as the left-turn collision avoidance system (3.4.1.) is performed in the right-turn collision avoidance system (Fig. 9). The on-board unit computes the TTC using the transmitted information on the counterpart vehicle, together with information on the vehicle's own position and velocity acquired from the on-board LAN. The system cautions the driver using the navigation display screen, an electronic tone, and voice

announcements when traffic signal current status information indicates a green light and the vehicle's own right turn signal is activated. When the oncoming vehicle is a passenger car, the system uses a combination of the navigation display screen and an electronic beep tone to alert the driver. When the counterpart vehicle is a motorcycle, the system prompts for additional caution using a combination of the navigation display screen and the voice announcement, "Watch out for the oncoming vehicle."

3.4.3. System installation conditions

Installation location L_2 (m) for the infrared beacon in the left-turn collision avoidance system is the road distance from the infrared beacon to the location where collision involvement is predicted. Installation location L_3 (m) for the infrared beacon in the right-turn collision avoidance system is the road distance from the infrared beacon to the center of the intersection (right turn wait line). Both L_2 and L_3 are calculated by Eq. (2).

$$L_2, L_3 = V^2/2\alpha + ((T_p + T_d) + V_i/\alpha - V/\alpha) V \quad (2)$$

The parameters shown below are preconditions for the function of this system.

- Vehicle velocity V : 11.1 m/s (40 km/h) in left turns, 13.9 m/s (50 km/h) in right turns
- Vehicle velocity allowing left turn or right turn V_i : 8.3 m/s (30 km/h) in left turns, 5.6 m/s (20 km/h) in right turns
- Deceleration α : 0.3 G
- Processing time for infrastructure equipment and on-board unit T_p : 2.0 sec
- Driver reaction time T_d : 2.0 sec

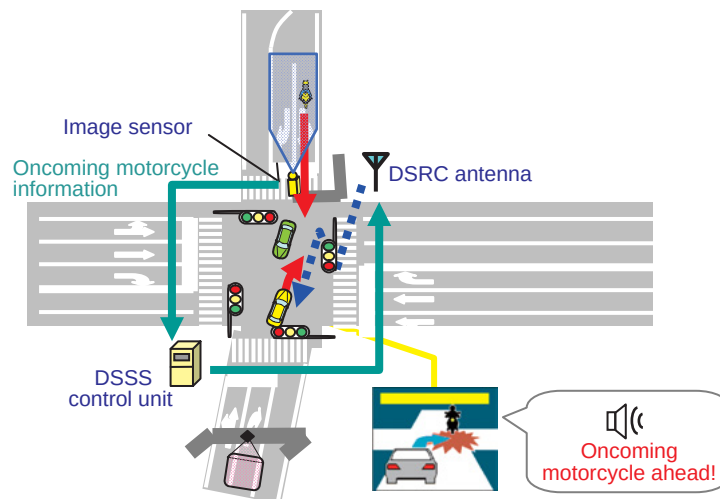


Fig. 10 Right-turn collision avoidance information service

4. Field Operation Test

4.1. Overview

Honda commenced field operation tests in September 2007, and held a public demonstration in November 2007. In addition, system effectiveness in collision reduction will be evaluated by the end of 2008.

4.1.1. Public demonstration

The participating companies jointly held a public demonstration of the Honda DSSS using road-to-vehicle communication on public roads in Utsunomiya City, Tochigi Prefecture from November 5th to 7th. The public demonstration covered the rear-end, left-turn and right-turn collision avoidance systems (Fig. 11). Dedicated on-board units were installed on FOT vehicles, and people were allowed to take test rides to experience the system. An exhibition of equipment, panel displays explaining the system, a driving simulator, and other displays were also held in conjunction with the demonstration (Fig. 12).

The event had a total of 278 visitors over a three-day period. Practically all of the visitors were able to experience a test drive.

4.1.2. Criteria for evaluation of effectiveness

It was decided to evaluate through the FOT the effect



Fig. 11 Situation of right-turn collision avoidance information service



Fig. 12 Exhibition site

of caution by the system on driver awareness, collision avoidance operation, or other driving behavior. For the left-turn collision avoidance system, the evaluation considered the effect on detection of a motorcycle approaching from the rear, and on involvement avoidance behavior. For the right-turn collision avoidance system, it considered the effect on detection of an oncoming vehicle and on collision avoidance behavior. For the rear-end collision avoidance system, it considered the effect on detection of traffic congestion ahead, and on deceleration behavior.

In order to evaluate effectiveness with regard to these various types of driving behavior, discernible indicators in each driving behavior are determined. Then the resulting effects on the indicator in cases when the system provided caution and cases when it did not are compared and verified.

Examples of indicators are shown below.

- The location where brakes or accelerator pedal (accelerator pedal release) are operated with respect to the objective position⁽⁵⁾
- Vehicle velocity change
- Differential time in right and left turns

Figure 13 shows the vehicle velocity change that takes place with left-turn collision avoidance. As a result of caution by the system for approaching motorcycle from behind, collision avoidance behavior such as deceleration occurs earlier. This indicates that an extra margin of time is available in the collision avoidance operation.

For the present testing, an evaluation was made using three FOT vehicles and six test subjects before going on to full-scale operation. The following section introduces the left-turn collision system using an example of results from previous verification testing.

4.2. Field Operation Test Results

4.2.1. Evaluations of the test drive

A questionnaire survey concerning the test ride experience was administered over the three-day period and 258 responses (approximately 93% of the total number of

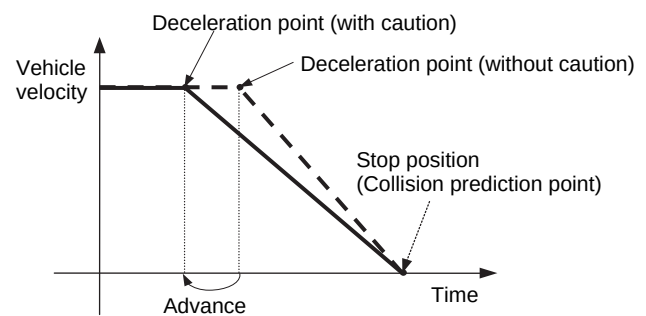


Fig. 13 Vehicle velocity change (left-turn collision avoidance system)

event visitors) were obtained. Figure 14 shows that the number of respondents who said they understood the left-turn collision avoidance system “well” or “somewhat” amounted to 100%. This indicates that even ordinary members of the public, without any prior knowledge of the system, were able to gain some understanding of it. About the necessity for a left-turn collision avoidance system, respondents who found that a system is “needed” or “needed depending on location” amounted to 99% of the total. Similar results were obtained regarding rear-end collision prevention and the right-turn collision avoidance system. Responses to the exhibition content were also favorable.

The above results show that the systems were viewed positively, not only by the people concerned but also by members of the public at large. The findings also confirm that there are people who look forward to widespread use of such systems in the future. These results provided data points that will be considered during future work to optimize the systems.

4.2.2. Left-turn collision avoidance system

(1) Conceptual approach of evaluation of system effectiveness

The intersection where testing of the left-turn collision avoidance system took place has large numbers of motorcycles passing through during the morning commuter period. Drivers who make left turns at the intersection must always watch out for the movement of motorcycles approaching from behind and motorcycles traveling to their left. If drivers delay in noticing a motorcycle approaching from behind them because of inadequate confirmation of safety or inattention, then they may involve the motorcycle in a collision during their left turn. Alternatively, they may experience a close call, for example if they notice the motorcycle just before their left turn and abruptly stop the turn.

Under these kinds of circumstances, drivers are likely to notice motorcycles sooner if they are informed in advance that a motorcycle is present before they make their left turn. Then they will have a greater margin of time for their involvement avoidance operation.

(2) Evaluation items

The driving actions taken by drivers making left turns at this intersection are assumed to be as in the following.

- Drivers decelerate to the target vehicle velocity before starting their left turn. At the left turn start location they conduct a steering operation and make their left turn.

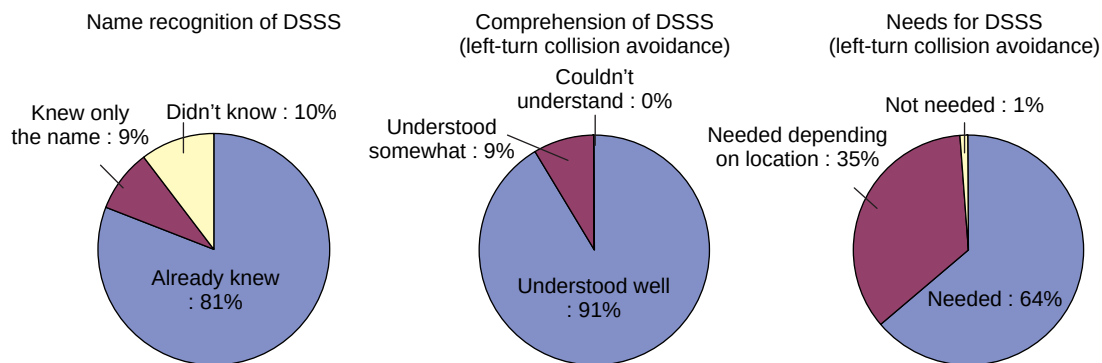


Fig. 14 Questionnaire results

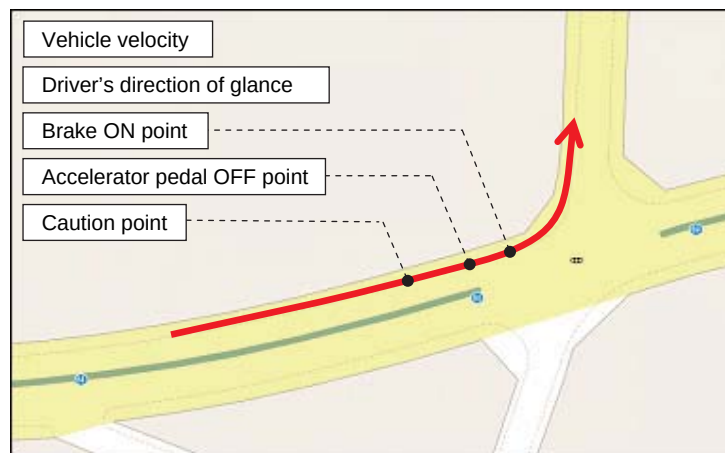


Fig. 15 Evaluation items (left-turn collision)

- When there is a motorcycle approaching from behind, they closely observe its movement up to the point where they begin their left turn, to determine whether safety is maintained.
- When drivers determine that a motorcycle will be passing on their left, they engage in involvement avoidance operation. In other words, they decelerate to a vehicle velocity that will enable them to stop before their left turn start location (the location where collision involvement is predicted).

The evaluation items are vehicle velocity, driver's direction of glance, caution point, brake ON point, and accelerator pedal OFF point (Fig. 15). Verification is

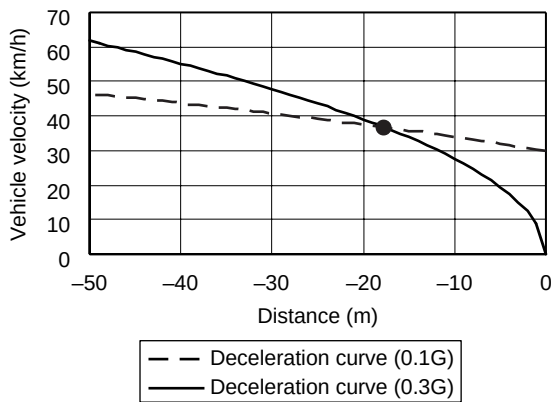


Fig. 16 Deceleration curve (0.1 G, 0.3 G)

made of the system's effects on driving actions.

The following standards were determined for evaluation of driving action: the standard vehicle velocity for starting a left turn at the intersection to be 30 km/h, the deceleration to the standard vehicle velocity due to release of the accelerator pedal to be 0.1 G, and the deceleration due to braking to be 0.3 G. Those deceleration curves (standard velocity curves), which plot velocity versus distance to starting point of each left turn, are shown in Fig. 16.

Distance and velocity at accelerator pedal OFF point were compared to 0.1-G standard velocity curve in order to evaluate the timing of accelerator pedal OFF point. If the accelerator pedal release data are plotted under the standard velocity curve, this would indicate that release of the accelerator pedal occurs earlier. Conversely, if the data are plotted above, this would indicate a delay.

Furthermore, vehicle velocity at the distance before beginning the left turn was compared to 0.3-G standard velocity curve in order to evaluate whether involvement avoidance operation occurred in time. If the vehicle velocity data are plotted below the 0.3-G deceleration curve, this would indicate that the involvement avoidance operation occurs with a margin of time. If the data are plotted above the curve, this would mean that the avoidance takes place with no extra margin of time. The intersection of the two deceleration curves signifies the cutoff point of distance and velocity for quitting the left

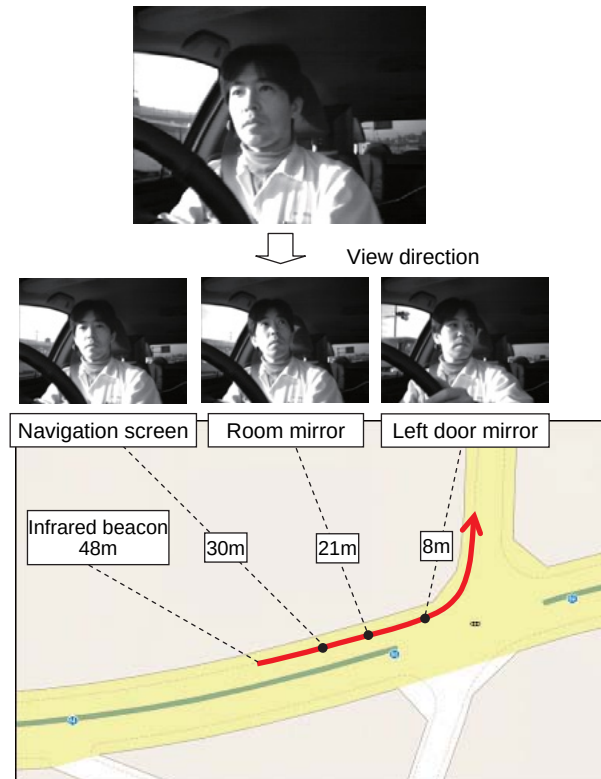
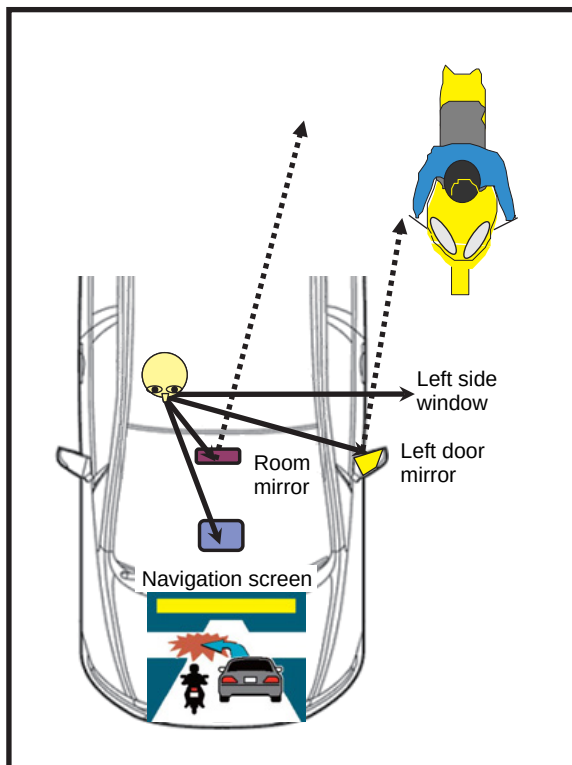


Fig. 17 Analysis of driver's view direction

turn and switching over to involvement avoidance operation. The standard distance for starting avoidance operation was determined to be -18 m, and the standard velocity to be 37 km/h. In this experiment, measured velocities were compared to the standard velocity at the standard distance for starting point of avoidance operation, in order to determine whether involvement avoidance operation is performed in time.

Actions taken to confirm safety with regard to a motorcycle approaching from behind can be determined from the direction in which the driver is looking. For example, when the driver looks in the direction of the room mirror or the left door mirror, this would mean that the driver is exercising caution with regard to a

motorcycle approaching from behind, and is closely observing the motorcycle's movement (Fig. 17).

(3) Test results

The test subject results on the test course are as indicated below.

Figure 18(a) shows the OFF point at which the accelerator pedal is released when there is a motorcycle approaching from behind before beginning the left turn. This plots the measured data when the accelerator pedal is released for the two cases in which system caution is and is not provided. The solid line in this graph indicates the standard velocity curve in 0.1-G.

Figure 18(b) shows accumulated components of distance difference between the standard velocity curve

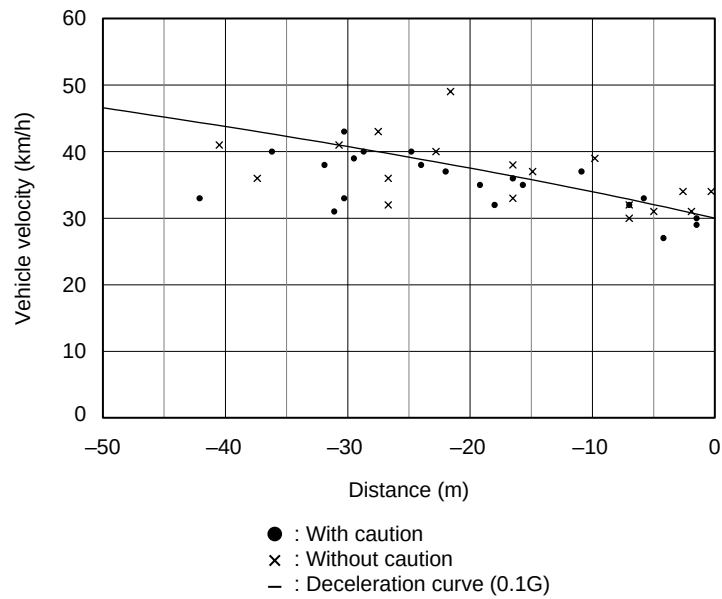


Fig. 18 (a) Accelerator pedal OFF point

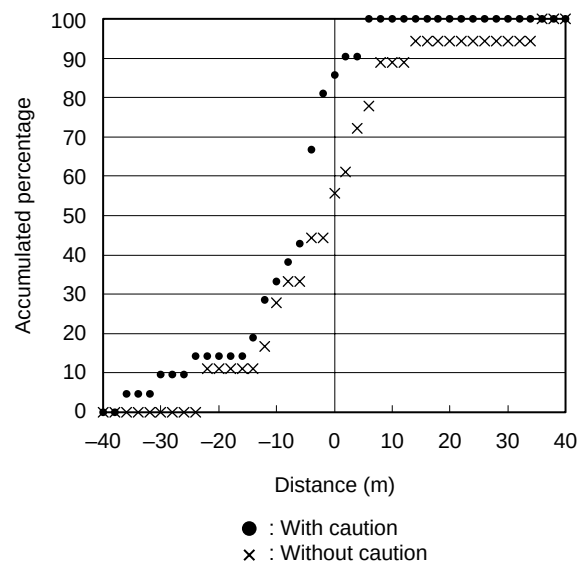


Fig. 18 (b) Accumulated percentage of accelerator pedal OFF point

and the OFF point at which the accelerator pedal is released. The distance on the standard velocity curve was set as a standard (0 m). If the distance difference on the lateral axis is negative, this would indicate that the timing of accelerator pedal OFF is earlier than the standard, and if positive, this would indicate the timing is later.

This graph indicates that the timing for accelerator pedal release when no caution is provided is earlier than the standard timing in less than 60% of cases. When caution is provided, however, accelerator pedal release comes earlier than the standard timing in more than 80% of cases. This shows that the caution provided by the system tends to make deceleration behavior occur earlier.

Figure 19 shows accumulated components of velocity difference from the standard velocity (37 km/h) at the

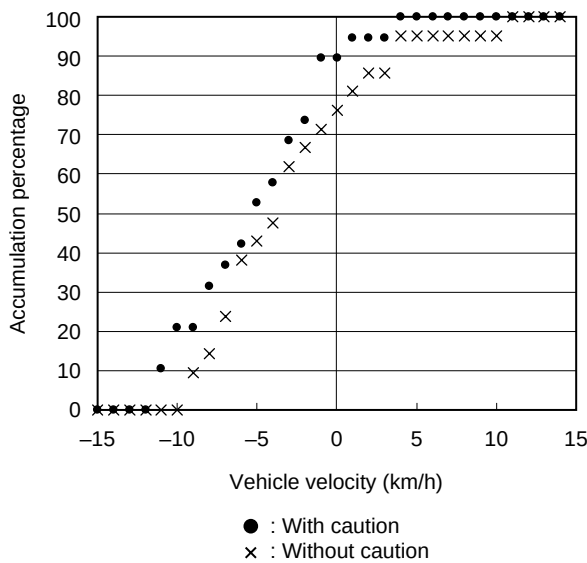


Fig. 19 Distribution of speed

standard distance for starting avoidance operation (18 m before beginning the left turn). If the velocity difference on the lateral axis is negative, this would indicate that the velocity is lower than the standard, and if positive, this would indicate the velocity is higher.

This graph indicates that the velocity when no caution is provided is lower than the standard velocity in less than 80% of cases. When caution is provided, however, the rate of such cases increased to almost 90%. This shows that the vehicle velocity when caution is provided by the system is lower than when not provided.

Figure 20 is a graph showing the distribution of the driver's direction of view before beginning the left turn. It indicates the number of times the driver checks for safety with respect to a motorcycle to the rear by looking at the room mirror or left door mirror. When system caution is not provided, the driver will look next at the room mirror and left door mirror an increasing number of times at a point about 15-25 m before beginning the left turn. When system caution is provided, the driver is looking at the car navigation screen after system caution at a point about 20-50 m before beginning the left turn, checking the caution displayed on the screen. An increasing number of times, the driver will look next at the room mirror and left door mirror at a point about 25-40 m before beginning the left turn.

This shows that the driver starts checking for safety with respect to a motorcycle to the rear by looking at the room mirror or left door mirror at an earlier point, when system caution is provided.

The above results confirm that the left-turn collision avoidance system has the effect of prompting drivers to check for safety with regard to a motorcycle approaching from behind, may cause deceleration behavior to occur earlier, and increases the margin of time available for involvement in avoidance operation.

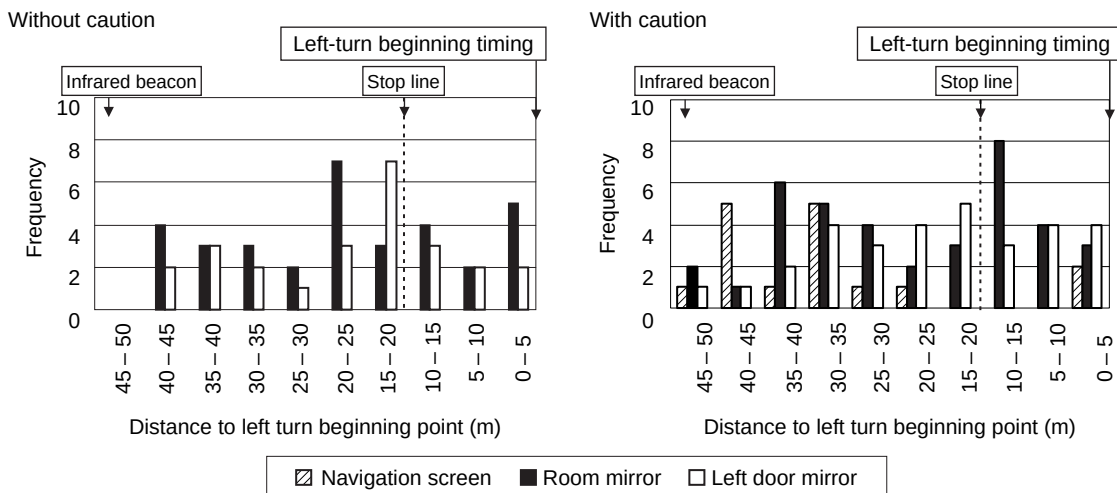


Fig. 20 Distribution of driver's direction of view

5. Conclusions

The results from Honda DSSS field operation tests are as follows.

- (1) Systems for rear-end, left-turn and right-turn collision avoidance were built on the basis of collision analysis results.
- (2) These driving support systems were provided by the coordination of continuous communications through infrared beacons and DSRC.
- (3) Test subject verification was performed under complex traffic conditions of public roads.
- (4) System effectiveness was verified by the verification.

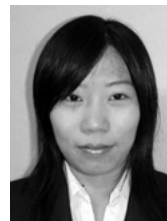
It was determined from the above results that the systems could be expected to be effective in traffic collision reduction.

In this paper, the results from previous verification tests were introduced for the left-turn collision avoidance system. For the future, testing of commuter vehicles will be conducted over the long term, and system effectiveness will be confirmed, including the rear-end and right-turn collision avoidance systems.

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■ Author ■



Yuriko INO



Tomo FUJIMAKI



Nobuyoshi ASANUMA

Introduction of Honda Driving Support System – Field Operation Test for Highway Infrastructure System (Smartway) –

Shugo KONDO* Nobuyoshi ASANUMA*

ABSTRACT

As an initiative of the Smartway project under the Ministry of Land, Infrastructure, Transport and Tourism, field operation tests have been conducted since May 2007 with the participation of private enterprises (nine automobile manufacturers and 17 electronics manufacturers) and other organizations. An advanced highway safety system was created that uses images and sound to enable safe, comfortable driving. Employing new on-board equipment for communications linking road and vehicles, the system was tested on the Metropolitan Expressway in Tokyo. Honda has been participating in the Smartway project with a view to practical application of information provision systems, as well as to verify the functionality of such systems and of the validity and acceptability of driver support.

A symposium was also held in October 2007 in order to generate widespread publicity inside and outside Japan for advanced ITS services on the Metropolitan Expressway. This Smartway 2007 Demonstration, performed as a joint public and private sector (with 30 private enterprises) program, provided people the opportunity to experience Smartway through test drives, presented exhibition displays, and the like. In questionnaires administered after the test drive, respondents gave favorable ratings for service effectiveness, indicating that Smartway holds promise for collision reduction impact.

1. Introduction

The Road Bureau of the Ministry of Land, Infrastructure, Transport and Tourism has been promoting Smartway, which integrates advanced ITS technology. In November 2004, the Smartway Project Advisory Committee established working groups. The National Institute for Land and Infrastructure Management (NILIM) and 23 private enterprises collaborated in summarizing the results of “Joint Research on Systems for Providing Advanced Road Services” in March 2006. The standards and specifications for roadside systems and on-board equipment were decided in March 2007.

The services that are to be realized in Smartway are the timely provision of Smart cruise information, the provision of regional guidance suited to location and needs, and smooth passage through all gates. These three services are to be realized by roadside systems and ITS on-board equipment. Field operation tests were conducted on the following driver safety support services: Providing information on obstacles ahead, merging assistance,

providing information on conditions ahead, information provision coordinated with digital road map data, and position information provision. These five support services were realized, and operation tests geared to full-scale operation have been underway since the end of 2007. Large-scale operation tests are planned to take place in Tokyo at the end of 2008. These will take place through the collaboration of the Cabinet Secretariat, the National Police Agency, the Ministry of Internal Affairs and Communications, the Ministry of Economy, Trade and Industry, the Ministry of Land, Infrastructure, Transport and Tourism, and private enterprises.

This paper reports the field operation test results obtained in Smartway and the details of Smartway 2007 demonstration.

2. Background

Japan has promoted various initiatives to develop Intelligent Transport Systems (ITS) and bring them to practical application. Such systems have spread rapidly,

* Automobile R&D Center

so that the number of VICS units has passed 18 million, while the number of vehicles equipped with ETC on-board equipment has surpassed 16 million.

In Fig. 1, the horizontal axis shows years, while the vertical axis shows shipment volume of navigation systems, VICS, and ETC units⁽¹⁾. The societal impact is starting to emerge, for example in the mitigation of congestion, the alleviation of environmental burdens, and the like. ITS is entering its second stage, in which it is expected to contribute to resolving societal issues and reforming society and people's lives.

Under these circumstances, the Smartway Project Advisory Committee submitted "ITS Enter the Second Stage" in August 2004⁽²⁾. This presents an organized summary of measures for the development of ITS as it enters its second stage.

The government announced the "New IT Reform Strategy" in January 2006⁽³⁾. This calls for the practical application of advanced highway safety systems utilizing vehicle-infrastructure cooperation in order to reduce traffic collision fatalities to 5000 or fewer, with the goal of making Japan's road traffic the safest in the world.

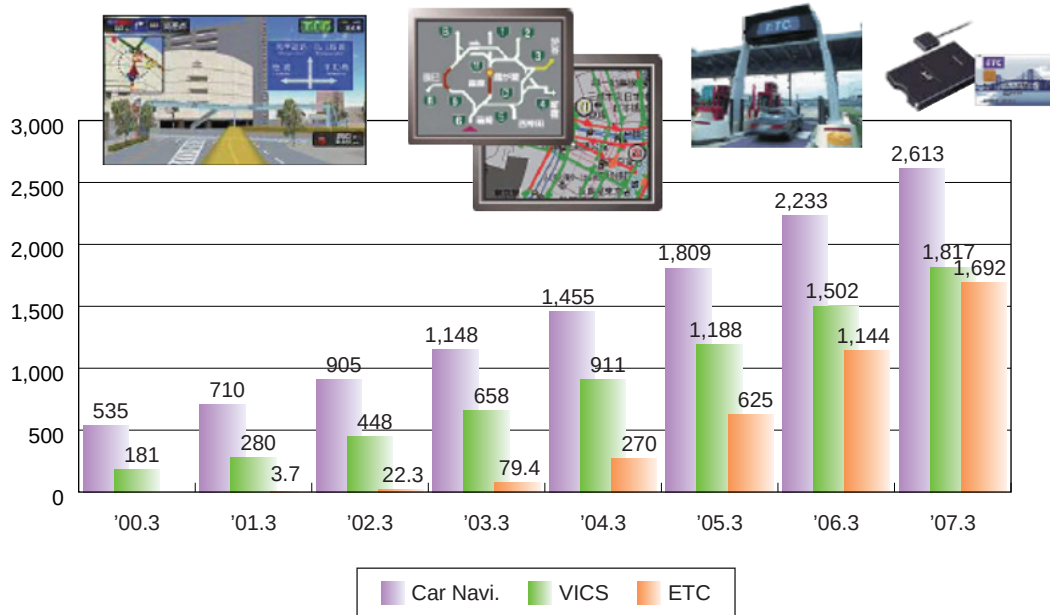


Fig. 1 Cumulative total numbers of navigation systems, VICS, and ETC units shipped

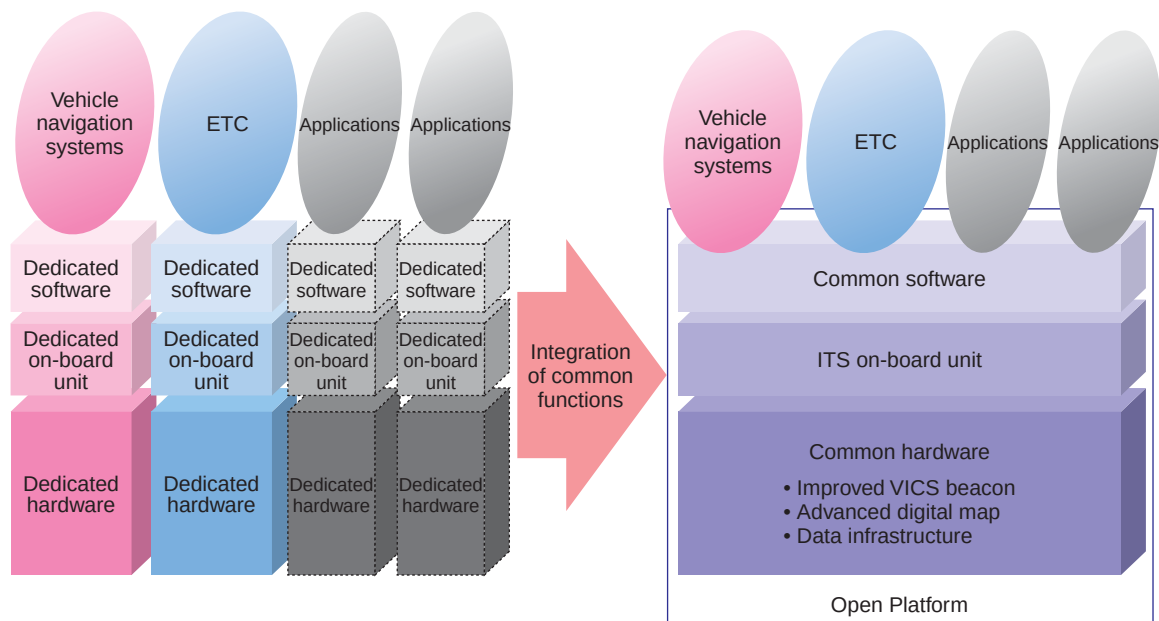


Fig. 2 Concept of platform

Measures to realize this goal have accordingly been promoted by the government and private sector in collaboration. Under such measures, a collaborative council with government and private sector members was formed early in 2006, and conducted an examination of those services and systems envisioned as effective, to include comparative examination of the characteristics of multiple media, as well as the substance of the corresponding operation tests. The result of this examination was the adoption of the concept of an open platform that would share dedicated hardware and software that had been performing their functions independently. Such a platform could mount multiple applications to realize system functionality. Figure 2 illustrates the concept of the platform for next-generation on-board equipment⁽¹⁾.

3. Smartway Technology

3.1. System Configuration

The system for realization of next-generation road services is made up of roadside systems, ITS on-board equipment, and road-vehicle communications. The roadside systems and ITS on-board equipment have a shared functionality among them that is used by different services [a basic application program interface (API)].

Figure 3 shows the structure of the Smartway system⁽⁴⁾.

3.2. Shared Functionality

The common functions for realization of various ITS services include instruction and response functionality, memory access functionality, IC card access functionality, push-type information distribution functionality, ID communication functionality, basic instruction functionality, and shared security functionality. The system has the basic shared functionalities (basic API) that incorporate the same functions.

3.3. Roadside System Configuration

The roadside system is equipped with road-vehicle

communications functionality, basic API, and individual applications. It provides services to the ITS on-board equipment in coordination with external systems. The roadside system is equipped with the functions shown in Fig. 4(1), according to the services to be provided. These are the functions for generating the information to be provided, editing it, and storing it, as well as probe data processing functionality, transaction charge processing functionality, parking area gate operation functionality, and parking charge display functionality⁽⁵⁾. Of these, only the necessary functions are selectively installed. Figure 4(2) shows a photograph of the dedicated short range communication (DSRC) antenna.

3.4. ITS On-board Equipment Configuration

The ITS on-board equipment provides an external interface, basic API, road-vehicle communications, and other such functionality for providing services to the driver.

The road-vehicle communications functionality in the ITS on-board equipment is made up of the DSRC protocol, the extended link control protocol, and the application sub-layer (ASL), which is comprised of non-Internet protocol (non-IP) communications and Internet protocol (IP) communications. This system realizes road-vehicle communications within the roadside system communications zone.

The car navigation function serves as the point of contact between the ITS on-board equipment and the driver. It is equipped with display, Web, and other such functionality. Since it also serves as the point of contact with external equipment, it is further equipped with the capability to acquire and uplink information from car navigation or other such systems.

Figure 5 shows the configuration of the ITS on-board equipment⁽⁵⁾.

3.5. Information Provision Content

The content of services provided in Smartway was as shown below.

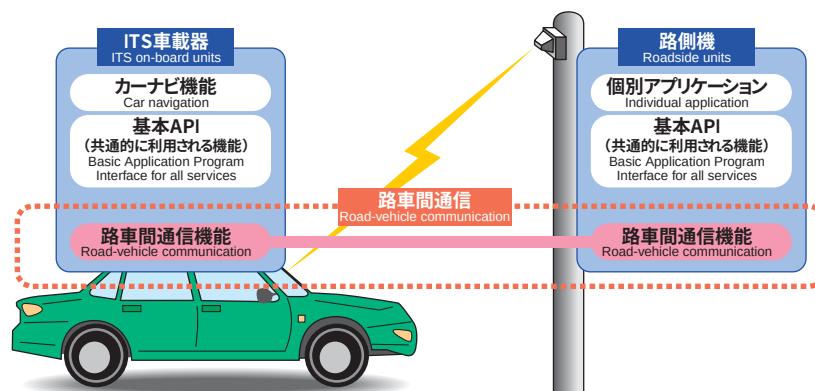
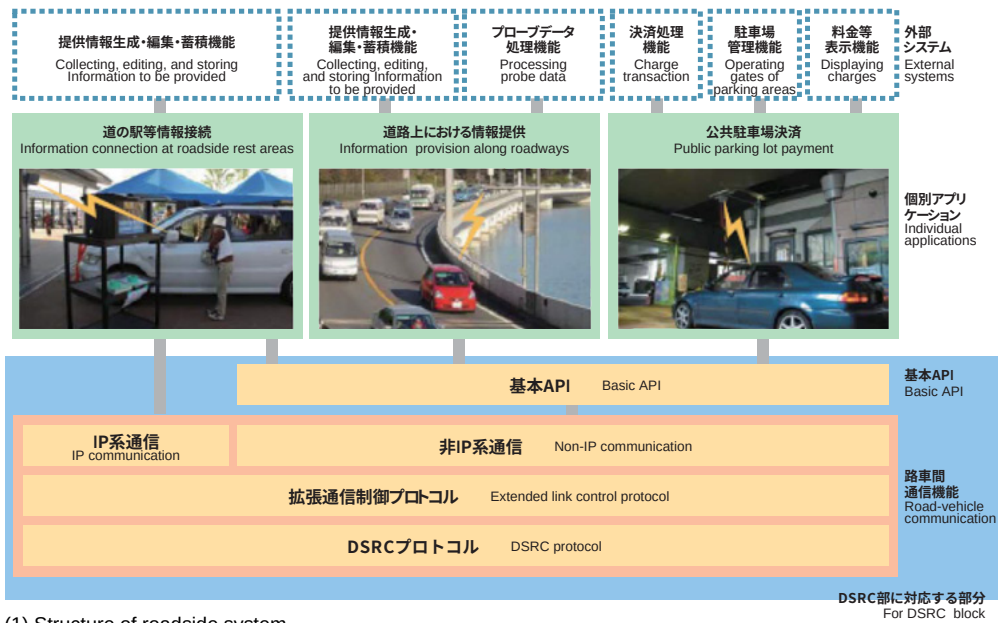


Fig. 3 Structure of Smartway system

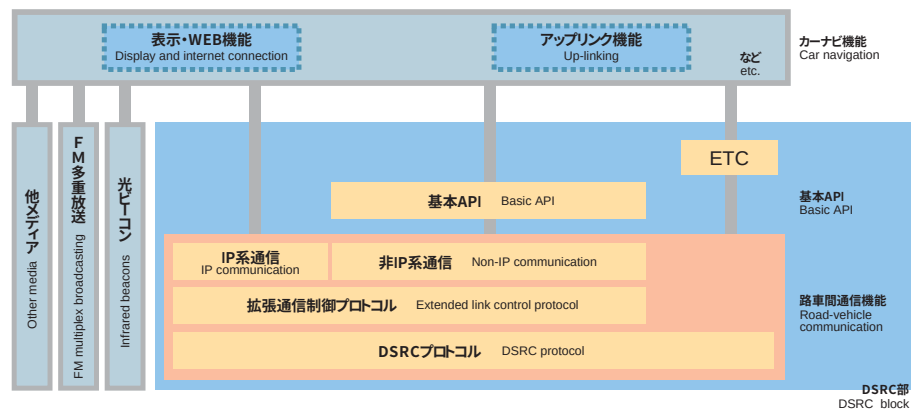


(1) Structure of roadside system



(2) DSRC antenna

Fig. 4 Roadside system



(1) Structure of ITS on-board unit



(2) ITS on-board unit

Fig. 5 ITS on-board system

(a) Providing information on obstacles ahead

This service uses roadside sensors to detect congestion and stationary vehicles ahead on curves with reduced visibility. It alerts drivers who are heading toward such curves by providing them with information by screen display or voice announcements (Caution! Congestion ahead) before they enter the curve.

Figure 6 shows a conceptual image of the service for providing information on obstacles ahead⁽⁶⁾.

(b) Merging assistance

This service uses vehicle detectors to detect the presence of merging vehicles. It alerts drivers by providing them with information about the presence of other moving vehicles by screen display or voice announcements (Caution! Merging from the left ahead) before reaching the merging section.

Figure 7 shows a conceptual image of the merging assistance service⁽⁶⁾.

(c) Providing information on conditions ahead

This service alerts drivers using screen display and



Fig. 6 Providing information on obstacles ahead



Fig. 7 Merging assistance

voice announcements to provide information on road conditions at tunnels and other locations subject to frequent congestion, in order to enhance drivers' sense of security, provide drivers with an extra margin of reassurance, and assist them with route selection.

Figure 8 shows a conceptual image of the service for providing information on conditions ahead⁽⁶⁾.

(d) Providing information on conditions ahead (highway radio)

This service uses voice announcements to provide information about congestion and other such road traffic conditions according to the driving location and direction of movement.

(e) Information provision coordinated with digital road map data

This service uses the digital map database that is built into the car navigation system to alert drivers through



Fig. 8 Providing information on conditions ahead (still image information)



Fig. 9 Providing information with digital road map data

screen display and voice announcements (Caution! Rear-end collisions happen often) about the presence of high-collision curves and other such locations according to the vehicle speed.

Figure 9 shows a conceptual image of the service for providing information with digital road map data⁽⁶⁾.

(f) Providing location information (electronic signs)

This service provides information to support



Fig. 10 Providing information with digital position data



Fig. 11 Internet connection



Fig. 12 Parking lot cashless payment

positional determination by the car navigation system where on-ramps, and the like, overlap with the main route itself on the map, and provides simple road sign information.

Figure 10 shows a conceptual image of the service for positional information provision⁽⁶⁾.

(g) Internet connectivity

This service provides an Internet connection environment for vehicles parked in service areas, parking areas, and other designated areas.

Figure 11 shows a conceptual image of the Internet connectivity service⁽⁷⁾.

(h) Parking lot cashless payment

This service enables vehicles to enter and leave parking areas smoothly and without any need to use cash by charging parking fees to a credit card.

Figure 12 shows a conceptual image of the parking lot cashless payment service⁽⁷⁾.

4. Field Operation Tests

4.1. Overview of Testing

From May 14 to June 8, 2007, field tests were conducted on the Metropolitan Expressway using a total of 60 test vehicles equipped with prototype on-board equipment and manufactured by each participating company. Public road test routes were set on sections of Route 4 Shinjuku Line, Route 5 Ikebukuro Line, and Inner Circular Line. Test subjects were provided with services at 18 points and then asked to respond to questionnaires. Verification of system functionality, which combines the roadside system, vehicle, and communications functions, was performed in the tests, and the service effectiveness and acceptability were verified separately by each participating company.

Field operation test results using Honda test vehicle will be described in section 4.2., and overall test results from participating companies in section 4.3.

Figure 13 shows the test routes and the point at which services were provided during the field operation tests.

- (a) Number of test days: 19 days
- (b) Operating hours: 172.5 hours
- (c) Test subjects: 55 subjects (cumulative)
- (d) Driving distance: 7 000 km
- (e) Offered services: Information provision on obstacles ahead, information provision on conditions ahead, merging assistance, information provision coordinated with digital road map data
- (f) Evaluation data: Data on vehicle behavior during service provision and collected test subject questionnaire data
- (g) Test vehicle: ODYSSEY

Figure 14 shows a scene of information provision during the field operation tests.

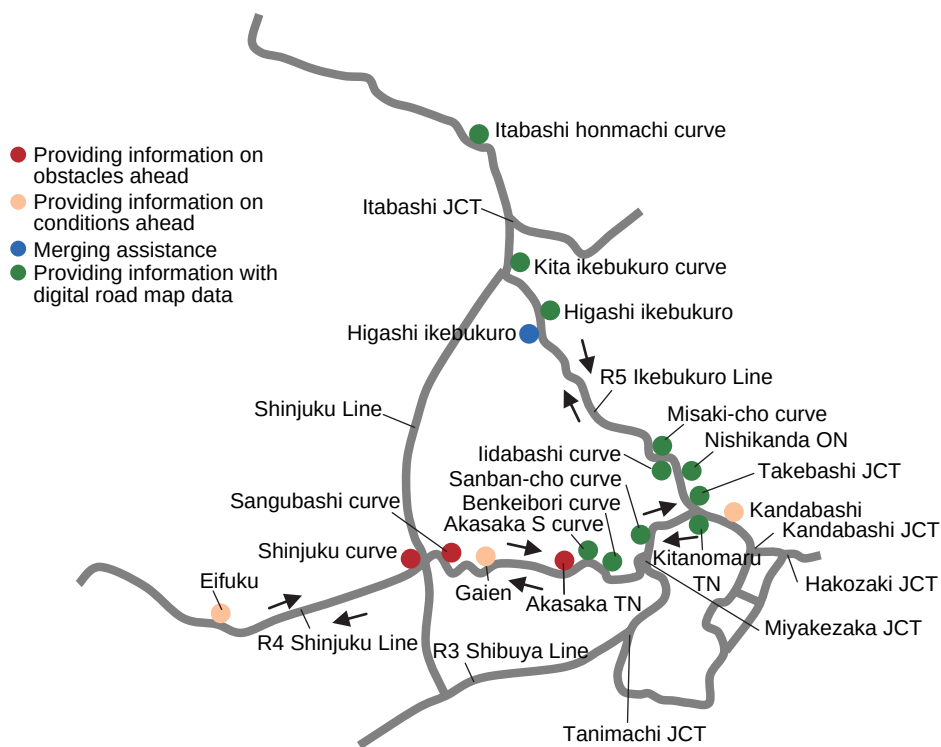


Fig. 13 Service test locations



(1) Providing information with digital road map data



(2) Providing information on obstacles ahead



(3) Providing information on conditions ahead



(4) Merging assistance

Fig. 14 Scenes of information being provided

Table 1 Changes that occur in drivers when information is provided

Changes in Drivers	Description	Findings
Changes in Awareness	• Warning prompted focus of awareness to shift to object, attention directed to safe side.	L
	• Few negative responses such as “startled and confused” or “troublesome”	L
Changes in Action	• Warning did not cause instant action, but line of vision moved in direction of object phenomenon.	L
	• Almost zero negative responses such as suddenly applied brakes	L

Table 2 Selected opinions

Service	Description	Findings
Providing information on obstacles ahead	Information is of low reliability.	L
Providing information on conditions ahead (still image)	Cannot tell what location the information refers to.	L
	Was too occupied with driving to look at screen.	L
	Screen display time too short	L
	Need voice announcement or sound explanation of screen image.	L
	Cannot tell what is intended by screen image.	L
Providing information on conditions ahead (voice announcement)	Too much information provided	M
	Provision timing is too slow.	S
	Cannot tell at all where information refers to.	L
	Manner of speech is difficult to understand because it is unnatural.	L
Information provision coordinated with digital road map data	Provision timing is too slow.	L
	Depending on vehicle speed, provision timing seems off.	S
	Depending on road traffic conditions, provision timing seems off.	L
	Function overlaps with function of commercial car navigation system.	L
General	Difficult to distinguish from existing VICS service.	L
	Warning tones all sound the same so differences are hard to distinguish.	S
	When you get used to it, you stop paying attention.	L
	Coming on top of VICS information, the system information becomes overload.	L
	It should be possible to select the amount of information.	M
Human Machine Interface (HMI)	HMI screen designs are inconsistent.	L
	Understanding the information necessitates time and knowledge.	L
	Voice announcements are too long.	L
	The language is ambiguous so the meaning is unclear.	M
	The road infrastructure side and Smartway services overlap sometimes.	L
	Functions should be narrowed down to just those that can be handled by the vehicle.	L

4.2. Test Results Using Honda Test Vehicle

The questionnaire results with respect to changes in drivers' awareness and action are shown below.

The results show the number of responses by three grades (L: large number; M: medium number; S: small number).

Table 1 indicates changes that occur in the drivers when information is provided.

Large numbers of subjects responded positively to all the items related to changes in awareness and action.

Table 2 shows selected opinions.

Large numbers of subjects responded positively to the majority of questionnaire items. However, a few responded negatively to the items regarding provision timing and quality of information.

4.3. Overall Test Results

Figure 15 shows questionnaire results of the necessity of each service⁽⁸⁾.

The majority of responses regarding the service for providing information on obstacles ahead and that for merging assistance were positive, indicating the services were felt to be necessary. Meanwhile, no clear results were obtained from the questionnaires regarding the necessity of information provision on conditions ahead and information provision coordinated with digital road map data.

Figure 16 shows overall questionnaire results of the effectiveness of the services in Smartway⁽⁸⁾.

Positive response indicating the services were useful or relatively useful amounted to 63% of the total.

4.4. Discussion

The Metropolitan Expressway is a demanding environment in terms of road shape and traffic conditions. With many sharp curves located throughout it, the

expressway frequently experiences congestion in areas with low visibility (after curves, at tunnel entrances) and other conditions likely to result in collisions. The test subjects felt that these services were necessary to deal with the rapidly changing traffic environment. They were hopeful that the services could help reduce the risk of collisions, and thought that the effectiveness of Smartway was promising. The test results show that provision of services directs driver attention to the objects of those services, and give reason to expect services will be effective.

As to acceptability, the affirmative responses received from test subjects vary by whether the content of the service is considered necessary or not. It will be necessary in the future to consider the amount of information provided, the quality of information, the provision timing, and other factors from the driver perspective in order to develop still better driver safety support.

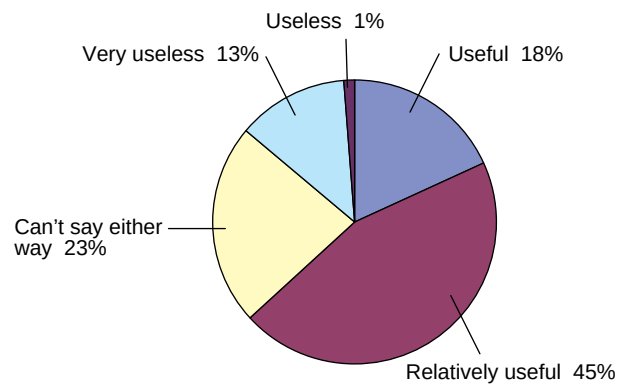


Fig. 16 Overall assessment of effectiveness

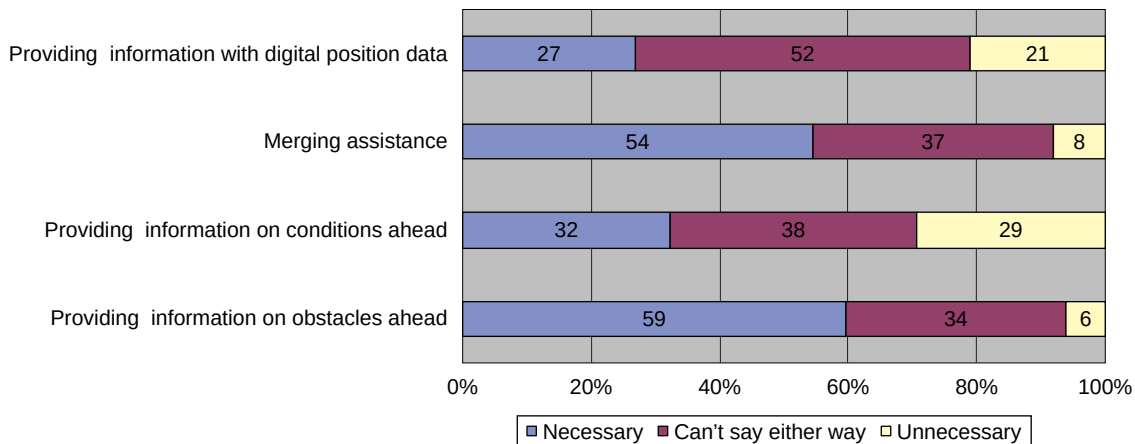


Fig. 15 Assessment of necessity of each services

5. Smartway 2007 Demonstration

5.1. Overview of Demonstration

The prototype ITS on-board equipment systems, developed by the various participating companies, were installed into demonstration vehicles. Visitors were offered test drives to experience ITS services on the Metropolitan Expressway, as well as in the Kajibashi Parking Area, where a roadside system was set up. The event also included symposiums and an exhibition.

A total of approximately 1 650 people attended the event. These included 50 people from other countries, and people from 35 media news organizations. Test drives were experienced by approximately 670 of the visitors.

(a) Dates: October 14–17, 2007

(b) Venue: Tokyo International Forum, Metropolitan Expressway (Route 4 Shinjuku Line, Route 5 Ikebukuro Line, Inner Circular Route)

(c) Content: Opening ceremony, inauguration of test rides, symposiums, keynote address, panel discussions, test drives (scenes of the test drives are shown in Fig. 17 and the test drive course is shown in Fig. 13), exhibitions, vehicle displays, and the like.

(d) Participants: ITS-concerned parties from Japan and other countries, Japanese government ministries and agencies concerned, expressway companies, experts and specialists, related organizations, media news organizations, Smartway 2007 demonstration participating companies, and other such related organizations.

5.2. Test Drive Content

The roadside system used for the test drive employed equipment that had been licensed as utility stations. The on-board equipment consisted of prototypes for use in Smartway 2007. The on-board equipment included both ITS on-board equipment for use with car navigation systems, which provide information by screen image and voice announcement in coordination with a car navigation system, and stand-alone ITS on-board equipment, envisioned for use in heavy duty vehicles, light motor vehicles, and other vehicles not equipped with car navigation systems, that provides information only by voice announcement. Test drives were performed. Eight types of services, which are to be provided in Smartway, were evaluated.

5.3. Evaluation of Test Rides

A brief questionnaire survey was conducted after the test drive and 511 responses were received (approximately



Fig. 17 Scene of test drive experience

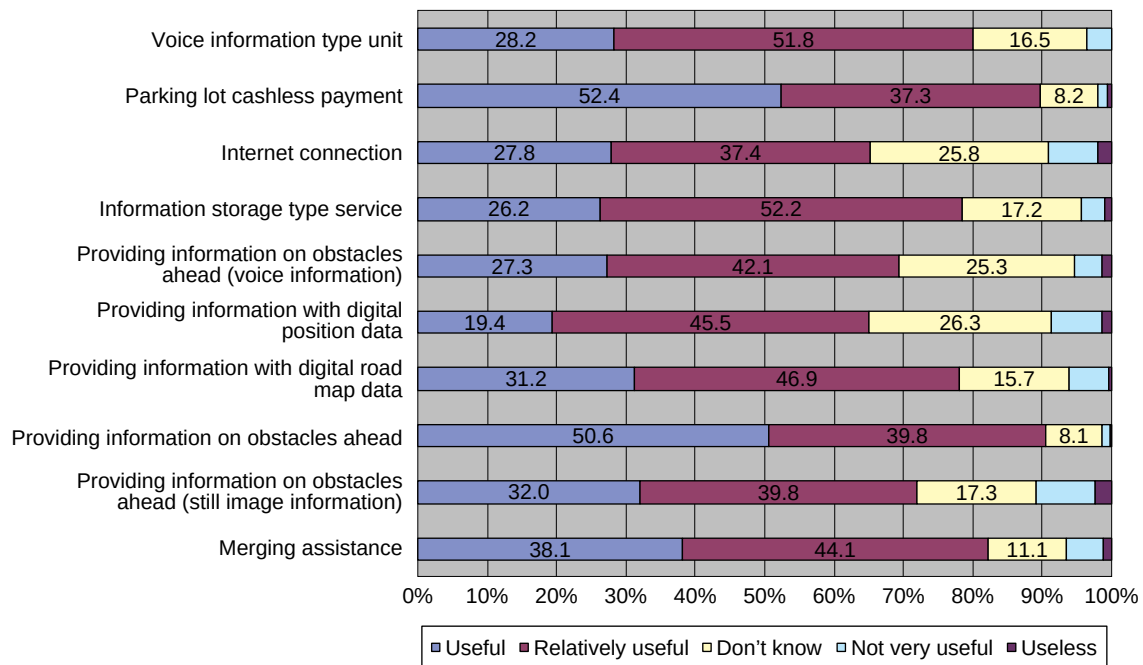


Fig. 18 Partial assessment of experienced service

83% of visitors who took test rides) over a four-day period. Responses that the subjects would like to buy or trade up their unit, indicating a willingness to purchase, were received from approximately 60% of respondents. Evaluations of merging assistance, providing information on obstacles ahead, and parking lot cashless payment were particularly positive (Fig. 18). Responses indicating a positive evaluation of the experienced services, finding them either useful or relatively useful, were received from approximately 90% of the respondents (Fig. 19).

The main reasons cited for finding the information provision service attractive were the use of voice announcements and the provision of information only when circumstances make it necessary⁽⁹⁾.

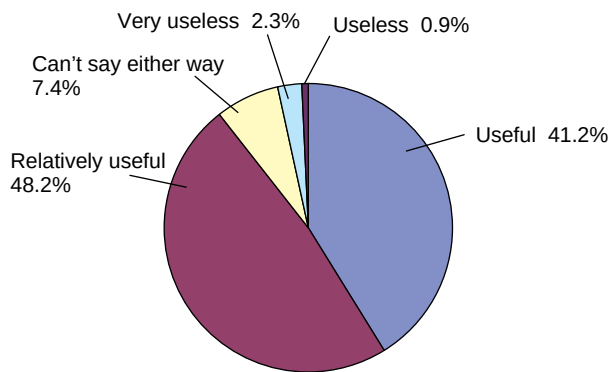


Fig. 19 Overall assessment of experienced service (domestic participant including Honda, excluding foreign invited guest)

6. Conclusions

Field operation tests conducted on public roads have successfully verified the functionality of Smartway project systems. The resulting findings were reflected in the Smartway 2007 Demonstration, where large numbers of visitors experienced test drives. They felt the promise of these services and the resulting efficiency in reducing collisions. The development of safety driving support systems with a high level of value and finish will be of great importance in determining the future configuration of Smartway services. As nationwide Smartway deployment proceeds, we anticipate major advances in the timing and content of information provision, as well as in the level of infrastructure development, including installation location. It will be necessary to continue making incremental enhancements to sustain system development.

Future Smartway development plans call for operation testing that is scheduled to take place in the Aichi, Hanshin, and Hiroshima regions in 2008. Measures have also been initiated to conduct specific studies for prioritization of Smartway deployment at high-collision locations.

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■ Author ■



Shugo KONDO



Nobuyoshi ASANUMA

Development of Chlorine-free Waterborne Primer for Bumper

Takanori HASHIMOTO* Takeshi OGAWA*

ABSTRACT

Changes in the production process of the polyolefin substrate used on automotive bumpers have created the need for increased coating adhesion performance. A method of lowering the melting point of polyolefin, the principal component of the primer, in order to fuse the polyolefin resin with the polypropylene (PP) substrate using the heat generated during pre-heating and baking, thus achieving increased adhesion through the interaction force between molecules, was therefore examined. Using a metallocene catalyst, the tacticity of the polyolefin was controlled and its melting point reduced, resulting in the achievement of a higher level of adhesion intensity than the conventional primer. The chlorine that was formerly added to the olefin is no longer used, reducing the burden on the environment.

1. Introduction

Regulations restricting environmentally harmful substances have been put into effect worldwide against a background of increasing concern with reducing the burden on the environment in order to contribute to the safety and well-being of our societies (Fig. 1). In Japan, revision of the domestic Air Pollution Control Act has put into place regulations restricting emissions of volatile organic compounds (“VOCs” below) from April 1, 2006. The reduction of VOC emissions from automotive coating lines has become an important issue, leading to an increased focus on the development of materials

containing reduced amounts of environmentally harmful substances.

Honda has therefore developed waterborne coatings in the bumper coating process in order to reduce emissions of VOCs. Honda has worked to reduce VOCs, introducing a waterborne primer to its North American bumper coating lines in 1983, and introducing a three-coat one-bake coating system using a waterborne primer, a waterborne base, and a super high solid clearcoat to achieve the world’s highest level of environmental performance to its UK lines in 1998. However, recent changes in the production process of bumper materials have led to the need for the achievement of greater adhesion intensity.

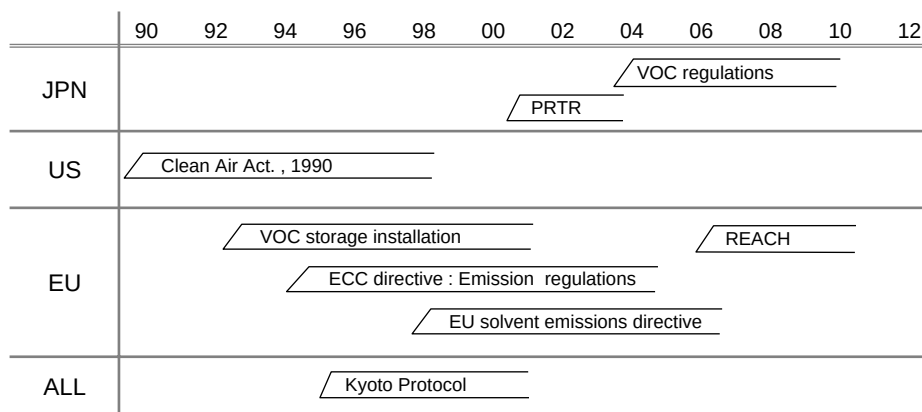


Fig. 1 Global environmental laws and regulations

* Automobile R&D Center

This paper will discuss a waterborne primer technology using catalytic control to reduce the melting point of the olefin resin used in the coating, thus increasing adhesion intensity with the substrate.

2. Status and Issues of Bumper Coating

PP is widely used as a material in the manufacture of bumper fascia in order to achieve weight savings, low costs, and other benefits. However, the surface energy of PP is low at less than 100 erg/cm^2 , and it is chemically inert. These properties make PP a substance that displays low adhesion intensity⁽¹⁾. Figure 2 shows the composition of a bumper coating. The most substantial contribution to adhesion between the coating film and the PP substrate is made by the primer, which is applied directly to the substrate. The solvents in the primer infiltrate the rubber component of the substrate, and promote adhesion by means of an anchor effect⁽²⁾. However, the stipulations concerning reduction of VOCs in recent environmental regulations have resulted in the use of reduced amounts of solvents in the coating films, making it a challenge to achieve an anchor effect by expansion and infiltration of the rubber component of the substrate. In addition, recent modification of processes in order to increase production efficiency⁽³⁾ has also resulted in the addition of a reduced amount of rubber to the PP substrate. Figure 3 shows the

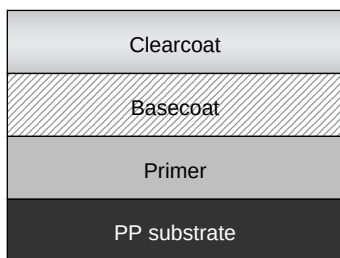


Fig. 2 Bumper coating

change in the structure of a cross-section of the substrate resulting from this reduction in rubber content (the white sections are rubber, and the black are PP). As is immediately clear, there has been a reduction in the amount of rubber close to the surface of the substrate, and this is a factor in the decline in the adhesion of the coating. Given these changes, it has become challenging to balance response to environmental regulations with the maintenance of coating quality.

3. Development of Waterborne Primer

3.1. Concept of Increased Adhesion

The use of an anchor effect or the exploitation of intermolecular interaction are available as means of increasing adhesion with PP substrates. Conventionally, the achievement of an anchor effect by controlling the molecular weight of the olefin, promoting infiltration of solvents, and the like, has been focused on in order to increase adhesion⁽²⁾. Because the mechanical promotion of adhesion using the anchor effect has now become challenging, as discussed above, the project discussed here focused on promoting adhesion via physical adsorption, specifically by means of increasing the intermolecular interaction at the boundary between the coating film and the substrate. Because adhesion is a phenomenon that occurs at the boundaries between materials, the achievement of a sufficient level of intermolecular interaction at the boundary is an important factor in determining whether or not the coating will adhere. Controlling the level of melting during pre-heating, baking, and the like, in order to achieve a sufficient contact area can be considered an important condition for increasing adhesion by exploiting intermolecular interaction. The project therefore focused on the melting point of the polyolefin resin that forms the main component of the primer. The reduction of the melting point of the resin in order to achieve a heat melting effect during pre-heating and baking was considered.

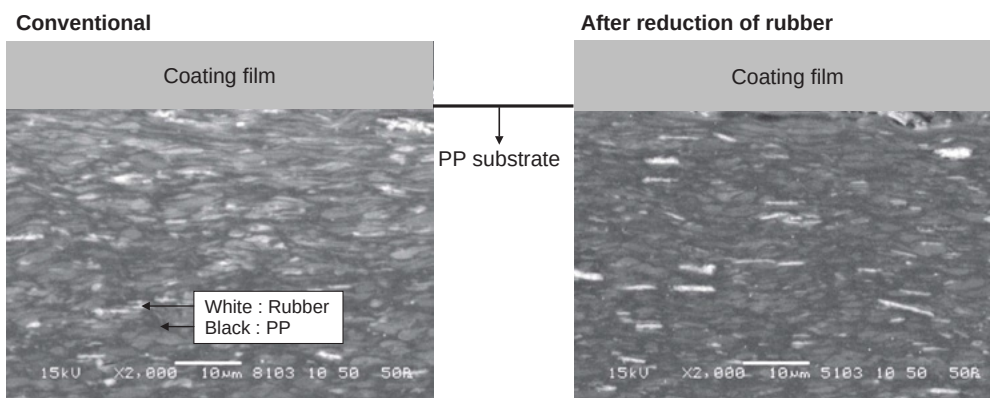


Fig. 3 Cross-section of PP substrate

3.2. Reduction of the Melting Point of Polyolefin

Reducing the crystallinity of polyolefin was considered as a method of reducing its melting point. The addition of chlorine to the olefin resin is a method of achieving this. Chlorine has conventionally been added to reduce the crystallinity of polyolefin in order to help ensure its solubility in the solvents used in the coating and to stabilize the coating. The relationship between the addition of chlorine and the melting point of polyolefin is shown in Fig. 4. As the volume of chlorine added is increased, the melting point becomes lower. Figure 5 shows the relationship between the content of chlorine in the olefin and peeling strength and fuel resistance. Because the peeling strength increases with the addition of higher quantities of chlorine, this relationship shows that a reduction in the melting point contributes to increased peeling strength. However, fuel resistance, which is necessary to the quality of the coating film, tends to decline. The reduction in the melting point and the increase in solubility achieved by the addition of chlorine are in a trade-off relationship, and a lower melting point can be considered to be a factor in reduced fuel resistance. The fact that the addition of chlorine to reduce the melting point results in a decline in fuel resistance as the melting

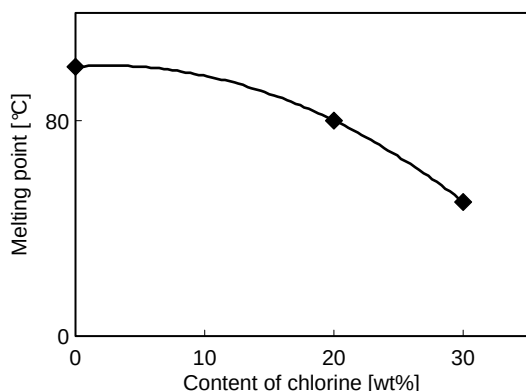


Fig. 4 Influence of chlorine weight % on melting point

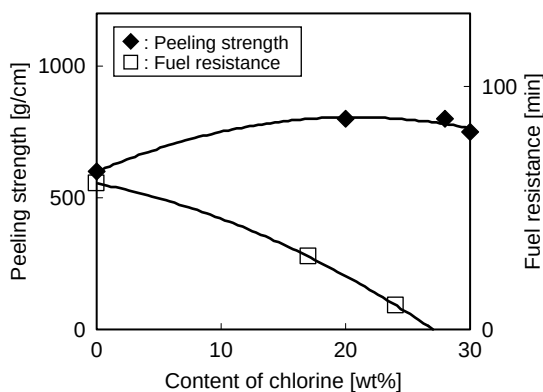


Fig. 5 Influence of chlorine weight % on peeling strength and fuel resistance

point is reduced makes it a challenge to achieve a balance between the two characteristics.

Next, the use of a catalyst technology as a method of controlling crystallinity was studied. The project considered using a metallocene catalyst⁽⁴⁾ enabling free and highly accurate control of the tacticity of PP (Fig. 6) to polymerize the low-melting point polyolefin, while

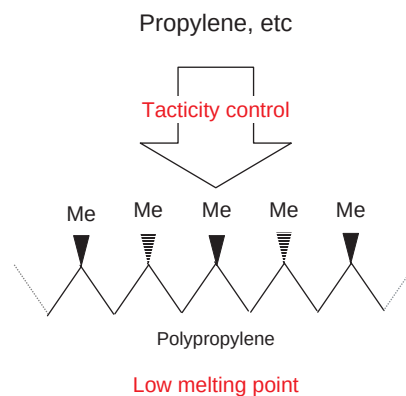


Fig. 6 Tacticity control of polyolefin by developed catalyst

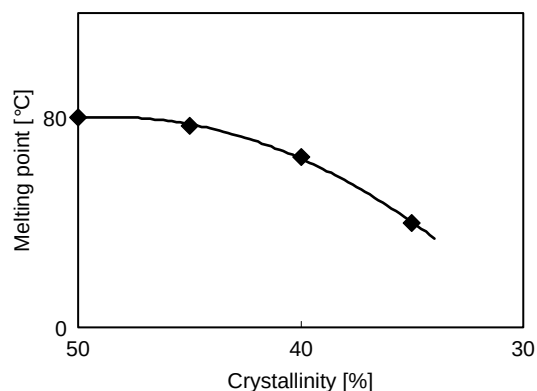


Fig. 7 Influence of crystallinity on melting point

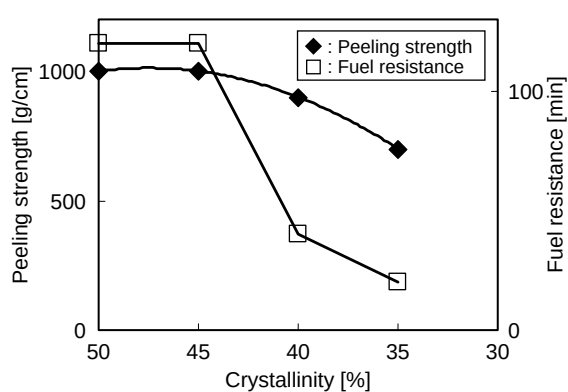


Fig. 8 Influence of crystallinity on peeling strength and fuel resistance

maintaining the poor-solubility characteristic of the olefin. Tacticity is an important primary structural factor which determines the higher-order structures of crystals, such as crystal morphology and crystallinity, that govern the properties of high molecular substances. The mesopentad fraction (mmmm) necessary for the formation of crystals was measured using the ^{13}C -NMR method, and defined as crystallinity. The results shown in Fig. 7 indicate that the reduction of crystallinity using the metallocene catalyst reduces the melting point. As Fig. 8 shows, peeling strength could be increased while maintaining fuel resistance. Because the polyolefin does not contain chlorine, solubility in solvents is low, and the substance displays superior fuel resistance. The low melting point of the olefin fuses the polyolefin resin with the substrate using the heat generated during pre-heating and baking, resulting in efficient adhesion. In addition, the removal of the formerly added chlorine from the olefin settles concerns over the burden on the environment represented by the emission of gases when the coating film is burned at disposal⁽⁵⁾.

3.3. Effect of Developed Primer

Figure 9 shows the melting points of the former and

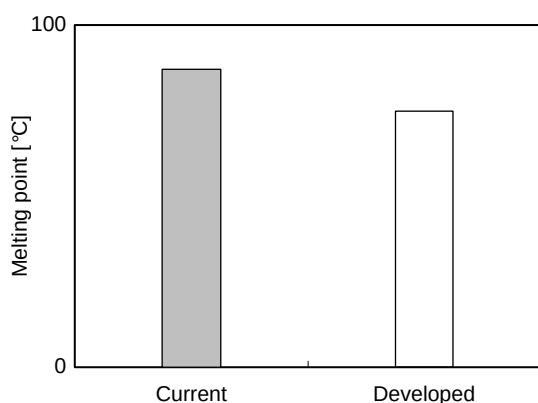


Fig. 9 Melting point of polyolefin

newly developed polyolefins. The results of measurements of the peeling strength of the primer in which the polyolefin is employed are shown in Fig. 10. The newly developed primer has enabled the achievement of a 28% increase in peeling strength against the former product.

4. Conclusion

A metallocene catalyst has been employed to control the tacticity of polyolefin in order to reduce its melting point, enabling the development of a waterborne primer displaying enhanced adhesion.

- (1) The developed primer displays a 28% increase in peeling strength against the former product, in addition to enhanced adhesion.
- (2) The use of a metallocene catalyst has enabled the development of a chlorine-free primer.

In addition to reducing the use of environmentally harmful substances, the outcome of the research discussed in this paper offers outstanding automotive coating performance. The developed product was introduced to Honda's UK coating lines in January 2008, and has been employed in the manufacture of the CIVIC and the CR-V (Fig. 11).

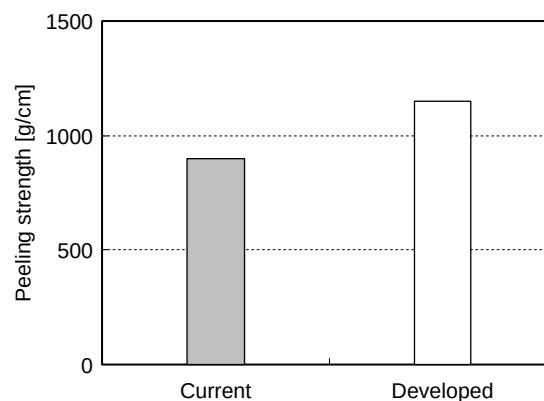


Fig. 10 Peeling strength of primer



Fig. 11 Honda CIVIC and Honda CR-V

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■ Author ■



Takanori HASHIMOTO



Takeshi OGAWA

Honda Grid System for Large-scale Heterogeneous PC-cluster Group

Fumito MAEKAWA*

Yoshihiro YAMAMOTO*

Hidemoto NAKADA**

ABSTRACT

In order to enhance the CAE computation environment, a Honda grid system for a large-scale heterogeneous PC-cluster group was developed. This system is necessary for increasing the operating rate of and effectively utilizing resources (CPUs), the number of which increases rapidly, along with the shift of the CAE computation environment from that of a supercomputer containing a high performance CPU to a PC cluster consisting of a large number of PCs installed on a rack.

By carrying out tuning that matches the environment based on a high quality scheduler while developing unique functions, the operating rate of the CPU was able to be increased. This, together with the effect of reinforcing the resources, enabled the mean computation time, including the waiting time, to be reduced to approximately one-tenth, regardless of the increase in the job number and job scale.

Also, using the same method, a CAD desktop PC could be effectively utilized, and the CAE computation environment on the PC was also enhanced.

1. Introduction

Up until several years ago, super computers that use a few powerful CPUs to perform high-speed calculations were predominately used to carry out CAE calculations. However, such systems are expensive and demand a large investment in order to expand the system scale.

In contrast, given that PC performance increases constantly over time, investment costs can be kept down, while securing performance approaching that of a super computer by introducing a PC cluster formed by clustering several PCs together.

However, to maintain the same level of performance, a PC cluster with 64 CPUs versus eight super computer CPUs is needed, resulting in an increase in system scale, from several dozen CPUs originally to several hundred CPUs.

In addition, as a result of heightened expectations regarding increased accuracy in CAE analyses and shorter development cycles, the amount of model data increases.

Further additions to the PC cluster in order to compute this data in an equivalent amount of time increases the number of CPUs needed, approaching 10 000 CPUs (core).

Clearly, system operations will break down in a situation like this, given the job management and operation techniques performed manually up to this point. This leaves no choice but to quickly automate operations and increase efficiency.

Honda has applied grid computing⁽¹⁾ technology as a means of addressing these issues. The system functions so that if there is a calculation request, the calculation is performed using unspecified resources connected somewhere on the in-house LAN and the result is then returned (Fig. 1).

This paper describes the technology and research topics related to the Honda Grid System (Fig. 2), which has succeeded in reducing the man hours associated with operation and management, while achieving the highly efficient operation of several thousand CPUs.

* Automobile R&D Center

** National Institute of Advanced Industrial Science and Technology

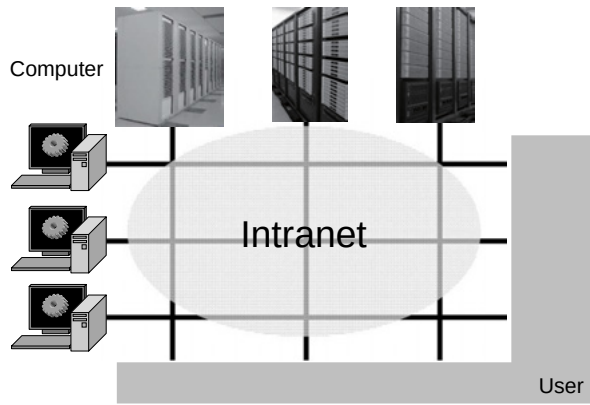


Fig. 1 Honda grid system

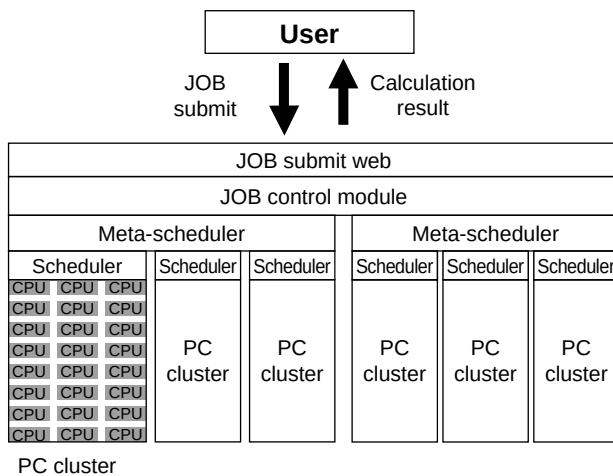


Fig. 2 Outline of Honda grid system

2. System Goals

The goal of developing the Honda Grid System is to take a PC cluster group consisting of PCs from different manufacturers that have different levels of performance, installed at different times, and run that PC cluster group efficiently, to reduce job calculation times without increasing the hours of operation.

The following goals were set for standard calculation of a large-scale analysis performed in-house.

- (1) A CPU availability rate of 85% or more
- (2) A calculation time, including wait times, of 7 hours or less
- (3) Reduction in management and operation man hours
- (4) Effective utilization of PCs for CAD

3. System Configuration and Implemented Functions

3.1. Job Scheduling Function

In building a grid system in a large-scale PC cluster environment, the most important function is job

scheduling. This function distributes jobs submitted to the system to each CPU to be used for calculation based on the status of CPUs being managed and policy of job processing. The software used to fulfill this role is called a “scheduler.”

Since the selection of a scheduler is the key to success even in building the Honda Grid System, the items to be evaluated were chosen and, of these, the following requisite features were established for the Honda grid. The system should:

- (1) Support both Linux and Windows
- (2) Be strong at (or capable of being made strong at) performing parallel calculations
- (3) Have a good performance record and offer stable operations
- (4) Keep costs down even with an increased number of CPUs

As a result of thorough investigation, Condor⁽²⁾ was selected.

3.2. Check Point Restart Function

This function temporarily saves information in the middle of calculations (check point), so that calculations can be restarted later based on this intermediate information (Fig. 3).

Since the number of CPUs and number of jobs are both low in a conventional computing environment, processing can be performed according to a simple first-in, first-out method where only the number of tasks being executed were limited for each user.

However, as the number of jobs increases along with the increase in CPUs, waste begins to arise in both job processing and CPU utilization conditions when using simple processing. To help eliminate this waste and achieve efficient job processing and CPU utilization, a Check Point Restart function was introduced. If a CPU is available, the restriction on the number of jobs is canceled and jobs that are accumulating are processed. If a high-priority job is submitted to the system after this, a CPU can be made available by taking the check point of an executing job and temporarily suspending it. The high-priority job is then processed. When other calculations are completed and a CPU becomes available, the suspended job is restarted from its check point.

The challenging task involves determining the job to temporarily suspend. Suspending a job once processing has started may give rise to feelings of distrust among users. Honda therefore considered the issue of execution priority to determine the order jobs are processed from a reverse perspective, in which an evacuation priority is determined for temporarily suspending jobs. By increasing the evacuation priority of jobs with later processing start times (lower processing priority), jobs that start processing later can be temporarily suspended as necessary.

A processing algorithm has been realized based on this thinking and, as a result of PR aimed at users, Honda was able to introduce the algorithm as a function that did not

give rise to distrust. In addition, because the implementation of this system differs for each application, a processing method suitable for each application was established.

By implementing this function, overall job processing performance was increased by 16%.

3.3. Meta-scheduler Function

Because Honda has used multiple calculation servers (PC clusters) at the individual department and application level up to this point, there was a tendency for server loads to fluctuate in accordance with the task load. Although differences in the load among servers would

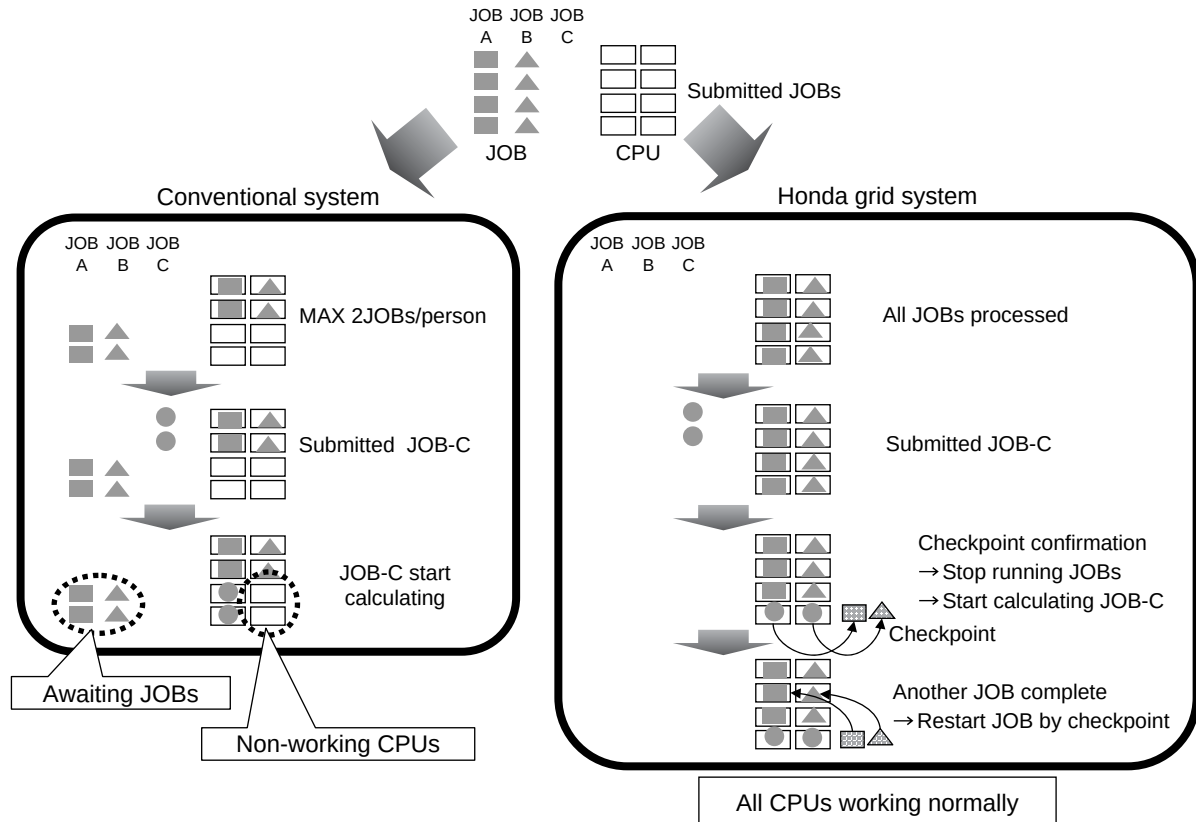


Fig. 3 Checkpoint and restart

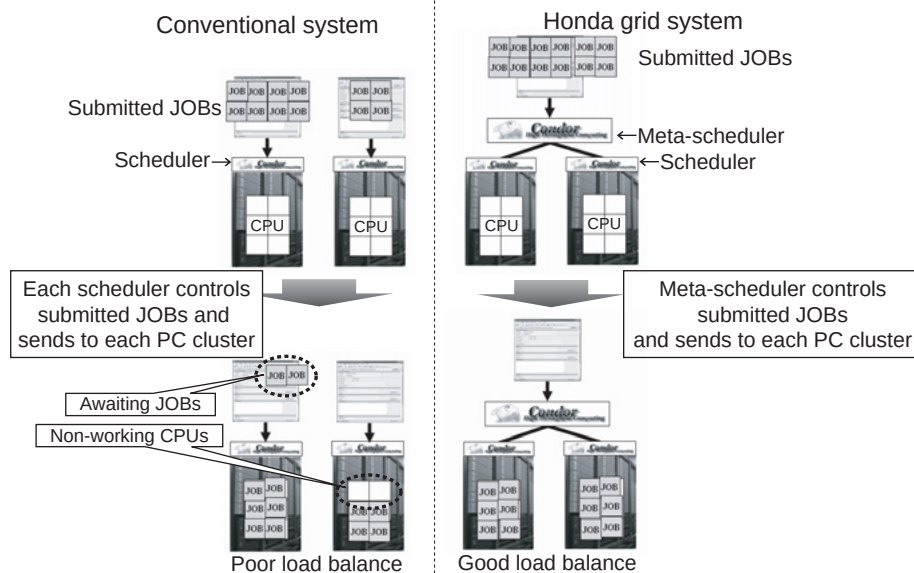


Fig. 4 Meta-scheduling

therefore arise, resulting in inefficient operations, it was almost infeasible to manually perform the task of assigning jobs in order to settle these differences.

A meta-scheduler function that manages multiple servers at once and distributes jobs appropriately has been developed in order to reconcile these differences automatically (to achieve load balance) (Fig. 4).

The flow of this function is as follows:

- (1) When a job is submitted, the meta-scheduler (parent) exchanges information with the scheduler (child) for each server under it, and the job is then passed to an appropriate scheduler based on that server's load information.
- (2) The scheduler to which the job is passed distributes the job to available CPUs underneath it and calculations begin.
- (3) When calculations have ended, the child scheduler informs the meta-scheduler that the job has ended. Finally, the job is removed from the pool.

This function allows operators to balance the load among servers and job execution performance of the overall CAE calculation environment is increased as a result.

3.4. Simple Job Submission Function

Since the method of submitting jobs to the system and the Message Passing Interface (MPI) software used for parallel calculations differ for each application being used, creating a job submission script is complicated, and mistakes in scripts arise due to users creating the scripts themselves, resulting in job issues.

By putting job submission that calls for these man hours on the Web, mistakes in scripts are minimized by automatically creating job submission scripts, thus reducing both the user man hours for submitting jobs and job calculation issues.

Furthermore, Honda determined a standard policy

regarding the order in which jobs submitted to the system are processed and developed a program for assigning priority in line with this policy. This allowed Honda to provide equal service to all users by counteracting unequal conditions among users regarding job processing.

3.5. Automatic System Monitoring Function

In the Honda grid system, where massive parallel calculations are processed in a large-scale environment, the early detection of issues is the most important factor related to stable operation. However, once a system reaches a scale of this size, monitoring by human operators as in the past is no longer feasible. New means of monitoring therefore need to be constructed, even for early recovery, when an issue is detected.

The following measures have therefore been taken with this system.

- (1) A 24-hour monitoring system is configured by launching a tool for constant automatic monitoring of the grid system. Under this system, an e-mail is sent to an administrator if an issue is detected.
- (2) Issues can be investigated and handled smoothly when they occur because grid system monitoring tasks have been split up and divided between the overall system department and grid-related departments.
- (3) Hotlines have been established with hardware vendors to achieve early resolution of hardware-related issues. With specialists in place on the vendor's side, issues can be investigated and handled smoothly.

These measures serve to reduce the man hours and time needed to recover from issues.

3.6. Benefits of the Honda Grid System

Figure 5 shows the benefits of this system configuration.

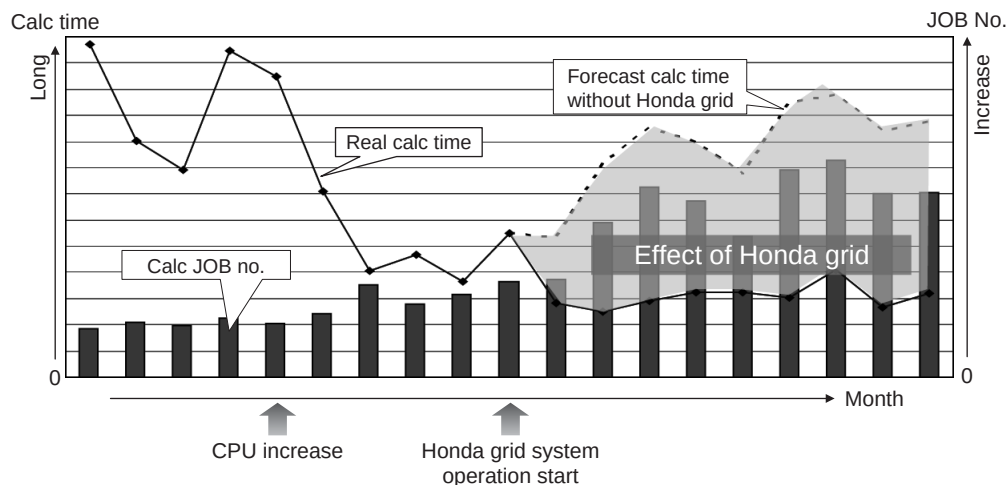


Fig. 5 Result of Honda grid system

4. Grid Application Development on the Desktop PC Group

Although many various CAE applications are being used under Windows today, prompt, responsive performance is reduced to the point that other tasks cannot be executed during calculations. However, even though one frequently sees cases of PCs going unused due to a supervisor being out of his or her seat for a meeting or for some other reason, it is not easy to assign calculations to these PCs. In addition, although one might like to perform calculations continuously throughout the night when no one is using the system, the environment is not set up to do so.

To meet these demands, a grid application using Condor, mentioned previously, has been developed for use on a desktop PC group (Fig. 6).

The issue at this point is what criteria to use in determining that a PC is in idle status. In other words, how should the system determine whether someone is merely out of their seat for a short time or that a PC is not being used for an extended period due to a meeting or other such reason?

The system has been designed so that machines

- (1) whose mouse and/or keyboard has not been touched for a set period of time, and
- (2) whose CPU has not been running for a set period of time, are assumed to be idle PCs to which jobs can be assigned.

The system is therefore designed so that if a machine

being used to process a job is used by a user, calculations are immediately stopped and restarted on another idle machine.

In addition, since users do not use their PCs in the middle of the night, the system has been designed so that the number of PCs needed to process submitted jobs is estimated, and only those PCs needed are run during the night to perform calculations.

These specifications were used to tune an effective and reliable environment for job processing without placing a load on PC users.

Furthermore, reduction in the man hours for submitting jobs to one tenth was achieved by allowing jobs to be submitted over the Web.

5. Conclusion

Having developed several functions for effectively running a large-scale PC cluster environment, Honda has built the Honda Grid System. The benefits of this system are listed below.

- (1) The CPU availability ratio has been increased from 50% to 85% and computing performance has been increased 1.7 times.
- (2) Along with this CPU increase, calculation time (including wait times) has been reduced to approximately one-tenth.
- (3) By forming desktop PCs into a grid, Honda has successfully realized a CAE calculation service under a Windows environment.

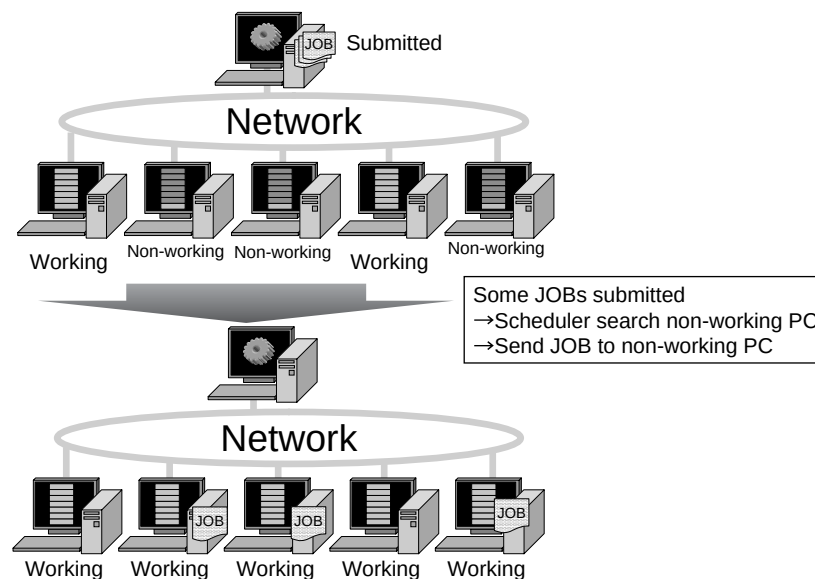


Fig. 6 Honda grid system on CAD PC

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■ Author ■



Fumito MAEKAWA



Yoshihiro YAMAMOTO



Hidemoto NAKADA

Development of Design for Realization of New Value Motorcycle “DN-01”

Kiyoshi KATAGIRI*

ABSTRACT

The new value motorcycle DN-01 is a product resulting from “bottom-up” proposals. The DN-01 represents an entirely new design that was sketched out beginning from the rider rather than the bike, making the rider the main focus, and was created to incorporate a new transmission system and to realize an “easy,” “relaxed,” and “comfortable” model concept. In order to create a design that did not fit into any existing category, the project approached the external design arrangement of the bike from the perspective of a four-wheel vehicle. The DN-01 was displayed at the 2005 Tokyo Motor Show as a show model, after which it went into mass production with the design image unchanged. This resulted in the realization of a long-nosed bike with long and low styling.

1. Introduction

The DN-01 motorcycle is provided with Human Friendly Transmission (HFT), an entirely new transmission system. With the comfort provided by the transmission shock-free HFT system as a central concept, “easy,” “relaxed,” and “comfortable” were established as the keywords for the development of the DN-01. An entirely new and unprecedented design was also needed in order to realize these goals. Drawings for the new bike commenced from images of riders stretching out their arms and legs in a natural posture. Approaching the external design arrangement of the bike from the perspective of a four-wheel vehicle to achieve an enclosed feeling resulted in the creation of a new-value design that did not correspond to any existing category. The new design, incorporating HFT to help prevent transmission shock from resulting in the bumping of helmets when carrying a pillion passenger, and a seat with a lower difference in level than a scooter seat, creates a comfortable space for two, enabling face-to-face conversation between rider and passenger. These elements are packaged using a design with outstanding presence, which catches the attention of even people who are not motorcycle enthusiasts. The packaging of the DN-01 enables its users to spend quality time together, presenting a new lifestyle option not previously available from a

motorcycle.

This paper will describe the development of the design of the DN-01 from the stage of proposal of the concept of a new-value motorcycle, to its presentation as a show model and subsequent mass production.

2. Product Out

In addition to a “market in” process, in which thorough market surveys are conducted to enable the development of a product matched to the requests of target users, motorcycle development often employs a “product out” process, in which Honda creates and offers new value to customers. The development of the BIG-1, the V-Twin Magna, and the Valkyrie Rune, for example, was based on the latter process. The development of the DN-01 was also a “product out” development, and presented the unique challenges for product realization that are associated with this approach.

At the initial stage of development, the concept of the bike stressed the use of existing sports bike layouts as a base to produce an entirely new styling that would be suitable for the employment of the HFT. However, few epoch-making ideas eventuated based on existing layouts, and the project became bogged down. It was the sketch, shown in Fig. 1, which opened a crack as the designers were groping in the dark.

* Motorcycle R&D Center

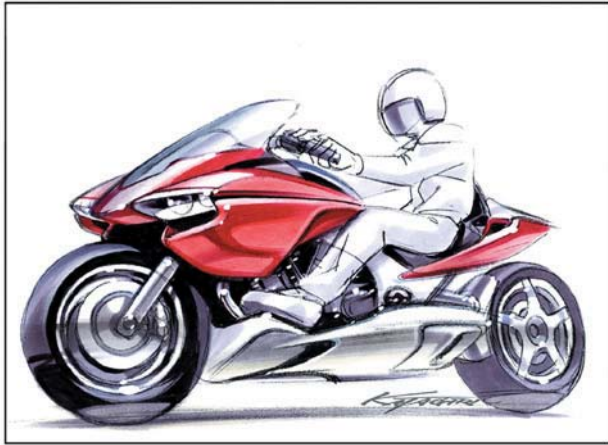


Fig. 1 Idea sketch

3. Sketches beginning from the Rider

In the majority of cases, the external design arrangement of a motorcycle is sketched on the side-on layout diagram in which the wheel base, engine positioning, the positioning of the seat, steering handles, and step, and the like, are determined. However, in order to produce an entirely new design for the DN-01, a method was adopted in which the design process commenced with sketches of riders in postures illustrating the design keywords “easy,” “relaxed,” and “comfortable,” and the external design arrangement of the bike was sketched later.

To express the “easy” concept, a wrap-around-style body cowl that would not interfere with the rider’s arms or legs when naturally stretched out was sketched. Sketches in which the rider’s extended arms were stretched out further were employed to express the “relaxed” concept, resulting in a long-nose design. In order to express the “comfortable” concept, sketches were made in which the cabin was slimmed down to help prevent interference with the rider’s arms and legs when naturally stretched out in front. In addition, the elegant flow of the surface of a long-nosed sports car was used to express the concept of “majestic.” As a result, the front and rear wheels were further separated in order to help prevent interference with the rider’s legs, which were stretched out with the rider seated in a low position, and the wheel base became longer than it had been when the concept was initially proposed (Fig. 1).

4. Design-focused Layout Changes

A low riding position and proud riding posture was also projected as being suitable for the DN-01 engine. Such a riding position and riding posture with long-nosed styling—the proportions—were thought to be appropriate for its realization. However, the dimensions that had been

projected up to that point were a wheel base of around 1450 mm, similar to a sports tourer. Moreover, the distance between the front and rear wheels in the design for the projected rider posture was well over 1600 mm, and turnability also differed entirely from the initial concept. To enable the realization of the design, it was necessary to reexamine the layout from the beginning. Having discussed the issues, the project team’s direction was fixed – the new design would be used even if this necessitated a circuitous process.

5. Employment of Four-wheel Vehicle Elements

While the concept had been accepted, the design was still in the idea sketch stage. With the design judged as lacking in originality, the process of design refinement commenced. The involvement of a large number of veteran designers was sought in order to develop the idea sketch shown in Fig. 1 into a more precise design, and the development of the design took place in a semi-competitive atmosphere.

The design of a bike in an entirely new category commenced with the designers forgetting the bike design know-how that they had attained up to that point. To express the “majestic” concept, this motif was identified with large vehicles with long noses, and the design process commenced attempting without use of bike styling and to seek a styling closer to that of a four-wheel vehicle.

The feel of a long-nosed and short-decked bonnet, a sense of large-diameter wheels obtained using a cantilevered rear fork, indicators integrated with body-mounted mirrors, thin and wide lights with a beltline extending from them, a sit-in riding position, and the like. With the team using four-wheel vehicle terminology not normally associated with bikes during the design review process, entirely new design ideas emerged (Fig. 2).

6. Challenges of “Design Out” Process

However, there were concerns about whether bikes for which the dimensions were determined prior to the design process were more stable and enjoyable to ride, and whether the sketched out design image would still be appealing when presented at full scale. A preliminary model was therefore produced, based on the current Shadow 750, in order to enable the team to get an idea of the feel of the bike.

A 1/10-scale clay model was produced, and a full-scale model of the cowl was produced from the sketches using Fiber Reinforced Plastic (FRP) based on measurements and 3D data, and fitted on the bike (Fig. 3). The riding feel of the bike was judged to be entirely new, different from any other existing model, and the development team expressed a strong desire to move ahead with the development on the basis of the current dimensions.

However, the designers were concerned that the bike had insufficient bulk, and that its imbalance would make the rider seem to have a big presence. This lessened the bold sense of presence shown in Fig. 2 and the sense of integration between rider and vehicle that results from the feeling of being enveloped, as in an automobile, reducing the appeal of this design. However, if the wheel base was extended any further, the bike would not turn easily. The responsibility for achieving the sense of presence illustrated in the design sketch while limiting the size of the bike lay heavily on the proposers of the concept.

7. Accentuating Dynamism

The designers and modelers continued their discussions while studying the prototype. “Low and long” designs are also used in some scooters. The characteristics of this design are a long nose and short deck. It was decided that the team would move in the direction of attempting to create the feeling of being enveloped, characteristic of an open sports car, by further accentuating the sense of a “bonnet” on the bike (Fig. 4). Specifically, the section that interfered with the tank when the steering handle was

turned was not rendered concave, and the bulges in other sections were designed to resemble a vehicle fender. This produced the sense of a continuous single unit extending from the nose of the bike. The sense of continuity of this feature has previously not been attained in a motorcycle. In addition, making this concave section continuous with the meter panel produced a sense of being enveloped that was close to that provided by a vehicle instrument panel. Attention was focused on these features when a rider was seated on the full-scale clay model, enabling the realization of the sense of rider balance that had previously represented a concern. As a result of accentuating the sense of a “bonnet,” the overall design took on a strongly assertive quality (Fig. 5).

8. Objective Opinions

This strong assertiveness certainly matched the goal of creating a design that had never previously been seen, but at the same time, as a commercial product, the appearance of the bike could not seem overly “self-satisfied.” Sketches showing different variations were created based

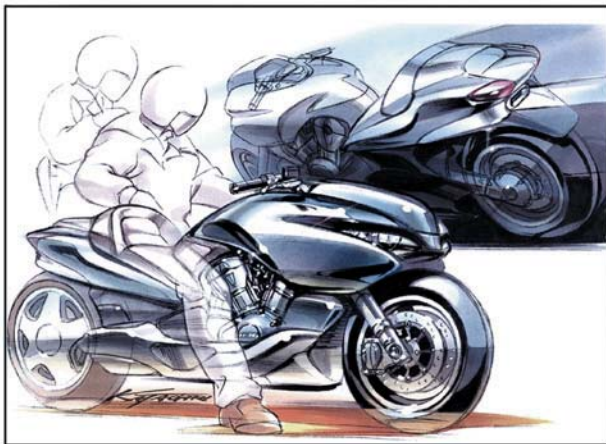


Fig. 2 Key sketch

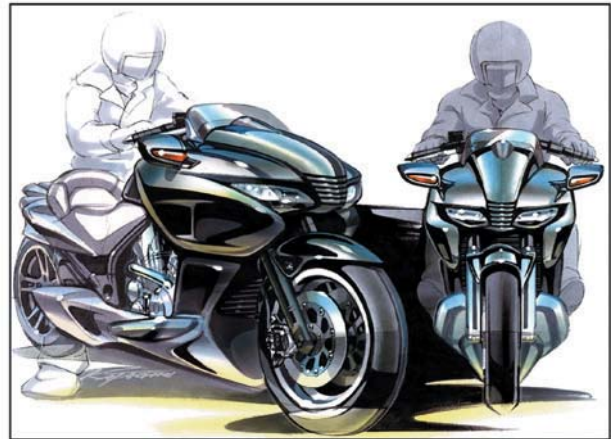


Fig. 4 Second design



Fig. 3 First FRP model



Fig. 5 Second model

on the sketch shown in Fig. 4, and feedback on the different designs was solicited by conducting a broad-based in-house survey to enable the team to review the styling again from an objective viewpoint. As a result, the headlight “grille” was eliminated and the right and left headlights were more closely integrated to provide a more relaxed impression with a sharper image around the front.



Fig. 6 Third design



Fig. 7 Third model



Fig. 8 Forth design

The conclusion was also reached that the bodywork surface quality, which had a strong organic feeling overall, should be given an appropriately sharp impression, and a third clay model was created on this basis (Figs. 6, 7).

This model was displayed at the 2005 Tokyo Motor Show, and on the Press Day, Honda's President Fukui himself unveiled the model and promised the release of the bike in the near future.

9. Development of Mass Production Bike

Thereafter, full-scale development commenced with a view towards mass production, but of course there were still various hurdles that had to be cleared in order to realize mass production. Therefore, the clay model was refined for the fourth time in order to address these mass production issues while retaining the attractive image of the show model (Fig. 8).

First, a variety of issues came to light in the running tests. In the show model, the main seat had been designed low to produce a low riding posture. The team gave consideration to togetherness when a rider and a passenger were using the bike, and lowered the rear seat.

In addition, the instrument panel was distant from the rider's line of sight, and it was judged necessary to increase its readability. The liquid crystal negative display employed in the show model was therefore replaced with a positive, concentrated display, increasing visibility and at the same time realizing a more fitting display for an automatic bike in an entirely new category (Figs. 9, 10).

From a design perspective, the specifications of each



Fig. 9 Display of show model



Fig. 10 Display of mass product model

part were reviewed to achieve the target cost; lamps and other parts were brought into compliance with laws and regulations; and the ABS unit, parking brake and other auxiliary units that had been left off the show model were incorporated into a superb body layout, while finely adjusting the cover shape, to retain as much of the show model image as possible. This involved substantial efforts on the part of both designers and modelers. With all of the team members generating ideas—dividing the fuel tank in two to balance slimness with the necessary fuel capacity, creating a path to guide air from the radiator under the engine to attempt to help ensure that it does not reach the rider’s feet, and the like—a mass production design model was completed while maintaining the design of the show model.

10. Conclusion

Honda pioneered “low and long” proportions in the Fusion. In order to build on that proud record of achievement, the mass production of the DN-01 was to be achieved at all costs. The author had nurtured the dream of a long-nosed bike for more than 10 years, even before joining the Honda team, and that dream was realized precisely because of Honda’s spirit of challenge. The author wishes to express his sincere thanks to all the team members who worked tirelessly to bring the DN-01 to mass production while remaining true to the initial design image, and hopes sincerely, in addition, that the new bike will contribute to offering new lifestyle choices to its users.

■ Author ■



Kiyoshi KATAGIRI

Development of 5-speed Dual-clutch Transmission System for All-Terrain Vehicle

Kinya MIZUNO*

Seiji HAMAOKA*

Eiji KITTAKA*

Masaki KOBAYASHI*

ABSTRACT

A dual-clutch transmission, Honda's first, has been developed for use in utility All-Terrain Vehicles (ATVs). The developed system targeted a comfortable ride and fuel economy while maintaining the off-road running-through performance typical of an ATV, by leveraging the aspect of the transmission efficiency of a dual clutch transmission being equal to that of a manual transmission. This system was developed without changing the transmission axis configuration, the locations at which the clutch and change gear are attached, or other basic elements of the layout, based on the engine being equipped with a manual transmission. The gearshift system was kept simple by adapting a motor-driven gear shift system and simplifying the hydraulic changeover circuit so that the number of operating devices could be held to a minimum. These features also allowed the developed transmission to be mounted on the same frame as that of the manual transmission vehicle.

1. Introduction

ATVs are used in a broad variety of applications, ranging from agriculture and stock farming to trail riding and hunting. ATVs are also driven by a wide variety of people, ranging from experienced users who are used to ATVs to novices who are unaccustomed to gearshift operation. Therefore, there is increasing demand among users for an easy ride, with a trend toward replacing manual transmissions with automatic systems. Meanwhile, the lower fuel consumption of manual transmission vehicles is being reassessed in view of measures to address global environmental issues.

Honda has sought to make the previous kind of manual gear shifting operation easier by developing a motorized gearshift (the Electric Shift Program, or ESP) in which the gear shifting movement is performed by a motor⁽¹⁾, a torque converter type 3-speed automatic transmission⁽²⁾, and a hydraulic-mechanical transmission (HMT)⁽³⁾, and marketing ATVs equipped with these various systems. Other manufacturers are selling ATVs equipped with belt-type CVTs. Table 1 presents a comparison of automatic mechanisms used in previous ATVs.

The issues ordinarily faced with automatic transmission mechanisms include diminished transmission efficiencies because of slippage or flexing resistance in the transmission mechanism, together with the resulting reduction in running through performance, diminished drive feeling accompanied by a sense of slippage, and reduced fuel economy. There was demand, therefore, for development of a new gearshift system that realizes an easy ride while also overcoming these automatic transmission issues.

On the other hand, from the perspective of product development, manual transmission vehicles and conventional automatic transmission vehicles are different greatly in their construction methodologies. Therefore, the parts employ different engineering design principles, and in some cases even the frame has to be reconsidered. Due to these issues, it is desirable for the two types of vehicles to have as many commonized parts amongst them as possible.

Given this context, a dual-clutch transmission system integrated with the engine was developed for use in ATVs. This paper will describe the construction of the developed system and its technological features.

* Motorcycle R&D Center

Table 1 Transmission categories for ATVs

Categories	Typical ATV	Merits	Demerits
Electric shift program (ESP)	Honda TRX420FM	The same basic construction as manual transmission The same off-road ability as manual transmission	Manual shift only Necessity for start clutch
Torque converter with 3AT	Honda TRX680FA	Mechanical control for start Smooth start and shift	Completely different in construction Lower transmission efficiency due to torque converter Necessity for shift device each gear
Hydraulic-mechanical transmission (HMT)	Honda TRX500FA	Smooth driving and low vehicle behaviour due to gear shift	Completely different in construction Lower transmission efficiency due to HFT Necessity for start device Lower capacity for transmission
Belt converter	Other	Smooth driving and low vehicle behaviour due to gear shift Lower in cost	Completely different in construction Unsuitable for ATV's usability Lower transmission efficiency due to belt converter Necessity for start device Lower capacity for transmission

2. Development Goals

The dual-clutch transmission system was developed for the automatic transmission version of the mid-class TRX420 series, with 420-cm³ displacement, which already had a line-up of manual and motorized gearshift vehicles. The developed system had the engine and transmission integrated so as to fit completely within a single casing. This is the same construction as general ATVs or motorcycles.

An automatic transmission for ATVs should provide running through performance equal to that of vehicles with manual transmissions while not giving riders undue shift shock or driving power reduction. It is particularly important to make certain that there is no reduction in the driving power when riding up a steep gradient, leaving the vehicle with insufficient climbing power.

It is characteristic of the dual-clutch transmission that the transmission elements are the same as in a manual transmission, for it to obtain equal transmission efficiency, as well. It is also known that reduction of driving power can be precluded by adjusting the shift from one clutch to the other when shifting gears. A dual-clutch transmission was therefore adopted with the goal of realizing an easy ride, while also providing running through performance and enhanced fuel economy relative to previous ATVs with automatic transmission, including hydraulic-mechanical transmission.

On the other hand, certain issues have been pointed out: When standing start control is used with a dual-clutch, a stable standing start control system is needed. Construction of the dual-clutch unit becomes complicated so that the size of the transmission needs to be reconsidered. The transmission section needs a large number of sensors and gearbox parts⁽⁴⁾. Also, automatic gear shifting may generate undue noise from operation of the gearbox.

The goals of the present development project were

therefore defined as follows:

- (1) The transmission should be mountable on the frame of an existing series vehicle without changing the shaft arrangement or the locations of the clutch and change gear from the manual transmission model.
- (2) The system should be made simple and lightweight.
- (3) Shift shock should be on par with those of previous vehicles using automatic transmissions with torque converters.
- (4) The transmission mechanism should not generate undue mechanical noise during automatic gear shifting.

3. Overview of the Dual-clutch System

An ATV equipped with the developed dual-clutch transmission is shown in Fig. 1, and its engine is shown in Fig. 2. The overall system configuration is shown in Fig. 3. The driving power of the engine is conveyed through the centrifugal-type starting clutch and primary gear to the dual-clutch transmission, and is transmitted through the transmission to the output shaft.

Both of clutches are connected with the gearbox. The first clutch controls the odd gears (first, third, and fifth) while the second clutch controls the even gears (second



Fig. 1 Appearance of TRX420FA

and fourth) and reverse. The gear shifting method is similar to that in the dual-clutch transmission on an automobile. The gear of the non-driven clutch side is prepared in advance to the targeted gear by gearbox. Switching between the driven clutch and the non-driven



Fig. 2 Appearance of engine

clutch completes the gear shifting. The way this gearbox prepares a gear train in advance is a distinctive feature of dual-clutch transmissions and is known as pre-shifting. The preparatory shifting on this system always has the next gear in the gear range for each clutch ready to be engaged. The preparatory shifting control is configured so that the gear range on the non-driven clutch side has readied the next gear up from the gear on the driven clutch side on which the vehicle is presently driving. The reason for this is that upward gear shifts, which mainly occur during acceleration, call for gears to be shifted in

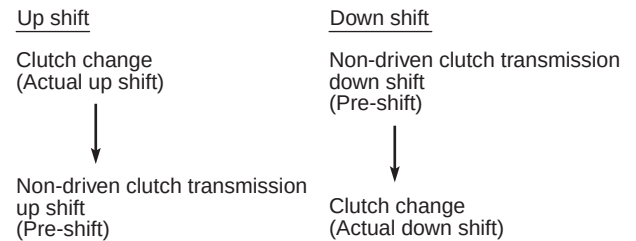


Fig. 4 Gearshift process

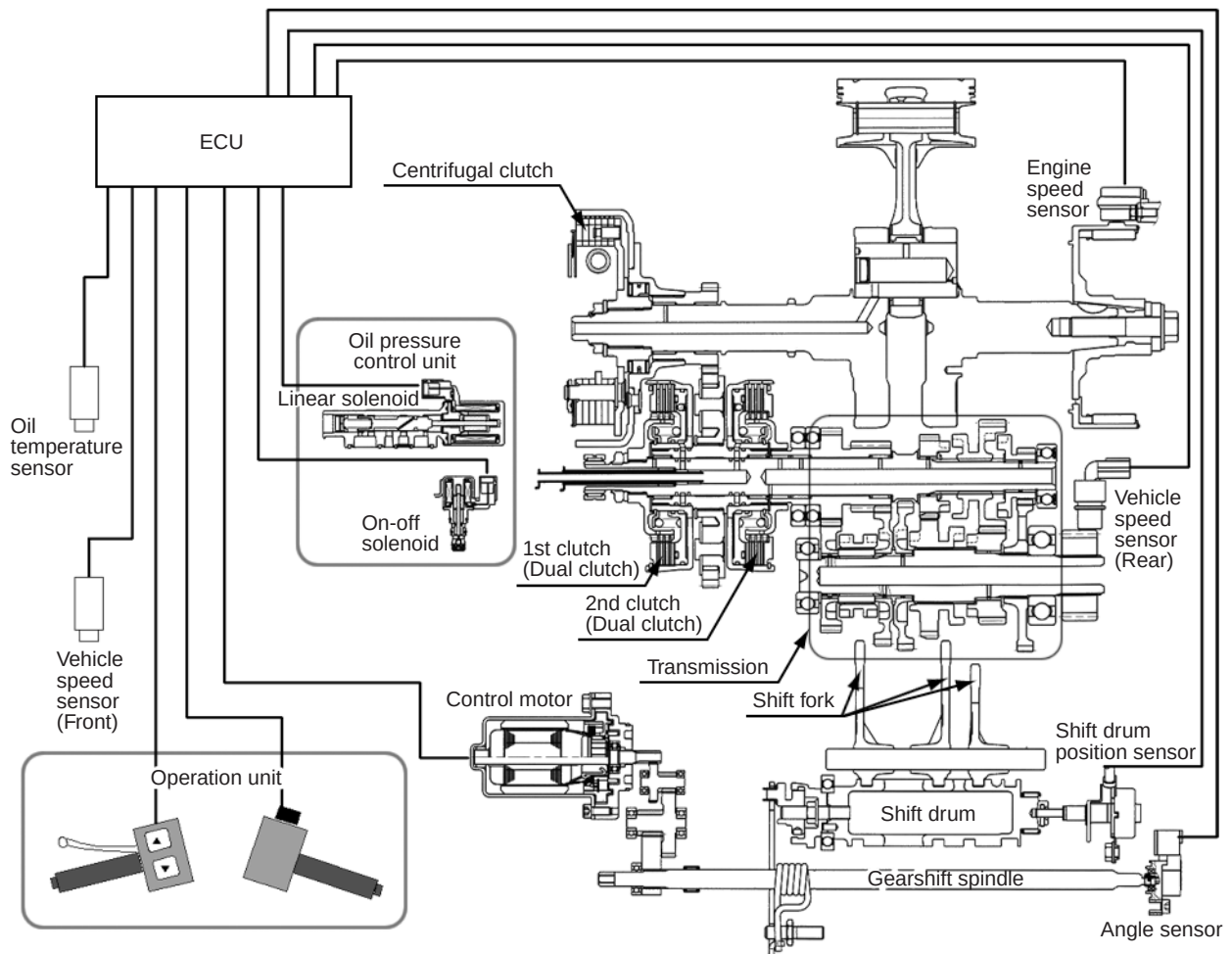


Fig. 3 System configuration

a short time so as not to lose the sensation of acceleration. Therefore, an upshift of gear can be accomplished simply by switching clutches. A downshift is performed by switching clutches after preparatory shifting takes place. Figure 4 shows the gear shifting process when upshifting and downshifting.

Switching between clutches is done by the on-off solenoid that switches hydraulic lines. The linear solenoid adjusts the hydraulic pressure during this process in order to reduce shift shock. Switching between gears in each gearbox is carried out by a motorized gearshift. The electronic control unit (ECU) computes the gear shifting by using the front wheel speed, rear wheel speed, throttle position, engine speed, and oil temperature. It checks these parameters against a transmission map and conducts the clutch switching control and preparatory shifting transmission control, accordingly.

4. Technical Features

4.1. Standing Start Control

Standing start control of vehicles with dual-clutch transmissions generally uses electronic hydraulic pressure control of the clutches to create a half-clutch state for a smoother start. Start control in the system developed here, however, uses the same mechanical control with a centrifugal-type clutch as used for ATVs with manual transmissions. This helps eliminate the need for an electronic control system and allows for a simpler system.

4.2. Construction of the Dual-clutch and Gear Train

Previous dual-clutch systems were known to have structurally complicated clutch units. The clutches used in this system are similar in construction to the single hydraulic clutch used in automobiles, and they are configured symmetrically on either side of the primary gear. Figure 5 shows the configuration of the transmission. Each clutch is connected by a clutch guide to the first and second main shafts, which are located on the same axis and supported in a double shaft structure so that they can rotate freely from each other. The gear engagement elements are of the dog type, generally used in motorcycles and ATVs with manual transmissions.

Adoption of this basic layout permits the dual-clutch attached to the primary gear to be placed in the location where a manual transmission vehicle has a multi-plate clutch attached to the primary gear. The construction of the first main shaft has added elements that are not found on the main shaft of a manual transmission vehicle, such as the clutch lubricating channel and the supports for the dual shaft unit. However, the shaft coordinates, shaft support method, gear coupling splines, and the shifter slide spline are among the elements they commonize. The countershaft differs in the placement of the gear train, but otherwise it is a commonized element.

4.3. Shift Shock Reduction Method

The method used to reduce shift shock is basically the same method using in general passenger automobiles. A linear solenoid is used to adjust the hydraulic pressure of the connecting side clutch to balance the driving force of the engine to the clutch capacity. This reduces the change in vehicle behavior resulting from sudden engagement of the clutch. A spool valve is used to gradually bleed off the hydraulic pressure of the disconnecting side clutch. Meanwhile, ATVs are fitted with low-pressure tires in order to drive off-road, and these tires also constitute a contribution to shift shock reduction. This has enabled reductions in the hydraulic circuit to be implemented, such as in the accumulation valve used to control the pressure regulating accuracy in ordinary passenger automobiles. The hydraulic circuit is shown in Fig. 6.

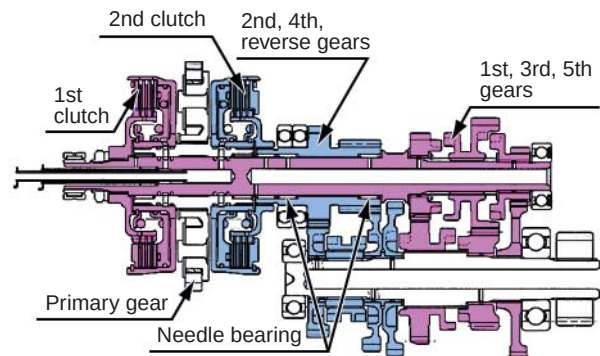


Fig. 5 Design of gears and clutches in transmission

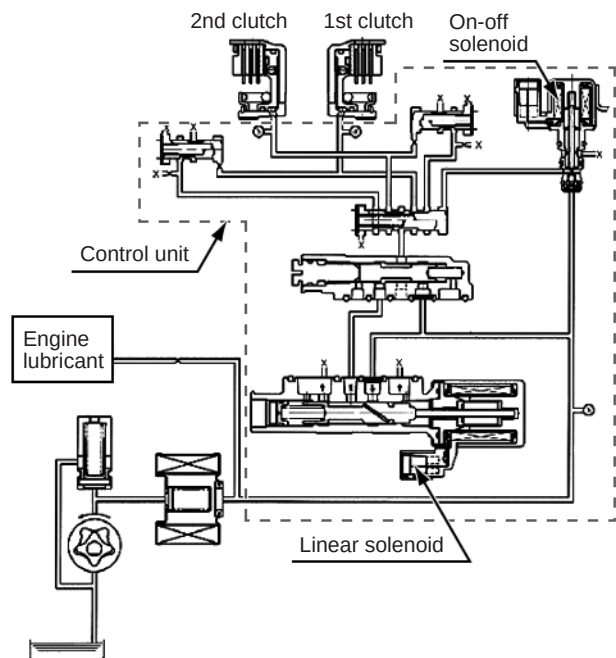


Fig. 6 Hydraulic circuit

The hydraulic control applied using the linear solenoid is divided into three phases. In phases 1 and 2, the system is rapidly placed in a half-clutch state. In phase 3, engine speed and throttle position are used to control the torque capacity of the hydraulic clutch. Figure 7 shows the control chart for the linear solenoid. At the same time that hydraulic control is applied, the engine ignition timing is also retarded slightly. This suppresses engine output so as to hold down the fluctuation in transmission torque when gear shifting takes place. This hydraulic control and ignition timing adjustment provide smooth gear shifting, and yield a shock during gear shifting that is on a par with vehicles that have an automatic transmission with a torque converter.

The valve body unit that performs hydraulic control was designed to hold the linear solenoid, on-off solenoid, and spool valve within a single unit in order to reduce faulty operation due to foreign matter contamination during manufacturing. This unit is attached to the front cover. From the front cover it is connected to the clutch by a feed pipe that supplies lubricant to the main shaft and the clutch. The parts containing high-pressure hydraulic channels were limited to these parts in order to simplify maintenance by limiting the parts subject to foreign matter control. The valve body units were designed to be assembled in a clean room exclusively for that purpose at the manufacturing plant, after which they

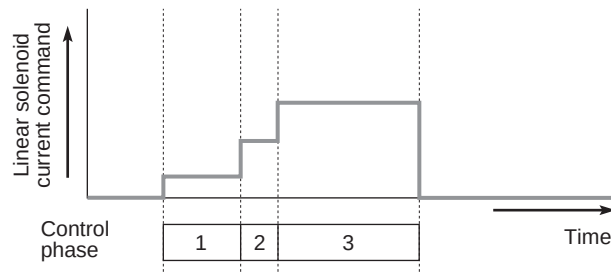


Fig. 7 Linear solenoid control chart

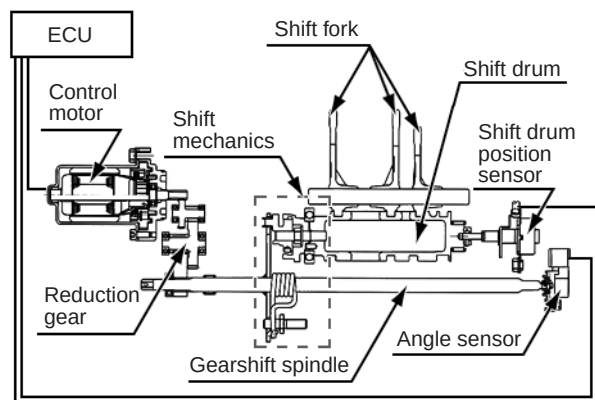


Fig. 8 Transmission system

are installed on the front cover. They are intended to be subject to verification of functionality and quality assurance.

4.4. Preparatory Shifting System and Its Method of Operation

The pre-shift on dual-clutch transmissions for conventional passenger automobiles have from three to four shift forks. These are all controlled in each. Therefore, as many operating devices, position sensors, and failsafe devices to reduce errors in joint operation were needed as there were shift forks. Motorcycles and ATVs, on the other hand, generally utilize sequential drum systems. Therefore, it was decided that the present system would also utilize the sequential drum method. The construction, operating devices, and control systems were accordingly carried over, without alteration, from the TRX420 motorized gearshift system. The transmission system is shown in Fig. 8.

Motorized gearshifts use a control motor to operate a manual transmission. The motor turns a change spindle, which makes the gearbox operate. The spindle position is detected by an angle sensor, while a shift drum position sensor determines when the travel is complete and the motor returns the spindle to a neutral position. A single motor operates the drum and moves three shift forks.

The drum is of the same construction as in an ATV with manual transmission. It differs, however, in that it is configured so that one position engages the respective gear trains of the first clutch and the second clutch. Moreover, the gear ranges in those respective gear trains are arranged so that the gears to be engaged are always adjacent. Table 2 shows the combinations of drum positions and engaged gears.

The preparatory shifting on the non-driven clutch side therefore changes two gear speeds ahead, and the relative difference in rotation when the dogs are engaged will be two gear speeds. This is why the dogs together generate undue noise. In order to reduce this noise when preparatory shifting occurs, a control function was added to bring it to a stop, once at a half-neutral position after disengaging from the previous gear, and before the next gear is engaged. Figure 9 shows the control of the control motor. The travel control applied by the control motor is divided into four phases. Phase 1 travels to a position halfway from the previous position to the next position.

Table 2 Shift drum position and engaged gears

Shift drum position (Sequential)		Reverse	Neutral	1-2	2-3	3-4	4-5	5
Engaged gear	1st clutch	—	—	1st	3rd	3rd	5th	5th
	2nd clutch	Reverse	—	2nd	2nd	4th	4th	—

Phase 2 holds at that position for a certain time. Phase 3 travels to the next position. Phase 4 returns the transmission to the original position. By means of this control, the speed on the non-driven side of the transmission is synchronized with the speed on the driven side without the addition of synchronizers or other such mechanisms. The relative difference in speed can be reduced to one gear speed before the next dog is engage. This reduces the impact energy of the dog engagement and limits the amount of rattling noise. Figure 10 shows the rotation difference when preparatory shifting takes place. This gear shifting operation only adds the control described above, stopping at half-neutral, to the previous motorized gearshift. The previous system was therefore able to be carried over, for the most part, without any change. This system was also made simpler than the preparatory shifting system in automobiles.

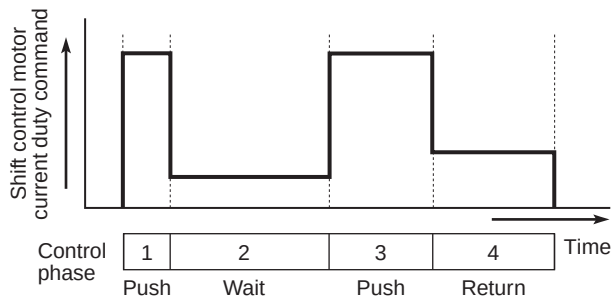


Fig. 9 Control of control motor

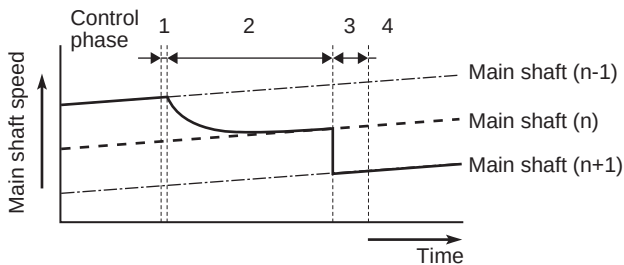


Fig. 10 Rotation difference

5. Shift Control Mode and Its Method of Operation

Selection of the gearshift mode and traveling range are by-wire operations. They are conducted using the upshift and downshift switches on the left handlebar and the gearshift mode switch installed on the right handlebar. The arrangement of the switches is shown in Fig. 11. The gearshift mode switch can be switched between “AUTO” for automatic gearshift mode, and “ESP” to select any gear range desired by the rider. In AUTO mode, the upshift switch can be pressed to choose to shift to R (reverse), N (neutral), or D (drive), in that order. The

downshift switch does the opposite, switching the selection from D to N to R in that order. When shifting from N to R, an interlock needs to be released at the same time, in the same way as in current models. In ESP mode, the upshift switch shifts up and the downshift switch shifts down. The rider can therefore freely select the desired gear range.

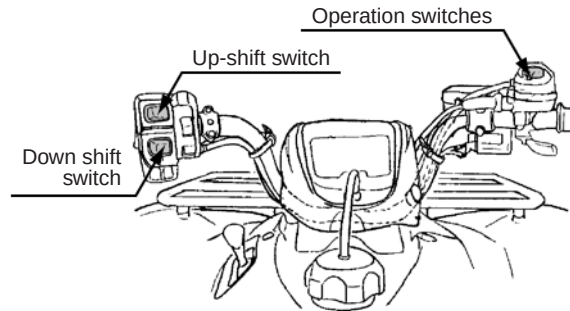


Fig. 11 Arrangement of switches

6. Conclusion

A dual-clutch transmission system was developed with the goal of providing running-through performance suitable for an ATV while also providing both an easy ride and an enhancement in fuel economy relative to previous ATV automatic transmissions, including the hydraulic-mechanical transmission.

- (1) The transmission shaft arrangement and the locations at which the clutch and change gear are attached were made the same as on a vehicle with manual transmission. The frame was also shared with the manual transmission vehicle.
- (2) Making use of distinctive ATV characteristics, the hydraulic control circuits were simplified and the motorized gearshift was carried over to the new system for use in preparatory shifting. These steps realized a system that is simpler, lighter in weight, and more compact than the previous systems in automobiles.
- (3) Shift shock was reduced to the level of an automatic transmission with a torque converter by using electronic control of the hydraulic pressure through a linear solenoid and by making use of distinctive ATV characteristics.
- (4) Preparatory shifting was accomplished without using a synchronizer or other such special mechanism. The use of motor control can reduce relative rotation speed as well as undue mechanical noise.

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■ Author ■



Kinya MIZUNO



Seiji HAMAOKA



Eiji KITTAKE



Masaki KOBAYASHI

Construction of Support System for Suspension and Steering Layout of ATV

Kazuhiro TAMURA*

Toyokazu NAKAMURA*

ABSTRACT

When designing All-Terrain Vehicle (ATV) suspension and steering, over 40 different items should be verified in the layout stage, such as the change in geometry when the vehicle attitude changes due to wheel stroke, and the swing angle of each joint. Thus far it has been necessary to use multiple tools depending on the item to be verified, which has hindered the efficient investigation of specifications. Therefore, a system that enables verification in conjunction with the CAD layout was constructed by integrating the layout support tool and the mechanical analysis function built into CAD. In addition, rubber part deformation when the vehicle attitude changes is calculated using the nonlinear analysis built into CAD, enabling verification of the clearance that takes into account rubber part deformation. This enhances the layout accuracy and reduces man-hours when designing ATV suspension and steering.

1. Introduction

The suspension and steering of an All-Terrain Vehicle (ATV)⁽¹⁾ is a complex assembly of parts that swings, so it is challenging to investigate the effects of vehicle attitude changes due to wheel stroke in the layout stage. For this reason, ATV designers thus far have verified the effects of vehicle attitude changes using multiple tools for individual purposes, and then reflected these results to the layout.

With the popularization of 3D CAD in recent years, tools that can perform mechanical analysis and nonlinear analysis for calculating items such as rubber part deformation in 3D CAD have appeared on the market, and designers are able to investigate an increasing range of items by themselves.

This development integrates multiple verification functions used thus far by ATV designers into a 3D CAD tool, incorporates advanced CAE technology into a verification tool, and reports the resulting enhancement in layout verification efficiency and accuracy.

2. Features of ATV Suspension and Steering Layout

As shown in Fig. 1, the typical ATV suspension and steering has a structure similar to that of a passenger car⁽²⁾. However, it also has the following features for

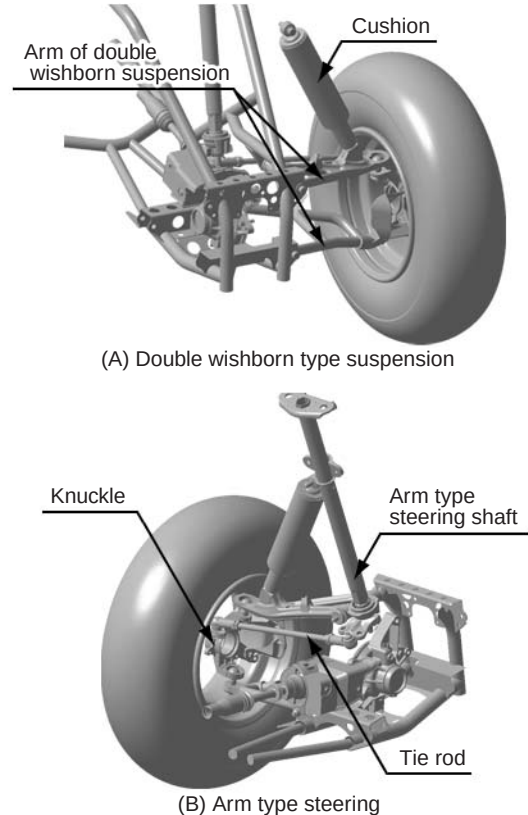


Fig. 1 Structures of assembled ATV suspension and steering

* Motorcycle R&D Center

applications such as sports running on dirt courses and carrying cargo on farm and ranch land.

- (1) The seating capacity is one person, and the tread width is approximately 1 m.
- (2) The tire camber angle, toe-in amount and other items vary greatly as shown in Fig. 2 to allow a large wheel stroke with a small suspension arm. In addition, the ball joint and arm pivot angles also vary greatly.
- (3) Center-arm-type steering is used, and a small steering angle produces a large tire turning angle.
- (4) The wheel radius is small, so the space around the wheel is narrow.

When developing an ATV, the layout needs to be investigated in consideration of these features.

3. Suspension and Steering Layout Issues

In an ATV suspension and steering layout, there are over 40 different items to verify, such as reducing changes in the geometry when the vehicle attitude changes due to wheel stroke, making sure that the swing angle of each joint does not exceed the operating limit, and securing clearance of the specified value or more. Issues when verifying these items in the layout stage are as follows.

- (1) Changes in vehicle attitude need to be verified using multiple calculation tools for different purposes, which has hindered the efficient investigation of specifications.

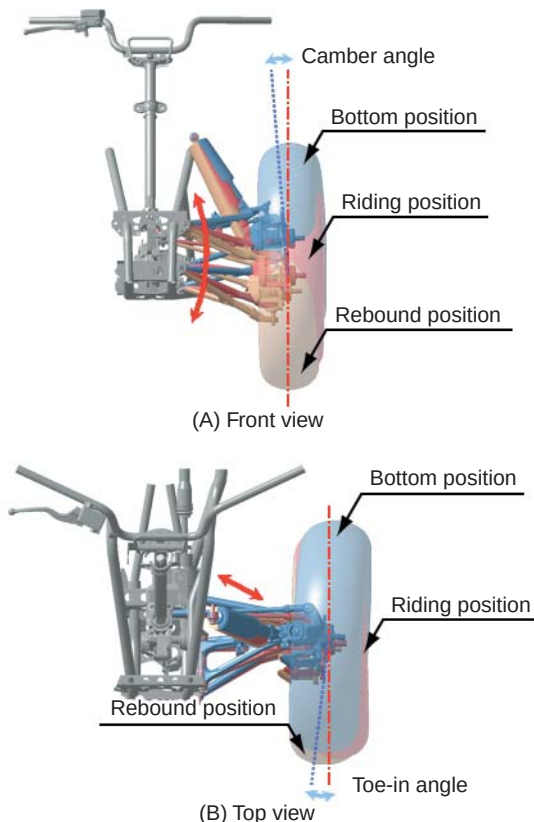


Fig. 2 Wheel stroke characteristics of wheel alignment

- (2) Tools thus far have offered only graph or numeric output of the change in camber angle or the joint swing angle during wheel stroke, and it was challenging for engineers to understand three-dimensional motion based on this data.
- (3) After the basic layout is determined, driving tests are conducted and the results are used to tune and adjust the toe-in and cushion lengths. However, it was challenging to accurately redraw the changed layout in CAD in order to reconfirm the verification items for the tuned layout.
- (4) The brake hose, rubber boot, and other rubber parts located around the suspension and steering undergo forcible bending and other deformation when the vehicle attitude changes. Nonlinear analysis is needed to investigate complex rubber deformation, but this entails the use of specialized technology, so application was limited.

4. Construction of an ATV Suspension and Steering Layout Support Tool

A tool was newly constructed that settles the issues noted in section 3 above. This tool enables calculation of changes in vehicle attitude and deformation of rubber parts around the suspension and steering on 3D CAD, allows designers to understand complex mechanical motion, and enhances layout accuracy. Figure 3 shows an outline of this tool.

4.1. Mechanical Analysis Models

A function was constructed that automatically creates a mechanical analysis model, such as that shown in Fig. 4. With this function, simply by selecting the ATV joint position in CAD, the change in vehicle attitude was

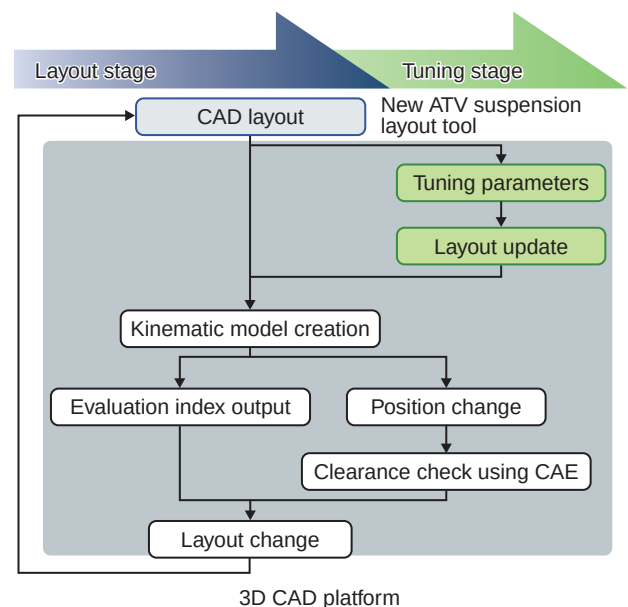


Fig. 3 Outline of new ATV suspension layout tool

calculated by using the mechanical analysis function built into 3D CAD. Automatic model creation is realized by preparing a basic mechanical analysis model for each ATV suspension and steering type, and reflecting the layout joint positions to these basic models.

The mechanical analysis models are created so that the vehicle attitude can be uniquely determined according to the wheel stroke and steering amounts. This is done by defining motion with limited degrees of translational and rotational freedom for the ball joints, rotating joints, and other joints at each joint position. This tool uses two mechanical analysis models: one for applications that calculate the change in vehicle attitude for an optional stroke amount, and one for applications that calculate the change in vehicle attitude for an optional steering amount.

• Model 1

Application: Calculating the vehicle attitude position for an optional wheel stroke amount, and calculating the cushion length for that attitude.

Configuration: This model defines a joint that restricts the vertical height from the ground-to-wheel center position to an optional value [Fig. 5(A)].

• Model 2

Application: Calculating the vehicle attitude position for an optional steering amount.

Configuration: This model defines a joint that restricts the cushion length to an optional value, and a joint that restricts the steering angle to an optional value [Fig. 5(B)].

Only model 1 is used to obtain the vehicle attitude change due to wheel stroke. When the attitude change by steering is obtained while restricting the vertical height of the wheel center using model 1, the cushion length changes unexpectedly. Therefore, the cushion length at the attitude obtained by model 1 is applied to model 2, and steering is performed while maintaining that cushion length.

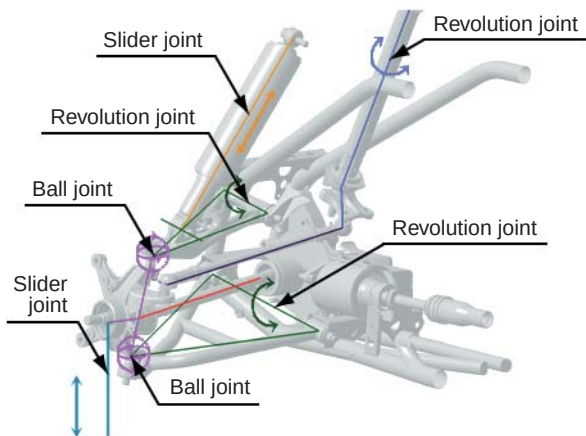


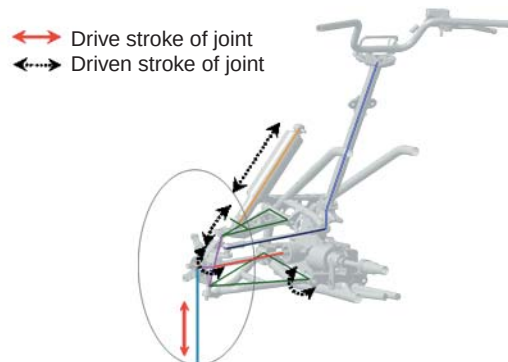
Fig. 4 ATV kinematics model

4.2. Calculation of Geometry

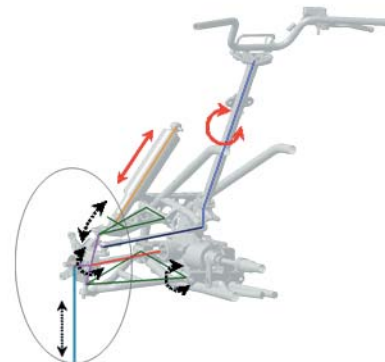
This tool recalculates and outputs the data in the form of characteristics graphs and evaluation values based on the main point coordinates of each part when the vehicle attitude changes within the mechanical analysis model. This enables evaluation of verification items such as the camber angle change and swing angle of each joint due to wheel stroke.

The geometric verification items include values that cannot be calculated as a simple angle in CAD, so these evaluation values are recalculated from the part position coordinates, and the following processing is performed by programs other than CAD.

- (1) When both the front and rear wheels are in contact with the ground, the vehicle attitude is tilted forward by 3° relative to the CAD layout, so the CAD coordinate values are converted to a coordinate system that uses the ground surface as the reference before calculating the evaluation values.
- (2) When the camber angle and toe-in angle are calculated from the coordinates at the top, bottom, front and rear edges on the tire outer, knuckle rotation around the wheel axis due to wheel stroke cannot be taken into account. Therefore, these angles are calculated from the coordinates of a reference point on the wheel center axis that is not influenced by rotation.
- (3) The swing angle clearance, until the ball joints at both



(A) Wheel stroke control kinematics model



(B) Cushion stroke control kinematics model

Fig. 5 Selection of ATV kinematics models

ends of the tie rod reach the operating limits when the vehicle attitude changes, is verified by the following method. Here, the side of the ball joint that swings is called the shank, the side that is fixed to the tie rod is called the housing, and the respective reference axes are defined as shown in Fig. 6(A). The torsional angle $\delta_{2(Limit)}$ at which the tie rod rotates on the central axis until the shank axis reaches the operating limit position at each vehicle attitude position is calculated by the following equations. Here, the swing range of shank axis is inside the cone, the radius of which (generating line) is “1”, and the vertex angle is the ball joint operating limit angle γ as shown in Fig. 6(B).

$$y = 1 \times \sin \delta_1, z = 1 \times \cos \gamma \quad (1)$$

$$x = \sqrt{\sin^2 \gamma - y^2} = \sqrt{\sin^2 \gamma - \sin^2 \delta_1} \quad (2)$$

$$\begin{aligned} \delta_{2(Limit)} &= \tan^{-1} \left(\frac{x}{z} \right) \\ &= \tan^{-1} \left(\frac{\sqrt{\sin^2 \gamma - \sin^2 \delta_1}}{\cos \gamma} \right) \end{aligned} \quad (3)$$

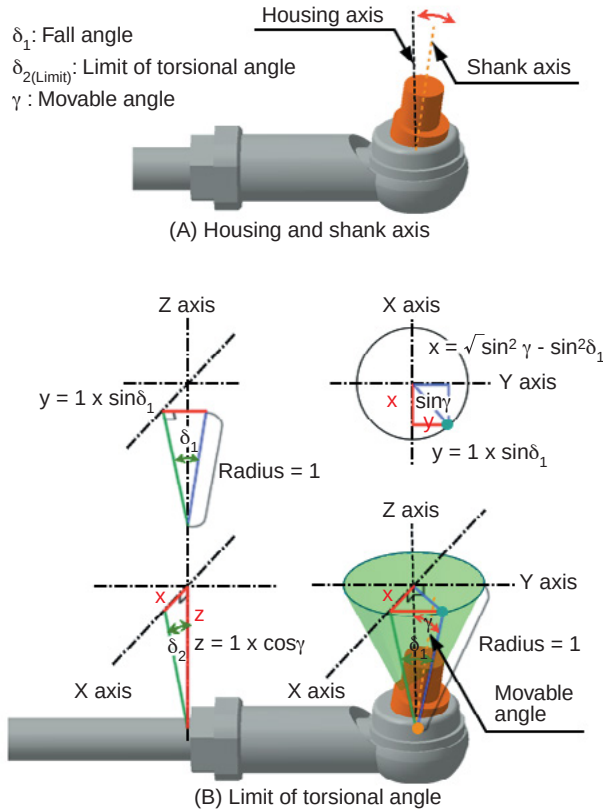


Fig. 6 Ball joint moving angle

y axis: Tie rod axis direction

z axis: Ball joint housing reference axis direction

γ : Ball joint swing operating limit angle

δ_1 : Fall angle by which the shank shaft tilts from the Z axis towards the y axis at each vehicle attitude position

δ_2 : Torsional angle by which the shank shaft rotates on the tie rod central axis (towards the x axis) from the z axis at each vehicle attitude position

- (4) In addition to the swing angle of the shank shaft on one side described in (3) above, the relative angle of the housing at both ends, as viewed from the tie rod axis direction, determines the clearance angle until the operating limit is reached. This relative angle is determined as the angle when the lock nut is tightened in the position with the tie rod twisted completely toward the front of the vehicle as shown in Fig. 7, in the vehicle attitude at the fully extended wheel stroke. Therefore, this angle was calculated by obtaining the torsional angle limits from the positions of the ball joints on both ends in the vehicle attitude at the fully extended wheel stroke.

In addition, when outputting the verification item calculation results, the layout information of each joint position and calculation results are registered in the tool, allowing to accumulate past layout information and overlay characteristics graphs to compare different layout specifications.

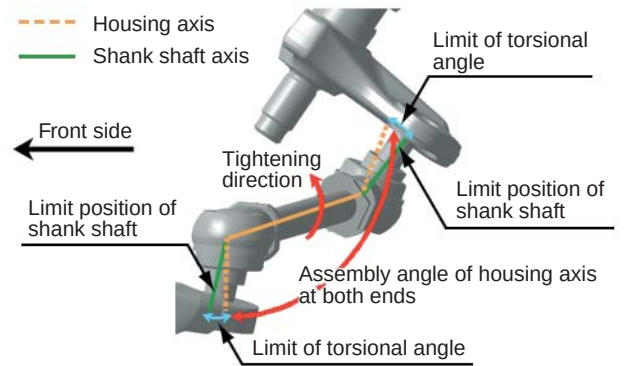


Fig. 7 Assembly angle of housing axis at both ends

4.3. Verification of Layout after Tuning

After driving tests, the layout is tuned by changing the tie rod length with the lock nut tightening position to adjust the toe-in amount, and by replacing the cushion to adjust the wheel stroke characteristics. The shapes of parts other than the tie rod and the cushion do not change, but each part is assembled via the joints, so the layout positions of other parts also change. Formerly, the layout had to be redrawn in CAD to reconfirm the verification items in the changed layout, but it was challenging to accurately obtain the position of each part that moved in

a three-dimensional manner. Therefore, a function was constructed that automatically obtains the layout position of each part, after tuning, simply by inputting numerical values of the toe-in amount and cushion length used for tuning.

To obtain the layout position of each part changed by tuning, translational joints that expand and contract the cushion and tie rod were incorporated into the mechanical analysis model that calculates vehicle attitude changes. First, the translational joint incorporated into the cushion is changed to the input cushion length by expanding and contracting. The translational joint that is incorporated into the tie rod is then adjusted in small increments so that the toe-in amount matches the input value. This translation joint adjustment process is repeated until the difference converges. This function enables the layout positions of other parts that are affected to be quickly and accurately ascertained when the toe-in amount and the cushion length change.

This tool has a special screen that obtains the layout positions after tuning by inputting the changed toe-in amount and cushion length based on the basic layout positions.

4.4. Verification of Clearance when Rubber Parts Deform

When evaluating the clearance between parts when the vehicle attitude changes, it is necessary to take into account how the brake hose, rubber boot, and other rubber parts located around the suspension and steering deform when the vehicle attitude changes. Thus far, rubber part deformation has been calculated by full-time analysts. However, a function was developed that calculates this deformation using the nonlinear analysis built into CAD, allowing designers to verify deformation at all vehicle attitudes.

Brake hose deformation is obtained by defining the hose end and clamp positions at the reference vehicle attitude, and then using the mechanical analysis model to calculate the hose end and clamp positions when the vehicle attitude changes by wheel stroke. The brake hose deformation at that vehicle attitude is calculated by nonlinear analysis, and the results are output as a CAD shape (Fig. 8). This function enables the CAD layout accuracy to be enhanced, such as by investigating clamp positions that can secure the necessary clearance.

Rubber boot deformation is obtained by reading the bend angle of the assembled joint when the vehicle attitude changes, calculating the deformed shape by applying this bend angle and the boot mounting location as conditions to the nonlinear analysis, and using these calculation results to create the CAD shape after deformation (Fig. 9). This calculation can reproduce the condition whereby the bellow ridges of the boot contact each other and bulge outward at large bend angles. Even complex deformation such as this could be reproduced to within an accuracy of approximately ± 3 mm.

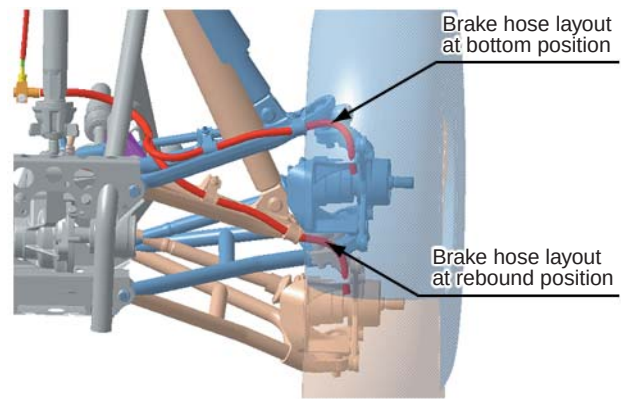


Fig. 8 Clearance with brake hose

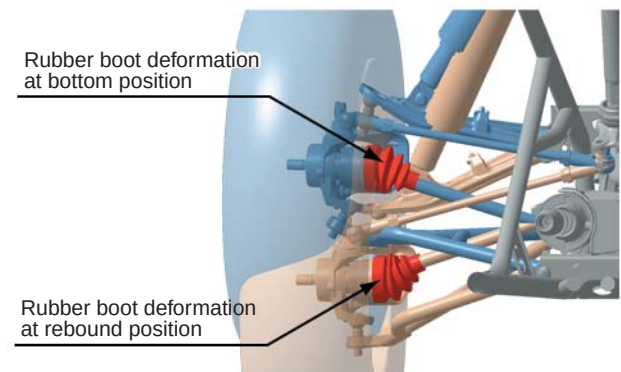


Fig. 9 Clearance with rubber boot

5. Effects of Application

Functions for calculating the suspension and steering geometry when the vehicle attitude changes were integrated, and functions for automatically calculating layout changes after tuning were combined into a tool. This reduced the man-hours needed to investigate suspension and steering layout specifications to 1/4 those of the previous level.

In addition, thus far layouts were investigated using only graphs and numeric results, but this tool enables 3D visual confirmation of conditions when the vehicle attitude changes, enabling investigation of complex mechanical motion.

Designers can also verify the clearance between parts, taking into account the rubber part deformation when the vehicle attitude changes.

6. Conclusion

A tool was constructed to evaluate verification items when ATV designers investigate suspension and steering layout specifications, and the following functions were realized.

- (1) Functions for evaluating verification items were integrated into a tool that uses 3D CAD mechanical analysis. This allows designers to understand the complex mechanical motion in three dimensions in conjunction with the CAD layout, and reduces the man-hours needed to investigate layout specifications to 1/4 those of the previous level.
- (2) The position of each part in the changed layout can be quickly and accurately calculated using mechanical analysis models in order to evaluate the layout after tuning by driving tests.
- (3) The deformation of the brake hose, rubber boot, and other rubber parts when the vehicle attitude changes can be calculated using the nonlinear analysis built into CAD, enabling verification of the clearance with surrounding parts.

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■ Author ■



Kazuhiro TAMURA



Toyokazu NAKAMURA

Motorcycle Riding Simulator Enabling Experience of Honda's Advanced Brake Systems

Katsuhito AOKI* Sadanao ICHIMI*

ABSTRACT

In order to spread motorcycles with advanced brake systems, the “Honda Riding Simulator” was utilized as a way to experience advanced brakes. Since Honda Riding Simulator only reproduced the brake feeling based on conventional brake systems at the time of braking, we modified the pitching motion of the simulator and the tire squeal sound generated by the sound system in order to enable users to experience the characteristics of advanced brake systems. In addition, a scenario was created enabling easy comparison of the differences between conventional and advanced brake systems. Test-ride events of the simulator were conducted at the 2007 Tokyo Motor Show. The modified Honda Riding Simulator fulfilled the development goal of providing a safe and efficient means of enabling users to experience the feature of advanced brake systems.

1. Introduction

Honda actively pursues research on advanced brake systems - the Combined Brake System (CBS), the Anti-lock Brake System (ABS), and the Combined ABS that brings the two systems together as an effective technological means to help support safe motorcycle riding. In order for these technologies to be accepted and achieve a greater level of use, it is important to increase user awareness of them. The modification of the Honda Riding Simulator, which was already in use in motorcycle riding schools, was therefore planned in order to provide a safe and efficient means for users to experience and understand the advanced brake systems.

2. Status of Collisions and Social Background

According to World Health Organization (WHO) statistics for FY2004, traffic collisions kill approximately 1.2 million people and injure more than 50 million people annually. Approximately 22.8% of fatalities resulting from injuries occur due to traffic collisions. This figure is 6.7

times the rate of deaths resulting from wars (3.4%) (Fig. 1). Traffic collisions ranked 9th in terms of cause of death in 1990, but are expected to rise to 3rd place in 2020. As the leader in the motorcycle industry, Honda has had a long-term involvement in efforts to increase motorcycle safety.

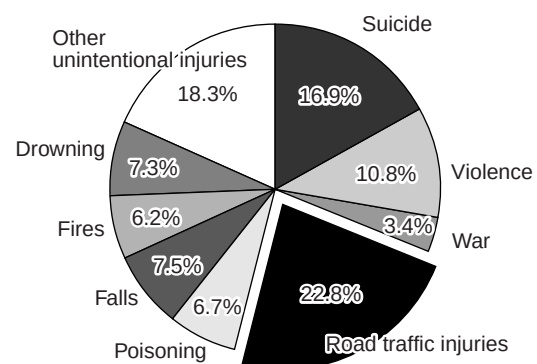


Fig. 1 Distribution of global injury mortality by cause (WHO 2004)

* Motorcycle R&D Center

3. Honda's Safety Technology Initiatives

3.1. Stages of Motorcycle Safety and Safety Technologies

Safety technologies can be broadly divided into two types: "Passive safety" technologies, which function to mitigate impact when a collision occurs, and "active safety" technologies, which function before a collision occurs. Depending on the margin of time before impact, active safety measures can be further classified into support for the rider's avoidance maneuvers, support for recognition of danger, and education to increase safety awareness (Fig. 2).

Honda has conducted research directed towards developing technologies for each of these safety stages. The airbag system fitted on the GL1800, advanced brake system, and inter-vehicle communication system (still in research phase), and the Honda Riding Simulator that is primarily used in Japanese motorcycle riding schools represent some of the outcomes of this research (Table 1).

3.2. Support for Avoidance Maneuvers

Technologies classified as "active safety" technologies

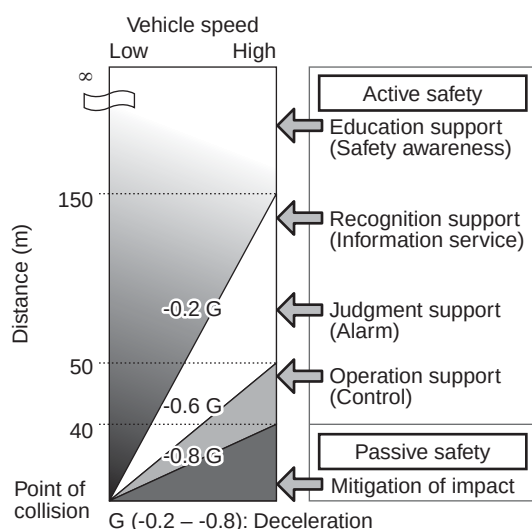


Fig. 2 Safety stages

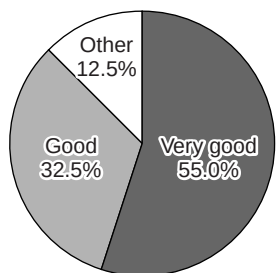


Fig. 3 User assessment of advanced brake systems

Table 1 Safety technologies for motorcycles

	☆ In use	□ Under research
Active safety		
Safety education	☆ Riding simulators	
Recognition support	□ Inter-vehicle communication systems	
Operation support	☆ Advanced brake systems	
Passive safety		
Mitigation of injury	☆ Airbag systems	

are helpful in reducing the number of collisions. Among these, advanced brake systems are technologies that enhance rider confidence.

3.3. Advanced Brake Systems

Motorcycle brake systems originally functioned separately for the front and rear wheels. The center of gravity in relation to the length of the wheel base is higher in a motorcycle than in a four-wheeled vehicle, and therefore there is a higher degree of change in the distribution of vertical load between the front and rear wheels during deceleration. This makes it necessary to appropriately adjust the operation of the front and rear brakes in order to generate a high degree of braking force. In addition, overly forceful application of the brakes can cause the wheels to lock. Honda developed its motorcycle advanced brake systems in order to address these issues. Honda has developed and put into use a number of advanced brake systems. The Combined Brake System (CBS) appropriately allocates braking force between the front and rear wheels with only the operation of the rear brake lever or a pedal. The Antilock Brake System (ABS) functions to help prevent the wheels from locking during braking. The Combined ABS combines these two functions. The goal of the development of the advanced brake systems has been the enhancement of the rider's sense of confidence.

At the same time that Honda continues to perform R&D of safety technologies, Honda also is seeking to increase usage of motorcycles fitted with advanced brake systems. Surveys of existing users of motorcycles fitted with CBS and Combined ABS in Europe show that 87.5% of users have positive opinions regarding the systems (Fig. 3). This indicates that although Advanced Brakes have received good ratings by existing users, in order to promote their widespread usage, it is important to ascertain how we can get new users to select models equipped with advanced brake systems.

4. Use of Riding Simulator to Increase User Awareness of the Advanced Brake Systems

This is where we came up with the idea of utilizing the "Honda Riding Simulator" to promote the features of Advanced Brakes to new users.

4.1. Features of Honda Riding Simulator

In 1988, research commenced on the Honda Riding Simulator as a new educational tool that would offer an experience beyond traditional lectures in the process of traffic safety education. The goal was to offer riders a simulated experience of various riding situations that can arise, to help increase their ability to anticipate and predict potentially dangerous situations on the road. The simulator enables riders to experience programmed scenarios that can result in representative motorcycle incidents while “riding” in a virtual environment of city streets. The base for the new model developed in the project discussed here is the second generation of the simulator, which has been marketed since 2001, chiefly to Japanese riding schools.

4.2. Configuration

Based on the concept of offering the user the sense that they are actually riding is important to the training effect of the unit. The simulator is provided with a motion unit controlled by AC servomotors, and has realistic image projection and high-quality sound production functions (Fig. 4).

4.3. Image System / Audio System / Motion Unit

The simulator’s image system uses a 3D graphics accelerator to project a realistic, high-quality image at 30 frames per second. The audio system uses digital

processing to synthesize sound in real time based on samples of actual sounds. The sound of the engine, for example, changes depending on engine speed and load, and is quite realistic. The audio system also produces a variety of other sounds that express different vehicle states, including the sound of the footrest scraping the road surface, tire squeals, and sounds of impact. The motion unit, which is controlled by a servomotor, can transmit to the rider the feeling of acceleration and deceleration, the sensation of roll, the shock of impact, and the feel of operating steering handles. The simulator excels at providing a realistic experience of the situations that lead to incidents.

5. Enabling Experience of Advanced Brake Systems on the Simulator

The project discussed here modified the conventional riding simulator to enable users to gain a better understanding of the features of the CBS and Combined ABS advanced brake systems.

5.1. Characteristics of Advanced Brake Systems to be Reproduced

- CBS

Based on operation of the rear brakes only, the characteristics of the CBS differ from those of conventional

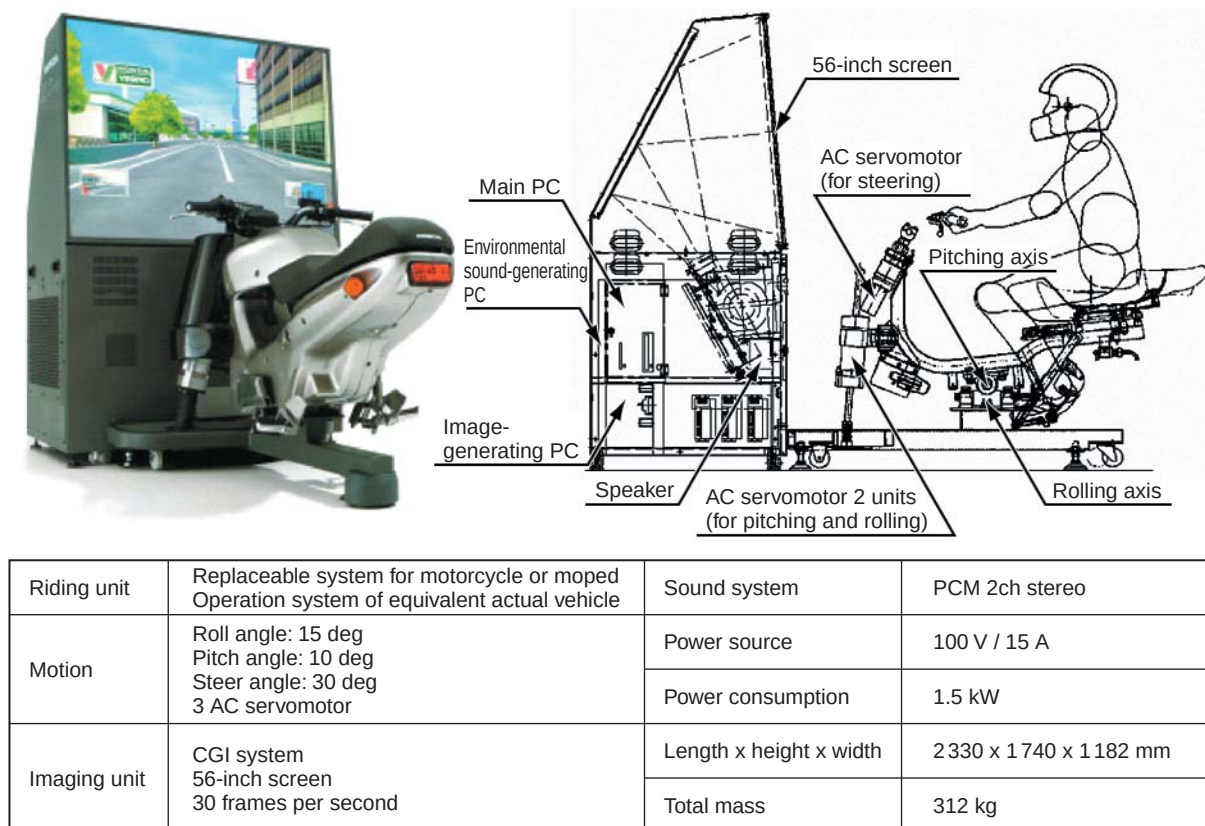


Fig. 4 Honda riding simulator

brake systems in the following respects:

1. Higher level of deceleration for the same application of force on the brake.
 2. Rear wheel reaches the slip limit first.
- Combined ABS

In addition to the features listed above, the Combined ABS is provided with a function that helps prevent the wheels from locking. It has the following characteristics:

1. Operates as CBS up to a level of rider brake input that causes the wheels to lock; operates as Combined ABS at higher levels of brake input.
2. Deceleration varies depending on the coefficient of friction of road surface during ABS operation.
3. Compared to the operation of ABS in a four-wheeled vehicle, it is easier in a motorcycle to feel changes in wheel speed through the chassis and the steering handles.

The development project sought to reproduce the characteristics listed above by making effective use of the simulator motion unit and sound system.

5.2. Reproduction of Operation of Advanced Brake Systems via Sound System and Motion Unit

Figure 5 shows the motion units pitch motion as well as the acceleration and deceleration calculations. The unit's steering equipment is identical to that of an actual motorcycle, and is fitted with sensors that detect the rider's steering operations and inputs them to the computer every 33 msec. Dynamic characteristics are expressed as equations of motion that employ the specifications of the motorcycle as constants. The state of motion for each time period is calculated by integration using the Runge-Kutta-Gill method [Fig. 5 (2)].

5.2.1. Reproduction of operation of CBS

In order to reproduce characteristics 1 and 2 of the CBS, as discussed in 5.1, the operating input to the rear brake is calculated for allocation to the front brake [Fig. 5 (1)]. The fact that the rear wheel has reached the braking limit is expressed by sounds such as tire squeals. Because the simulator is not provided with a motor enabling the motion unit to move longitudinally, longitudinal acceleration and deceleration is expressed by converting it into pitch motion of the motion unit. During deceleration, the front of the motion unit moves as if sinking in response to the level of deceleration. This is to give riders the experience of deceleration by pitching them forward so that their body weight rests on their arms. Pitch motion on an actual motorcycle occurs due to the extension and compression of the suspension, but pitch motion has a different meaning in the simulator because of the reproduction of longitudinal acceleration and deceleration. Because the rider on the simulator experiences the level of deceleration through the pitch motion of the motion unit, as indicated above, the changes in posture of the simulated motorcycle differ from those of a real motorcycle; when the calculated level of

deceleration is high, deceleration is reproduced through a greater degree of pitch motion.

5.2.2. Reproduction of operation of Combined ABS

When the rider's operation of the brakes causes the wheel lock limit to be reached, based on the coefficient of friction of the road surface that is being simulated as defined in the computer, the system is judged as being in ABS operation. Because prediction of ABS performance is not the goal of the system, ABS motion itself is not simulated, and the operating cycles used in the reproduction of ABS behavior are based on actual data for each coefficient of friction of the road surface [Fig. 5

Com: Coefficient of pitching motion of longitudinal acceleration and deceleration component
Cfm: Coefficient of pitching motion of front brake operation component
Crm: Coefficient of pitching motion of rear brake operation component
Ngain: Smoothing coefficient of longitudinal acceleration and deceleration
Scl: Longitudinal acceleration and deceleration adjustment factor depending on velocity
Cbf-Cbr: Brake force distribution coefficient
Pabs: Amplitude of pitching motion expressing ABS operation
Bfi: Rider's front brake input
Bri: Rider's rear brake input
Bfc: Front brake force
Brc: Rear brake force
hz: ABS operation cycle depending on road μ
E: Adjusted longitudinal acceleration and deceleration
Pit: Pitch angle of motion unit
T: Elapsed time
dt: Time increment

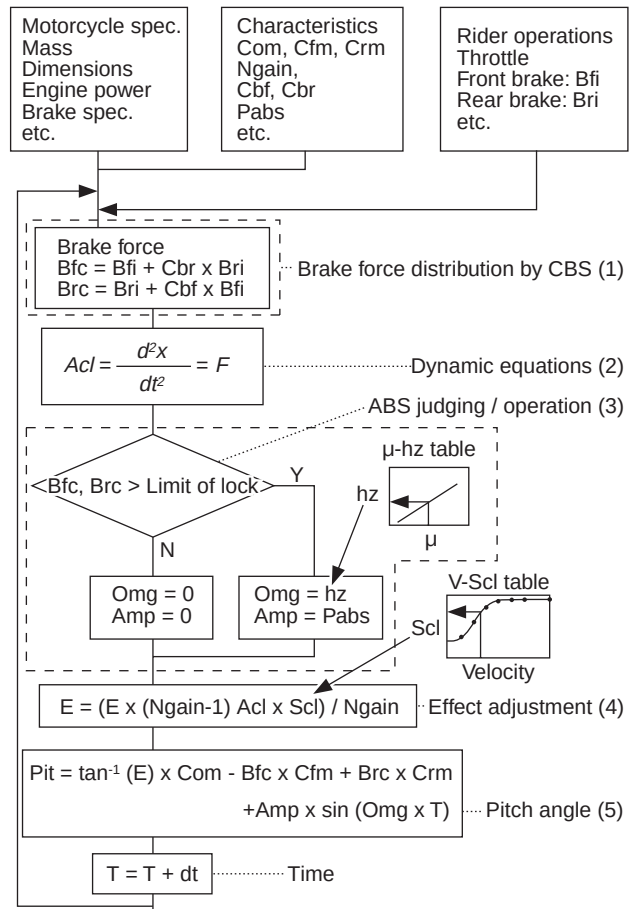


Fig. 5 Motion system control flow

(3)]. Until the wheel lock is reached, the brake system operates as a CBS and reproduces characteristic 1 of Combined ABS as listed above. To enable the rider to experience the behavior of a real motorcycle, characteristic 2 of Combined ABS is reproduced during ABS operation via pitch motion corresponding to the level of deceleration, which varies depending on the coefficient of friction of the road surface. Adding the sine wave component of the operating cycle that expresses ABS operation [(Fig. 5 (5))], which is determined by the coefficient of friction of the road surface, enables reproduction of characteristic 3 of Combined ABS. The amplitude of the sine wave component is determined by sensory evaluation. The squeal noise during ABS operation is produced by multiplying the conventional brake squeal noise by a folded-back waveform of the negative component of the sine wave that represents half the operating cycle for the ABS (Fig. 6). In this simulator, when the motion unit is moving, the calculated longitudinal acceleration and deceleration includes high-frequency noise, and is discontinuous at take-off and when the motorcycle is stopped. Because of this, the pitch angle is not substituted for longitudinal acceleration and deceleration as is, but is multiplied by the longitudinal

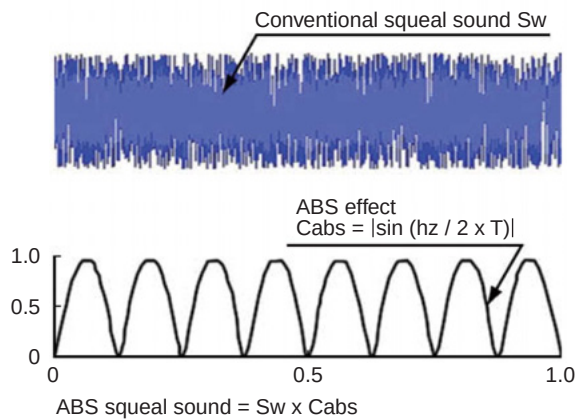


Fig. 6 ABS squeal sound

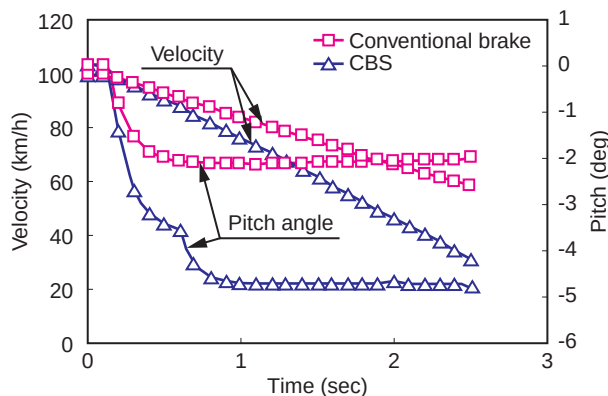


Fig. 7 CBS versus conventional brake

acceleration and deceleration adjustment factor depending on velocity, which is low at low speeds, and then further smoothed, for use as pitch motion. In addition, one of the characteristics of motorcycles is that the moment of inertia in the direction of pitch is lower than in a four-wheeled vehicle, and the vehicle's posture therefore responds more quickly to operation of the brakes. In order to reproduce this characteristic, the level of pitch motion is increased based on brake operation [Fig. 5(4), (5)].

Figure 7 shows a comparison of conventional brakes and CBS brakes calculated on the simulator. The graph shows deceleration and pitch motion resulting from use of the rear brake in isolation. This example shows that the CBS produces 1.78X the deceleration of the conventional brake system for the same level of the rear brake operation. Adding in the increase in pitch motion that is used to express deceleration, as discussed above, the CBS produces 2.28X the deceleration than the conventional brake system, enabling the user to experience a high level of deceleration. Figure 8 shows a comparison of a real motorcycle and the simulator during ABS operation. Wheel speed in the real motorcycle and pitch motion on the simulator are superimposed on the graph. This example is for braking on dry asphalt. The speed of the front wheel of the real motorcycle varies in a cycle of approximately 8 Hz. Also, in the simulation, the pitch angle varies approximately 8 Hz. This motion expresses the deceleration occurring due to the change in the speed of the front wheel in the real motorcycle. Figure 9 shows the amplitude of the tire squeal noise produced at the time of operation. The waveform of the noise also displays an

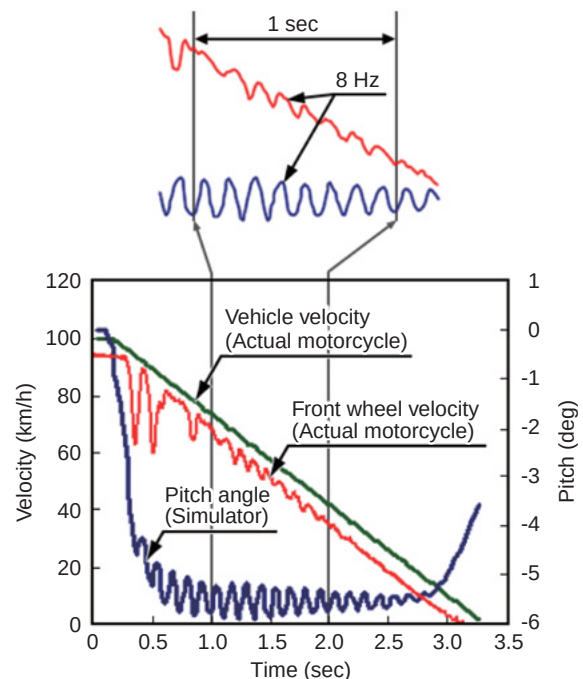


Fig. 8 Pitching motion as feeling of deceleration

8 Hz cycle. This indicates that the modes of expression of the characteristics of a real motorcycle incorporated in the simulator enable users to realistically experience the operation of advanced brake systems.

5.3. Simulator Scenarios

The monitor on the simulator control panel features a brake system menu enabling switching between conventional and advanced brakes. Initially, the concept of the system was to switch brake systems while the user was riding down the virtual street in order to enable the difference between conventional and advanced brake systems to be experienced. However, it was decided that given the fact that the vehicle speed at which a particular scenario was entered and the response to the scenario varied from rider to rider, and that the rider's attention was directed to other factors, in many cases the rider would not be attentive to the difference between brake systems. Based on these considerations, a specialized

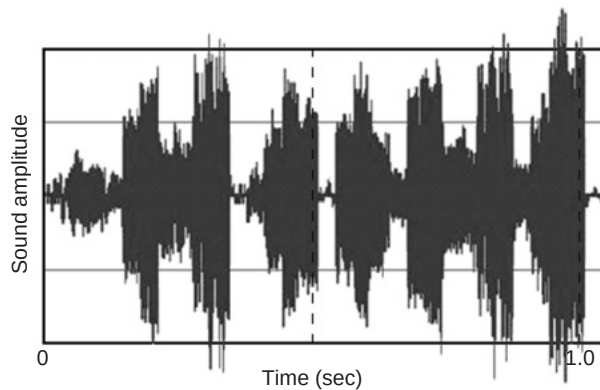


Fig. 9 Squeal sound during ABS operation

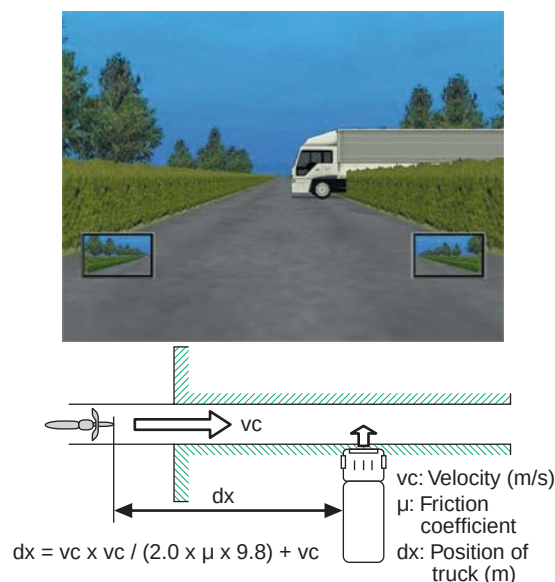


Fig. 10 Brake test course

course enabling a simpler comparison of the brake systems was employed. This is a straight course, rather than a street scene, in order to enable the rider to experience the difference in the brake system efficiently in a short period of time. Both sides of the road are planted with a hedge in the image to prevent the rider from having an unobstructed view of the sides of the road. In the scenario presented, the rider is riding down the straight road at a predetermined speed, when a truck emerges from the hedge in front into the path of the motorcycle (Fig. 10). This models a type of head-on collision that could easily occur in an urban environment. The position at which the truck appears is a position established by the shortest braking distance as determined by the coefficient of friction of the road surface when the motorcycle commences deceleration from the predetermined speed, plus a "free running distance" corresponding to a 1-second delay in brake response. The rider is instructed to apply the brakes quickly when the truck appears in order to bring the motorcycle to a halt and prevent a collision. The rider goes through this emergency braking scenario twice, using conventional and advanced brake systems. Following this, the free running distance, which shows the speed of the brake response, and the braking distance needed for application of the brakes until the motorcycle was stopped are displayed on the monitor. In addition, if a collision or crash has occurred, the reason for the crash (wheels locking, collision with truck) is displayed. Because this scenario is simple, it makes it easier for the rider to comparatively understand the difference between the brake systems.

6. Test Rides of Simulator at the 2007 Tokyo Motor Show

In addition to being the automotive industry's largest exhibitions, motor shows represent an opportunity to publicize company activities both domestically and overseas and to increase awareness of company concepts. They are therefore ideal venues in which to conduct activities aimed at increasing awareness of the advanced brake systems. Accordingly, a riding simulator enabling users to experience advanced brake systems was set up in the Honda motorcycle booth of the 2007 Tokyo Motor Show, and visitors were offered test rides for approximately two weeks. The scenario discussed above was employed, and the Combined ABS was set as the brake system. The speed of the motorcycle when the truck appears was set at 60 km/h, which is the speed limit on local streets in Japan. During the Tokyo Motor Show, 2 255 people experienced the operation of advanced brake systems on a demonstration simulator. An average of 140 persons rode the simulator on a trial basis, daily. Since test rides on the simulator took less time and effort than it would to ride a real vehicle, they were deemed efficient. In addition, although it would be very difficult to carry

out an actual brake operation on a real vehicle for one riding scene in which a track jumps out suddenly onto a course, the special features of the simulator enabled the rider to experience the advanced brake system safely.

7. Conclusion

Almost all the visitors who used the simulator during the Tokyo Motor Show responded positively when asked their impressions of the advanced brake system. Respondents felt more relaxed when braking using the advanced brake system, and understood the system. This demonstrates that the simulator is effective in enabling users to safely and efficiently experience the advanced brake systems. In the future, other uses for the simulator will be considered, including its use in safety education conducted from the perspective of helping prevent riders from overestimating advanced brake systems.

■ Author ■



Katsuhito AOKI



Sadanao ICHIMI

Diagnostic System for Power Products Using Automatic Recognition Technology for Multi-protocols

Manabu DOBASHI*

Makoto YAMAMURA*

Asagi MIWA*

Makoto YAMANAKA*

ABSTRACT

Honda has adopted serial communication, CAN communication and K-Line communication as communication protocols for its power products equipped with ECUs. Capabilities for automatic recognition of these multi-protocols were examined, using the difference in communication voltages and the difference in the ECUs' product ID data. An interface unit (I/F unit) conducts analog-to-digital conversion of the input communication voltages, transfers the data to the CPU, and determines the communication protocols. A technology has been developed to automatically and precisely recognize the protocols by checking the threshold voltages and recheck for return values of the acquisition command for product ID data.

A diagnostic system with this technology has been developed to check the operation data and malfunction data of power products. The PC application software used in the system provides functions to assist in conducting precise diagnosis. The screen display is easy to use for a wide range of operators, regardless of whether it is used indoors or outdoors. Data for new power products can also be easily built in to the system.

1. Introduction

Honda power products include products that are equipped with ECUs, such as general-purpose engines⁽¹⁾, snow throwers⁽²⁾, outboard motors⁽³⁾, electrical generators, and 4-wheel scooters⁽⁴⁾. On the other hand, many power products without ECUs are also sold. At service shops that handle power products without ECUs, in many cases the service operations that are performed depend on the skills of the operators. At service shops where power products with ECUs are also handled, there is a limit to the service operations that can be performed with human skills alone.

The use of ECU diagnostic systems has become normal practice at automobile service worksites. The ECU diagnostic system for power products was first introduced in 2003 as a special system for outboard motors, to coincide with the start of sales for the BF225⁽⁵⁾ outboard motor. However, dealers that handled products other than outboard motors also wanted a diagnostic system that could properly visualize operation data and malfunction data.

Different communication protocol types are used for different power products that are equipped with ECUs. For example, general-purpose engines and snow throwers use a half duplex asynchronous communication type of serial communication, which is a unique format. Outboard motors use K-Line communication, which primarily has been used for automobile communication. Four-wheel scooters use CAN communication as a Drive by Wire (DBW) system. An ECU diagnostic system was examined that has a function for automatically recognizing multiple communication protocols in power products equipped with ECUs.

This document describes the technology in an I/F unit that has a function for automatically recognizing multiple communication protocols, and a diagnostic system that can be used at outdoor service worksites.

2. Multi-Protocol Automatic Recognition System

In previous diagnostic systems, a special I/F unit was

* Power Products R&D Center

used for each communication protocol. For example, a special diagnostic system was used for outboard motors, and rewriting software⁽⁶⁾ was used for general-purpose engines, with ECU diagnostics performed separately for each product.

To enhance usability, a system was developed with a common I/F unit that could connect with all these communication protocols.

2.1. Communication Protocol Automatic Recognition

Table 1 shows three different communication protocol specifications: serial communication, CAN communication, and K-Line communication. Figure 1 shows a system block diagram of an I/F unit with a function for automatically recognizing these communication protocols.

During communication protocol recognition, the CPU conducts analog-to-digital conversion of the communication voltages that are input to the I/F unit. It then compares these converted values to the thresholds to determine the communication protocols. In Fig. 1, the voltage on the side sending (T_x) to the ECU is set as V_{Tx} , and the voltage on the side receiving (R_x) from the ECU is set as V_{Rx} . The sampled values in a 1 ms cycle are set as V_{Tx_i} and V_{Rx_i} , respectively, and the feedback values for determining the communication protocols are shown in Eqs. (1) and (2), and are set as V_{Txave} and V_{Rxave} , respectively.

$$V_{Txave} = \frac{\sum_{i=1}^n V_{Tx_i}}{n} \quad (1)$$

$$V_{Rxave} = \frac{\sum_{i=1}^n V_{Rx_i}}{n} \quad (2)$$

Table 2 shows the threshold for determining each communication voltage. The system compares the

Table 1 Communication protocols

Protocol	Signal voltage level (V)	Communication speed (kbps)
Serial	$0 \leq V \leq 5.0$	9.6
CAN	$1.5 \leq V \leq 3.5$	500.0
K-Line	$0 \leq V \leq \text{Battery voltage}$	10.4

Table 2 Threshold level of communication voltage

Signal	Pin name		Communication voltage level (V)
No device	Tx	0	$V \leq 0.7$
	Rx	0	$V \leq 0.7$
Serial	Tx	$0 \leq V \leq 5.0$	$V \leq 0.7$
			$4.0 \leq V \leq 5.2$
	Rx	$0 \leq V \leq 5.0$	$V \leq 0.7$
			$4.0 \leq V \leq 5.2$
CAN	CAN_L	$1.5 \leq V \leq 3.5$	$1.4 \leq V \leq 3.6$
	CAN_H	$1.5 \leq V \leq 3.5$	$1.4 \leq V \leq 3.6$
K-Line	K-Line	$8.0 \leq V \leq 16.0$	$6.0 \leq V \leq 16.0$
	SCS	$0 \leq V \leq 5.0$	$4.0 \leq V \leq 5.2$
Error	Tx	$17.0 \leq V$	$17.0 \leq V$
	Rx	$17.0 \leq V$	$17.0 \leq V$

feedback values of the communication voltage to these thresholds to determine the communication protocol.

When the communication protocol satisfies Eqs. (3) and (4), it is determined to be CAN communication.

$$V_{Txave} \geq 1.4 \wedge V_{Txave} \leq 3.6 \quad (3)$$

$$V_{Rxave} \geq 1.4 \wedge V_{Rxave} \leq 3.6 \quad (4)$$

When it satisfies Eqs. (5) and (6), it is determined to be K-Line communication.

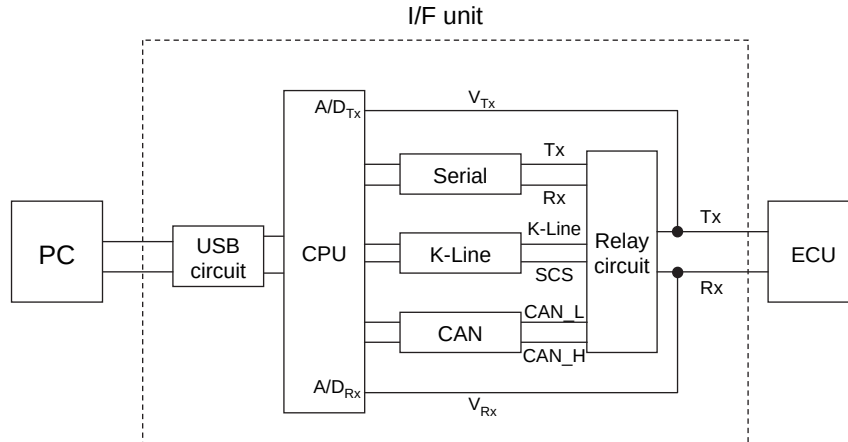


Fig. 1 I/F unit system block diagram

Table 3 Average voltage of each communication protocol

Signal	Test result voltage level (V)		Threshold voltage level (V)
Serial	V_{Txave}	0	$V \leq 0.7$
		4.8	$4.0 \leq V \leq 5.2$
	V_{Rxave}	0	$V \leq 0.7$
		4.8	$4.0 \leq V \leq 5.2$
CAN	V_{Txave}	2.5	$1.4 \leq V \leq 3.6$
	V_{Rxave}	2.5	$1.4 \leq V \leq 3.6$
K-Line	V_{Txave}	11.8	$6.0 \leq V \leq 16.0$
	V_{Rxave}	4.9	$4.0 \leq V \leq 5.2$

$$V_{Txave} \geq 6.0 \wedge V_{Txave} \leq 16.0 \quad (5)$$

$$V_{Rxave} \geq 6.0 \wedge V_{Rxave} \leq 16.0 \quad (6)$$

In serial communication, because the initial communication voltage is indefinite, the system cannot determine whether it is connected or not. If the condition in Eq. (7) is satisfied, the I/F unit provisionally determines the protocol to be serial communication and performs provisional communication. At this time, the I/F unit sends a product ID acquisition command to the ECU. If there is a response from the ECU within 5 seconds, including the retransmission, the system determines the communication protocol to be serial communication. If there is no response from the ECU within 5 seconds, the system determines that it is not connected.

$$V_{Txave} \leq 0.7 \wedge V_{Rxave} \leq 0.7 \quad (7)$$

If Eq. (8) is satisfied at this time, the system determines that there is an error in the communication voltage, and sends a voltage error generation alert to the diagnostic application software.

$$V_{Txave} \geq 17.0 \vee V_{Rxave} \geq 17.0 \quad (8)$$

After the communication protocol is determined in the CPU, the relay circuit is activated to open the communication line and establish communication with the I/F unit. After communication is established, the common product ID acquisition command for all the communication protocols is sent to the ECU from the diagnostic application software, and the ECU product ID data is acquired. This enables a recheck of whether the diagnostic application software and I/F unit have correctly recognized the communication protocol. This communication protocol automatic recognition function also continuously monitors the communication voltage during communication. The CPU determines if there are interruptions in the ECU communication line or communication voltage errors, and immediately switches to a non-connection condition.

Table 4 Pin assignment of connector

Pin No.	Serial	K-Line	CAN
1	GND	GND	GND
2	Tx	K-Line	CAN_L
3	Rx	SCS	CAN_H
4	—	12 V	—

2.2. Test Results

Table 3 shows examples of V_{Txave} and V_{Rxave} based on measured values for Eqs. (1) and (2) during the recognition for each communication protocol. It can be seen that V_{Txave} and V_{Rxave} are within the minimum and maximum thresholds. Based on these results, it was proven that when determining the communication protocol, no duplicate judgments are made for different protocols.

3. Power Products Diagnostic System

Figure 2 shows a diagram of the power products diagnostic system. The following issues were addressed when constructing this power products diagnostic system.

- (1) Integration of communication connectors
- (2) Development of diagnostic application software
- (3) Addition of new power product data

3.1. Communication Connector

A common communication connector to connect the communication unit and the power products was needed to enhance connection convenience and reduce costs. As a result, the communication connector that was installed on power products was integrated with the connector for the outboard motor ECU diagnostic system. Table 4 shows the pin layout for function assignment in each communication protocol.

3.2. Diagnostic Application Software

In ECU diagnostics, in addition to the I/F unit, diagnostic application software is needed to perform power products diagnostics using a computer. This section describes the functions of this diagnostic application software.

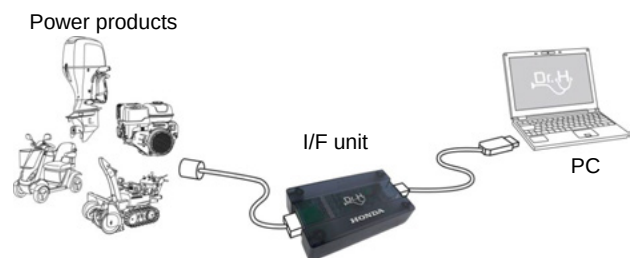


Fig. 2 Diagnostic system diagram

Table 5 Functions of application software

Screen	Detail
Data list	Checks data in real time
DTC (Data trouble code)	Checks operational errors, warnings and failures
All history	Checks history of past errors
Initial setting	Checks ECU setting information
Simple diagnostic	Chooses data and checks data automatically
Function test	Function test of single device
Graph mode	Graph and storage function
Service manual linkage	Service manual linkage
Service report function	Outputs service report

3.2.1. Diagnostic functions

The following issues needed to be addressed for the diagnostic application software to be able to retrieve ECU information and perform precise diagnostics.

- (1) Able to perform accurate diagnostic assistance
- (2) Able to quickly locate the malfunctioning location

Table 5 shows these functions and their characteristics.

“Data list”, “DTC”, “All history”, and “Initial setting” in Table 5 are functions that check the operation status and history status. “Function test” is a function that tests whether a single device that is connected to the ECU, such as a throttle motor or fuel injector, operates correctly. As shown in Fig. 3, “Graph mode” is a function that displays and saves the waveforms of data stored in the ECU. This can be used to check the instant at which the malfunction occurred.

3.2.2. Graphical User Interface (GUI) design

Different service operations for power products have different work environments, such as maintenance that is

performed outdoors. A wide range of age groups are also likely to perform the operations, so the computer screen needed to be easy to read and use.

Careful consideration was given to the coloration and brightness balance of the background and text to help achieve good visibility in sunlight and in darker environments. A friendlier GUI design was developed by creating color-coordination with good visibility over all parts of the screen, allowing use under all kinds of light conditions.

Also, to open up the tool to as many people as possible, efforts were made to create friendlier screens by carefully designing operation buttons and display text, and by representing the power products on the screen with illustrations (Fig. 4). In this way, it was hoped that operators would feel a connection with the system, which might otherwise be seen as a cold and mechanical tool.

3.3. Function for Adding New Power Product Data

Previously, whenever a power product underwent a model change or minor change, it often took a long time to develop additions to the diagnostic application software. Excel data called an “ECU data appropriation list” is used in the diagnostic application software. In this ECU data



Fig. 4 Top screen of diagnostic system



Fig. 3 Graph mode screen

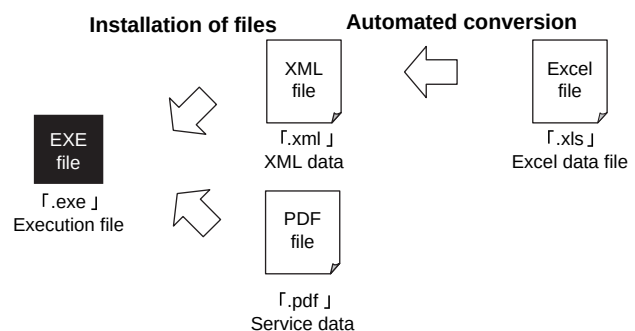


Fig. 5 File generation

appropriation list, information such as data names, physical value conversion formulas, and units are input into specific address cells. In this development, file conversion software was created that automatically converts the ECU data appropriation list from Excel data into Extensible Markup Language (XML) data.

This file conversion software enables product data to be easily added for power products undergoing a model change or minor change. Figure 5 shows the flow for adding new power products. All these flows can be performed just by a file conversion. They allow low-cost and short time-period development, even when adding new power product data.

Based on the concept of performing diagnostics for all power products, this power products diagnostic system was named “Dr. H” (symbolic of a Honda service operator who performs product diagnostics).

4. Conclusion

The following benefits will result from introducing the new power products diagnostic system that is proposed.

- (1) Enables the automatic recognition of communication protocols using the difference in communication voltages and in the ECUs’ product ID data.
- (2) A power products diagnostic system was constructed that can be connected to power products that are equipped with ECUs.
 - The communication connectors were integrated to enhance convenience when connecting to the power products diagnostic system.
 - File conversion software enables the addition of new power product data in a shorter period of time.
 - The GUI design was created by taking into account visibility, operability, and user-friendliness, to enhance its use as a work tool anywhere, by anyone.

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■ Author ■



Manabu DOBASHI



Makoto YAMAMURA



Asagi MIWA



Makoto YAMANAKA

Running Gait Generation for Biped Robot

Toru TAKENAKA*

Takashi MATSUMOTO*

Takahide YOSHIIKE*

Shinya SHIROKURA*

ABSTRACT

An attempt to increase the running speed of a biped robot often results in a loss of balance due to slip and spin generated just before lifting or after landing its foot. In order to solve this issue, we developed the gait generation technology that assures dynamic stability while suppressing slip and spin by defining the allowable range of friction force between a sole and a floor. The range of the friction force changes according to the vertical ground reaction force, and is used to adjust the horizontal acceleration, bend, and twist of the robot's torso. By adjusting the range of the friction force, a uniform approach can be taken to generate various gaits such as walking on a floor with low friction coefficient, or slow jogging. Slow jogging is an intermediate motion between walking and running in which the vertical ground reaction force does not quite reach zero. The gait generation technology realizes a running speed of 10 km/h in an experimental robot of the same size as ASIMO.

1. Introduction

Honda is engaging in research on humanoid robots that bring new value by coexisting and collaborating with people, and by providing a useful function in society⁽¹⁾⁻⁽³⁾. Research on running is crucial to the mobility of such robots in order for them to be able to move rapidly to keep up with the movement of people and respond agilely when a person appears suddenly in front of them.

Running allows a longer stride than walking because it involves forward movement above the ground. Consequently, running is advantageous for rapid movement. Running also generates a vertical ground reaction force of about twice the body weight in the vicinity of the lowest point in the vertical movement of the torso. This raises the limit of the ground friction force, thus enabling sudden acceleration and facilitating a more agile response to changing conditions.

In 2004, Honda announced that ASIMO was able to run in a straight line at 3 km/h, and in 2005, that it was able to run in a straight line at 6 km/h and in a circular pattern at 5 km/h [Figs. 1(a) and (b)]⁽³⁾. Continuing

research subsequently yielded an experimental robot the same size as ASIMO that is able to run at 10 km/h. This paper describes the gait generation technology that was at the core of these running achievements.



(a) Running (6 km/h)



(b) Running in circular pattern (5 km/h)

Fig. 1 Running posture of ASIMO

* Fundamental Technology Research Center

2. ZMP-criteria Gait Generation

One of the general methods for biped robot walking control is to generate a desired gait and make the actual robot follow that gait. Honda robots also make use of this method. The desired gait is defined in terms of the trajectory of the foot position and posture and the trajectory of the torso position and posture, which are needed to determine the robot's joint angles. Providing dynamic stability is crucial to generating the desired gait.

The concept of the zero moment point (ZMP) has been proposed as an approach to providing this stability⁽⁴⁾. The resultant force of the gravity and the inertial force generated by the robot moving is called the total inertial force. The point on the floor at which the moment (strictly speaking, this excludes the component of rotation around the vertical axis) created by that total inertial force becomes zero is the ZMP (Fig. 2). In order for the robot to be dynamically stable, the ZMP should be within the ground contact area of the supporting leg [the region with

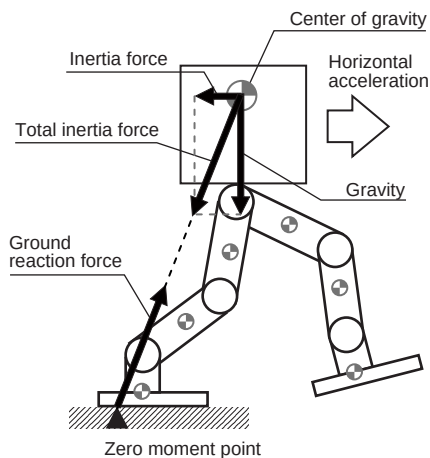


Fig. 2 Zero Moment Point (ZMP)

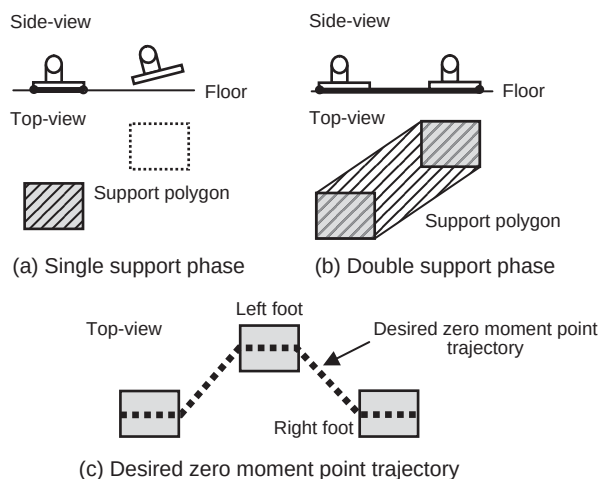


Fig. 3 Desired zero moment point trajectory

slanted lines in Fig. 3(a)] during a single support phase, and within the ground contact envelope of the two legs [the region with slanted lines in Fig. 3(b)] during a double support phase. When the ZMP departs from these bounds, then the robot cannot necessarily be provided ground contact and it may fall over.

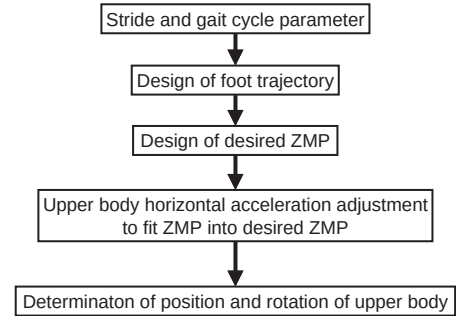
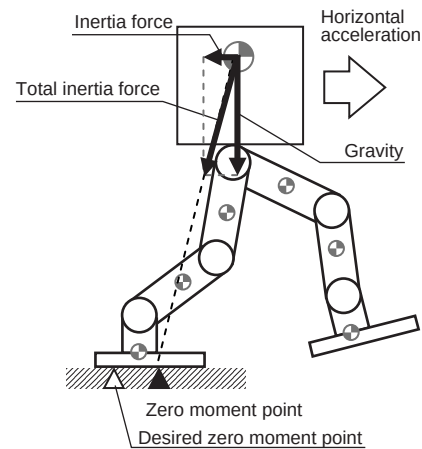
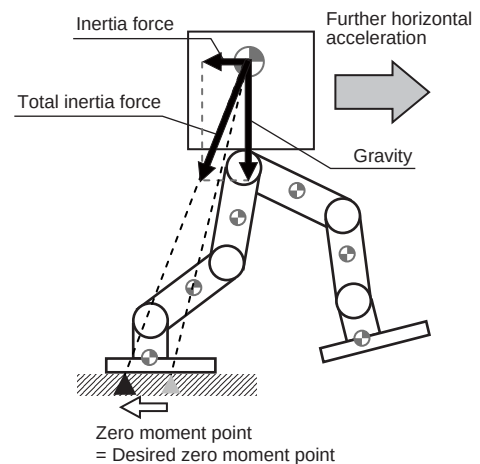


Fig. 4 Flow of ZMP based walking gait determination



(a) Before adjustment by horizontal acceleration



(b) Adjustment by horizontal acceleration

Fig. 5 Balancing motion via upper body horizontal acceleration

The ZMP-criteria gait generation method sets the trajectory of the desired ZMP to satisfy the above conditions, and generates the desired gait so that it will realize that trajectory. This method (Fig. 4) involves first setting parameters such as stride and gait cycle. These parameters then provide the basis for determining the foot trajectory. Next the desired ZMP trajectory is set within the range of stability. Specifically, the desired ZMP trajectory is set so that, from the instant when the foot stepping forward (the leading foot) touches the floor to the instant when the trailing foot lifts up, it moves from the toe of the trailing foot to the heel of the leading foot, and during the single support phase, the trajectory moves from the heel to the toe of the foot [Fig. 3(c)].

Horizontal acceleration of the torso of the robot is defined so that ZMP calculated from the robot's movement agrees with the desired ZMP. This is to make the moment generated by the total inertial force around the desired ZMP equal to zero. As shown in Fig. 5(a), for example, when the ZMP calculated from movement locates farther forward than the desired ZMP, the robot's torso position is accelerated further forward. Then the ZMP calculated from movement shifts toward the rear so that the moment around the desired ZMP will be zero [Fig. 5(b)].

Finally, the horizontal acceleration of the torso is doubly integrated to obtain the horizontal position of the torso. The torso posture is kept in a predetermined posture (an upright posture, for example). This method will generate a gait that provides stability.

In this method of ZMP-criteria gait generation, the horizontal acceleration of the center of gravity is determined dependently. The horizontal ground reaction force is also determined dependently.

3. Issues with Previous Technology

Generation of a running gait necessitates generation of movement for which the ground reaction force is zero when the robot is above the ground. Therefore, the ZMP-

criteria gait generation method, in which the ground reaction force is determined dependently, cannot be applied to generation of a running gait.

To address this issue, a method has been proposed that will generate the gait while switching the movement constraint conditions at the boundary between the flight phase when the robot is above the ground and the landing phase when it is in contact with the ground⁽⁵⁾. During the flight phase, the robot's position and posture will be determined such that the movement of the center of gravity describes a parabola while the angular momentum around the center of gravity is conserved. Meanwhile, during the landing phase, the robot's position and posture are determined by the ZMP-criteria gait generation method.

Use of this method allows the generation of a gait in which the ground reaction force will be zero during the flight phase. During the period just before lifting or after landing its foot, however, when the vertical ground reaction force is small (the slanted part of the line in Fig. 6), the dependently determined horizontal ground reaction force exceeds the limit for the ground friction force, so there is a possibility that the robot will slip or spin. Slip can cause the robot to fall over to the front or back. Spin can not only reduce the rectilinearity of the robot's movement, but it can cause deviation of the landing position, and it can also generate a large centrifugal force when the robot is running fast, making the robot fall over laterally.

In the case of slow jogging, at least one of the feet is always in contact with the ground. Therefore, the vertical ground reaction force is not zero and there is no period when the robot is above the ground. The switching method cannot be applied in such a case. On the other hand, use of the ZMP-criteria gait generation method throughout will leave open the possibility of a fall due to slip or spin even during the double support phase, when the vertical ground reaction force is small (Fig. 6).

4. Running Gait Generation

In order to address the issue described in the previous section, a method was developed to set the allowable range of the friction force, which changes continuously with the changing vertical ground reaction force, and, on that basis, to generate a gait that will suppress slip and spin. This method can provide dynamic stability just before the robot lifts or after it lands its foot. The method is also capable of uniform gait generation for movement that is in between walking and running, and for walking on a floor with a low friction coefficient. This gait generation method is introduced below.

4.1. Torso Trajectory Generation Taking the Horizontal Ground Reaction Force into Consideration

The desired ZMP trajectory during the landing phase

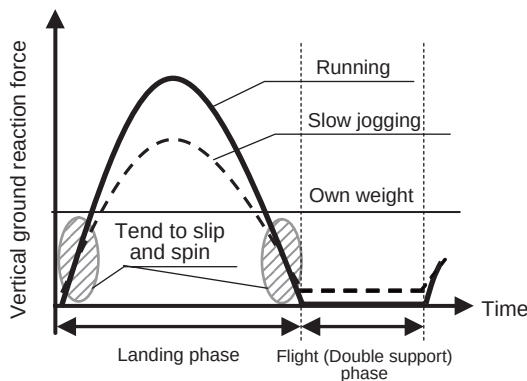


Fig. 6 Vertical ground reaction force of running and slow jogging

is set using a method similar to that for ZMP-criteria gait generation. No ground reaction force is generated during the flight phase, so that the total inertial force becomes zero, and the moment around arbitrary action point becomes zero. As a result, movement unrelated to the desired ZMP is generated. The desired ZMP can therefore be placed in an arbitrary location, but it is set to move continuously to the next landing position in order to provide continuity with the landing phase.

Figure 7 shows the flow for generating the trajectory of the torso position and posture (excluding the component of rotation around the vertical axis) when running. First, the pattern of the vertical ground reaction force is designed according to the duration of the flight phase, the duration of the landing phase, and the desired height of the center of gravity at the instant when the robot lands on the ground (Fig. 8). Next, the allowable range for the horizontal ground reaction force is set according to this vertical ground reaction force (fine line in Fig. 9).

The horizontal acceleration of the torso for the desired gait is adjusted so that the moment around the desired ZMP will be zero, as was done for the walking gait. The horizontal ground reaction force is obtained dependently

by this adjustment (dotted line in Fig. 9). If this horizontal ground reaction force is within the allowable range, this adjusted torso trajectory provides both dynamic stability and suppression of slip. If it is not, then the horizontal acceleration of the torso is limited so that the horizontal ground reaction force will come within the allowable range (heavy line in Fig. 9). This limitation, however, means that the conditions for dynamic stability, in which the moment around the desired ZMP is zero, are no longer satisfied. Dynamic stability is therefore satisfied by modifying the rate of change of the angular momentum around the center of gravity of the robot (Fig. 10). This is done by making the torso rotate (bend) while keeping the horizontal component of the acceleration of the center of gravity of the robot unchanged. This rotation of the torso has the same effect as displacing the action point of the total inertial force, so that the moment around the ZMP can be made zero.

The adjustment of the horizontal acceleration of the torso together with the compensation by the rotation of the torso satisfy the conditions for dynamic stability. However, the compensation that is brought about by rotating the torso modifies the rotational acceleration of the torso because of the limit on the horizontal ground reaction force. Consequently, the torso posture ends up diverging because of the time integral. The torso of an

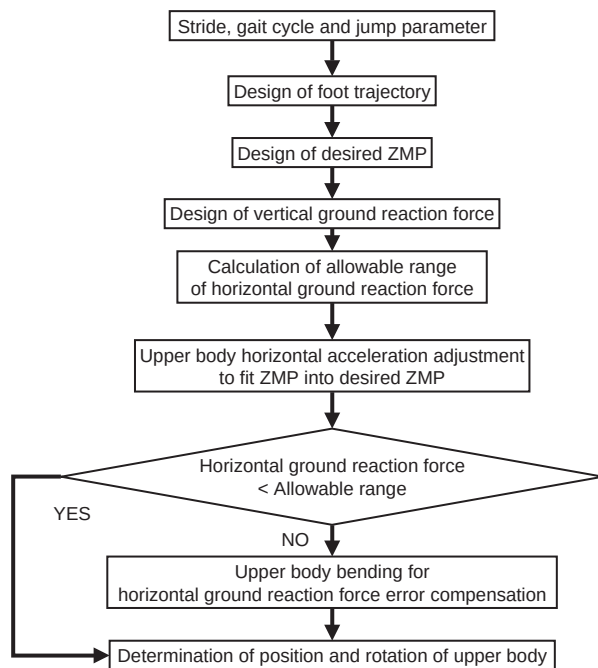


Fig. 7 Flow of running gait determination

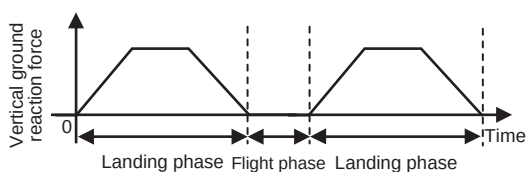


Fig. 8 Vertical ground reaction force

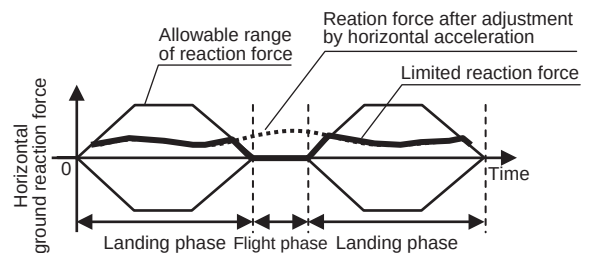


Fig. 9 Distribution of horizontal force

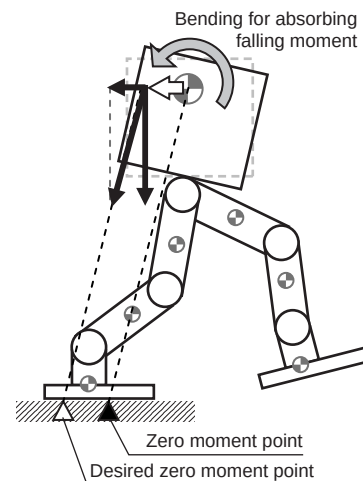


Fig. 10 Balancing motion via upper body bending

actual robot cannot be rotated endlessly, so this becomes an unrealizable gait. The torso posture is therefore recovered while the vertical ground reaction force is large.

4.2. Generation of the Trajectory of Rotation by the Torso around the Vertical Axis

When the yaw moment of the ground reaction force exceeds the moment limit of the ground friction force, a spin is generated. This yaw moment becomes large when the robot moves its legs forward at high speed, such as when running or walking fast. As noted earlier, dynamic balance is lost when spin is generated, increasing the possibility that the robot will fall over. In order to help prevent this, the allowable range of the yaw moment is set according to the vertical ground reaction force and friction coefficient between a sole and a floor (Fig. 11). When the yaw moment exceeds the allowable range, compensation is applied by the torso rotating around the vertical axis (twist) (Fig. 12), providing dynamic stability around the vertical axis. However, a modification is applied to the rotational acceleration of the torso around its vertical axis, so that the angle of rotation (twist angle) of the torso around its vertical axis ends up diverging because of the time integral, just as in the generation of

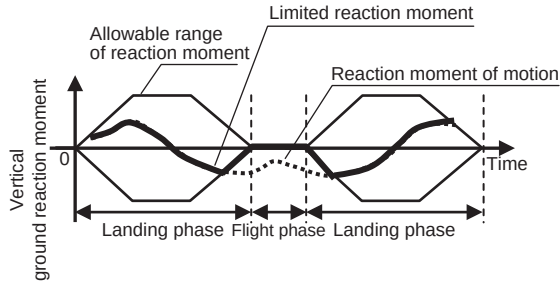


Fig. 11 Limit of vertical ground reaction moment

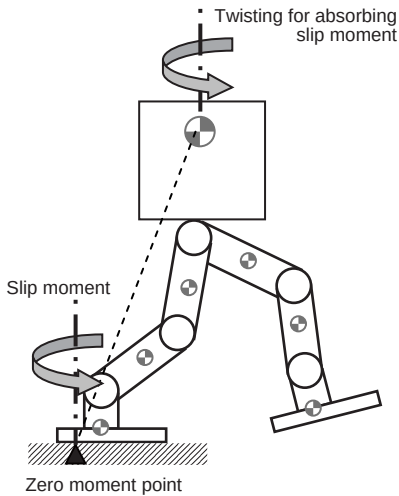


Fig. 12 Vertical moment control

the torso position and posture trajectory in the previous section. Therefore, the rotation of the torso around the vertical axis is recovered at a time when the vertical ground reaction force is large.

The method described in this section is able to provide dynamic stability just before lifting or after landing the foot. It is also capable of generating a uniform gait just by changing the allowable range of ground friction force, even during slow jogging, without switching the movement constraint conditions. This method can also generate a gait that suppresses slip, even on a floor with a low friction coefficient, just by reducing the allowable range of the ground friction force.

5. Gait Generation Results

Computer simulation of a robot with the same leg length as ASIMO was used to generate the desired gait for running at 10 km/h. The stride was set at 730 mm and

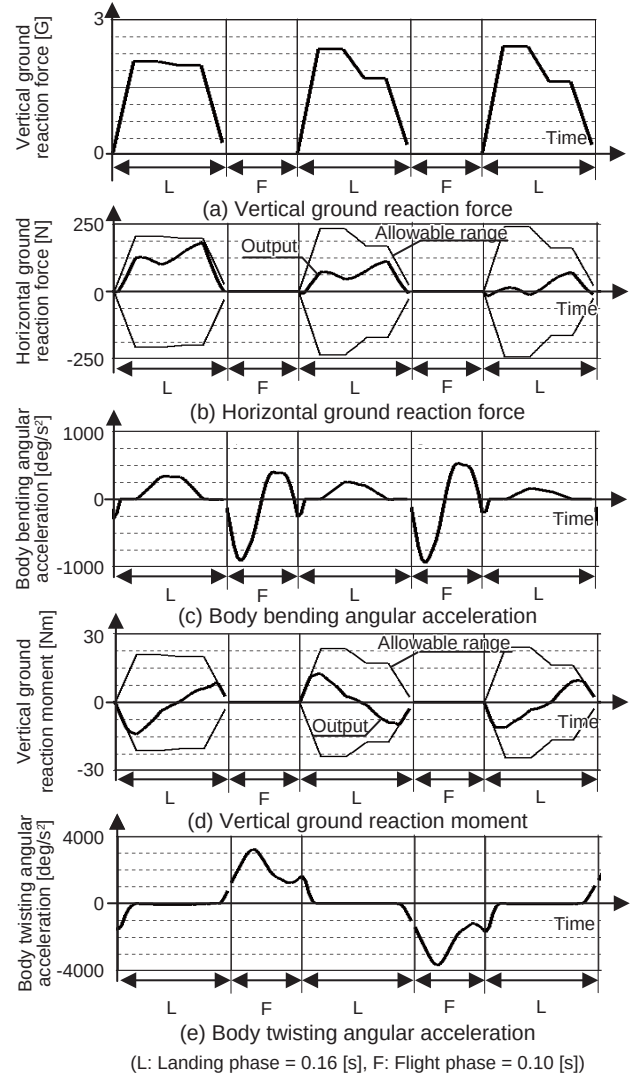


Fig. 13 Force and moment design at 10 km/h running

the gait cycle at 0.26 s (0.10 s during the flight phase, 0.16 s during the landing phase). Figure 13 shows the design value for vertical ground reaction force when running (a), the allowable range for the horizontal ground reaction force [fine line in (b)] and horizontal ground reaction force in the generated gait [heavy line in (b)], the angular acceleration of the bending of the torso (c), the allowable range for the yaw moment [fine line in (d)] and yaw moment of the generated gait [heavy line in (d)], and the angular acceleration of the twisting of the torso (e). The first landing phase in these graphs occurs in a state of accelerated running, and the subsequent two landing phases occur in the state of constant running at 10 km/h.

As shown in Figs. 13(b) and (c), the horizontal ground reaction force in the generated gait comes within the allowable range. An angular acceleration that compensates for the excess portion is generated during the flight phase and in its vicinity, in which the allowable range is reduced, and an angular acceleration for recovery is generated during the landing phase.

As shown in Figs. 13 (d) and (e), the yaw moment of the generated gait also comes within the allowable range. An angular acceleration that compensates for the excess portion is generated during the flight phase and in its vicinity, in which the allowable range is reduced. This angular acceleration is generated alternately to the left and to the right, each canceling out the other, so that the angular acceleration for recovery becomes more or less zero.

The above results indicate that this method is able to generate gaits that provide dynamic stability while also suppressing slip and spin just before the robot lifts or after it lands its foot.

6. Running Experiment

An experiment in running at 10 km/h was conducted using a robot with legs that are the same length as

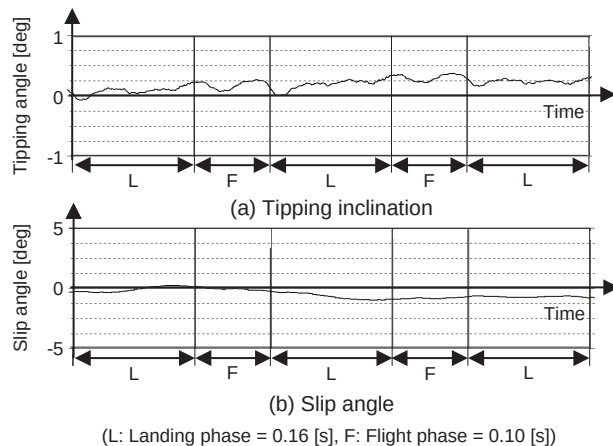


Fig. 14 Inclination error at 10 km/h running

ASIMO's but lighter in weight, and actuators with enhanced output. The robot was controlled so as to follow the desired gait generated in section 5.

With an actual robot, a tracking error occurs with respect to the desired gait because of the influence of actuator tracking capability, mechanical deflection, and other such factors. The desired gait therefore does not obtain the necessary ground reaction force, dynamic balance is lost, and the robot tips over. As this leaning is likely to result in the robot falling over, ground reaction force control (so-called compliance control)⁽¹⁾,⁽²⁾ was exercised during the landing phase so as to generate the necessary ground reaction force. In addition, a ground reaction force moment according to the inclination of the robot was generated around the desired ZMP during the landing phase to stabilize the robot's posture. Meanwhile, the robot's posture during the flight phase was stabilized using the reaction from the torso rotating around the center of gravity⁽³⁾. Figure 14(a) shows the tipping angle of the robot while Fig. 14(b) shows the spin angle at that time. The tipping angle is an estimated value from a gyrosensor and acceleration sensor. The tipping angle is held to within ± 0.4 deg while the spin is held to within ± 1 deg. It is apparent that stable running was achieved with no deterioration of behavior due to slip and spin just before the robot lifted or after it landed its foot.

7. Conclusion

Gait generation technology for a biped robot was developed that sets a continuously changing allowable range for the friction force between a sole and a floor according to the vertical ground reaction force. In order not to exceed this allowable range, the method adjusts the horizontal acceleration and bending and twisting of the torso. This technology makes available a method for providing dynamic stability during running while suppressing the slip and spin of the robot just before lifting or after landing its foot. The technology can be applied in a uniform manner just by changing the allowable range of the ground friction force for various patterns of walking, including slow jogging and other forms of intermediate movement between running and walking in which the vertical ground reaction force does not completely reach zero, as well as for when walking on a floor with a low friction coefficient.

By use of this technology, running at a speed of 10 km/h was achieved in an experimental robot the same size as ASIMO.

For the future, this technology will be used as a basis for further enhancement of movement capabilities. The goal is the realization of a robot that is able to respond in an agile manner when people appear suddenly in front of it, so that the robot can coexist and collaborate with people and fulfill a useful function in society.

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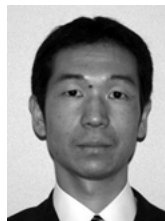
Toru TAKENAKA



Takashi MATSUMOTO



Takahide YOSHIKE



Shinya SHIROKURA

Mechanism of Occurrence of Ghost Noise in Helical Gears Resulting from Thread-type Tool Machining

Ryo EGAMI*

Tadami KONDO*

Motoki IKUMA**

ABSTRACT

Specifying the origins of ghost noises produced at non-integral multiples of gear meshing order frequency from the order at which they occur represents a challenge, and determining countermeasures for these forms of noise has involved a process of trial and error. Today, with the widespread use of a method of continuous generation involving the use of a multiple-thread-type tool for high production efficiency in the process of grinding the flanks of the helical gears used in automotive transmissions, new ghost noises have appeared. The research discussed in this paper clarified the mechanism behind the production of the ghost noises that have become noticeable as a result of the use of the multiple-thread-type tool, and identified the relationship between the conditions used in the grinding process and the orders at which the noises occur. By verifying these findings in an actual transmission and studying countermeasures, the project demonstrated that ghost noises could be reduced by modifying the conditions used in the grinding process.

1. Introduction

Gear noise normally occurs at the meshing order frequency and integral multiples of the meshing order frequency. When noise is generated at non-integral multiples of the meshing order frequency, it is termed ghost noise⁽¹⁾. Ghost noises result from cyclical undulations produced on the surface of the gear teeth during machining; it is known that ghost noises can be produced even if the amplitude of these undulations is extremely low ($1\ \mu\text{m}$ or less)^{(2), (3)}. Their cycle per tooth is not an integer, and as such, a complex periodicity is displayed in which the frequency orders at which ghost noises are produced are non-integral multiples of the meshing order frequency. Because of these, it is challenging to specify the cycles of the undulations using usual tooth form measurements^{(2), (3)}. In addition, this complex periodicity makes it challenging to infer the mechanisms by which these undulations are produced. In this paper, these undulations on the surface of the gear tooth, for which the cycle per tooth is not an integer and which represent a factor in the production of ghost noises, will be termed ghost undulations.

Ghost undulations have been shown to be produced by

transmission errors generated by gears on shafts of continuous generating grinding machine using thread-type tools⁽⁴⁾. Today, with the widespread use of multiple-thread-type tools in order to achieve production efficiency in the process of grinding the flanks of the helical gears used in automotive transmissions, ghost noises of unknown origin have appeared. It has therefore become necessary to determine the origin of these ghost noises, and to examine methods of reducing their incidence.

2. Characteristics of Data Concerning Tooth Forms Generating Ghost Noises

Figure 1 shows the drive train pathways of an automotive transmission. Figure 2 shows transmission case vibration data measured in order tracking analysis using this transmission. The horizontal axis shows order frequencies for the benchmark driveshaft rotation, and the vertical axis shows driveshaft rotation speed. At the benchmark rotation of the driveshaft, the meshing order frequency of the final gear set is the 71st order and the meshing order frequency of the 4th gear set is the 137.6th order, and these orders appear as peaks in transmission case vibration. The 331st and 662nd orders that appear in

* Automobile R&D Center

** Honda Motor Co., Ltd.

Fig. 2 are ghost order frequencies which are not harmonics of gear meshing order frequency. By a cross-check between transmissions that produce these ghost noises and transmissions that do not, it turned out that they originate in final driven gears, the flanks of which have been ground in a process of continuous generation grinding using a thread-type tool.

The ghost undulations that have been shown to be produced by transmission errors in the drive shafts of machine and other factors are produced on both the tooth profiles and the lead forms. Research has been conducted on a method of evaluating the cycle and the amplitude of the undulations using tooth profile measurement data⁽²⁾. However, the ghost noises that formed the subject of the research discussed in this paper are produced by ghost undulations in the lead form, not the tooth profile.

Figure 3 shows continuous lead form measurement data for four teeth of the final driven gear that produced the ghost noises shown in Fig. 2. The undulations have cycles of approximately 3 mm in wavelength. In addition, the undulations display regularity: the phase of the undulations on each successive tooth differs by approximately 1 mm, and the phases of the undulations on the first and fourth teeth are virtually identical.

The order frequency of the undulations is able to be specified by FFT analysis of the lead form measurement data. As shown in Fig. 4, one pitch of the lead length on the plane of action is termed L_p , and this length L_p is extracted from the lead form measurement data. This extracted data is connected for all teeth in the direction of movement of the contact line, and lead form

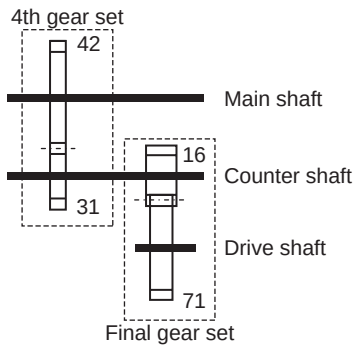


Fig. 1 Drive train of transmission in 4th gear

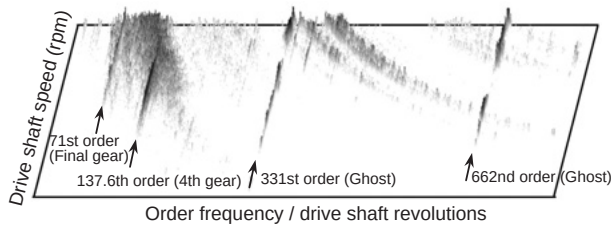


Fig. 2 Waterfall plots of transmission case vibration

measurement data for one rotation of the gear is subjected to FFT analysis as a single set of continuous waveform data. Figure 5(a) shows the results of an FFT analysis of all teeth using lead form measurement data for the gear that produced the ghost noises shown in Fig. 3. Figure 5(b) shows the results of an FFT analysis of all teeth for a gear that did not produce ghost noises. The order frequencies at which the peak amplitudes of the undulations occur in Fig. 5(a) correspond to the ghost order frequencies shown in Fig. 2, while no conspicuous undulation peaks are observed at ghost order frequencies in Fig. 5(b). This indicates that it is the cyclic undulations on the lead forms of the gear teeth that produce the ghost noises.

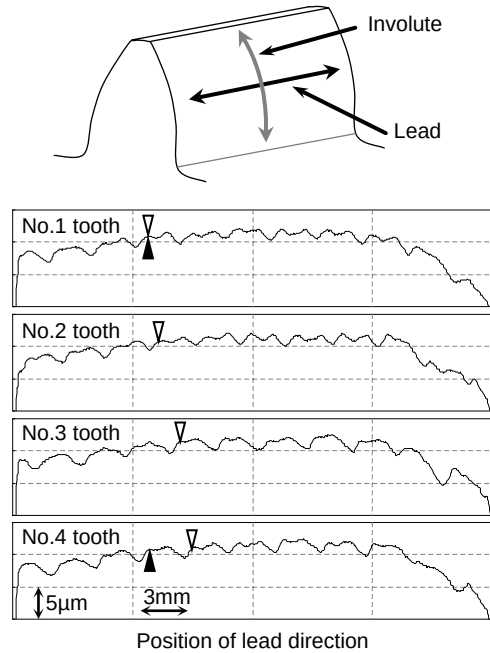


Fig. 3 Measured lead form of 'Ghost' gear

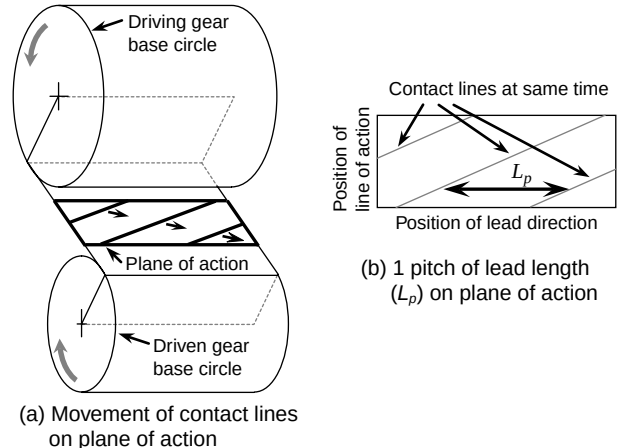


Fig. 4 Contact lines on plane of action

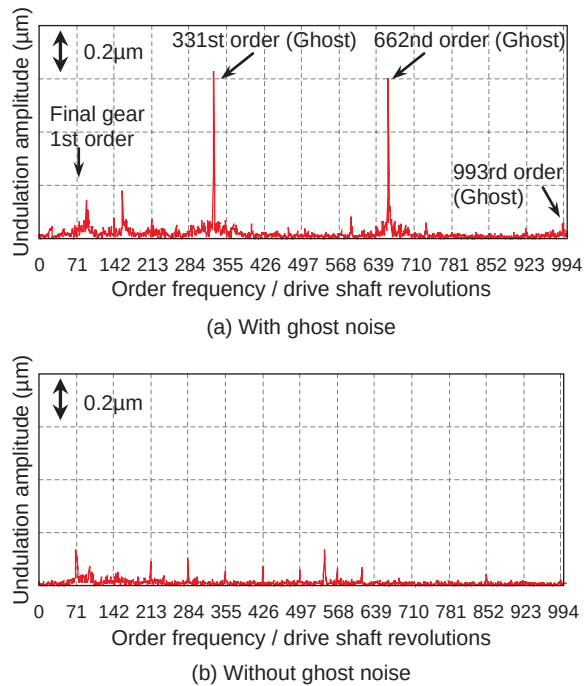


Fig. 5 Spectra of FFT analysis for compound data of lead forms of all teeth

3. Mechanism of Occurrence of Ghost Undulations by Continuous Generation Gear Grinding Using Thread-type Grinding Wheel

3.1. Mechanism of Occurrence of Ghost Undulations by Grinding Wheel Feed Pitch

Figure 6 shows the motion of the work and the grinding wheel during continuous generation gear grinding using a thread-type grinding wheel. The work and the grinding wheel rotate as if meshing to generate the tooth profile, and the feeding action of the grinding wheel in the direction of the work axis generates the lead form.

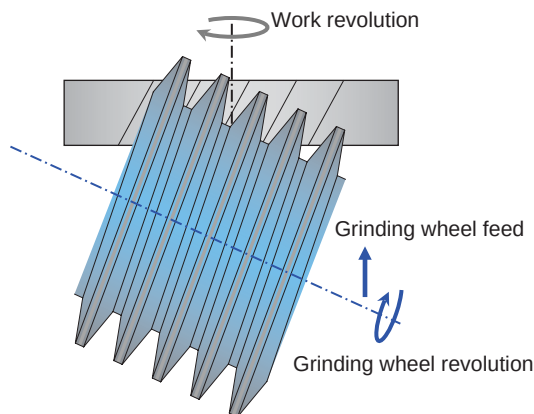


Fig. 6 Motion of grinding wheel and work during continuous generation process in gear grinding

Table 1 Simulation conditions

(a) Gear specifications

Number of teeth (Z_w)	10
1 pitch lead length (L_p)	3 mm

(b) Grinding conditions

Feed pitch (F_p)	1 mm / work rev.
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Because the grinding wheel acts discontinuously on the lead form, the feed pitch of the grinding wheel produces undulations of the type shown in Fig. 7 on the surface of the tooth. F_p represents the pitch of grinding wheel feed per rotation of the work. The feeding action of the grinding wheel continues while generating the same point of profile on the remaining teeth. Because of this, if the number of work teeth is designated Z_w , the phase of the undulations on the neighboring tooth shifts by δ_1 , as determined by Eq. (1).

$$\delta_1 = \frac{F_p}{Z_w} \quad (1)$$

Figure 8 shows the lead form undulations on a helical gear of the specifications and produced under the machining conditions shown in Table 1 as sine waves. Three-cycle feed pitch undulations are generated for each one pitch of lead length (L_p), and the phase of the

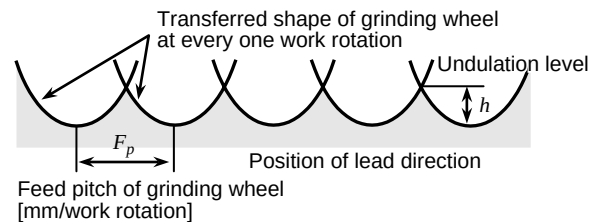


Fig. 7 Undulation of wheel feed pitch generated on lead form

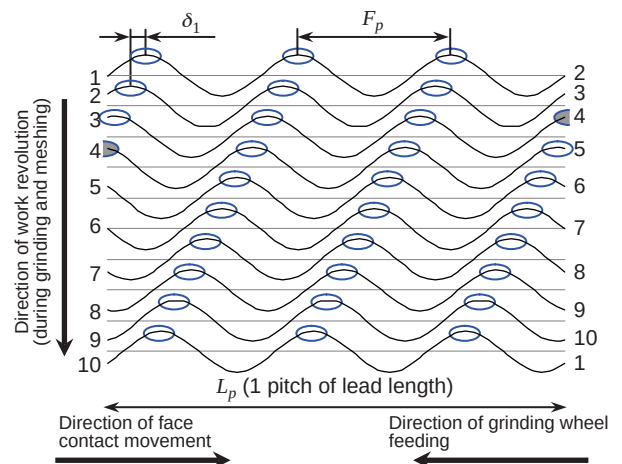


Fig. 8 Mechanism of ghost order generation by feed pitch undulation cycle

undulations shifts by δ_1 on each neighboring tooth. Figure 9 shows the results of an FFT analysis of data connected in the same direction as the movement of the contact line. A peak can be observed at the 31st ghost order frequency. This peak occurs just one order frequency from the 30th order, the triple order of the meshing frequency, because δ_1 , the phase difference between neighboring teeth, overlaps to the extent of the integral of the number of teeth, and becomes equivalent to the wavelength F_p . In the example shown in Fig. 8, the feed direction of the grinding wheel is opposite direction of the contact line movement. Because of this, as each tooth meshes, each phase difference δ_1 builds up, and the undulation for a wavelength increases per rotation of the work. By contrast, under machining conditions in which the direction of meshing and the feed direction of the grinding wheel are the same, the undulation for a wavelength decreases per rotation of the work. Given this, the ghost order frequency produced by the grinding wheel feed pitch undulations, G_1 , can be expressed by Eq. (2).

$$G_1 = Z_w \cdot N_1 \pm 1 \quad (2)$$

$$N_1: \text{Integer} \doteq \frac{L_p}{F_p}$$

Regarding the plus/minus sign in the equation, the sign will be minus when the direction of meshing and the direction of grinding wheel feed are the same, and will be plus when the directions are not the same. When the

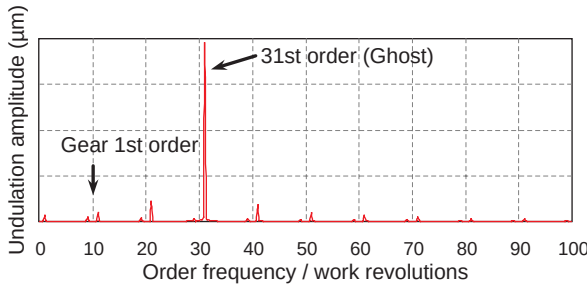


Fig. 9 Spectra of simulated ghost undulation (with Table 1 conditions)

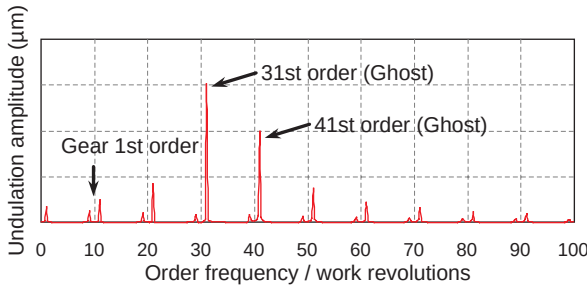


Fig. 10 Spectra of simulated ghost undulation (with Table 1 conditions when $F_p = 0.857$)

direction of rotation of the work during machining is opposite that of the work during meshing, the direction in which the phase difference δ_1 is produced is reversed, and the plus and minus signs will therefore be reversed. Figure 10 shows the results of an FFT analysis of simulated 3.5-cycle-per-pitch undulations (as sine waves) under the conditions shown in Table 1, and with $F_p=0.857$ (mm/work rev). In this case, the integer N_1 in Eq. (2) is 3 or 4, and there are amplitude peaks at the 31st and 41st orders. N_1 is determined by the grinding wheel feed pitch, F_p , and the plus or minus sign of the second right-hand term is determined by the direction of the grinding wheel feed and direction of rotation of the work. As a result, the ghost orders vary with machining conditions.

The undulation level h shown in Fig. 7 can be calculated approximately from the geometric form of the grinding wheel acting on the work, and the grinding wheel feed pitch. Figure 11 shows the shape of the spiral surface on the grinding wheel approximated as a disk. With the grinding point on the work as the point of origin, the x axis shows the direction of the lead, the y axis shows the direction of the disk axis, and the h axis shows the direction of tooth surface normal. A conic plane is formed

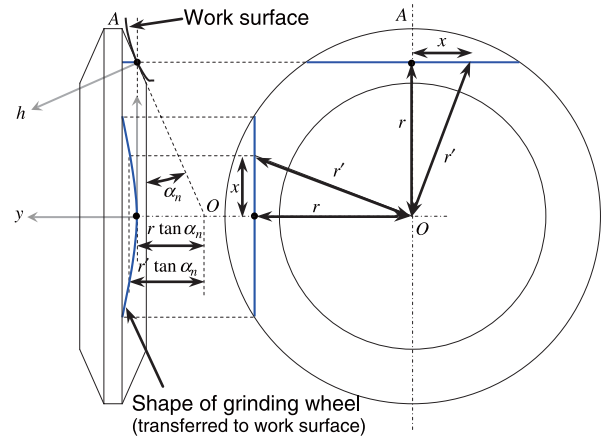


Fig. 11 Shape of grinding wheel transferred to lead shape on work

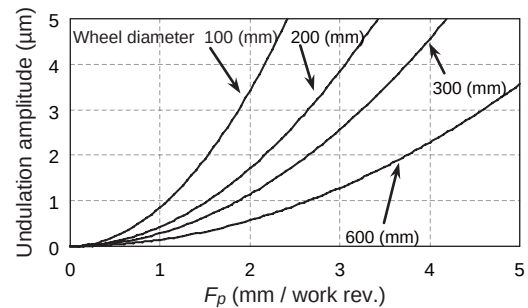


Fig. 12 Relation between F_p and undulation amplitude ($\alpha_n=20\text{deg}$)

with point O as the apex and OA as the generatrix. By projecting a cross-section of plane xy onto plane xh , the shape of the undulations transferred onto the lead form can be obtained. The following equation can then be obtained from the geometric relationship.

$$x^2 + r^2 = r'^2, \quad y = (r' - r) \cdot \tan \alpha_n, \quad h = y \cdot \cos \alpha_n$$

Reordering the simultaneous equations for h gives Eq. (3). (Because $h \ll r \sin \alpha_n$, an approximation of $(r \sin \alpha_n + h)^2 \doteq (r \sin \alpha_n)^2 + 2rh \sin \alpha_n$ is employed).

$$h = \frac{x^2 \cdot \sin \alpha_n}{2r} \quad (3)$$

With F_p representing the pitch feed, the undulation level, h , is obtained by substituting $x = F_p/2$ for Eq. (3). Figure 12 shows examples of quantitative calculations of undulation level. Undulation level h increases in proportion to the square of the feed pitch F_p , and in inverse proportion to the diameter of the grinding wheel.

3.2. Mechanism of Occurrence of Ghost Undulations by Errors in Multiple-thread Grinding Wheels

An example of a single-thread wheel is shown at the left of Fig. 13, and a double-thread wheel is shown at the right. The lead angle of the thread is larger on the double-thread wheel than on the single-thread wheel. Because of this, when grinding at the same wheel rotation speed, the rotation of the work is twice as fast for the double-thread wheel, increasing production efficiency. This type of multiple-thread grinding wheel is today extensively used in the grinding of gear flanks.

It is normal to set a number of tool threads that does

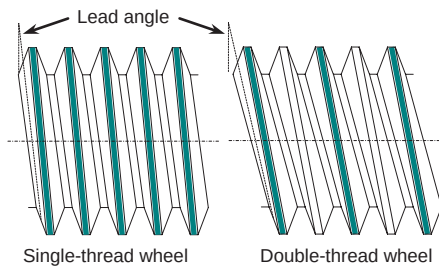


Fig. 13 Single-thread wheel and multiple-thread wheel

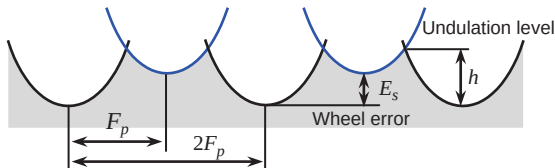


Fig. 14 Undulation of lead form generated by grinding with double-thread wheel

not represent a divisor of the number of teeth in the work. When this is the case, a different thread surface on the grinding wheel grinds the work tooth surface each rotation of the work. If there is pitch error or offset error in the thread surfaces of grinding wheel, the grinding depth varies with each rotation of the work, so undulations are produced. Figure 14 shows the relationship between grinding wheel error, E_s , and the level of undulation, h , when a double-thread grinding wheel is employed.

In order to predict the ghost order frequencies per rotation of the work, it is necessary to establish patterns based on determination of how many threads have left grinding marks on the finished work surface. These

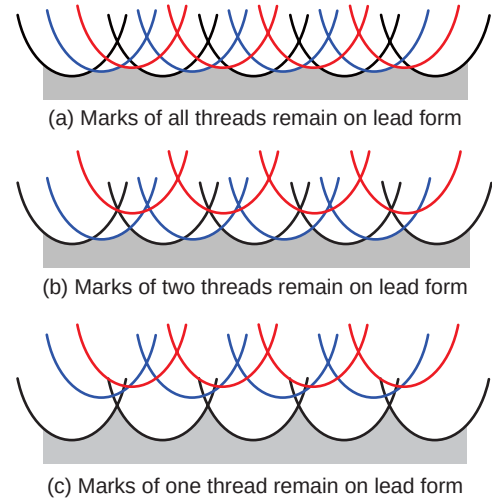


Fig. 15 Ghost undulation patterns in case of grinding with triple-thread wheel

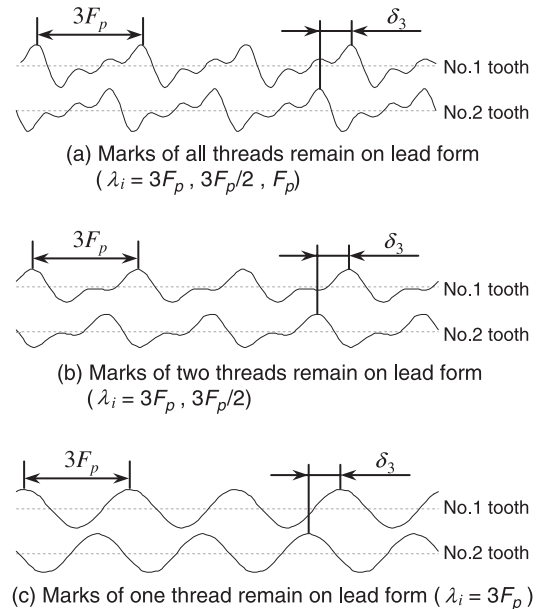


Fig. 16 Simulated ghost undulation patterns in case of grinding with triple-thread wheel

Table 2 Value of N_m

	$Z_j = 2$	$Z_j = 3$	$Z_j = 4$	$Z_j = 5$
$m = 1$	1	2	3	4
$m = 2$	–	1	–	2
$m = 3$	–	–	1	3
$m = 4$	–	–	–	1

Table 3 Relation between ghost undulation pattern and ghost order number in case of grinding with triple-thread wheel

Undulation pattern	Wavelength of synthetic wave	Ghost order frequency
Fig. 16(a)	$F_p, 3F_p/2, 3F_p$	G_1, G_{32}, G_{31}
Fig. 16(b)	$3F_p/2, 3F_p$	G_{32}, G_{31}
Fig. 16(c)	$3F_p$	G_{31}

patterns exist for the number of threads on the grinding wheel, Z_j . Figure 15 shows ghost undulation patterns that might be produced when $Z_j=3$. The waveforms of these ghost undulation patterns can be expressed by Eq. (4), where x represents the position of the lead form.

$$h = \sum_{i=1}^{Z_j} a_i \cdot \sin\left(\frac{2\pi}{\lambda_i} \cdot x\right), \quad \lambda_i = \frac{Z_j \cdot F_p}{i} \quad (4)$$

Differences in the patterns of the ghost undulations can be expressed by changing the value of a_i . Figure 16(a) shows a pattern produced under the conditions $Z_j=3$ and $a_1=a_2=a_3$. In this pattern, there are errors in all three threads, and all the threads leave grinding marks on the finished work surface. Figure 16(b) shows a pattern produced when $Z_j=3$ and $a_1=a_2 \neq 0, a_3=0$. In this pattern, one thread has a larger error than the other two threads, and two threads leave grinding marks on the finished work surface. In Fig. 16(c), $Z_j=3$ and $a_1 \neq 0, a_2=a_3=0$. In this pattern, one thread leaves grinding marks on the finished work surface.

Because the sequence of the threads grinding the teeth changes for each rotation of the work due to the combination of the number of teeth in the work, Z_w , and

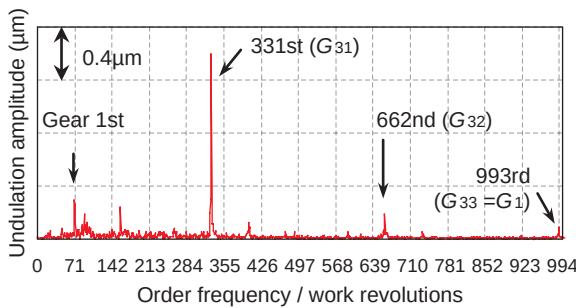


Fig. 17 Example of ghost order frequency in case of grinding with triple-thread wheel

the number of threads on the tool, Z_j , the phase difference between ghost undulations on neighboring teeth is determined by Z_w and Z_j , in addition to the feed pitch, F_p . The relationship between Z_w and Z_j can be expressed as Eq. (5), using the natural numbers n and m .

$$Z_w = n \cdot Z_j + m \quad (5)$$

The phase difference between the ghost undulations on neighboring teeth δ_{Zj} , can be found using Eq. (6), substituting values for the integer N_m determined from the values of Z_j and m as shown in Table 2.

$$\begin{aligned} \delta_{Zj} &= \frac{F_p}{Z_w} + N_m \cdot F_p \\ &= \delta_1 + N_m \cdot F_p \end{aligned} \quad (6)$$

Using Eqs. (4) and (6), the waveform of the ghost undulations on the k^{th} tooth, h_k , can be found from Eq. (7).

$$h_k = \sum_{i=1}^{Z_j} a_i \cdot \sin\left(\frac{2\pi \cdot i}{Z_j \cdot F_p} \cdot (x \mp k \cdot \delta_{Zj})\right) \quad (7)$$

The plus or minus sign in the equations that determine the phase is determined by the direction of rotation of the work and the direction of feed during grinding. The ghost order frequencies can be specified by the FFT analysis for the waveform which connected and united the waveform in Eq. (7) of all the teeth. The ghost order frequency produced when $a_i \neq 0$, G_{ji} , can be found from Eq. (8).

$$\begin{aligned} G_{ji} &= \frac{N_{ji} \cdot Z_w \pm i}{Z_j} \\ N_{ji} &= Z_j \cdot N \pm i \cdot N_m \div \frac{i \cdot L_p}{F_p} \end{aligned} \quad (8)$$

(N : Integer)

Table 3 shows the relationship between the ghost undulation patterns produced by a triple-thread wheel shown in Fig. 16 and the ghost order frequency. The FFT analysis for all teeth that was shown in Fig. 5(a) in chapter 2 was conducted on a gear ground using a triple-thread wheel. For the ghost undulation shown in Fig. 5(a), the 993rd-order ghost frequency corresponds to G_{33} ($=G_1$), the 662nd-order ghost frequency corresponds to G_{32} , and the 331st-order ghost frequency corresponds to G_{31} . Of these, the amplitudes of the 331st-order (G_{31}) and 662nd-order (G_{32}) are dominant, and it can be conjectured that the grinding during machining was close to the pattern shown in Fig. 16(b). Figure 17 shows the results of an FFT analysis for all teeth of the reverse faces of the teeth

of the same gear as shown in Fig. 5(a). In Fig. 17, the 331st order (G_{31}) is dominant compared to the 662nd order (G_{32}) and 993rd order (G_{33}), and it can be conjectured that the ghost undulations follow a pattern of the type shown in Fig. 16(c).

The ghost undulation level is highest when only one thread leaves grinding marks on the finished work surface, as in Fig. 15(c). Denominating the wheel diameter D and substituting $x = Z_j \cdot F_p / 2$ in Eq. (3), the maximum ghost undulation level, $h_{\max}$, can be found from Eq. (9).

$$h_{\max} = \frac{Z_j^2 \cdot F_p^2 \cdot \sin \alpha_n}{4D} \quad (9)$$

4. Multiple-thread Hobs

In addition to the grinding wheels discussed up to this point, the hobs used in the hobbing process are another type of multiple-thread tool. The mechanism of production of ghost undulations during hobbing with a multiple-thread hob is the same as that of the grinding wheels discussed in this paper. Figure 18 shows the results of an FFT analysis of lead form measurement data for all teeth of a 44-tooth work hobbed using a five-thread hob. The ghost undulations produced are the type determined from Eq. (8) as G_{ji} . This demonstrates that the order frequencies of ghost undulations on the lead forms can be accurately predicted, even in the case of hobbing using a five-thread hob.

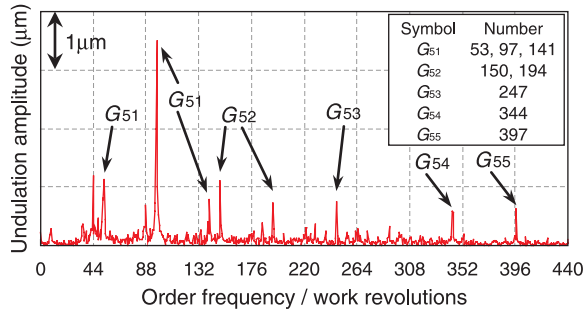


Fig. 18 Example of ghost order frequency in case of hobbing with five-thread hob

5. Measures to Ameliorate Ghost Undulations Produced by Multiple-thread Grinding Wheels

As indicated above, the ghost undulation level that occur on the lead forms of gear teeth during grinding using multiple-thread grinding wheels is affected by the feed pitch and the precision level of the grinding wheel. Figure 19 shows the undulation level reaches a maximum when there is a large degree of error in a double-thread

wheel. Figure 20 indicates that the amplitude of the undulations could be reduced by reducing the feed pitch. The effectiveness of reducing the feed pitch in ameliorating ghost undulations and ghost noises was therefore examined.

Figure 21(a) shows the results of an FFT analysis for all gear teeth machined at a feed pitch F_p , which produced a 310th-order ghost noise at the benchmark work rotation, and two reduced feed pitches, $F_p \times 0.77$ and $F_p \times 0.62$.

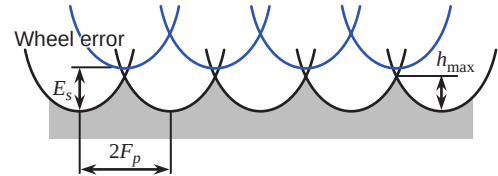


Fig. 19 Ghost undulation when marks of one thread remain on lead form ground by double-thread wheel

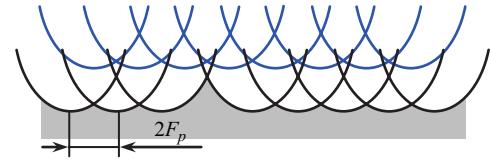
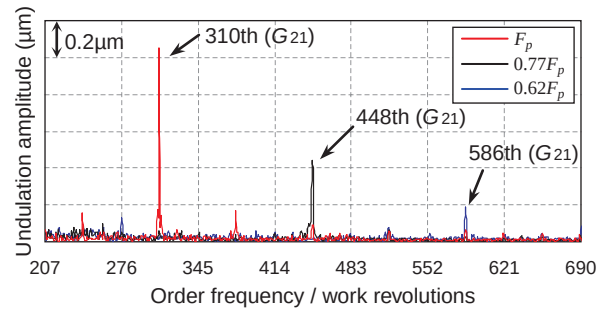
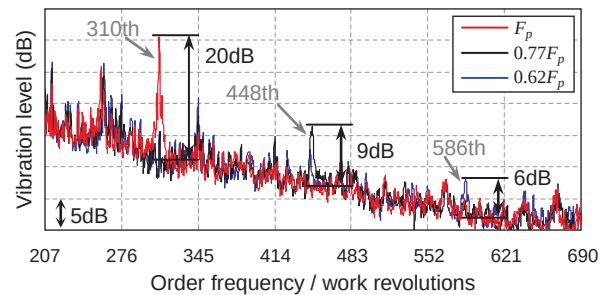


Fig. 20 Enhancement of undulation by reduction of feed pitch



(a) Spectra of FFT analysis for measured lead forms in case of changing feed pitch of grinding wheel



(b) Spectra of transmission case vibration occurring when changing feed pitch of grinding wheel

Fig. 21 Experimental results for reduction of ghost noise by reduction of wheel feed pitch

With the reduction of the feed pitch, the ghost order frequencies change to the 448th order (G_{21}) and 586th-order (G_{21}), and the amplitude of the ghost undulations is reduced. The ghost order frequencies change due to changes in the value of N_{ji} in Eq. (8). Figure 21(b) shows a comparison of transmission case vibration spectra measured when these gears were employed in a transmission. As a measurement condition, the frequencies at which ghost noises were produced were adjusted taking into consideration the frequency characteristic of the transmission case. The reduction of feed pitch reduced the vibration peak level of ghost orders against the background vibration level. This demonstrates the effectiveness of the method in reducing ghost noises even in an actual transmission.

6. Conclusion

The research discussed in this paper examined the mechanism of occurrence of undulations on tooth flank, which are a factor in the occurrence of ghost noises, during grinding the gears using a thread-type wheel. The following results were obtained:

- (1) The order frequency of ghost undulations was found to be determined by the number of threads on the grinding wheel, the feed pitch, level of precision and feed direction of the grinding wheel, and the direction of rotation of the work; the principles of determination of the order frequencies were formulated as equations.
- (2) The order frequency of ghost undulations predicted by the formulated equations corresponded to that of ghost undulations produced on gear teeth during actual machining.
- (3) The reduction of the feed pitch of the grinding wheel was shown to be effective in reducing undulations on the surfaces of gear teeth and ameliorating ghost noise in an actual transmission.

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■ Author ■



Ryo EGAMI



Tadami KONDO



Motoki IKUMA

Development of Front Pillar for Visibility Enhancement

Yoshiharu HASEGAWA*

Hiroshi FUJITA*

Takeharu ENDO*

Masaaki FUJIMOTO*

Junya TANABE**

Masaki YOSHIDA***

ABSTRACT

In order to reduce blind spots during driving, a front pillar has been developed that enhances the driver's diagonal forward visibility.

Physiological and psychological studies conducted internally indicated the existence of a more favorable front pillar width that enables essentially a continuous field of vision between the front window and the side windows, and which would not induce a sense of visual incongruity to the driver.

A new pillar structure was developed in order to realize a target pillar width, and at the same time maintain vehicle performance. The new structure uses ultra-high-strength hydroformed steel tubes as the main framework. Indirect spot welding and arc laser hybrid welding are used to mount the new structure in the body framework.

1. Introduction

It is said that approximately 90% of the external information that a driver needs to drive a car is taken in visually. It is therefore clear that visibility performance is an important element in controlling a vehicle. A blind spot produced by the front pillar is one factor that impedes forward visibility during normal driving. The reduction of this blind spot could be expected to be effective in enhancing visibility.

The reduction of the size of the blind spot to less than the human interocular distance would reduce it to zero, and the development of a pillar structure to realize this has been reported⁽¹⁾. The research discussed in this paper newly focused on understanding visibility performance from a dynamic perspective, and determined the physiological and psychological effect of the size of physical blind spots on the percipient when moving objects are perceived visually. In addition, the project determined a more favorable pillar width, and developed a body structure and production technology that would enable a pillar of this width to be used without adversely affecting the rigidity, strength and performance features of automobiles.

2. Study on Pillar Visibility from a Dynamic Perspective and Establishment of Appropriate Pillar Width

2.1. Concept of Visibility Involving Front Pillar

The front pillar sometimes acts as an impediment when a driver is attempting to follow an object with their eyes.

However, during driving, visual information is in constant motion, and it can be conjectured that if the thickness of the pillar was decreased to a specific level or below, the exterior scene flowing between the front window and side window would be experienced by the driver as seamless. That is, even when, in physical terms, the visual field was discontinuous, the driver would be continuously aware of objects of visual perception without the pillar acting as an impediment. Studying pillar width from this perspective would enable a more favorable pillar width for visibility performance to be determined.

2.2. Establishment of Driving Scenarios

In order to study pillar visibility, it was first necessary to establish driving scenarios. Turning left or right at an intersection and cornering can be projected as typical scenarios in which the front pillar affects visibility.

* Automobile R&D Center

** Honda Engineering Co., Ltd.

*** MARUJUN CO., LTD.

Observation of drivers in these scenarios shows that when turning at an intersection, there is rapid and extensive movement of the driver's line of sight around the front pillar as they attempt to visually distinguish a variety of obstacles. In addition, as the vehicle turns extensively, the front pillar moves across the object of visual perception in a short period of time.

The movement of the driver's line of sight in relation to the pillar is therefore rapid, and the impediment of visibility represented by the pillar is instantaneous.

By contrast, during cornering, the driver's line of sight is stable in order to focus on a point of the curve, and the objects of visual perception are in the distance. Because of this, the movement of the line of sight is slow, and the impediment of visibility represented by the front pillar is continuous.

Studying the field of vision during cornering and conducting a performance evaluation would therefore represent a valid method of establishing target values. Curves are constructed with various curvature radii, and are designed based on the magnitude of the radius and the road structure ordinance⁽²⁾ that stipulates vehicle speed in relation to the curvature radius.

Test results showed that during normal driving, the driver's line of sight is focused on a point 3 seconds ahead in the direction of motion, to enable prediction of the situation in front of the vehicle. Superimposition of the driver's line of sight over curves constructed on the basis of road structure ordinances indicated that an R30 curve represented the most severe condition for drivers, with the front pillar sitting in the line of sight and becoming an impediment. Studies were therefore conducted on the basis of a scenario of a vehicle traveling around an R30 curve at 30 km/h, the speed established for a curve of this radius by road structure ordinances, while the driver watched a forward obstacle. Figure 1 shows the driver's line of sight when turning different curves.

2.3. Method of Study

Visual tracking tests using a test stand were conducted as a method of studying driver's continuous visibility when turning.

First, test equipment was prepared consisting of a test stand modeled on a real vehicle on which the width of the front pillars could be changed, with a large screen in front. Figure 2 shows the test equipment. A point modeling the movement of an obstacle at a point 3 seconds ahead of a vehicle traveling around a simulated R30 curve at 30km/h was projected on the screen. Test subjects sat in the test stand and followed the point projected on the screen with their eyes, while the movement of their line of sight at different front pillar widths was monitored using an eye mark recorder. Test results for different front pillar widths were compared.

Next, a subjective evaluation was conducted using the same equipment. An image of front view when cornering

an R30 curve at 30 km/h was projected on the screen, and test subjects evaluated the feeling of the visual image over a pillar using a point system. The rated evaluation items for different pillar widths are the level of visibility and presence or absence of odd feelings.

2.4. Test Results

2.4.1. Ocular tracking characteristics

Figure 3(a) shows an example of results of measurements of the changes of the test subjects' line of sight when they were demonstrating smooth pursuit eye movement following the point on the screen, as measured using an eye mark recorder.

The vertical axis shows the various pillar widths, while the horizontal axis shows horizontal location where the object of visual perception was moving on the screen. Circles show the positions at which the test subjects' eye movements stopped; the size of the circles indicates the length of time for which eye movements were stopped. The shaded part shows an area in which the pillar represented a visual obstacle. These results indicate that the subjects' eye movements stopped even outside the area in which the pillar represented a visual obstacle. That is, even when the lines of sight of the test subjects appeared

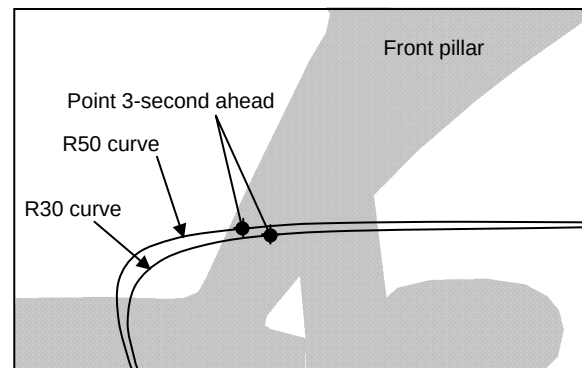


Fig. 1 Driver's visual field on driver side curve

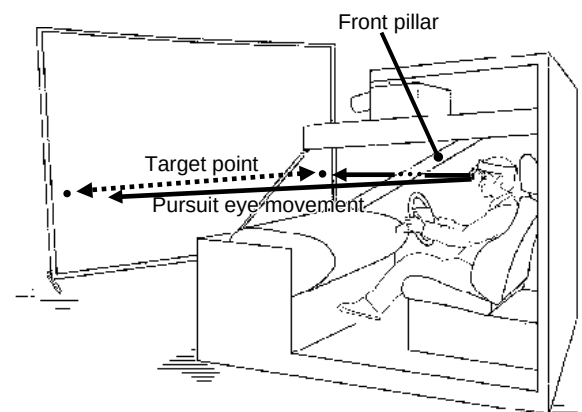


Fig. 2 Test equipment

to be moving smoothly following the point in the absence of a visual obstacle, the data showed that slight stops in eye movement occurred as the point was being followed. This is believed to be a physiological phenomenon of human visual perception that occurs during smooth pursuit eye movement. It was judged that when a visual obstacle corresponding to a pillar is part of the visual field, when the point being tracked visually crosses the pillar, the line of sight shifts to the point at which the point is expected to reappear, and the eyes stop, waiting for the reappearance of the point. The project therefore focused on the change in the length of time for which the eyes stopped moving in the area in which the pillar represented a visual obstruction as the width of the pillar was increased.

First, the length of time for which the eyes stopped moving outside the area in which the pillar represented an obstruction was studied, after which lengths of time for which the eyes stopped that were longer than these lengths of time were isolated from the data. Figure 3(b) shows

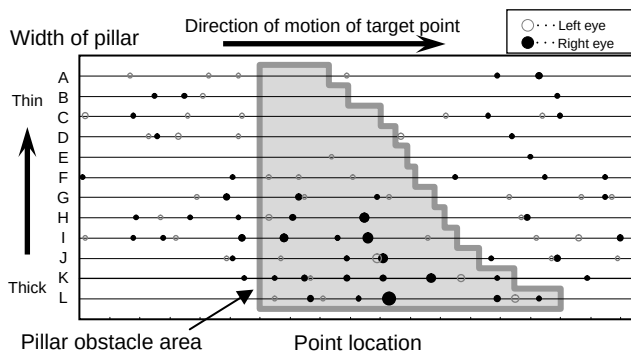


Fig. 3 (a) Points at which eye movement stops

points, isolated from the data in Fig. 3(a), at which the eyes stop for an extended period.

The results showed that the points at which the length of time for which the eyes stopped was extended almost all occurred within the area in which the pillar represented an obstacle, and in addition, that when the pillar width was narrowed beyond a specific width, no points at which the eyes stopped moving remained. Based on this, it was judged that there is no longer any effect from the pillar as a visual obstacle, and that pillars narrower than a specific width would enable an object of visual perception to be tracked continuously, without breaks.

2.4.2. Subjective evaluation

Test subjects' evaluation of their feeling of the visual image became better as the pillar became narrower, but evaluations of the feeling of visibility peaked at specific pillar widths, and the points given declined as the pillar became narrower than these specific widths. Figure 4 shows the subjective characteristics. Subjects commented

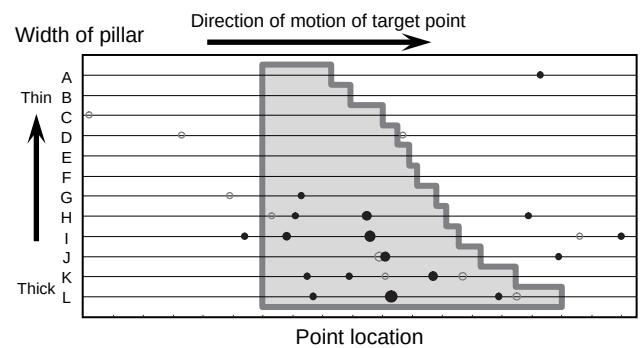


Fig. 3 (b) Points at which eye movement stops

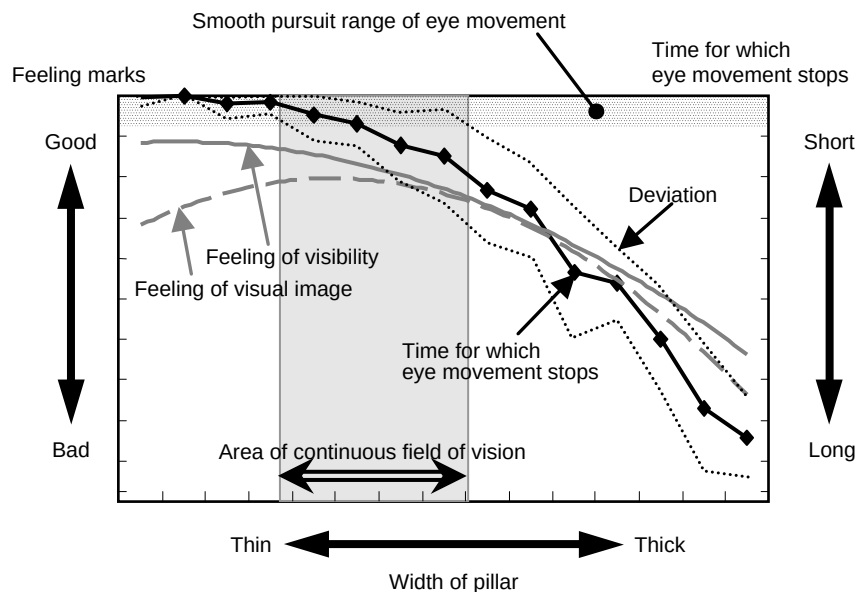


Fig. 4 Eye movement characteristic and subjective rating

that when the pillar became too thin, they felt a sense of oddness, and that the sense of the pillar appearing and disappearing made them uneasy. When a subject watches a distant screen when the width of the pillar is less than the width of the eyes, either the left or right eye can see the image on the screen, and in theoretical terms the blind spot is zero. It is thought that under these conditions, one eye can see the pillar, while the other eye cannot, producing a state of binocular rivalry⁽³⁾, which occurs when the eyes present differing visual information. From these results, it may be concluded that there is a more favorable pillar width.

2.5. Setting Pillar Width Targets

The study of ocular tracking characteristics, a human physiological phenomenon, and subjective characteristics, a human psychological phenomenon, showed that there was a more favorable pillar width at which the eye was able to track an object of visual perception continuously, and at which there was little sense of incongruity in the field of vision. Figure 4 shows ocular tracking characteristics and subjective evaluations. The range for ocular tracking characteristics includes variations in the periods of time for which the movement of the eyes have stopped within the area of the pillar. The time for which movement is stopped decreases as the width of the pillar decreases, until it becomes less than the length of time that the eyes stop movement during smooth pursuit movement. If the subjective characteristic discussed above is superimposed over this, it can be seen that the length of time that the eyes stop for the pillar width, close to the peak of the subjective characteristic, is also less than the length of time that the eyes stop during smooth pursuit movement. From these results, this pillar width was adopted in this project as the target width in order to realize continuous visibility.

3. Basic Structure of Front Pillar for Visibility Enhancement

3.1. Analysis of Main Factors in Pillar Width in Conventional Structure

The target established in the preceding section was considerably narrower than the conventional front pillar and could not be achieved using the conventional pillar structure. Therefore, it was judged that it would be necessary to consider a new structure. First, the conventional structure was reexamined from the perspective of reducing the width of the pillar. The cross-section of the normal orientation of a conventional front pillar shown in Fig. 5 indicates that the width of the pillar in relation to the direction of the line of sight is determined by three elements: the windshield adhesive area, the door sash area, and the closed section area between them. Reducing the width of these elements would reduce the width of the front pillar. However, the

windshield adhesive area and the door sash itself are basic structural sections that are necessary to the existence of surrounding parts, and they cannot be removed. In addition, they are parts that are already minimized to some extent. Given this, it was clear that the key to narrowing the front pillar was moving the windshield adhesive area and the door sash closer together in the remaining area.

3.2. Method of Narrowing Front Pillar Cross-section

Based on the results of the study conducted in the preceding section, the following two methods were judged as being effective in reducing the width of the front pillar: First, ultra-high strength tubes would be used to reduce the width of the closed section area, while maintaining the achievement of a high level of strength.

In addition, as a method of moving the door sash and the windshield adhesive area closer together, the flange that represents one element determining the width of the front pillar would be removed, and the windshield adhesive area would be situated directly on top of the closed section area. Using these methods, the goal of the project was to balance the achievement of the target reduction in the sectional width with the achievement of strength and rigidity.

3.3. Details of Formation of Section Using Hydroformed Tubes

Figure 6 shows an outline of the front pillar section proposed above, in which the windshield adhesive area is moved directly on top of the closed section area and the closed section area is formed using hydroformed tubes. The section is discussed in detail below.

First, the pillar width in the direction of the line of sight was reduced by removing the flange from the windshield adhesive area, and connecting the outer panel directly to the closed section area. Indirect spot welding,

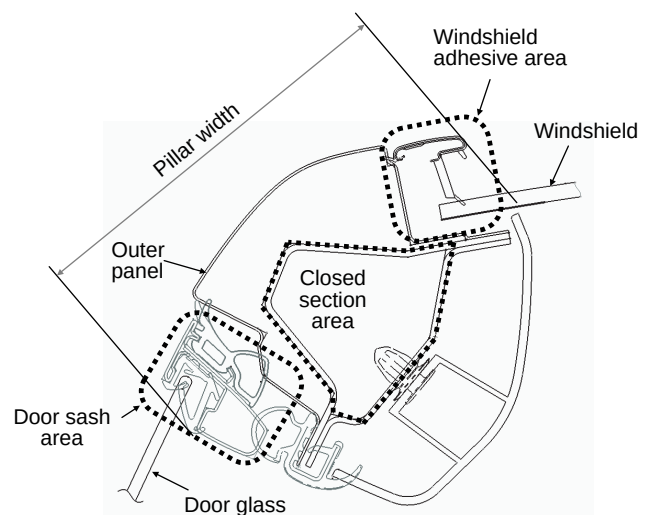


Fig. 5 Conventional section of front pillar

a single-sided access welding technique, was used to connect the two. Indirect spot welding will be discussed in detail later in this paper. Reinforcing structures were employed to enhance the performance of the section, and arc laser hybrid welding was used to connect these to the closed section area. This is also a single-sided access welding technique, and will be discussed in more detail later in the paper. All the components to be mounted in the cabin were fitted in an area in which they would have no effect on the width of the direction of the line of sight. Settings were also reexamined to further minimize the door sash and the windshield adhesive area. The hydroformed tube was used to form the basic section as a quadrangular structure. A simple sectional shape is maintained around the entire general area of the pillar by allocating each of the different functions – the outer connection face, the earth face for the single-sided spot weld, the mounting face for the mounted parts, and the seam welded section of the tube – to different sides of the quadrangle. This helps prevent sudden changes in the section, and increases formability while maintaining impact strength.

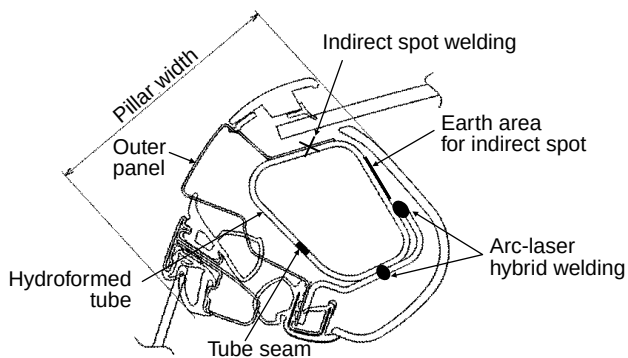


Fig. 6 Proposed section for increased visibility

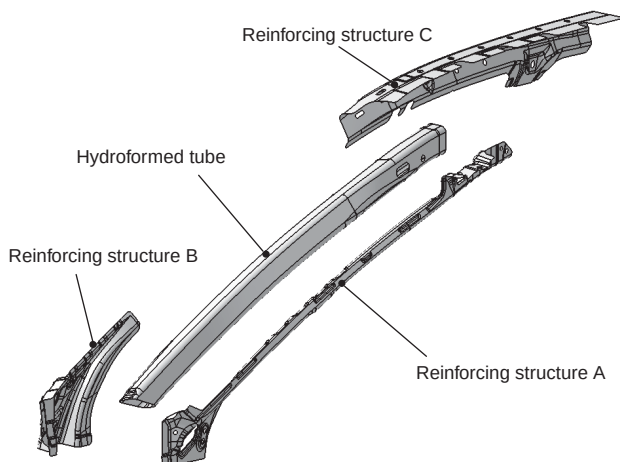


Fig. 7 Structure of front pillar using hydroformed tube

3.4. Structure of Front Pillar Using Hydroformed Tubes

Figure 7 shows a schematic of the structural parts, which will be discussed in detail in this section. The hydroformed tube has a straight form.

Given that the reduction of the cross-sectional area would reduce the load capacity of the hydroformed tubes and thus increase the degree of deformation of the pillar in a front collision, this structure helps ensure that the input during a collision is received by the section as much as possible in an axial direction rather than as bending force in order to reduce the deformation of the pillar. It also results in the simplification of the shape and a consequent increase in productivity. The face of the pillar inside the vehicle body is subject to compressive stress due to bending in a front collision. To help ensure strength during a front collision, reinforcing structure A is attached to the entire inner face of the pillar as an efficient means of increasing strength. To help ensure effective transmission of input during a front collision in the axial direction of the hydroformed tube and produce a sectional area equivalent to that of the conventional section at the same time, reinforcing structures B and C are attached to the surrounding body framework parts, resulting in the achievement of an equivalent level of strength and rigidity in the body as a whole as a conventional model, thus satisfying body performance specifications. The ability to easily optimize structural parts, the materials from which they are formed, and their thickness, in order to reinforce the uniform (hydroformed) closed section area parts in the appropriate places with the appropriate structures can be pointed to as one of the merits of hydroformed structures. Highly specialized single-sided access welding techniques were used to connect the hydroformed parts and the reinforcing structures. The parts in Fig. 7 are used to form the framework of the pillar, and single-sided access welding employing arc laser hybrid welding was used for connections within the complete front pillar, minimizing the need for special equipment and procedures.

In addition, in achieving the connection between the completed front pillar and the surrounding body structural parts, mating flanges were used with reinforcing structures to enable direct spot welding to be employed, except where indirect spot welding to the outer panel was employed.

The use of equipment that enables procedures to be conducted using standard facilities, and the accuracy that can be guaranteed using conventional management techniques, makes this a structure that can be considered for mass production.

4. Formation Techniques

4.1. Hydroformation

4.1.1. 980 MPa-class steel tubes

To form the structures described in Sections 3.2 to 3.4, hydroformable 980 MPa steel tubes were developed. The

basic concepts underlying this development will be discussed in this section.

Dual phase steel was selected for the steel sheet, to help ensure that the tube remained formable following the machining history applied in the production process, i.e., to help ensure surface cracks are prevented from forming in the corner radius section and produce stretchability during hydroformation, in addition to producing a weldable, corrosion-resistant, and more available hydroformed tube.

In setting the characteristics of the tubes, values were chosen that enabled the achievement of a balance between strength and formability. The lower limit for strength was determined by the strength of the part. It was necessary to set the upper limit for strength with consideration of the considerable effect of the bending resistance of the material on stretchability during hydroformation. Figure 8 shows the relationship between deformation resistance and stretchability. As the deformation resistance of the material increases, the corner radius that can be formed also increases. Therefore, the strength range was set between the upper limit strength at which the corner radius needed for the pillar could be formed, and the lower limit strength needed to help ensure the strength of the part.

4.1.2. Hydroformation

Because the deformation resistance of the material was high, high pressures had to be applied during hydroformation.

Preforming and pre-bending procedures are normally conducted prior to hydroformation, but in this project, modifying the press direction in the clamping process of hydroformation and performing section squeezing at the process enabled the preforming procedure to be done away with. The pre-bending procedure uses a press bending process that does not employ standard bending equipment such as rotary draw bending machines, thus reducing the initial investment for equipment.

Modifications have also been made to the hydropiercing procedure used to cut holes in the material during hydroformation, with the main purpose of enabling

accurate holes to be cut to function as position references in the following procedures. Reducing press droop around hole edges and handling scrap material represent issues in the hydropiercing procedure.

If the droop around the hole edges is large, the holes will not function as position references. The relationship between variables such as hydro pressure and hole diameter and the degree of drooping was therefore studied, and the hydropiercing conditions were optimized to enable the holes to function as position references. As an example of the results of these studies, Fig. 9 shows the relationship between hydro pressure and degree of drooping.

It was predicted that scrap material from the piercing might remain in the front pillar cross-section or settle in the die, thus reducing production efficiency and potentially resulting in production mishaps and the like. In order to assist in avoiding these issues, a method was developed in which the hole is not completely punched through, leaving the scrap attached to the piece being produced (Fig. 10). After the hole has been used as a position reference in later procedures, a laser is used to cut a complete circle around the hole (the dotted line in Fig. 10), removing the scrap material from the finished product.

In addition to enabling the pillar to be narrowed, as discussed in Section 3.2., hydroformation of the part in this manner has a variety of other benefits.

The first of these is that it enables a sufficient degree of freedom in the design of the front pillar, because it

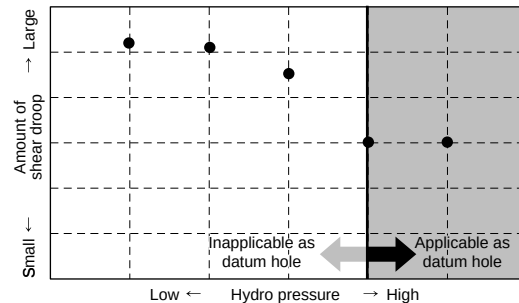


Fig. 9 Relation between hydro pressure and amount of shear droop

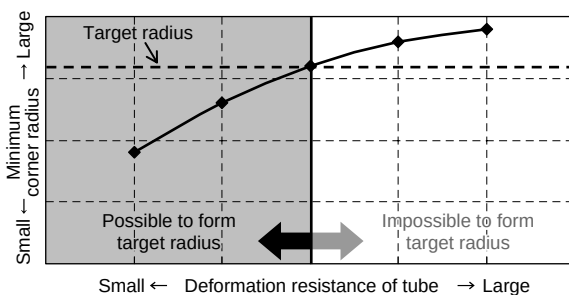


Fig. 8 Relation between deformation resistance of tube and corner radius

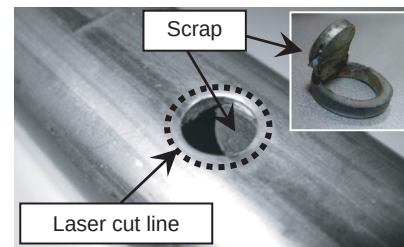


Fig. 10 Shape of hydro pierce

enables continuous variation of the sectional shape. Figure 11 shows representative sectional shapes of the front pillar developed in this project.

In addition, the rigidity of the formed component, resulting from the use of the closed section, helps to ensure good form stability with minimal spring-back. Figure 12 shows the points at which dimensional accuracy in the new front pillar was measured, and Fig. 13 shows the results of measurements of dimensional accuracy for the prototype in the initial prototype stage. These results indicate that a high level of dimensional accuracy was obtained from the initial prototype stage.

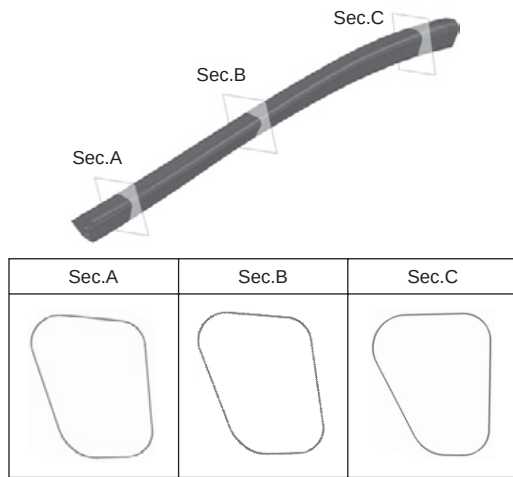


Fig. 11 Cross - section of hydroformed front pillar

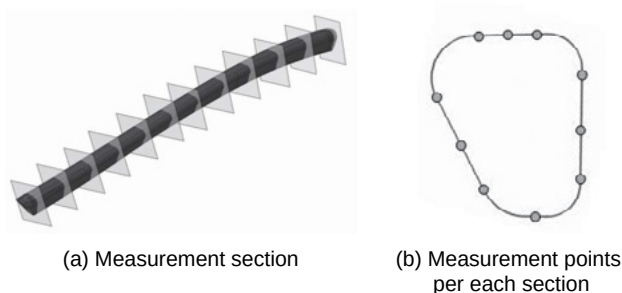


Fig. 12 Measurement positions of dimensional accuracy

Dimensional accuracy (Difference from target dimension)	Frequency
0.0-0.5 mm	80%
0.6-1.0 mm	11%
1.1-1.5 mm	6%
1.6-2.0 mm	3%

Fig. 13 Dimensional accuracy of front pillar at early trial stage

The achievement of a good level of form stability also increases the ease of die quality maturation. In the new front pillar, dimensional accuracy has been increased in the early stages of manufacture, chiefly by adjusting the position of the die cross-section, and this can be indicated as also having the advantage of reducing the time for die quality maturation.

4.2. Welding

Because the hydroformed part is a closed section structure, direct spot welding [Fig. 14(a)], the standard technique used on mass production lines, cannot be used. Instead, a welding technology that can form a connection with single-sided access is needed.

In selecting the welding techniques to be used in the development of the new pillar, the specifications of the materials to be welded and the structure of the mass production line were considered. As a result, indirect spot welding [Fig. 14(b)], which can be incorporated into the existing internal mass production line, was chosen as the welding technique for hydroformed part and thin sheets (the outer panels), and arc laser hybrid welding was applied for welding thick parts (reinforcing structures A-C in Fig. 7) to achieve desired part performance (strength and joint shape).

4.2.1. Indirect spot welding

Indirect spot welding was expected to display high productivity equivalent to that of the existing mass production method, but the fact that the current pathways are longer than in the case of direct spot welding promotes shunt current, and it is challenging to concentrate the current in the area being welded. This makes the determination of a method of controlling shunt current an issue in obtaining a stable welding quality. The following modifications were made to enable indirect spot welding to be conducted at the same level of tooling as the current mass production system:

- Optimization of welding conditions
- Optimization of earth position for control of shunt current
- Optimization of the order of the welds to enable control of shunt current

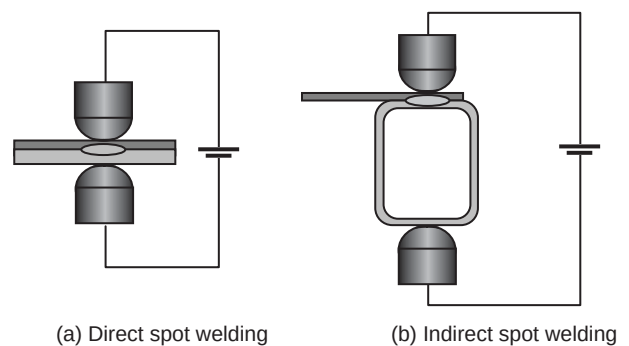


Fig. 14 Schematic design of spot welding

These modifications enabled the achievement of a stable weld quality. In addition, they enabled the maximum use to be made of existing mass production facilities, enabling facilities to be shared without the need for new dedicated welding procedures.

4.2.2. Arc laser hybrid welding

Development was conducted to enable the use of arc laser hybrid welding to produce filler weld joints and lap joints [Figs. 15(a) and (b) respectively], depending on the type of joint that was necessary between the components.

Because the laser can also be used for filler weld joints, a deeper welding bead can be achieved from the initial stages of welding compared to conventional arc welding. Welding can be conducted at three times the conventional speed, and inaccuracies due to heat deformation of the product being welded can be controlled. In addition, the ease of execution of the technique produces a stable weld quality, even if there is a gap between the materials being welded.

As Fig. 16 shows, to produce a lap joint, the laser can be used to cut a keyhole (a deep wedge-shaped groove

for the welding bead) at the same time the hole is being widened and filled using arc welding. As a result, undercutting, which represents an issue when laser welding is used in isolation, can be controlled, even when there is a gap between the materials being welded. In addition, the technique produces a broader welding area than is the case when laser welding is employed in isolation, and helps to ensure stable welding quality. The use of the technique in mass production was focused on and a variety of factors that can result in variations in production (floating of the components, trimline deviations and the like) were considered, enabling welding conditions that would result in stable production to be determined.

These innovations have enabled the application of arc laser hybrid welding, which combines the strengths of arc welding and laser welding, in the manufacture of the newly developed pillar, and have established the conditions for stable production.

5. Conclusion

A more favorable front pillar width for the enhancement of visibility while driving was determined on the basis of ocular tracking characteristics, a physiological phenomenon, and subjective visual evaluation, a psychological phenomenon. The project produced the following outcomes:

- (1) If the width of the front pillar is reduced to a specific width, the pillar will not create any blind spot, and the driver's eyes will essentially be able to continuously follow an object of visual perception.
- (2) The more the width of the pillar was reduced, the better the test subjects' evaluation of their sense of the visual image became; however, if the pillar became too narrow, subjects experienced visual incongruities, indicating the existence of a more favorable pillar width.

That pillar width was narrower than the conventional width; in order to enable its realization, hydroformation and vehicle mounting techniques were developed, with the following results:

- (3) The properties of the steel tubes were set based on considerations of the strength and formability of the part, allowing stable production of hydroformed parts under fixed forming conditions.
- (4) Indirect spot welding and arc laser hybrid welding enabled the hydroformed pillar framework to be mounted in the vehicle, resulting in a sectional structure that reduced the width of the pillar.

The project discussed in this paper was able to determine methods of addressing the antagonism between the reduction of front pillar width for the achievement of a continuous field of vision and the ability to help ensure vehicle performance. The application of the front pillar for enhancing visibility, developed in this project, was enabled by the outstanding fit between the design concept of the new framework structure and the superior

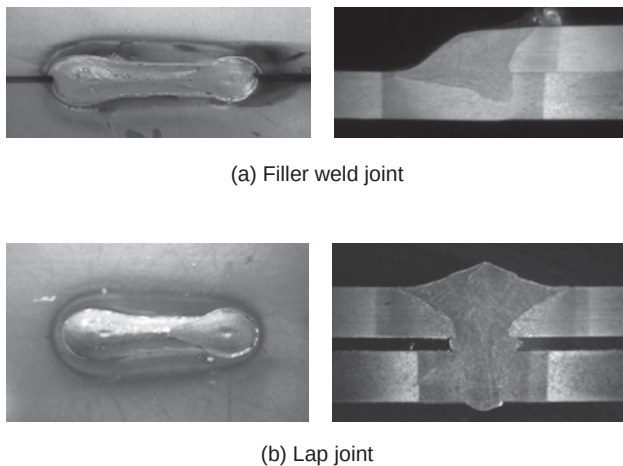


Fig. 15 Weld bead shape

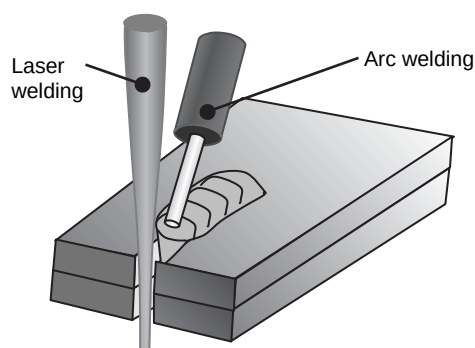


Fig. 16 Schematic design of arc-laser hybrid welding.

performance of hydroformation as a means of producing ultra-high-strength steel tubes, which are generally considered challenging to form.

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■ Author ■



Yoshiharu HASEGAWA



Hiroshi FUJITA



Takeharu ENDO



Masaaki FUJIMOTO



Junya TANABE



Masaki YOSHIDA

Development of Magnetic Stiffener

Masanobu YAGUCHI*

ABSTRACT

A magnetic stiffener was developed to enhance workability and increase the degree of freedom in its range of application. A tentative holding function that does not use release paper and does away with the need for a human-powered press-bonding process is needed to enhance workability. This feature was achieved by selecting non-sticky resin materials, and by mixing ferrite particles into and magnetizing the resin layer. In addition, the optimum magnetic force was ascertained in order to help prevent drop-out in the mass production process, and material design was performed to also simultaneously achieve conventional reinforcement performance.

1. Introduction

Stiffeners are materials that are applied to localized areas, at which the surface stiffness is insufficient with respect to outer parts of an automobile, as represented by the doors, hood, trunk, and the like, to provide necessary stiffness⁽¹⁾. Historically, technology for simulating the surface stiffness of the outer parts had not yet been developed, so detailed correlation between the predicted values and the measured values obtained using an actual vehicle could not be obtained. In addition, when structural measures could not be adopted, stiffeners were used. However, because of the need to lighten the body in recent years, the panel thickness of the parts has become increasingly thinner, and the number of parts where surface stiffness is called for has increased. Consequently, at present, stiffeners are often used from the initial design stage.

As shown in Fig. 1, a general stiffener is of a 3-layer construction consisting of a glass cloth, which has reinforcing performance and strain mitigation ability, a resin layer, which has reinforcing performance, temporary retentivity prior to curing, sticking power subsequent to

curing, and release paper. The stiffener is applied by peeling off the release paper, aligning the stiffener with the target location, bonding it to the mating surface while taking care to ascertain that there is no air inclusion, and then applying pressure to the stiffener with an application device that uses a roller, for example. From the perspective of the current environment and to reduce the working load (ergonomics), it is predicted that eliminating the pressure application, which entails working time, and the release paper to keep the factory clean and reduce garbage, and reducing the panel thickness of outer parts, other than that of the above-mentioned parts and their associated weight reduction, will be promoted in the future, so the quantity of stiffeners is tending to increase each year. Consequently, to reduce the workload, the development of stiffeners that meet the above needs is being demanded.

2. Setting the Development Targets

As shown in Fig. 2, stiffeners are applied at the weld shop where the outer parts are bonded to each other in an automobile manufacturing process, and are cured using a heat source at the paint shop where rust-proofing and painting are applied. A stiffener should of course satisfy the main function of reinforcing performance subsequent to the curing process, and prior to curing it should be in a condition such that it can be readily applied to the bonding surface and temporarily retained. In addition, it should be completely cured at the paint shop without



Fig. 1 Structure of stiffener

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dropping out. Accordingly, in the development of the stiffener, the following two items were set as the development targets.

- (1) To be able to eliminate the release paper.
- (2) The stiffener should not drop out from when it is applied in the welding section to when it is cured and baked in the painting section.

3. Philosophy Concerning Stiffener Design

Figure 3 shows the material formulation of the resin layer of the conventional stiffener. For the basic specifications, it was decided to study a blend based on the conventional resin and curing agent to enable the main function of reinforcing performance to be secured. Particularly, a study of stiffener design that could enable the targets to be achieved by adding functions to the filling agent that is used as an extender was performed.

3.1. Philosophy Concerning the Elimination of the Release Paper

The composition of the resin used in many of the conventional stiffeners consisted of epoxy resin and/or SBR or BR rubber, either as a mixture or a single material⁽²⁾, and full play was given to the stickiness power prior to curing, as well as the adhesiveness and reinforcing performance after curing, by using the physical properties of these materials. In order to eliminate the release paper, it is necessary to reduce the stickiness power, and thus it is necessary to select a material that realizes both adequate stickiness power prior to curing and adequate adhesiveness and reinforcing performance after curing. With the

conventional stiffeners, the adhesiveness generated prior to curing was large because liquid rubber was used. It is therefore considered that the stickiness power can be reduced by selecting solid rubber.

3.2. Philosophy Concerning Drop-out Guarantee Prior to Curing

3.2.1. Method of temporary retention

A method of realizing temporary retention instead of stickiness power was studied. As a method of temporary retention of a stiffener to a steel panel to which rustproofing oil or press oil, for example, adheres, the use of magnetic force instead of stickiness power was studied from the following two perspectives.

- (1) The fact that steel material is used for the greater part of an automobile body
- (2) The reduction of the working load (automation)

In general, the addition of ferrite particles is used as a method of generating magnetic force in the resin curing agent⁽³⁾, and it is considered that temporary retention, prior to curing, can be realized by substituting the abovementioned filling agent with ferrite.

3.2.2. Setting the magnetization force

Figure 4 shows the mechanism of magnetization. The ferrite particles dispersed in the resin layer are magnetized by orienting them by means of a magnetizer that uses a permanent magnet, for example. Because the ferrite raw material itself has a high specific gravity, a high magnetic force can be produced by increasing the amount of ferrite added. There is however a possibility that the weight of the ferrite may affect the impact resistance and reinforcing

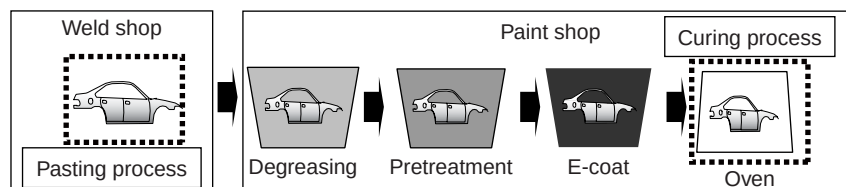


Fig. 2 Automotive product process

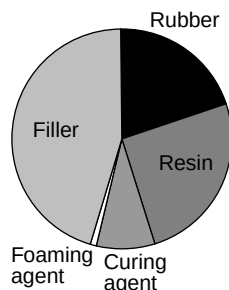


Fig. 3 Material formulation

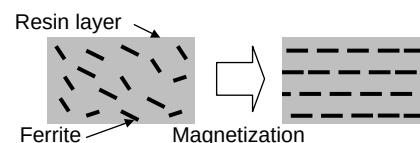


Fig. 4 Mechanism of generating magnetism

performance. It is considered that a high magnetic force can be produced with a relatively small amount of ferrite by either using ferrite of small particle diameter and large surface area, or by using a means of aligning the magnetization axes.

3.2.3. Demagnetization timing

It is considered that because the magnetic force is not needed after the resin has cured, the stiffener can be demagnetized by physically destroying the orientation of the ferrite particles after the resin has cured. Because drop-out will occur unless the adhesive strength is generated prior to demagnetization, the timing of demagnetization and adhesion after the resin has cured is important. It is considered that drop-outs can be avoided by prompting the resin to foam in order to destroy the orientation of the ferrite particles and thus bring about demagnetization, while ascertaining that foaming starts after the adhesive strength is generated.

4. Evaluation Method

4.1. Reinforcing Performance

As shown in Fig. 5, a three-point bending test is used to evaluate the reinforcing performance⁽⁴⁾. The reinforcing performance is defined as the load value for a 2 mm displacement obtained from the load-displacement curve shown in Fig. 6. Likewise, measurement after the environmental durability test is performed under the same condition. The conditions of the environmental durability test are shown in Table 1.

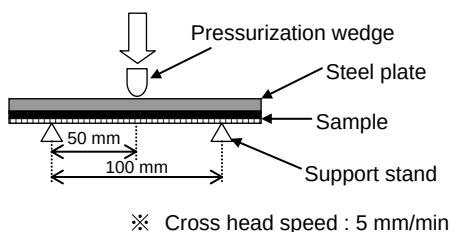


Fig. 5 Three-point bending test

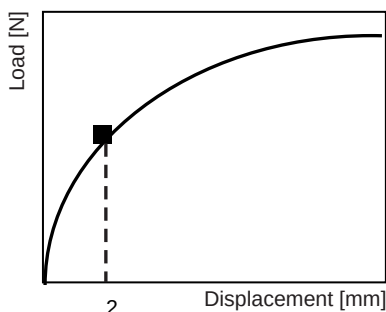


Fig. 6 Load-displacement curve

Table 1 Test conditions

Durability item	Conditions
Heat resistance	100 °C
Water resistance	50 °C warm water
Humidity resistance	50 °C 95%Rh

4.2. Impact Resistance Performance

The impact resistance test used was a slamming test⁽⁵⁾, a general test method employing the tester shown in Fig. 7. The test piece is placed on the slamming tester, and an impact is applied to it from an angle of 10 degrees. If separation or misalignment does not occur, the impact is applied repeatedly at the same angle to a total of 10 times. If neither of the above phenomena occurs during this time, the angle is increased 10 degrees at a time and the same test performed. The impact angle and the number of test cycles at which separation or misalignment occur were defined as the impact values.

4.3. Magnetic Force Measurement

The magnetic force was measured using a gauss meter [T-301 (main body GM-301) made by Denshijiki Industry Co., Ltd., or equivalent). The gauss meter probe was drawn along the line on the magnetizing face of the stiffener in the horizontal magnetization direction, in close proximity to it but without actually touching it, and the maximum magnetic force at this time was measured.

4.4. VOC Measurement

Regarding measurement of VOCs (Volatile Organic Compounds), a measurement method based on the method of the Japan Automobile Manufacturers Association, Inc.^{(6), (7)} was decided upon and implemented.

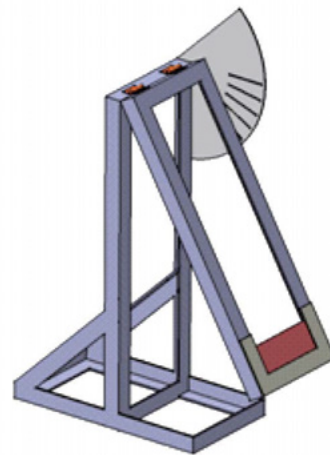


Fig. 7 Slamming test machine

5. Results and Examination

5.1. Design of Resin Layer that has no Stickiness Power, and Reinforcing Performance

As mentioned above, a resin layer that had no stickiness power was realized by changing the rubber component from a liquid type to a solid type. Also, it was decided to continue to use butadiene rubber for the rubber component in order to minimize the reduction of reinforcing performance after the durability test. The blend composition of the developed stiffener is shown in Fig. 8, and the results of performing a comparative reinforcement test of the conventional stiffener and that of the developed stiffener are shown in Fig. 9. It was found that compared to the conventional stiffener the developed stiffener showed no reduction of performance, including that of the durability test items. It is thought that this is because the absolute quantity of the composition of the resin layer that is affected by a reduction of the reinforcing performance is less in the case of the developed stiffener, making the developed stiffener less affected by the passage of time while the durability test is taking place.

5.2. Setting the Magnetic Force

Figure 10 shows the results of studying the ferrite content. The results show that it is necessary to increase the amount of ferrite in order to generate a high magnetic force, but that as the amount of ferrite is increased, the relative resin content decreases, leading to reduced

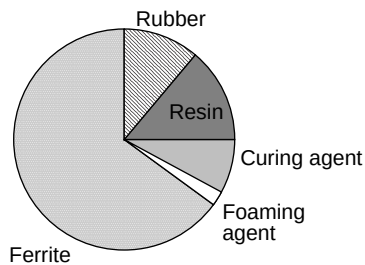


Fig. 8 Material formulation

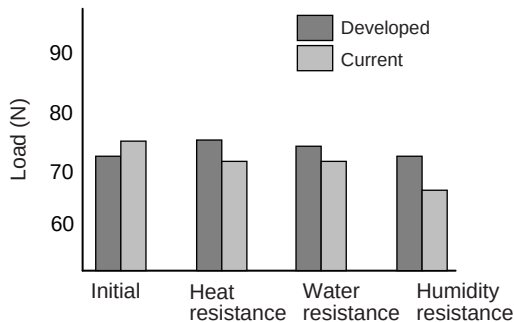


Fig. 9 Reinforcement performance

Table 2 Influence of ferrite particle diameter

Particle diameter (μm)	Magnetism (mT)	Impact resistance
1.0	13	30 deg, 1 time
1.2	13	30 deg, 1 time
2.8	13	30 deg, 1 time

Table 3 Influence of ferrite kind

Ferrite kind	Magnetism (mT)	Impact resistance
Isotropy	13	30 deg, 1 time
Anisotropy	13	30 deg, 1 time

reinforcement performance. It was also found that the magnetic force obtained by substituting the filling agent of the conventional blended material with the same amount of ferrite (45 wt%) did not provide satisfactory impact resistance. Consequently, a study of various types of ferrite was performed with the objective of obtaining high magnetic force through the addition of a relatively small amount of ferrite. The results are shown in Tables 2 and 3. It was predicted that small particle diameter material, which produced a larger magnetization area overall, would result in a larger magnetic force, and that large particle diameter material would result in a smaller magnetic force, but the results obtained in both cases were the same. The reason for this is thought to be that because only the maximum value of the magnetic force is measured, there is no change in the maximum value of the magnetic force, so as long as equal quantities of ferrite are added. Regarding the difference due to the grade of ferrite used, it is said that generally the magnetic force increases by using aeolotropic ferrite and magnetizing it from the magnetization axes⁽⁸⁾. An attempt was made to generate a high magnetic force by aligning the magnetization axes by rolling the ferrite dispersed in the resin material several times, though no difference with respect to isotropic ferrite was obtained. It is thought that

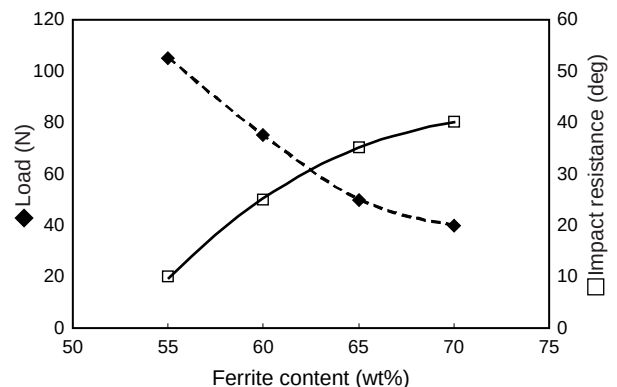


Fig. 10 Influence of ferrite content

this was because the resin layer is a rigid body, which suppresses the orientation movement of the ferrite particle in the resin layer from occurring freely. Based on the above, isotropic ferrite that had a particle diameter of 1.2 μm was selected from the viewpoint of versatility, and the blending ratio of the ferrite was made at least 65% in order to help ensure sufficient impact resistance. A study of how to increase the reinforcing performance in the region where the impact resistance performance is satisfied was performed.

5.3. Increasing the Reinforcing Performance

As a method of increasing the reinforcing performance, a study for increasing the thickness subsequent to curing was performed. Figure 11 shows the results of a study concerning the foaming rate. It was expected that by enhancing the foaming rate the thickness subsequent to curing would be increased, thus increasing the reinforcing performance. It was confirmed that the reinforcing performance increased as the foaming rate was increased, and that the impact resistance performance was satisfactory at all levels. However, drop-out was found to occur during baking. It is thought that this is due to the abovementioned difference in the timing between demagnetization and the generation of adhesive strength during baking. A foaming agent that resulted in a suitable timing difference was selected. Figure 12 shows the

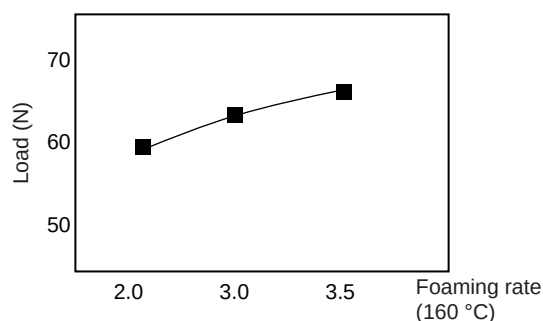


Fig. 11 Influence of foaming rate

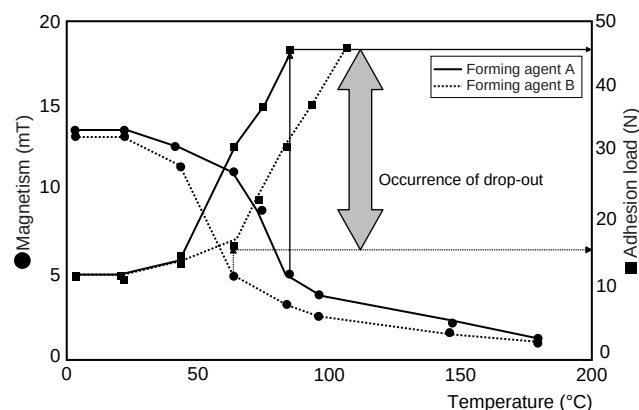


Fig. 12 Magnetism-load curve

Table 4 Impact resistance test

Process	Magnetism (mT)	Impact resistance
Degreasing	13	30 deg, 1 time
Pretreatment	13	30 deg, 1 time
E-coat	13	30 deg, 1 time

relationship between the disappearance of magnetic force and the occurrence of adhesiveness when foaming agents A and B were used. The minimum magnetic force for supporting the weight of the stiffener is 5 mT. From the figure, it can be seen that in the case where low temperature foaming type foaming agent B is used, foaming starts to occur early, speeding up demagnetization so that it occurs faster than adhesion to the steel sheet, which results in the stiffener dropping out. In the case where high temperature foaming type foaming agent A is used, adequate adhesive strength is generated by the time that the magnetic force reaches 5 mT. Consequently, it was decided to use foaming agent A and increase the thickness of the stiffener material prior to curing as the method of enhancing the reinforcing performance. Figure 13 shows the results of studying the thickness of the material prior to curing. It was found that the minimum thickness of the resin layer necessary for obtaining satisfactory reinforcing performance was 0.8 mm, so the thickness was set to 0.8 mm.

5.4. Effect of Paint Shop Zone Flow on the Magnetic Force

In mass-production, a stiffener is subjected to various chemical and physical factors such as warm-water washing (40°C) and pre-treatment by chemical liquids, and electric current due to electrodeposition, prior to being cured, so there are concerns about the effect of these factors on the magnetic force. Table 4 shows the results of the impact resistance test performed on the developed material. It was found that no demagnetization occurred due to the paint shop zone flow. Regarding temperature, demagnetization starts from the vicinity of 50°C, so it is considered that demagnetization did not occur in the

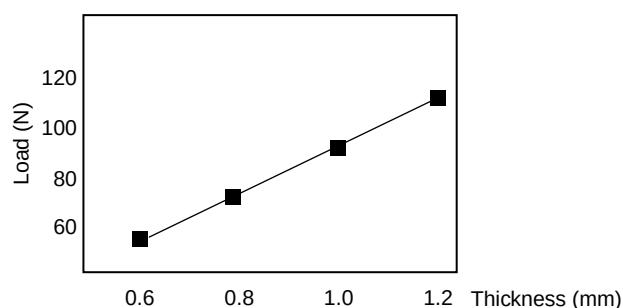


Fig. 13 Relation between thickness and load

vicinity of 40°C. Also, it is considered that the reason why no reduction of magnetic force due to acids, alkalis or other chemical liquids, or electric current was found is because, as mentioned in the sub-section concerning the mechanism of magnetization, the generation of magnetic force takes place as a result of the orientation movement of the ferrite particles, and there is no disappearance of magnetic force other than that due to the physical destruction of the orientation of the ferrite particles due to the foaming that occurs subsequent to curing.

5.5. VOC Measurement Results

Although stiffeners are used on outer parts, measurement was performed and countermeasures devised with a view to widening the range of applications in the future. Figure 14 shows the VOC measurement results for the developed stiffener and the conventional stiffener. These results indicate that, with the exception of formaldehyde, the developed stiffener produces roughly the same measured values of VOC as those of the conventional stiffener. It is thought that this is due to the fact that there was no change in the organic contents of the resin layer, and thus the developed stiffener did not contribute to an increase or decrease of the associated volatile matter. Regarding formaldehyde, the conventional stiffener uses melamin resin during the glass cloth molding process, and it was found that residual formaldehyde from the melamin resin manufacturing process remains in the product. With the developed material, the generation of formaldehyde is suppressed as a result of the change from melamin resin to acrylic resin.

6. Results of Confirmation on an Actual Vehicle

The developed material was applied to an actual vehicle, and a validation, which included the impact when parts are being bonded in the weld zone and when the body is being transferred from the suspension hanger

while moving from the weld zone to the paint zone, and the water shower during the pre-treatment process in the paint zone, was performed. Figure 15 shows the parts to which stiffeners are applied and the application positions. As a result, it was found that no deviation or drop-out occurred due to the curing process and there was no loss of quality in terms of appearance, so it was judged that there was no problem with the developed material. Also, it was confirmed that the rigidity of the outer panel satisfied the constraints of an actual vehicle, and that the developed stiffener, which satisfies the target reinforcing performance and impact resistance performance, is suitable for mass-production conditions.

7. Conclusion

A new stiffener that does not have release paper, nor needs to have pressure applied, and has enhanced workability was developed, and the following results were obtained.

- (1) The butadiene rubber in the resin layer was changed from a liquid to a solid type in order to reduce the sticking



Fig. 15 Application part and position

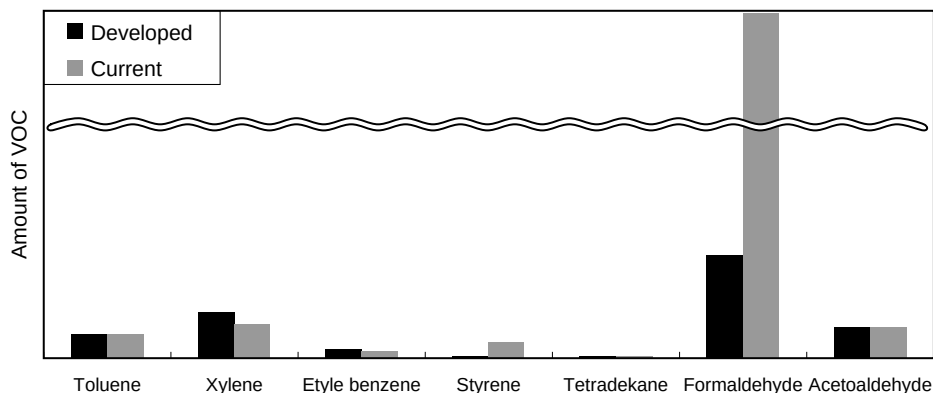


Fig. 14 VOC measurement result

power, thus enabling the release paper to be eliminated.

- (2) The ferrite blending ratio was studied and a foaming agent adopted, thus permitting material design that simultaneously realized high impact resistance and reinforcing performance.

The newly developed magnetic stiffener is used in the manufacture of the hood of the European ACCORD, which went on sale in 2008.

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■ Author ■



Masanobu YAGUCHI

Development of Lightweight Forged Piston Material by Controlling Needle-type Intermetallic Compound Size Using Carbon Mold Continuous Casting Method

Hidenori FUJII*

Tomoo OKA*

Masaki YONEYAMA*

Toshiaki DEGUCHI*

Hirotaka MURAKAMI*

ABSTRACT

The addition of transition metals to piston materials is an effective method of increasing their strength at high temperatures. However, the addition of transition metals increases specific gravity and promotes the crystallization of coarse intermetallic compounds, reducing forgeability. To respond to these issues, a new piston material has been developed in which the type and amount of the added transition metals and the size of the crystallized intermetallic compounds has been optimized. In order to control the size of the intermetallic compounds, the material is not produced by conventional metal mold casting, but by a carbon mold continuous casting method that enables a fine and uniform structure to be obtained. The optimization of the elements forming the intermetallic compounds and the quantity of these elements added, in addition to the addition of refining elements, to enable dispersion of needle-type intermetallic compounds throughout the alloy, has resulted in the achievement of levels of specific gravity and forgeability similar to those of a conventional forged piston material, while increasing strength at high temperatures by approximately 50%.

1. Introduction

Heat treatments are applied to forged piston materials such as A4032 in order to exploit the precipitation strengthening mechanism and help to further ensure strength. However, one issue has been the fact that the coarsening of precipitated compounds at around 300 °C results in a decline in strength. To respond to this issue, cooling mechanisms such as oil jets have been employed to reduce the temperature of the crown of the piston, but the higher power of modern engines has placed limits on the degree of cooling achieved using oil jets, and high strength at temperatures of around 300 °C is essential in order to achieve weight savings in the piston crown.

The project discussed in this paper developed an aluminum forged piston material that displayed equivalent specific gravity and forgeability to conventional materials while achieving increased high-temperature strength, by promoting the dispersion of needle-type intermetallic compounds of 3-30 μm in diameter through the aluminum matrix. The new material has helped to enable the achievement of a reduction in piston mass. This paper will discuss the issues associated with conventional forged

piston materials, the control of the microstructure of the developed material and the resulting increase in strength, and the characteristics obtained in the developed material.

2. Method of Development of Forged Piston Material

2.1. Method of Strengthening Developed Material

Table 1 compares the composition, specific gravity, size of intermetallic compounds, and method of piston manufacture for the developed and conventional piston materials. Figure 1 shows schematic diagrams of the microstructures of the conventional (A) and developed (B) piston materials. The precipitation strengthening mechanism^{(1), (2)} has been employed in order to increase high-temperature strength at temperatures of 250 °C and below in the conventional piston material, and the crystallization of intermetallic compounds has been controlled in order to help prevent a decline in material characteristics due to the coarsening of the compounds. In the project discussed here, the strengthening mechanism employed was dispersion strengthening that made direct use of the intermetallic compounds in order to achieve

* Motorcycle R&D Center

Table 1 Chemical composition, specific gravity, size of intermetallic compound and continuous casting mold of conventional forged piston and developed forged piston

	Chemical composition (mass%)								Specific gravity	Size of intermetal. compound (μm)	Continuous casting mold
	Si	Mg	Cu	Ni	Fe	Zr	Ti	Al			
Developed forged piston material	12	1	1	2	0.6	0.1	0.1	Bal.	2.70	Ave. 8 (3 - 30)	Graphite mold
Conventional forged piston material	12	1	1	1	–	–	–	Bal.	2.69	0	Metal mold

increased high-temperature strength at temperatures of around 300 °C. The issues associated with the dispersion strengthening mechanism are controlling the weight and the size of the secondary dispersion layer. The use of a metal matrix composite (MMC)^{(3), (4)} has been considered as a means of responding to these issues. An MMC is formed by adding “whiskers” or other strengthening materials to an alloy matrix. Because of this, it is easy to control the weight and size of the strengthening materials, and high-temperature strength can be increased using this method. However, producing MMC entails complex manufacturing processes and special manufacturing facilities. If alloy design could be used to enable dispersion strengthening, existing manufacturing processes could be employed. The project therefore sought a method by means of which needle-type intermetallic compounds could play the role of the strengthening material, and a fine and uniform dispersion could be achieved in the aluminum matrix in order to help prevent a decline in the material characteristics resulting from coarsening of the compounds. The project attempted to control the cooling rate of the material during manufacture, and to control the size of the intermetallic compounds using an alloy design suited to the cooling rate employed. The goal in using this method was to crystallize intermetallic compounds with the addition of the minimum amount of transition metal, to increase high-temperature strength while achieving an equivalent specific gravity to the conventional material, and to maintain forgeability.

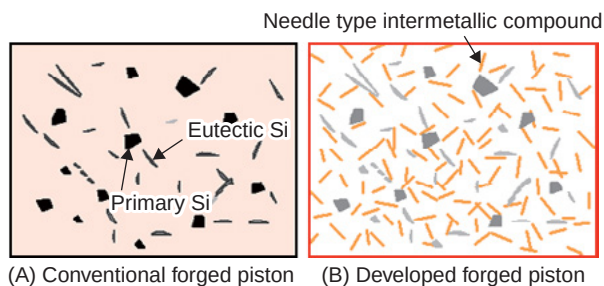


Fig. 1 Schematic illustration of microstructure of conventional forged piston and developed forged piston

2.2. Method of Producing Forging Billet for Developed Material

Control of the cooling rate in the solidification process during manufacture is an important factor in helping to enable control of the dispersion and the size of the needle-type intermetallic compounds in the aluminum alloy matrix. The relationship between the cooling rate of the piston material and the size of the compounds for different manufacturing methods is shown in Fig. 2. Using the standard method of manufacture for pistons, gravity die casting (GDC), it has been necessary to slow the rate of cooling in order to help ensure castability. Because of this, the addition of small amounts of transition metals results in a tendency towards the production of coarse compounds, placing limitations on the ability to actively exploit intermetallic compounds. More recently, the speed of cooling in HPDC has been exploited to enable the addition of large quantities of transition metals in order to produce pistons with higher levels of high-temperature strength^{(5), (6)}. HPDC pistons display high productivity, but the high rate of cooling during casting tends to result in an increase in the amount of transition metals in solid solution in the matrix, and a refinement of the size of the intermetallic compounds. As a result, it is necessary to add a high quantity of transition metals in order to crystallize sufficient intermetallic compounds to achieve the desired increase in high-temperature strength. This increases the

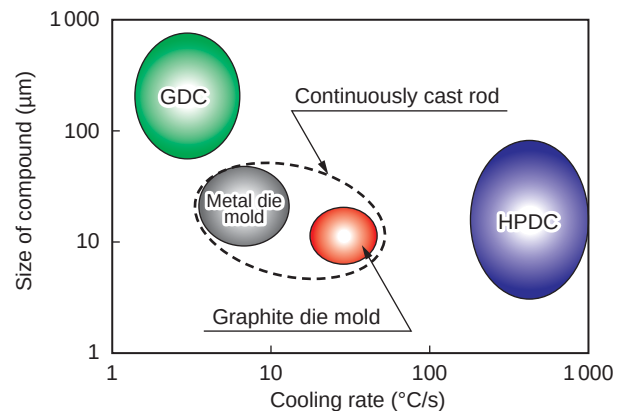


Fig. 2 Relationship between cooling rate of piston material and size of compound

specific gravity of the material, reducing the ability to achieve weight savings in the piston. In both of these casting methods, there are limitations on the ability to control the microstructure because the rate of cooling is determined by castability. By contrast, the use of billets for forging separates the processes of forming the piston and solidifying the material, resulting in a high degree of freedom in the selection of the cooling rate. Using this method, a range of cooling rates intermediate between those of GDC and HPDC can be employed. This cooling rate range has not previously been a subject of research focus. Exploiting this cooling rate range, the effect of coarse intermetallic compounds can be minimized, while also controlling the amount of transition metals in solid solution in the matrix. A carbon mold continuous casting method was employed to produce the billet for the developed material in order to achieve a faster and more stable cooling rate. The billet is produced using a metal mold in the normal continuous casting method. However, when this method is used cooling commences from the surface in direct contact with the mold. Because cooling takes place from the external circumference, the rate of cooling of the center is slow, resulting in a tendency towards coarsening of the microstructure [Fig. 3(A)]. When a carbon mold is used, the cast material can be cooled by means of direct contact with cooling water, enabling the formation of a fine and uniform microstructure⁽⁷⁾. [Fig. 3(B)]

2.3. Forging Process

In contrast to casting, forging presents issues in the formation of near-net shapes. Because of this, when the material has been heated once, a “1 heat 1 press” (1H/1P) formation process is desirable in the forging process. Heating the material multiple times and forging is effective in the case of materials that are challenging to forge, and in the formation of complex shapes. However,

correction in each forming process in order to attempt to prevent forging defects and the necessity of using multiple molds reduces productivity. Forming the product using a 1H/1P process enables the achievement of consistent quality in the forged material, in addition to high productivity. In order to realize 1H/1P formation, the high-temperature elongation of the forging billet should be close to that of a conventional forged material. The project therefore sought an alloy design and a method of control of intermetallic compounds that would help to ensure increased high-temperature strength and high-temperature elongation.

3. Method of Alloy Design for Developed Material

3.1. Design of Base Alloy

The quantities of the main constituents, Si, Mg and Cu, to be added in order to achieve the basic, essential characteristics of an alloy for piston use, were determined in the following ways.

3.1.1. Setting quantities of Si and Mg to be added

In order to attempt to prevent forgeability from declining due to crystallization of primary Si crystals, the ratio of Si to be added was set at 12 mass%, the upper limit of the eutectic point. Mg is effective in increasing creep strength, but the ratio was set at 1.0% in order to attempt to prevent breakage during forging as a result of crystallization of Mg_2Si .

3.1.2. Setting quantity of Cu to be added

The coexistence of Cu and Ni promotes the crystallization of Al-Cu-Ni ternary intermetallic compounds, and this has an effect on high-temperature strength. The effect of the addition of differing quantities of Cu and Ni to the Si 12%-Mg 1% alloy on the high-temperature tensile characteristic at 300 °C was therefore examined. Figure 4 shows the results of varying the quantity of Cu added at a Ni content of 3%, and Fig. 5 shows the results of varying the quantity of Ni added at a Cu content of 1%. Virtually no change was observed with the addition of Cu, while the addition of Ni had a substantial effect. Figure 6 shows a secondary electron image of an alloy with ratios of Ni3% and Cu1%, and Fig. 7 shows a secondary electron image of an alloy with ratios of Ni3% and Cu5%. Needle-type Al-Ni intermetallic compounds crystallize at a Cu ratio of 1%, but at a Cu ratio of 5%, the crystallization of dendritic Y alloys impedes the crystallization of needle-type intermetallic compounds. There is therefore no observable increase in strength at 300 °C with the addition of Cu. Increasing the quantity of Cu added increases the specific gravity of the material, and the ratio of Cu was therefore set at 1%, the minimum content essential to achieving strength up to 250 °C.

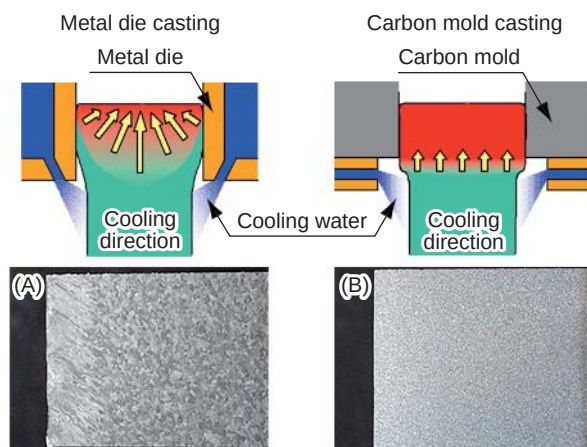


Fig. 3 Schematic illustration of casting method and microstructure of conventional casting material and carbon mold casting material

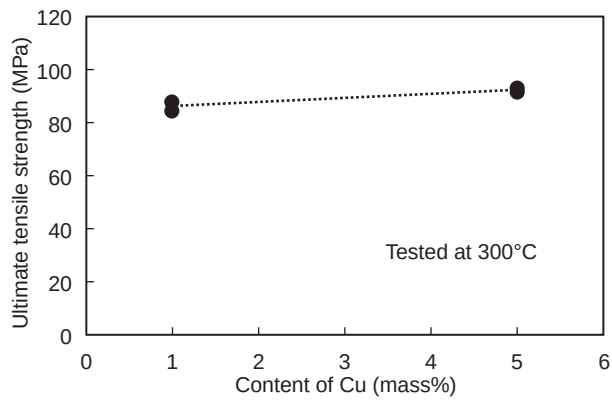


Fig. 4 Effect of Cu addition on tensile strength of Al-Si12-Mg1-Ni3 alloy

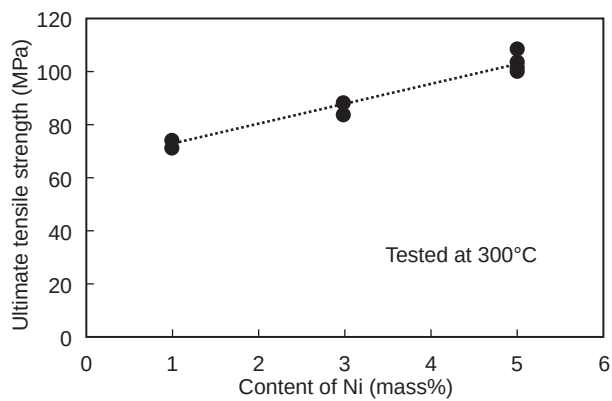


Fig. 5 Effect of Ni addition on tensile strength of Al-Si12-Mg1-Cu1 alloy

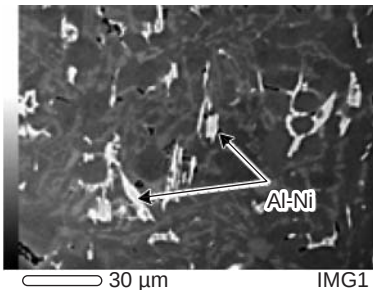


Fig. 6 Microstructure of Al-Si12-Mg1-Cu1-Ni3 alloy

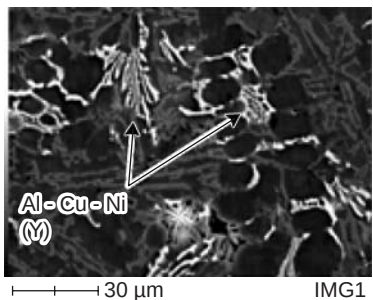


Fig. 7 Microstructure of Al-Si12-Mg1-Cu5-Ni3 alloy

3.2. Selection of Transition Metal

The addition of a transition metal is an effective means of increasing strength at 300 °C and above, but adding transition metals in excessive quantities can result in a decline in forgeability and increased specific gravity. Transition elements were therefore screened from the perspectives of forgeability and specific gravity. Figure 8 shows the results of high-temperature tensile tests conducted on test pieces taken from continuously cast rods (pre-forging) and forged pistons produced from an Al-Si12-Mg1-Cu1 alloy to which different transition metals had been added. The tests were conducted following exposure of the test pieces to a temperature of 300 °C for 100 hr. In the A range, the addition of high quantities of transition metals increased the strength of the continuously cast rod, but strength declined considerably following forging, and the overall strength increase following forging was small in comparison to the increase in specific gravity. This phenomena results from the intermetallic compounds fracturing, brought about by the plastic deformation that occurs during forging. The agglomerated intermetallic compounds that crystallize when the

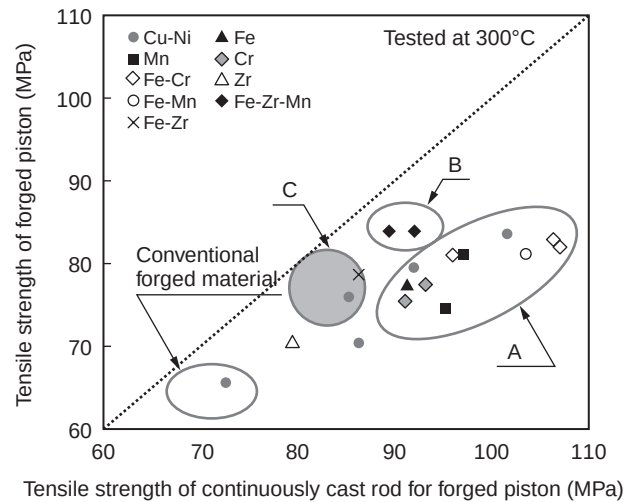


Fig. 8 Tensile strength of continuously cast rod and forged piston

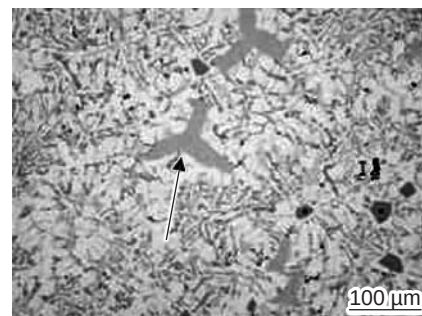


Fig. 9 Shape of intermetallic compound of Mn, Cr added alloy

quantities of Cr and Mn added exceed specific amounts (Fig. 9) also substantially reduce strength following forging. The crystallization of coarse intermetallic compounds was observed in the B range, resulting in the formation of cracks during the forging process. The target range for this project was the C range, which displayed good forgeability, low specific gravity, and high high-temperature strength following forging. Based on the results of the study of transition metals, Ni and Fe were selected for addition. Ni has a low solid solubility limit in aluminum, and is effective in helping to ensure the crystallization of intermetallic compounds. The addition of this element is considered to be effective in increasing high-temperature strength while having a minimal effect on the thermal conductivity of the alloy^{(8), (9)}. In the screening also, the addition of Ni increased strength through the formation of fine, needle-type intermetallic compounds. Fe, by contrast, is recognized as a harmful element that promotes the coarsening of intermetallic compounds, and has conventionally been controlled as an impurity. In the screening, however, Fe was found to not be harmful in the cooling rate range associated with the carbon mold continuous casting method, and its addition was found to be able to increase high-temperature strength through the crystallization of fine, needle-type intermetallic compounds. This project therefore made active use of Fe as an element that was effective in increasing high-temperature strength, and examined the potential for an alloy containing both Ni and Fe.

3.3. Study of Potential for Alloy Containing both Ni and Fe

A study of the potential for an alloy containing both Ni and Fe was conducted based on the relationship between the amounts of each element added, forgeability, and the state of crystallization of intermetallic compounds. Figure 10 shows the results of piston forging tests using Al-Si12-Mg1-Cu1 alloys to which Ni had been added in

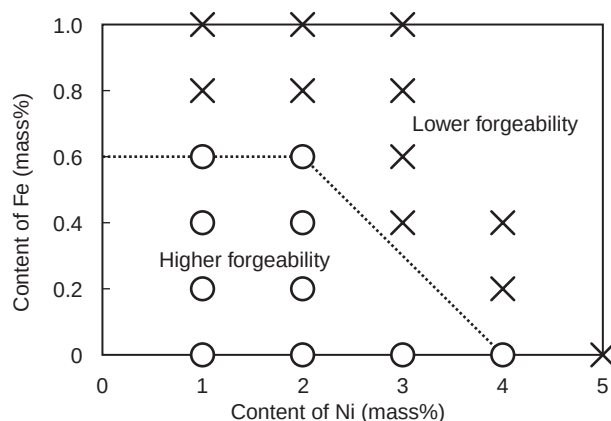


Fig. 10 Effect of Ni and Fe addition on forgeability of Al-Si12-Mg1-Cu1 alloy

isolation and to which Ni and Fe had been added at the same time. Alloys to which Ni was added in isolation displayed no cracking and adequate forgeability up to a ratio of 4%. Forgeability was adequate in the alloys to which Ni and Fe were added simultaneously up to a ratio of Ni 2% and Fe 0.6%. While the acceptable quantity of Fe was low in comparison to Ni, a range was determined in which forgeability was adequate while Ni and Fe were both present in the alloy. Next, a study was conducted from the perspective of the microstructure. The addition of Ni in isolation resulted in the crystallization of needle-type intermetallic compounds at ratios of 2% and higher, and the size of the compounds increased as the ratio of Ni increased. The addition of Ni and Fe together resulted in the crystallization of ternary Al-Fe-Ni needle-type intermetallic compounds. The addition of the elements together promoted increased coarsening of the intermetallic compounds in comparison to the addition of Ni in isolation. However, the crystallization of needle-type intermetallic compounds responded more readily to the amount of Fe added than Ni, and Fe can therefore be regarded as a useful element in enabling specific gravity to be reduced. The effective use of the range of additive amounts in which no coarsening of the intermetallic compounds was observed would therefore enable specific gravity to be reduced. Figures 11 and 12 show the coarse intermetallic compounds in the alloy when Ni is added in isolation in a ratio of 5% (Fig. 11) and when Ni and Fe are added in ratios of 3% and 1% respectively (Fig. 12). The coarse intermetallic compounds produced both by the

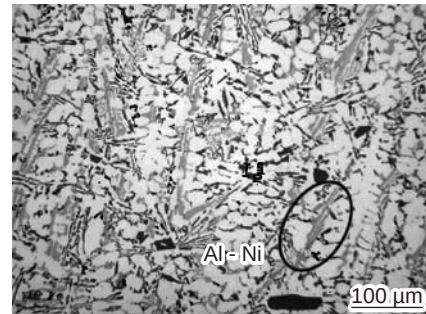


Fig. 11 Microstructure of Al-Si12-Mg1-Cu1-Ni5 alloy

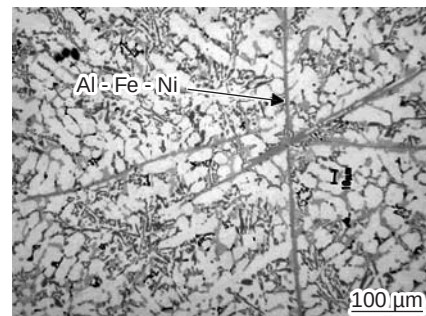


Fig. 12 Microstructure of Al-Si12-Mg1-Cu1-Ni3-Fe1 alloy

Table 2 Chemical composition, specific gravity and average size of intermetallic compound of tested alloy and conventional alloy for forged piston

	No.	Ni	Fe	Mn	Cr	Zr	Ti	Specific gravity	Average size of intermetallic compound (μm)
Tested alloys	1	3	—	—	—	—	—	2.71	20
	2	3	—	1	—	—	—	2.73	25
	3	3	—	—	0.2	—	—	2.71	15
	4	2	0.6	—	—	0.1	0.1	2.70	8
Conventionally forged piston alloy	1	—	—	—	—	—	—	2.69	0

addition of Ni in isolation and of Ni and Fe together would result in cracking during the forging process. Because of the high correlation between the microstructure and forgeability, the refinement of the intermetallic compounds is important to the achievement of good forgeability. A ratio of Ni in isolation of 3% (taking specific gravity into consideration) and a ratio of Ni 2%-Fe 0.6% when Ni and Fe are added together were selected as the highest quantities of transition metals that could be added while retaining the potential to achieve good forgeability, and alloys containing these quantities of the metals were studied comparatively.

4. Size of Needle-type Intermetallic Compounds and Mechanical Properties

4.1. Refining Elements and Size of Needle-type Intermetallic Compounds

Refining elements were added to control the size of the intermetallic compounds in the alloys under study. The addition of Ti+Zr, Mn, Cr and other elements is known to refine the intermetallic compounds in aluminum alloys⁽¹⁰⁾. The relationship between the refining elements and the size of the intermetallic compounds in the alloys was studied in order to clarify the type of high-temperature characteristics that would be obtained as a result of adding these refining elements in order to control the size of the intermetallic compounds. Table 2 shows composition, specific gravity, and average size of needle-type intermetallic compound for the test alloys and a conventional material. Ti+Zr were added to an alloy to which Ni and Fe had previously been added together. Mn and Cr were added to alloys to which Ni had previously been added in isolation. Neither Mn nor Cr was added to alloys to which both Ni and Fe had been added, because this would result in agglomeration of the needle-type intermetallic compounds. In all cases the matrix was an Al-Si12-Mg1-Cu1 alloy. Table 2 indicates that the size of the intermetallic compounds can be controlled by adjusting the quantities of transition elements and refining elements added to the alloy. The study showed that the addition of Ti+Zr to an alloy containing both Ni and Fe

was effective in enabling the formation of fine intermetallic compounds.

4.2. Size of Needle-type Intermetallic Compounds and High-temperature Elongation

The relationship between the average size of the intermetallic compounds and elongation at 300 °C was studied in order to determine the intermetallic compound size at which an equivalent level of forgeability to a conventional forged material was obtained (Fig. 13). Elongation declined as the size of the intermetallic compounds increased. The conventional material displayed an elongation of 35%, while the highest level of elongation among the test alloys was displayed by No. 4 at 23%. The elongation of the conventional material and alloy No. 4 was measured at high temperatures of 300-500 °C in order to study methods of achieving increased elongation as the temperature of the billet was increased during forging (Fig. 14). The results showed that increasing the temperature to which the billet was heated to approximately 50 °C above the conventional forging temperature range of 300-400 °C enabled the achievement of a level of elongation similar to that of the conventional material, indicating that an equivalent level of forgeability

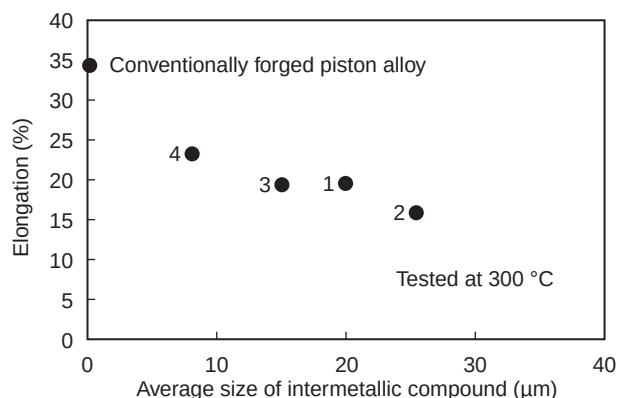


Fig. 13 Relationship between average size of intermetallic compound and elongation

could be achieved with no modification of the conventional forging process. However, at an average intermetallic compound size of 20 mm and larger, there was a substantial decline in elongation, which might necessitate an increase in the number of forging processes or restrict the potential shapes of the pistons. In order to achieve adequate forgeability, it would be necessary to control the average size of the intermetallic compounds to below 20 mm, in addition to controlling the coarsening of the compounds.

4.3. Size of Needle-type Intermetallic Compounds and Fatigue Strength

From the perspective of forgeability, it was desirable to reduce the size of the intermetallic compounds, but this would generate the possibility that sufficient fatigue strength at high temperatures could not be achieved. The relationship between the average size of the intermetallic compounds and fatigue strength at 300 °C was therefore studied. The results are shown in Fig. 15. The test pieces were taken from the crown of a piston following forging

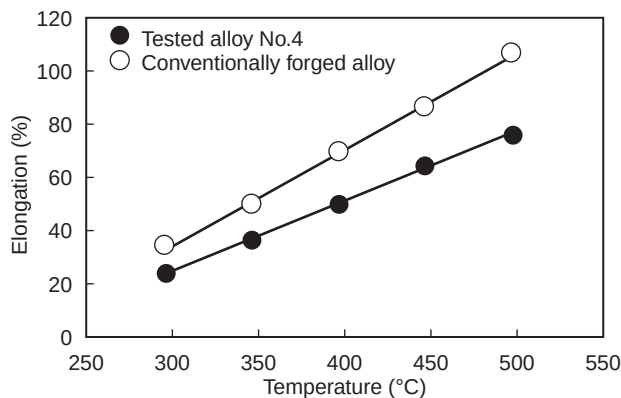


Fig. 14 Elongation of tested alloy No.4 and conventionally forged piston

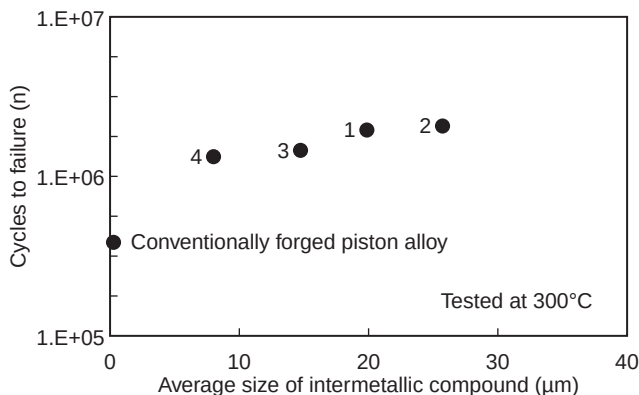


Fig. 15 Relationship between average size of intermetallic compound and cycles to failure

and the application of heat treatments. Fatigue strength was expressed by cycles to failure in a 30 MPa pulsating tension fatigue test. Fatigue strength was increased even in alloy No. 4, which displayed the lowest average intermetallic compound size of 8 mm, and little change in fatigue strength was observed even as the average size of the compounds increased. This indicates that an adequate increase in strength could be achieved at an average intermetallic compound size of 8 mm.

4.4. Needle-type Intermetallic Compounds in Developed Alloy

From the results discussed in sections 4.2. and 4.3., alloy No. 4, containing both Ni and Fe, was selected as the optimum alloy enabling the achievement of both adequate forgeability and high-temperature strength in the piston. The microstructure of the developed material is shown in Fig. 16. A microstructure in which needle-type intermetallic compounds of 3-30 mm are dispersed can be observed. The use of the minimum essential amount of transition metals in this alloy has enabled the achievement of a specific gravity similar to that of a conventional material. The addition of Fe at a ratio of 0.6% has also enabled the use of reclaimed aluminum, contributing to material recycling.

5. Properties of Developed Material

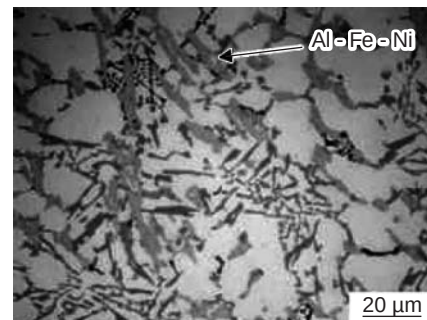


Fig. 16 Microstructure of Al-Si12-Mg1-Cu1-Ni2-Fe0.6-Ti0.1-Zr0.1 alloy

5.1. High-temperature Fatigue Strength

Figure 17 shows the results of fatigue tests between 200 and 300 °C conducted on test pieces from pistons manufactured using the developed material (alloy No. 4), a conventional forged material, and a conventional cast material. The developed material displays higher fatigue strength than the conventional materials from a low temperature of 200 °C to a high temperature of 300 °C. At 300 °C in particular, the fatigue strength of the developed material is approximately 50% higher than that of the conventional forged material.

5.2. Creep Strength

Pistons used in high-temperature environments can become deformed by creep during engine operation, resulting in a decline in engine performance. In achieving weight savings in the pistons, it is therefore necessary to consider creep characteristics in addition to increased fatigue strength. Figure 18 shows the creep strength of a conventional forged piston material and the developed material. Creep strength was evaluated by the time to failure when test pieces from pistons forged using the materials were subjected to a 47.8 MPa stress in a high-temperature environment of 300 °C. The developed material displayed creep strength more than twice as great as that of the conventional forged material.

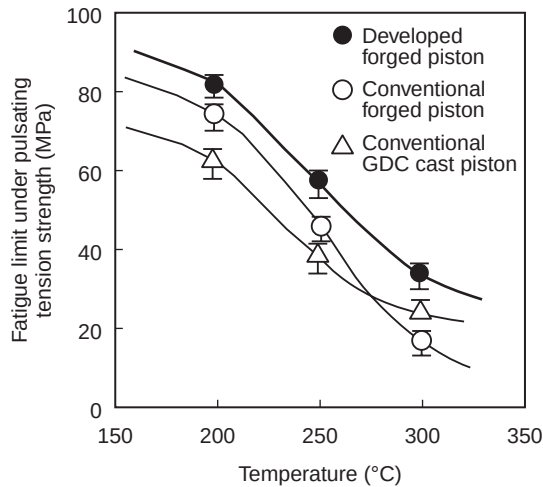


Fig. 17 Pulsating fatigue strength of pistons tested at various temperatures

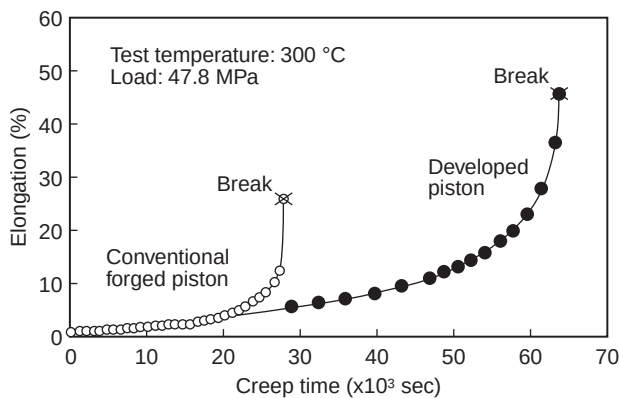


Fig. 18 Creep strength of developed piston and conventionally forged piston

6. Verification in Actual Engine

Based on the results of the tests of the material, its effectiveness in an actual engine was verified. The engine employed was a water-cooled in-line four-cylinder 998 cm³ model. The weight of the pistons was reduced by 5% by reducing the thickness of the crown by 18% against conventional pistons and removing the pin boss ribs. CAE analyses and engine durability tests were then conducted. The shape of the pistons employed in the engine durability tests is illustrated in Fig. 19. The results of the CAE analyses are shown in Fig. 20. The analysis showed that the lighter pistons displayed a similar rate of safety to conventionally-shaped pistons. In addition, the lighter pistons satisfied performance criteria including sliding performance and dimensional stability in engine durability tests under the same loads as the conventional pistons, indicating that they presented no issues in terms of durability.

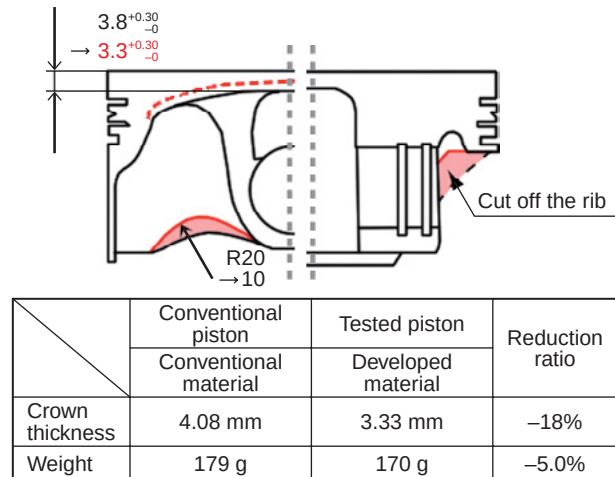


Fig. 19 Schematic diagrams of tested piston and actual crown thickness and piston weight

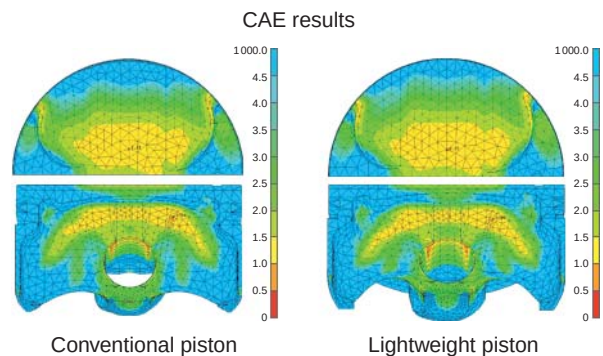


Fig. 20 Fatigue factor of tested piston

7. Conclusion

The project discussed in this paper set out to realize a lightweight piston that could be manufactured using a "1 heat 1 press" (1H/1P) forging process, and developed an Al-Si12-Mg1-Cu1-Ni2-Fe0.6 forged alloy using a carbon mold continuous casting method that enabled the cooling rate to be controlled.

- (1) The exploitation of the refining effect of Zr+Ti to control the size of the needle-type intermetallic compounds to 3-30 μm and the addition of the minimum quantities of Ni and Fe have enabled the achievement of a similar level of specific gravity and forgeability to that of a conventional forged material.
- (2) The developed material displays higher high-temperature strength than conventional forged and cast materials from 200-300 $^{\circ}\text{C}$, and approximately 50% higher high-temperature strength than a conventional forged material at 300 $^{\circ}\text{C}$.

The pistons manufactured using the developed material have already begun mass production, beginning with those that are employed in the 2008 CBR1000RR super sports motorcycle.

Acknowledgments

The authors wish to take this opportunity to thank Toyama Alloy Co., Ltd., Miyamoto Industry Co., Ltd., and Honda Foundry Co., Ltd for their generous assistance in the development of this product and technology.

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■ Author ■



Hidenori FUJII



Tomoo OKA



Masaki YONEYAMA



Toshiaki DEGUCHI



Hirotaka MURAKAMI

Time-sharing PWM Control for Multi-motor Drive

Manabu KUROKAWA*

ABSTRACT

The necessity for the use of multi-motor drive systems is increasing, for example in hybrid vehicles, fuel cell vehicles and electric vehicles. Multiple-output inverters, which are able to drive multiple motors simultaneously, are available as a means of preventing the system enlargement with the increase in the number of motors.

The project discussed in this paper applied time-sharing PWM control to a multiple-level multiple-output inverter. The viability of proposed control method was confirmed and power loss characteristics were clarified. Within the operating range in which the sum of the voltages supplied to the multiple motors did not exceed the voltage of the power source, an output current control characteristic equivalent to that of a conventional independent inverter was achieved. In addition, the project demonstrated that the use of switching devices with superior current conduction characteristics in the multiple-level system was an effective means of increasing efficiency.

1. Introduction

Demands for the reduction of environmental impacts by automobiles are increasing recently. Energy conservation technology by electric motor represents one means of responding to these demands. Efficient systems using three-phase motors are being employed as vehicle drive systems producing several tens of kW or more, as systems driving electric air conditioners producing several kW in hybrid and electric vehicles, and as systems driving electric power steering in large vehicles.

Figure 1 shows the conventional parallel configuration of the three-phase inverters that are necessary to drive

multiple motors. Three-phase inverters used for motor drive employ multiple power switching devices, and the number of switching devices should be reduced to lower costs and to reduce size. One means of reducing the number of switching devices is to employ multiple-output inverters that share some switching devices among multiple inverters. The circuit configurations of multiple-output inverters can be divided into multi-phase systems that reduce the number of phases by sharing phases [Fig. 2 (a)]^{(1), (2)} and multi-level inverter systems that use series-connected switching devices for each phase and share the some of the switching devices [Fig. 2 (b)]. A number of R&D projects are being conducted on multi-phase systems, but the number of components that can be reduced is small in the case of these systems, and they have not yet come into practical use. A type of multi-level system in which the carrier waveform is divided into several voltage levels has been proposed^{(3), (4)}. However, the detailed analyses of power conversion efficiency for multi-level inverters have not been conducted yet.

The research discussed here proposed the use of time-sharing PWM control for multi-level inverters. The proposed time-sharing PWM control strategy enables to drive multiple motors simultaneously and independently, with less switching devices than conventional inverters or multi-phase inverters. In addition, power conversion efficiency was calculated from the loss characteristic of the switching devices, and compared with the efficiency of a conventional parallel three-phase inverter.

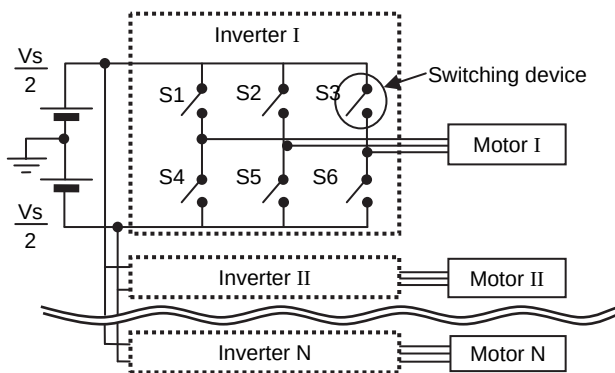
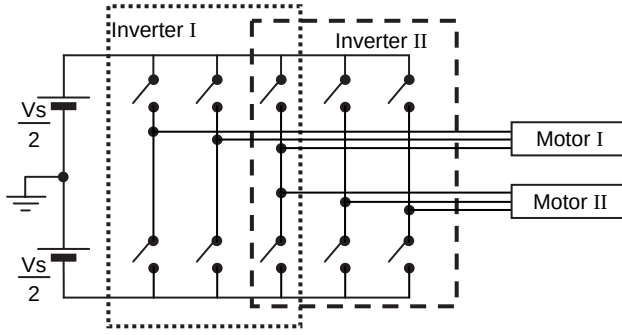
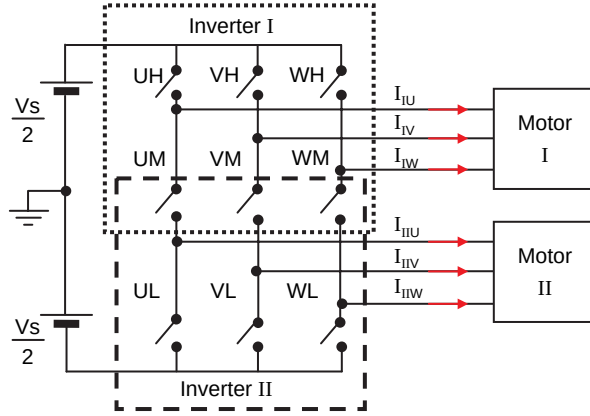


Fig. 1 Multiple-output inverter by parallel 3-phase configuration

* Automobile R&D Center



(a) Multiple-output inverter by multi-phase configuration



(b) Multiple-output inverter by multi-level configuration

Fig. 2 Multiple-output inverter

2. PWM Control of a Multiple-output Inverter

2.1. Conventional Inverter

Figure 1 shows the configuration of a conventional three-phase inverter. Each phase of the inverter is formed by two series-connected switching devices on high- and low-side, and three phases (U/V/W) of the inverter are connected in parallel. The motor is driven by controlling the total switching patterns of six switching devices (S1-S6). Figure 3 shows the timing signal (PWM signal) for the ON and OFF states of the U phase switching devices. The signal for each switch is determined by comparing the target voltage waveform, V_u , with the carrier waveform.

2.2. Configuration of Multiple-output Inverter

As Fig. 2 (b) shows, a three-level two-output inverter employs three switching devices (high, middle, and low) for each phase of a three-phase inverter. Inverter I, which drives motor I, consists of the six high- and middle-level switching devices of the three phases (UH, UM, VH, VM, WH, WM). Inverter II consists of the six middle- and low-level switching devices (UM, UL, VM, VL, WM, WL),

and drives motor II. This configuration enables to reduce the number of switching devices that are necessary to control two motors from 12 to 9.

Figure 2 shows examples of control of two-motor system. A conventional inverter and multi-phase inverter need $6 \times N$ switching devices and $4 \times N + 2$ switching devices, respectively. By contrast, a multi-level inverter is able to control N motors using only $3 \times N + 3$ switching devices.

2.3. Proposed Time-sharing PWM Control

Figure 4 shows a control method for each switching device that enables simultaneous independent control of two motors by a three-level, two-output inverter. The figure shows the method for U phase. As shown in Fig. 4 (a), when a three-level two-output inverter is used for simultaneous independent control of two motors, the formerly used carrier cycle, T_s , is divided into two stages, the inverter I control period T_I and the inverter II control period T_{II} . T_I and T_{II} are determined by the ratio between the target voltages V_{IU} and V_{IIU} , and the equations for them are shown in (1) and (2).

$$T_I = T_s \times V_{IU} / (V_{IU} + V_{IIU}) \quad (1)$$

$$T_{II} = T_s \times V_{IIU} / (V_{IU} + V_{IIU}) \quad (2)$$

Carrier waveform I is triangular during period T_I , and has a constant value given by $V_s/2$ during period T_{II} . Carrier waveform II is triangular during period T_{II} , and has a constant value given by $V_s/2$ during period T_I .

Due to the division of the control interval, the average line to line output voltage that is determined by comparison of the target voltage waveform and the carrier waveform is reduced to T_I/T_s for inverter I, and to T_{II}/T_s for inverter II, respectively. Therefore, the voltage waveforms V_{IU}' and V_{IIU}' are determined by multiplying

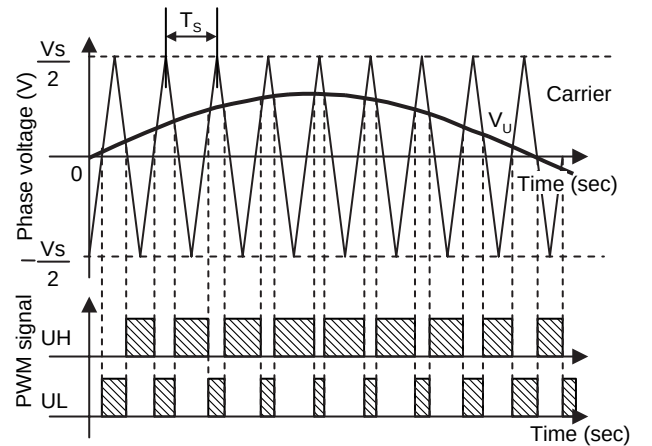


Fig. 3 Conventional PWM control

the target voltage waveforms V_{IU} and V_{IUI} by T_s/T_I and T_s/T_{II} . PWM signals are generated by comparing these V_{IU}' and V_{IUI}' with the carrier waveforms. The relationship between V_{IU}' , V_{IUI}' and V_{IU} , V_{IUI} is expressed by Eqs. (3) and (4).

$$V_{IU}' = V_{IU} \times T_s / T_I \quad (3)$$

$$V_{IUI}' = V_{IUI} \times T_s / T_{II} \quad (4)$$

Figure 4 (b) shows the method of generation of the PWM signal during periods T_I and T_{II} for two cycles of the conventional carrier cycle. During period T_I , UH is ON and UM is OFF in the range in which $V_{IU} > \text{carrier I}$, and UH is OFF and UM is ON in the range in which $V_{IU} < \text{carrier I}$. UL is ON for all ranges during period T_I . During period T_{II} , UM is ON and UL is OFF in the range in which $V_{IUI} > \text{carrier II}$, and UM is OFF and UL is ON in the range in which $V_{IUI} < \text{carrier II}$. UH is ON for all ranges during period T_{II} . In the cases of V phase and W phase also, the target voltage waveforms V_{IV} and V_{IW} are compared with carrier waveform I; V_{IIV} and V_{IIW} are compared with carrier waveform II; and control signals are generated in the same way.

Controlling the switching devices using this signal generation method enables motor I to be controlled by inverter I during period T_I , and motor II to be controlled by inverter II during period T_{II} .

Figure 5 (a) shows the current pathway of inverters I and II during period T_I , and Fig. 5 (b) shows the current pathway of the inverters during period T_{II} . Because the low-side switching devices for all phases are ON during period T_I , inverter I is equivalent to a conventional three-phase inverter with respect to motor I. In addition, because the inverter II current flows back into the lower switching devices, which are switched ON, there is no interference between it and the current output of inverter I. In the same

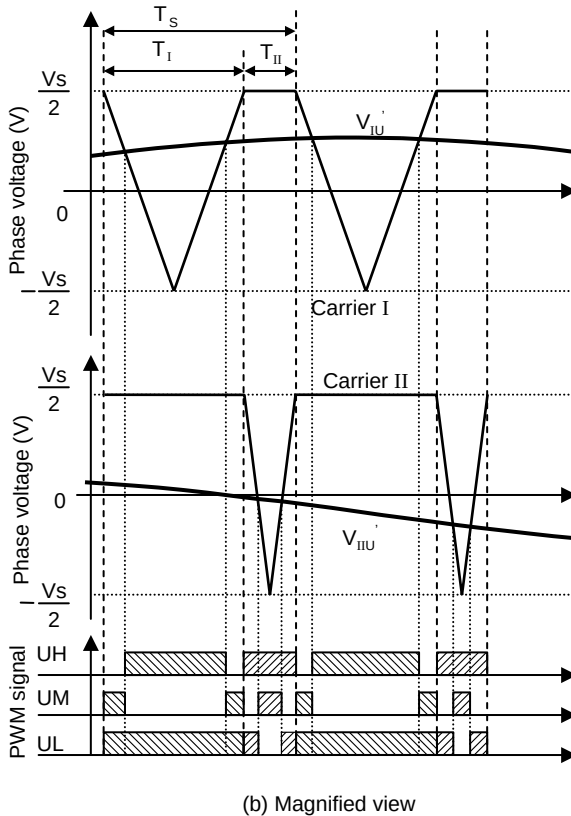
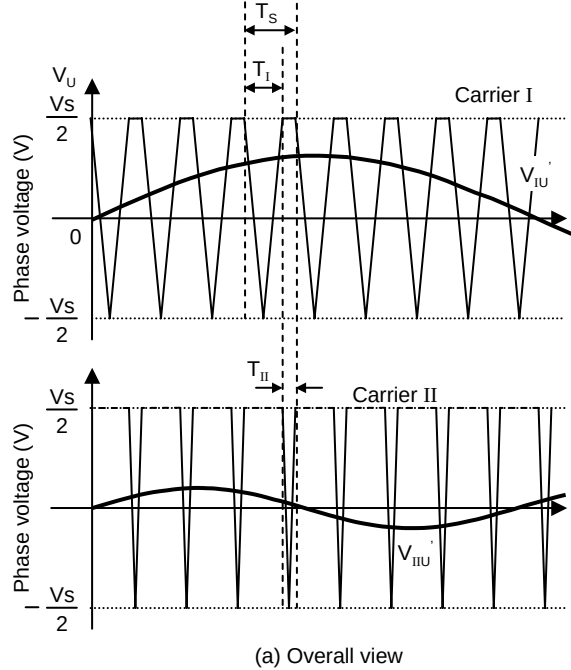


Fig. 4 PWM control in 3-level 2-output inverter

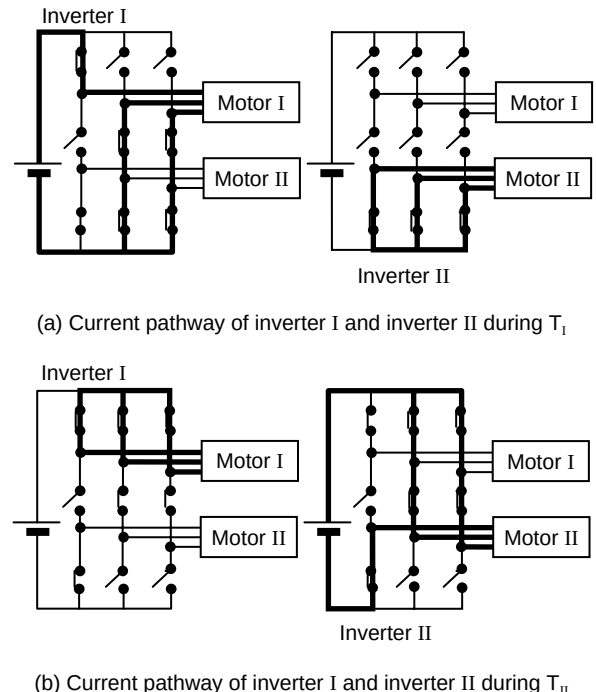


Fig. 5 Current pathway of 3-level 2-output inverter

way, during period T_{II} , inverter II is equivalent to a conventional three-phase inverter with respect to motor II, because the high-side switching devices are switched ON. The inverter I current flows back into the upper switching devices, which are switched ON, and therefore there is no interference between it and the current output of inverter II. This enables motors I and II to be controlled simultaneously and independently. However, the fact that the voltage output of the inverters is zero during the period in which current is flowing back places limitations on the output voltage during control periods T_I and T_{II} . If the average output voltage of inverter I is defined V_I and the average output voltage of inverter II is defined V_{II} , then

$$V_I \leq V_s \times \frac{T_I}{T_s} \quad (5)$$

$$V_{II} \leq V_s \times \frac{T_{II}}{T_s} \quad (6)$$

Given

$$T_I + T_{II} = T_s \quad (7)$$

then

$$V_I + V_{II} = V_s \quad (8)$$

and the range of voltage that can be output by V_I and V_{II} is as shown in Fig. 6.

2.4. Reduction of Switching Times

Using the method of generation of PWM signals shown in Fig. 4, switching devices are switched ON or OFF a total of seven times for each carrier period T_s (UH: twice; UM: four times; UL: once). The number of instances of switching has increased from four for a conventional parallel three-phase inverter. This is a factor in increased switching loss. To reduce switching times for each carrier period, PWM generation method by a sawtooth waveform is proposed. As shown in Fig. 7, switching devices are turned on and off only 3 times for each carrier period (UH: once; UM: once; UL: once). Substantial reduction of total switching times can be realized by using PWM generation with sawtooth waveform.

3. Simulation and Analysis Results

3.1. Analysis Method

An electric circuit simulation was conducted in order to verify the viability of the proposed time-sharing PWM control for a multiple-output inverter. Ideal switching devices (switching time: 0 sec; resistance: 0 Ω) were used in this simulation. PSIM was employed as the electric

circuit simulator. The control logic used in the simulation is shown in Fig. 8. The PWM signals UH (VH, WH), UM (VM, WM) and UL (VL, WL) were determined by comparing voltage waveforms V_{IU} (V_{IV} , V_{IW}), V_{IIU} (V_{IIV} , V_{IIW}) and the carrier waveforms in periods T_I and T_{II} , respectively.

3.2. Output Voltage Characteristic

Figure 9 shows the three-phase output current waveform for a three-level two-output inverter during steady-state operation. The frequency and amplitude of the output current are 100 Hz and 50 A, respectively, for inverter I, and 300 Hz and 20 A, respectively, for inverter

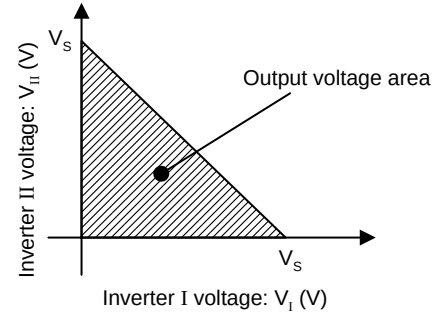


Fig. 6 Output voltage area of 3-level 2-output inverter

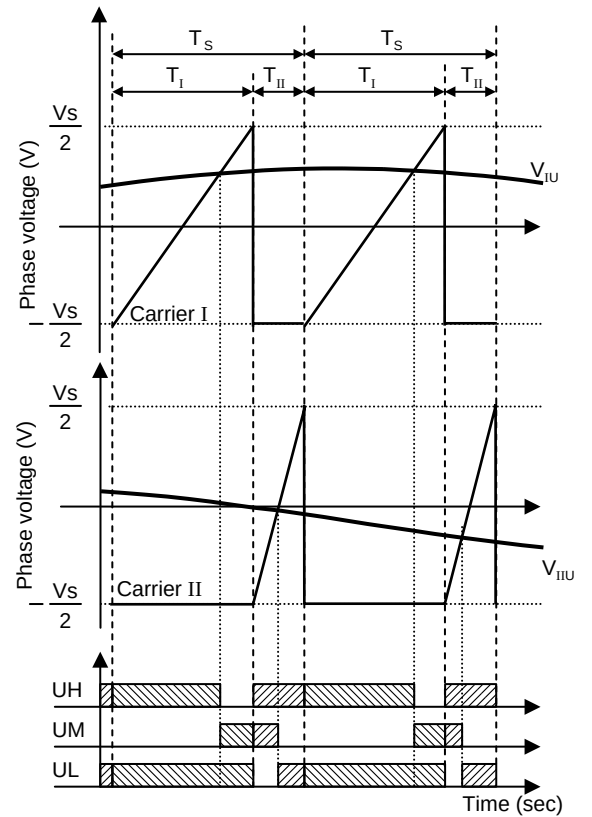


Fig. 7 Enhanced PWM control in 3-level 2-outputs inverter

II in this multi-level system. This indicates that current for two motors can be controlled simultaneously and independently.

Figure 10 shows the transient response of the three-phase current waveforms of three-level two-output inverter. The results show that the system is able to respond to the command value, changing the frequency of the output current of inverter I from 200 to 400 Hz, and the amplitude of the output current of inverter II from 25 A to 50 A.

3.3. Power Conversion Efficiency Characteristic

Figure 11 shows the power conversion efficiency characteristic of the three-level, two-output inverter using

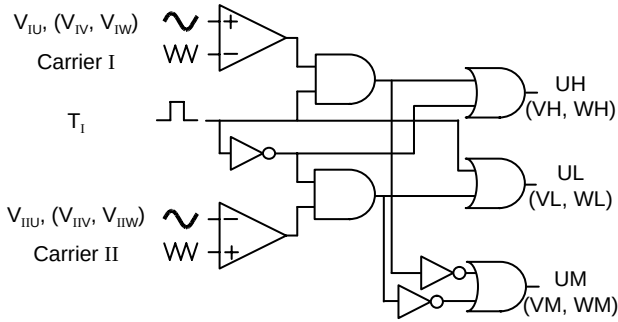


Fig. 8 3-level 2-output inverter control logic

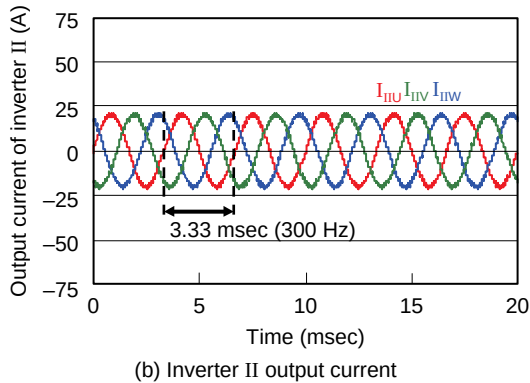
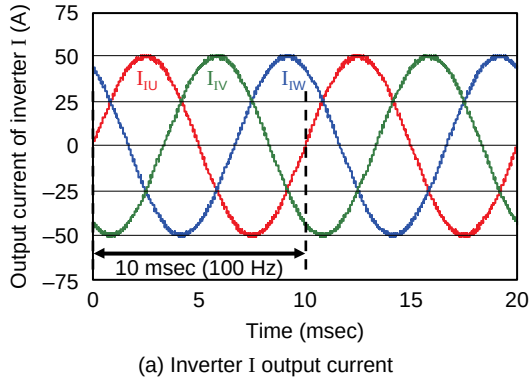


Fig. 9 Steady-state operation of 3-level 2-output inverter

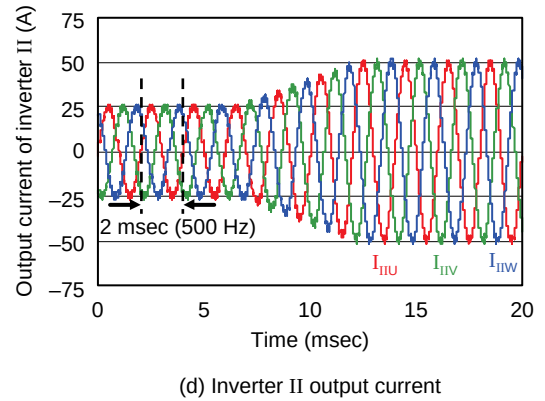
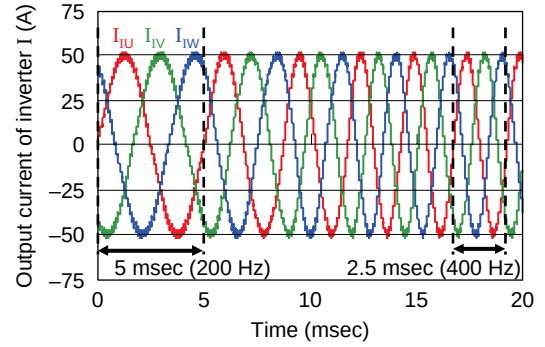
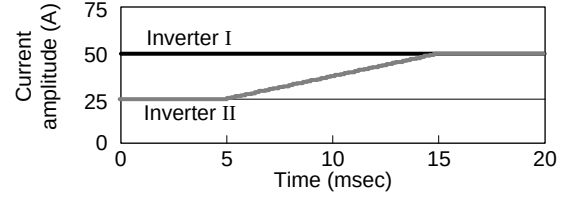
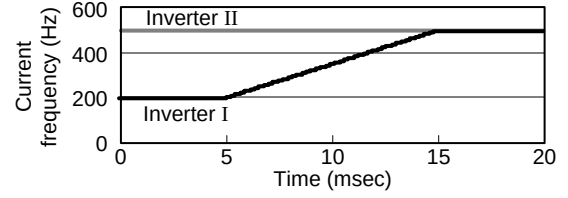
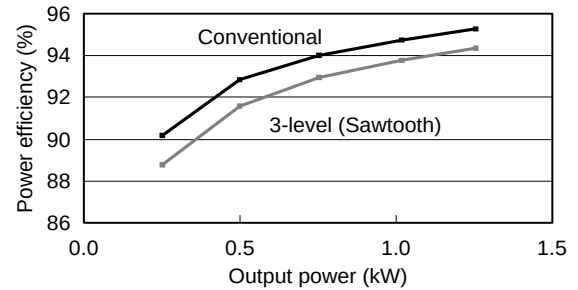


Fig. 10 Transient characteristic of 3-level 2-outputs inverter



sawtooth-wave PWM control and a conventional two-parallel two-output inverter. Figure 12 shows the results of a power loss analysis for both inverters at 1 kW. MOSFET (100 V, 75 A) were used as the switching devices in this analysis of power conversion efficiency. Both switching loss (ON loss and OFF loss) and conduction loss ($I_D \times V_{DS}$, $I_S \times V_{SD}$) were taken into consideration. A DC source voltage of $V_s = 48$ V and a motor speed of 1000 rpm were set as the operating conditions for the inverter. Phase and amplitude were identical for motors I and II. Switching loss was calculated from the switching loss characteristic in relation to voltage and current, and conduction loss was calculated from the ON voltage characteristic in relation to the current.

Figure 11 shows that the power conversion efficiency of the new inverter is lower than that of the conventional inverter across the entire operating power range. The results of the analysis of power loss in Fig. 12 indicate that conduction loss is the main factor in the lower power efficiency of the new inverter since there was an increase of 0.8 W in switching loss and 9.8 W in conduction loss for the proposed inverter. Three-level two-output inverter indicates increased conduction loss, because it employs a larger number of switching devices in the current pathways than a conventional inverter (three devices as opposed to two). The phase currents for two motors pass through each switching devices at the same time. Therefore the switching loss increases by 0.8 W despite the reduction in the amount of switching.

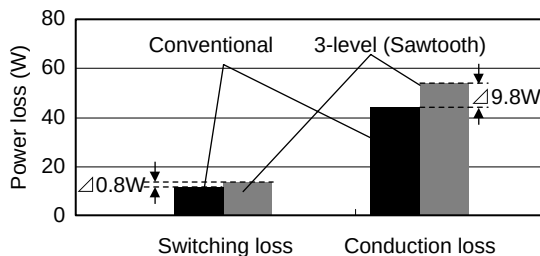


Fig. 12 Power loss analysis of 2-output inverter

4. Conclusion

The research discussed in this paper proposed a previously unconsidered form of PWM control that enables to drive multiple motors simultaneously and to reduce the amount of switching devices. The viability of this form of control was demonstrated by a simulation analysis. The results of this verification showed that the frequency and amplitude of the three-phase output current could be controlled independently for each motor, demonstrating the validity of the new system. However, the analysis indicated that there are also limitations on

output voltage in multi-level inverters like multi-phase inverters. The carrier waveform was converted to a sawtooth wave in order to reduce the amount of switching, and an analysis showed that switching loss was similar to that of a conventional inverter. The analysis results of power conversion efficiency showed that the increase in the number of switching devices in the current pathways had resulted in an increase in conduction loss and that switching devices with good conduction loss characteristics would be suitable for use with this method. A current phase difference between inverters I and II affects conduction loss. Adjusting the current phase difference may be necessary to enhance efficiency.

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■ Author ■



Manabu KUROKAWA

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Editorial Afterword

Four years have passed since I was first given the responsibility of providing guidance to our engineers in writing technical papers. My first impression, when I took up this position, was that some first-time writers of technical papers focused too much on the type of flow of information that one would associate with a PowerPoint presentation, and that they were not able to appropriately organize their thoughts into the form of a written argument. "Well, that's the nature of the modern age," I thought, and went right back to square one and convinced the authors of the importance of transmitting their ideas to others in writing.

Then, as I read paper after paper while pondering how to make them clearer and more logical in their presentation, I came to realize that it wasn't just a matter of modifying a given expression, but rather, I felt the need to fully engage in the content of the papers, and experienced the excitement of creating a paper in discussion with its author. The thought processes of our authors are extremely impressive; drawing these out to the fullest possible extent is the real thrill of being an editor. And from now on, we must consider what people coming to oral presentations are seeking. How should we aim to communicate in our presentations? What I want to create is an opportunity for people to come together to discuss ideas.

(Eitaro KOYA)

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CustomerService@sae.org
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