

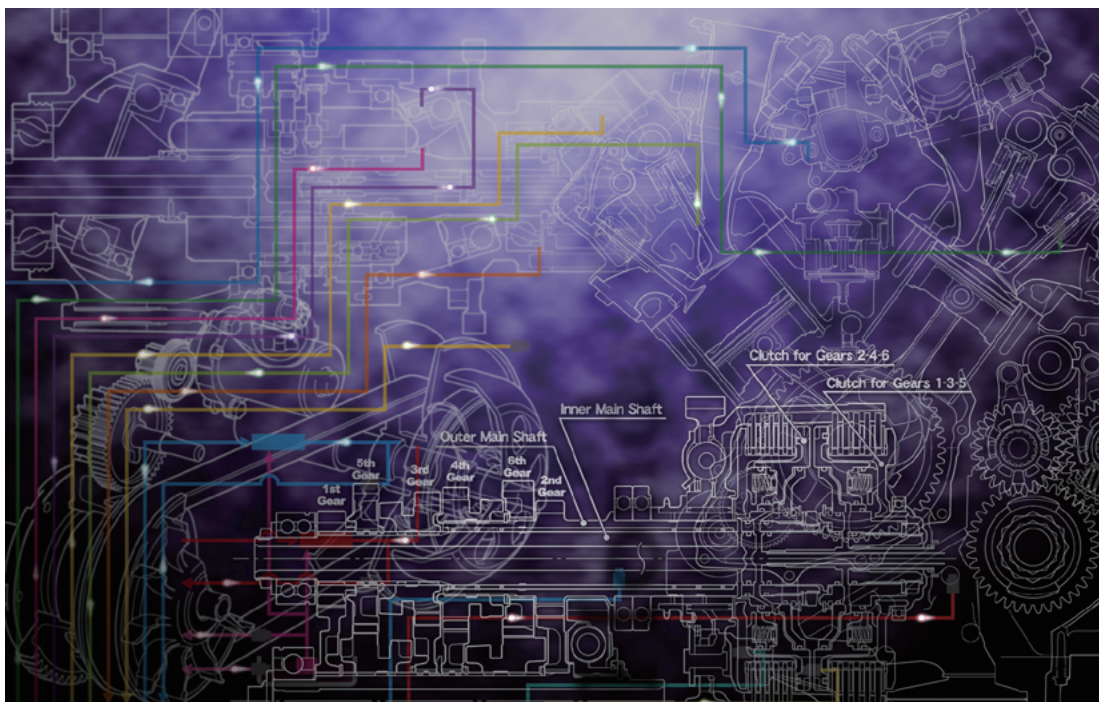
Honda R&D

Technical Review

Clutch for Gears 2-4-6

Clutch for Gears 1-3-5

April 2010
Vol.22 No.1



Comment on the Cover

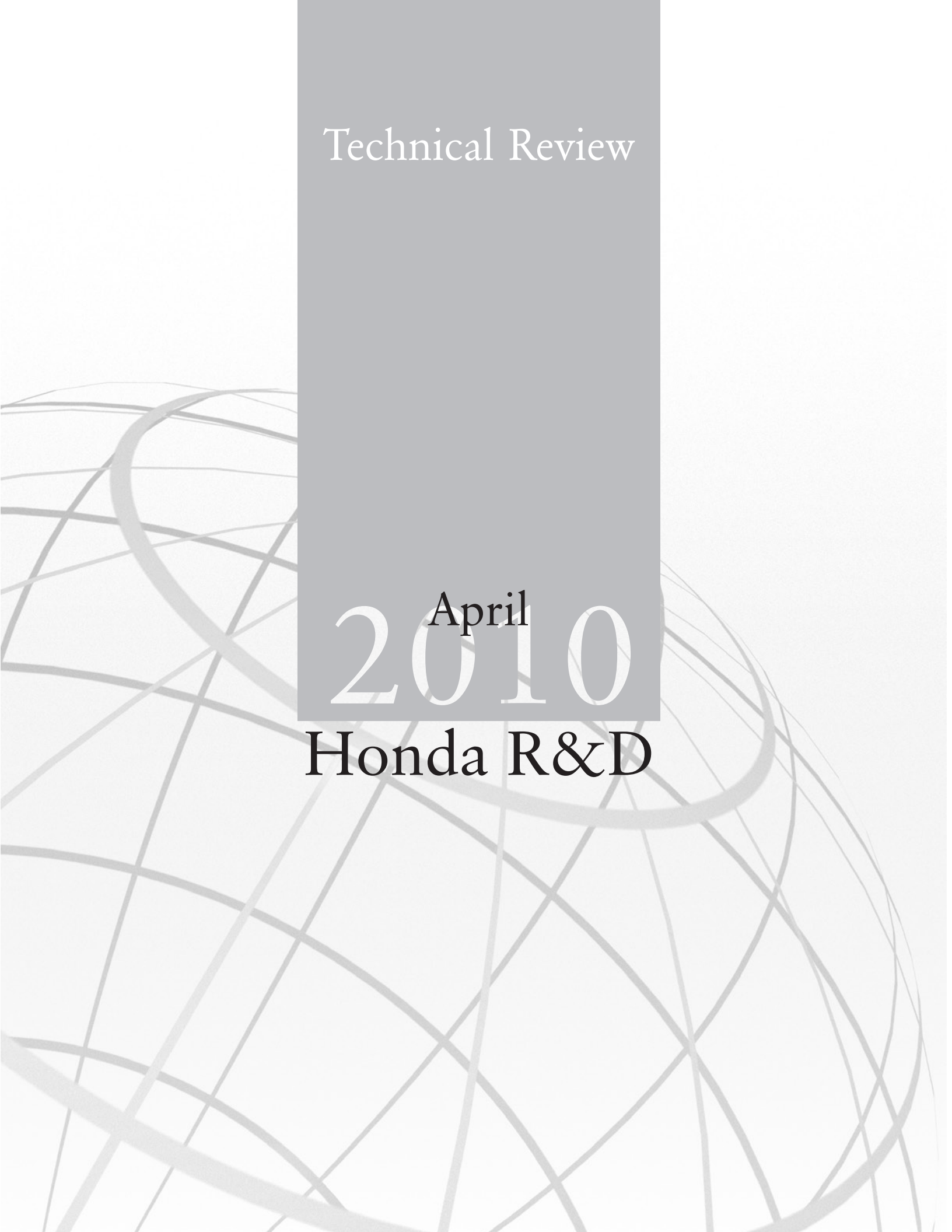
AT technology has evolved at a breakneck pace since Honda produced its first AT, for a two-wheeled scooter, some 55 years ago. Today, the protection of the environment is a major global concern, and engine performance has advanced dramatically from the environmental perspective.

The cover design for this issue of the Technical Review has a certain fantasy element, and is our attempt to provide an image of what could be when designers produced an “AT world,” bringing new engine configurations into being.

As a Honda employee, but also a Honda fan, I eagerly await the new technologies that the future will bring.



Motorcycle R&D Center
Takamaru Hoshino

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

Technical Review

April
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Honda R&D

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Styling Design of STEP WGN

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ABSTRACT

The newly released fourth-generation STEP WGN is the result of a full model change based on the concept of a “minivan maximizing comfort for everyone.” It was developed as a product creating lifestyle richness with greater functionality and space than users expect in a “5 number” (small class) utility minivan.

“Friendly Max” was chosen as the keywords for the design development.

The STEP WGN’s exterior design was conceptualized for spaciousness, ease of use and simplicity. Honda has achieved a “Friendly Max Exterior” that is at once functional, yet is simple and warm with rich surfaces, projecting fenders and a lively-looking form taking advantage of the minivan’s square frame. It gives the impression, at first glance, of a large space and the generous amount of glass surface-area evocative of the joy of driving.

The interior design uses the same concepts of spaciousness, ease of use and simplicity to achieve a “Friendly Max Interior” that embodies both functionality and friendly-feeling qualities with a form that projects both space and usability at first glance, and a display and operating system that anyone can see and use with ease. Color designs, moreover, were chosen to draw out these characteristics.

1. Introduction

The first-generation STEP WGN debuted in 1996 as a new form of family car to let families enjoy their weekends to the fullest. It became the first “5 number” (small class) FF utility minivan and created a new market in Japan.

Subsequent generations of the STEP WGN have adopted a “low floor/low center of gravity frame” that retains that same interior spaciousness while lowering the floor and enhancing cruising stability and day-to-day usability.

The latest full model change was conceptualized to take advantage of the low floor/low center of gravity frame and thereby create a tool with remarkable convenience and friendly-feeling offering maximum spaciousness and day-to-day usability in a 5 number minivan (Fig. 1).



Fig. 1 Approach to development

* Automobile R&D Center

Taking “Friendly Max” as the design keywords, lifestyle richness was sought to be created in the new STEP WGN for the target family users by putting them front and center under the concept of spaciousness, ease of use and simplicity (Fig. 2).



Fig. 2 Design concept

2. Package Layout

As a convenient tool to enrich family life, the STEP WGN aims to provide maximum space and excellent usability in a 5 number minivan based on a low floor/low center of gravity package.

The package retains the door openings that rise from a low floor with no steps as used in earlier generations of the STEP WGN, while setting an overall height so that the interior would be high enough for a small child to walk without bending down between the first and third rows. Overall length is 4690 mm, the maximum for a 5 number vehicle. The minivan offers the maximum spaciousness in a 5 number FF utility minivan with 8 seats (Fig. 3).

The wider, horizontally toned window surface areas and increased head clearance (accomplished by an increase in overall height by 45 mm compared to the previous model) create an open feel where every seat offers a good view as well as comfort.

The back-edge position of the second row has been set

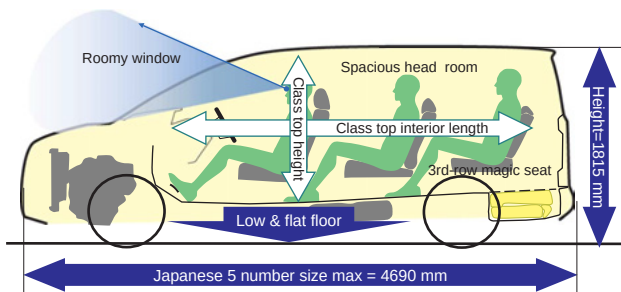


Fig. 3 Vehicle package highlights

100 mm farther back than in the previous model, and the rear-seat door opening was given bigger dimensions to ease access for young children and elderly passengers (Fig. 4).

Trunk room is a vital function of a minivan, and the STEP WGN aims to provide not only storage space for everything from daily purchases to bicycles, but a square area that functions as a new “living space,” along with a first-in-class third row that tucks into the floor (Fig. 5).

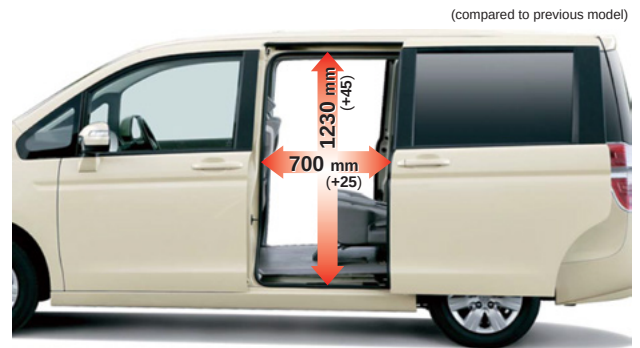


Fig. 4 Easy rear sliding door access



Fig. 5 For my family room

3. Exterior Design

To achieve the “Friendly Max Exterior,” Honda aimed to use the concepts of spaciousness, ease of use and simplicity to create a new frame not previously available in a 5 number utility minivan.

The basic motif was to surround passengers in the long-loved “simple” style consisting of a square “box section” which, with just a glance at the exterior, gives the impression of being able to hold plenty of cargo, and a “glassy-surround cabin” enclosed by large, horizontally toned windows to keep the interior bright and make driving feel easier and more enjoyable (Fig. 6).

The “dynamic form” based on this frame that gives the impression of a refreshing and pleasant drive is brought to life by a roof line with a smooth profile that looks like wings in cross-section, a rear bumper that angles upward,

and character lines that run up to the edge of the tailgate. In addition, the use of wide grooves to make the front and rear fenders appear to stretch beyond their actual dimensions lets the STEP WGN express a “wide tread” for the feeling of stable and confidence-inspiring driving (Fig. 7).

By working on such characteristics, including carefully examining the processing of pliant-tension, high-quality surfaces and corner parts, the STEP WGN aims to express a new sense of functionality giving a warmer, more people-friendly impression of a family car while remaining functional and simple.

The front design is built around a solid hood that has the appearance of a strong block of material, the chrome plated grill and headlights are lined up horizontally to seem wide and intrepid, and the large, round, multi-reflector headlights with four lamps aim for a bright, highly functional expression.

The rear design expresses strength and richness with a large three-dimensionally rounded rear window, a pliant-tension rear panel, and a rear bumper designed to appear as if it were in a higher position. Rear combination lamps wrap widely around the side while leaving a generous tailgate opening with the chrome plated housing and inner

lens cut expressing a high-class look. (Figures 8 and 9 show the final rendering.)

The “Friendly Max Exterior” was created as a new design for a 5 number utility minivan that is simple and people-friendly even as the frame gives the appearance of spaciousness, ease of use and functionality. (Figures 10 and 11 show the final exterior design.)



Fig. 8 Final rendering



Fig. 9 Final rendering



Fig. 10 Final exterior



Fig. 11 Final exterior

Friendly Max Exterior

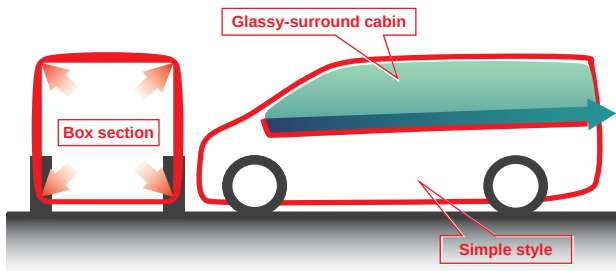


Fig. 6 Exterior design highlights (1)

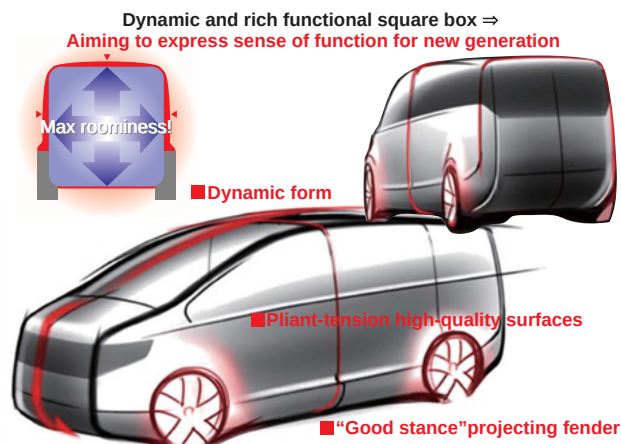


Fig. 7 Exterior design highlights (2)

4. Interior Design

To make the “Friendly Max Interior” a reality, Honda designed for spaciousness, ease of use and simplicity while pursuing all the functionality expected of a 5 number utility minivan (Fig. 12).

The instrument panel aims to provide utility (such as storage), as well as display and operation interfaces. The mass of the upper part is offset with the left side brought closer to the front row and the right pushed forward away from the front row. The space thereby created can function as a tray with the part brought closer to the driver housing the controls to the audio and navigation systems (Fig. 13).

Displays for meters, air conditioner and the like are set farther forward away from the driver, which makes them easier to read instantly and helps to create wider cockpit space. When the vehicle is being driven in a fuel-efficient way, the light in the center ring of the meter changes to green providing instantly comprehensible information on fuel economy.

Another aim was to achieve more intuitive operation of the close-up air conditioner interface by making the switch position consistent with the display position (Fig. 14).

The seats pay attention to lateral space while giving the appearance of a spacious, comfortable place to sit. The third row of seats in particular was designed to provide the function of cargo space when tucked neatly into the floor (Fig. 15).

The door lining was given a clean horizontal tone to increase the sense of space from front to back while providing a convenient place to stow away goods.

To express unprecedented spaciousness, the sky roof with a large opening and the plain wood-like flooring were utilized for greater brightness and pleasant feel (Fig. 16).

Taken together the overall interior space succeeds in creating a “Friendly Max Interior,” a functional space with simple design and all the spaciousness and usability one would expect from a utility minivan (Fig. 17).

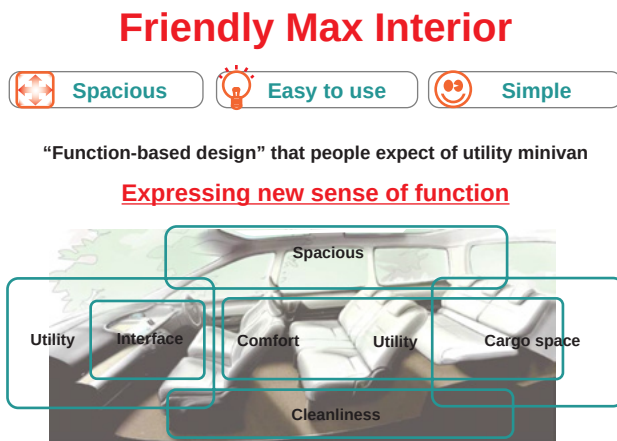


Fig. 12 Approach to interior design

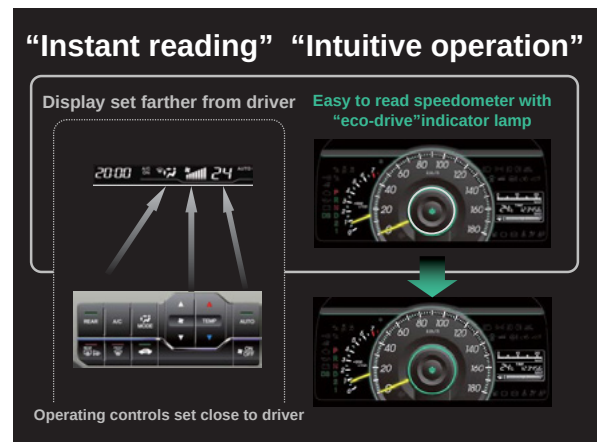


Fig. 14 Driver's interface

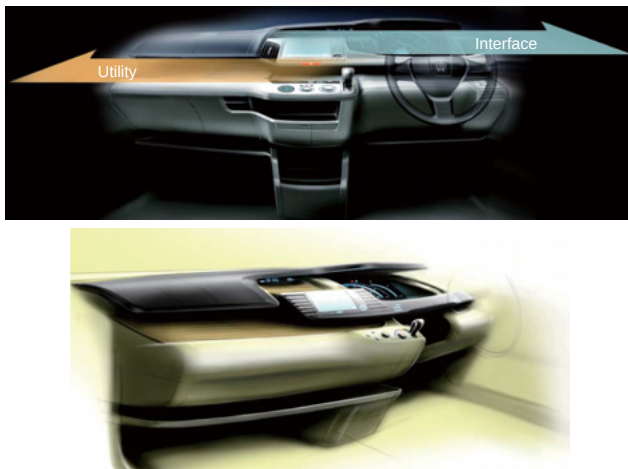


Fig. 13 Instrument panel

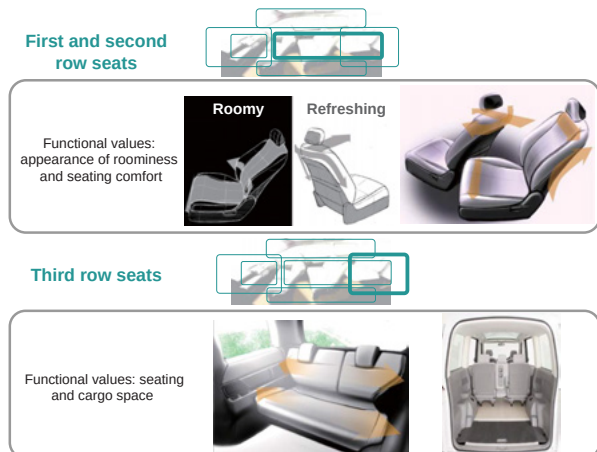


Fig. 15 Seats

Cabin
floor

Fig. 16 Roof and floor



Fig. 17 Final interior

5. Color Design

5.1. Exterior

The styling of the STEP WGN is intended to be warm even as the auto is the functional result of a “box-shaped, people-friendly form with pliant-tension quality surfaces.” The theme of color development was to use the synergy between these characteristics and color to create a tool that is even easier to use and more friendly-feeling.

The theme colors chosen were Premium Hidamari Ivory Pearl and Aozora Blue Metallic: colors that are gentle and fit naturally into people’s family lives.

Expressed with three-coat pearl, Premium Hidamari Ivory Pearl shares some of the composition of bright existing colors that draw out the warmth and brightness of a spot of sunlight while following expressive surface changes that reveal changes in color.

This emphasizes the vehicle’s characteristic warmth and makes the vehicle look larger.

Aozora Blue Metallic simply emphasizes the pliant-tension quality of the new STEP WGN’s forms. Aozora

Blue Metallic has a feeling of depth and a look of transparency which brings out the color of a perfectly clear, blue sky (Fig. 18).

The lineup of other body colors includes Super Platinum Silver (with greater brightness and metallic feel than standard Light Silver), Deep Bordeaux Pearl (refined yet showy), Crystal Black Pearl, White Pearl and Polished Metal Metallic (gunmetal color) (Fig. 19).

5.2. Interior

The aim of the interior’s design was to create “a bright and comfortable space with natural elements,” presenting a spacious area suitable for many different uses.

The colors of the interior were coordinated by contrasting the coloring of the overhead space with that of the utility space below. The color scheme projected images of natural elements throughout the interior, with seat cover and ceiling colors, for example, emphasizing the interior spaciousness of the vehicle (Fig. 20).

The interior colors selected were Gray and Black. Bright Max Ivory pillars and ceiling areas impress the occupants with the interior’s spaciousness enhanced by the floor’s Dark Gray and Black.



Fig. 18 Inspirations for exterior colors

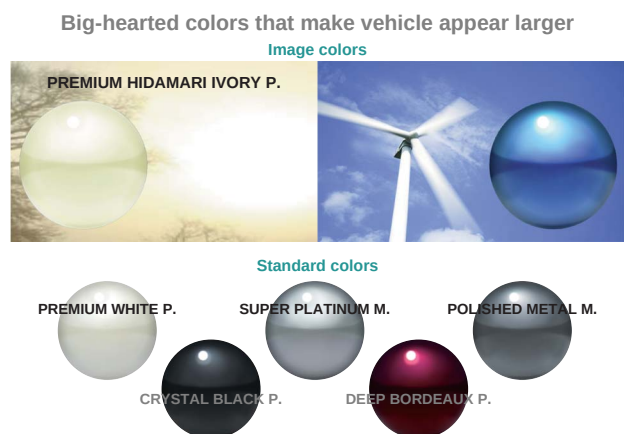


Fig. 19 Exterior color options

The grain of the overall surface evokes an image of layers of sunlight filtering through trees, with the synergy between it and the color coordination presenting a neutral, easy-to-use space (Fig. 21).

Max Ivory was used on the ceiling to maximize the brightness of the overhead space thus giving the occupants a feeling of spaciousness. This color choice makes the light from outdoors look brighter and makes the interior space seem larger than its actual size.

Light Gray and Black were chosen as colors for the seat fabric. Light Gray seats offer two tones of gray, one for the seating surface and one for the back, which brings variety to the space and makes it easier to use.

Patterning was chosen to bring out the interior design's characteristic horizontally toned form and sense of spaciousness, while the horizontal tone itself evokes an image of the wind. The base design is processed to add a sense of depth to the interior while resisting soiling.

The suede tricot fabric is pleasant to the touch.



Fig. 20 Interior inspirations



Fig. 21 Coordinated interior color scheme

6. Conclusion

The STEP WGN's debut pioneered a new genre in Japan known as FF 5 number utility minivan. Since then it has continued to offer solutions for family happiness and fun. In its 13 years of evolution, Honda has created minivan value beyond customer expectations with unprecedented spaciousness, usability and economy. The new, people-friendly design of the fourth-generation model brings the high functionality of the vehicle into sharp focus and enhances product appeal.

STEP WGN aims to express a new sense of functionality including: the package layout that offers open feel unprecedented in the class within the frame of 5 number utility minivan and usability as typically seen in the third row of seats that tucks into the floor; the exterior design that gives the impression of a refreshing and pleasant drive in a box shape; the interior design that projects spaciousness and functionality; Color designs that were chosen to draw out the vehicle characteristics with friendly-feeling and comfort.

Honda believes that more customers will come to accept the new STEP WGN as a vehicle that puts the family users front and center while working with them to realize bigger dreams.

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Styling Development for 2010 Acura ZDX

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ABSTRACT

In keeping with Acura's core value of being an innovator of vehicle segments, the ZDX team identified an opportunity to create a segment that would satisfy a trend-setting luxury customer and raise Acura's prestige.

The ZDX styling concept of "Suspenseful Emotion imagining an epicurean [luxurious] journey" provides a driving and living experience of emotion and style. The "Motion Surfacing" exterior concept gives the vehicle an immediate and strong visual impression. The "Two plus Freedom" packaging concept combines characteristics of sport sedans, coupes, and SUVs to provide spirited, coupe-like driving and strong all-road performance. The "Destination Hotel" interior design concept extends the living experience of a modern and luxurious getaway.

ZDX's coupe proportions featuring large wheels and tires and a small, tapered cabin allowed stylists to create very dramatic shapes, such as the deep shoulders over the rear tires. The all-glass panoramic roof maximizes the feeling of interior openness while minimizing exterior size. The interior features wrap-around leather surfaces, ambient lighting, and a center command area called the "Blended Technology Monolith," which blends the climate, entertainment, and navigation systems.

ZDX's purpose of inviting a couple to explore and drive in a spirited fashion led to a new mix of sport SUV driving and utility combined with the comfort and seating of a sport sedan or coupe.

1. Introduction

Acura has established itself in the US market as a luxury brand with balance between performance, comfort, and value by innovating new vehicles in luxury segments or by creating entirely new vehicle segments. Luxury vehicle brands have divided their lineups into the main categories of stylish coupes, versatile sedans, and flexible SUVs. For example, Acura TSX and TL impress as sport sedans while Acura RDX and MDX created and continue to define their niche as sport SUVs.

Acura set forth with the ZDX project to add to their lineup of sedans and SUVs; however, in keeping with the Acura's core value of being an innovator of vehicle segments, the ZDX team identified an opportunity to create a new segment that would satisfy a trend-setting luxury customer and raise Acura's prestige.

This paper examines the styling concept development process from initial market understanding to final vehicle creation, and describes the achievement of key components, including the deep one-piece rear quarter panel, all-glass panoramic roof, disappearing instrumentation, and stitched leather interior surfaces.



Fig. 1 2010 Acura ZDX



Fig. 2 Acura ZDX positioning concept

* HONDA R&D AMERICAS, INC.

2. Vehicle Concept

2.1. Target Customer

The design team targeted a customer of a certain mindset, versus age or gender. The ZDX target customers are active, adventurous individuals who appreciate cutting-edge design, innovative thinking, and fresh ideas. They are self-assured trendsetters and understand good design beyond preconceived categories. They value sleek, edgy, exotic or emotional design and find appeal in craftsmanship and high technology.

ZDX customers are in a life stage without children. They might be considered coupe category buyers since their key purchase reasons include styling and performance; however, due to their active lifestyle, they find great value in vehicles with utility and capability. They enjoy weekend getaway trips, traversing twisty mountain roads, and golfing with friends. Such activities require their vehicles to handle tough urban driving conditions year-round.

2.2. Styling Concept

The ZDX styling concept was “Suspenseful Emotion imagining an epicurean [luxurious] journey.” The designers imagined the emotion one feels when approaching this vehicle as suspenseful, that is, a mixture of anticipation, excitement, and not knowing what lies around the corner. They desired to capture the feeling of suspense and excitement a skier feels at the moment just before launching down a slope. ZDX targeted the idea of a vehicle that invites two people to join in a driving and living experience of emotion and style. The experience may be one of just two people enjoying an invigorated drive on different roads, a trip where spontaneous shopping may occur, or a planned trip with another couple to a special destination. Glancing at the ZDX, the target customer would dream of partaking in these types of positive experiences and activities, which the design team called the “Epicurean Journey.” This styling concept set the tone for the exterior, packaging, and interior design concepts.

2.3. Exterior Styling Concept

The exterior concept of “Motion Surfacing” required the vehicle to make an immediate and strong visual impression. Since the overall styling concept focuses on

the feeling of suspense when a person walks up to or sees the vehicle, the overall proportion needed to make a rapid impression. The secondary visual sensations of line, shape, and detail needed to enhance the overall proportion and draw on the customer’s experiences and emotions. The designers wanted all of the surfaces to be interactive with and generate from adjacent surfaces as the viewer moved around the vehicle and viewed it from new vantage points. The shapes and lines will in effect never end. They will just lead to new, dynamic surfaces.

The ZDX target customer is rich with experience in many types of vehicles; therefore, they are able in a glance to categorize a vehicle as SUV, sedan, or sport coupe. ZDX designers wanted to draw from the customer’s experiences with these different types of vehicles and combine the advantages offered by all three. For example, traditional SUV shapes and proportions seek to maximize passenger and cargo volume while providing all-road strength and security; sport coupes convey personal and dynamic character through a low and wide stance, small passenger compartment and windows, and smooth, flowing shapes; and sedans focus on passenger comfort with four equal seating areas and a separate cargo area.

2.4. Packaging Concept

The “Two plus Freedom” concept guided the packaging design. The packaging sought to combine some of the stand-out characteristics of sport sedans, coupes, and SUVs to provide the driver and passenger with spirited, coupe-like driving and strong all-road performance.

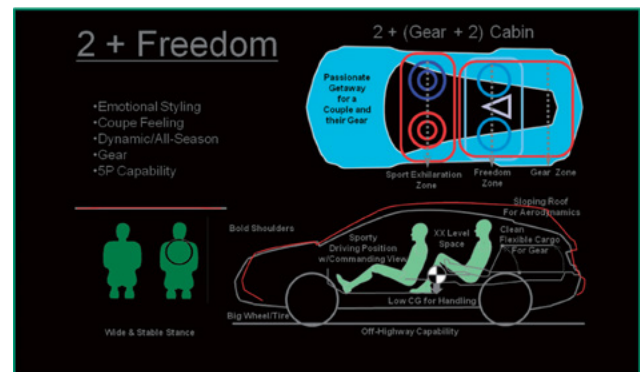


Fig. 4 Concept document describing styling shape and proportion of ZDX compared to existing vehicle

Image not displayed for copyright reasons.

Fig. 3 Key images of ZDX usage environments

The cockpit layout provides a coupe-like feel for the front passengers. The rear area provides sport sedan seating for three adults or converts into luggage and utility space. To access the seating areas with sedan ease, four doors were required, and a rear hatch chosen for access to the utility space. The vehicle's high, protective beltline and low roof create a secure feeling from inside the cockpit. However, the passengers do not feel confined due to the integration of an all-glass roof. This layout supports the concept of a personal, sport luxury vehicle that inspires spirited driving while adding an adventurous feeling. The designers imagined the confidence of sitting higher in traffic than a coupe when looking out and focusing on other vehicles, yet having the emotional experience of a coupe when looking forward over the front fender forms or when looking rearward and seeing the haunches of the rear shoulders lifting over the large wheels and tires.

2.5. Interior Styling Concept

The interior concept was “Destination Hotel,” which centers on the feelings of adventure and luxury for the high-end customer. The adventurous couple who enters the “destination hotel” is rewarded with a modern design combined with seductive textures and an exhilarating personal ambiance. The designers wanted the ZDX interior to be an extension of this hotel living experience.

The interior highlights the dominant front passenger area by surrounding the front passengers with rich leather,

advanced ambient lighting, and modern technology. The key to this is the center “Blended Technology Monolith” area where climate, entertainment, and navigation systems blend and then disappear into crafted fine metal and leather materials.

3. Design Achievement

Achieving ZDX's exterior, packaging, and interior concepts required close collaboration between Styling, Engineering, and Manufacturing. This section looks at the key design breakthroughs from Styling's perspective. Critical to achieving the ZDX's styling and engineering breakthroughs was the clear communication of the designers' styling and package concepts to the engineers. The designers felt that the key to communicating the concepts was having the engineering team personally see a styling mock-up of the ZDX while viewing the concepts. This made clear the ways in which ZDX was different than SUVs, sedans, and coupes, and helped them understand how vital uncompromised styling was to the character of ZDX.

3.1. Deep Dimension Body Shapes

ZDX's coupe proportions featuring large wheels and tires and a small, tapered cabin allowed the stylists to create very dramatic shapes, such as the deep shoulders over the rear tires. The idea was to express the “Suspenseful Emotion” concept in the overall mass and



Fig. 5 Initial ZDX interior concept



Fig. 7 Description of ZDX form



Fig. 6 Interior concept set tone for ZDX interior mood



Fig. 8 Description of ZDX stance

stance of ZDX. The designers imagined a sling shot pulled back in tension and in ready. They desired the overall appearance of the form to be uncompromised by the realities of passengers, door openings, and functioning parts. Yet they knew that ZDX needed to have comfortable seating and utility space for four.

For ZDX to accommodate four adults comfortably, four independent doors were necessary. The goal was to have the doors designed into the overall form so as not to compromise the deep, dramatic shoulders. In addition, the front rear door side pillar was positioned as far rearward as possible, similar to the position in a two door, to mimic the driving environment of a sport coupe. To allow comfortable passenger ingress, the cut lines for the doors needed to be as far fore and aft in the cabin area as possible and flow with the side window and body design lines. This design led to very deep stamping draw conditions in the rear cutline area of the rear door and wheel opening.

To achieve the crisp body line on the flowing rear wheel area and allow for a door cut line immediately forward of the wheel opening presented stamping challenges. The depth of stamping draw from the furthest out point on the rear wheel arch form into the door opening seal line was much greater than any previous construction. The team did not want to compromise the styling, so they did not request major shape changes to ease the stamping difficulty.

To achieve the design, engineers investigated two



Fig. 9 Rear wheel opening and shoulder area of ZDX has deep shapes with crisp edge



Fig. 10 Stamping achievement of deep draw ZDX rear quarter side panel

development plans concurrently. The primary plan, if successful, would result in a one-piece rear quarter panel. Styling and Engineering worked together to relieve some of the depth of draw and acute angle conditions that were making the stamping improbable. Using stamping simulation tools, engineers were able to achieve a panel design and stamping process plan that showed probability of producing the deep draw while maintaining reasonable scrap rates. To confirm the plan, dies were made to test the stamping before committing to the one-piece panel.

If the one-piece stamping could not be achieved, the backup plan was to use known technology to achieve the shape in two stamped pieces; however, this would add weight, cost, and complexity to manufacturing. In the end, the dramatic deep shoulder styling was achieved with a high-quality one-piece panel that helped meet weight and cost targets.

3.2. Panoramic Roof

ZDX's overall purpose of inviting a couple to explore and drive in a spirited fashion led to a new mix of sport SUV driving and utility combined with the comfort and seating of a sport sedan or coupe. This results in a more reclined seating position, but higher off the ground like an SUV and a small tapered cabin like a fast back coupe. While these characteristics satisfied many of the emotional and dynamic targets of the customer, they raised functional issues with visibility and passengers' perception of freedom versus confinement. The panoramic roof was pioneered to meet these seemingly contradicting needs and wants.

The goal of the panoramic roof was to maximize a feeling of interior openness while minimizing exterior size. It needed to provide maximum up, side, front, and rear visibility while maintaining the coupe cabin form. In keeping with the styling image of simple flowing forms, the roof could not be overflowing with visual detail and complexity.

The resulting roof is a smooth glass surface from base of the windshield to the rear license area that is disrupted only by a performing rear air spoiler and four minimal on-surface cut lines between panels. In plain view, the glass surface extends from left to right roof side rails without cuts. The visibility from within the cabin was maximized and tuned to create up-and-outward visibility.



Fig. 11 Long and wide expanse of glass of ZDX panoramic roof



Fig. 12 The ZDX panoramic roof allows unparalleled upward and outward visibility experience



Fig. 13 Sun shades roll out from center roof beam in fore and aft directions to provide protection to front and rear passengers

The mechanisms and structures of the sliding center section were minimized and moved outboard to create an amazing open roof experience. To protect passengers from excess sunlight, the roof glass provides 70% sun protection, and powered sun shades span out from the center transverse roof beam that block all but 3% of sunlight.

3.3. Disappearing instrumentation

Achieving the center control stack “Monolith” theme was an interesting blend of engineering creativity and styling dream. The styling team envisioned the front passengers in personal wraparound areas of fine leather, and in between them would be a center command area referred to as the “Blended Technology Monolith.” This area would allow passengers to interact with many

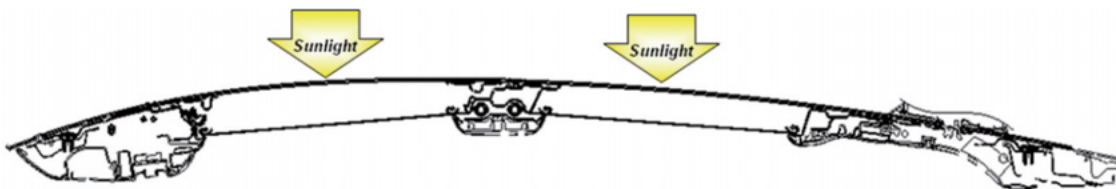


Fig. 14 Section of panoramic roof and sun shade mechanism



Fig. 15 Images of Blended Technology area in off and on conditions used to demonstrate idea during development of ZDX

functions of the vehicle, such as climate, sound, and navigation, but not be overly complicated or cluttered.

Engineers had already been working on a process to have a simple surface that could transform into an interactive display when lit. They had no specific product to apply this technology until they learned of the idea the ZDX designers had in mind for the Blended Technology Monolith area.

3.4. Stitched Leather Interior Surfaces

Creating the wraparound leather surfaces to achieve the interior design concept required new business and manufacturing processes. The design idea was to have leather wrapped and stitched around the interior parts and appear seamless. This required the door and part cuts to be minimized and the leather texture, color, and stitching to be perfectly consistent and flowing from one part to another. Three suppliers, each with excellent experience in providing stitched leather parts, were chosen to make the interior leather surfaces.

In the past, appropriate matching of different parts from different suppliers was achieved by creating a master sample and distributing it to each supplier. However, a new level of matching accuracy would be required since the ZDX would have naturally varying leather and actual stitching. Since the suppliers had slightly different techniques to select and trim leather and used different materials and thread, the design team had to establish new verification processes to guarantee consistency.



Fig. 16 ZDX interior design sketch expressing wraparound shapes and hand-stitched leather surfaces

4. Conclusion

The ZDX defines a new vehicle class for a new class of Acura owner. ZDX combines the emotional and dynamic character of a sport coupe with the utility and confidence of sport sedans and SUVs. The flavors of these luxury categories are blended in an incredible way to truly delight an active, discerning customer pursuing life.

Achieving the exterior, packaging, and interior concepts posed challenges that required innovation and close collaboration between Styling, Engineering, and Manufacturing. The key design achievements included the following:

- (1) The deep rear quarter panel required a depth of stamping draw greater than any previous construction. Using stamping simulation tools, engineers were able to achieve a one-piece panel design that met the deep draw requirement while maintaining reasonable scrap rates.
- (2) The reclined seating position and small tapered cabin raised functional issues with visibility and passengers' perception of freedom versus confinement. The all-glass panoramic roof allows unparalleled upward and outward visibility that maximizes the perception of interior openness while minimizing exterior size.
- (3) The Blended Technology Monolith allows passengers to interact with many functions of the vehicle, such as climate, sound, and navigation, but not be overly complicated or cluttered.
- (4) The wraparound interior leather surfaces required a new level of matching accuracy to accommodate naturally varying leather and hand-stitching. The design team established new processes to guarantee consistency across multiple suppliers.

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Development of New Model CR-Z Honda Small Hybrid Sports Car

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ABSTRACT

The 2009 model year INSIGHT platform and hybrid system have been used to develop a fun car with a new generation hybrid configuration.

A sporty feeling was given by a 1.5-L Integrated Motor Assist (IMA) engine in combination with a six-speed transmission. The “Low, short, and wide” design emphasized its nature as a sport package, and its advanced sporty form was further realized by Honda’s very first light-emitting diode, daytime running light (LED DRL) headlights and other such advanced items. A sporty quality was produced in the interior by means of advanced three-dimensional gauges and smart sport seats, and a “soft feel” instrument panel and metallic film were used to enhance the quality. The handling was made agile for driving performance, and the adoption of a 3-Mode Drive System adapted to different driving situations realized a completely new feeling of acceleration and exhilarating driving performance.

The development achieved a high-level balance between driving performance and fuel efficiency. With a design that produces a sense of sportiness, an unprecedented small sporty hybrid car was created that possesses the new value of the IMA.

1. Introduction

The spread of ecological awareness due to global environmental issues, together with sharply rising gasoline prices, have lead to the intensification of worldwide demand for fuel-efficient vehicles. Meanwhile, the Honda brand has gradually been gaining name recognition in Europe over the past several years, while Honda had been waiting for a car that could serve to spur even further recognition. The proposal was therefore made to create a balance of fuel efficiency and sportiness in the kind of three-door small car that is popular in Europe. Such a new value vehicle would reinforce the company’s brand presence in Europe while also capturing the youth market and reducing the corporate average fuel consumption. At the same time, the aim was also to position this as a global car that could be marketed in North America and Japan, as well. This was how development of the CR-Z began.

2. The Concept

With the aim of creating a new appeal that would give form to originality and innovativeness and open the way

forward to a new era, the grand concept was formulated as “Advanced! Hybrid Café Racer” (Fig. 1). This would have hybrid technology as its core, and would seek to achieve a balance of Honda’s environmental stance and its sportiness, which can be considered an expression of the company’s DNA.

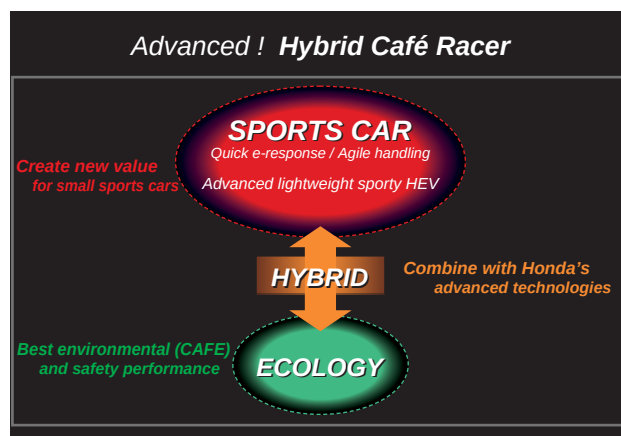


Fig. 1 Grand concept

* Automobile R&D Center

The name Café Racer originated with the Ace Café in the UK, where motorbike aficionados get together. “Café Racer” is intended to convey the desire for an urban sports car that people could boast about to their friends.

In order to create the new value of a small sports car for a new era, development was carried out with the following items identified as main topics:

- (1) Advanced sporty form
- (2) Balance of totally new acceleration and fuel efficiency
- (3) Quick and agile handling
- (4) Comfort in daily use

3. Advanced Sporty Form

3.1. Exterior Design

3.1.1. Design concept

“Advanced Sensual Bullet” was termed for the exterior design concept, which expresses the “massiveness” of highly concentrated power and the “dynamic feeling” of the chassis that thrusts up out of that massiveness. A large grill is positioned low down at the front, conveying the notion of the car’s low center of gravity. The blacked out front pillars and the highly curved front window form a clear, unified glass area that realizes advanced quality and good visibility. The deeply pared back front and rear fenders produce a dynamic feeling. The rear extra window, which forms a continuous line with the fenders, and the rear combination lights give a sense of breadth and powerful steadiness. The finished design yields a premium sports car with a stylish presence, which possesses the appeal of a sports car together with the advanced nature of a hybrid system and satisfies adult sensibilities (Fig. 2).

3.1.2. Form with low total height and short overhang

In order to realize the low total height that is a necessary condition for a sports car, there was an issue to be faced: how to satisfy the requirement for pedestrian protection performance while somehow lowering the engine hood height.

Removing the “torque up” resonator as well as altering the structure of the intake manifold and air cleaner reduced the engine height by 14 mm relative to that of the 2008 model year FIT, which was the base. The use of simulation to optimize the hood sheet thickness, optimize design surfaces and collision areas, as well as modify the connecting structure between the upper members and the damper base as shown in Fig. 3, enabled the rear edge of the hood to be lowered by 23 mm relative to the 2009 model year INSIGHT. As a result of these measures, the height of the car roof was reduced by 75 mm to 1350 mm compared to the 2009 model year INSIGHT (overall height including antenna: 1395 mm).

Optimization of the stiffener shape in the rear frame structure and the addition of cross members made the rear panel into an energy-absorbing structure. This

allowed the car to clear U.S. Federal Motor Vehicle Safety Standard (FMVSS) 301 concerning rear impacts while also making the rear overhang 225 mm shorter, with an overall length of 4080 mm, than in the 2009 model year INSIGHT (Fig. 4).



Fig. 2 Exterior design

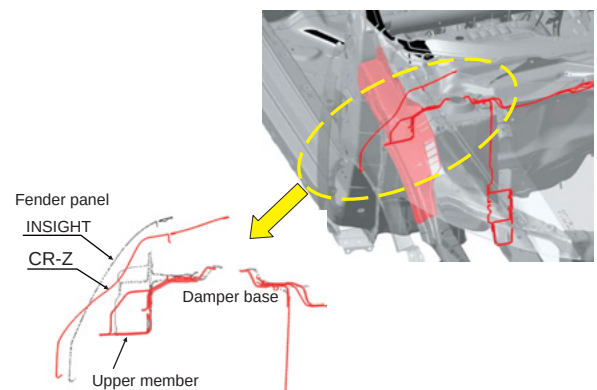


Fig. 3 Upper member

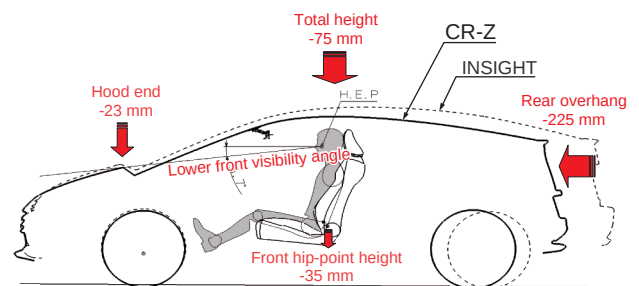


Fig. 4 Package design

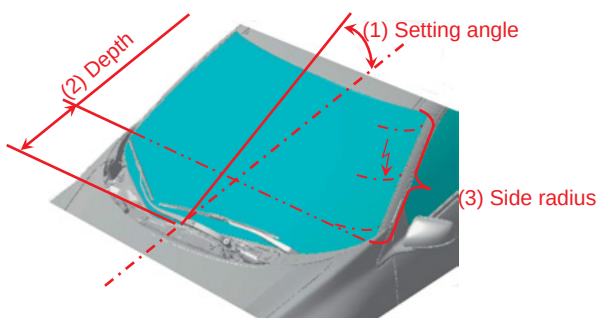
3.1.3. High curvature front window

It was necessary for the front window glass to have a high curvature in order to realize the cornering visibility and advanced form appropriate for a sports car.

When window glass is given a high curvature, distortion occurs at both ends of the glass. Moreover, the glass is set at an oblique angle, making this the most demanding shape with regard to distortion among all Honda vehicles to date (Fig. 5). Rather than gravity casting, therefore, the glass was manufactured instead using a stamping process that tends to cause less distortion to be generated. This was combined with repeated verification that the formability of the window was matched with its design conducted through molding simulations. As a result, a high curvature was achieved in the front window.

Garnish was also used on the front pillars in order to produce an impression that the front glass and side glass are unified. In order to enhance the unified impression of the front glass and the pillar garnish and to reduce wind noise, a flush surface front pillar that is integrated with the glass was adopted. This forms a water drainage channel by the combination of the edges of the glass at the ends with the front pillar garnish, and it achieves a balance of water drainage performance and wind noise reduction (Fig. 6).

Through the adoption of a high-curvature front window, good cornering visibility suitable for a sport car is obtained. Figure 7 shows front pillar visibility indices, with cornering evaluation points for visibility in a far-away direction on the vertical axis, intersection evaluation points for the scarcity of “blind spots” around the front pillar on the horizontal axis.



Model	(1) Setting angle	(2) Depth (mm)	(3) Side radius (mm)
INSIGHT	23.1°	66.3	1100 – 1600
CIVIC 2D	21.9°	60.0	1100 – 2150
ELYSION	27.0°	145.0	880 – 1140
FIT	26.0°	56.6	3000 – 4600
NSX	24.2°	176.0	800 – 1030
CR-Z	22.3°	143.4	600 – 1000

Fig. 5 Front window glass

3.1.4. Light-emitting diode daytime running lights (LED DRL)

In order to anticipate the EU DRL regulations that will be going into effect in 2012, and to enhance product appeal, a thin format, low power consuming DRL that uses white LEDs was developed. Figure 8 shows the light in an illuminated state. This high-density headlight, which incorporates a DRL function into a form factor no different from that of conventional headlights, is one major instance of the advanced design that characterizes the CR-Z.

Figure 9 shows the sectional structure of the DRL reflector. LEDs are highly directional, and taking advantage of this characteristic of LEDs, the reflector was made small. While possessing light distribution

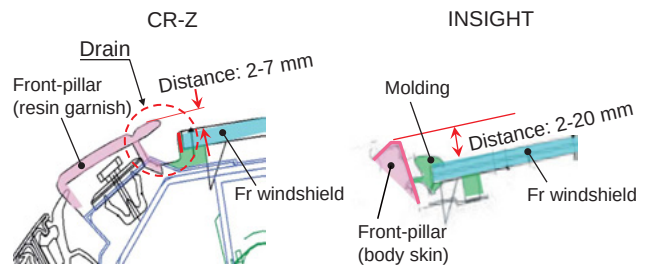


Fig. 6 Front pillar section

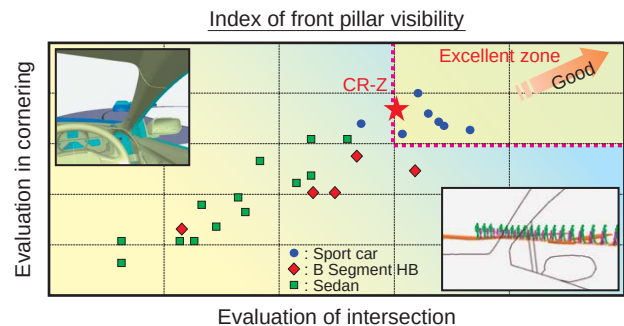


Fig. 7 Index of front pillar visibility



Fig. 8 LED DRL

performance on a par with conventional bulbs, LEDs have a greater service life, at 6000 hours, and consume approximately half the power, at 24 W.

The Japan specification uses one half of LEDs as position lamps.

3.1.5. Low profile antenna

A stylish AM/FM antenna with sufficient reception performance was developed as a decorative item on the roof in this advanced design (Fig. 10).

Conventional AM/FM micro antennas had an antenna element rod (with windings inside) approximately 180 mm in length. The rod type antenna would receive signals by resonating at one-sixteenth the wavelength l ($l/16$) in the FM band while the low profile antenna (LPA) would receive signals by resonating at approximately one-fiftieth of that ($l/50$).

When a wire wound antenna element is used as it is to resonate at a wavelength of $l/50$, the reception band will become narrower and the usefulness of the antenna will be lost. The LPA has the antenna element in a plane form to enable wideband reception. A miniature AM/FM antenna with this structure that has been made practical to use is a world first.

The antenna element of the LPA is made up of a metal element and a coil in order to broaden the reception band. In combination with a low noise high gain amplifier, with which matching is a challenge, it realizes reception performance on a par with or better than a micro antenna.

Figure 11 shows a comparison of reception performance by the LPA and the micro antenna.

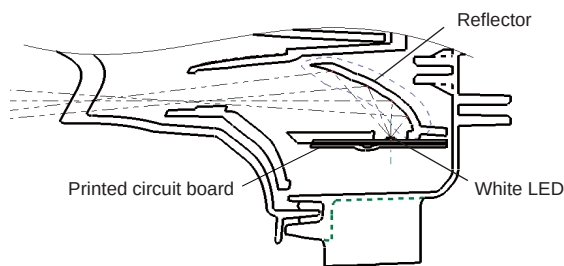


Fig. 9 Sectional structure of DRL



Fig. 10 Low profile antenna

3.2. Interior Design

3.2.1. Design concept

“Smart Cockpit” was the design concept for the interior (Fig. 12). The themes were a sporty cockpit, the kind of advanced feeling one would expect of an IMA car, and the highest level of functionality for a 2+2 package.

The instrument panel was configured with a “floating upper pad” that achieves a balance of a tight, sporty feeling and an exhilarating feel together with an “advanced and sports meter” (gauge cluster) that has a concentrated focus on the functions required when driving. The seats were made “smart sport seats” that achieve a balance of holding performance and safety functionality.

3.2.2. Sporty and high quality

A “soft feel” was adopted for the upper part of the instrument panel in order to further highlight the design and to enhance quality. This was the first “soft feel instrument panel upper” to be used by Honda. A “high shine metal film garnish” was adopted for the garnish connecting the instrument panel and the console and for the grip portion of the door lining. This was the first such garnish used in the automobile (interior) industry.

The “soft feel instrument panel upper” is fabricated by setting the already formed instrument panel in a die, injecting two types of liquid into the die, and producing a surface film of urethane by the resulting chemical reaction. Because the surface film is produced inside a die,

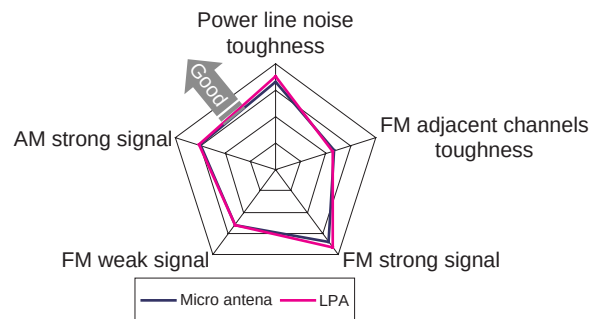


Fig. 11 Local compatibility



Fig. 12 Interior design

it has high transferability of grain and it readily allows adjustment of the gloss, thereby yielding a high-quality appearance. The surface film produced by this approach is only 1 mm thick and therefore cannot yield enough to actually feel soft to the touch, but it does convey an extremely high-quality impression.

The “high shine metal film garnish” is formed by using compressed air vacuum forming wherein the film structure shown in Fig. 13 is applied onto a base material that has been formed in advance. The first layer of the film has tin applied by deposition, after which a layer (genuine metal layer) that has had a glossy surface process applied to it is inserted. This approach yields an appearance that is on par with plating.

The film allows more freedom in design because of its high extensibility and can be applied even to deep-drawn shapes like the door grip. This film possesses a metallic gloss that cannot be reproduced through coating. On the other hand, this method also enables the film to be applied to complex shapes that cannot be formed from metal plate. Adoption of this technology therefore realized a high-quality appearance.

The “soft feel” instrument panel upper and the high shine metal film garnish enabled the creation of an interior one would not expect to find in a small car, with a level of finish typical of cars one class above it.

3.2.3. Advanced three-dimensional gauges

The gauges were created to communicate the feeling of the design concepts of “advanced” and “the future” at a single glance. There is a single round gauge with crystal accents placed in the center. The luminescence of the finely-worked dial markings with radial blue lines and the virtual image technology realized a three-dimensional view of the gauge that suggests an image from outer space. The main functions required when driving are concentrated in the central area, enabling enhanced readability (Fig. 14).

3.2.4. The smart sport seat

The front seats were developed with an emphasis on

the ability to cradle their occupants and provide solid support during sporty driving. A sporty bucket seat design was adopted and the headrest was also made a shock mitigation seat with safety functionality for the neck. In order to give the seats a unified appearance, garnish was applied to the portions connecting the headrest to the seat back.

Seat fabric that conveys the impression of being advanced was used on the main, side, and support portions (Fig. 15).

The seat’s ability to firmly cradle its occupants was realized by the following three items of technology:

- (1) The rising portions of the seat cushion and seat back were made into separate pieces and the urethane was made softer. These measures enabled the seat shape to provide support for people with smaller bodies while the shape of the seat reaching further outward from the seat could provide support for people with larger bodies. The seat is thus able to hold people regardless of their physique, so that, for example, it could eliminate the discomfort felt when steering operations cause a seated person’s body to lean in the direction opposite to the direction being steered.
- (2) Support wire was added to the side of the base seat frame to control the sideways movement of the seated person’s body during cornering.

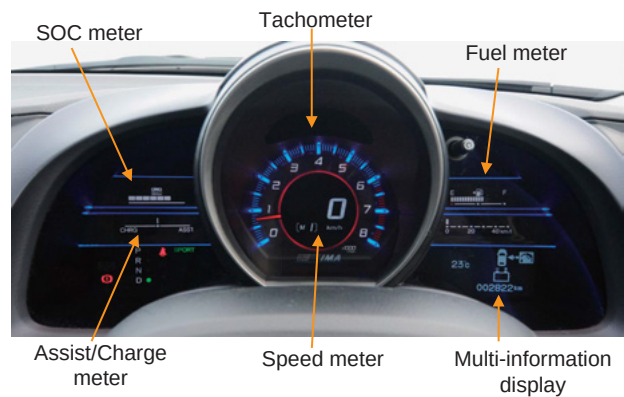


Fig. 14 Advanced 3D meter

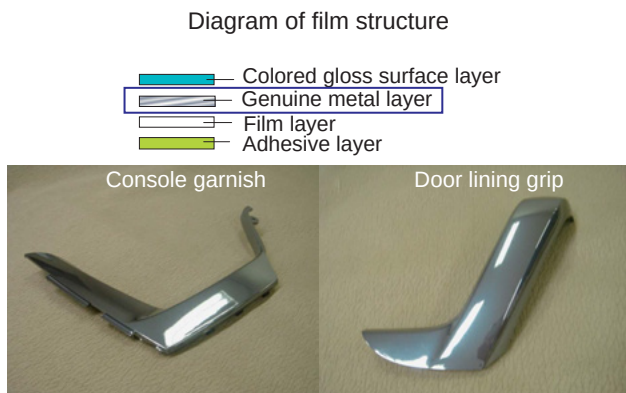


Fig. 13 Door grip and console garnish

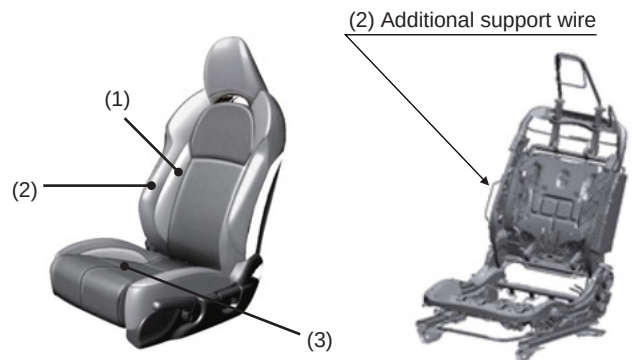


Fig. 15 Design and structure of front seat

- (3) The angle of the seat under the seated person's thighs was set so that the buttocks would be less likely to slide across the seat cushion during acceleration and deceleration. Attention was given not just to the angle but also to the coefficient of friction of the seat fabric, and fabric that would tend to keep the buttocks from shifting out of position was chosen.

The front seat was developed to achieve a high level of integration of design and functionality. The Honda team therefore feels certain that customers will be able to experience with their bodies how enjoyable it is to drive this car.

4. Balance of Totally New Acceleration and Fuel Efficiency

4.1. Power Train

4.1.1. Engine

The 1.5 L, 4-valve engine of the 2009 model year FIT was taken as the base engine, and some enhancements were made in order to mount the IMA. The engine was further modified to incorporate a variable valve timing and lift electronic control (VTEC) system with one-valve idling, which enhanced its environmental performance and brought it a California Air Resources Board (CARB) emissions rating as an Advanced Technology Partial-Credit Zero Emission Vehicle (AT-PZEV). Incorporating the IMA made up for the engine's reduced output in the high speed range due to its enhanced environmental performance, while the IMA motor also increased torque in the low speed range. As a result, the output curve became flatter with respect to engine speed, and both the power responsiveness at low to medium speeds and the ease of handling in practical use were enhanced (Fig. 16).

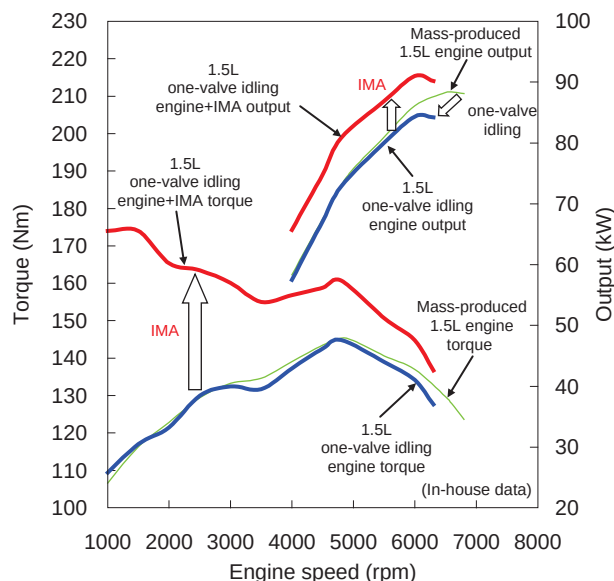


Fig. 16 Power train performance

4.1.2. Transmission

The world's first such combination of a hybrid engine and a six-speed manual transmission were developed for this car, and measures were taken to give it a sporty feel. The following changes were made to the base six-speed manual transmission in the course of transmission development:

- (1) For the purpose of incorporating the IMA, the mission case was changed, a neutral sensor switch was added, and a new clutch was developed especially for IMA use.
- (2) In order to enhance the car's sporty feel, the shift stroke was shortened (from 50 mm to 45 mm) and the operational load of shifting was reduced.
- (3) The transmission ratio was configured to pursue a balance between fuel efficiency and driving performance.

The Japanese- and American-specification cars were also configured with the same continuously variable transmission (CVT) as that of the 2009 model year INSIGHT.

4.1.3. Integrated Motor Assist (IMA)

The IMA motor, battery, and IPU (Intelligent Power Unit) were carried over from the INSIGHT specification. The cables and the intake and exhaust ducts were modified to fit the chassis shape. The combination of a 1.5-L engine, IMA, and six-speed transmission enabled the car to clear the AT-PZEV regulation values while also achieving a balance of fuel efficiency performance above the level for a car with 1.5-L displacement, and driving performance at or above the level for a car with 1.8-L displacement (Fig. 17).

4.2. 3-Mode Drive System

4.2.1. System aims

In order to allow drivers greater enjoyment of the distinctive characteristics of a hybrid sports car, the drive system was configured with three modes: SPORT, NORMAL, and ECON. In NORMAL mode, a balance of agile handling and fuel efficiency by means of smooth

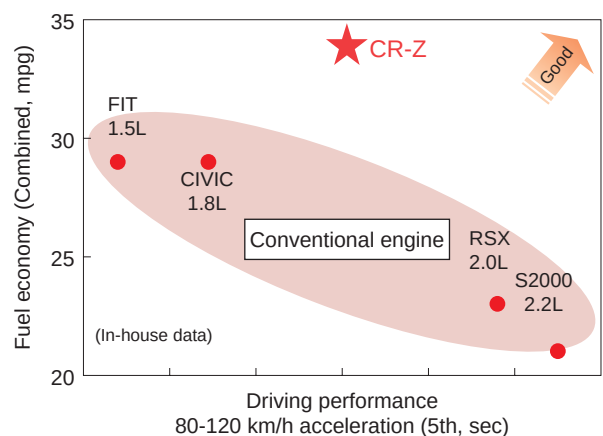


Fig. 17 Driving performance and F/E

driving is achieved at a high level. SPORT mode enables enjoyment of an exciting feeling by means of driving performance characterized by high response and power and firm handling. In ECON mode, support for ecological driving provides the vivid experience of enhanced fuel efficiency (enhanced from 5% to 10% at most). The enjoyment of working these modes to suit the driving situation was also part of the aim.

4.2.2. System configuration and settings

Figure 18 shows a conceptual image of the 3-mode system and its settings. Each mode switches to its steering effort, engine response, motor assist, and air conditioner settings. When the engine starts, the car is in NORMAL mode, and after that it can easily be switched to either of the other modes by pressing a button. The color of the ambient meter also changes with the mode. NORMAL and ECON modes are blue when the car is being driven. That changes to green when the car goes to fuel efficient driving, and it stays constantly red in SPORT mode, producing a sportier quality.

This 3-mode drive system has realized three different kinds of driving performance, something that could only be found in a hybrid vehicle.

4.3. Quick and Agile Handling

4.3.1. Chassis

The suspension was carried over from the 2009 model year INSIGHT specification. The front tread was enlarged by 20 mm. The rear tread was enlarged by 27 mm, with a change from 1° to 1.5° in the camber angle in light of tire wear performance as well as a modification to the shape of the reinforcement inside the trailing arm. Special dedicated settings were also used for the springs, damper characteristics, and high-efficiency brushless EPS characteristics, thus realizing a ride feel that achieves a high-level balance of driving stability that is sporty yet quick and highly stable with ride comfort that has flexibility (Fig. 19).

Large-diameter 195/55R16 tires, a size widely distributed on the market, were used. Tires were

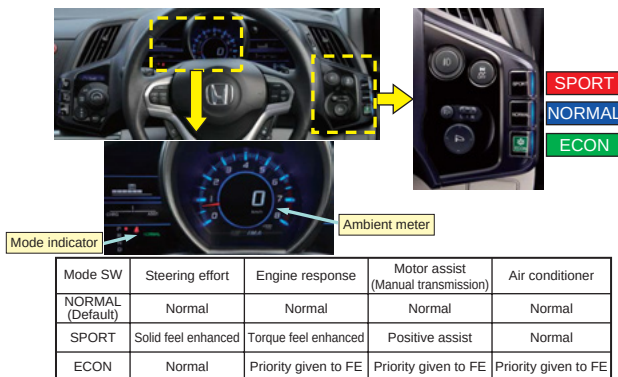


Fig. 18 3-mode drive system

developed exclusively for summer season use and all-season use. This achieved a high-level balance of driving stability that was also fuel efficient with ride comfort.

The mount system supporting the power train employed a system with three principal axes of inertia with a torque rod similar to that in the 2009 model year INSIGHT, and the mount characteristics were given special settings. These measures contributed to the enhancement of driving stability and ride comfort.

Aluminum-forged front suspension lower arms and aluminum wheels designed for lighter weight were also used. These measures contributed to a reduction in the vehicle weight and enhancement in driving stability and ride comfort.

4.3.2. Body rigidity

Driving stability performance that is quick requires high body rigidity. However, the CR-Z is a two-door hatchback coupe, which means that its design includes doors with larger openings and a tailgate opening. Consequently, gussets were added in the corner areas of the upper front bulkhead and patches were efficiently placed between the front damper and the dashboard upper

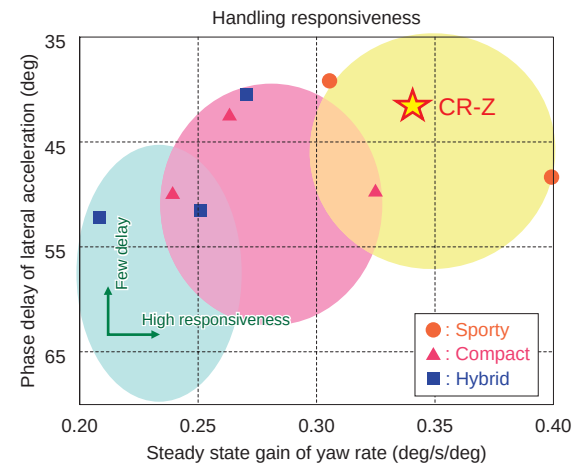


Fig. 19 Chassis dynamics

member. The structure between the center pillar and the roof rail was straightened and a closed section member was used all around the periphery of the tailgate opening. These measures were taken to increase the body rigidity (Fig. 20).

In addition, the trailing arm attachment points in the rear suspension employed H-shaped performance rods. This contributed to enhancement of driving stability performance.

4.3.3. Sporty sound

Measures were also taken with regard to the car's sound in order to underscore the development concept of the "hybrid sports car."

The results of sensory evaluation showed that mid-frequency sound during acceleration produces a sporty impression. Therefore the focus was turned to engine acceleration sounds and exhaust sounds, and this led to optimization of the characteristics of the mounts, which are parts that transfer the engine vibration. The exhaust system, which was limited as to layout because of the short overhang, was designed specially to produce a powerful impression in cars with manual transmission and a clean impression in cars with CVT.

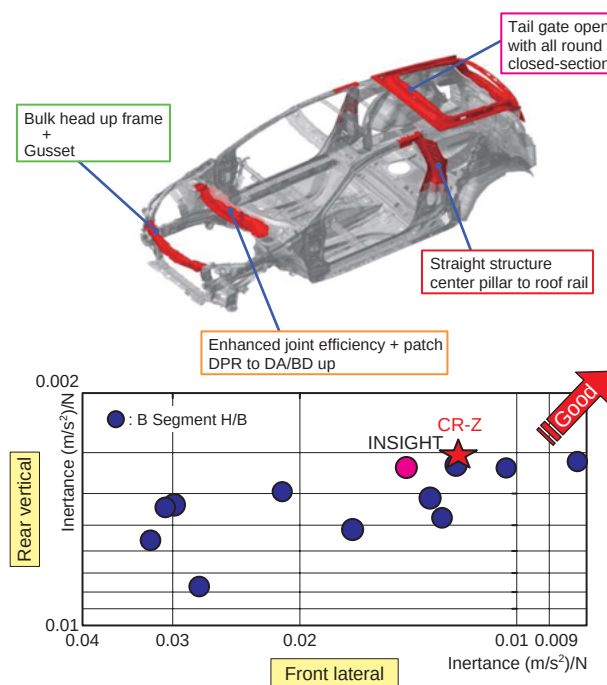


Fig. 20 Body structure and rigidity

5. Comfort in Daily Use

5.1. One-Touch Fold-Down Seats

In order to achieve a balance of sporty design that has all excess pared away and luggage space functionality for daily use, a luggage space with a simple structure was developed.

The luggage space can hold two small suitcases (655 x 480 x 255 mm) or one large suitcase (750 x 540 x 280 mm) even with the rear seats in their upright position. The luggage space accommodates an optional tonneau cover and can be used in three modes, giving it good everyday usability.

The rear seat backs form a single unit that can be folded over easily by the one-touch release of a locking knob at the center. This expands the luggage space to a flush area that is all on a single level and is spacious enough to hold two popular 9.5-inch golf bags (Fig. 21).

Cars with the instant mobility flat tire repair system (IMS) specification have a large under-floor box for storage of items that people want to have in the car but that are only infrequently used (such as the car inspection certificate, triangular emergency sign, car washing implements, and so on). Cars with the spare tire specification have a smaller storage box.

Furthermore, cars for the North America market were made in a two-seater specification without rear seats (Fig. 22). A separator and trays were installed instead of seats. The bottom of the trays are lined with carpeting material so that objects placed in the trays will not rattle. A damper is built in to reduce the noise the separator would make hitting the trays when it is folded over, and this realizes a high-quality operational feel. A central locking knob was installed to allow the same usability as in the rear seat specification.

Efforts were made to design common parts in the two-seater development by carrying out the development with efficiency. For example, the frame of the separator carried over the frame of the rear seat, and the holes for the rear seat belts were used to install lights to illuminate the trays, resulting in the common use of the side lining. These measures allowed a balanced implementation of the two different packages.



Fig. 21 Luggage space

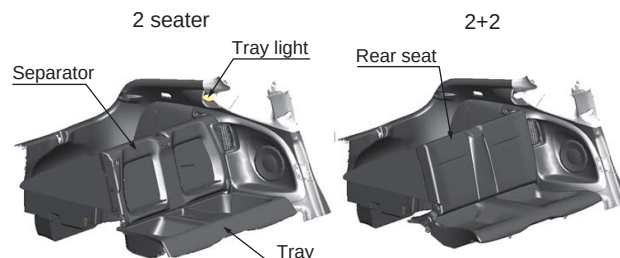


Fig. 22 Rear package

5.2. Rear Wide-Angle Camera

In order to further enhance the usability of the rear view camera, a wide-angle camera that increased view angle from conventional 130° to 180° was developed. Figure 23 shows an image from that camera.

Simply displaying a 180° image will result in extreme distortion of the kind found with fisheye lenses, making it difficult to recognize objects.

What was done, therefore, was to specify the physical size of vehicles to the rear of the car and vehicles 180° to either side that are displayed in the camera image. The view of the central area is then displayed in an image equal to that from a conventional camera, and the track of an object that is moving across the view from side to side is drawn so that it will appear as a straight line. This modification allows recognition of objects across a wide angle of view without a sense of strangeness or discomfort.

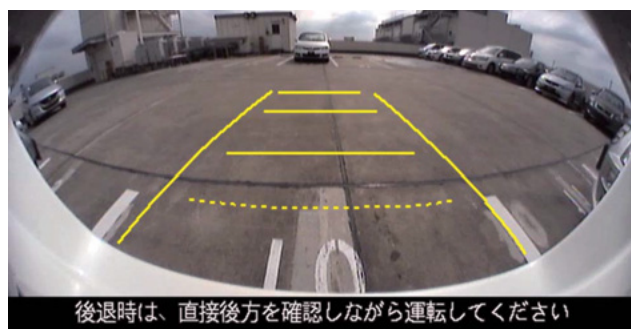


Fig. 23 Image from wide-angle camera

6. Conclusion

The CR-Z that was created in this development effort has low, sporty styling and quick, precisely maneuverable handling together with a new generation sporty sensation from a totally new acceleration based on the car's hybrid configuration, while the design also provides fuel efficiency that takes everyday usability into consideration. Therefore the CR-Z has been given new value, while offering sheer driving pleasure, as a car that can be used to suit ordinary days and extraordinary days, depending on how the driver feels at the time.

The CR-Z has been brought to a level of styling and performance sufficient to establish its own distinctive presence among them. It will have more than enough presence to give customers the joy that comes from driving a car as well as the joy that comes from owning one. The challenge was "to create new value in a small sports car," and the response of the development team is "the CR-Z." Honda is certain that this will make good sense to more and more customers.

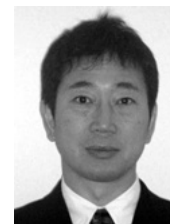
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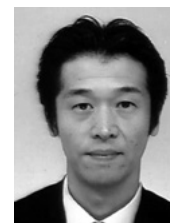
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Toru MATSUI

Innovative Technologies for Compactification of Inverter Generator, “EU3000i Handi”

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ABSTRACT

In addition to their use as an emergency backup power source during power outages and as a power source for construction applications, portable engine generators are also used in leisure activities. In the latter category, their main application is a power source for the air conditioners used in camper vans and recreational vehicles, necessitating a quiet and highly portable 3 kVA-class generator. The EU3000i Handi was developed in response to this need.

The volume of the EU3000i Handi has been reduced by 36% and its weight by 42% from the professional-use EU3000is (3 kVA), realizing class-leading lightness and compactness.

The maximum speed of the engine used in the new generator has been increased and full transistor ignition timing control has been employed, helping to reduce engine displacement to 163 cm³ from the 196 cm³ engine used in the EU3000is. A plastic monocoque structure reinforced with magnesium alloy is used for the frame, further reducing overall weight in comparison with that of the steel frame. In addition, the use of a Hershel resonator has reduced air intake noise, and the modification of the housing configuration has increased the absorption and shielding of mechanical noise. This has resulted in the achievement of the noise level of L_{WA} 91 dB, when the unit is being operated at rated output, equivalent to that of the previous recreational use generator, the 2 kVA EU2000i.

1. Introduction

Portable generators are widely used to provide an emergency backup power source in case of power outages and natural disasters, or as a power source for electrical devices and tools on construction sites, and as an outdoor power source for leisure activities. The generators are also extensively used as a backup power source for work sites and in daily life in developing countries, where power outages frequently occur, as rapid economic development generates imbalances in power supply and demand. With increasing economic activity worldwide, the demand for power and the need for portable generators are increasing as well.

The production of light, compact, quiet, vibration-free, and fuel-efficient portable generators with good performance is necessary in order to respond to these various needs.

Inverter generators display better electrical performance⁽¹⁾ than that of conventional double-pole field rotary generator systems, and their ability to vary engine speed in response to generator output helps ensure lower

noise and vibration and higher fuel efficiency⁽²⁾.

Given these characteristics of inverter generators, in 1998 Honda met the market's needs by producing the first EU series of soundproof-type inverter generators, the 3 kVA EU3000is. Since then, generators with outputs ranging from 1 to 6.5 kVA have been included in the series, and have proven successful with Honda customers. The 1-3 kVA-class are mainly positioned for recreational use, providing power at camp sites and for camper vans and RVs, while the 3-6.5 kVA class are mainly used to provide emergency backup power in power outages during natural disasters and as power sources for electrical devices and tools on construction sites.

Figure 1 shows the relative weights and output levels of EU series generators, the EM3800SX, an open frame-type double-pole field rotary generator, and other soundproof-type inverter generators.

The weight of the EU3000is inverter generator was reduced from that of the EM3800SX open frame-type due, among other factors, to the use of a multi-pole outer rotor-type alternator⁽²⁾ that also functions as a flywheel. In addition, the mass per output of the EU3000is is higher

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than that of the EU1000i or the EU2000i, models mainly employed for recreational use.

The EU3000i is designed for professional use, for example on construction sites, and as an emergency backup power source for offices and other places of business, and is therefore provided with a high-capacity fuel tank and a starter motor and battery. In addition, because the use environment of the unit on construction sites is more severe than that of the EU2000i, the use of the latter's simple plastic monocoque frame would have been impractical from the perspectives of strength and stiffness, and a frame configuration using multiple steel plates was therefore employed for the EU3000i. This is a major factor in the difference in mass per output for the two units.

However, in many cases the output of the EU2000i is insufficient to enable the unit to act as a power source for RV air conditioners, their main use in the North American market. Because of this, despite the need for low weight, compactness, and portability, the professional-use EU3000i is being used to power RV air conditioners. The development of a lightweight, compact, and easily usable 3 kVA-class generator was also desirable from the perspective of providing a generator for recreational use as well as emergency domestic power supply.

The 3 kVA-class EU3000i Handi was developed in response to these needs. This paper will discuss the technologies employed in making the unit lightweight and compact.

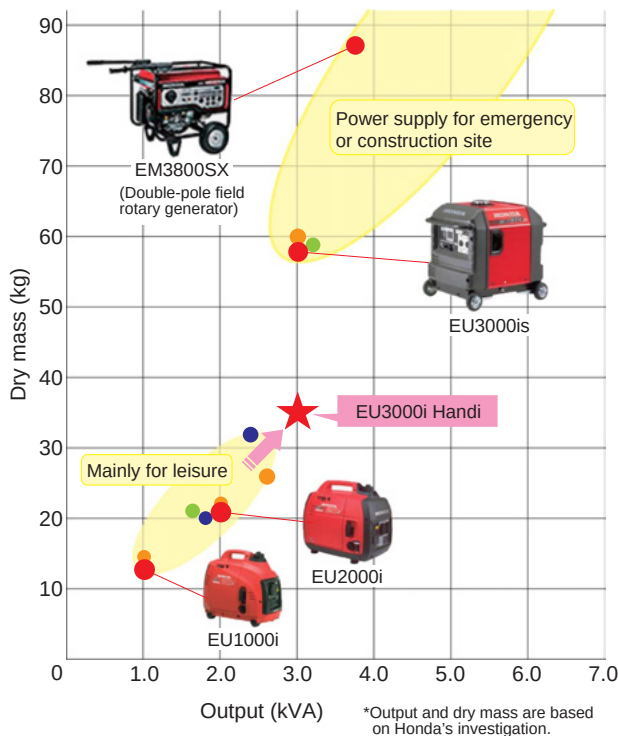


Fig. 1 Output and dry mass

2. Development Goals

The following goals were established for the development of the EU3000i Handi in order to respond to market needs for a lightweight and compact 3 kVA-class inverter generator.

(1) Easy portability and handling

In general, generators for recreational use are transported to a camping site in a camper van or truck, and then manually unloaded from the vehicle and moved to the location where they will be used. Focusing on the achievement of sufficient lightness to enable one person to manually unload the unit from a truck or camper van and then comfortably move it to the desired location, subjective tests were conducted on prototype models. Based on the results of these tests, target figures of a dry mass of 35 kg with gasoline and engine oil removed and a pulling load of 80 N with wheels and pulling handle attached were set for the unit.

Figure 2 shows the EU3000i used as a power source for a camper van.

Generators are often placed next to a camper van for them not to be stolen. To help ensure that the exhaust outlet of the generator does not face the body of the camper van, and also because the plug and the engine switch are located on the opposite side to the exhaust outlet, it is normal to position the units parallel to the camper van body, in order to make it easy to use the power cable and operate the engine switch. Given this, the space necessary for positioning the unit would be reduced in proportion to its total width, and a reduction of same by 20% from that of the EU3000i was established as a target.

(2) Power output for RV air conditioner

The target established for the unit here was, while maintaining a maximum output of 3 kVA, to be capable of instantaneous power output sufficient to start an RV air conditioner with a cooling performance of 4 kW.

(3) Achievement of quiet performance

The target established in this area was to achieve a sound power level, L_{WA} , of 91 dB when operated at rated output, equivalent to the level of the EU2000i when operated at its rated output of 1.6 kVA.



Fig. 2 Example of use as camper power source

3. Technologies for Reduction of Size and Weight

3.1. Use of Low-displacement Engine

The 163 cm³ GX160 engine, with a displacement 17% lower than that of the 196 cm³ GX200 engine used in the EU3000is, was employed in order to reduce the weight of the new generator. The use of this engine reduced the weight of the unit by approximately 500 g.

In order to help ensure the achievement of a maximum output of 3 kVA, equivalent to that of the EU3000is, despite the reduction of engine displacement, the four changes discussed below were made to the standard specifications of the GX160. As the result of these changes, the output of the engine at wide-open throttle was increased by 11% at the target speed for maximum output, 4000 min⁻¹.

(1) Increased engine speed at maximum output

Because the EU3000i Handi is an inverter generator, the frequency of the output voltage is not dependent on engine speed⁽¹⁾. Given this, the target engine speed at maximum generator output was increased to 4000 min⁻¹ from the figure of 3800 min⁻¹ for the EU3000is.

(2) Increase of valve operating angles

Because output at high engine speeds was prioritized, the valve timing was modified, with the valve-operating angle increased by 20° on the intake side and 10° on the exhaust side against the standard specifications of the GX160.

(3) Expansion of carburetor venturi diameter

From considerations of startability, idling stability, and ability to track loads, the charging efficiency of the engine at high speeds was increased by increasing the area ratio of the carburetor venturi diameter by 21% against the base GX160 specifications.

(4) Ignition timing control

Because the transistor-type magneto ignition used in the EU3000is applies constant ignition timing irrespective of engine speed or load, the ignition timing across the entire operating range is mainly determined by the knocking limit and the strength of components at the engine speed for peak torque. Full-transistor ignition was employed in the EU3000i Handi to enable control of ignition timing at each engine speed, helping to realize the optimum ignition timing for startability and maximum output at each engine speed.

Figure 3 shows the engine performance curve.

3.2. Engine Layout and Cooling System Configuration for Reduced Size

In order to reduce the space necessary for positioning the generator when used as a power source for a camper van, the reduction of the unit's total width was prioritized and attention was focused on the packaging.

Figure 4 compares the orientation of engine placement, total width, and the channels for the flow of

cooling air in the EU3000is and the EU3000i Handi.

In the EU3000is, the direction of the width of the generator unit is at a right angle to the direction of the crankshaft to help realize short, straight-line cooling channels, thereby reducing cooling air pressure loss in the housing. By contrast, in the EU3000i Handi, the direction of the crankshaft, in which the engine width is narrowest, is the direction of the width of the generator. This helped to achieve a total width of 379 mm for the generator body, a reduction of 21% from that of the EU3000is.

With the change in the positioning of the engine, cooling air no longer flowed in a straight line, and the high-density arrangement of parts reduced the amount of available space in the housing. In the initial specifications of the new generator, this increased cooling air pressure loss and impeded cooling air flow in comparison to the EU3000is. However, application of computational fluid

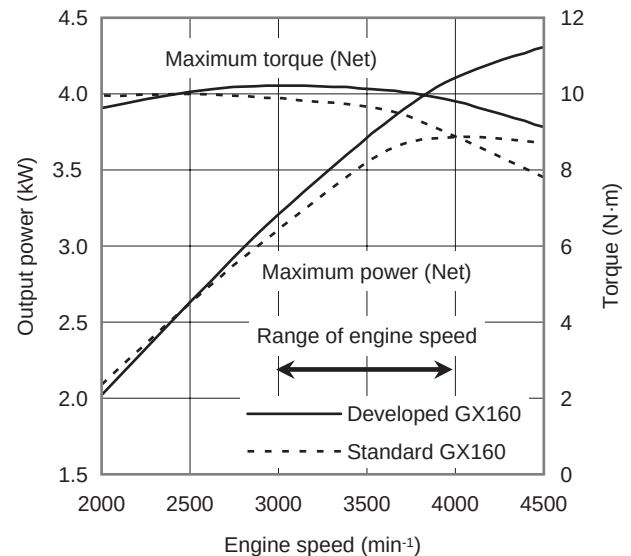


Fig. 3 Engine performance

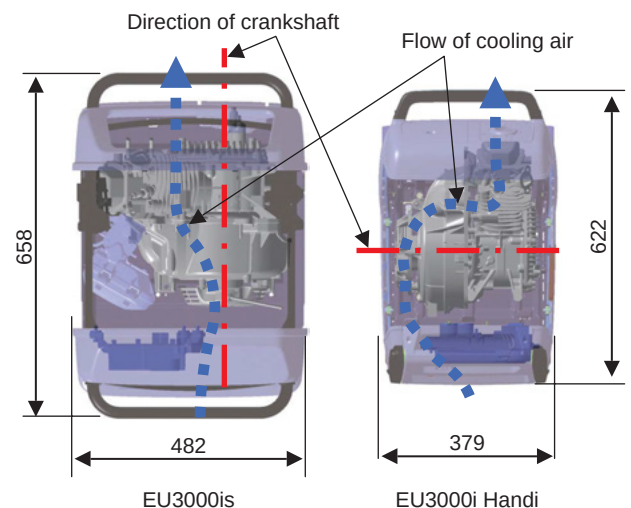


Fig. 4 Location of engine in relation to full width of generator

dynamics (CFD) analyses helped to realize good cooling performance through optimization of the form of the cooling channels.

Figure 5 shows an example of the optimization of the cooling air flow around the engine cylinder head using CFD.

As can be seen, the insufficient flow of cooling air around the underside of the cylinder head in the initial specifications has been corrected.

Figure 6 shows an image of the flow of cooling air around the interior of the body of the generator.

The center frame separates the parts that produce a low level of heat, such as the inverter unit, from those that produce a high level of heat, including the alternator, engine and muffler. Cooling air flows from the low-heat area to the high-heat area for efficient cooling.

3.3. Simple Frame Configuration

One method employed to reduce weight in relation to the frame structure was the use of multiple plastic sheets in place of the steel sheets used in the EU3000is.

Figure 7 shows the structure of the frame and cover of the EU3000is, while Figs. 8 and 9 respectively show

the structure of the frame and the cover of the EU3000i Handi.

Because the EU3000is is designed for professional use on building sites and similar situations, the frame and the covers, with the exception of the front and rear covers, are constructed of steel sheets to ensure durability.

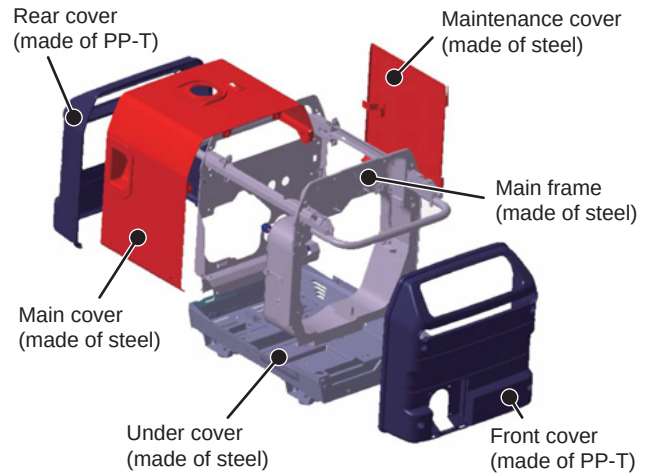


Fig. 7 Structure of frame and cover (EU3000is)

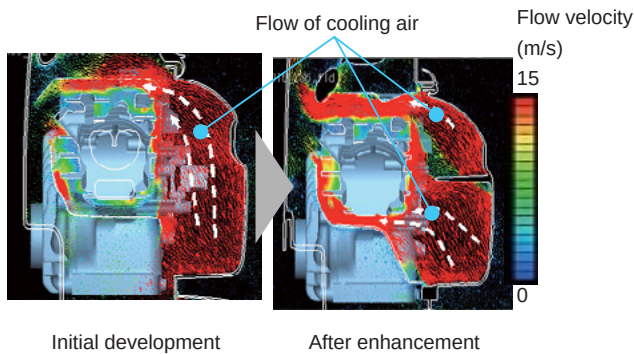


Fig. 5 Cooling air flow dynamics around cylinder head

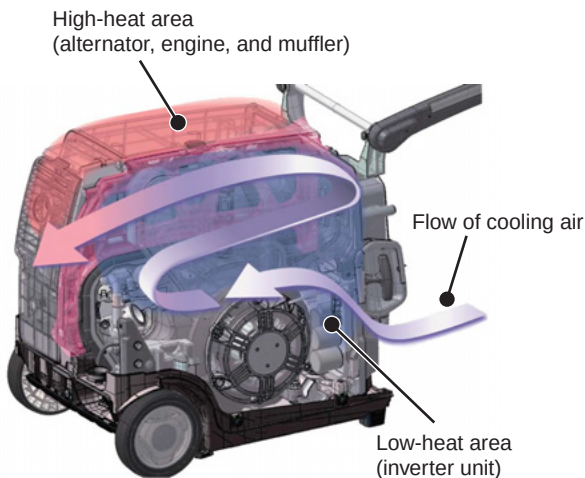


Fig. 6 Flow of cooling air

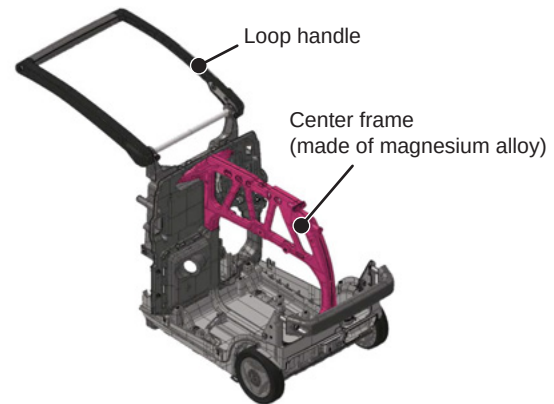


Fig. 8 Structure of frame (EU3000i Handi)

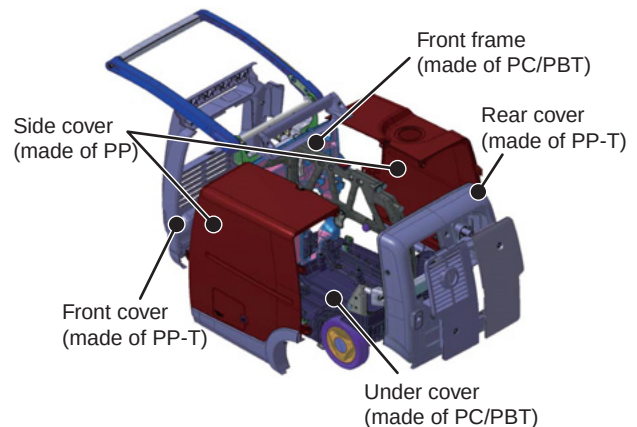


Fig. 9 Structure of cover (EU3000i Handi)

By contrast, since the EU3000i Handi was designed for recreational use, it was necessary to create a light unit. For this reason, a plastic monocoque structure reinforced by a center frame was selected for the new generator.

The center frame, which is the main frame that connects the pulling handle and the wheels, is positioned close to the muffler. Because of this, thermal radiation from the muffler immediately after operation of the unit stops and the cooling air ceases to flow would cause its temperature to increase. In order to maintain the stiffness of the frame when its temperature increased, it was necessary to employ a metal in which the modulus of elasticity had a low level of temperature dependence. A magnesium alloy was selected for the EU3000i Handi.

The use of a high-strength, high-stiffness center frame that was not significantly affected by heat made it possible to use a plastic monocoque structure fashioned exclusively from a general-purpose plastic for the other parts of the frame and the covers. This had the merit of not only realizing weight savings, but also of reducing costs.

As a result of these efforts to achieve weight savings, the total weight of the frame and the covers of the EU3000i Handi has been reduced to 6 kg, a reduction of approximately 60% from the 14 kg of the EU3000is.

3.4. Inverter Unit

Because the maximum speed of the engine used in the EU3000i Handi had been increased against that of the engine in the EU3000is, the voltage range of the alternator sub-winding had been increased, necessitating a power supply circuit configuration able to respond to a wide range of input voltages.

The use of a power supply circuit configuration identical to that employed in the EU3000is would result in increased power circuit loss in the high engine speed range, and would increase the size of the unit. A low-loss ringing choke converter system able to operate stably at a wide range of input voltages from low to high engine speeds was therefore employed in the EU3000i Handi.

The use of software to conduct waveform feedback control helped to enable the number of parts to be reduced, in turn reducing the printed circuit board area. Figure 10 shows the inverter units of the EU3000is and the EU3000i Handi. Despite the fact that the noise filter, which was a separate part in the EU3000is, has been built into the inverter unit, the external dimensions of the unit are identical to those of the EU3000is inverter unit, while its weight has been reduced by 13%, or 500g.

In addition, a simple hour meter function has been newly built-in to the inverter unit. In contrast to professional-use situations, generators for recreational use are often used intermittently, making it challenging for users to calculate hours of use and time oil changes and other maintenance procedures. To make users aware of recommended times for unit maintenance, the inverter unit of the EU3000i Handi records the operating time of the

generator, and, when specific cumulative operating times have been reached, indicates this fact to the user by flashing the output indicator light at engine start. The light will not flash before 100 hours has been reached. After 100 hours, the light flashes once and increases the number of flashing times by one for every 100 hours, until it flashing five times at 500 hours.

3.5. Alternator

Figure 11 shows the alternators used in the EU3000is and the EU3000i Handi.

In the EU3000i Handi, the use of a high-magnetic-force magnet and the realization of an increased permeance coefficient (determined by the form of the magnetic circuit and the magnet) in the magnetic circuit has helped to reduce the thickness of the magnet, increasing the winding space. In addition, the use of the magnetic flux density distribution simulation shown in Fig. 12 to analyze the magnetic flux flowing in the stator core during its design helped to reduce the thickness of the stator by 5 mm and its weight by 14% (-730 g) against the EU3000is while maintaining the same generating capacity.

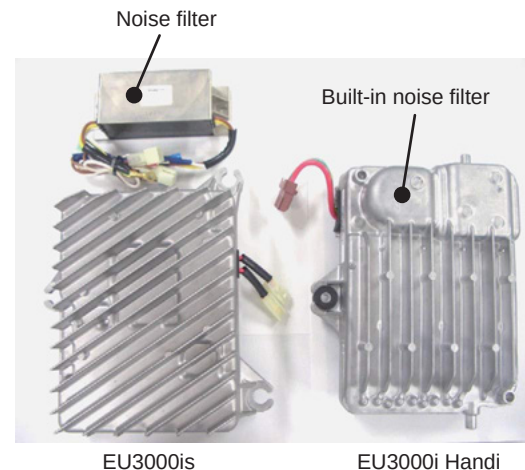


Fig. 10 Inverter unit

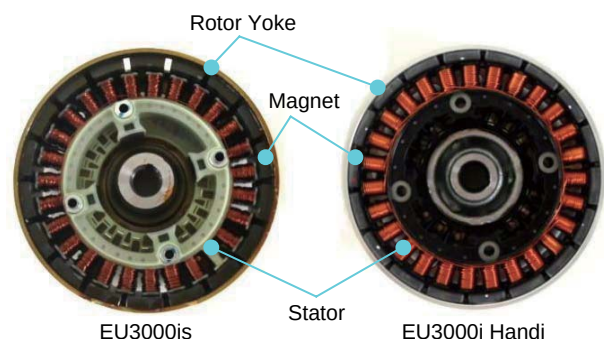


Fig. 11 Alternator

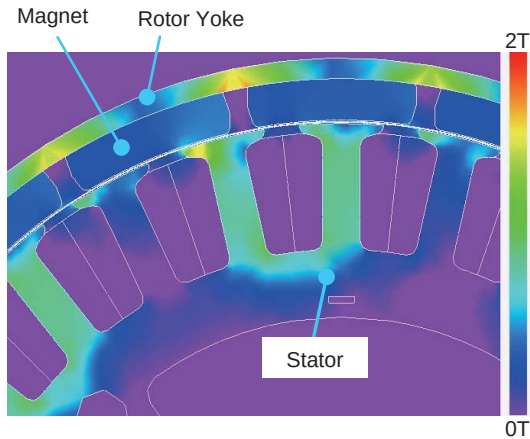


Fig. 12 Distribution of magnetic flux density of alternator

4. Power Generation Performance

Inverter generators are provided with an overload protective function that protects the loading devices and the generator itself when an overload or short-circuited output occurs. The over-current protective function counts the time for which the detected value of the output current exceeds the over-current threshold level, and shuts off power generation if the over-current continues for a predetermined period of time. In addition to this overload protective function, when short-circuited output occurs, an over-current limit circuit stops switching operations in the semiconductor power devices in order to control the peak current of the output short-circuit current and protect the generator. While this is happening, a microcomputer detects the over-current, and shuts off power generation after a predetermined period of time.

Air conditioners with a cooling performance of 4 kW are the most common type among RV air conditioners, and a generator with a 3 kVA output is generally sufficient to operate them. However, if switching occurs rapidly a number of times in an RV air conditioner, the air conditioner compressor locks and a lock current flows in the system, making an output of more than 3 kVA instantaneously necessary. Because the output current at this time exceeds the over-current threshold level, unless the compressor returns to normal operation within the predetermined period of time, the overload protective function will go into operation, and the generator will be unable to power the RV air conditioner. For this reason, despite the fact that at normal times a 3 kVA generator is sufficient to operate an RV air conditioner with a cooling performance of 4 kW, a generator with an output of 3.5 kVA or higher is actually necessary.

High-capacity semiconductor power devices able to resist lock current have been used in the EU3000i Handi, and the period for operation of the overload protective function has been extended to ten seconds from the conventional five seconds. These modifications have

helped to enable the 3 kVA unit to operate an RV air conditioner with a cooling performance of 4 kW until the compressor returns to normal operation.

However, if the same period for operation of the overload protective function was used in the case of short-circuited output, the period for which output short-circuit current flowed would double, and the semiconductor power devices would be destroyed by the current. A new function for the detection of short-circuited current has therefore been added. If short-circuited current is detected for five seconds, the generator output is shut off in order to protect the unit.

5. Noise Reduction Technologies

(1) Reduction of air intake noise

The EU3000i Handi is the first portable engine generator to feature an intake air resonator in order to reduce air intake noise. Figure 13 shows the intake port, the intake air resonator, and the carburetor.

The space under the carburetor has been used to fit a Hershel resonator⁽³⁾.

Figure 14 shows the noise characteristics around the air inlet with and without the resonator fitted.

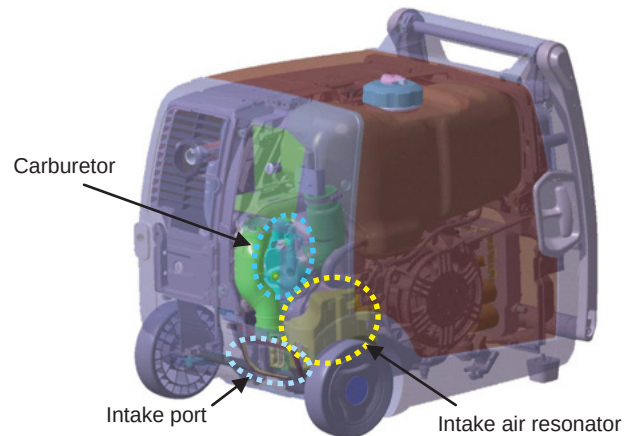


Fig. 13 Location of intake port, resonator and carburetor

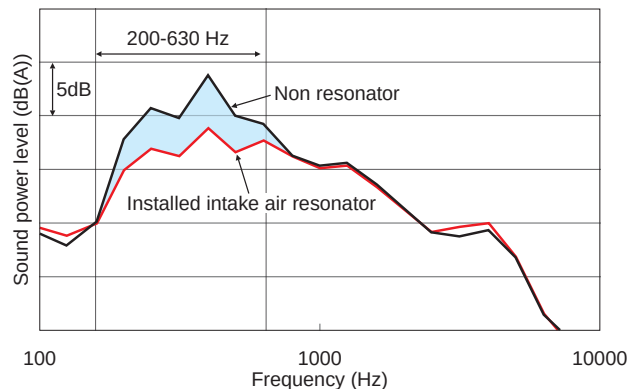


Fig. 14 Noise characteristics around inlet

The volume and shape of the resonator were designed using CAE in order to control noise in the 200-630 Hz intake noise band. As a result, a noise reduction of 5 dB(A) or more was achieved at positions 500 mm from the intake port at an angle of 45°.

(2) Reduction of radiated noise

Measures were also taken to reduce mechanical noise. A sound-absorbent material has been welded to the inside of the plastic covers to absorb noise, and large components (the inverter unit and muffler) have been positioned between the openings of the housing (the inlet and outlet for cooling air) and the engine, a source of mechanical noise, in order to increase the shielding effect. These measures have reduced the level of radiated noise to the exterior of the housing.

Urethane foam and felt were used as sound-absorbent materials in the EU3000i. The new generator, however, employs a polypropylene non-woven fabric⁽⁴⁾ that has proven to display a good acoustic absorption coefficient in the automotive field. The new material achieves an identical acoustic absorption coefficient at an approximately 40% reduced thickness, and has contributed to the reduction of the total width of the unit. The use of the new material has also reduced weight by approximately 200 g.

The use of these noise reduction technologies has helped to achieve the target noise level of L_{WA} 91 dB at the rated output of 2.6 kVA. The noise level is 3.5 dB lower at an output of 1.6 kVA, the rated output of the EU2000i. The EU3000i Handi has thus achieved a level of quiet operation one rank higher than that of the other unit.

Figure 15 shows the noise characteristics of the EU3000i Handi in the practical use range and the noise level of the EU2000i when operating at rated output.

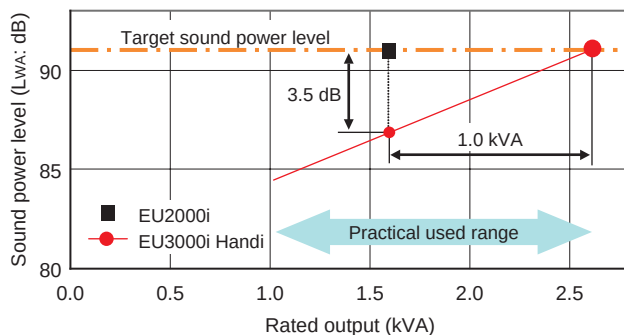


Fig. 15 Noise characteristics of EU3000i Handi

6. Conclusion

Each component used in the EU3000i Handi was reduced in size and weight, and a simple frame structure was employed. In addition, the specifications of the fittings and the range of continuous operation were

Table 1 EU3000i Handi specifications

		EU3000i Handi
Engine	Displacement (cm ³)	163
	Speed (min ⁻¹)	3000-4000
Generator	Max. AC output (kVA)	3.0
	Rated AC output (kVA)	2.6
Continuous operating hours (h)		3.5
Operating noise level (L _{WA} : dB)		91
Dimensions	L×W×H (mm)	622×379×489
	Dry mass (kg)	35.2
	Wet mass (kg)	40.1

modified to suit the unit to recreational use. The cell starter used in the professional EU3000i was removed, and the unit is operated using only the recoil starter. The fuel tank was reduced in size, while at the same time the 3.5 hrs of continuous operation at rated output necessary for recreational use was also realized. These measures have helped to enable the realization of a compact, lightweight, and easy-to-use 3 kVA-class generator.

Figure 16 shows an exterior view of the EU3000i Handi, and Fig. 17 shows the unit being moved using the pulling handle. Table 1 shows the specifications of the new generator.



Fig. 16 Appearance of EU3000i Handi



Fig. 17 EU3000i Handi with handle

The details of the technologies used to reduce the size and the weight of the EU3000i Handi can be summarized as follows:

- (1) The reduction of the size and weight of each component and the use of a plastic monocoque structure reinforced with a magnesium alloy center frame helped to achieve a dry mass of 35.2 kg, a comfortable weight for users. This represents a reduction of 42% in weight from that of the previous EU3000is, a generator with the same output. The unit is provided with two wheels and a handle that can be folded away. The generator is surprisingly portable, and with a pulling load of 80 N, can be comfortably moved by one person. The total width of the new generator is 103 mm lower than that of the EU3000is at 379 mm, and its volume has also been reduced by 36%.
- (2) The weight of the generator has been reduced through the use of an engine with a 17% lower displacement than the engine used in the EU3000is, while maintaining an output of 3 kVA and realizing the instantaneous output necessary for use with an RV air conditioner.
- (3) A noise level of L_{WA} 91 dB at rated output has been achieved, equivalent to that of the EU2000i, a generator with a rated output approximately 40% lower.

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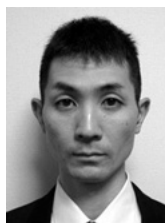
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Development of New Honda Techmatic System

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ABSTRACT

A new manually operated “Honda Techmatic System” auxiliary driving device that supports the independence of people with lower limb disabilities was developed and installed in a Honda FIT.

This development aimed to secure safety and reliability as official Honda product features, greatly advance operability from simply providing “the ability to drive” to “enjoying driving (Fun for Everyone),” secure compatibility with various car models, and enable easy installation.

This paper describes the specific structure used to achieve these goals, and the control lever design that blends with the cabin interior. It also introduces the history of driver’s licenses for disabled persons, and this market.

1. Introduction

The normal operation system layout for automobiles enables smooth driving by performing accelerator and brake operations with the feet and steering and various switch operations with the hands. This makes driving an automobile a challenge for persons with disabilities, congenital or otherwise, who mainly use self-propelled wheelchairs in their daily lives. The term “barrier-free” has been used for some time, but there are still many obstacles hindering the efficient movement of wheelchair users by means of public transportation. This makes cars an even more indispensable tool for wheelchair users than for fully able persons, and manually operated auxiliary driving devices enable accelerator and brake operations to be performed by hand to support the social independence of wheelchair users.

Most manually operated auxiliary driving devices are retrofitted after the license plates are acquired, and do not require any particular application for conversion. However, as these are important devices that control running and stopping, reliability equivalent to that of the accelerator and brake pedals of the base vehicle is required. The goals when developing the new Techmatic System were to secure safety and reliability as official Honda product features, to greatly advance operability

from just “the ability to drive” to “enjoying driving (Fun for Everyone),” to make the system compatible with various car models, and to make installation easy.

2. History and Types of Manually Operated Auxiliary Driving Devices

The first driver’s license in Japan was issued in 1907. Some 53 years later, the Road Traffic Act revision of December 1960 changed the conditions for obtaining a driver’s license, enabling disabled persons to also operate automobiles with the assistance of devices (Fig. 1)⁽¹⁾. As of 2008, there are approximately 77.96 million people with regular automobile driver’s licenses in Japan, and of these approximately 210000, or roughly 0.27%, hold conditional driver’s licenses limited to “vehicles for the physically disabled.”⁽²⁾

The market scale is approximately 5000 vehicles per year, of which two specialty manufacturers account for approximately 90%, while the three automobile manufacturers, Honda, Toyota, and Nissan, have also entered the market.

Honda’s efforts in this area began in 1968 with installation of manually operated auxiliary driving devices by the former Techland Co., Ltd. under the trademark “Techmatic System.” This was then carried over to the

* Automobile R&D Center

** Honda Motor Co., Ltd.

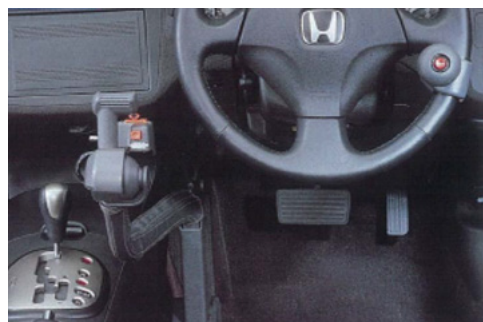
former Honda Tokuso Co., Ltd., and currently continues with Honda Logistics Inc.

Manually operated auxiliary driving devices have approximately a half-century of history since their commercialization. Recently, drive-by-wire types that use special actuators have been developed in Europe and the U.S., but most are still mechanical types that do not rely on electronic devices, and the basic structures have not changed much.

Mechanical-type manually operated auxiliary driving devices should operate the brake and accelerator pedals, which are normally operated by the feet, without multiplying the operating force. Therefore, the conventional structure uses relatively long control levers to operate the pedals using link mechanisms added to the



Fig. 1 Manually operated auxiliary driving device



(a) Floor type



(b) Column type

Fig. 2 Types of auxiliary driving device operated by hand

pedals, via rods or cables.

As shown in Fig. 2, there are two types of control lever mounting systems: the floor system that mounts the rotating axis mechanism on the vehicle center side of the driver's seat floor; and the column system that mounts the rotating axis mechanism below the steering column.

In addition, there are also two types of accelerator and brake operation systems. The first type is the single axis system that uses an integrated control lever, and operates the accelerator by pulling the lever toward the rear, and the brake by pushing the lever toward the front of the vehicle. The second type is the dual axis system that has an additional rotation axis on the upper part of the control lever. This system operates the accelerator by rotating only the upper part of the lever toward the rear around the upper axis, and the brake by rotating the entire lever toward the front around the rotation axis at the base of the lever.

3. Structure of the New Techmatic Model

In creating the new model Techmatic System, the basic structure of the control lever was addressed first out of consideration for operability that enables "enjoying driving." This led to adoption of a single-axis short-lever system that combines a shorter control lever, which reduces the distance the elbow moves, with a single-axis system, which is a clearly defined operation system.

In a single-axis short-lever system, the lever is located at a level higher than the driver's seat to reduce interference with the seat cushion, even when the seat is slid forward or backward. This also better secures the necessary control lever stroke for use with instrument panels of different shapes, and is advantageous in terms of mountability on the vehicle.

Safety reliability was further satisfied by installing a double brake cable, a drum with built-in equalizer mechanism, and a forced drum rotation mechanism. An operation gain variable mechanism and a built-in switch were adopted for greater driving enjoyment. Additional measures taken for mountability were adoption of a floor-mounted mechanism and of a cable mechanism as the means for transmitting the operating force. Figure 3 shows the overall structure. It is composed of the control lever assembly (C. L. A.) in the input section and the pedal link assembly (P. L. A.) that moves the vehicle brake pedal and accelerator pedal. This explanation will center on these two parts.

3.1. Control Lever Assembly

3.1.1. Design and functions

The control grip is an important interface that the driver touches directly with their hands to perform accelerator and brake operations, so not only is good operability demanded, but this lever is located in a highly visible position for the user, which also necessitates a

good design. The conventional control grip was considered to have good operability, but it could be said that the design was somewhat lacking in many cases. This development also gave sufficient consideration to design aspects, and realized a control grip that does not clash with the interior of the vehicle.

Grip shapes include the “I-type,” that is grasped from the side and the “T type” that is grasped from on top. The newly developed device was based on the T-type to enable a wider range of users. Design verification and practical verification were repeated using a mock-up model shown in Fig. 4. The optimal switch layout, grip sizes, and section shapes were verified, and the design was fine-tuned down to the smallest detail, including the angle R of parts touched by the user’s hand. An aluminum die-cast grip frame was used to secure sufficient strength and rigidity, and the surface was covered with real leather to give a high-class appearance for enhanced product appeal, and also to effectively reduce slipping due to perspiration.

Aluminum die-cast was also used for the housing that

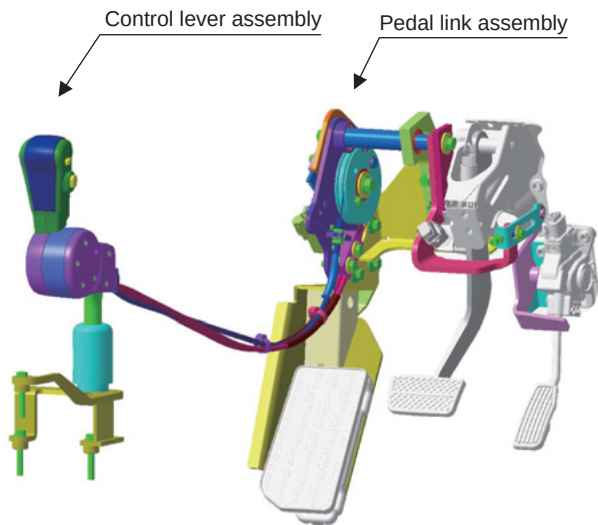


Fig. 3 New Honda Techmatic System



Fig. 4 Mock-up model

supports and houses the drive drum to which the control lever is attached. This secured strength and rigidity while also providing a high degree of shape freedom. The structure clasps these reinforcing parts between plastic cover sheets on both sides, and an elastic material (thermoplastic resin) was used in locations that might contact the user’s thighs and knees during driving.

The C. L. A. and the P. L. A. at the driver’s feet are connected by three cables and a wire harness inside polyester mesh covers, which provides a streamlined appearance for good product appeal. Figure 5 shows the device installed on an actual vehicle. These parts are all coordinated in basic black to blend with the cabin design and enable easy, lateral applications to various models.

3.1.2. Operating force transmission mechanism using three pull cables

The next matter to consider after the operation system had been decided was how to transmit the lever operating force to the pedal. Manually operated auxiliary driving devices are retrofitted to completed vehicles for which the package has already been decided, so mountability should be given sufficient consideration. The rod system, hydraulic system, and cable system were examined as media during the development stage, and eventually the cable system was adopted because it could achieve the purpose without mutual loss in operability, mountability, or safety reliability.

Thicker cables generally have an advantage in strength, but the allowable minimum bending radius also increases, thus imposing limitations on mounting layout. Therefore, to achieve a more compact structure, the high load during brake operation is constantly received by two $\phi 2.0$ mm-diameter twisted-pair cables made of stainless steel wire material via an equalizer. This approach secured sufficient strength, including durability, equal to or better than conventional approaches. In addition, the accelerator uses a single cable with the same specifications as the brake cables, but wired in the opposite direction of rotation. As



Fig. 5 Auxiliary driving device operated by hand (New type)

shown in Fig. 6, this cable layout is the same as the throttle forced return mechanism of a normal motorcycle. It allows direct input in the return direction using the lever, and even in the unlikely event of some malfunction in the accelerator linkage system, moving the lever in the brake direction would be certain to restore the accelerator.

3.1.3. Drum with built-in equalizer mechanism

The system is structured to transmit the brake operation load equally in the two cables by means of the equalizer. Even in the unlikely event that one brake cable breaks, therefore, the remaining cable would be certain to enable stopping.

The following matters were considered in designing the equalizer.

- (1) When the equalizer operates (when one cable has been severed), the remaining cable and the inner cable end should not be subject to excessive force.
- (2) The equalizer operation amount should be secured so that the driver can recognize the cable failure.
- (3) Compact

As shown in Fig. 7, a structure was devised to house the equalizer mechanism inside the drive drum in order to address these matters.

Figure 8 shows the equalizer in operation. The absence of a fixed rotating axis is a feature of this equalizer mechanism. In the event that one cable breaks, the cable-end connecting portion of the equalizer will move in a straight-line path along the side wall of the recessed space in the drum. This will cause the cable end to move 7 mm without adding any excessive force to the remaining cable. As a result, the invalid stroke amount on the control grip is 2.5 times or more the normal amount

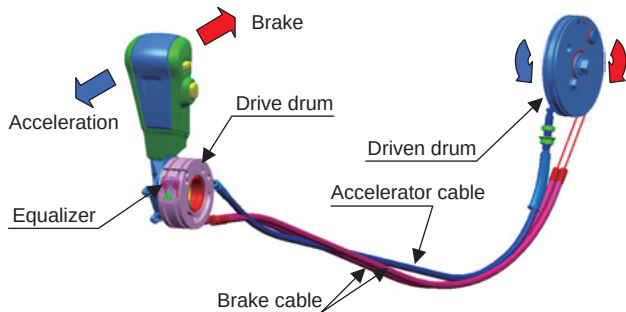


Fig. 6 Pull cables used for driving

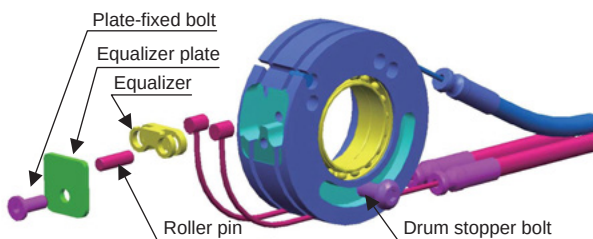


Fig. 7 Drum with built-in equalizer

of play, allowing the driver to recognize the malfunction.

In addition, an indicator was also installed on the top of the control lever housing, as shown in Fig. 9, to enable objective recognition of cable failure. The normal looseness on the indicator does not go past the ● mark, and in the event of a broken cable, it will exceed the range of looseness for two cables.

3.1.4. Switches built into grip

In driving operation with manually operated auxiliary driving devices, the right hand operates the steering wheel and the left hand operates both the accelerator-brake lever and the AT selector lever. This can sometimes make it a challenge to perform the various switch operations required for smooth and safe driving. To address this situation, the control grip of a manually operated auxiliary driving device should be equipped with auxiliary switches that have the equivalent functions to those of the base vehicle switches. The required auxiliary switches differ slightly among users, but the developed device was equipped with the following five switches to support as many users as possible (Fig. 10).

- (1) Brake lock
- (2) Turn signal
- (3) Horn
- (4) Headlight Hi/Lo-beam
- (5) Hazard

These switches basically made use of mass-produced car and motorcycle parts from the viewpoints of reliability and cost reduction. The brake lock switch (1) is unique

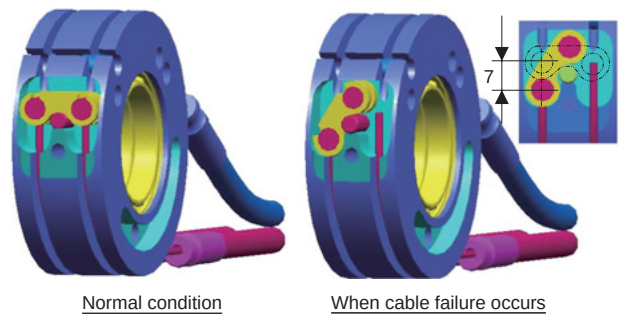


Fig. 8 Equalizer operation

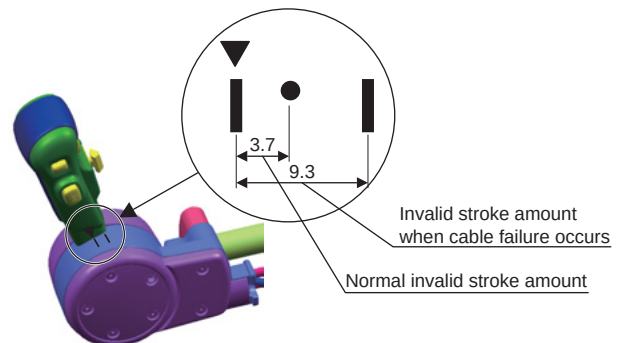


Fig. 9 Cable failure indicator

to manually operated auxiliary driving devices, and is not present on the base vehicle. This switch is indispensable, and is the most frequently used switch. When shifting the AT select lever from the P position to a different position, the select lever should be operated in the state with the brake pedal pressed. However, when using a manually operated auxiliary driving device, brake operations are performed by the left hand, which makes this a challenge. When the control lever is pressed to the brake side (towards the vehicle front) while pressing the brake lock switch, the brake pedal is held in the pressed state even when the user releases the control lever. This enables select lever operation.

As shown in Fig. 11, the five switches installed in the control grip and the wire harness added inside the instrument panel are connected by eleven low-voltage electrical wires (AVSSH0.3F⁽³⁾: super-thin wires for automotive use with nominal diameter of 0.3) inside corrugated tube with an inner diameter of 7 mm. To absorb harness movement due to accelerator and brake operations in the minimum space possible, the structure passes the wires through a space provided in the center of the control lever rotating part, and the harness bends (oscillates) within that space.

In addition, endurance testing was conducted to verify that the electrical wire had sufficient breaking resistance

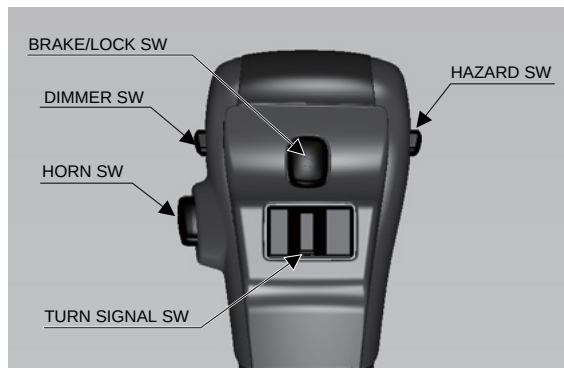


Fig. 10 Grip with built-in auxiliary switches

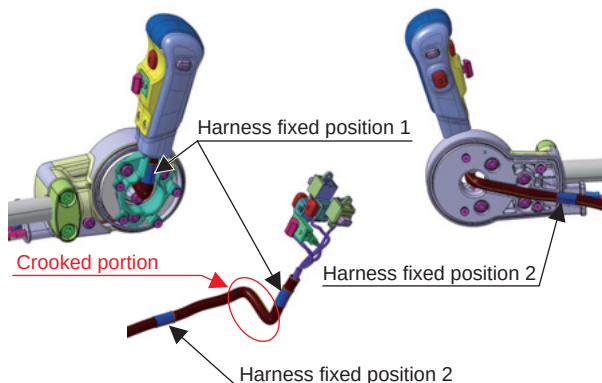


Fig. 11 Harness layout inside grip

with respect to bending (oscillation). The endurance testing was conducted in a mode that took frequency of use and other such factors into consideration by combining multiple operating angles, as well as a mode that considered low-temperature operating environments in which wire does not bend readily.

3.2. Pedal Link Assembly

The mechanism that embodies the single-axis short-lever system is the P.L.A. As shown in Fig. 12, the driven drum with an offset rotating axis and the cam plate are placed on either side of the main plate. Through the roller, the system converts the pulling force on the cable to rotary force on the main shaft.

The plate that is used at the point of contact between the cam plate and the roller is about one-fifth the thickness of plate that resists plastic deformation. This is based on “surface pressure design that takes into consideration work hardening due to plastic deformation,”⁽⁴⁾ whereby minute deformations (collapses) in the cam rolling contact surface are allowed. The width of the P. L. A. was thus reduced by the greatest extent feasible.

The brake lock mechanism described above is also incorporated into the P. L. A. The structure is shown in Fig. 13. Operation of the solenoid triggers the ratchet to

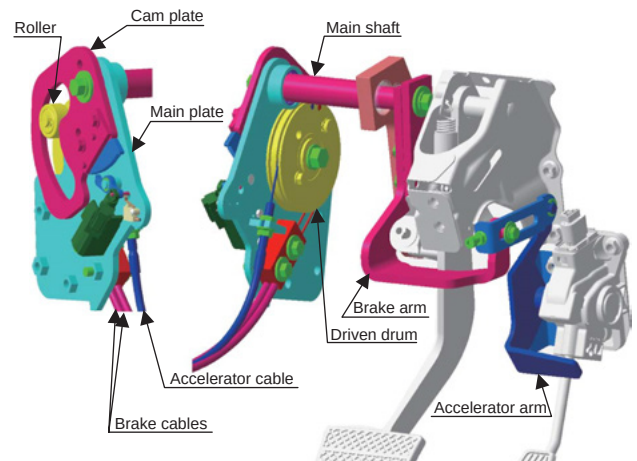


Fig. 12 P.L.A. (Pedal link assembly)

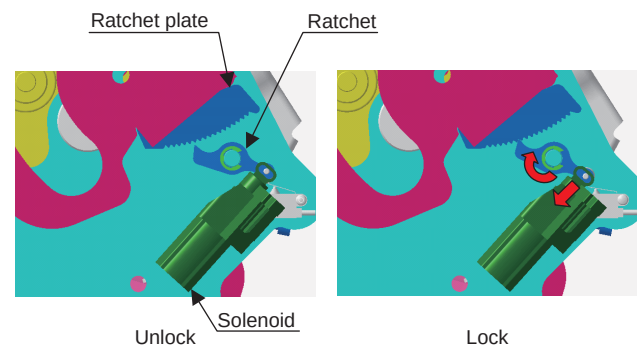


Fig. 13 Brake lock mechanism

rotate and engage the ratchet plate that is fixed in place on the cam plate, temporarily locking the main shaft so that it does not rotate. This serves to hold the brake pedal in the depressed state.

3.2.1. Operation Gain Variable Mechanism

Since the ordinary brake pedal secures a stroke double the normal operating range to cope with the failure of one of the hydraulic systems, the manually operated device should also secure a lever operating range that supports the failure of one of the hydraulic systems. However, the relationship between the lever stroke of conventional manually operated devices and the pedal stroke is determined by the geometry of the link mechanism, which comprises a rotating arm and rods. This made it a challenge to obtain the optimum characteristics over the entire stroke range.

Therefore, a gain variable mechanism that enables active setting of the operation gain (pedal stroke/lever stroke) was realized by the combination of the cam and roller. This allows further lowering of the small gain in the brake normal operating range to reduce the operation load, and increasing the gain to suppress the stroke over the operating range where the stroke increases due to failure of one of the hydraulic systems.

As shown in Fig. 14, the gain variable mechanism comprises an output axis (main shaft), an input axis (driven drum) located away from the output axis, and a roller located above the driven drum offset from the center. The input from the driven drum is output to the brake pedal rotating axis and the coaxial main shaft via the roller and the cam plate.

In a cable system, the number of parts can normally be reduced by locating the driven drum on the brake pedal rotating axis. When the device is actually mounted on a vehicle, however, the rigging of the brake cables can result in severe bending so that the sliding friction of the cables increases. The use of a cam and roller allows the

drum to be positioned lower down while at the same time enabling rotation of the drum to be reversed. As a result, the cables can be rigged to be closer to a straight line, which also contributes to lessening of the operation load and enhancement of cable endurance.

Figure 15 shows the operation gain characteristics chosen for the present device. The horizontal axis is the length of the arc of movement of the tip of the lever, which is taken as the lever stroke, and the vertical axis is the pedal stroke. On the right side of the neutral position, the brake characteristics are shown as a red line, and on the left side, the accelerator characteristics are shown as a blue line.

The brake characteristics include the pedal stroke divided into four regions that are indicated by different background colors. In order from the bottom, these are the normal braking region (light blue), the slam-on-the-brake region (yellow), the hydraulic one-line failure region (pink) and the brake fluid replacement region (lavender).

In the case of a drive drum with which the lever is fixed, the stroke is limited by the stopper bolt. The system was configured, however, so that the maximum pedal stroke would coincide with the maximum lever stroke.

The intermediate characteristics are lower gain and limited operation load in the normal operating region, and higher gain for limited stroke in the operating region where the stroke increases when one of the hydraulic systems fails.

The accelerator characteristics are set similarly so that the maximum pedal stroke coincides with the maximum lever stroke. In the normal operating range, the gain is reduced and the operation load is limited. In the operating range where the throttle is opened wide, the gain is increased and the stroke is limited.

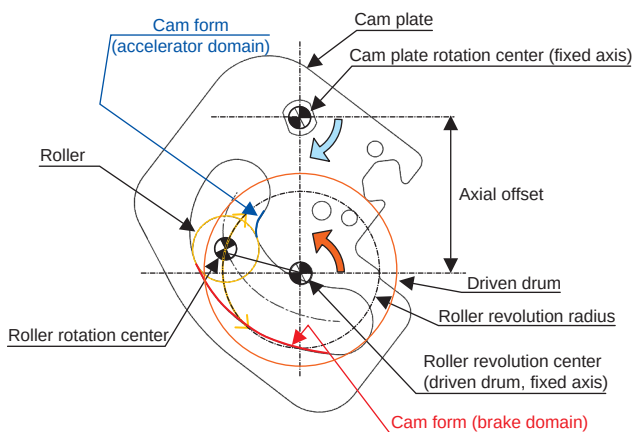


Fig. 14 Gain variable mechanism

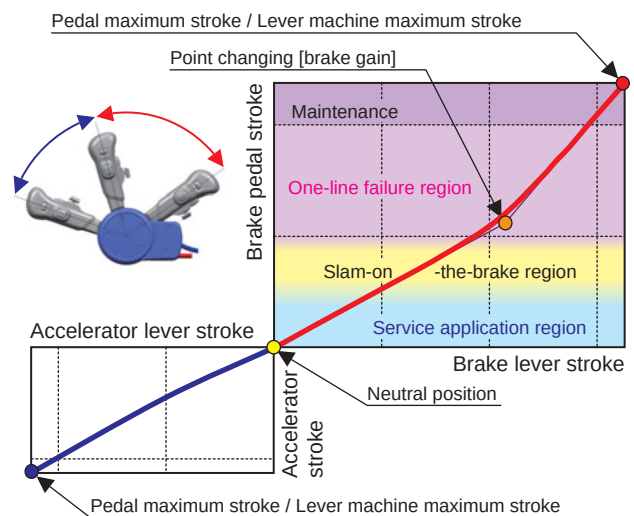


Fig. 15 Input/output characteristics

4. Verification on an Actual Vehicle

4.1. Operability

Figure 16 shows the lever stroke and load characteristics when the accelerator is operated. In order to minimize unnecessary play in the device as a whole, ball bearings were used on the main rotating axis. This enabled the position of Point A in Fig. 16, where the accelerator pedal starts to move, to be limited to 10 mm or less.

The distance to Point B shown in Fig. 17, where the brake pedal starts to move, was a similarly small 5 mm. This enabled smooth lever operation during continuous operation from one side of the neutral position to the other. It also allowed the load at Point A in Fig. 16 to be limited to 12 N or less and realized characteristics equivalent to those of the accelerator pedal on the base vehicle.

The brake operation characteristics shown in Fig. 17 allow the driver to clearly perceive Point C, where the brakes actually become effective. The driver can control braking from that point while receiving a definite sense of tactile response in the lever. This yielded operation feeling equivalent to that of the base vehicle.

To enable users to “enjoy driving,” it is considered essential to set the proper load and hysteresis for controls,

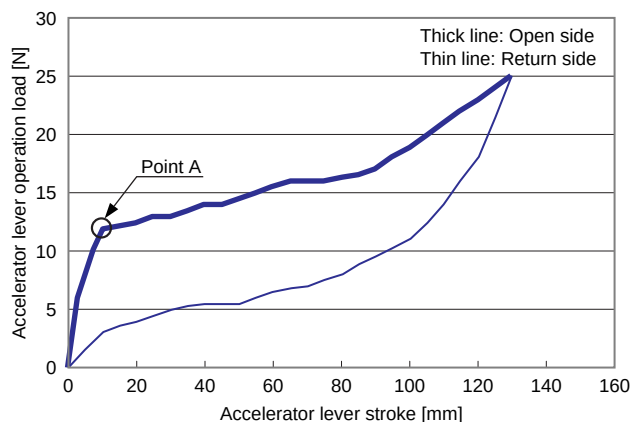


Fig. 16 Accelerator operation characteristics

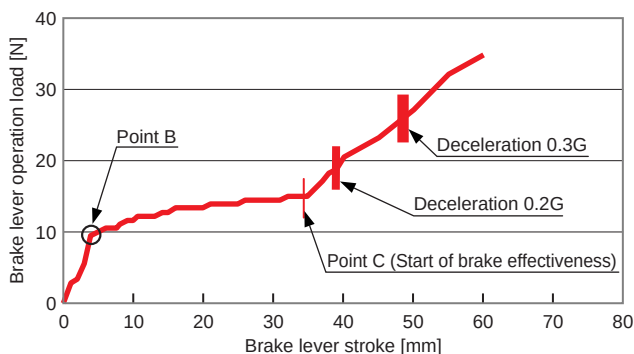


Fig. 17 Brake operation characteristics

and to minimize unnecessary motion. Cables and a gain variable mechanism were selected as the means of achieving this, and realized a direct sensation as though the driver were pressing directly on the pedal.

4.2. Mountability

The manually operated auxiliary driving device here was developed on the assumption that it would be mounted on vehicles by dealers. In order to secure assembly reliability at the dealers, therefore, it was decided to ship the system with the C. L. A., the P. L. A., and the cables already assembled and with the tension in the cables adjusted in advance.

Installation of the parts makes use of existing attachment points, requiring no new holes to be opened in the vehicle at the dealer. Further measures, such as the use of protective members, were also taken to prevent damage to the cables.

Figure 18 is a diagram of the system assembled and mounted on a vehicle. The only parts that have to be removed once before installing the system are the center console and other such resin parts.

These measures yielded a reduction of approximately 20% in installation labor compared to Honda's conventional product.

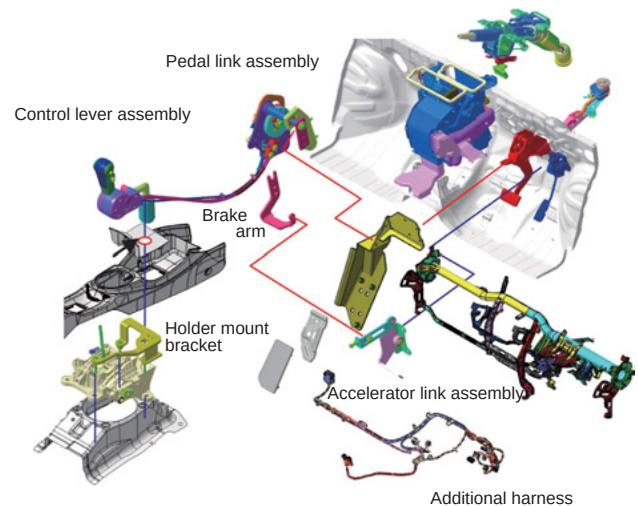


Fig. 18 Construction of New Honda Techmatic System

5. Conclusion

A new manually operated auxiliary driving device was developed and mounted on the FIT.

Safety and reliability as official Honda product features have been secured, and in terms of operability, major advances from just “the ability to drive” to “enjoying driving (Fun for Everyone)” were made, and the system is now compatible with various car models and is easy to install as well.

The authors are confident that the many features not previously available will delight an even greater number of customers.

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Verification Test of Home Energy Station IV System Using Natural Gas Reforming

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ABSTRACT

A compact hydrogen supply system (Home Energy Station, HES) has been developed that reforms natural gas to supply hydrogen to fuel cell electric vehicles at households, while at the same time also enabling supply of electricity and heat. The Home Energy Station IV (HES IV) is the 4th-generation prototype model of this system, and was operated to verify both hydrogen supply to a Honda FCX and power generation that can be interconnected with the commercial power grid.

The results showed that introducing the HES IV together with a fuel cell electric vehicle to a typical household can reduce CO₂ emissions by 30% and energy costs by 54%. Verification tests demonstrated that this is a new energy supply system format that can simultaneously reduce the environmental load while providing merits to users.

1. Introduction

Honda has for some time considered hydrogen fuel cells as the ultimate next-generation clean power plants, and worked to develop fuel cell electric vehicles from the viewpoints of finding alternatives for fossil fuels, reducing exhaust emissions, and reducing the effect on global warming.

Achieving widespread acceptance of fuel cell electric vehicles requires the social approach of constructing an infrastructure for supplying hydrogen. Recently, research and development on these technologies has been led by the infrastructure industry and public agencies, and the advanced introduction of hydrogen stations is proceeding. However, regions where these facilities have been introduced are still limited, and the number of locations is insufficient considering widespread future use.

Honda therefore has promoted research on a compact hydrogen supply system that can be installed in households, with the aim of early widespread acceptance

of fuel cell electric vehicles. Experimental operation of a Home Energy Station (HES) that generates hydrogen from natural gas and supplies it to a fuel cell electric vehicle, and has a cogeneration function to supply electricity and heat to a household, commenced from 2003 at the Honda R&D Americas, INC. Los Angeles Center (California, U.S.A.)⁽¹⁾⁻⁽³⁾. Natural gas, selected as the raw material for hydrogen production, is considered to be the most appropriate fuel for promoting the early spread of fuel cell electric vehicles. This is because the infrastructure for supplying natural gas to households already exists, and less CO₂ is emitted when producing hydrogen compared to other fossil fuels.

Verification tests have been continuing since 2007 on a 4th-generation prototype model HES IV (Fig. 1) that has been downsized and made more efficient, based on the assumption of its more realistic use within households. This paper introduces the HES IV, and reports the energy cost merits and CO₂ reduction effects of practical application calculated based on the verification test results.

* Automobile R&D Center

** HONDA R&D AMERICAS, INC.



Fig. 1 External view of Home Energy Station IV

2. Overview of HES IV

2.1. External View

Figure 1 shows an external view of the HES IV. The HES IV comprises a main unit that produces compressed hydrogen and generates electricity, a dispenser unit that fills the hydrogen tank of a fuel cell electric vehicle with hydrogen, and a hot water tank that accumulates the exhaust heat as hot water.

The system configuration was reviewed and the auxiliaries were downsized in consideration of its installation in households, and resulted in the main unit successfully being downsized by approximately 70% compared to the 1st-generation prototype model. The system is designed for outdoor installation, but it is more realistic to assume that filling work will be performed with the fuel cell electric vehicle parked inside a garage, so the configuration uses a separate dispenser unit that enables a flexible layout.

A touch panel control monitor that enables simple operation by users in an interactive format is installed in the dispenser unit (Fig. 2).

Figure 3 shows the main unit and dispenser unit sizes.

2.2. System Configuration and Specifications

Figure 4 shows the HES IV system configuration. The main system components are a fuel processor, dual mode

stack (DMS), pressure swing adsorption (PSA), hydrogen compressor, and interconnection inverter.

The fuel processor employs auto thermal reforming (AFR) using a reforming catalyst and an oxidation catalyst



Fig. 2 Dispenser unit with touch panel controller

as the natural gas reforming method to shorten the start-up time. A water gas shift (WGS) reactor and a CO preferential oxidation (PROX) reactor are located after that to reduce the carbon monoxide (CO) in the reformat.

The DMS device has both a hydrogen membrane pump function using a proton exchange membrane, and a power generation function as a fuel cell. Figure 5 shows a schematic drawing of these two functions. When functioning as a hydrogen membrane pump, the hydrogen in the reformat component is conveyed selectively to the cathode side using externally supplied electricity [Fig. 5(a)]. At the same time, the hydrogen pressure is boosted to the pressure required for PSA operation. The hydrogen transferred from the DMS contains moisture and minute levels of impure gas, so combination with the PSA secures a hydrogen purity that can be supplied to a fuel cell electric vehicle.

On the other hand, when the DMS generates electricity, air is supplied to the cathode and the DMS functions as a fuel cell [Fig. 5(b)]. The conventional system required a separate pressure booster device for hydrogen refining and fuel cell for electrical generation. However, the DMS enabled switching between both functions with a single device, which enabled a smaller system.

The hydrogen production capacity was set at 3 Nm³/h, which can fill enough hydrogen within 6 hours for the

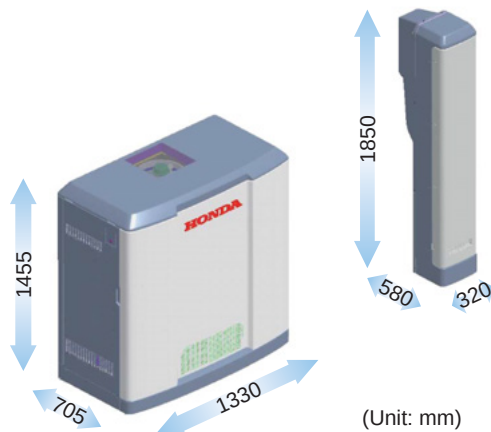


Fig. 3 Dimensions of HES IV

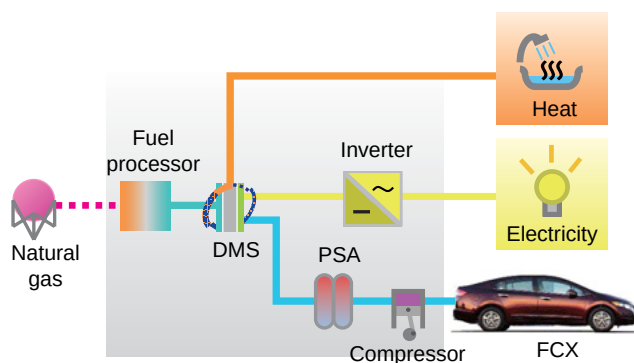


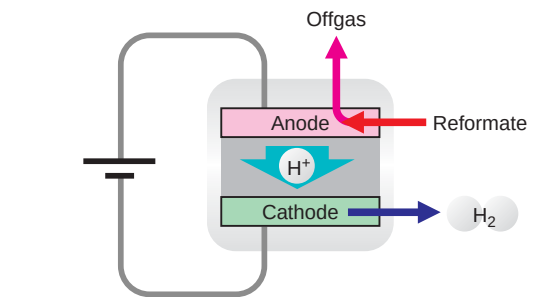
Fig. 4 System configuration of HES IV

Table 1 Specifications of HES IV

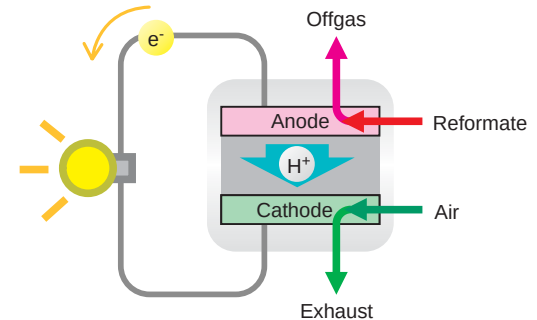
Hydrogen production	Flow rate	3 Nm ³ /h
	Filling pressure	35 MPa MAX
	Purity	> 99.99%
	Efficiency	40%
Power generation	Rated output	4 kW
	Voltage / Frequency	120/240 VAC / 60 Hz
	Efficiency	30%
Heat recovery		80°C hot water
Utility	Electricity	120/240 VAC
	Gas	Natural gas
	Water	City water
Fuel processor		ATR + WGS + PROX
Purifier		DMS + PSA
Compressor		4-stage piston compressor

FCX to run 100 miles. This assumes that the user connects the dispenser to the fuel cell electric vehicle after returning home in the evening, fills the vehicle's hydrogen tank with hydrogen through the night, and then drives the vehicle again the next morning. In this manner, the HES IV does not have a hydrogen storage tank, and instead employs the slow fill method that fills the vehicle's hydrogen tank directly from the compressor.

In addition, the power generation capacity was set at 4 kW, which is capable of supplying sufficient electricity to a typical household in the Los Angeles district. Table 1 lists the main specifications of the HES IV.



(a) Hydrogen membrane pump



(b) Fuel cell

Fig. 5 Two functions of Dual Mode Stack

3. Operating Modes

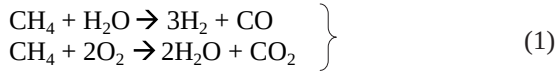
The HES IV operates while switching between two operating modes. In hydrogen production mode, compressed hydrogen, generated by the main unit, is filled directly to the hydrogen tank in a fuel cell electric vehicle via the dispenser unit. In power generation mode, electricity, generated by the DMS, is converted to AC by the inverter and used as electric power.

3.1. Hydrogen Production Mode

System operation in hydrogen production mode is as follows.

(1) Fuel processor

The natural gas that is supplied to the fuel processor undergoes the reaction shown in Eq. (1) on the surface of the ATR catalyst, together with the simultaneously supplied water and oxygen (air), and is reformed into hydrogen-rich reformat. After that, the hydrogen concentration of the reformat is increased by the CO shift reaction shown in Eq. (2) on the shift catalyst in the WGS. The CO concentration is then reduced to 10 ppm or less by the preferential oxidation action of the PROX catalyst as shown in Eq. (3), and ultimately reformat with a hydrogen content of approximately 40% to 50% is supplied to the DMS.



(2) DMS

Hydrogen is selectively separated by the proton exchange membrane by having the DMS function as a hydrogen membrane pump. In this stage the hydrogen concentration is increased to approximately 99%; at the same time the pressure is boosted to 0.8 MPa and the hydrogen is sent to the PSA.

(3) PSA

Non-hydrogen gases are eliminated by the PSA, and the hydrogen purity is increased to 99.99% or higher, which enables filling to a fuel cell electric vehicle.

(4) Compressor

The hydrogen is compressed by the hydrogen compressor and filled to the fuel cell electric vehicle.

Note that the auxiliary power needed when operating in hydrogen production mode is supplied from the commercial power grid.

The user can fill the vehicle with hydrogen at an optional time by operating the touch panel. The system stops automatically when the hydrogen filling pressure of the vehicle's hydrogen tank reaches 35 MPa, but the user can also designate that filling stop when an optional pressure is reached.

3.2. Power Generation Mode

System operation in power generation mode is as follows.

(1) Fuel processor

Like in hydrogen production mode, the natural gas supplied to the fuel processor is reformed into hydrogen-rich reformat and supplied to the DMS.

(2) DMS

The DMS generates power as a fuel cell using the reformat supplied to the anode side and the air supplied to the cathode side.

(3) Inverter

The electric power generated by the DMS is converted to AC by the inverter and used interconnected with the commercial power grid.

In power generation mode, the system operates automatically based on the power consumption of each household, but the system can also be started up and stopped as designated by the user.

4. HES IV Verification Tests

4.1. Hydrogen Production Mode

Figure 6 shows an example of the amount of produced hydrogen and the change in pressure inside the vehicle's hydrogen tank when the system is operated to fill the FCX with hydrogen.

Filling to the vehicle's hydrogen tank started in 10 minutes, and the rated hydrogen output (hydrogen production amount = 3 Nm³/h) was reached in 50 minutes after system start-up. In this example, filling by the slow fill method started from a hydrogen tank residual pressure of 15 MPa, and system operation stopped automatically when the tank was fully filled (tank pressure = 35 MPa) approximately 7 1/2 hours after start-up.

In addition, heat energy of 27.2 kWh was recovered as hot water during the approximately 8 hours of hydrogen production mode operation in this example. This shows that the heat recovery capacity of this system in hydrogen production mode is approximately 3.4 kW.

Figure 7 shows the system flow diagram during hydrogen production mode operation, and the rated hydrogen production efficiency and total efficiency. The hydrogen production efficiency (η_{H_2}) is the value obtained by dividing the generated hydrogen energy (E_{H_2}) by the sum of the input natural gas energy (E_{NG}) and the input electric power (E_{elec}). However, considering the power generation efficiency of the power station and the electric transmission loss up to the household, the input electric power was calculated with an energy efficiency of 38% when it reaches the household. Likewise, the total efficiency ($\eta_{H_2, total}$) is the value obtained by dividing the sum of E_{H_2} and the recovered heat (E_{heat}) by the sum of the input energy.

$$\eta_{H_2} = E_{H_2} / (E_{NG} + E_{elec} / 0.38) \quad (4)$$

$$\eta_{H_2 total} = (E_{H_2} + E_{heat}) / (E_{NG} + E_{elec} / 0.38) \quad (5)$$

4.2. Power Generation Mode

Figure 8 shows the electrical output when the system is operated for approximately 6 hours in power generation mode.

The DMS commenced power generation as a fuel cell in 10 minutes, and reached the rated power output of 4 kW

in 1 hour after system start-up. The DMS electrical output at this time was 5.5 kW (DC). The difference between the DMS electrical output and the system output is due to the electric power consumed by auxiliaries and the energy loss from DC/AC conversion inside the inverter.

Heat energy of 23.8 kWh was recovered as hot water during the approximately 6 hours of power generation mode operation in this example. This shows that the heat recovery capacity of this system in power generation mode is approximately 4 kW.

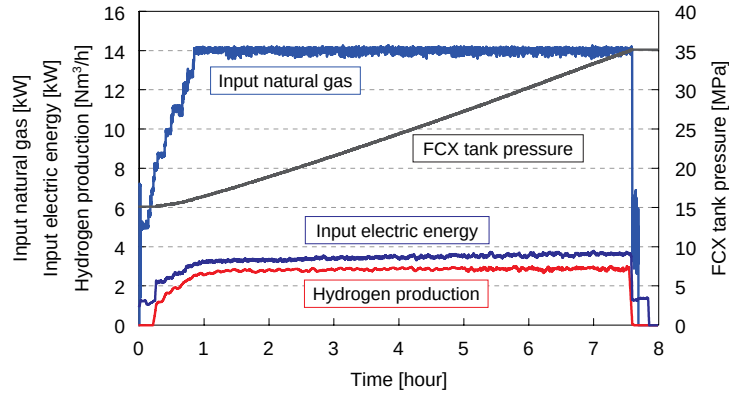


Fig. 6 Example of hydrogen production

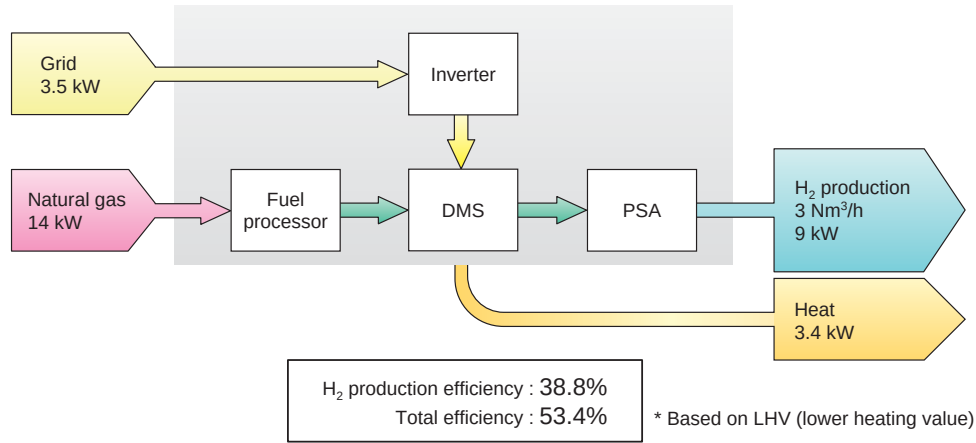


Fig. 7 System efficiency in hydrogen production mode

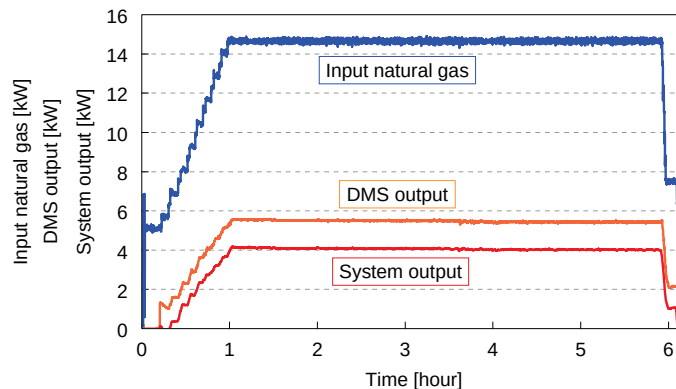


Fig. 8 Example of power generation

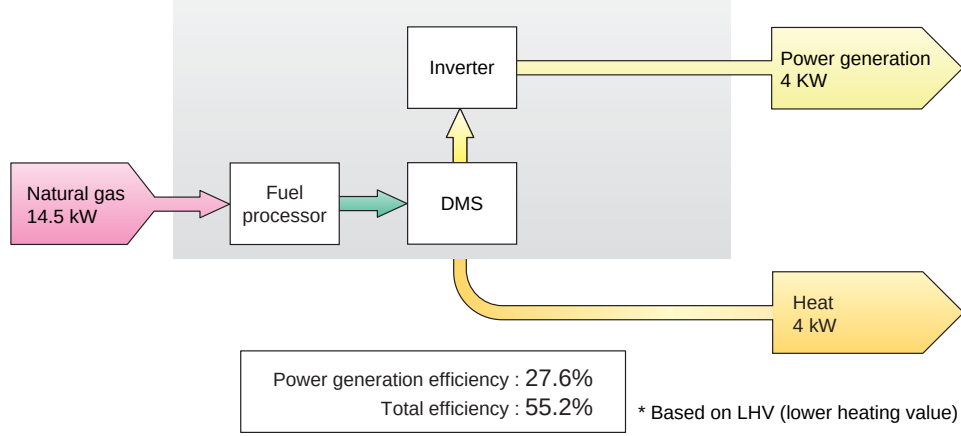


Fig. 9 System efficiency in power generation mode

Figure 9 shows the system flow diagram during power generation mode operation, and the rated power generation efficiency and total efficiency. The power generation efficiency (η_{AC}) is the value obtained by dividing the system electric output (E_{AC}) by the input natural gas energy (E_{NG}), and the total efficiency ($\eta_{ACtotal}$) is the value obtained by dividing the sum of E_{AC} and the recovered heat (E_{heat}) by the input natural gas energy (E_{NG}).

$$\eta_{AC} = E_{AC} / E_{NG} \quad (6)$$

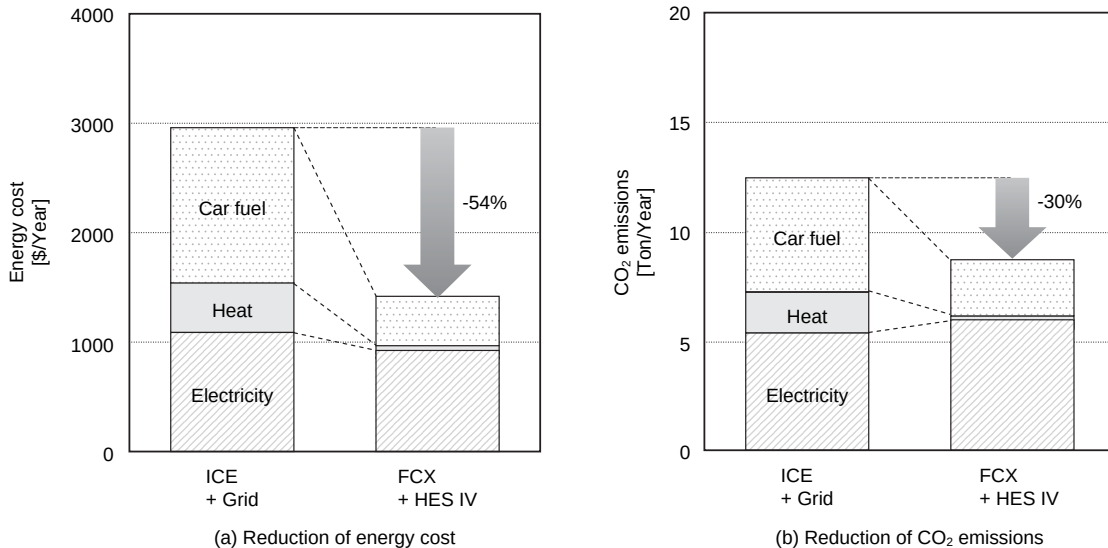
$$\eta_{ACtotal} = (E_{AC} + E_{heat}) / E_{NG} \quad (7)$$

5. Energy Cost and CO₂ Emissions Reduction Effects of Introduction

The energy cost and the effects to reduce household CO₂ emissions when the HES IV is introduced to a household were calculated based on the series of data obtained by the verification tests. The effects were

estimated by comparing the cases when using a conventional gasoline-fueled reciprocating engine vehicle and the commercial power grid (ICE + Grid), and when introducing the combination of the HES IV and the FCX. The amounts of electric power and heat energy consumed by the household used as the basis for the calculations were the average annual energy consumption amount and the energy cost for a typical household in the Los Angeles district (California, U.S.A.). In addition, the annual distance traveled by the vehicle was set at 10000 miles. Note that the CO₂ emissions from the vehicle were evaluated as the amount generated from extraction of the fuel to running (Well-to-Wheel).

Figure 10 compares the effects on annual energy cost and CO₂ emissions for a typical household. This shows that system introduction reduces the energy cost and CO₂ emissions due to heat use to almost zero. This means that almost the entire heat demand required by the household can be covered simply by the heat energy that can be


 Fig. 10 Reduction effects of energy cost and CO₂ emissions in typical household

recovered in hydrogen production mode and power generation mode. Focusing on the amount of CO₂ emitted by vehicle movement [Fig. 10(b) Car fuel], hydrogen is generated by reforming natural gas, so while CO₂ emissions cannot be completely eliminated, these results indicate that the emissions amount can be reduced by half, compared to the conventional method.

The calculation results showed that using the HES IV and the FCX together can reduce the energy cost by 54% and simultaneously reduce the amount of CO₂ emitted from the entire household by 30%. This provided prospects for establishing the HES IV as an energy supply system that provides cost merits to users while also reducing the environmental load.

6. Conclusion

Verification tests were performed for a Home Energy Station that aims to address energy issues and reduce the environmental load, and the following results were obtained.

- (1) A new energy supply system format was presented that can not only fill a fuel cell electric vehicle with hydrogen at the household, but also supply both the electric power and heat required by that household.
- (2) The system was downsized by approximately 70% compared to the 1st-generation prototype, which realized a size capable of installation in a typical household.
- (3) Introducing the combination of the HES IV and the FCX to a household can reduce the energy cost by 54% and simultaneously reduce CO₂ emissions by 30%.

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Development of New Honda 6-speed Automatic Transmission for MDX

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ABSTRACT

A new Honda 6-speed automatic transmission (AT) for the MDX has been developed with the aims of enhancing fuel economy, raising efficiency, and achieving greater compactness. The unit was built on a parallel-shaft structure similar to the previous Honda AT, which has high torque transmission efficiency. The new AT incorporated an additional gear stage and the ability to handle higher input torque from the engine. Meanwhile, providing for a common gear train and placement of the forward-reverse shift mechanism on the input shaft were performed. As a result of these and other measures, the overall length of the new transmission becomes 18 mm shorter than the 2007 model 5-speed AT for the MDX. A multi-plate lock-up clutch (LC) structure with a separate chamber in the torque converter was also adopted so that LC capacity could be increased and the LC control area expanded. In combination with the new hydraulic control system, this new configuration achieves a 5% or greater increase in fuel economy over that of the 2007 model year MDX while realizing greater power and driving performance with enhanced direct feeling.

1. Introduction

Global warming due to increasing levels of greenhouse gases is a shared global issue, and responses are demanded from all sectors of industry. There are growing demands on the automotive industry to increase fuel efficiency in order to reduce CO₂ emissions, and urgent responses are necessary. In addition, the achievement of increased commercial appeal and enhanced drive feel is also demanded.

In the area of automotive transmissions, Honda has worked to increase fuel efficiency through measures including employing CVT⁽¹⁾,⁽²⁾ and IMA⁽³⁾, and expanding the LC control range, primarily in small and medium-sized vehicles. A 6-speed AT has now been developed as a new structure with the aim of increasing fuel efficiency in large vehicles.

The total length of the new 6-speed AT has been reduced, and at the same time, increased fuel efficiency as well as enhanced dynamic performance and drive feel have been achieved. This paper will discuss the aims for the development of the new 6-speed AT and the main technologies employed in the unit.

2. Development Goals

Controlling engine speed during vehicle operation to a low level reduces the amount of fuel consumed, and the use of an additional gear stage is therefore an effective method of increasing fuel efficiency. When shift clutches and shift gears are added to a conventional parallel-shaft transmission, the overall length of the transmission increases, and this can affect crash safety by reducing clearance with the body frame, and can reduce the flexibility of installation in car bodies. Given this, it is necessary to reconcile the increase in the number of gear stages with the reduction of the overall length of the transmission unit.

With regard to the torque capacity of the transmission, taking the higher torque of next-generation engines into consideration, it is necessary to design the transmission on the basis of an upper limit target torque for the structure.

In addition, while responding to demands for increased fuel efficiency, it is also necessary to enhance drive feel and increase commercial appeal.

Based on the above, the following three main aims

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were set for the development of the new 6-speed AT:

- (1) Reconcile increase in the number of gear stages with reduction in overall length
- (2) Realize high torque capacity for next-generation engines
- (3) Reconcile increased fuel efficiency and enhanced drive feel

3. Basic Structure and Main Specifications

Figure 1 shows the main section of the new 6-speed AT. The main specifications of the transmission are listed in Table 1. Figures 2 and 3 show a vehicle speed map and the transmission appearance, respectively.

The new AT uses the parallel-shaft structure of a conventional Honda AT, which displays a good friction characteristic. The following innovations have been applied:

- (1) The 2nd, 3rd, and reverse gear trains are shared.
- (2) The 4th, 5th, and 6th gear trains are shared.
- (3) The forward-reverse shift mechanism, previously positioned on the counter shaft, has been positioned on the third shaft.
- (4) Bolt tightening structure rather than nut tightening is used for the counter shaft and secondary shaft tightening.
- (5) A new form of engine brake control has been employed, enabling the low-hold clutch to be eliminated.

These five innovations have enabled the use of an additional gear stage while reducing its overall length by 18 mm.

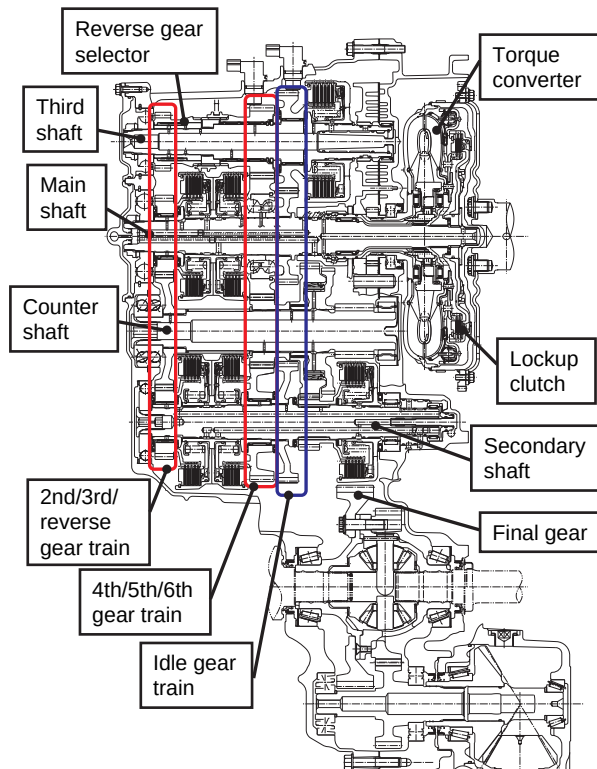


Fig. 1 Main section of Honda 6AT

Table 1 Specifications of 6AT

Transmission type		6-speed automatic
Torque capacity [Nm]		400
Ratios	1st	3.359
	2nd	2.094
	3rd	1.484
	4th	1.065
	5th	0.754
	6th	0.555
	Reverse	2.269
	Idle	1.282
Ratio range		6.05
Overall length [mm]		375

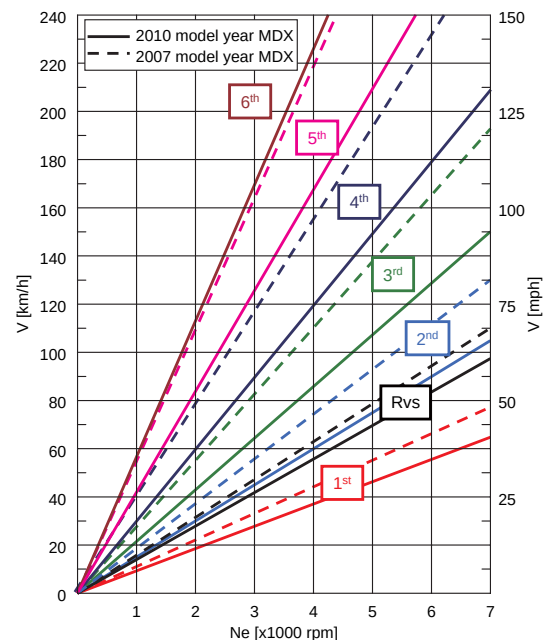


Fig. 2 Vehicle speed map

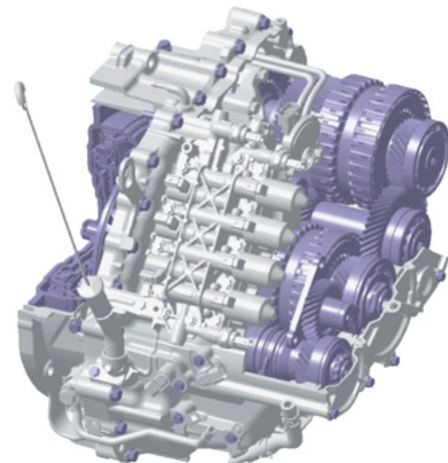


Fig. 3 Appearance of Honda 6AT

With the use of an additional gear stage, it becomes possible to employ wide-range gear ratios, and lower gear ratios at low gear stage. The ratio of 1st gear is 17% lower, and the ratio of the top gear (6th gear) is 3% higher than those of the 2007 model year MDX. These gear ratio settings enabled increased fuel economy to be reconciled with enhanced drive feel.

The use of a thin, wet-type multi-plate clutch with large-diameter⁽⁴⁾ for the LC enabled the LC capacity to be increased and the LC control range to be expanded⁽⁵⁾, helping to realize increased torque capacity and increased fuel economy.

4. Main Technologies

4.1. Study of Strength of Gears, Shafts, and Cases

Using CATIA V5, a stiffness analysis of a full 3D model of the transmission was conducted, enabling high-accuracy prediction of gear and shaft inclinations and the load on the bearings⁽⁶⁾. This analysis enabled the case specifications and multi-shaft gear mechanism to be optimized, resulting in a reduction in overall length and enhanced gear meshing.

All the torque transmission and support parts, including the gears, shafts, and cases, were 3D modeled, and an analysis of stiffness under conditions of maximum engine torque input was conducted with consideration of the clearance of each part and other factors. The analysis model is shown in Fig. 4. The model took gear inclinations, shaft deflections, and other factors into consideration. Because the stress analysis of the cases could be conducted under conditions approximating those in an operating transmission, prediction accuracy was increased, and it was possible to front-load development. Figure 5 shows the results of the stress analysis of the cases.

The pressure distribution on the surfaces of the gear teeth and the cumulative stress on the gear roots were

determined with consideration of the gear and shaft inclinations obtained from the abovementioned analysis⁽⁶⁾. With a strength map of the gear materials obtained using a unit gear tester as a reference, the strength of the gears for each speed and the final gear was examined, and durability predictions were made for the gears. At the same time, the same method was used with the teeth in contact during input of cruise torque, enabling prediction of the effect on gear noise. Figure 6 shows the distribution of pressure on the surface of the final gear. Figure 6(a) shows the cumulative pressure distribution for all gear speeds when full torque is transmitted, prior to correction of the shape of the gear teeth. The pressure is concentrated at the irregular sections of the gear teeth, and the peak pressure is high. Figure 6(b) shows the cumulative pressure distribution for all gear speeds following correction of the shape of the teeth based on the results of the analysis discussed above. The results show that the corrected tooth shape prevents localized concentration of pressure by dispersing pressure across the entire width of the tooth, and reduces peak pressure, thus increasing durability. Figure 6(c) shows the pressure distribution for the corrected tooth shape when 6th gear cruise torque is being transmitted. The results show that while the tooth shape takes durability into consideration, at the same time a good area of contact is obtained in meshing in the standard-use range. Figure 7 shows the results of NV measurements for the final gear. NV measurements of the tooth shape for which noise

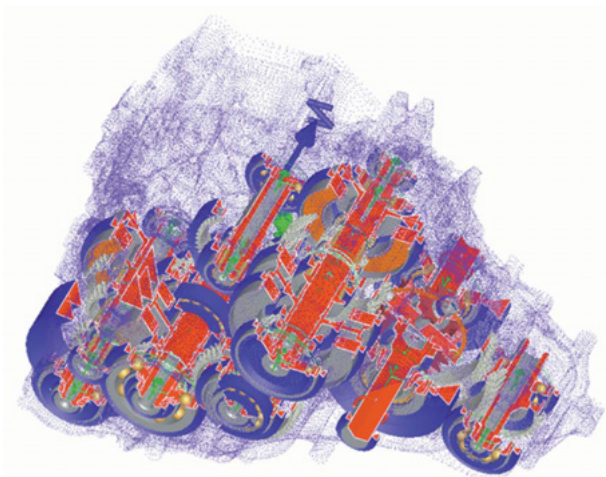


Fig. 4 Analysis model

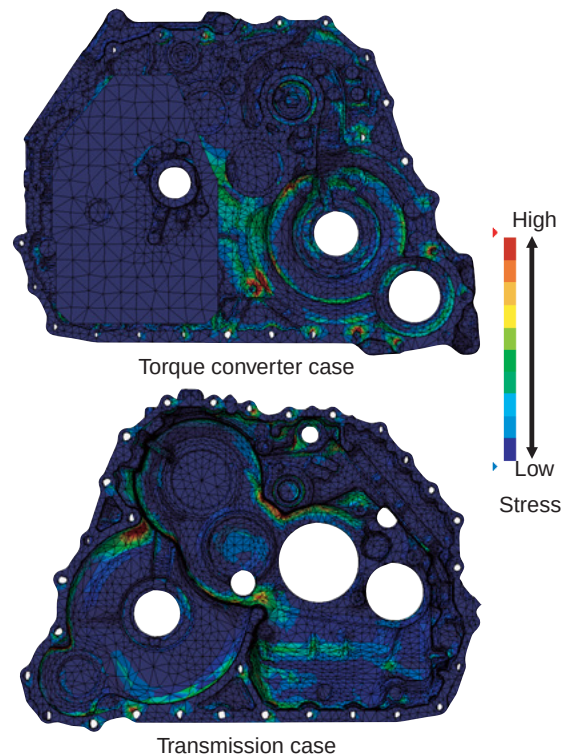


Fig. 5 Stress analysis result for cases

predictions were made for tooth contact during transmission of full torque and cruise torque show that the tooth shape satisfies the targets set for it.

4.2. Multi-plate Lock-up Clutch

A multi-plate lock-up clutch (LC) in a separate housing was employed to enable response to high-torque engine input. Figure 8 shows the oil flow when the torque

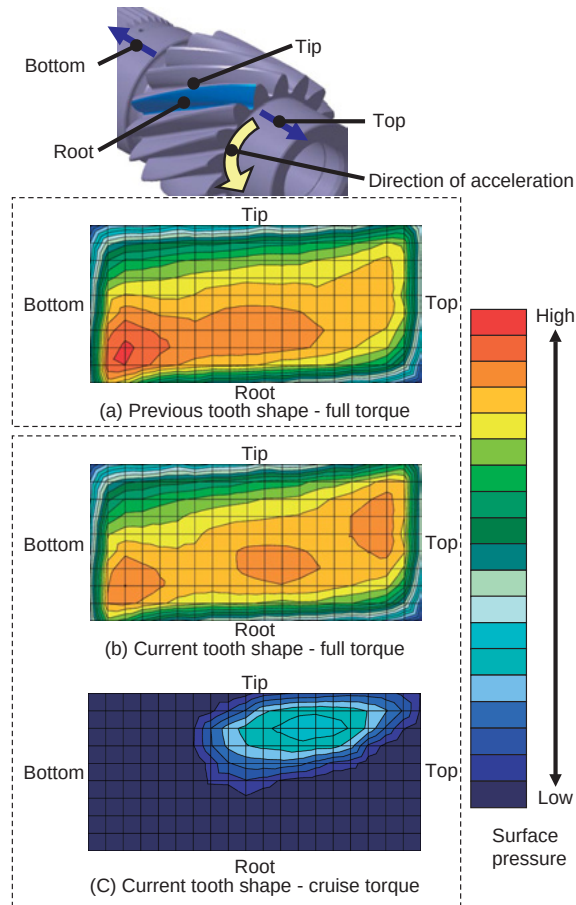


Fig. 6 Pressure map of final gear

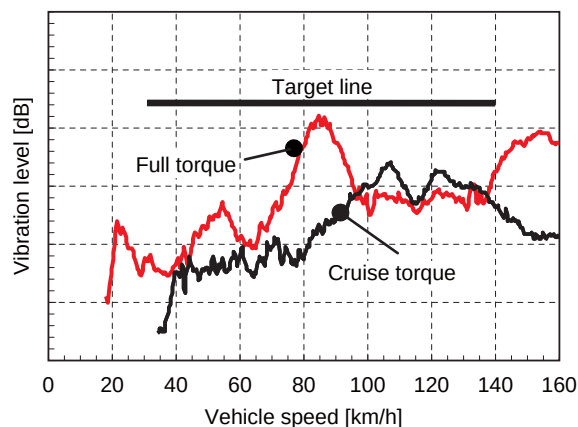


Fig. 7 Final gear NV

converters used in the AT in the 2007 model year MDX and in the developed 6-speed AT are engaged⁽⁵⁾. In order to increase the range for engagement against the former single-plate LC clutch, a wet-type multi-plate clutch with the same structure as the shift clutches has been employed. To help ensure space with the use of a multi-plate clutch, the flatness rate of the torque converter was reduced to 51% from the 65% of the unit used in the 2007 model year MDX. While LC control could be applied simply by adjusting the internal pressure of the torque converter for the single-plate LC, in the case of the multi-plate LC, the LC capacity is controlled by maintaining the internal pressure of the torque converter constant and adjusting the back pressure of the pistons, as in the shift clutches.

In the previous single-plate LC, the LC facings could not be cooled by the oil flow in the torque converter when the LC was engaged. The use of a new LC control method and an LC facing provided with lubrication grooves in the new multi-plate LC has enabled the LC discs to be cooled by the oil flowing in the torque converter, enhancing LC

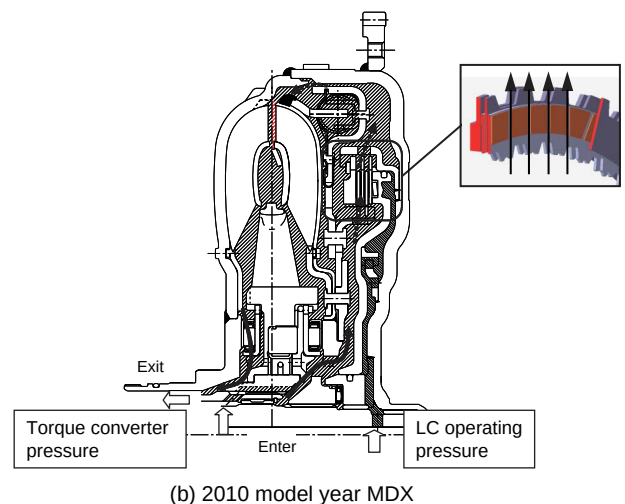
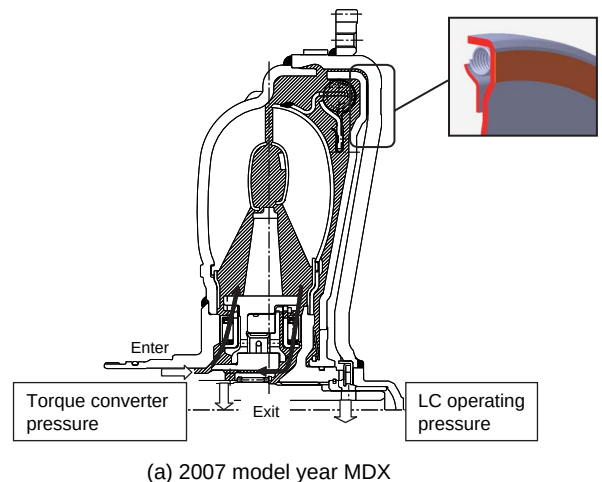


Fig. 8 Structure of torque converter

cooling performance and increasing allowable heat generation. This has enabled the LC to be engaged under transient operating conditions, such as wide-open throttle. Figure 9 shows the LC control range⁽⁵⁾. The expansion of the LC control range to the high throttle opening and low engine speed ranges has helped to increase fuel efficiency and realize a drive feel with a powerful directness.

4.3. Low-friction Hydraulic Circuit

A new hydraulic control circuit was employed in order to increase fuel efficiency and reliability. Figure 10 shows the hydraulic control circuit used in the new 6-speed AT.

In the 2007 model year MDX, the line pressure was maintained constant, while ON/OFF control was applied to the lubrication pressure in D range, and lubrication cut-off control was applied in the P, R, and N ranges. In the new 6-speed AT, two-stage Hi/Lo switching control is applied to the line pressure, reducing the load on the oil

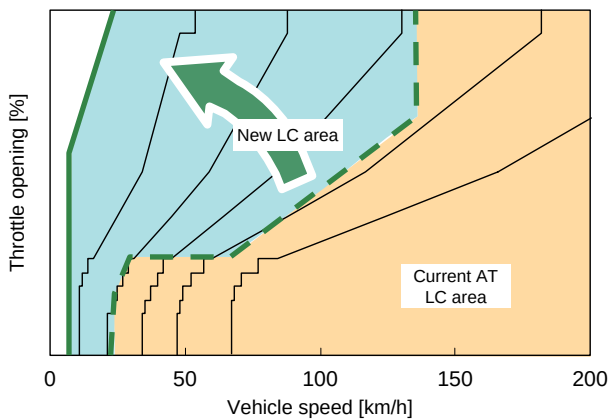


Fig. 9 Lock-up clutch map

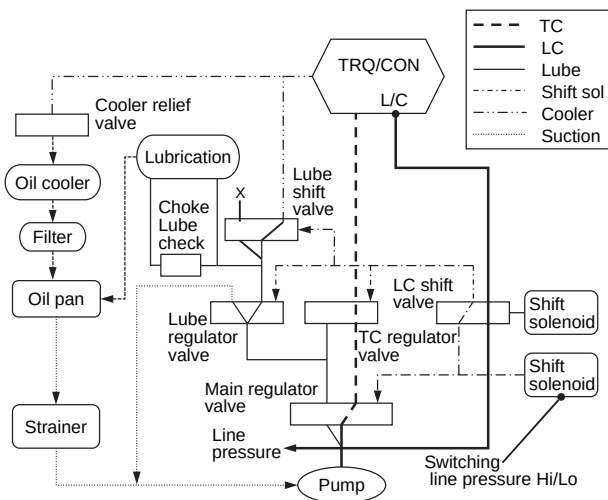


Fig. 10 Hydraulic circuit

pump; lubrication pressure is adjusted in order to help ensure stable supply of lubricating oil; lubrication control is applied in the P, R, and N ranges; and the amount of lubrication at low temperatures is limited by the choke. These measures have helped to ensure the durability of the mechanical parts under high loads, and have reconciled this with the achievement of increased fuel efficiency through the reduction of the load on the oil pump at low loads and the reduction of clutch friction by the lubricating oil.

4.4. Modularized Valve Body

The electrical devices in the 6-speed AT were modularized in order to reduce size, increase quality and performance, and realize greater ease of maintenance. Figure 11 shows the modularized valve body (secondary valve body). Bodies formed from linear solenoids and shift solenoids are in a back-to-back structure with the valve body, which includes accumulator pistons.

In the 2007 model year MDX, clutch pressure was adjusted by means of the linear solenoid units. The modular configuration employed in the new 6-speed AT enables adjustment of pressure and maintenance of hydraulic pressure with the constituent parts of the hydraulic circuit such as the shift valves and the accumulator pistons incorporated, and this enables clutch pressure to be maintained with a higher degree of accuracy. Figure 12 shows the method of linear solenoid pressure regulation used in the modular configuration. With the starting pressure supplied by the line pressure hydraulic channel of the secondary valve body and a prescribed current impressed in the linear solenoid, the adjust cap is pressurized until the prescribed value for pressure in the output hydraulic channel is reached, and the load on the spring set is adjusted. This method of pressure regulation has increased the accuracy of clutch pressure, and has reduced shift shock.

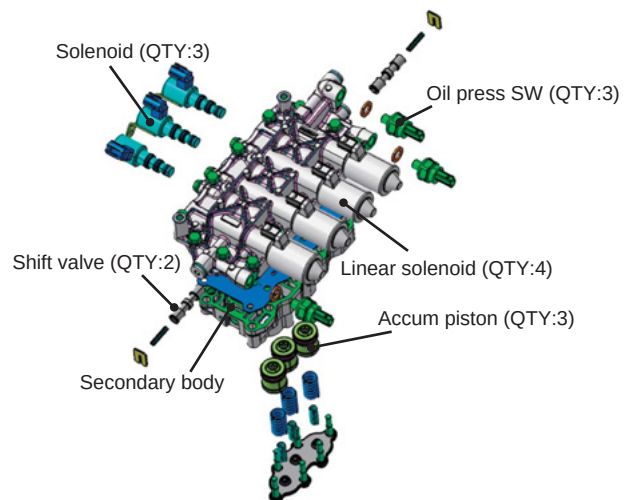


Fig. 11 Secondary valve body assembly

4.5. Lubrication Oil Management

In order to increase cooling efficiency by reducing friction and controlling aeration and to enable the use of oil splashes to lubricate the surfaces of the gear teeth, a plastic baffle plate molded in one piece is installed in the transmission case. Figure 13 shows the baffle plate, and Fig. 14 shows the mounting of the baffle plate and the flow of lubricating oil.

The baffle plate is positioned so as to cover the underside of the main shaft, and the ribs of the case are set around the differential. This has divided the interior of the case into three compartments, for mechanical components, the differential, and the strainer. This arrangement prevents the lubricating oil that is supplied from the shaft to the clutches, gears, and bearings from directly striking the oil around the strainer, thus controlling aeration of the oil. In addition, the form of the baffle plate follows the arrangement of the gears for each speed, enabling lubrication of the surfaces of the gear teeth by the oil flowing on the baffle plate. This has contributed to the achievement of increased durability in the gears.

The baffle plate has been provided with oil supply holes, and the supply of lubricating oil to the differential and the recirculation of lubricating oil to the area around the strainer inlet have optimized the oil level while driving and reduced agitation resistance due to the gears.

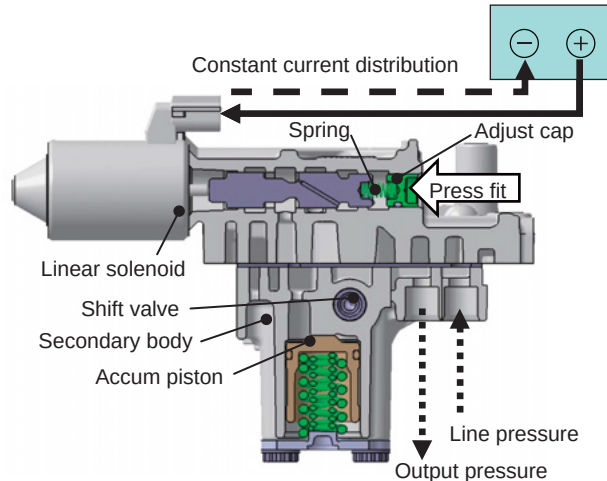


Fig. 12 Linear solenoid pressure regulation method

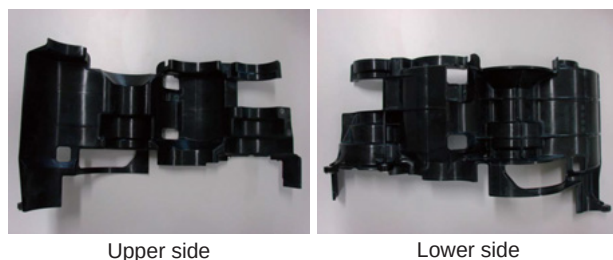


Fig. 13 Baffle plate

4.6. New Engine Brake Control

When the engine brake is applied in an AT vehicle, the shift lever is operated, and the position is shifted to the L range, setting 1st gear as the drive gear. When 1st gear is set as the drive gear in an AT with a one-way clutch below 1st gear, the one-way clutch idles during deceleration, making it necessary to engage a low-hold clutch in order to set 1st gear as the drive gear.

The low-hold clutch has been done away with in the new 6-speed AT by means of the gear ratio settings, shifting during application of the engine brake, and LC control. Figure 15 shows the details of the new engine brake control, and Fig. 16 shows a comparison of deceleration torque with the 2007 model year MDX.

In the new engine brake control system, the vehicle

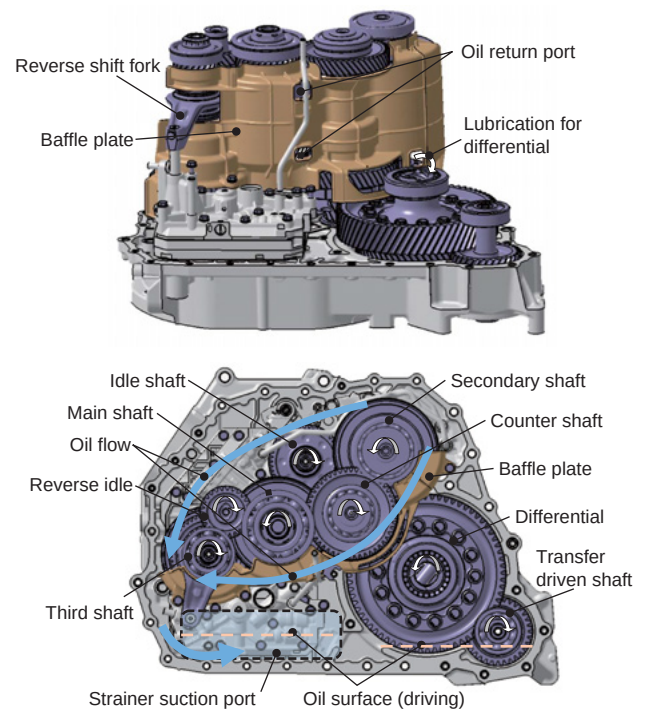


Fig. 14 Assembled baffle plate

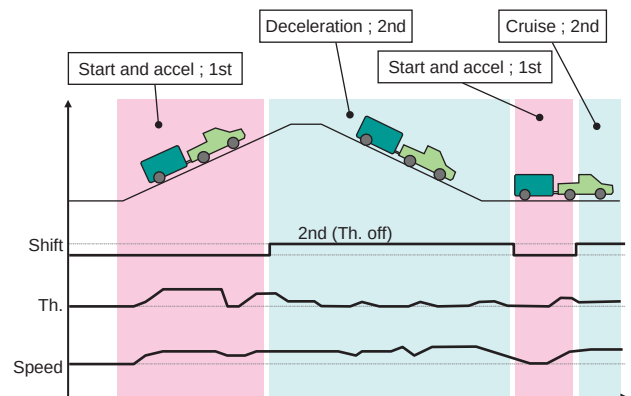


Fig. 15 New engine brake control

is operated in 1st gear when driving at low speed in the L range, and at high speed when the gear shifts to 2nd gear. The system kicks down from 2nd gear to 1st gear when acceleration torque is necessary. When the engine brake is necessary, such as when descending a slope or when the accelerator pedal is fully closed, 2nd gear is set as the drive gear, preventing idling of the one-way clutch. In order to help ensure torque during deceleration, the ratio of 2nd gear has been set 22% lower than the ratio used in the 2007 model year MDX. In addition, the use of a multi-plate LC has enabled the LC to be engaged during deceleration, increasing torque during deceleration by 7% compared to the level when the LC is not engaged. The use of a low ratio for 2nd gear and the engagement of the LC have enabled the realization of an optimum deceleration torque setting in 2nd gear drive, even without the use of the low-hold clutch. The previous use of L range when descending a hill necessitated frequent operation of the accelerator and manual switching between 1st and 2nd gears in order to maintain a high vehicle speed. In the new 6-speed AT, the ratio of 2nd gear is set for linear deceleration torque in relation to the vehicle speed, making frequent accelerator operation and manual switching unnecessary, thus increasing drivability.

4.7. Protection Control

The range of operation of protective control (hardware protection) against user misuse has been increased in comparison to a conventional AT vehicle. The new control achieves the same or higher level of performance compared to that of a conventional AT vehicle. The control encompasses torque limitation, vehicle speed limitation, and shift limitation in order to protect the forward-reverse switching mechanism when the vehicle is put in gear at a high engine speed, forward operation is engaged when the vehicle is in reverse, and the forward-reverse switching mechanism is operated when the vehicle is on ice. In addition, stall torque is limited in order to help prevent an increase in the ATF temperature when the vehicle is continuously stalling.

Table 2 shows an overview of the forms of control

Table 2 New control of mode protection targets

Mode protection target	Protection items	Control	Control target
In-P on ice by mistake	Parking	Speed limit	FI coordination (F/C)
Normal drive	Clutch	Shift limit	AT (shift)
	LC	Slip ratio limit	AT (LC)

newly applied in the developed 6-speed AT for the various control targets. Projecting a situation in which the P range is accidentally engaged when the driver is attempting to get off a snowdrift by switching between forward and reverse and the drive wheels are spinning on the snow, the speed of rotation of the drive wheels is limited in order to protect the parking gear and the parking brake pawl. In addition, a new form of control has been employed in order to control heat generation by the LC and the shift clutches. An ECU predicts the level of heat that will be generated by each clutch, and judges whether or not protective control should be applied. The target value for heat generation is set relatively higher for driving on flat roads, and for the transient use range of the slip control under high throttle opening conditions at low engine speeds or immediately before a kick down, as long as normal LC control is applied. However, when the level of heat generated remains continuously at the target value for a specified period or longer, the target value is changed to the target value for the other operating ranges. If a high level of heat continues to be generated despite this, the LC is disengaged. In addition, for protection of the shift clutches, shift delay control and other forms of control have been employed in order to prevent the temperature of the clutch plates from increasing. These protective control measures were tested in an actual vehicle, and were found not to result in excessive intervention of control, and to have no effect on the normal usability of the vehicle.

5. Fuel Economy and Driving Performance

5.1. Friction and Transmission Efficiency

The use of two-stage Hi/Lo switching control for line pressure in response to the torque that must be transmitted by each clutch and the reduction of oil agitation resistance by means of the use of a plastic baffle plate molded in one piece and modification of the shape of the case have reduced friction torque by 20% at 100 km/h against the 2007 model year MDX. Figure 17 shows measurements of friction torque.

In addition to this reduction of friction torque, the optimization of gear tooth contact with consideration of gear inclinations has increased transmission efficiency during cruising by 0.7%. Figure 18 shows measurements of transmission efficiency during cruising.

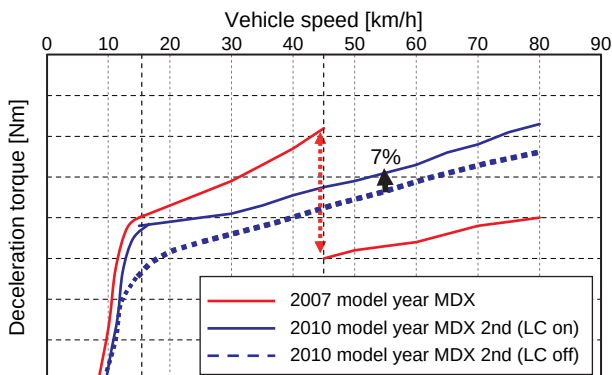


Fig. 16 Deceleration torque

5.2. Fuel Economy

The use of an additional gear stage, the expansion of the LC control range through the use of a multi-plate LC, the reduction of friction torque, and the realization of increased torque transmission efficiency have increased fuel efficiency by more than 5% against the 2007 model year MDX. Figure 19 shows the rate of increase in fuel efficiency against the 2007 model year MDX in each testing mode. Figure 20 shows a breakdown of the different factors in the realization of increased fuel efficiency in U.S. combined mode.

Figure 19(a) and (b) shows the rate of increase of fuel efficiency in the current testing modes and in the new testing modes scheduled for use from 2012, respectively. The new model displays increased fuel efficiency by more than 5% in both the current and new modes. The measurement conditions used in the new modes are particularly close to actual use conditions, and these results therefore suggest that fuel efficiency in actual use has also increased.

A number of factors that have contributed to this

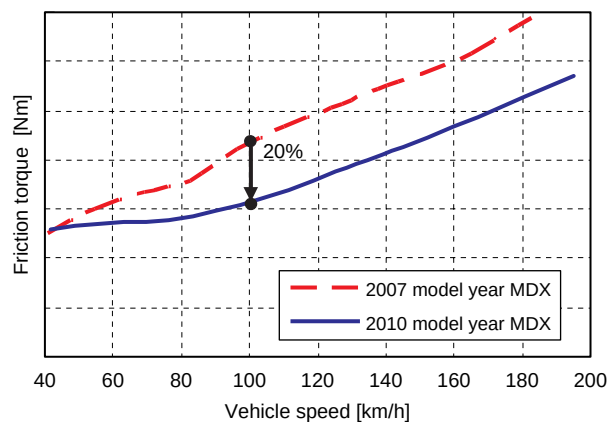


Fig. 17 Friction torque

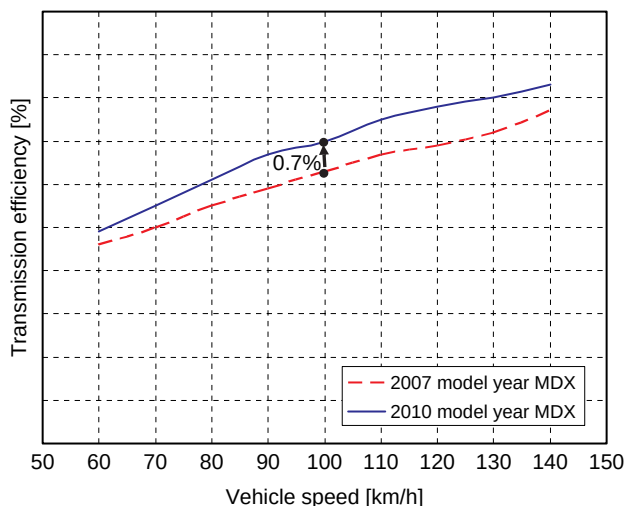


Fig. 18 Transmission efficiency

increased fuel efficiency can be enumerated, including the achievement of increased torque transmission efficiency, the expansion of the LC control range, and the use of an additional gear stage. The achievement of increased torque transmission efficiency has contributed to increased fuel efficiency in all modes. The contribution of the expansion of the LC control range was small in HWY mode, which features little acceleration and deceleration. However, the expansion of the LC control range was confirmed to contribute to increased fuel efficiency in other modes, during acceleration and deceleration and under low-temperature operation. The use of an additional gear stage also had an effect in increasing fuel efficiency by reducing the input engine speed and extending the period for which fuel is cut.

5.3. Driving Performance

Lower gear ratios at low gear stage and wider LC direct drive range have reduced the time for acceleration from 0-60 mph by 0.8 sec and the time for acceleration from 50-70 mph by 0.9 sec with a wide-open throttle against the 2007 model year MDX. In addition, wider LC direct drive range has resulted in the achievement of a drive feel with powerful directness, as obtained from a manual transmission vehicle.

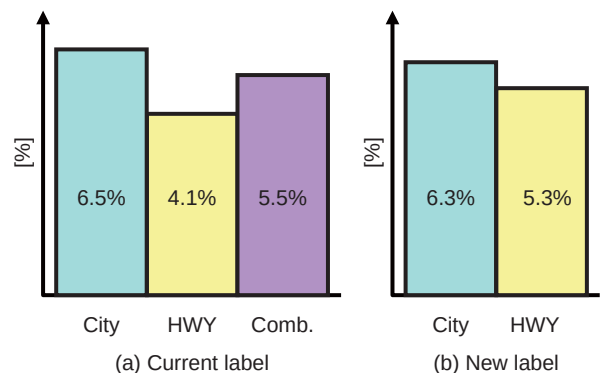


Fig. 19 Fuel economy enhancement

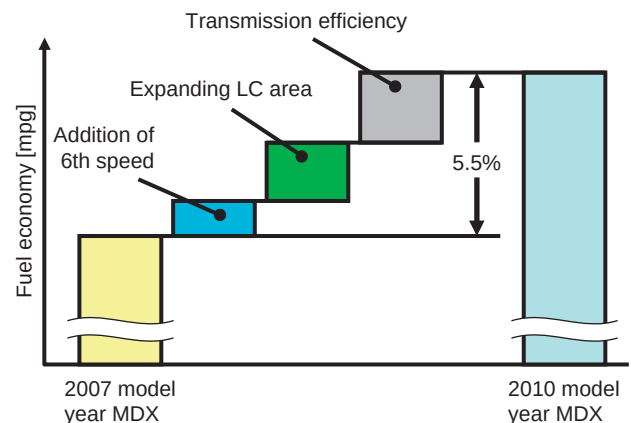


Fig. 20 Detail of fuel economy enhancement

6. Conclusion

A new 6-speed AT was developed in order to increase fuel economy, achieve increased efficiency, and realize greater compactness. The following results were obtained:

- (1) The use of shared gear trains, positioning the idle gear on a separate shaft, positioning the forward-reverse switching mechanism on the third shaft, the use of bolt tightening structure for the shaft, and the application of a new form of engine brake control have enabled the low-hold clutch to be done away with, enabling overall length to be reduced by 18 mm, despite the employment of an additional gear stage, and the realization of increased torque capacity.
- (2) The expansion of the LC control range through the use of a multi-plate LC, and the reduction of agitation resistance through the use of a new form of hydraulic control and lubrication oil management have increased fuel efficiency by more than 5%.
- (3) While fuel economy has been increased, a wider range of gear ratios, lower gear ratios at low gear stage, in addition to wider LC direct drive range, have enhanced driving performance and enabled the realization of a drive feel with a powerful directness.

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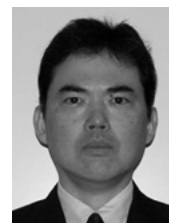
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Using Road Simulation Technology for Durability Verification of Plastic ATV Fenders

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ABSTRACT

To validate the durability of ATV fenders in the lab, it is necessary to apply a testing method that utilizes inertial reactions with large displacements and high accelerations that are present in actual vehicle usage. A relatively simple test rig was devised utilizing an ATV frame, hydraulic actuators, and beam structures. The test rig can reproduce many high amplitude events common to ATVs such as jumps, whoops, and log crossings. By combining this test rig with road simulator technology, a simulation test method was developed. A correlation study to actual riding durability testing demonstrated the test method's ability to reasonably recreate testing conditions and validate ATV fender durability. With this method, efficient and timely part validation was achieved. This simulation method was successfully applied during development and continues to be utilized in additional research and development projects.

1. Introduction

In recent development of new models and revamping of existing models, the plastic fenders on ATVs have typically been parts of major focus. To maintain a styling advantage in the market, new part configuration and construction has been desired. Also with ever increasing economic pressures, finding methods that allow reduced part cost or part count is a focus. These two factors, along with others, lead to designs such as the 2008 TRX400X ATV. This model utilizes a multi-layer fender structure that includes a base fender and an over fender. With this structure, it was possible to get a revolutionary new look while reducing part cost and increasing part toughness.

With this new part configuration, the methods used to evaluate the strength and durability of the fenders in the past may not have been suitable. Methods such as actual vehicle riding durability can take a long time to verify the parts, while methods such as impact testing may not accurately indicate weak areas of the construction. Fatigue testing methods of a point applied load with constant amplitude or a point applied displacement with constant amplitude were also determined not able to accurately indicate the weak points of the structure. Even inertial reacted testing with constant amplitude or frequency was considered to be deficient in accurately

reproducing the actual situation. It was determined that the best method to efficiently test the new construction would be to use an inertia reacted Remote Parameter Control TM test method, more commonly termed road simulation.

This paper describes the techniques used in the creation of a method for applying road simulation technology to verify the durability of plastic ATV fenders.

2. Considerations in designing road simulation test rig

When it was determined that durability testing would need to be conducted on the new fender structure, many test methods were considered. One of the first methods considered, primarily due to simplicity, was to measure deflection at the tip of the fenders during actual vehicle operating conditions, then to conduct a displacement controlled constant amplitude test by applying the load/displacement near the tips of the fenders. The amplitude of displacement would be determined by common strain damage correlation procedures. This test method was deemed inappropriate as the deflection shape of the fender would most likely be different than actual operating conditions. The deflection shape is largely influenced by the inertial input loads due to large accelerations as well as the frequencies at which these inputs occur. The

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geometry of the fender is generally quite complex and has several bolted connections in addition to contact points with the frame body. In addition to strain damage, proper deflection shape is necessary to accurately reproduce wear on contacting surfaces as well as possible slip loads at bolted connections. The fenders are relatively long, wide, and thin, with a significant portion of the mass cantilevered from the frame. During high speed jumps with large accelerations, a distributed load applied to the surface would be more appropriate, yet difficult to reproduce in the lab.

In order to recreate the proper deflection shape, inertial reaction would need to be used. Briefly considered was a method where a single hydraulic actuator would be attached near the center of the ATV frame-body and control the displacement amplitude at either a constant frequency or through a frequency sweep. This method was not chosen because it was thought that it would also not be able to sufficiently reproduce all the deflection shapes that occur in the highly dynamic usage conditions. It was then determined that a method using two vertical inputs to reproduce vertical and pitch motions should be sufficient for a first try at recreating the actual usage. Using existing equipment and fixtures, the first generation test rig was constructed and operational in a very short time. The test rig consisted of 2 hydraulic actuators with 250 mm (10 inches) of stroke capability, an aluminum link with roller bearings on each end attached to a bracket that was bolted to the rear of the frame, a collar welded to the front of the frame with roller bearing, and the ATV frame body with fenders attached. Fig. 1 shows the 1st generation test rig.

3. Preliminary simulations

With the test rig design selected, data from actual vehicle operation needed to be collected. Initially, the best metric (acceleration vs. strain) and locations for the parameter to be reproduced was not certain. With the unknown of which parameters would be the desired signals for reproduction and which would be used for reference, many parameters were recorded simultaneously. Acceleration data was acquired at 2 points on the ATV frame body and 2 points near the tips of the fenders (1 front, 1 rear). Strain measurements were taken at 6 points on the fenders as well. Once the data was collected, the instrumented fenders along with the frame mounted accelerometers were transferred to the test rig.

When the test rig was ready to run, a white noise displacement signal was input into the hydraulic actuator system and the response of each of the above mentioned parameters was measured. All being performed within road simulation software, the frequency response function (FRF) and coherence were determined. It was quickly discovered that the coherence, or how much of the output signal is related to the input signal, of the accelerometers attached to the tips of the fenders and the strain gages were very low, and thus using these as controlling parameters would have caused difficulties in creating good reproductions. Being far removed from the displacement input points; these locations have responses that are heavily dependant on many other factors aside from vertical displacement. With the frame mounted accelerometers attached in close proximity of the displacement input points, coherence was good, and thus

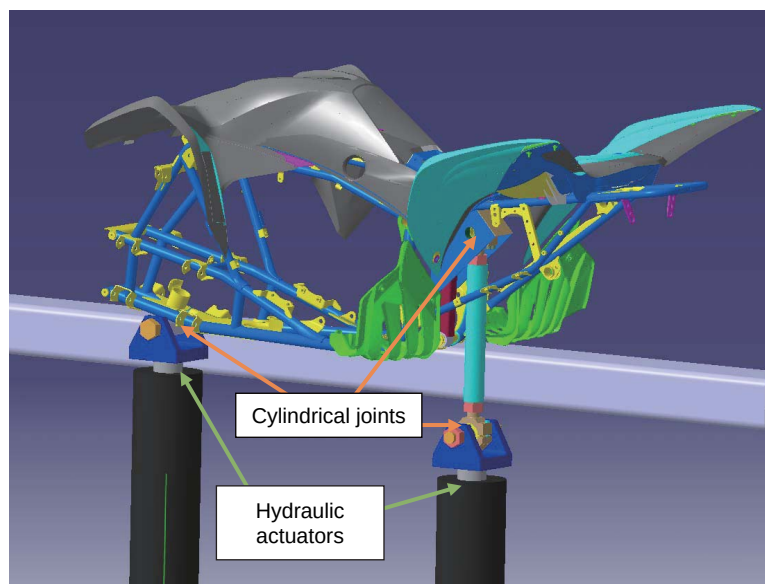


Fig. 1 Computer model of 1st generation test rig

they were chosen as the controlling parameter.

With the measured FRF, the inverse FRF was calculated within the simulation software and the iterative process of reproducing began. During this stage, it was discovered that the available stroke of the hydraulic actuators were not going to be sufficient for reproducing acceleration of several key events a sport ATV experiences such as a triangle jump or running a whoops section. In addition to the lack of stroke capability, it was observed that during other events such as staggered log crossings, the lack of body roll was significantly reducing the calculated strain damage. With these two short falls, it was determined that a more elaborate test rig would need to be developed.

4. 2nd generation test rig and simulations

In determining the design of the next generation test rig, there were 2 primary requirements: (1) there must be adequate displacement to reproduce jump events and (2) body roll motion should be simulated in addition to vertical and pitch motions. Primarily due to the shape and orientation of the fenders, it was determined that forward, lateral, and yaw motions would cause limited additional motion and damage to the parts. To determine the amount of vertical displacement needed, a linear drive estimate was performed using unitary gain in a single iteration on the 1st generation test rig. The displacement that the simulation software indicated plus some additional buffer was calculated to be approximately 500 mm (20 inches). With the rig requirements and much consideration to full DOF (Degree Of Freedom) constraint, the 2nd generation test rig was designed.

As shown in Fig. 2, the method chosen to obtain the required displacement was to use simple lever arrangements with 3 hydraulic actuators. The center pivot of the levers were made such that the fulcrum points could be changed, thus allowing for additional displacement if needed. The ATV frame-body was still used as the base for attaching the fenders as in the 1st generation test rig. The front suspension components were also used with the exception that the dampers were replaced with rigid links. In order to attach the actuator/lever assemblies to the front spindles, simple links with spherical joints at each end were used. The rear of the frame was connected to the third actuator/lever assembly via a single spherical joint. This left the entire structure with a single uncontrolled DOF, so a double arm arrangement was designed to laterally support the front of the vehicle with a spherical joint. The design allowed for the roll axis of the vehicle to be placed easily based on the location of two spherical joints. The roll axis was initially chosen to be the same as the suspension roll axis. This arrangement caused a majority of the mass to be above the roll axis and was quite violent to the test rig structure. The roll axis was then moved closer to frame-body center of mass. This resulted in a much smoother operating rig and had minimal effect to damage correlation of the test parts. An additional revision that was made at a later time was to add 2 additional links. One was between the rear of the body and the actuator/lever assembly. The other was a very long arm between the rear of the body and a reaction fixture. This revision reduced the lateral and fore/aft forces transmitted to the actuator/lever assembly, improving stiffness and test rig component life. Motion capability of final test rig was over 500 mm (20 inches)

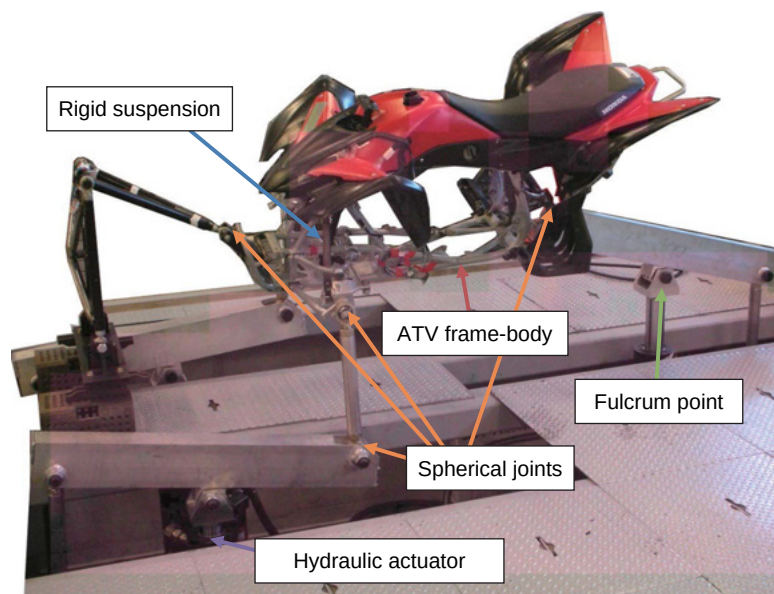


Fig. 2 2nd generation test rig

of vertical motion and more than 20 degrees of both pitch and roll motion.

With the 2nd generation test rig having 3 actuators, there were 3 control channels for the simulation software, requiring 3 response signals for simulation. From findings during the 1st generation preliminary simulations, it was determined that frame body accelerations would be the best control signals. Positions on the frame body were chosen so that each of the 3 accelerometers would be primarily sensitive to only 1 input. Preventing significant cross couplings in the control/response systems aids in quicker and less difficult simulations. Due to the frame body structure of the TRX400X not being very wide in the front, a light-weight rigid beam with the accelerometers secured to it was attached to the front of the vehicle. This allowed for placement of an accelerometer near each of the front actuator/lever assembly connection points. A coherence check confirmed the beam mounted accelerometers to be adequate control signals for simulations with the 2nd generation test rig. A quick try of controlling with the fender tip mounted accelerometers was also checked, but as with the 1st generation test rig, coherence was low. As stated before, maintaining close proximity between input points and response measurement points aids in achieving good coherence.

5. Additional considerations during simulations

One of the parameters to be considered during simulation is the range of frequencies that will be reproduced. With ATVs, the more severe events that one wants to reproduce are jumps, which require large amounts of stroke to reproduce the actual events exactly. When looking in more detail at a jump, it can be broken up into 4 separate events. The first section is the launch. This is when the vehicle changes directions to include some significant portion of vertical velocity. This typically occurs quickly and can be somewhat damaging.

This is shown by the red zone in Fig. 3. The second portion is the upward motion portion shown as green in Fig. 3. This event is not very damaging, but can cover a vertical motion of 2 m (6 feet) or more. The 3rd portion, shown as blue, is the fall. This also is not very damaging, and covers basically the same displacement as the upward motion. The last portion is the impact and suspension oscillations that follow. This is the orange section in Fig. 3. This high acceleration is very damaging for inertial loaded parts. So in considering the jump, damaging events for the fenders occur quickly, or at a relatively high frequency, while the majority of the vertical motion occurring at a relatively low frequency is not very damaging.

An easy way to remove this large, undamaging motion is to apply a high pass filter to the acceleration data. The question then becomes how high of frequency content can be filtered out before integrity of the test begins to diminish. This can be answered a couple different ways. The field data can be filtered and compared to unfiltered data through a damage or pseudo-damage calculation. Another way is drive the test rig with a sinusoidal displacement, starting with a very slow frequency and gradually increasing the frequency while either recording frame to fender tip displacement, strain on the fender, or simply visually observing the frame to fender motion.

The upper frequency limit of the band for simulation is often determined by the design and capability of the test rig. In order to achieve high accelerations, the rig was designed to be light. However, with lightness often comes lower stiffness. This results in lower natural frequencies of the system. Ideally, the rig is designed with detailed CAE analysis to optimize this weight/stiffness balance. Once the test rig is built, it is fairly easy to determine this upper limit. During the FRF measurement process, the coherence will become low once this limit is reached at higher frequencies. This lower coherence means that it will be difficult for the simulation software to accurately reproduce the desired response at higher frequencies. To determine if it should be attempted

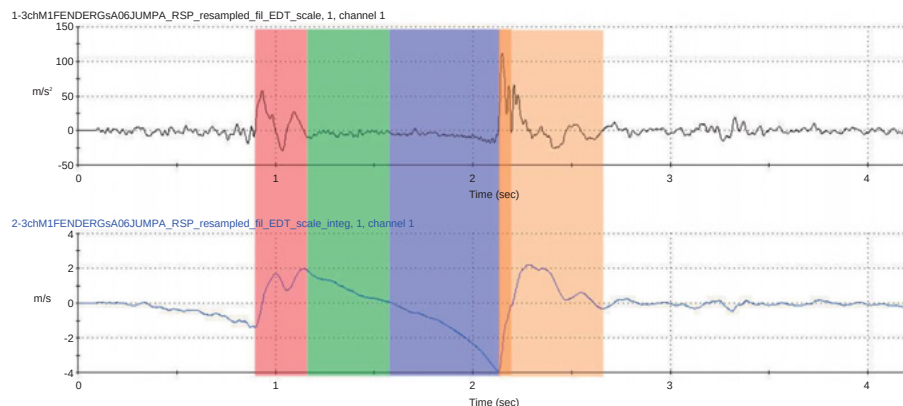


Fig. 3 Acceleration versus time (top) and velocity versus time (bottom) of frame near front left corner of ATV

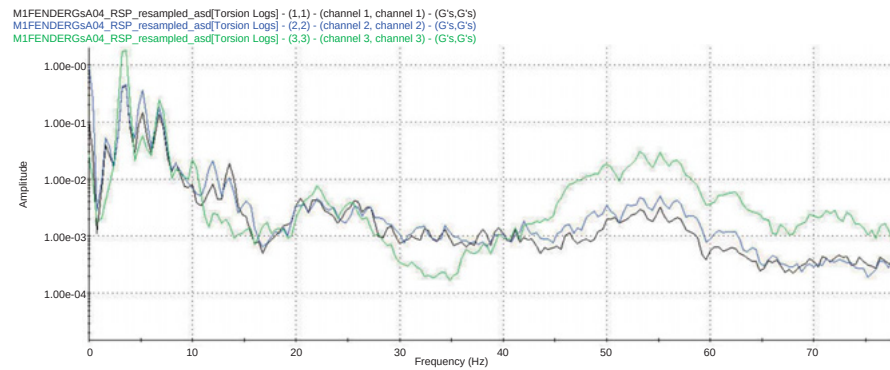


Fig. 4 Spectrum analysis of field data acceleration measurements at 3 different locations

to reproduce higher frequencies, the damage effect of this higher frequency portion of the data needs to be investigated. This can be done as mentioned above by filtering the field data and then comparing the damage back to the unfiltered data. A quicker, but less accurate way, is to simply look at the spectrum analysis of the field data, such as an example shown in Fig. 4. As seen in Fig. 4, there is substantial spectrum content in the 50 to 60 Hz range that could cause significant damage, especially for the location indicated by the green line. Pursuing a simulation to include up to 60 Hz would lead to a more valid simulation for this example.

6. Qualifying results

Once the drive files were created, it was necessary to validate if the simulation was adequately reproducing wear

and damage. To quantify this, strain measurements were taken at the same locations as the initial field measurements. A damage analysis comparison was done to compare field versus lab results. The target for pseudo-damage values from lab testing was to be in the range of 0.5 to 2.0 times damage values from field data. As seen in Fig. 5, only 2 of 6 channels achieved this target. The remaining channels were in the range of 0.25 to 4.0 times. In the case of the initial test for the TRX400X, it was determined that the damage correlation was sufficient. An additional method to verify the simulation was to compare actual damage and wear to vehicle durability testing. This type of analysis was done, and confirmed the 0.25 to 4.0 times damage range to be acceptable. On later projects, limit testing has become more of a focus. In this case, the field acceleration data can be scaled to produce more severe events and accelerate damage accumulation with

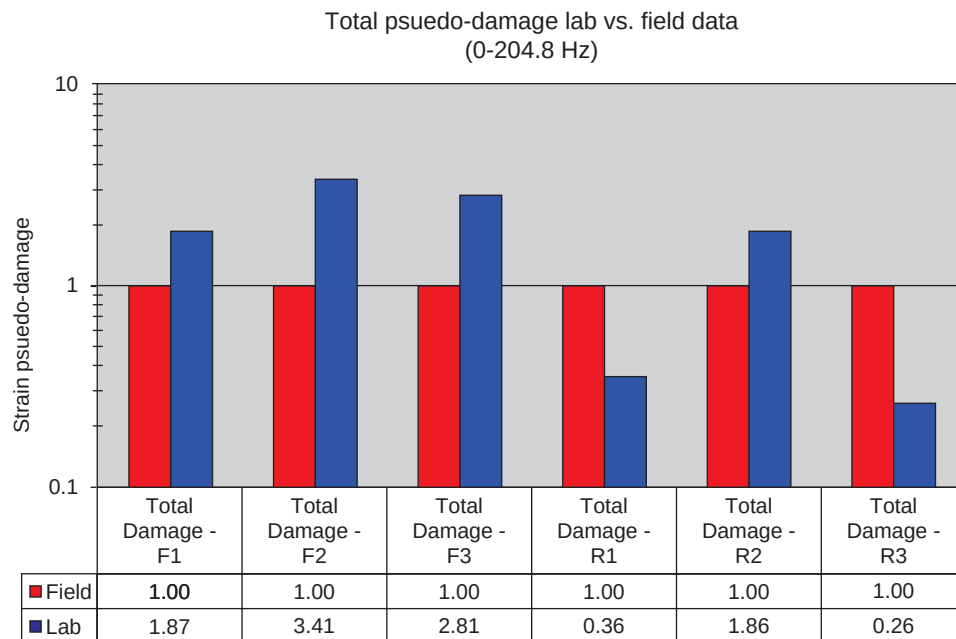


Fig. 5 Pseudo-damage of field and lab data at 6 different locations

relatively accurate results. Through experience with this and other projects, it has been demonstrated that the durability verification of plastic ATV fenders can be done with road simulation technology.

7. Conclusion

A unique test rig and method were developed to validate the durability of ATV fenders. This rig was able to reproduce high displacement events such as jumps as well as higher frequency vibrations. The result was the creation of an effective method for reproducing durability testing and quickly evaluating durability of the ATV fenders.

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Scott Batdorf

Development of High-strength, Nitrocarburized Crankshaft with Control of Vanadium Carbonitride Precipitation by Normalization

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ABSTRACT

A high-strength crankshaft with a high productivity has been developed. This crankshaft development sought to balance high fatigue strength, machinability and straightening ability. Vanadium is known as a precipitation strengthening element, and the development focused on the fact the amount of solid solution of vanadium in steel is dependent on the heating temperature. A material was developed in which a high level of strength could be achieved through control of the normalizing temperature, for which conditions could more easily be set, rather than controlled cooling following hot forging. Optimization of the normalizing temperature helped to enable the achievement of a 30% or more increase in fatigue strength against the existing material, while maintaining machinability and straightening ability.

1. Introduction

In addition to the traditional demands for high power and compactness, the achievement of further weight reductions in motorcycle engines is now demanded in order to increase fuel efficiency and thus reduce CO₂ emissions. This has made the achievement of weight savings in the crankshaft, the heaviest rotating engine part, an important issue. The crankshaft is required to be strong enough to resist the bending stresses generated by the explosive force from the combustion chamber and the inertial forces of the pistons, connecting rods, and other reciprocating parts. The necessity to reduce the weight of the crankshaft increases the load on the component.

Because motorcycle crankshafts pass through a wide variety of machining processes, including lathe machining, milling, drill machining, grinding, and gear machining, the materials employed are required to possess a high level of machinability. For this reason, it is necessary to consider a balance with productivity when seeking to increase crankshaft strength.

The development project discussed in this paper designed a hardness transition curve in an attempt to develop a crankshaft balancing high fatigue strength with machinability and straightenability. Vanadium, known as

a precipitation strengthening element, was added in order to increase fatigue strength⁽¹⁾⁻⁽³⁾, and the heating temperature-dependence of the amount of V in solid solution in the steel was focused on. A material was developed in which a high level of strength could be achieved through control of the normalizing temperature, for which conditions can be set more easily, rather than controlled cooling following hot forging. Optimization of the normalizing temperature helped to enable the achievement of a 30% or more increase in fatigue strength against the conventional alloy steel, while maintaining machinability and straightenability.

2. Development Goals

The majority of integrated crankshafts used in motorcycle engines are manufactured from quenched and tempered or normalized steels to which soft nitriding or nitriding treatments have been applied. The manufacture of crankshafts from quenched and tempered steels is advantageous from the perspective of fatigue strength, but their homogeneous metallic microstructures give them insufficient cuttability. Normalized steels have a ferrite-pearlite mixed microstructure and display better cuttability than steels with a homogeneous metallic microstructure;

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nitrogen also rapidly diffuses in the ferrite microstructures during soft nitriding or nitriding. The steels therefore display good productivity. A steel for which the normalizing process has been omitted but which achieves identical characteristics to normalized steels has also recently been put into practical use⁽⁴⁾.

Previously, the authors had employed a calcium-sulfur free-cutting steel for which the normalizing process could be omitted for standard crankshafts in order to reduce costs. For crankshafts in which high fatigue strength was necessary, a lead-sulfur free-cutting nitrocarburizing steel to which Al had been added in order to obtain high near-surface hardness following soft nitriding was used in order to help assure fatigue strength in areas in which stress concentrated. However, the near-surface hardness of these existing soft-nitrided materials was excessive in relation to the hardness transition curve demanded by the gradient of the stresses on the crankshafts.

The aim of the development was therefore set as the development of a material displaying the following characteristics, in order to reconcile the achievement of high fatigue strength with machinability and straightenability:

- (1) Able to achieve high level of strength without special heat treatments
- (2) Identical level of productivity to existing materials
- (3) Able to be made lead-free

3. Breakthrough Points in Development of High-strength Crankshaft Material

3.1. Design of Hardness Transition Curve

A hardness transition curve was designed prior to the design of the material composition (Fig. 1). In order to increase fatigue strength, it is necessary to obtain a hardness profile that exceeds the hardness transition curve demanded by the stress gradient^{(5), (6)}. To achieve increased strength, the internal hardness would be increased against the hardness transition curve of the existing material, and the nitrided diffusion zone deepened. In addition, the near-

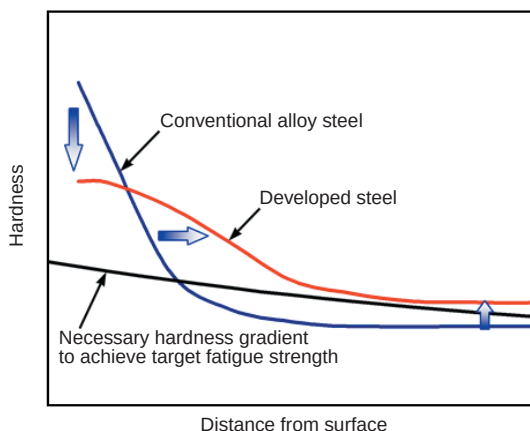


Fig. 1 Hardness transition curve design

surface hardness would be reduced in order to help guarantee straightenability. Given the need for increased nitrided depth and reduced near-surface hardness, the target for the practical depth of the nitrided case was set at 0.3 mm or more, and the target for near-surface hardness (hardness at a distance of 0.05 mm from the surface of the base material) was set at Hv500 or less. To help avoid a decline in machinability, the target for hardness following normalization was set at HRB100 or less.

3.2. Characteristics of Conventional Crankshaft Material and Study of Methods for Achievement of Increased Strength

Figure 2 shows the metallic microstructures of the steel used in Honda motorcycle crankshafts subject to standard loads (termed “conventional carbon steel” below) and in crankshafts from which high fatigue strength is demanded (“conventional alloy steel” below). Because the conventional carbon steel is not normalized following hot forging, its microstructure is coarse with a large crystal grain diameter. However, both materials display ferrite-pearlite microstructures, helping to assure a high level of machinability.

In the majority of cases, the hardness of the material used in large motorcycle crankshafts is increased by quenching and tempering, and the material’s fatigue strength is increased by soft nitriding or nitriding. However, homogeneous metallic microstructures such as a tempered martensite microstructure display insufficient cuttability in comparison to mixed microstructures such as ferrite-pearlite microstructures. In addition, because the speed of diffusion of N during nitriding is slower in martensite than in ferrite, it is necessary for nitriding to be conducted for a longer period in an attempt to increase the depth of the nitrided diffusion zone, resulting in a decline in productivity. Because of this, maintaining the ferrite-pearlite microstructure unchanged and strengthening the ferrite, which is the starting point for fatigue fractures, is an effective means of increasing fatigue strength without reducing machinability.

The addition of V and Ti for precipitation strengthening and the addition of P and Si for solid solution strengthening can be considered as methods for strengthening ferrite. The addition of Ti and P reduces the

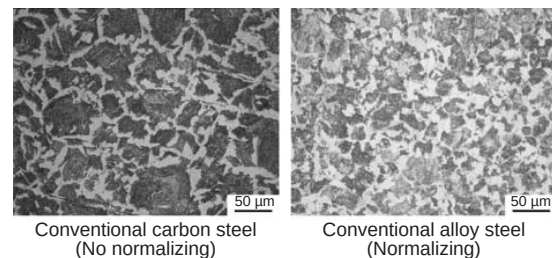


Fig. 2 Microstructure of conventional carbon steel and conventional alloy steel

impact value of the material, and is therefore undesirable in materials used in the crankshafts of large motorcycles, in some of which the primary drive gears are cut directly into the crankshaft. In addition, the fatigue strengthening effect of Si is low in comparison to the effect of V when added in identical amounts^{(1), (2)}.

Based on the above considerations, it was decided to attempt to increase fatigue strength in the developed material while minimizing any increase in hardness by adding V.

3.3. Study of Methods of Controlling Amount of V in Solid Solution

V-added steel is also known as microalloyed steel. Because identical characteristics to those displayed by quenched and tempered steels can be obtained despite the omission of the quenching and tempering process following hot forging, these steels are widely used for parts including crankshafts, connecting rods, and wheel hubs. Figure 3 shows the relationship between the heating temperature and the amount of V in solid solution, calculated using the total thermodynamics calculation system Thermo-Calc. The amount of V in solid solution increases as the heating temperature increases, but the additive amount in the case of microalloyed steels is generally 0.2% or less in order to achieve complete dissolution of the V at the heating temperature prior to hot forging (1100-1300°C).

Following hot forging, microalloyed steel crankshafts undergo controlled cooling using large blower fans, and their hardness reaches HRC20 and above⁽⁷⁾. The crankshafts are either machined at this level of hardness, or they are annealed in order to reduce their hardness. Given the challenge of finely controlling the cooling speed during controlled cooling under blower fans following hot forging, the development discussed here attempted to control hardness by controlling the amount of V in solid solution through adjustment of the normalizing temperature.

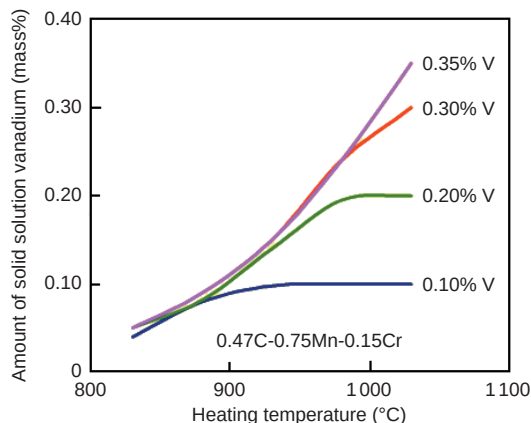


Fig. 3 Relationship between heating temperature and amount of dissolved vanadium

4. Alloy Design

4.1. Basic Composition

In order to increase wear resistance, timing sprockets and other parts that are cut directly into motorcycle crankshafts are subjected to high-frequency quenching, and quenching hardness values of HRC50 and above are obtained. To help assure stable high-frequency quenching hardness, a lower limit of 0.4% C was set. Al was not added because it bonds readily with N, markedly increasing near-surface hardness and impeding the diffusion of N during soft nitriding. S forms MnS in the steel, contributing to cuttability, but it can result in the occurrence of quenching cracks during high-frequency quenching. Because of this, the JASO S1 level of 0.06% was added. In order to promote the breaking of cutting chips, sulfite inclusions were finely dispersed in the steel, and by doing so, Pb was omitted as a free-cutting component. In addition, the amounts of C, Mn, Si, and V were adjusted.

C, Mn, Cr, and V were either added or removed in relation to the conventional alloy steel, and multiple alloys were produced in which the carbon equivalent, Ceq, was adjusted to achieve a hardness of HRB100 or below following normalization. Ceq was calculated using Eq. (1) below.

$$\text{Ceq} = \text{C} + 0.07\text{Si} + 0.16\text{Mn} + 0.19\text{Cu} + 0.17\text{Ni} + 0.2\text{Cr} + [\text{V}] \quad (1)$$

[V]: The amount of V that can exist in γ at the normalization heating temperature

Following normalization under identical conditions to the conventional alloy steel, these alloys were salt bath soft-nitrided, and the near-surface hardness and practical depth of the nitrided case were measured. Figure 4 shows the relationship between the amount of Cr and near-surface hardness. When the additive amount of Cr was greater than 0.3%, near-surface hardness following soft

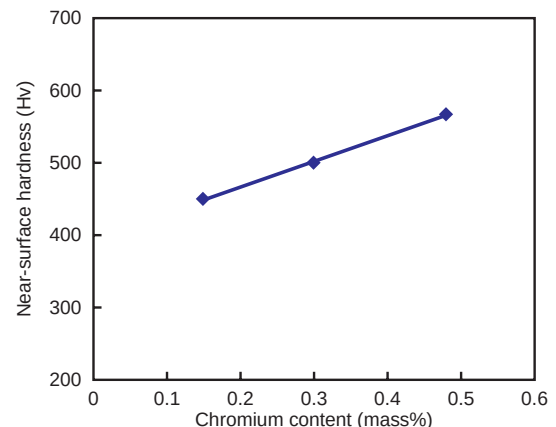


Fig. 4 Relationship between chromium content and near-surface hardness

Table 1 Chemical composition of conventional alloy steel and developed steel

Steel Grades	C	Mn	S	Cr	V	Al	Pb
Conventional alloy steel	0.23	1.00	0.055	1.00	-	0.016	0.08
Sample I	0.47	0.75	0.048	0.15	0.30	-	-
Sample II	0.45	0.74	0.049	0.30	0.29	-	-
Sample III	0.43	0.75	0.047	0.30	0.15	-	-
Developed steel *	0.47	0.75	0.060	0.15	0.30	-	-

* Based on Sample I

nitriding exceeded the target upper limit, and the additive amount was therefore set at 0.3% or below. Figure 5 shows the relationship between near-surface hardness and the practical depth of the nitrided case. The fact that no clear correlation existed between near-surface hardness and the practical depth of the nitrided case suggested that it would be possible to reduce near-surface hardness while increasing the nitrided depth by adjusting the additive amounts in the alloy.

4.2. Study of Fatigue Strength

The three steels that fell within the target area for near-surface hardness and practical depth of the nitrided case in Fig. 5 were subjected to an Ono-type rotating-bending fatigue test. The chemical compositions of the steels are shown in Table 1. The test specimens were shaped to

produce a stress distribution corresponding to the stress gradient on the R section of a crankshaft fillet (Fig. 6). Figure 7 shows the results of measurements of near-surface hardness and fatigue strength. Because the normalizing temperature of the test specimens was 900°C or below, identical amounts of V returned to solid solution in each when heated for normalization (Fig. 3). However, fatigue strength was lowest in sample III, which contained a lower additive amount of V than the other samples. It is conjectured that the crystal grains became coarse and fatigue strength declined in this sample because of the low amount of undissolved V when it was heated for normalization. Sample I, which displayed a good balance between near-surface hardness and fatigue strength, was selected as the new steel.

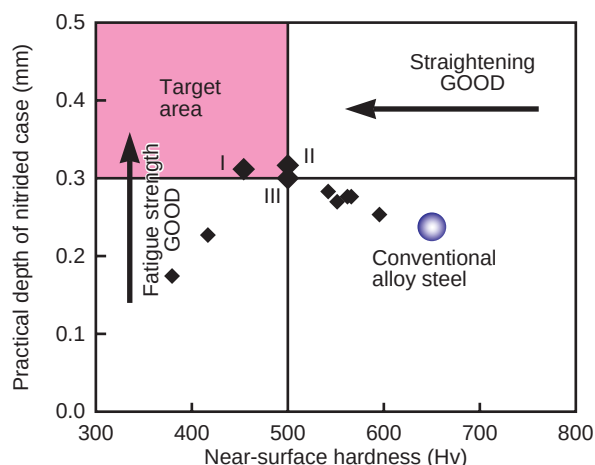


Fig. 5 Relationship between near-surface hardness and practical depth of nitride case

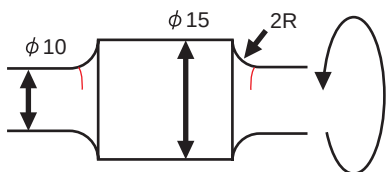


Fig. 6 Geometry of rotating-bending fatigue test specimen

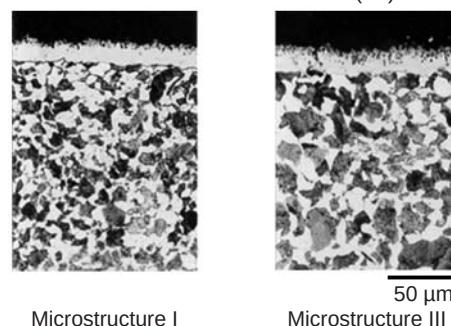
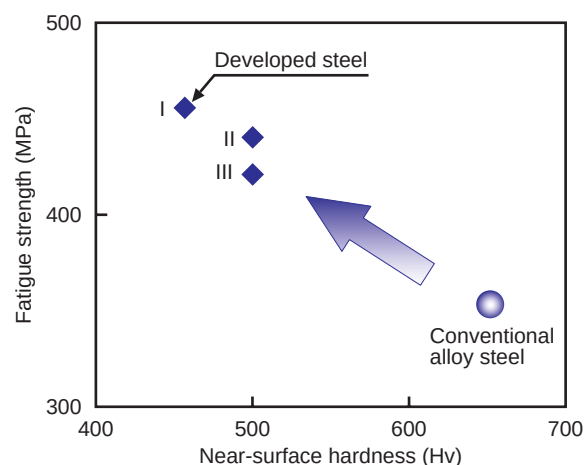


Fig. 7 Near-surface hardness and fatigue strength

5. Characteristics of Developed Steel

5.1. Normalization Characteristic

Because steels are normalized in order to restore the microstructure that has been disrupted by hot forging and to adjust hardness, it is necessary to use AC3 temperatures or higher. In the majority of cases, considering temperature variations and other factors, the work is conducted by heating the steel to AC3 temperature plus 50°C or more, followed by air cooling. The standard heating temperature range is around 50°C. The AC3 temperature of the developed steel was calculated as 770°C using Eq. (2) below, necessitating a heating temperature of 820°C or above.

$$AC3 = 854 - 180C + 44Si - 14Mn - 17.8Ni - 1.7Cr \quad (2)$$

In order to study the effect of the normalizing heating temperature, 40 mm x 40 mm test specimens were heated at temperatures of between 830 and 980°C, and their hardness was measured following air cooling. Figure 8 shows the relationship between the normalizing temperature and hardness. Hardness tends to increase with increased normalizing temperature. It is considered that this results from the V carbonitrides precipitated during cooling, following hot forging, returning to solid solution when the steel is heated for normalization; a greater amount of V carbonitrides return to solid solution with increased temperature, producing a greater effect of precipitation hardening during the following cooling process. This indicated that when a greater amount of V was added than the amount of V in solid solution calculated using Thermo-Calc, normalizing temperature control at temperatures lower than the temperature used in the calculations would make it possible to obtain any desired hardness level.

In order to study the effect of the cooling speed, hardness was measured after the steel was heated to temperatures of 820 and 880°C and cooled at speeds of between 0.1 and 2°C per second. Figure 9 shows the effect of cooling speed on hardness after normalizing.

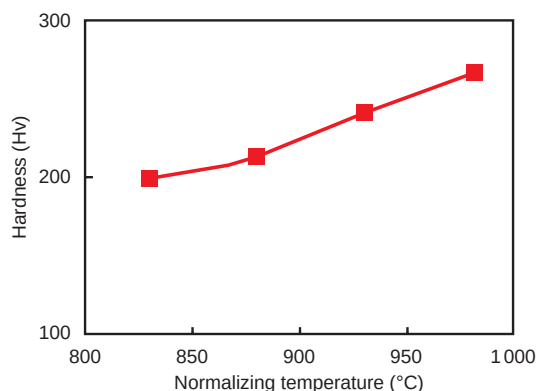


Fig. 8 Relationship between normalizing temperature and hardness

Hardness tends to be greater with faster cooling speeds. This is thought to be because at higher cooling speeds the V carbonitrides become finely dispersed in the cooling process and their ability to precipitation harden the steel increases, while at lower cooling speeds the V carbonitrides agglomerate, and their ability to precipitation harden the steel declines. In addition, the crystal grains become coarse at lower cooling speeds, and this may have a negative effect on fatigue strength and straightenability.

5.2. Fatigue Strength Characteristic

Ono-type rotating-bending fatigue tests were conducted on the test specimens shown in Fig. 6 in order to determine the effect of the normalizing temperature on the fatigue strength of the developed steel. 40 mm x 40 mm test specimens were normalized, machined, and salt bath nitrided. Figure 10 shows the relationship between the

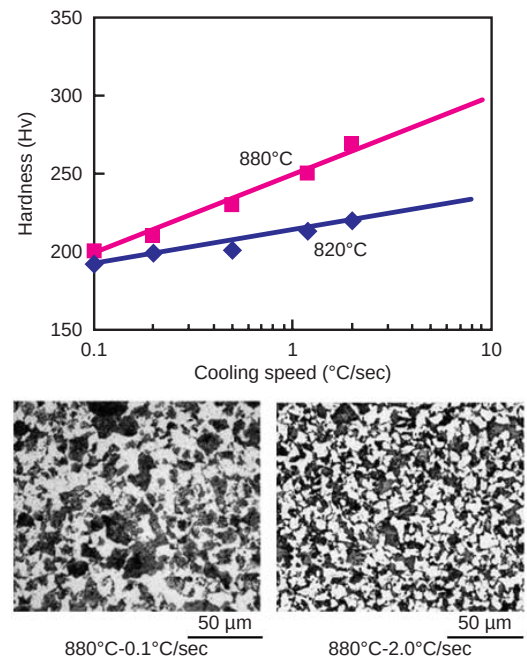


Fig. 9 Effect of cooling speed on hardness after normalizing

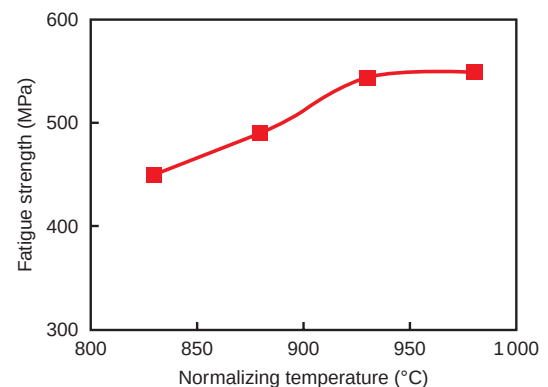


Fig. 10 Relationship between normalizing temperature and fatigue strength

normalizing temperature and fatigue strength. An increase in internal hardness is thought to be the reason that fatigue strength increases with increased normalizing temperature. However, fatigue strength at a normalizing temperature of 980°C is no higher than fatigue strength at a temperature of 930°C. Figure 11 shows the microstructures of the steels at normalizing temperatures of 930 and 980°C. It is conjectured that fatigue strength did not increase with increasing normalizing temperature due to a coarsening of the crystal grain. At temperatures of 930°C and below, it would be possible to control fatigue strength by means of controlling normalizing temperature.

5.3. Straightenability

3-point static bending tests were conducted using test specimens of the type shown in Fig. 6 in order to study the effect of the normalizing temperature on straightenability. With supports spaced 106 mm apart, the central sections of the test specimens were loaded until cracks appeared, and the change of runout was measured when the load was removed. Figure 12 shows the relationship between normalizing temperature and change of runout, and Fig. 13 shows the relationship between internal hardness and near-surface hardness. Internal hardness increases with increasing normalizing temperature, but there is no change in near-surface

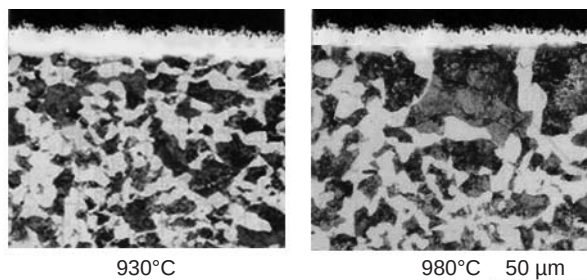


Fig. 11 Microstructure of developed steel at normalizing temperature of 930 °C and 980 °C

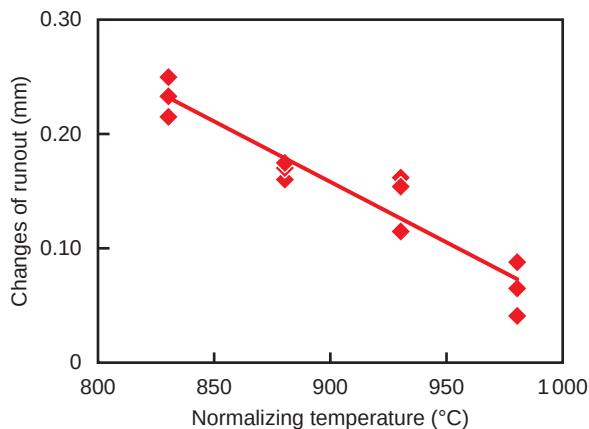


Fig. 12 Normalizing temperature and changes of runout

hardness, and therefore no change in the ductility of the surface, where stress is high. However, it is thought that straightenability declines with increasing normalizing temperature because the increase in hardness reduces the level of internal plastic deformation under bending.

6. Crankshaft Characteristics

6.1. Fatigue Strength Characteristic

Figure 14 shows the shape of the sample crankshaft. The normalizing temperature for the crankshaft was set at 820°C, but the machining processes were identical to those used for crankshafts manufactured from conventional alloy steel. Figure 15 shows hardness transition curves for the developed steel and the

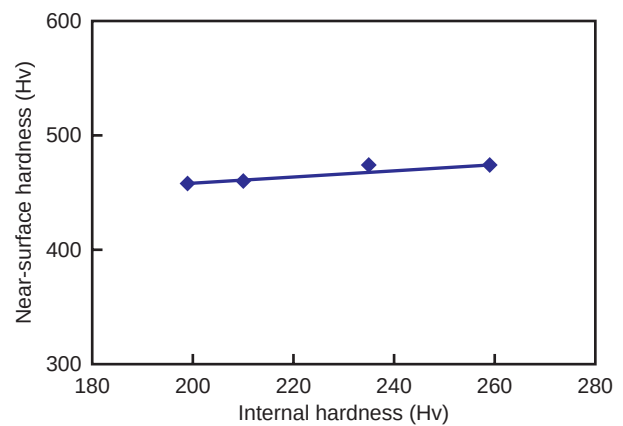


Fig. 13 Relationship between internal hardness and near-surface hardness

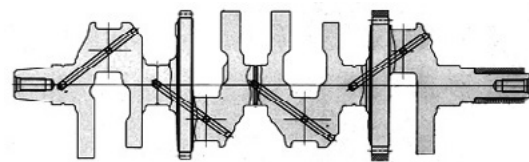


Fig. 14 Shape of sample crankshaft

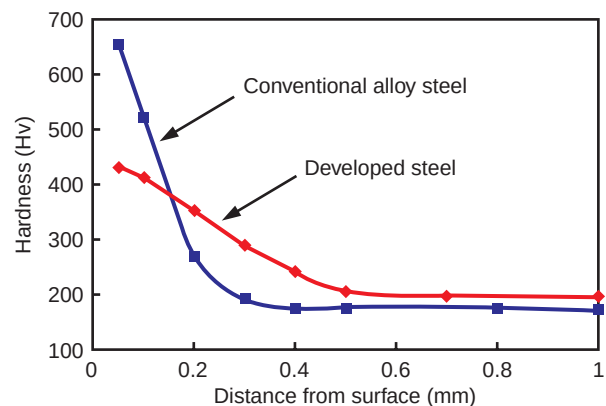


Fig. 15 Hardness transition curve of developed and conventional alloy steel

conventional alloy steel. The crankshaft manufactured from the developed steel displays greater internal hardness and depth of nitrided diffusion zone and lower near-surface hardness than the conventional alloy steel crankshaft. Next, the journals at the ends of the crankshafts were supported, and the central journals were subjected to reversed excitation. Strain gauges attached to the fillet sections of the center journals were used to measure the stresses generated (Fig. 16). Figure 17 shows the results of the bending fatigue tests. The developed material displayed a 30% increase in fatigue strength against the conventional alloy steel, even at a normalizing temperature of 820°C, at which fatigue strength is lowest.

6.2. Straightenability

The journals at both ends of a crankshaft of the shape shown in Fig. 14 were supported on V blocks, the center journal was loaded until cracks appeared, and the change of runout of the center journal was measured after the load was removed (Fig. 18). Figure 19 shows the relationship between the normalizing temperature and the change of runout. Straightenability declines with increasing normalizing temperature. Taking productivity into consideration, it would be desirable to set the normalizing temperature low. However, even when the normalizing temperature is set high, the fatigue strength of the

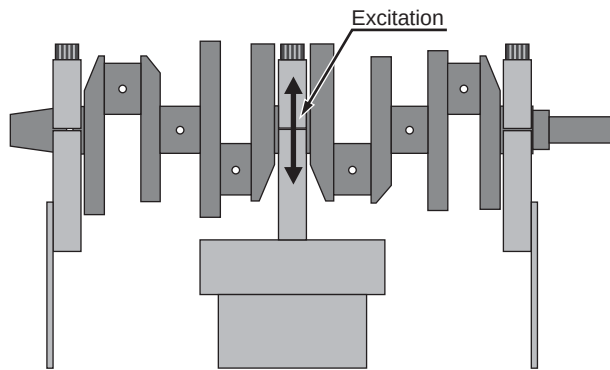


Fig. 16 Bending fatigue test method for crankshaft

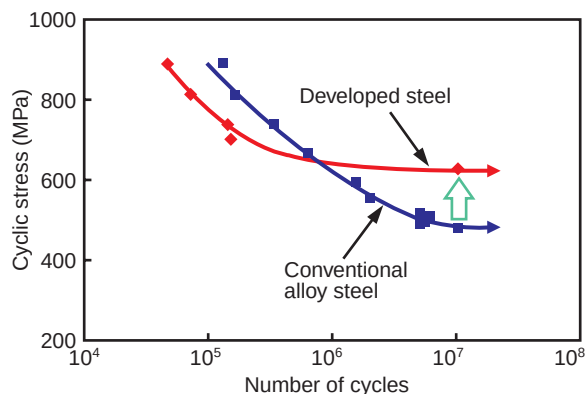


Fig. 17 Results of bending fatigue test for crankshaft

developed steel can be further increased, with no change in basic processes, through the achievement of increased accuracy in machining and the development of a method of controlling the amount of distortion following soft nitriding.

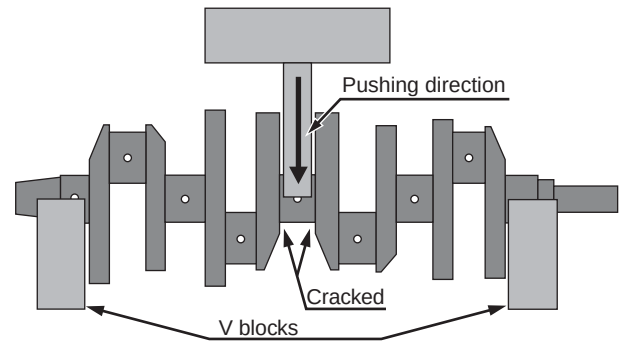


Fig. 18 Changes of runout test method for crankshaft

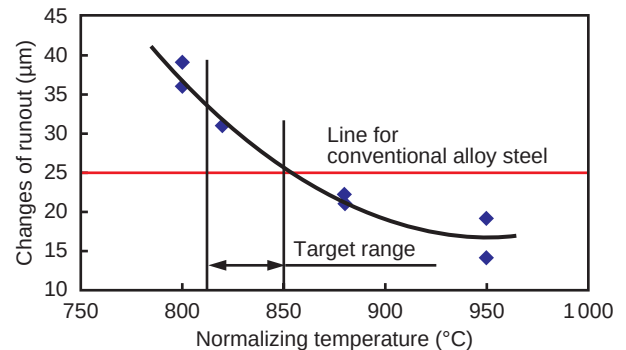


Fig. 19 Relationship between normalizing temperature and changes of runout

7. Effect of Application of Developed Steel

Crankshafts produced using the developed material was employed in the 2010 model VFR1200F (Fig. 20), which is fitted with a new geometry V4 engine. The use of the developed steel helped to enable the diameter of



Fig. 20 Mass production motorcycle VFR1200 using developed crankshaft



Fig. 21 Crankshaft with new geometry for new V-4 cylinder engine

the crank pins to be reduced by approximately 2 mm and the engine to be reduced in size and weight against the use of the conventional alloy steel (Fig. 21).

8. Conclusion

A crankshaft has been produced using a new material that achieves greater internal hardness and depth of the nitrided diffusion zone and reduced near-surface hardness against the conventional alloy steel. The following results were obtained in this development:

- (1) The addition of a greater amount of V than the amount of V in solid solution at the normalizing temperature made it possible to adjust hardness by controlling the normalizing temperature.
- (2) When the crystal grains became coarse due to the low amount of undissolved V while heated for normalization, a greater additive amount of V would result in a high level of fatigue strength.
- (3) Optimization of the normalizing temperature increased fatigue strength by more than 30% against the conventional alloy steel while maintaining machinability and straightenability.

The application of crankshafts produced using the new technology will enable the achievement of reduced size and weight of motorcycle engines, and contribute not only to higher power, but also to increased fuel efficiency, resulting in reduced CO₂ emissions.

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Development of High Fatigue Strength Valve Spring Using Control of White Layer by Nitriding

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ABSTRACT

Elemental addition to a material to reduce its non-dissolved carbide content while increasing its resistance to temper softening has helped to increase internal hardness after nitriding while maintaining its cold-coiling-ability. Conventionally, high strength has been achieved by applying nitriding as a surface treatment to the valve springs, without consideration of the compound layer. However, when the strength of wire coils, the hardness of the nitrogen-diffused layer, and internal hardness are maximally increased, the white layer becomes the points of origin of breakages, and fatigue strength does not increase. The white layer was therefore judged detrimental to fatigue strength, and a nitriding method was developed that attained nitriding without the white layer and the achievement of increased hardness in the nitrogen-diffused layer, which had previously been incompatible.

1. Introduction

Motorcycle valve trains generally use valve lifters or rocker arms to drive the valves, and speeds can exceed 15000 rpm in high-speed engines. As engine speed increases, valve springs experience a resonance phenomenon known as surging, as a result of which the stress on the springs exceeds design stress. For this reason, high-fatigue-strength valve springs are necessary in motorcycles, which are used frequently in the high-engine-speed range. The strengthening of valve springs enables reduced size of valve trains and contributes to weight reduction of the engine and vehicle body. In addition, as strength increases, the loads on the valve springs increase, and a decline in their reaction force (permanent set-in fatigue) occurs under the high-temperature environments generated during high-speed operation. This reduces the range of motion of the valves, and set-in fatigue resistance is therefore also an important characteristic.

The strengthening of materials and the advancement of surface treatment technologies play a significant role in the achievement of high strength in valve springs. The

tensile strength of a material is generally proportional to its hardness, and development projects were therefore conducted in which attempts were made to increase hardness^{(1),(2)}. However, the achievement of a high level of hardness can simultaneously reduce toughness, resulting in a decline in cold coilability. In addition, while a high level of hardness may be achieved, the internal hardness of the material declines when tempered in the nitriding surface treatment process, making it necessary to increase temper-softening resistance. Temper-softening resistance also has a significant effect on the set-in fatigue characteristic, an important factor of the valve spring characteristic. However, elements that increase temper-softening resistance also reduce cold coilability due to the formation of carbides. 0.4% V is added to SWOCN-V, a high-strength material that has been put into practical use, helping to enable good temper-softening resistance to be obtained. However, an increase in the amount of undissolved carbides in the material due to the addition of the V reduces its margin of cold coilability. Nitriding is used as a surface treatment technology for valve springs. Increasing the nitriding temperature promotes the

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diffusion of nitrogen, resulting in increased surface hardness. However, the nitriding conditions affect the material's internal hardness and the thickness of the white layer. Because these changes affect the fatigue strength of the material, the selection of nitriding conditions is important.

This paper will discuss the development of high-strength valve springs, with a focus on alloy composition and nitriding conditions.

2. Material Composition Design

2.1. Aims for Alloy Composition

The development was conducted with consideration of the following points as the design concept for the material:

- (1) Development of a material that balances high strength with cold coilability, and also displays good nitridability;
- (2) Achievement of high level of temper-softening resistance, in order to help prevent a decline in internal hardness under nitriding temperatures; and
- (3) Development of a material that can be produced using existing facilities, with no special treatments, in order to control cost increases.

2.2. Study of Alloy Composition

A study was conducted to determine an alloy composition that balanced high strength with cold coilability. For high-strength valve spring steels, V is important not only in increasing surface hardness during nitriding, but also in helping to maintain internal hardness following heating during the nitriding process. V is generally effective as an additive to heat-treated products, like nitrided spring steel, due to its precipitation hardening effect at temperatures of 400-700°C. However, numerous points must be taken into consideration when using V in industrial applications. If the V added to the material does not dissolve but forms coarse undissolved carbides, not only is the element's precipitation hardening effect reduced, but the material can also be negatively affected, as a result of the reduction of coilability by the coarse undissolved carbides themselves. It is therefore important to employ an effective solution of V or an ultra-fine form of V in order to control the formation of undissolved carbides and help prevent negative impacts on the material, in addition to obtaining a precipitation hardening effect and achieving temper-softening resistance during the nitriding treatment applied in the valve spring manufacturing process.

This can also be predicted from the fact, as shown in Fig. 1, that the effect of V becomes saturated with higher additive amounts. This result was used to find the maximum additive amount of V that could be effectively used in the thermal history of the manufacturing process, from rolling to production of the valve spring, to help ensure that a practically effective rather than an excessive amount could be added.

Other elements were similarly studied from the perspectives of strength and carbide formation. Cr was judged unsuitable for short heat treatments such as high-frequency treatments, because Cr carbides would not readily dissolve.

Si is known as an element that both has a positive effect on the set-in fatigue characteristic of valves and affects carbide formation behavior, and the optimum additive amounts of this element and Cr were therefore studied from the perspectives of strength and carbide formation.

Figures 2(a) and 2(b) show the results of these studies. The amount of undissolved carbides present in the material during coiling and its internal hardness following nitriding were evaluated with actually used materials as the benchmarks. The amount of undissolved carbides was calculated using Thermo-Calc, while the internal hardness of the material was judged using the hardness of previous steels in a hardness database as the criterion. The upper section of each box in Fig. 2 represents hardness, while the lower section represents carbide precipitation behavior. Better performance than previous examples is marked with an A, a decline in performance with a C, and a level of performance close to the critical value with a B. The results indicate that a basic composition of 0.73%C-2.2%Si-1.0%Cr-0.1%V is desirable from the perspectives of carbide behavior following quenching and tempering (which affects coilability and strength following nitriding) and internal hardness following nitriding.

The addition of Mo, which impedes the formation of undissolved carbides and assists in precipitation strengthening, can also be considered to be effective in helping to achieve a high level of strength. However, the addition of excess Mo affects quenchability, and results in production defects such as cracks due to increased hardness as a result of the fact that it promotes the production of super-cooled microstructures such as martensite and bainite during the various heat treatments from rolling to production of the spring. A maximum additive amount for Mo that would not result in the production of super-cooled microstructures in the

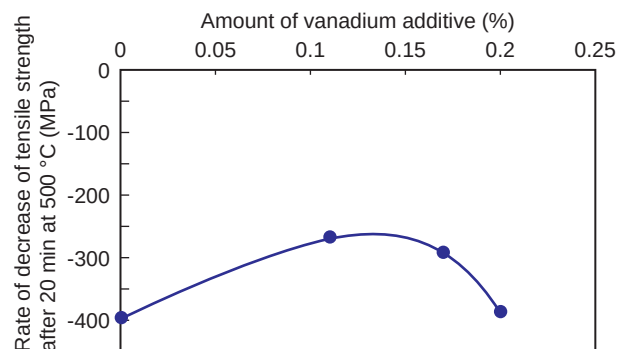


Fig. 1 Relation between vanadium additive amount and temper softening characteristic

projected thermal history of the material was therefore also established.

As in the case of Mo, the addition of W, which dissolves easily under heating during quenching, was also studied. While the precipitation hardening effect of W is not as strong as that of Mo, its effect on quenchability is low, and its addition is not prone to result in production issues due to the formation of super-cooled microstructures. W was therefore added to help ensure increased internal hardness following nitriding.

In this way, the elements related to the achievement of the necessary performance and manufacturability for the springs were comprehensively studied, and the

Table 1 Composition of developed steel

	(mass%)						
	C	Si	Mn	Cr	Mo	V	W
Developed steel	0.75	2.16	0.71	0.98	0.13	0.10	0.15
SWOCN-V (Previous)	0.73	1.67	0.58	0.56	0.15	0.41	—

appropriateness of the composition of the steel was verified in tests. Table 1 shows the composition of the steel, and Fig. 3 shows the material's hardness (temper-softening resistance) and notch-bending angle (coilability) after 3 hr at 500°C. The results indicated that the composition of the developed steel, with Mo and W added to the basic composition, balanced strength with workability.

2.3. Characteristics of Developed Alloy

Figures 4 and 5 respectively show the results of self-diameter winding tests and notch-bend tests on oil-tempered wires produced from wire dissolved in an actual oven. These results provide indices of cold coilability, and

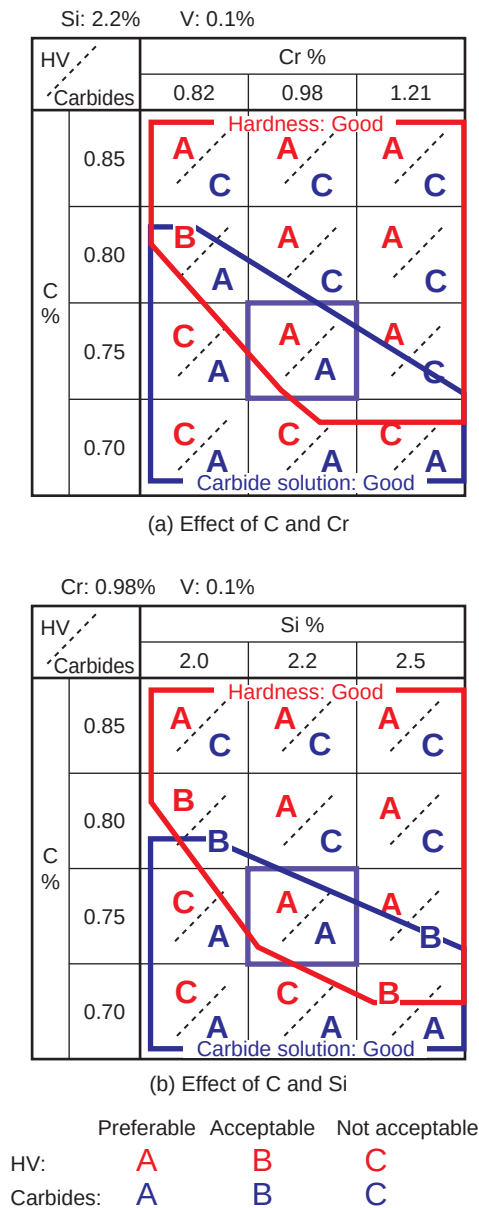


Fig. 2 Example of predicted performance of wires in terms of carbide state and hardness after heat treatments

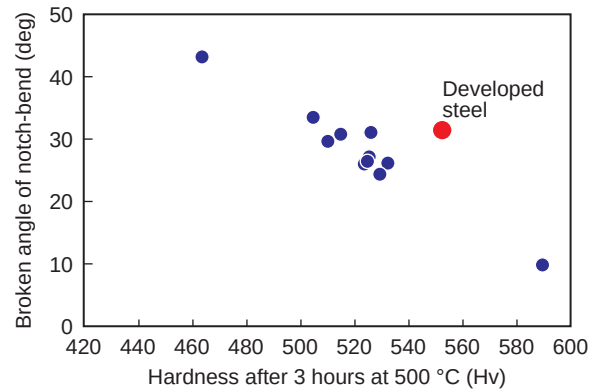


Fig. 3 Results of material composition study

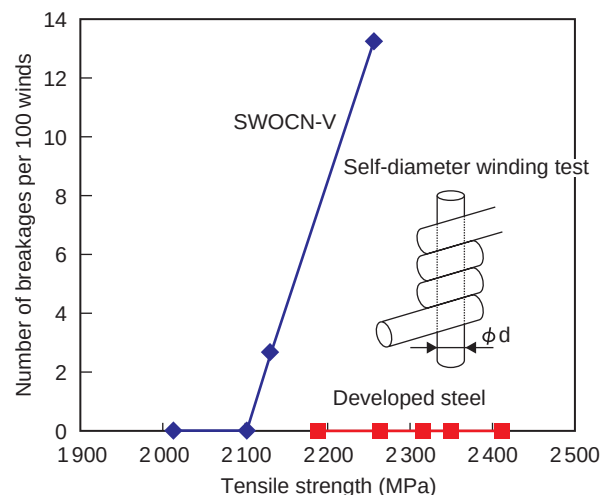


Fig. 4 Result of self-diameter winding test

the developed steel clearly demonstrated better performance than SWOCN-V in terms of coilability against strength. This is thought to be a result of the reduction in the amount of undissolved carbides in the material due to the reduction in V, as shown in Fig. 6.

Figure 7 shows the low-temperature tempering characteristic of the material. Even on the high-strength side, the hardness of the oil-tempered wire was approximately the same at 2250 MPa or more when tempered at a temperature corresponding to the nitriding temperature, and the tensile strength of the oil-tempered wire could therefore be set at 2250 MPa. Figure 8 shows the low-temperature annealing characteristic of the oil-

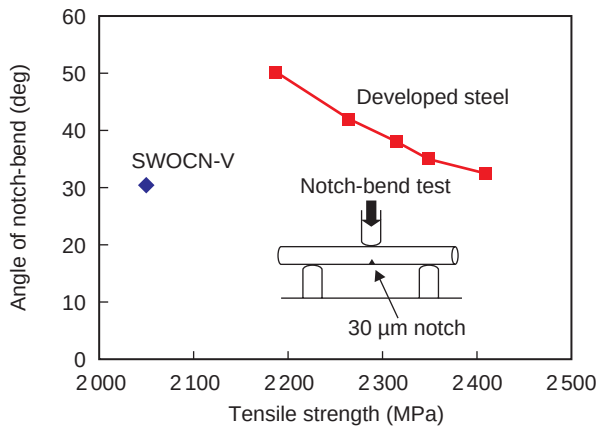


Fig. 5 Result of notch-bend test

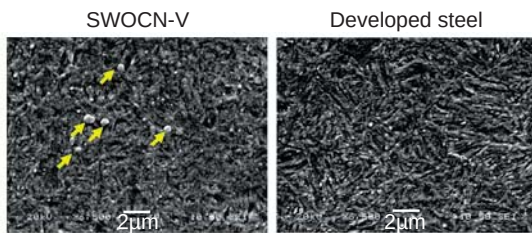


Fig. 6 Identification of non-dissolved carbide

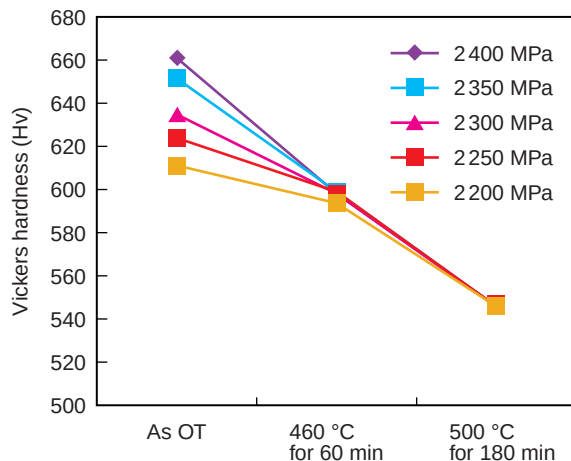


Fig. 7 Results of low-temperature tempering

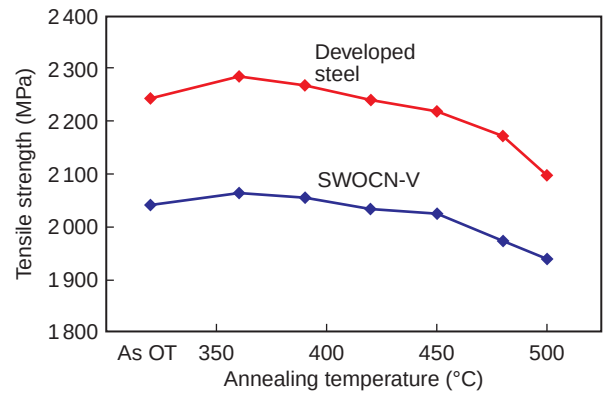


Fig. 8 Results of low-temperature annealing

tempered wire. Tensile strength is increased against SWOCN-V by approximately 200 MPa across virtually the entire temperature range. Existing facilities could also be used for the developed steel from dissolution to oil tempering.

3. Study of Nitriding Treatment for High-strength Valve Springs

3.1. Nitriding Quality Aims

With regard to nitriding conditions, in general when the nitriding temperature is increased the hardness of the nitrogen diffusion layer increases but internal hardness decreases, while the opposite occurs when the nitriding temperature is decreased. The sectional stress on the valve springs is highest on the surface of the internal diameter, and to achieve a high level of fatigue strength it is important to achieve high hardness by forming a high N-concentration nitrogen diffusion layer as the surface layer. However, because the formation of this type of nitrogen diffusion layer using normal gas nitriding necessitates the decomposition of large quantities of NH_3 gas in order to produce large quantities of activated H, a high quantity of N is produced at the same time. Therefore, a high-N concentration white layer is formed on the outermost surface layers of steel materials to which nitriding treatment has been applied. The toughness of this white layer is low, and an excessive white layer would result in reduced fatigue strength⁽³⁾. The aim of the development discussed here was to determine an optimum nitriding quality for fatigue strength by controlling the nitrogen diffusion layer hardness and the white layer⁽⁴⁾.

3.2. Optimization of Nitriding Conditions

Whether or not a white layer is formed on the outermost surface layer of the steel material is determined by the amount of N that penetrates the surface layer of the material; i.e., the balance between the amount of NH_3 that is decomposed and the amount of N that diffuses into the material. In order to control the formation of the white

layer, it is necessary to control the rate of decomposition of the NH_3 gas to attempt to ensure that the concentration of N at the outermost surface of the material, where the concentration is highest, remains below approximately 5 wt%.

The rate of decomposition of the NH_3 gas and the speed of diffusion of N into the steel material are both affected by the temperature. In addition, because the speed of diffusion of N in the steel material is also affected by the composition of the material, it is important to adjust the atmosphere in order to produce an NH_3 decomposition rate suited to different steel materials in order to help prevent the formation of a white layer on the outermost surface layer.

However, by merely adjusting a gas composition and nitriding temperature that help to prevent the formation of a white layer, the oxide film that exists on the surface of the valve spring is insufficiently reduced, and nitriding becomes unstable. The authors therefore decided to conduct the nitriding after replacing the oxide film on the surface of the valve spring with a fluoride prior to the process. Because this fluoridization process would be conducted at a temperature lower than that for low-temperature tempering following cold coiling, there was no concern regarding a decline in hardness. The fluoridized steel surface would be more active in the nitriding atmosphere than a surface covered with an oxide film, and nitriding would therefore be able to proceed even at a comparatively low NH_3 potential without forming a white layer on the outermost surface layer.

In attempting to determine a nitriding treatment applicable to the developed material, it was decided that the nitriding temperature and time should be decided first. Tests were conducted in which valve springs were nitrided at varying temperatures and for varying period of time in a small nitriding oven following replacement of the oxide film with a fluoride. The internal hardness and depth of the nitride diffusion zone were then measured. During the tests, nitriding was conducted with the ratio of NH_3 and H_2 held constant. The results are shown in Fig. 9. The target for internal hardness was set at Hv610 or above, and the nitriding period was set at two hr in consideration of mass producibility. Under these conditions, 470°C was determined to be the optimum nitriding temperature. Figure 10 shows the results for the formation of a white layer on the developed material at different temperatures; if a white layer was formed, the symbol ▲ is used, and if no white layer was formed, the symbol ● is used. The results show that when the temperature is fixed, the formation or non-formation of a white layer depends on the ratio of NH_3 and H_2 in the nitriding atmosphere. Therefore, conducting nitriding at a ratio in the zone where no white layer was formed or close to the boundary line would make it possible to form a high N-concentration nitrogen diffusion layer on the surface of the material with no white layer.

3.3. Study of Effect of White Layer on Fatigue Strength

Two valve spring samples nitrided with different ratios of NH_3 and H_2 were produced in order to clarify the effect of the white layer. The SEM photographs in Fig. 11 show the thickness of the white layer for each sample. Sample A displays a 2 μm -thick white layer, while no white layer can be observed on Sample B. The fatigue strength of these samples was studied using a spring fatigue tester. Figure 12 shows an S-N diagram of the results. The symbol ■ represents a sample in which a crack originated internally, and the symbols □ and ○ represent samples in which cracks originated on the surface. For Sample A

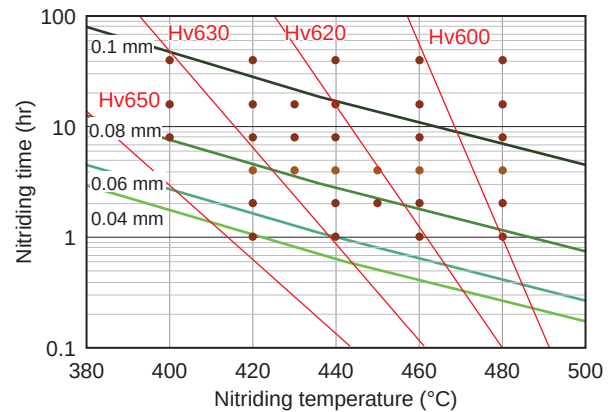


Fig. 9 Relation among temperature, time depth of nitriding, and internal hardness

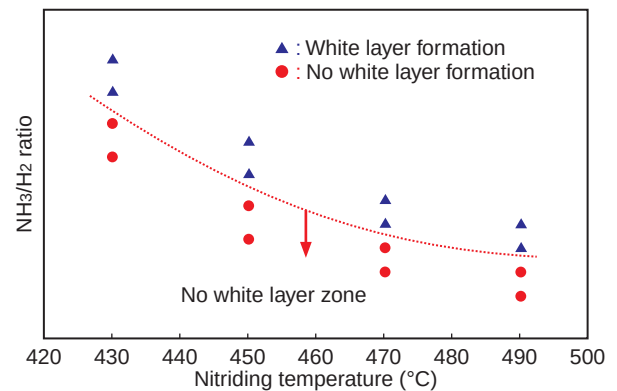


Fig. 10 Relation among gas composition, nitriding temperature, and white layer

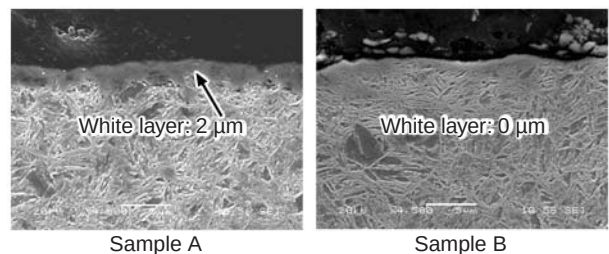


Fig. 11 SEM image of fatigue test specimens

test pieces, which displayed a thick white layer, all cracks originated on the surface. By contrast, for Sample B test pieces, which displayed no white layer, numerous cracks originated internally. Fatigue strength was higher in Sample B test pieces, in which cracks originated both internally and on the surface, than in Sample A test pieces, in which cracks originated exclusively on the surface. This phenomenon is explained in Fig. 13, which shows the stress distribution close to the outermost surface layer of the valve spring. It is conjectured that a high level of hardness due to nitrogen diffusion increases fatigue strength close to the surface of the internal diameter of the valve spring, where stress is highest, but because the white layer (ϵ -Fe₃N) is formed from hexagonal close-packed crystals it is hard but brittle⁽⁵⁾, and when it exists on the outermost surface layer, fatigue cracks spread from

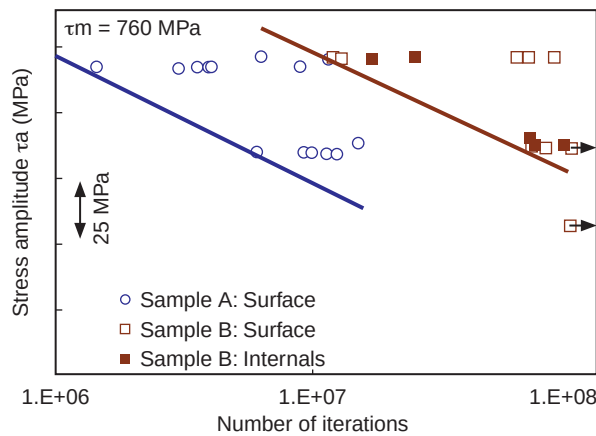


Fig. 12 Fatigue strength effect due to white layer thickness

minute notches. It is thought that in the SWOCN-V, by contrast, while white layers of 2 μm in thickness were present, because the stress on the outermost surface layer was low in the low-load range, a decline in the strength of the white layer did not become noticeable.

4. Characteristics of High-strength Valve Springs

4.1. Fatigue Characteristic

Figure 14 shows the results of spring fatigue tests on the developed high-strength valve spring material and SWOCN-V. The fatigue tests were initially conducted at $\tau_m=686$ MPa, but no cracks were observed in the developed material, and they were therefore conducted again at $\tau_m=760$ MPa. Because of this, the S-N data for SWOCN-V used as a comparison in Fig. 14 has been reduced by the difference in τ_m .

The developed material clearly performed better than SWOCN-V, displaying a 20% increase in fatigue strength. This is considered to be due to the fact that the hardness curve of the developed material used in the fatigue test valve springs is higher across the entire range than that of the SWOCN-V (Fig. 15), and because the lack of a white layer helps to prevent a decline in surface fatigue strength.

4.2. Set-in fatigue

When valve springs are used in high-load engine environments, high-speed operation exposes them to high temperatures, and declines in their reaction force can occur. This can result in a decline in valve motion, and may lead to the breakdown of the engine. It is therefore

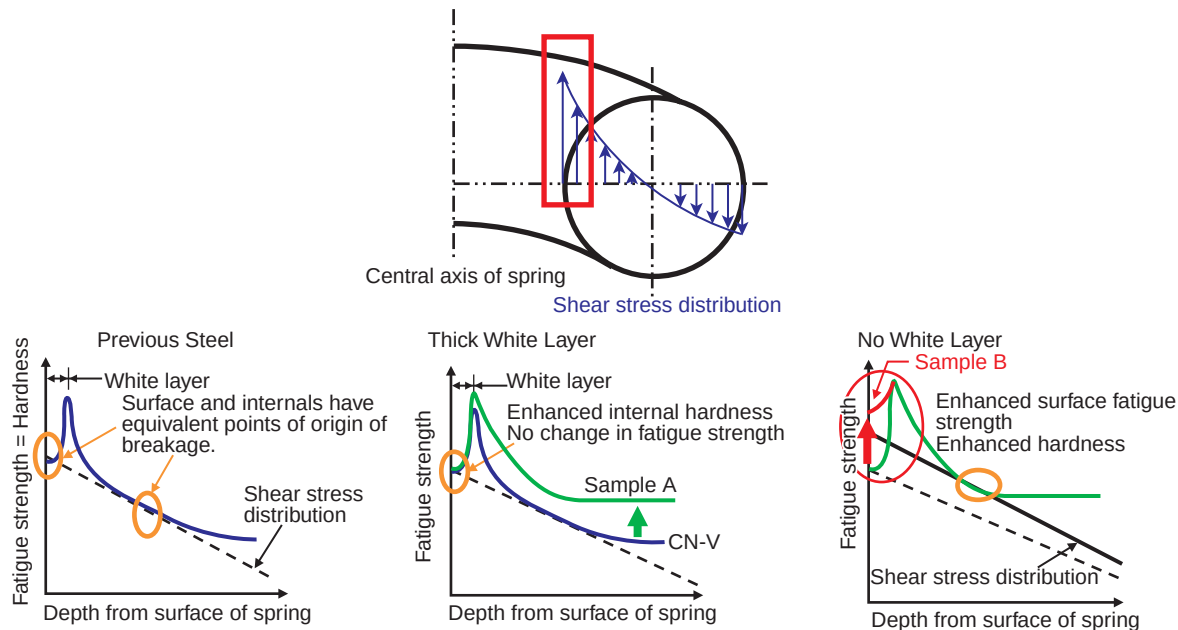


Fig. 13 Schematic diagram of influence of white layer thickness on fatigue strength

necessary to increase set-in fatigue resistance in response to increases in fatigue strength. The set-in fatigue characteristic of the springs was evaluated by measuring the degree of reduction in their reaction force following 100 hr at 150°C at varying constant stresses. Figure 16 shows the results of these measurements. The developed material displays good set-in fatigue resistance, with a

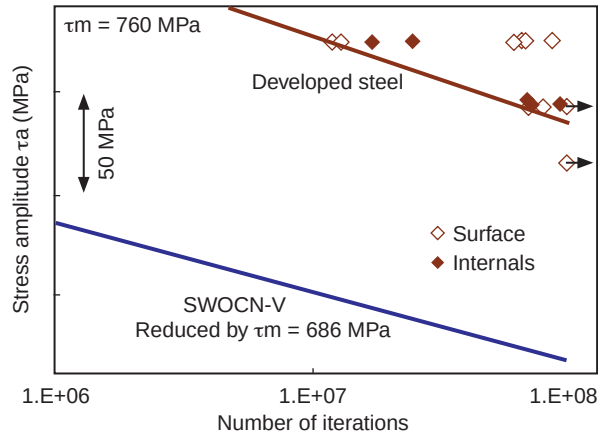


Fig. 14 Fatigue characteristic of developed steel

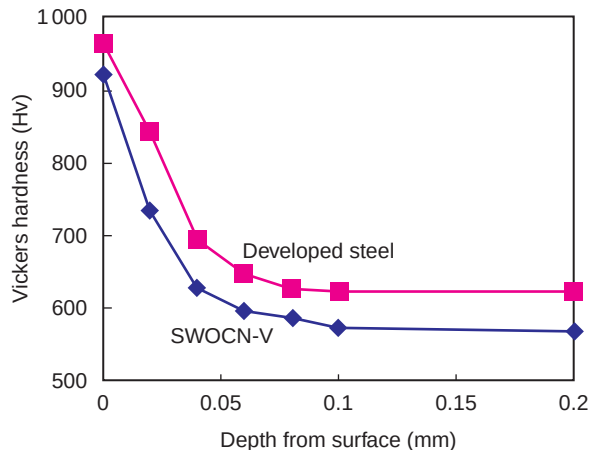


Fig. 15 Results for hardness after nitriding

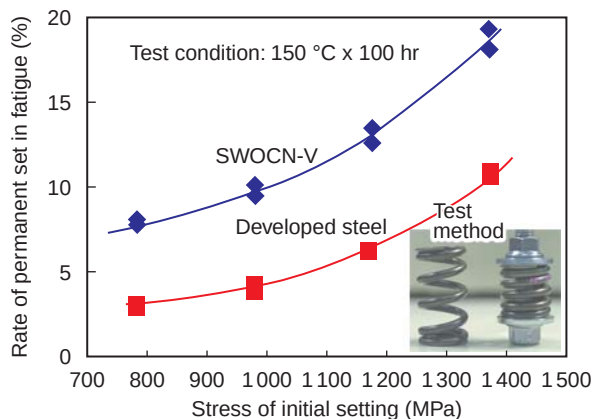


Fig. 16 Permanent set in fatigue characteristic of developed steel

reduction in set-in fatigue of approximately 25% against SWOCN-V. Strengthening of the martensite matrix by Si and Mo and the precipitation of microscopic carbides of Cr, V, and W are thought to be factors in this result.

5. Practical Application of High-strength Valve Spring

The use of the developed high-strength valve spring in the 2006 MotoGP machine helped to enable the height of the cylinder heads to be reduced, maximum engine speed to be increased, and the degree of cam lift to be increased. Riding this machine, Nicky Hayden took the MotoGP world championship in 2006.

Using the know-how gained from MotoGP, the developed valve springs were applied in mass-production for the first time to a motocross racing bike, the CRF250R (Fig. 17). The use of the developed high-strength valve springs helped to enable increased engine speed to be balanced with high durability, resulting in an increase in engine power in the high-speed range. The potential for the application of the developed valve springs to further mass-production models is being examined.



Fig. 17 CRF250R 2010 model using developed valve springs

6. Conclusion

- (1) Using SWOCN-V as a base, a composition that would balance high strength and high toughness was studied, resulting in the selection of 0.73C-2.2Si-1.0Cr-0.1Mo-0.1V-0.15W for the developed steel.
- (2) The necessary hardness following nitriding was achieved while also satisfying demands for cold coilability of oil-tempered wire.
- (3) The effect of the white layer on fatigue strength was clarified, and the necessity for nitriding conditions that would not produce a white layer in order to achieve high fatigue strength was ascertained. Nitriding without the production of a white layer was realized by using an activation reaction produced by fluoridization.
- (4) The high-strength valve springs produced using the developed material display 20% higher fatigue strength and 25% higher set-in fatigue resistance than SWOCN-V.

Afterword

The research conducted by the authors is aimed not only at achieving increased power from high-performance engines, but also, by contributing to the achievement of increased fuel efficiency by reducing the size and weight of automotive engines, to create products that will appeal to Honda's customers.

Acknowledgment

The authors would like to take this opportunity to express their gratitude to all the staff of NHK Spring Co., Ltd., who offered their generous assistance in this project.

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Development of Double-reflection Mirror System for Front-side Blind Spots

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ABSTRACT

A mirror system was developed that enables viewing of front-side blind spots. This mirror system uses a double-reflection optics system that eliminates image inversion and produces intuitive images that help prevent misconception. The effects when mounted on a vehicle were verified, and the mirror system was confirmed to enhance abilities to pull up to curbs and drive through narrow passages.

1. Introduction

Vehicles structurally have spots that cannot be directly viewed from the driver's seat, known as "blind spots." In particular, there are blind spots near vehicles, and many users feel some unease concerning these blind spots. This is especially true for users who are not used to driving, many of whom feel that they are not good at pulling up to side objects, parking, or performing other more complicated steering maneuvers. Recent advances in camera technology have enabled practical application of technologies that display images of the areas around vehicles on LCD screens such as those used in navigation systems, and have been favorably received.

This paper introduces a method that enables users to intuitively view front-side blind spots that could not adequately be viewed before, using a combination of a flat mirror and a convex mirror. Intuitive here refers to the ability to view properly aligned images without the vertical or horizontal inversion characteristic of mirrors.

2. Double-reflection Optics System Configuration

Mirrors are practically applied in various formats to view blind spots around vehicles. In particular, trucks, buses and other large vehicles have blind spots, and multiple mirrors for viewing these blind spots are attached (Fig. 1).

However, these are single mirrors, so images are always inverted vertically or horizontally, and experience is needed for accurate recognition. A combination of two

mirrors such as in a periscope (Fig. 2) can be used to obtain properly aligned images without inversion. This study used the principle of a periscope with the aim of viewing properly aligned images of front-side blind spots that could not adequately be viewed before.



Fig. 1 Example of mirrors for blind spots

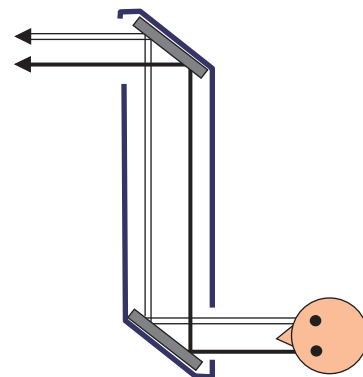


Fig. 2 Principle of periscope

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2.1. Target Blind Spots

One scene that users who are not good at driving feel uneasy about is passing oncoming vehicles on narrow passages. This is because the blind spots that extend forward from the front wheel fenders (front-side blind spots) make it necessary for each driver to judge the distance between their vehicle and the curb, gutter or other object using only their vehicle sense⁽¹⁾. In addition, due to design trends and the desire to provide an exhilarating field of view, few recent vehicles enable the driver to see the hood, making it even more difficult to sense and discern areas near the vehicle. Therefore, the target was to enhance visibility of the front-side blind spots (Fig. 3), which can be expected to increase convenience in practical situations, such as when pulling up to a curb or a toll booth, or driving through narrow passages.

2.2. Optics Design

Figure 4 shows a schematic diagram of the optics system that enables viewing of the front-side blind spots. This optics system can present a properly aligned image using the double-reflection of two mirrors installed inside and outside of the cabin.

The mirrors inside the vehicle (inner mirrors right, left) are flat mirrors, and the mirrors outside the vehicle (outer mirrors right, left) are convex mirrors. Table 1 shows the parameters of each mirror.

2.3. Image Appearance and Visible Area

Cameras installed on the door mirror housing as shown

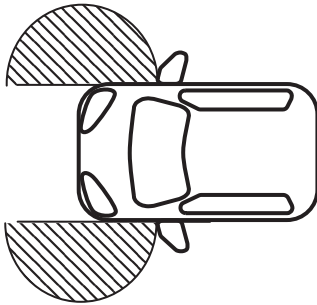


Fig. 3 Target field of view

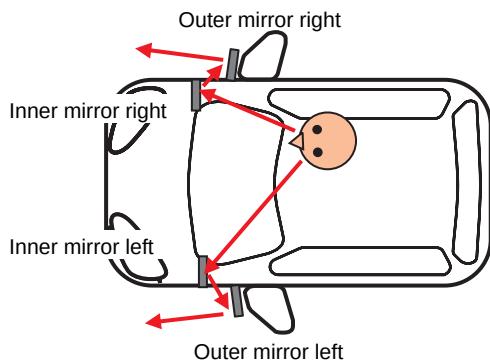


Fig. 4 Mirror layout

Table 1 Mirror parameters

Spec.	Right	Left
Eye point-inner mirror distance	924 mm	1262 mm
Inner mirror-outer mirror distance	175 mm	175 mm
Inner mirror area	1945 mm ²	3148 mm ²
Outer mirror area	5277 mm ²	6674 mm ²
Inner mirror curvature	Flat	Flat
Outer mirror curvature	R100 mm	R120 mm
Inner mirror angle	24.8 deg	48.7 deg

in Fig. 5(a) that enable viewing of the front-side blind spots have already been practically applied. The image displayed by these cameras is a properly aligned image of the area near the front wheel as viewed directly from near the door mirror as shown in Fig. 5(b), so it is easy to recognize.

The position of the flat mirror and the curvature of the curved mirror in this mirror system were investigated to provide a forward birds-eye view centering on the area near the front wheel, similar to the above noted camera images. A small mirror size is better for reducing the design and structural limitations. Therefore, this mirror system was designed to effectively utilize a small mirror area by showing a larger image of blind spots the driver wishes to see in the sideways direction, alongside the vehicle, and a smaller image that displays a wide range in the forward direction [Fig. 6(a)]. Fig. 6(b) shows an example of how the image appears generated using mirror optics simulation. As shown in Fig. 6(c), the visible area extends forward from the rear edge of the front wheel to the horizon, and to the side for approximately 1 m from the side of the body. This visible area can be changed by changing the mounting position and curvature of each mirror.

2.4. Mirror Layout

The Life (2004 year model) was selected as the base vehicle, and this system was installed.

The inner mirrors (inside flat mirrors) were installed at the base of the A pillar garnish near the front edge of the triangular window so as not to obstruct the direct field of view [Fig. 7(a)].



(a) Camera on mirror housing



(b) View of front-side

Fig. 5 Front-side view camera system

In addition, the outer mirrors (outside convex mirrors) were fit smoothly to the surface of the door mirror housing to avoid creating any new outside protrusions and realize an integrated design [Fig. 7(b)].

2.5. Exterior Design

Figure 8 shows the exterior design of a vehicle equipped with those mirror systems. As long as the vehicle has a triangular window like that of the Life used as the base vehicle, each mirror can be installed without greatly affecting the interior or body design.

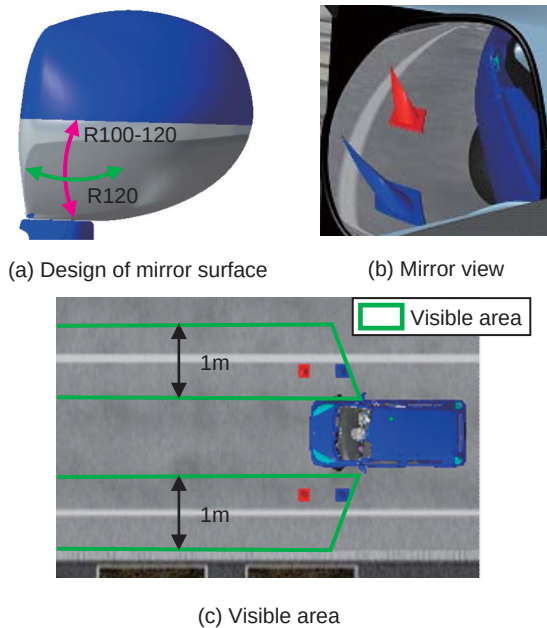


Fig. 6 Result of mirror view simulation

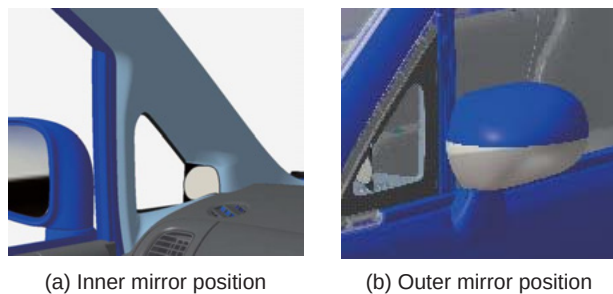


Fig. 7 Layout of each mirror



Fig. 8 Exterior view of test vehicle

3. Verification of Effects

3.1. Setting Verification Items

Pulling up to a curb and driving through narrow passages, during which a sense of where the front of the vehicle is needed, were selected as the verification items. Figure 9 shows the courses used for verification. The scene for pulling up to a curb imagined parallel parking, and care was taken to verify ease of use in conformance with reality. Driving through narrow passages imagined passing between two pylons that cannot be seen from the driver's seat in a short distance, and the ease of mind during passage was evaluated. Verification was performed after explaining the test contents to subjects solicited in-house, and obtaining their agreement.

3.2. Test Results

Figure 10 shows the change in ability to pull up to a curb for the cases with and without the mirror system. While there was no large difference in distance from the curb to the vehicle for subjects who are used to driving,

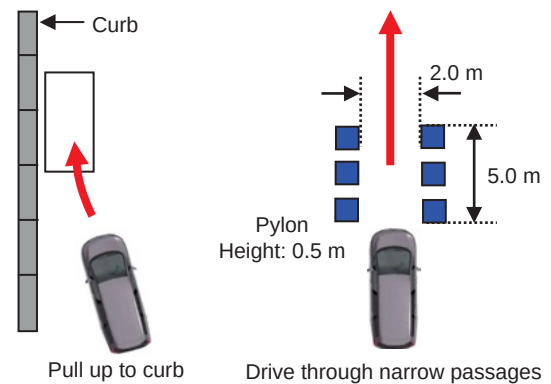


Fig. 9 Course for experiments

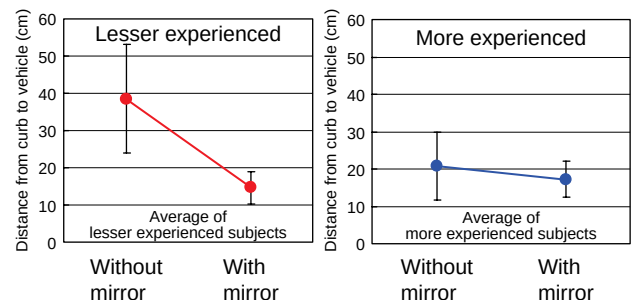


Fig. 10 Result of experiment 1

a noticeable difference was seen for subjects who were not used to driving, which confirmed that the initially targeted performance was obtained.

Next, Fig. 11 shows the sensory evaluation results for ease of mind when driving through narrow passages with and without the mirror system. Here, the responses from lesser experienced subjects in particular indicated a noticeable increase in ease of mind. Enhanced ease of mind was also confirmed for more experienced subjects due to being able to see the target objects.

Following the end of the tests, each subject's impression was obtained using a free description format. Many affirmative opinions were received, such as the following:

- It was intuitively easy to understand.
- The ability to see where one is going provides ease of mind.
- It was good to be able to understand the results of pulling up to a curb, which drivers normally perform rather casually.

In addition, issues for future enhancement were also elucidated such as the following:

- It would be better if the images were a little larger.
- Looking back and forth between the right and left mirrors is troublesome.

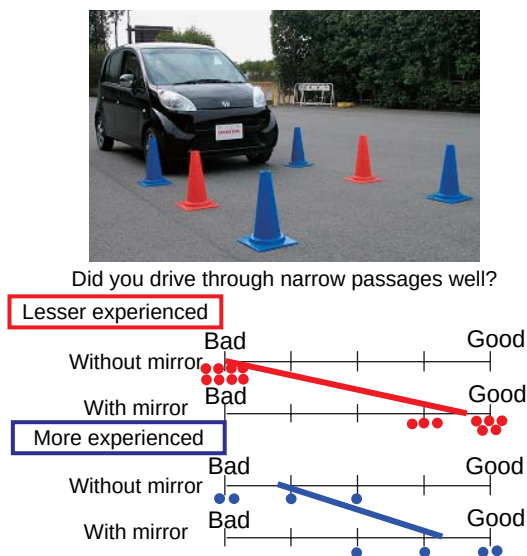


Fig. 11 Result of experiment 2

4. Conclusion

A mirror system that enables viewing of front-side blind spots was developed and mounted on a vehicle to verify the effects. The results confirmed that the mirror system enhanced abilities to pull up to curbs and drive through narrow passages. Future plans include the ergonomic investigation of mirrors to enable easier recognition, and quantization of mirror parameter requirements.

Note that the results of this study were applied as side view support mirrors⁽²⁾ to the 2010 year model STEP WGN.

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Next Generation Vehicle Safety System Research via Wireless Communication

Sue Bai*

ABSTRACT

In 2007, more than 41,000 people were killed in traffic collisions in the United States (NHTSA 2007). As the number of vehicles on U.S. roads continues to increase, and we at Honda continue to research autonomous safety systems, a new area of safety research has emerged – specifically, wireless communication-based active safety technology. Using wireless communication, vehicles exchange highly reliable, low latency safety messages with each other and the infrastructure to help prevent collisions. This effort represents a major step toward the goal of “vehicles that do not crash.”

This paper presents an overview of the projects in DSRC communication based Vehicle-to-Vehicle (V2V) and Vehicle-to-Infrastructure (V2I) safety technology research. Prototype system development and testing results are presented, as well as USDOT proposed further research plan.

1. Introduction

A wireless communication enabled safety system has the potential to complement or in some instances, provide alternative approaches to autonomous safety system.

In 1999, the US Federal Communications Commission (FCC) allocated a wireless spectrum in the 5.9 GHz frequency range for use by Dedicated Short Range Communications (DSRC) systems for supporting Intelligent Transportation Systems. National Highway Traffic Safety Administration (NHTSA) and automakers collaborated through the Crash Avoidance Metrics Partnership (CAMP) Vehicle Safety Communications Consortium. The potential of Vehicle to Vehicle (V2V) and Vehicle-to-infrastructure (V2I) communication to enhance vehicle safety was investigated. In 2006, reflecting the results of the research experience, the FCC refined the rules for DSRC to explicitly consider “vehicle-to-vehicle collision avoidance and mitigation.”

Honda is dedicated to providing our customers with top level safety features. We are continuously searching for new solutions to enhance safety for our customer.

Since 2006, Honda R&D Americas has engaged in cooperative agreements with CAMP to conduct pre-competitive system development research on V2I and V2V communication-based safety applications in the Cooperative Intersection Collision Avoidance System – Violation (CICAS-V) and the Vehicle Safety Communications-Applications (VSC-A) projects.

2. Cooperative Intersection Collision Avoidance System (CICAS-V) project

Intersection-related collisions account for 12% of fatal accidents in the U.S., based on NHTSA Fatality Analysis Report System 2006 data. The CICAS-V project focuses on intersection signal violation warning application through DSRC communication between a vehicle and an intersection.

The objective of the CICAS-V project was to design, develop, and test a prototype system to help prevent intersection collisions by warning the driver when the in-vehicle system determines a potential intersection traffic signal or sign violation.

2.1. Concept of operation

The concept is shown in Fig. 1. A CICAS-V equipped intersection has a Roadside Equipment (RSE) that continuously broadcasts DSRC messages over the air. The messages contain the signal phase and timing (SPAT) of the traffic signal controller output for each direction of travel, a digital representation of the intersection geometry (GID) with high accuracy, and the GPS differential corrections.

A CICAS-V equipped vehicle has an Onboard Equipment (OBE) that receives DSRC messages from the RSE as the vehicle approaches a CICAS-V equipped intersection. The OBE continuously monitors the distance from the vehicle to the intersection stop bar, the vehicle

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speed, the lane of travel, as well as the signal phase and timing of the traffic light corresponding to the lane in which the vehicle is traveling. The OBE then determines whether the vehicle has enough time to pass the intersection before the traffic light turns to red. If it is determined that a violation is likely to occur, the OBE issues a warning to the driver.

For stop sign intersections, the OBE determines the potential stop sign violation by calculating whether the vehicle has enough time to come to a stop before reaching the stop bar.

2.2. Geometric intersection description (GID)

In order for the vehicle OBE system to determine the correct traffic signal that the vehicle is heading toward, an accurate intersection digital map is necessary. In CICAS-V project, the intersection digital map was based on a 30 cm level accurate road and lane geometry for all the lanes and roads near the intersection.

The road and lane geometry in the original survey data are too large to fit into one DSRC packet size of 1.4KB. Therefore, the geometry information is compressed into a simplified form, which contains an array of points that are packaged into a single DSRC message. The vehicle's OBE parses the package and recreate the accurate intersection geometry map.

Figure 2 illustrates the contents of a typical GID. In a GID, all geometry points are offsets from a single reference point with absolute latitude, longitude and altitude location information, and this point serves as the origin of the GID coordinate.

Lanes and roads are described as an array of points with lane width information at each point. The order of the points starts from the stop bar towards the outskirt of the intersection in that lane/road. Each of the lane points represent the location of the center of the lane or road. The array of points for each approach covers a 300 meter distance from the stop bar.

2.3. Signal phase and timing

The RSE in a CICAS-V equipped intersection broadcasts the DSRC message, which contains the SPAT

information in a fixed frequency. Normally the frequency is set to 10 Hz. The SPAT message contains the current signal phase, time to next signal phase change and the approaches that are associated with that signal, for each of the signals in that traffic signal control unit.

The signal phase information is obtained through the traffic control unit in real time, and the information is packaged into a single DSRC message.

2.4. GPS corrections

In order for the vehicle to map itself to the correct lane in the GID received by the vehicle's OBE system, it is necessary to apply external position corrections to the vehicle's GPS positioning system. In the CICAS-V project, the RSE is designed to be able to collect differential GPS (DGPS) information through an external high-accuracy GPS receiver, package the DGPS correction data into a single DSRC packet in the standard Radio Technical Commission for Maritime Service (RTCM) format, and broadcast the correction packet over the air.

Figure 3 illustrates the structure of the GPS correction module of the RSE system.

The RTCM 1001 form is selected to be an adequate form with good accuracy improvement and a modest over-the-air data volume requirement. In most cases, the RTCM 1001 correction message is less than 100 bytes.

2.5. OBE system architecture

Figure 4 illustrates the system architecture of the OBE in a CICAS-V vehicle. Figure 5 shows the OBE and other hardware installed in a vehicle.

From the bottom of the Fig. 4, the CICAS-V WSU software service API module interfaces with the vehicle's Control Area Network (CAN) to obtain vehicle information, such as brake status, vehicle speed, and acceleration. The GPS positioning data obtained from the on-board GPS receiver feeds into other modules through the API module as well. The DSRC radio interfaces with the radio handler and data demux module, and sends and receives CICAS-V DSRC messages. The radio handler/data demux module packages outgoing DSRC messages, parses incoming DSRC messages, and feeds the parsed

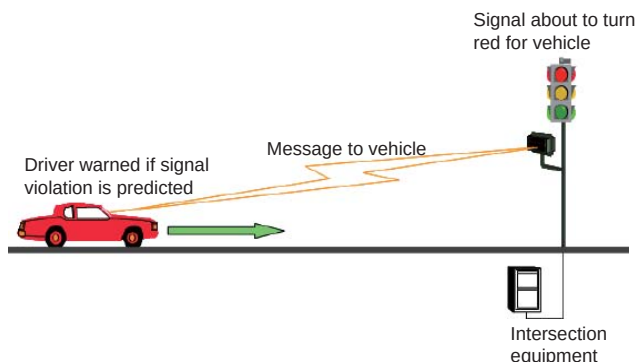


Fig. 1 CICAS-V concept of operation

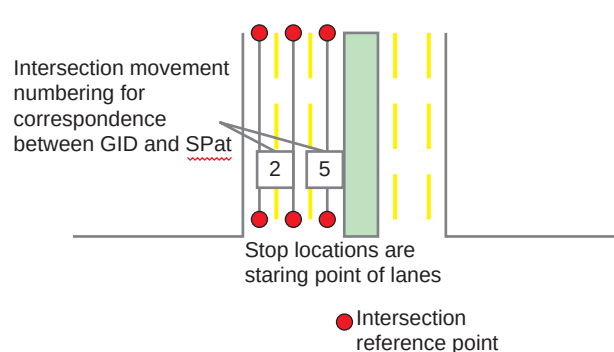


Fig. 2 GID description

data into other modules for further processing. The GID module maintains the latest GID information. The SPaT module parses the signal phase and timing data and feeds the information to other modules. DAS logger is a data logging module that records the vehicle status and other module status constantly for post-processing.

The intersection identification module determines the intersection the vehicle is heading toward based on the vehicle heading direction, current location, and the GID mapping.

The map matching module determines the most likely lane and approach the vehicle is traveling and the distance to the stop bar of that approach or lane.

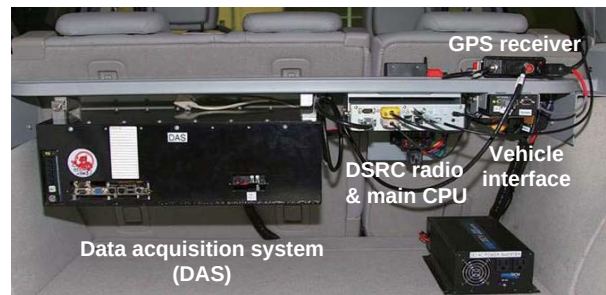


Fig. 5 CICAS-V components in a vehicle

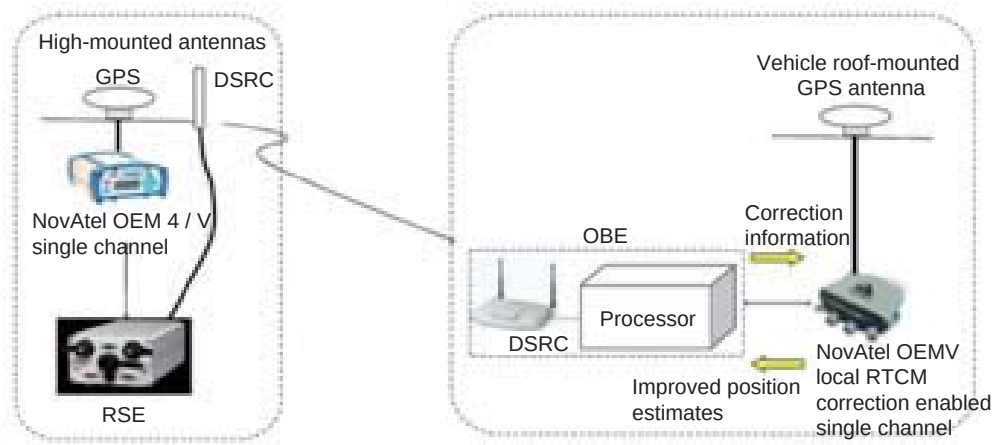


Fig. 3 CICAS-V GPS correction equipment

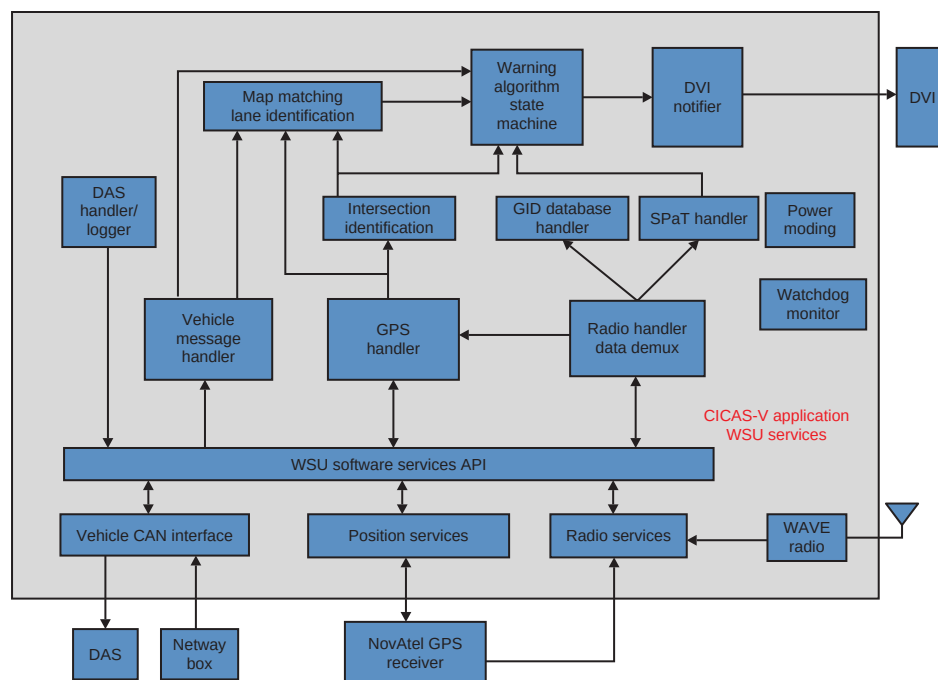


Fig. 4 CICAS-V OBE architecture

Once the correct intersection is determined and the vehicle is mapped on a specific lane and the corresponding approach with its signal phase and timing known to the OBE, the warning algorithm module determines whether a signal violation is likely to occur, and warns the driver when necessary.

A generic version of a driver-vehicle interface (DVI) is implemented with a colored intersection icon and an audible warning output. It receives output from the warning algorithm module and then issue the visual and/or audible warning depending on the warning level determined by the warning algorithm.

2.6. Objective testing

A set of test procedures was developed by the team and an objective test of the CICAS-V prototype system was performed in 2008 at the Virginia Tech Smart Road on a closed test track. Signalized intersections and stop sign intersections were used for the tests. Both true-positive and false-positive tests were conducted. The system passed all tests.

2.7. Conclusion

The prototype system was implemented successfully. It demonstrated the feasibility of implementing such systems in multiple locations across multiple vehicle manufacturers. The DSRC messages created in CICAS-V project are incorporated into the SAE J2735 standard and are being further refined in the next standard publication.

3. Vehicle Safety Communications – Applications (VSC-A) project

VSC-A project is a three year project that started in December, 2006. This project focused on V2V safety applications. The technology development objectives of the VSC-A program are to develop scalable and common vehicle safety communication architecture, protocols and messaging framework or interfaces necessary to achieve interoperability among different vehicle manufacturers. It also seeks to develop accurate and affordable vehicle positioning technology in conjunction with the 5.9 GHz DSRC to support most of the safety applications with high potential benefits, and to define a set of DSRC + positioning-based vehicle safety applications and establish a set of minimum system performance requirements.

3.1. Application Selection

Based on the 2004 NHTSA's General Estimate System (GES) data, the V2V applications such as rear-end, head-on, angle and sideswipe types are applicable to approximately 66% of the 6.2 million crashes. Seven crash scenarios were chosen based on their crash rankings by frequency, cost, functional years lost and a composite crash ranking based on the above three ranks.

The following applications were selected to correspond

to each of the selected top crash types:

- Emergency Electronic Brake Light (EEBL) – the EEBL application enables a vehicle to broadcast a self-generated emergency brake event to surrounding remote vehicles. Upon receiving such event information, the surrounding vehicles determine the relevance of the event and, if necessary, warn the driver. This application is particularly useful when the driver's line of sight is obstructed by other vehicles or bad weather conditions, such as fog or heavy rain.
- Forward-Collision Warning (FCW) – The FCW application warns the driver of an impending rear-end collision with a vehicle ahead in traffic in the same lane and the same direction of travel. FCW is intended to help drivers avoid or mitigate rear-end vehicle collisions in the forward path of travel.
- Intersection-Movement Assist (IMA) – The IMA application is intended to warn the driver when it is not safe to enter an intersection due to high collision probability with other vehicles.
- Blind-Spot Warning (BSW) + Lane-Change Warning (LCW) – During a lane-change attempt, the BSW + LCW application informs the driver when a vehicle is detected in the driver's blind spot zone, and warns the driver if the zone the vehicle intends to change to is, or will soon be, occupied by another vehicle traveling in the same direction.
- Do-Not-Pass-Warning (DNPW) – When a driver attempts to pass a slower vehicle and the passing zone is or will be occupied by a vehicle traveling in the opposite direction, the DNPW application informs the driver and, if a potential collision is determined, the system warns the driver.
- Control-Loss Warning (CLW) - The CLW application enables a host vehicle to broadcast a self-generated control loss event to surrounding remote vehicles. Upon receiving such event information, the remote vehicle determines the relevance of the event and, if necessary, warns the driver.

3.2. System Architecture

In this project, vehicles equipped with the DSRC radio and vehicle positioning system periodically broadcast their location, travel path, and vehicle status (e.g., brake, acceleration, etc.) over the air through the DSRC radio. In addition to broadcasting messages, the DSRC safety system in each vehicle also receives over the air messages from other vehicles, and then determines the relevance of the contents in the message to the safety applications. If a potential collision is detected, advisory or warnings are presented to the driver through a common Driver-Vehicle Interface (DVI) in both audio and visual means.

The DSRC safety system consists of the following subsystem/modules:

- Interface subsystem: Vehicle Controller Area Network (CAN) to OBE interface module, data logger and engineering graphic user interface
- Communication subsystem: Wireless Message Handler (WMH), Security Module and DSRC radio

- Positioning Subsystem: GPS module and Forward-looking camera module
- Supporting Subsystem: Sensor Data Handler (SDH), Path History module (PH), Path Prediction module (HVPP), Target Classification module (TC)
- Application Module: all six applications
- Threat Arbitration module (TA)
- Engineering Driver Vehicle Interface Notifier (DVIN) module

Figure 6 shows the architecture design of the DSRC V2V system. Figure 7 describes the hardware components of the system.

In Fig. 6, from the bottom, the system interfaces with vehicle sensor output through the CAN bus, receives GPS

location information from an external GPS receiver and antenna, transmits and receives DSRC over the air messages through an embedded DSRC radio, has a security module that signs outgoing messages and verifies incoming messages, and has a wireless message parsing/scheduling module –Wireless Message Handler (WMH). The received vehicle sensor data, GPS location data and remote vehicle's data are then processed through the sensor data handler, and feeds into the Target Classification module (TC) and Path Prediction Module (PH) to further determine the relative position of each of the remote vehicles. The applications receive TC output and determine the relevance of the remote vehicle's position and status, and then output the threat level. The

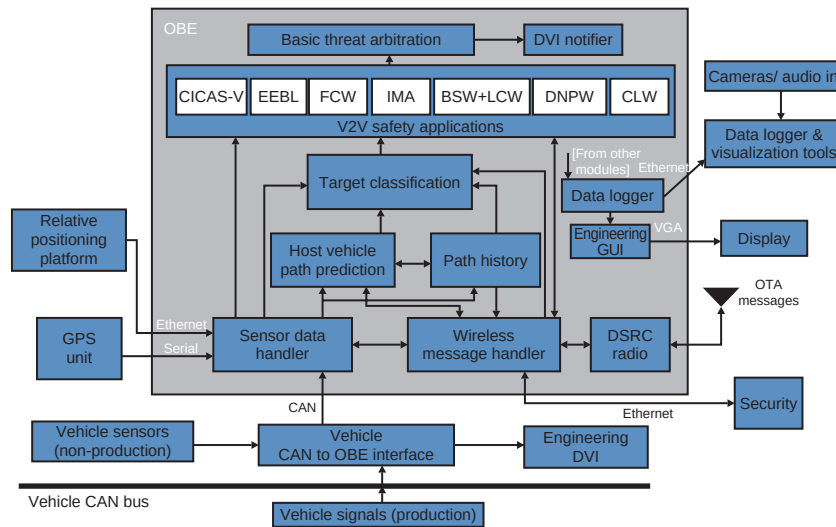


Fig. 6 VSC-A system architecture

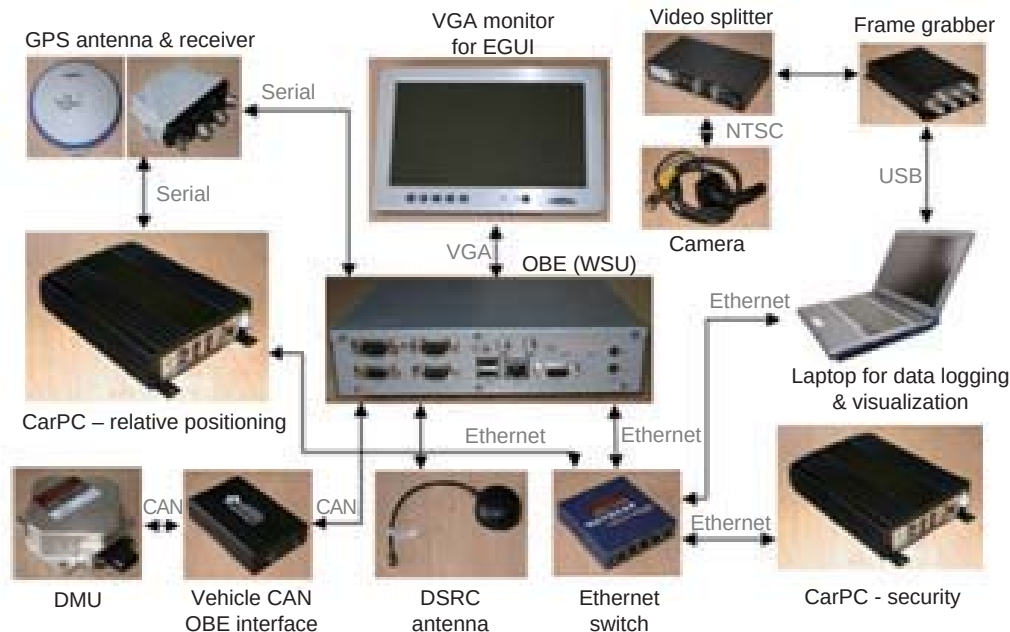


Fig. 7 System main hardware components

Threat Arbitration module (TA) arbitrates the threats when multiple threats are present from multiple applications, determines the ranking of the threats, and outputs the rankings and threat-relative data to the DVI module that controls how and when to inform and warn the driver.

3.3. Sub system design – path history

One of the challenges in identifying surrounding vehicles' locations relative to the host vehicle (e.g., ahead, ahead left/right, intersecting left/right, etc.) is that the instantaneous position and heading of the remote vehicle is not sufficient to generate the relative coordinate due to the road shape. For example, on a curved road, the leading vehicle may be several meters distance away laterally at the middle of the curved road while the host vehicle is just entering the curve in the same lane. In this case, if only the instantaneous latitude, longitude and heading of the remote vehicle are utilized to calculate the relative positioning, the result will point the remote vehicle toward far right from the host vehicle. In order to improve the performance on target classification, the vehicles generate a set of points that represents the past 300 meter distance of travel, which is called the Path History. The path history of each vehicle is then transmitted over the air, to assist the surrounding vehicle in determining the vehicle's classification.

Due to the limitation on DSRC bandwidth, the data exchanged over the air needs to be minimized. Therefore, the path history object was further refined and compressed using a concise point representation, depicted in Fig. 8.

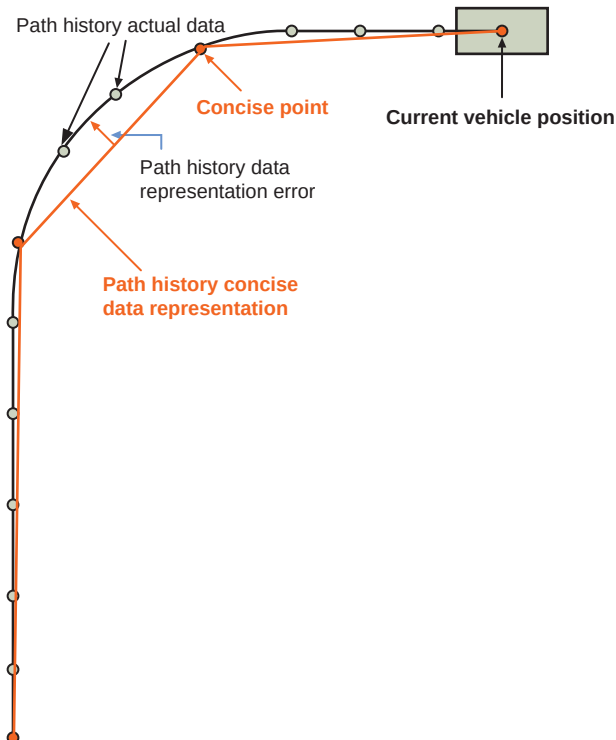


Fig. 8 Path history concise point representation

The Path History module maintains an array of the vehicle's recent position and sensor data points over a certain distance, and computes a concise representation of the actual path history of the vehicle based on allowable position error tolerance between the actual vehicle path and its concise representation. The concise representation is updated periodically and then transmitted over the air by Wireless Message Handler (WMH) module.

After intensive testing, the error tolerance of one meter was found to be adequate for good performance of all the safety applications developed in this project.

Figure 9 shows a typical concise path history representation for a vehicle traveling on an oval track consisting of straight paths, tight curves with estimated radius of 279 meters, and wide curves. With one meter allowable error, for a 300 meter distance path history, the number of concise points that was required varied from two points to nine points.

3.4. Positioning

One of the objectives of the project is to develop an accurate and affordable vehicle positioning solution. DSRC communication enables the vehicle to keep track of surrounding vehicles and the potential threat. In order to determine the potential threat, lane level positioning precision is required so that the potential threat can be identified as to which lane it presents.

Two methods are implemented as shown in Fig. 10. One is to use GPS latitude and longitude data over the air exchange and difference the current vehicle location from the over the air surrounding vehicle data. The second method is to exchange pseudo range data from the vehicle's GPS receiver over the air and each vehicle processes the surrounding vehicle's pseudo range data and generates high-accuracy relative positioning data using an onboard RTK engine.

Extensive testing was conducted and the comparison between these two methods of relative positioning solution is illustrated in Fig. 11. In the test scenario, two test

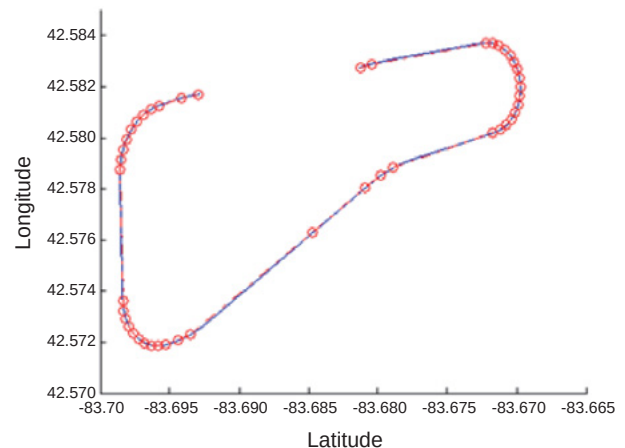


Fig. 9 Oval track vehicle trajectory and path history concise representation

vehicles were driven in the same lane on the straight road, so the lateral distance between the vehicles was maintained close to zero. The positioning system was configured to estimate the lateral distance using Real Time Kinematic (RTK) engine and using the latitude/longitude differentiation method (Single point method). The top plot in Fig. 11 shows the comparison of the output using both relative positioning approaches. Although both methods were able to provide less than one meter error in most parts of the driving, the output from the single point method was found to be more sensitive to the differences in the satellite visibility to each vehicle.

3.5. Performance specification and objective testing

A set of performance specifications for V2V safety systems were developed. The performance requirements include a common domain that applies to all of the VSC-A applications, such as the vehicle types, roadway characteristics, traffic density, GPS coverage constraints and weather conditions, the minimum requirements for the overall correctness of true-positive and false-positive warnings, and the detailed performance requirements for each application including the operating speeds, the detection/warning zone each application shall cover, and both the true positive and false positive aspects of the

application performance.

A set of objective testing procedures was developed and the system was tested in June 2009 on a closed test track at the Transportation Research Center in East Liberty, Ohio using the objective testing procedures also developed by the project. The scenarios represent some of the typical crash types that each application is designed to help prevent. The performance specification and the test procedures can be used in the future to test application performance across different vehicles and application developers.

3.6. Interoperability and standard development

Another objective of this project was to develop a scalable and interoperable vehicle safety communication systems architecture, including channel considerations, protocols, and security.

The application level over the air data exchange content based on the previous SAE DSRC J2735 standard was implemented. During the project development, the data contents continue to be refined and compressed in order to optimize the application performance while minimizing the over the air data volume.

The application level message consists of two main parts:

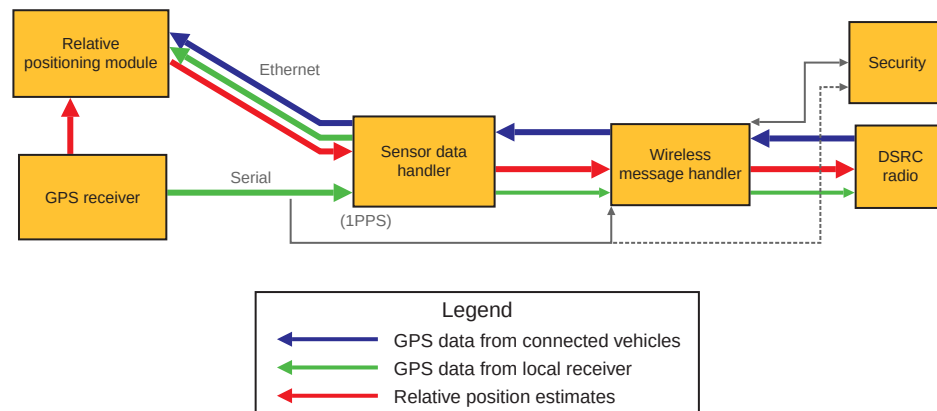


Fig. 10 Relative positioning module

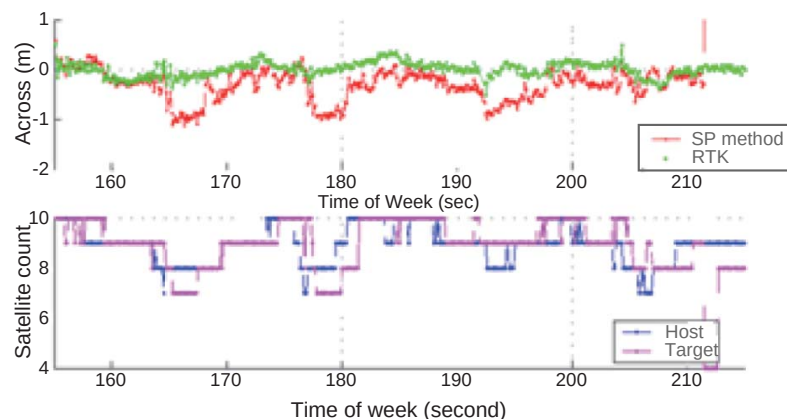


Fig. 11 Comparison of V2V lateral distance estimates

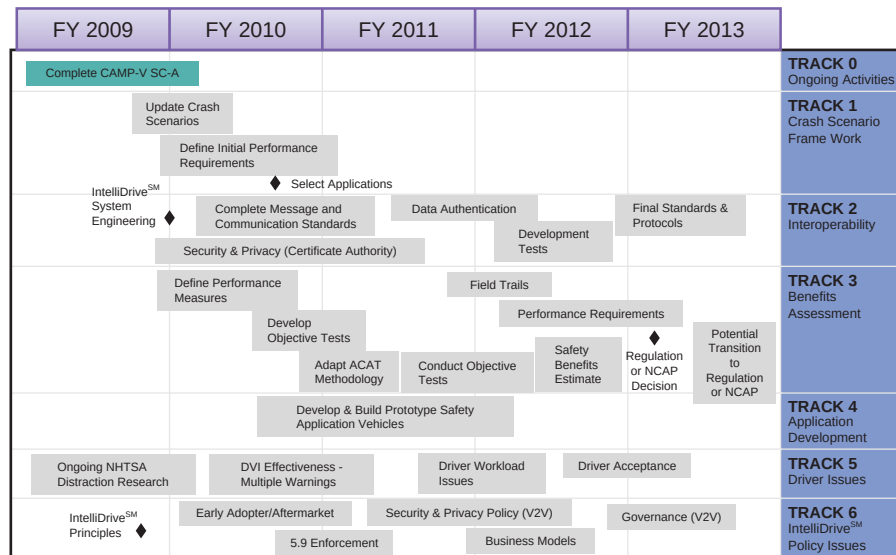


Fig. 12 Intellidrive V2V safety research plan

- Part I - Contains basic vehicle state, such as vehicle ID, GPS time, location information, motion, vehicle size, etc.
- Part II - Contains additional information, such as vehicle event, path history, path prediction, and RTC messages that are used to transmit GPS pseudo range data for high-accuracy relative positioning engine in the DSRC safety system.

A DSRC message with Part I is transmitted at 10Hz frequency as the default rate, while Part II data can be transmitted at equal or lower frequency.

The finalized message content is now included in the SAE J2735 version 2 which was published in November, 2009.

3.7. Conclusion

The VSC-A project was completed in December, 2009. The current implementation of the V2V safety system performs well in good conditions. The test results indicate that these types of test applications and systems potential can be deployed. We anticipate safety benefits by deploying such a system.

Through the development experience, it is identified that from the technical challenge perspective, further work is necessary, especially in V2V positioning and security implementation.

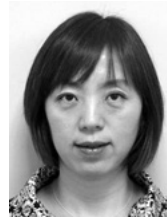
4. Next Step

The next phase of the V2V and V2I research is being planned. The 3-year research plan is shown in Fig. 12. It focuses on interoperability with larger scale implementation and is scheduled to start in December, 2009, immediately after the VSC-A project's completion. The USDOT's V2V research plan for the next phase is shown in Fig.12. It is anticipated that new V2I applications prototypes will be implemented as well.

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■ Author ■



Sue Bai

Research in Technology for Simulating Dynamic Behavior of Cam Drive Mechanism in Single-Cylinder Engines

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ABSTRACT

A technology has been established for theoretical evaluation of timing chain systems, including drive loss and slapping noise at idling engine speed, in single-cylinder engines. To date, almost no theoretical evaluation reports have been published about drive loss and slapping noise in cam drive systems including timing chains. The specifications for tensioner lifters and tensioner guides that satisfy slapping noise and friction loss requirements have therefore been determined only through tests with actual engines. The highly accurate mechanism simulation model developed in this research is populated with data obtained through bench tests with actual engines. This includes the static and dynamic characteristics of the cam chain and tensioner guide as well as the cam sprocket and cam chain contact stiffness and friction coefficient. The results have confirmed a high correlation at the absolute value level between the simulation values and the actual engine test values for the load generated when cam chain slapping noise occurs as well as drive loss in the cam chain system. This research enables examination at the initial design stage of cam chain systems intended to achieve a high-level balance of enhanced fuel efficiency and low noise in engine performance.

1. Introduction

Motorcycles equipped with single-cylinder engines make up the mainstream among small motorcycles, and there is demand for reduction of drive loss in cam drive systems in order to achieve greater fuel efficiency. Proper layout of the cam drive system can enhance the fuel efficiency of the engine. In addition, cam chain systems are used as the cam drive system in small motorcycles because of their compactness and the durability when driven under a high tensile load. There is demand for reduction of the noise generated when the cam chain and cam sprocket come into contact. Consequently, there is a need for development of a numerical analysis system to enable quantification of drive loss in the cam drive system as well as noise generated by contact with the cam chain. Numerical analysis has lately been employing structural analysis that makes use of three-dimensional models. In order to conduct structural analysis of cam drive systems, the valve assembly cam torque, the cam chain behavior, and the tensioner guide behavior must be determined simultaneously. This necessitated the creation of complex,

large-scale analytical systems, such that practical application became an issue. The modeling of the cam chain further involves issues with the large number of constituent elements and the quantification of contact with the cam sprocket. Meanwhile, advances in recursive methods and other such analytical algorithms in the domain of numerical computation have resulted in reductions in the time taken by analysis of multi-degree-of-freedom particle system models, and these methods are becoming practical tools for examining design parameters. Given this context, the present system was developed to perform comprehensive simulation of the behavior of the cam chain and cam drive system as well as the behavior of the valve assembly, and research was conducted on practical applicability of the developed system⁽¹⁾. This article presents an overview of the developed system, describes the method for modeling cam chain-related parts, and explains predictive evaluation performed by simulating the behavior that occurs when the cam sprocket and cam chain come into contact in the idling speed range as well as drive loss in the cam drive system.

* Motorcycle R&D Center

2. Overall Profile of System Development

The valve assembly mechanism to be developed was designed in SOHC configuration with a single cam shaft above the combustion chamber as frequently adopted in small motorcycles. Figure 1 shows an overview of the developed system. The cam torque changes at each engine speed due to the increase in the inertia force that results from acceleration of the valve system at high speeds. Consequently, the cam, valve, valve spring, and rocker arm were modeled so that the cam torque could be taken into account at all normal engine speeds. The tensioner guide, chain guide, and tensioner lifter, which influence cam chain behavior, were also modeled (hereafter referred to as the cam chain system model). The operating conditions of an actual engine were recreated by performing numerical integration in the time domain for each of these constituent elements using the Newton-Raphson method.

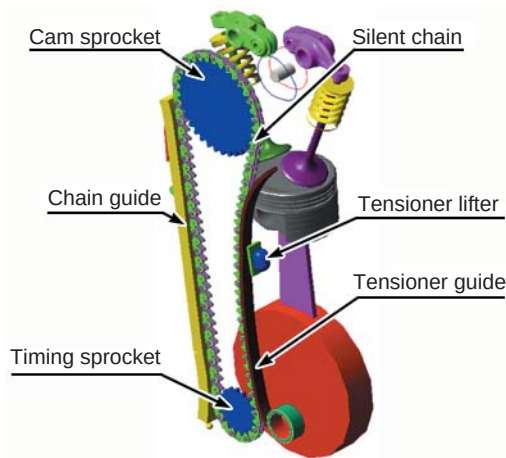


Fig. 1 Overview of developed system

with the cam sprocket and contact between the tensioner guide and chain, the actual shapes of the individual chain links were reflected in the model without reducing the links into simplified particle systems that have no shape. This system, as shown in Fig. 2, was made a multi-degree-of-freedom model in which each chain link is assigned a mass and moment of inertia and the connections between chain links are represented by non-linear springs. For the figure for stiffness between chain links, the cam chain was subjected to a tension test using an Amsler testing machine, and the correlation was confirmed from the load displacement. The accuracy of the resonant frequency of the cam chain, as representative of the dynamic characteristics, was also verified using the measurement tool shown in Fig. 3. Excitation was applied by an electromagnetic oscillator for each pre-tension and vibration acceleration, and the amplitude in the part marked A in Fig. 3 was measured. The results from actual measurement and simulation are shown in Fig. 4(a) and Fig. 4(b).

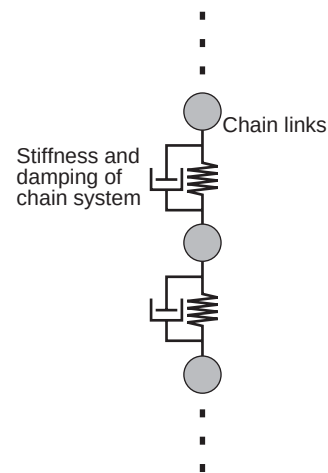


Fig. 2 Multi-degree-of-freedom model of chain links

3. Cam Chain System Modeling Method and Verification of Characteristic Model Values

3.1. Issues in Cam Chain System Models

The cam chain, which transfers the movement of the crankshaft to the cam shaft, is made up of a large number of link elements. The stiffness and attenuation of the chain as a whole possess non-linear characteristics because of such factors as friction and contact stiffness between chain links. In order to quantify the state of contact and friction between the cam chain and other parts, the characteristic values of the numerical model must also be obtained. Methods for modeling the chain system were therefore considered.

3.2. Cam Chain

In order to create an accurate model of engagement

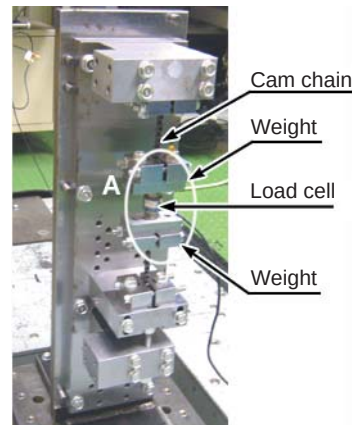


Fig. 3 Measuring tool for chain link dynamic characteristics

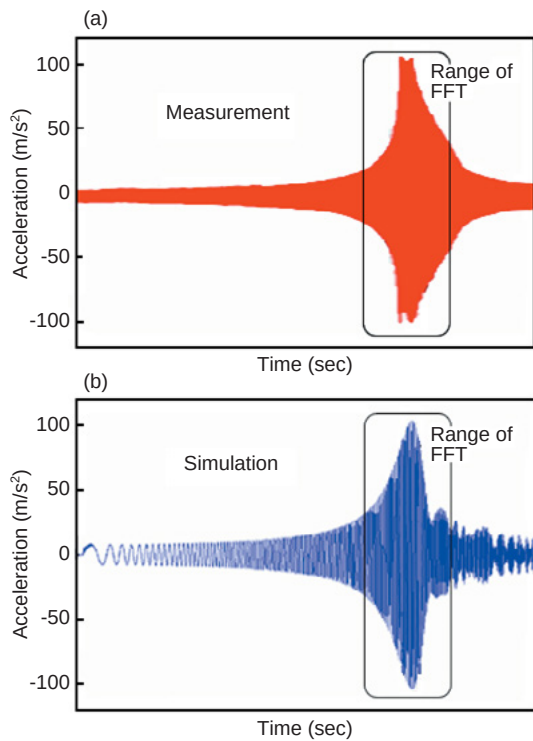


Fig. 4 Measurement and simulation results for the dynamic characteristics of chain links

Table 1 shows the FFT results from actual measurement and simulation. This confirms the close correlation of resonant frequency and amplitude when resonating.

3.3. Chain Guide

Figure 5 shows the model of the chain guide. The characteristics of the chain guide shape were expressed by dividing the chain guide into 20 beam elements. The amount of bending deflection of each part under unit load obtained by numerical calculation was matched to the actual measurement to identify the stiffness characteristic value for the beam elements.

3.4. Cam Sprocket

Figure 6 shows the model of the cam sprocket and cam chain. The engagement with the cam sprocket in the model was represented by incorporating consideration of contact stiffness, contact attenuation, and friction in the portion that comes in contact with the cam chain. The pendulum test shown in Fig. 7 was conducted in order to determine the characteristics of this engaged portion. This was done to find the change over time of the total energy of the system and to verify the correlation of the measurement and simulation. As shown in Fig. 8, the total energy is reduced over time, and it was confirmed that the calculation results provide a good match with this rate of reduction.

Table 1 FFT results of measurement and simulation

	Measurement	Simulation
Resonance frequency (Hz)	424.1	430.0
Amplitude (m/s ²)	31.7	31.9

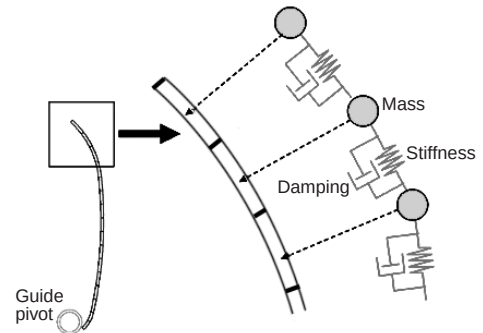


Fig. 5 Model of chain guide

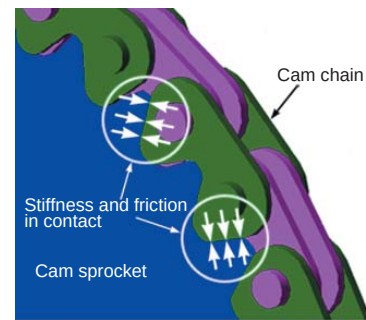


Fig. 6 Model of cam sprocket and cam chain

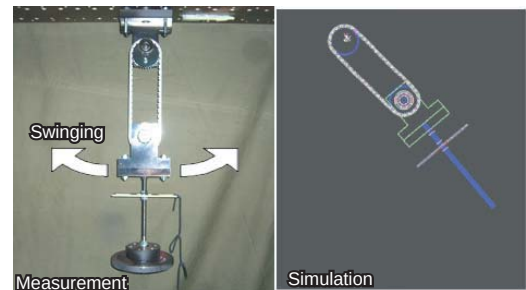


Fig. 7 Pendulum test for measurement of mesh area contact stiffness and friction

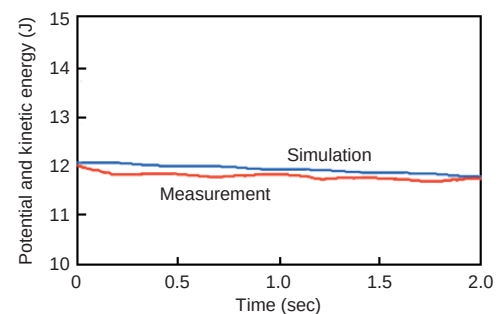


Fig. 8 Comparison of simulation result and measurement result of potential and kinetic energy

4. Verification of Cam Chain System Model Correlation with Actual Engine Operating State

The simulation figures for the cam drive torque and the load generated on the lifter were compared with the actual measurements to verify that the developed cam chain system model accurately represents the operating state of an actual engine. Figure 9 shows the actual engine used, which was a single-cylinder 150 cm³ engine. Measurements of the actual engine were made from 1400 rpm to 9000 rpm. Figure 10 shows a comparison of the

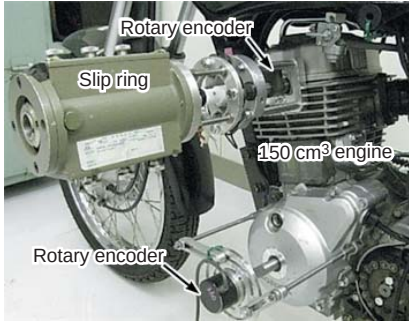


Fig. 9 Test engine and measurement system

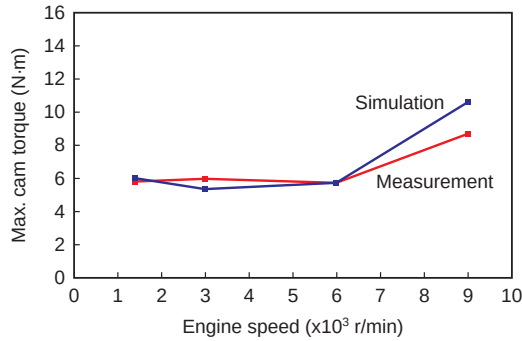


Fig. 10 Comparison of simulation result and measurement result of maximum values for timing sprocket drive torque

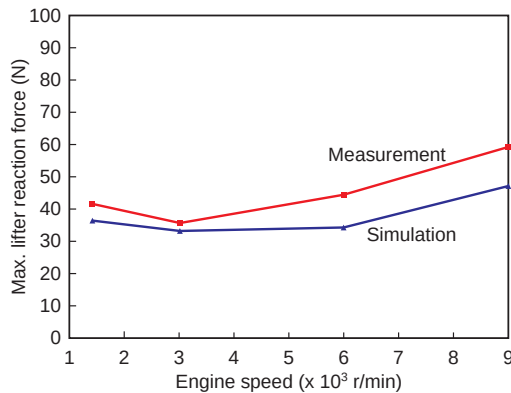


Fig. 11 Comparison of simulation result and measurement result for lifter reaction force

maximum values for cam drive torque over a single cycle. Figure 11 shows the results from comparison of load generated on the lifter. Some discrepancy can be seen at speeds of 6000 rpm and above, but the correlation is positive in the 1400-6000 rpm range. Given these results, the developed model can be said to accurately recreate the actual engine operating state.

5. Applied Research on the Developed System

5.1. Prediction of Cam Chain Drive Loss

This section describes the method for calculating cam chain drive loss. In the developed system, the crank sprocket drive torque and the cam drive torque are calculated. Methods for using these computed figures to calculate drive loss in the cam chain were examined. Equation (1) defines the drive loss torque FT for the average crank sprocket drive torque Tf and average cam drive torque Tc .

$$FT = Tf - Tc \quad (1)$$

$$Pslc = FT \cdot N \cdot \frac{2\pi}{60} \quad (2)$$

FT : Average cam chain drive torque (Nm)

Tf : Average crank timing sprocket drive torque (Nm)

Tc : Average cam drive torque (Nm)

$Pslc$: Horsepower drive loss (kW)

N : Engine speed (r/min)

This was taken as the basis for performing the comparison of cam chain drive loss in the engine as described in Chapter 2. Figure 12 shows the results.

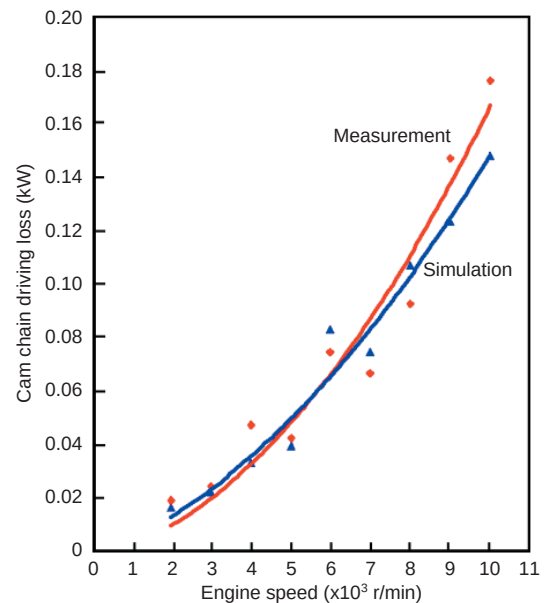


Fig. 12 Cam chain driving loss

Although fluctuations were apparent in the calculations for 6000-7000 rpm, the results show a close correlation.

5.2. Cam Sprocket and Cam Chain Contact Behavior and Slapping Noise

The noise that is made from the cam chain chamber during idling can cause discomfort and it damages the product appeal. The acceleration occurring in the vicinity of the part where the noise was generated was measured using an acceleration sensor to comprehend the timing of generation of this noise⁽²⁾. Figure 13 shows the measurement value. It was determined that the slapping noise occurred at exhaust top dead center due to the suddenly increased amplitude of acceleration. In order to further examine this phenomenon, measurements were made with a Morris damper, which attenuates the seating load, attached at the place where the cam chain is seated on the cam sprocket. The slapping noise vanished. It was discovered from the above that the slapping noise was generated by contact of the cam sprocket with the chain at exhaust top dead center. Next, in order to conduct theoretical evaluation of this slapping noise, an attempt was made to measure the force with which the cam chain and cam sprocket collide and thus generate the slapping noise. The measurement values were compared with the simulation values. The actual engine testing used the engine shown in Fig. 9. Figure 14 shows the tool used to measure the noise-generating excitation force. A three-component force gauge was fabricated to enable measurement of the X and Y components of the load on the plane surface of the cam sprocket and the cam torque, and measurements were made. Figure 15 shows a comparison of the actual engine measurement results and the calculation results with the engine running at 1400 rpm. The maximum load amplitude and the timing of noise generation both showed positive correlation, and this has enabled predictive evaluation by means of simulated slapping-noise-generating excitation force.

5.3. Evaluation of Cam Chain Layout

The main design parameters for the cam drive system include the chain guide shape, stiffness, and support point locations, and the angle of attachment and location of the tensioner lifter. These parameters have an influence on the layout of the engine as a whole. In order to achieve a

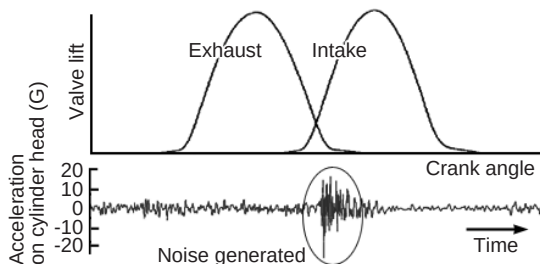


Fig. 13 Variation of acceleration on cylinder head

high-level balance of greater fuel efficiency and quietness, these design parameters have to be examined in the initial design stage. In the event that the simulations performed here were used for such a parameter study and the design parameters were changed accordingly, the change that would take place in the slapping noise was confirmed. Figure 16 shows the results of simulation evaluation of

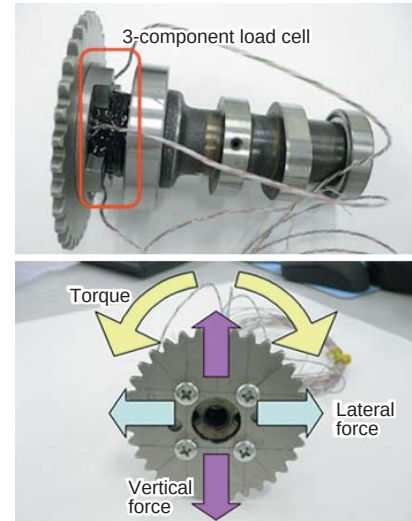


Fig. 14 3-component load cell for measuring excitation force

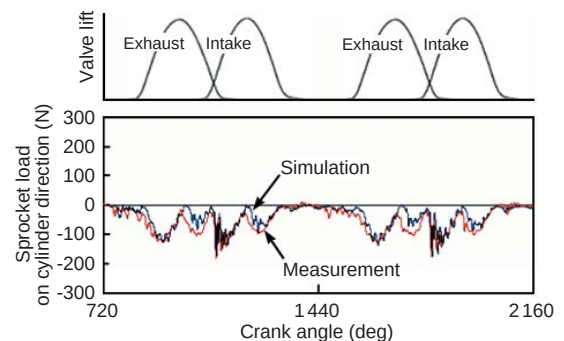


Fig. 15 Comparison of calculation result and actual engine measurement result for sprocket load on cylinder direction at 1400 r/min

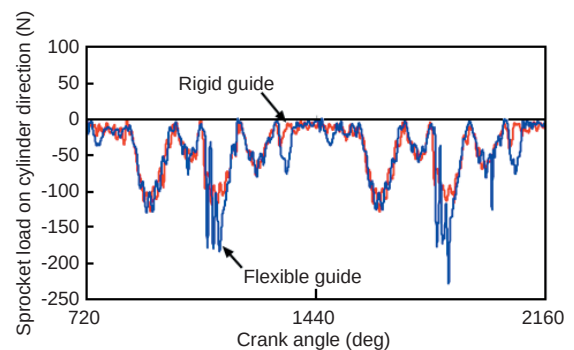


Fig. 16 Comparison of rigid guide and flexible guide result for sprocket load on cylinder direction at 1400 r/min

the excitation force that generates the cam chain slapping noise for differing degrees of stiffness in the chain guide at 1400 rpm. In the case of this engine, increasing the stiffness of the chain guide enabled a reduction in the noise-generating excitation force that was the primary factor in the cam chain slapping noise at 1400 rpm. It was found that this allowed greater quietness to be maintained.

6. Conclusions

- (1) Development of a system for coupled simulation of the cam drive system, reciprocating parts, and valve assembly enabled accurate simulation of the behavior of the cam drive system.
- (2) The phenomenon of the slapping noise that occurs when the cam sprocket and cam chain come into contact was explained, and a method for predictive evaluation of cam chain slapping noise in the idling range by means of simulation was found.
- (3) The research has enabled calculation of cam chain drive loss from the crank sprocket drive torque and the cam drive torque.

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Development of Class A Surfacing CAD System

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ABSTRACT

Class A data (or Class A surface data) is high-quality surface data used to express product styling. During the process of design development of four-wheeled vehicles, Class A CAD is used to generate Class A surface data based on 3D scanned data from clay models. An original CAD system for the generation of Class A styling data has been developed in order to respond to the increasing complexity and diversity of shape expression in four-wheeled vehicle styling, and at the same time to increase the precision of Class A data and the development efficiency of the process of generating Class A surfaces. The new system has displayed positive results in the areas of shape expression, shape accuracy, and development efficiency.

1. Introduction

In the final phase of the styling design development process, shape data for styling called Class A surfaces are created referring to 3D clay model measurement data. “Class A surface” is a collective term for smooth and continuous high-accuracy and high-quality surfaces that represent the interior and exterior shapes of the vehicle.

While the increasing density of today’s styling necessitates the use of Class A data that expresses styling shapes with a high degree of accuracy, it is also necessary to respond to further reductions in the development period, demanding the simultaneous realization of accurate reproduction of styling shapes and a high level of work efficiency from systems for the creation of Class A data.

When using conventional CAD systems, it is necessary to repeat shape operations and shape evaluations in order to create the styling shapes projected by the styling designer, and the designer must be aware of the mathematical formulas for expression of curves and surfaces when conducting these shape operations. In order to resolve these issues, the new technology discussed in this paper offers intuitive surface operations based on the behavior of plate-shaped bodies in the real world and referencing the evaluation method used by styling designers, also offers a method of correcting subject surface and curve shapes by directly operating the visualized shape evaluation result. In addition, the use of new algorithms in shape-processing calculations has

simultaneously reduced calculation time and increased the quality of the output shapes.

The new Class A data generation system was realized using original methods. This paper will provide an overview of three representative functions of the system [(1) a scanned data fitting function, (2) a high-speed and high-accuracy surfacing function, and (3) a global deformation function], and will discuss the results of using the system to create Class A surfaces in the product development process.

2. Development Goals

The development of the new system aimed to increase the speed with which Class A surfaces are brought to appropriate quality levels and to reduce the period for modeling. The target for the developed CAD system was to halve the man-hours expended on the creation of Class A data.

2.1. Developed System and Process of Creation of Class A Data

Figure 1 shows the process by which the new system creates Class A surface data using 3D scanned data from clay models.

2.2. Method of Reducing Man-hours Employed in Creation of Class A Data

In order to increase both data quality and the efficiency

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of modeling work, the development of the new system divided the process up to the stage of completion of the styling design data into the following three phases, and realized the established targets by employing new technologies for CAD functions in each work phase:

- (1) Phase 1: Surfacing based on scanned data
- (2) Phase 2: Creation of high-accuracy Class A surfaces
- (3) Phase 3: Correction of Class A surfaces

Figure 2 shows representative technologies used in each of the three work phases.

3. Outline of System Configuration

Figure 3 shows an outline of the configuration of the

new system. The lower section of Fig. 3 represents the system platform, while the upper section indicates its practical functionality. While the efficiency of system development was increased through the use of certain components available on the market, the system configuration was designed to help enable the efficient application of the three new technologies shown in Fig. 2, in order to realize the target of a 50% reduction in the man-hours employed in the creation of Class A data. The main characteristics of the system are as follows:

- (1) Loose coupling of the platform and functional sections of the system helps to ensure independence of the functions incorporated in the system.
- (2) The platform geometry kernel is provided with an original

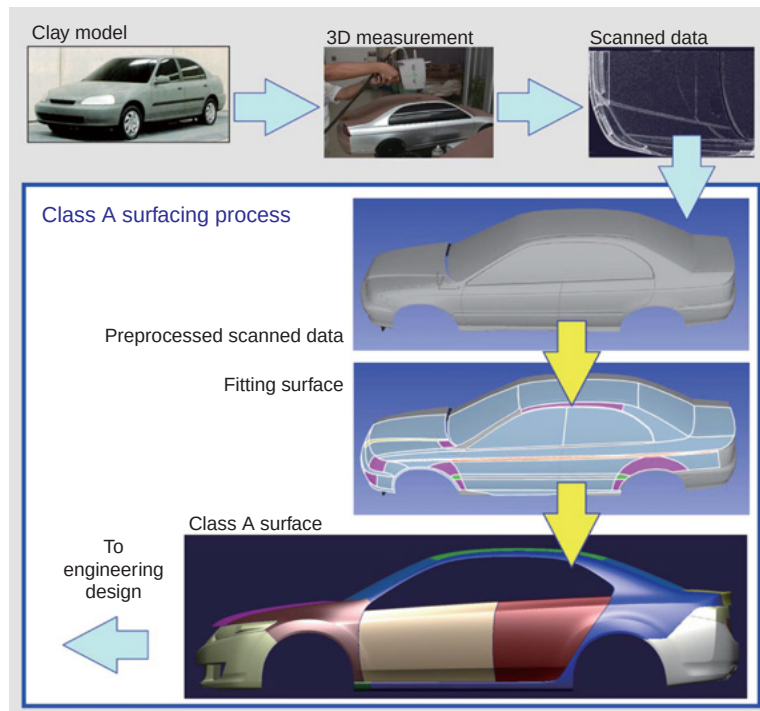


Fig. 1 Class A surfacing in styling process

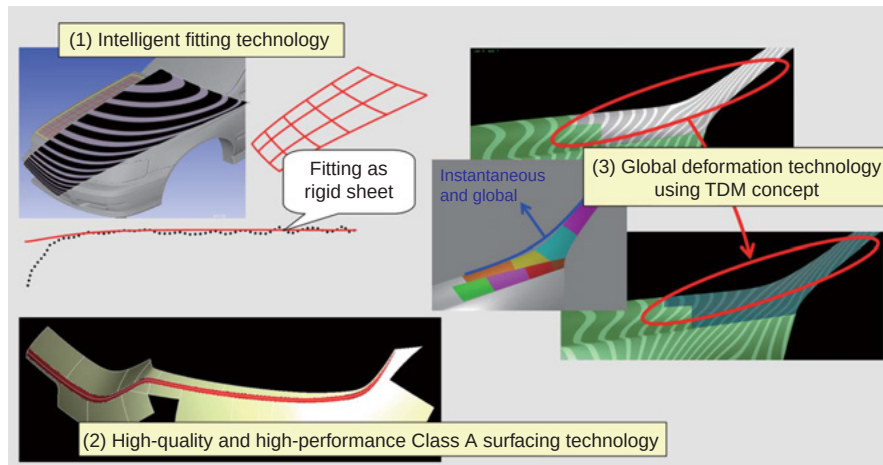


Fig. 2 Three typical technologies corresponding to three phases of Class A surfacing process

polygon kernel to help enable efficient management of large scanned data. This is shown as (A) in Fig. 3.

- (3) The three new technologies shown in Fig. 2 are incorporated in the functional section, which is unconstrained by the platform section. These are shown as (B), (C), and (D) in Fig. 3.

(A) to (D) in Fig. 3 have the following roles:

(A) is the original polygon kernel, and helps to enable efficient memory management for the use of large scanned data.

(B) is the fitting surface creation technology, the most upstream segment of the process of creation of Class A data. This technology helps to enable high-quality surfaces to be created from clay model shapes that include miniscule undulations and noise.

(C) is the core technology for the creation of Class A data. This was also regarded as the most important technology in the previous system, but the output surface calculation speed and quality have been increased in the new system.

(D) is the core technology that realizes the modeling method, the unique concept of the new system. Helping to enable intuitive deformation operations, this technology has contributed significantly to the realization of the system targets.

Due to the loose coupling described in (1) above, in addition to Class A CAD, the platform of the new system can support other CAD systems that aid styling design development.

4. Outline of CAD Functions

This chapter will provide an outline of technologies (A) to (D) above, and will introduce the main functions that have been realized through the use of these technologies.

4.1. Original Polygon Mesh Kernel

Pre-processing, including polygon mesh shape analysis of the scanned data input to the system and cutting out at the necessary points, should be conducted in the phase of preparation for Class A surfacing. The new system has been provided with an original polygon mesh shape processing kernel as the base technology to support these functions.

This kernel is the core processing section for polygon mesh shape processing, and exists as a standalone independent module. The application system links to the kernel for reference. The fact that the kernel is self-contained helps to enable it to be easily used with other applications, thus increasing versatility.

The 3D scanned data handled by the new system is large data with several million vertices. It was necessary to make it possible to work with this data interactively. The data storage and memory management systems were therefore the major issues for the system. These issues were resolved through the employment of three element technologies: (1) Half-edge structure technology; (2) Original memory pool technology; and (3) Original garbage collection technology. These technologies will be discussed below.

4.1.1. Half-edge data structure

In order to conduct efficient polygon mesh shape processing, it is necessary to be able to efficiently refer to the mutual contiguity and reference relationships of three topological elements of the polygon mesh: edges, faces, and vertices. The kernel therefore employs half-edge data structure⁽¹⁾. Figure 4 shows an overview of the half-edge data structure. The introduction of the concept of the half-edge at both sides of an edge helps to enable the efficient construction of reference relationships for mutual contiguity between topological elements. The

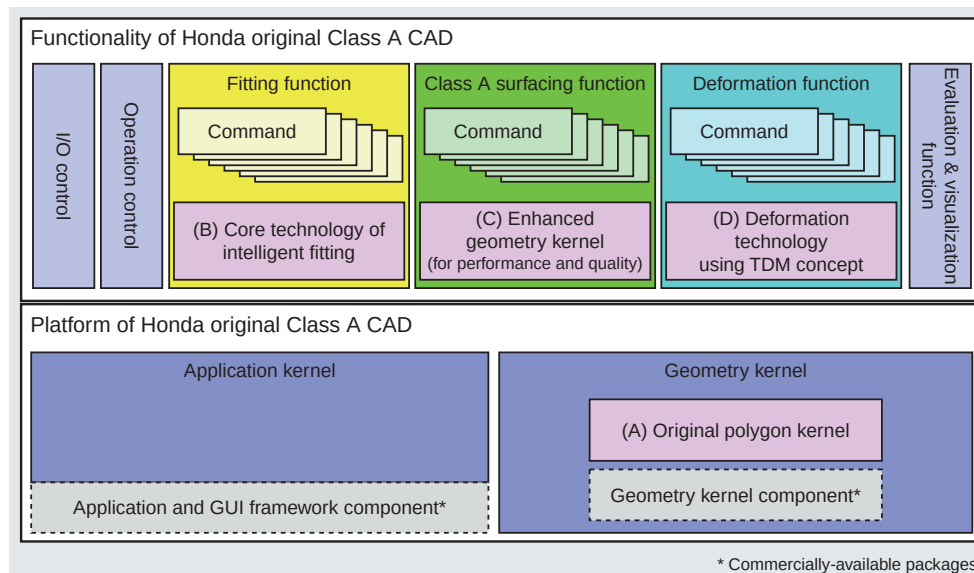


Fig. 3 Outline of system configuration

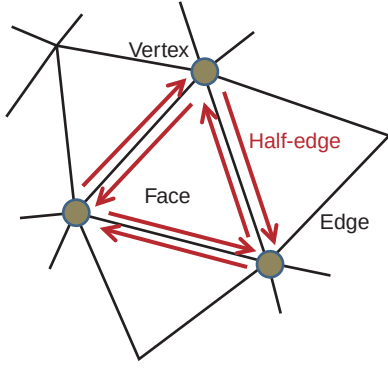


Fig. 4 Half-edge data structure

reference methods are as follows:

- For each half-edge, the vertex that forms the point of origin, the parallel edge and the half edge on the opposite side, the face in the same category, and the other half edges that share that face, are referenced.
- For each vertex, an arbitrary half-edge that is taken as its point of origin is referenced.
- For each face, an arbitrary half-edge above it is referenced.
- Finally, for each edge, one of the half-edges on the sides of the edge is referenced.

4.1.2. Fast and efficient memory fragment allocation via unique memory management

In order to help ensure memory allocation for an enormous number of objects in the range of several million and to efficiently release memory, the kernel employs a unique form of memory management based on a small object allocator⁽²⁾. Figure 5 shows an overview of the memory management mechanism. The system allocates and releases memory regions using small units of memory called “chunks.” The youngest generation of chunks are called “current chunk,” and memory fragments are provided from this chunk for each request for memory from the application. When the current chunk is filled, they are added to the chunk manager list, and new chunk is generated.

Because the number of categories of objects to be managed is limited, the kernel is provided with a total of five chunk lists, one for each of five topological elements (faces, edges, loops, half-edges, and vertices). The design of such an exclusive memory pool specialized for the polygon mesh topological element objects has helped to implement a simple and high-efficiency memory pool.

4.1.3. Garbage collection function

The repetition of shape processing operations results in objects that are no longer being referenced from the interior or exterior of the kernel being left over, and this impedes efficient memory use.

The polygon mesh kernel employs garbage collection (GC) based on a mark-sweep-compact garbage collector⁽³⁾. The GC monitors the reference status of all objects and

efficiently sweeps and releases regions occupied by unnecessary objects, and also applies anti-fragmentation processing to help prevent fragmentation of regions occupied by objects which are being referenced.

In addition, the employment of GC has made explicit memory release for individual topological elements unnecessary, and thus offers the advantage of resolving issues such as memory leaks and double free errors. An overview of the GC mechanism shown in Fig. 6 is provided below.

In panel (1) in Fig. 6, referenced and unreferenced nodes are mixed in each memory chunk while the user is repeating operations. If GC was not applied and this situation was allowed to continue, the memory chunk lists would continue to be filled with unreferenced nodes, and the system memory capacity would be used up to no effect.

Panel (2) in Fig. 6 shows the marking of all the referenced nodes to help discriminating between nodes which are referenced and unreferenced from the interior and the exterior of the kernel.

Continuing on from panel (2), panel (3) shows defragmentation for all the referenced nodes, at the same time as memory occupied by chunks containing no referenced nodes is successively released. In addition,

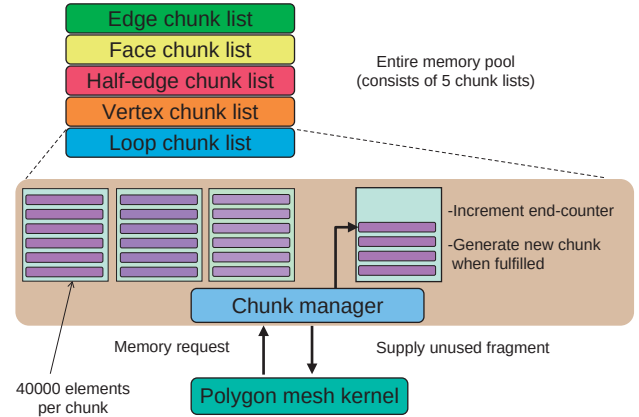


Fig. 5 Memory pool for topological nodes

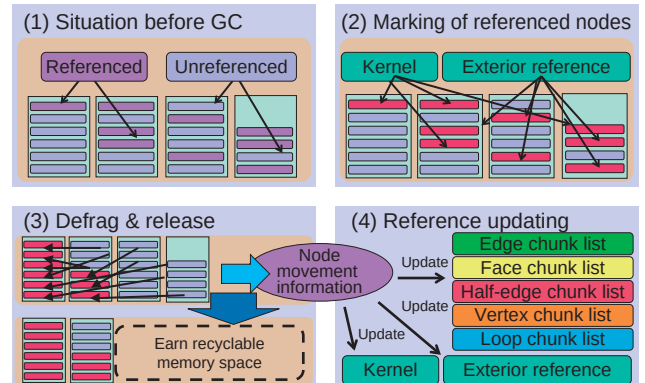


Fig. 6 Garbage collection

during this process movement data is recorded for the nodes.

Panel (4) in Fig. 6 shows updates of the reference sources for all the nodes in the kernel and exterior to the kernel, based on the node movement data recorded in (3).

Mark-sweep-compact garbage collection was applied in the system because the polygon mesh half-edge data structure used in the kernel employs reciprocal loop reference between the nodes, and other GC paradigms which are not able to work with reciprocal loop reference (such as reference counters⁽⁴⁾) would therefore have been unsuitable for topological element management in the system.

As indicated in the previous section, in this kernel space is allocated in response to memory requests from current chunks. However, memory spaces which have not yet been secured for which sweep processing⁽³⁾ and compact processing are unnecessary always exist in the current chunks, and there are therefore frequent exceptions in relation to other chunks. Full mark-sweep-compact processing is therefore only applied to saturated chunks other than current chunks. By this means, the algorithms could be defined with saturation of all the chunks other than the current chunks as a precondition, making exceptional processing for the current chunks unnecessary. This not only made it easier to incorporate the garbage collection function, but also offered advantages from the perspective of performance.

The design discussed above helped to enable the realization of a simple, practical, and high-efficiency GC function.

4.2. Fitting Technology

In the process of generation of Class A data, 3D scanned data from clay models is input, and Class A surface data that accurately represents the styling shapes is generated. Previously, this involved a repeated process in which the surfaces were generated manually, the match or discrepancy between the scanned data and the surface data was evaluated, and corrections were made manually. Because this process necessitated an enormous expenditure of man-hours, numerous mechanisms for the automatic generation of surface shapes that represent the styling shapes from scanned data have been considered. This will be defined as “technology to fit scanned data.”

Because undulations in clay models and measurement

errors were incorporated without alteration in the output, the previous fitting technology was unable to ensure the necessary surface quality for Class A data. The new system therefore removes clay model undulations and measurement errors while accurately representing the styling shapes, thus generating surfaces that fulfill Class A data quality demands. This is termed “intelligent fitting technology.” The rest of this section will provide an overview of this technology, and will discuss the differences in output quality between this technology and the previous technology.

First, the results output by the previous technology will be considered. Figure 7(a) shows a simulated reproduction of a highlight in scanned data. This image demonstrates that minute levels of noise and unevenness exist in the scanned data. Figure 7(b) shows the highlight resulting from least square fitting (LSF), a conventional technology, and Fig. 7(c) shows G2 (curvature) values. Noise in the scanned data produces unevenness and bending in the output surface, and the G2 curves also display undulations. The existence of these disturbances prevents the data from achieving Class A quality.

Next, an overview of the intelligent fitting technology will be provided and its output will be considered.

In order to maintain both surface quality and continuity between surfaces, it is necessary to control the tangency (G1) and curvature (G2) of the surfaces. In addition, as indicated above, undulations in the curvature affect quality, and because of this it is also necessary to control the rate of change of G2.

The concept of stiffness has been employed in the new system to help enable these parameters to be controlled. As Fig. 8 shows, when a metal ruler is pressed over an actual model, it follows the tendency of the surface, but the minute concavities and convexities are ignored. This is because the ruler has a specific degree of hardness, and will not bend more than a specific amount. Substituting a mathematical constraint for this concept and employing it as a condition during fitting of surfaces helped to enable surfaces obtaining Class A quality to be output.

Figures 9(a) and 9(b) show output surfaces incorporating this concept. Compared to the results shown in Figs. 7(b) and 7(c), these surfaces obviously obtain smooth highlights and changes in G2, while keeping the tendency of the scanned data. However, because fitting was conducted up to this point for each individual surface,

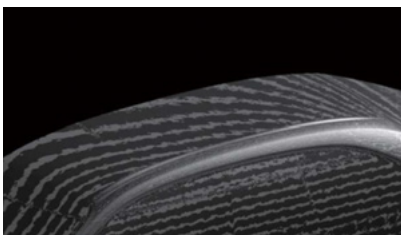


Fig. 7 (a) Scanned polygon

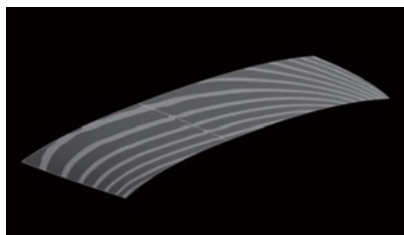


Fig. 7 (b) Highlight of LSF surface

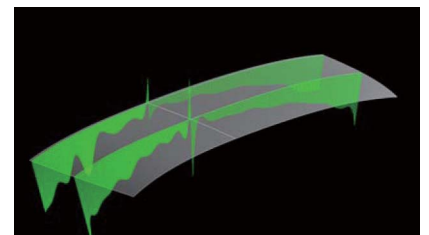


Fig. 7 (c) Curvature plot of LSF surface

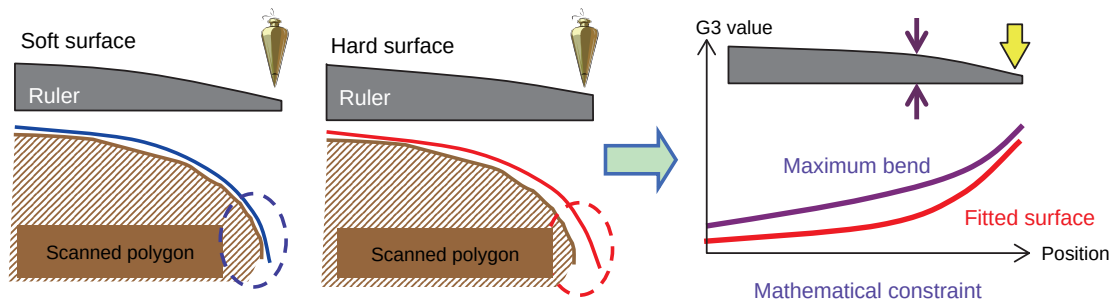


Fig. 8 Concept of stiffness

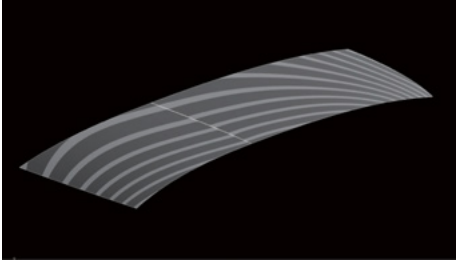


Fig. 9 (a) Highlight of intelligent fitting surface

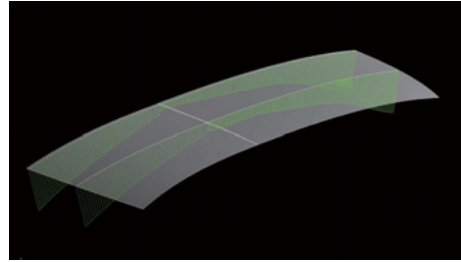


Fig. 9 (b) Curvature plot of intelligent fitting surface

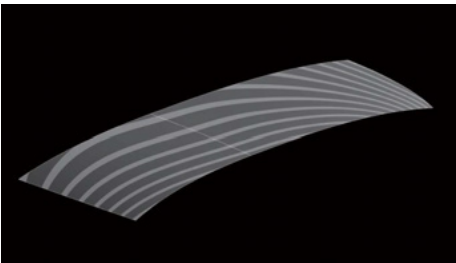


Fig. 10 (a) Highlight of merged surface

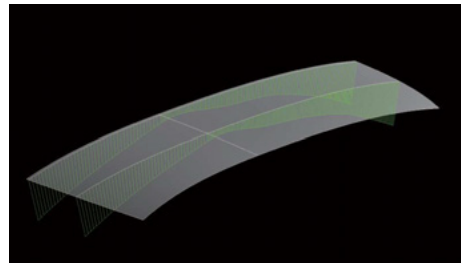


Fig. 10 (b) Curvature plot of merged surface

there was unevenness in terms of position and curvature at the boundaries between surfaces.

A “merge” technology was developed to resolve this issue.

Ultimately, Class A data is composed of multiple surfaces which have mutual continuity. “Continuity” here refers to position, G0; tangency, G1; and curvature, G2. In the past, the continuity was ensured by uniquely specifying the state of connection. However, because this method resulted in conspicuous undulations at the exterior of areas in contact, it was inadequate for Class A quality.

In the new system, mathematical constraints have been utilized for maintenance of continuity, and degrees of freedom have been granted to states of connection. This has helped to ensure continuity with minimal overall undulation. Figures 10(a) and 10(b) show output surfaces generated using the new system. Surface undulation has been controlled while continuity of position and curvature has been achieved in areas in contact.

4.3. High-speed and High-accuracy Class A Surfacing Technology

In order to realize the images produced by styling

designers, the method adopted by Honda for the generation of Class A data has necessitated the production of surface structures across which highlights flow, using curves termed “character lines” as a guide. In addition, a high level of accuracy has been demanded in order to satisfy engineering design conditions. Due to the inability of market CAD systems to satisfy the necessary conditions, and in order to simplify the generation of complex shapes necessitating the expenditure of a huge volume of man-hours, the company has developed original CAD systems using the latest technologies. However, the shapes used in recent automotive styling have become increasingly complex, and the number of surfaces employed has approximately doubled against the situation five years ago. Increased speed and increased accuracy were demanded for the process of creating Class A data. The Class A surfacing functions of the new system carry over the practical expertise incorporated in traditional data generation commands. At the same time, their speed and accuracy have been increased, and a command set extending to 185 functions has also been developed.

As a representative example, Fig. 11 shows the curve-smoothing function. The curve-smoothing function realizes

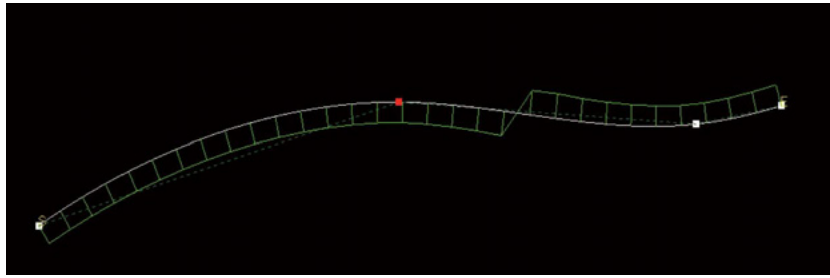


Fig. 11 Curve editing

intuitive operations that make it possible to directly select and move edit points, and has reduced the man-hours employed in correcting curves. In addition, the function also makes it possible to make corrections using control points and to make corrections while maintaining G2 continuity on the terminal using 5-degree curves, and has therefore expanded the scope of curve correction, reduced the man-hours involved in curve correction, and increased the accuracy of continuity.

For surfaces, an original Honda four-boundary surfacing command applying four-boundary Coons surfaces was developed. Because the previous system used 3-degree surfaces, the tendency of the curvature of the interior of the surfaces was sometimes erratic. The new system generates four-boundary surfaces using 5-degree C2 surfaces, significantly increasing interior continuity. Figure 12 shows the curvature of cross-section curves of four-boundary surfaces generated using the old and new systems.

In addition, the generation of fillet surfaces has been increased in speed, and the quality and accuracy of the surfaces have been increased. A fillet is a shape that rounds the angle formed by two surfaces. Fillets are important elements in satisfying the safety standard that stipulates that automotive surfaces must have no projections with a radius of curvature of less than 2.5 mm, and they make up approximately 10% of the total number of Class A surfaces. Redevelopment of the fillet-generation algorithms in the new system has increased computation speed and helped to enable the generation of fillet surfaces obtaining greater G2 continuity with support surfaces. In order to increase calculation speed, a method of calculating fillets using normal vectors of the support surfaces, the normal vectors of section planes, and the

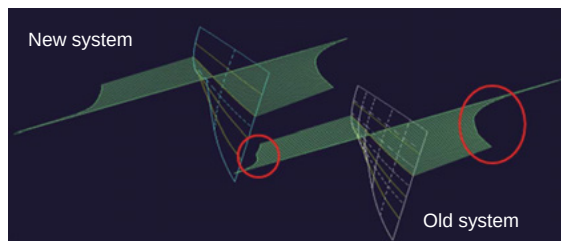


Fig. 12 Four-boundary surface

normal curvature of support surfaces was developed to replace the traditional calculation using intersections. To achieve greater G2 continuity with support surfaces, fillet surfaces were designed so that G2 continuity with peripheral surfaces was satisfied by means of staged Gordon surfacing with the cross-boundary derivative (CBD), which provides information concerning the derivative values for the peripheral surfaces, as an input. In addition, not only were the fillet-generation algorithms extensively redeveloped, but an interface was developed that synchronizes the creation of fillet surfaces with a surface quality evaluation (Fig. 13), contributing to a reduction in the man-hours employed in fillet creation.

4.4. Global Deformation Technology

When clay models are altered during the maturation process of a styling design, the Class A data created on the basis of the clay models should also be updated. In the previous system, data was corrected by means of repeated manipulation of the mathematical parameters of the existing surfaces. Because this was an indirect method of correction in relation to the target surface shapes, an enormous amount of time was required for operations conducted by trial and error. At times, it was faster to simply regenerate the surfaces. Using this method of creating Class A data, updates to the data were unable to keep pace with alteration of the clay models.

For the new system, technologies were developed to

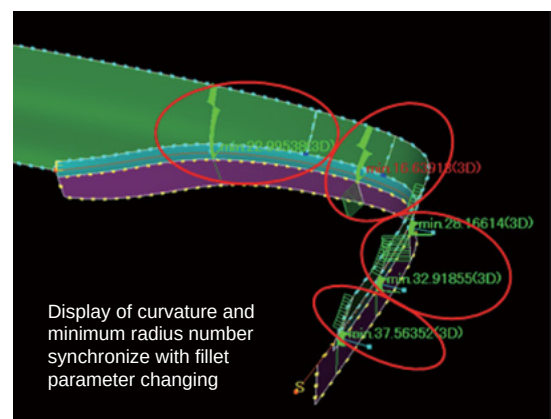


Fig. 13 Creation of fillet with dynamic evaluation

make the correction of existing surfaces more intuitive and efficient, using a method of correction of surface shapes in which styling designers directly input shape targets in accord with their aesthetic intention. Because shape targets are directly input in this method, it is termed target-driven modeling (TDM). This section will provide an example of the application of this new technology.

4.4.1. Highlight control

Highlight control applies the concept of target-driven modeling to the 2D highlights which represent one of the standards for quality evaluation of Class A data. When a styling designer designates the target highlights, this function deforms the existing surfaces to produce those target highlights. Figure 14 shows the results produced by the function in relation to the target highlight. The upper image shows the situation prior to deformation, and the lower image shows the results of deformation. The blue lines in the images are the surface highlights rendered visible. In this example, the highlight shown by the yellow line was deformed in order to follow the white line. Rendering the highlight visible shows that the highlight traces the specified white line on the surface following deformation.

4.4.2. Curvature control

Curvature control is a function that deforms existing curves by means of input of target focal curve for a

curvature plot in relation to the existing curves. In the previous system, the target curves were obtained by repeatedly adjusting the positions of the control points or pass points of the curves one-by-one. Based on the concept of target-driven modeling, in the new system the existing curves can be deformed to produce the target curvature plot simply by inputting the existing curves and the target curvature plot. Figure 15 shows that specifying a target curvature plot in relation to an existing curve has resulted in the achievement of a curve that follows the specified target.

4.4.3. Surface tension control

Surface tension control is a function that adjusts surfacing tension in relation to the existing surface. A curve is defined to help enable tension to be controlled on the surfaces which are being corrected. If the current tension of the surface is defined as 0, adjustments can be made in the range of $\pm 100\%$. By this means, the surfaces can be controlled when styling designers wish, for example, to apply slightly more tension. Figure 16 shows the results of applying tension increases of 10%, 30%, and 50%. Because the tendency of original input shapes is successfully kept well in the image as a whole, the difference is barely noticeable. When cross-sections at the curve defined for tension control are compared, however, it is clear that the tension of the surface has been increased to the degree specified.

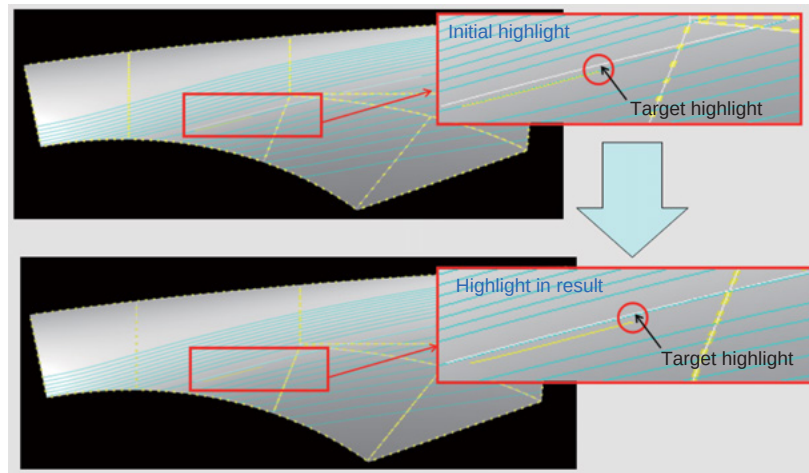


Fig. 14 Highlight control in sample case

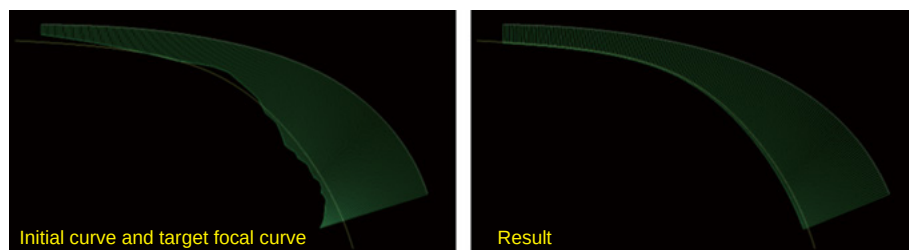


Fig. 15 Curvature control in sample case

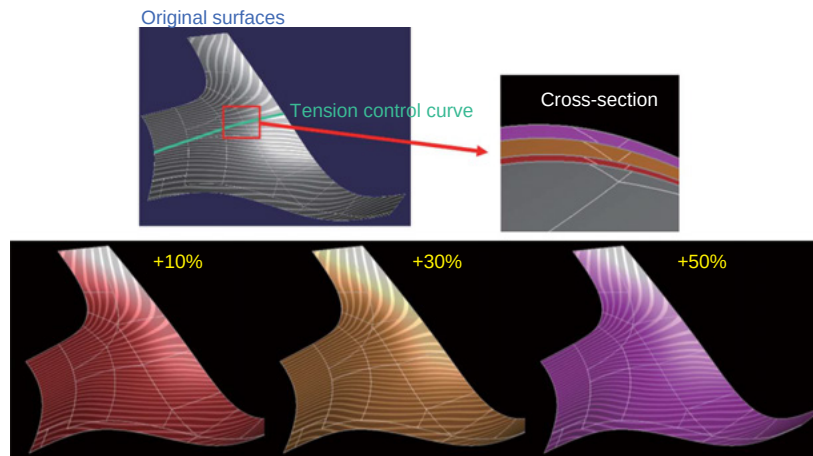


Fig. 16 Tension control in sample case

5. Effect of System in Reducing Man-hours and Analysis

The effect of the developed system in the process of product development to completion of the Class A data was evaluated in relation to the development target of the achievement of a 50% reduction in the time necessary for the creation of the data. The effect of individual functions was also analyzed.

5.1. Effect of System in Reducing Modeling Man-hours

Tables 1 and 2, respectively, show the time employed for the creation of Class A data for various parts and the reduction in modeling man-hours achieved through the use of the new system.

Tables 3 and 4, respectively, show the time employed for the creation of Class A data in different phases of styling development and the reduction in modeling man-hours achieved through the use of the new system.

While variations in the effect produced by the system of around 10% can be observed depending on vehicle

type, part, and styling development phase, it can be seen that the system has satisfied the development target.

5.2. Analysis of Effect in Reducing Modeling Man-hours

Table 5 shows the effects of the main functions incorporated in the new system, and Fig. 17 shows shapes output by the respective functions. The times in Table 5 represent the total calculation time necessitated by operations conducted until the completion of the shapes, and do not include the time necessary for shape evaluation or the selection of work methods by the system user.

A more-than-twofold difference was observed between the effect of the function that produced the greatest effect (No. 6) and that of the function that produced the least effect (No. 4), and the causes of this difference were analyzed. An effect of reduction of modeling man-hours satisfying the target established for the system was displayed by functions No. 1 and 2, for which entirely new calculation algorithms were employed, and functions No. 3, 5, and 6, for which new approaches not employed in the previous system were adopted. However, while

Table 1 Modeling time by parts (hours)

Production model	Recent model	Past model A	Past model B
Applied system	New system	Past system	Past system
Modeling time of parts L	235 h	823 h	340 h
Modeling time of parts M	134 h	188 h	440 h
Modeling time of parts N	125 h	225 h	169 h
Total modeling time	494 h	1236 h	949 h

Table 2 Reduction ratio by parts (%)

Model	vs. Past model A	vs. Past model B
Reduction of modeling time (total time)	60%	48%

$$\% = (\text{Past model X} - \text{Recent}) / \text{Past model X}$$

Table 3 Modeling time by styling phase (hours)

Production model	Recent model	Past model X	Past model Y	Past model Z	Ave.(X,Y,Z)
Applied system	New system	Past system	Past system	Past system	Past system
Styling phase P	92 h	190 h	129 h	182 h	167 h
Styling phase Q	202 h	823 h	340 h	230 h	464 h
Styling phase R	932 h	2204 h	1029 h	1453 h	1562 h

$$\text{Ave.}(X,Y,Z) = (X+Y+Z)/3$$

Table 4 Reduction ratio by styling phase (%)

Phase	Phase P	Phase Q	Phase R
Reduction of modeling time (average for phase)	45%	56%	40%

$$\% = (\text{Ave.}(X,Y,Z) - \text{Recent model})/\text{Ave.}(X,Y,Z)$$

Table 5 Analysis of effect by main functions implemented on new system

No.	Function	Model	Computing time* in past system	Computing time* in new system	Reduction of computing time
1	Fitting surface on measured mesh	Fig. 17 (a)	13 min.	3 min.	77%
2	Fillet surface (Class A)	Fig. 17 (b)	14 sec.	3 sec.	79%
3	Blending surface (Class A)	Fig. 17 (c)	18 sec.	2 sec.	89%
4	Matching surface (Class A)	Fig. 17 (d)	64 sec.	35 sec.	45%
5	Global deformation (Deformation)	Fig. 17 (e)	15 h	1.5 h	90%
6	Highlight control (Deformation)	Fig. 17 (f)	1 h	43 sec.	99%

* Computing time represents the total computing time for iterative operations.

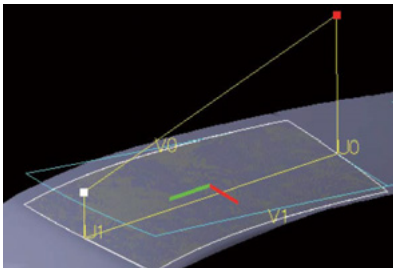


Fig. 17 (a) Fitting surface

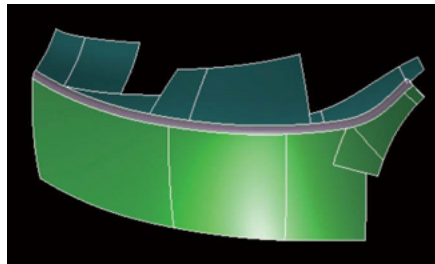


Fig. 17 (b) Fillet surface

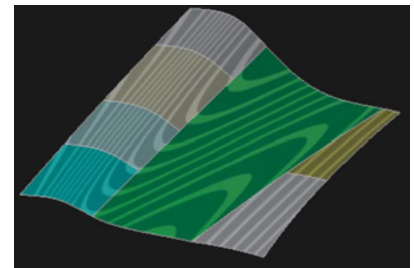


Fig. 17(c) Blending surface

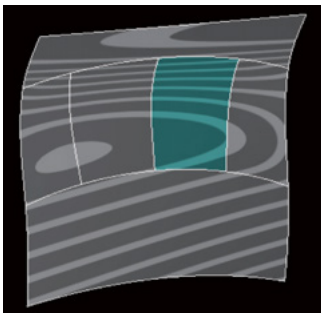


Fig. 17 (d) Matching surface

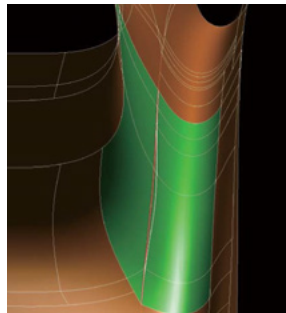


Fig. 17 (e) Global deformation

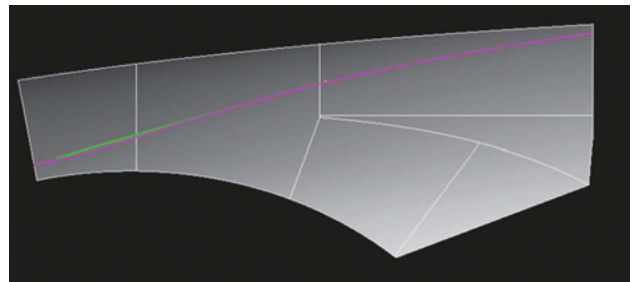


Fig. 17 (f) Highlight control

function No. 4 produced a reduction of 45% in modeling time, its effect is low in comparison with the other functions. This is because the matching surface function had already reached a high level of maturity in the previous system, and the modification of the method in the new system therefore did not result in a conspicuous change.

6. Conclusion

- (1) A new system was developed in order to reduce the man-hours employed in the creation of Class A data by 50%, and satisfied this target.
- (2) Class A data creation functions were developed for the new system on the basis of the creation of an original software platform.
- (3) The development of Class A data creation functions focused on three phases in the process of creating the data, and employed new and original technologies to realize the desired functions in each phase.

7. Afterword

Ambitious targets were realized in the development of the system discussed in this paper by bringing together system developers and the members of project teams involved in the creation of Class A data to squeeze their knowledge and hone potential solutions. The authors believe that the project produced another outcome in addition to increasing the efficiency of the creation of Class A data. This was the experience of combining the efforts of an entire team in a process of trial and error in order to resolve every issue encountered in the course of seeking to fulfill a challenging target. We believe, this must be a wonderful provision for the engineers who will continue to produce next-generation technologies into the future.

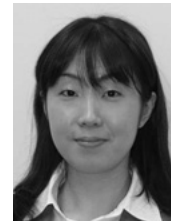
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Implemented Algorithms of Challenging CAD Technology aimed at Increasing Efficiency of Class A Surfacing

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ABSTRACT

In order to satisfy a development target for a 50% reduction in the man hours expended in the generation of Class A surfaces, the development of the system discussed in this paper focused on three phases of modeling leading to the completion of Class A surfaces. New technologies were developed to innovatively increase work efficiency in each area, and these were realized as CAD functions. This paper will discuss the technologies that helped to realize the functions that have had the greatest effect on performance.

First, for the generation of basic surfaces fitted over scanned clay model data, a technology was developed that defines surface behavior and quality as an optimization model based on hints from the stiffness of physical plates. The solution of the optimization model generates high-quality fitting surfaces.

Next, the computational algorithms for the generation of the fillet surfaces that represent approximately 10% of the number of Class A surfaces were fundamentally revised, resulting in a simultaneous increase in speed and accuracy.

For the final phase of Class A data generation, a target-driven modeling (TDM) function was realized that helps enable designers to make minor adjustments in accordance with their aesthetic judgment. This function has increased the efficiency of finishing work for Class A surfaces, which demands the expression of finely detailed design shapes.

1. Introduction

In order to reduce the man-hours necessary for the creation of Class A surface data by 50%, the project discussed in this paper focused on three phases of the modeling process up to the stage of completion of the Class A surfaces, and sought to develop CAD functions which would increase work efficiency in each phase⁽¹⁾. (A) to (C) below were established as the functional conditions necessary to increasing efficiency in the three work processes:

- (A) In the phase of creating fitting surfaces from scanned clay model data, the ability to efficiently create surfaces that satisfy Class A data quality standards while accurately reflecting the scanned data.
- (B) In the phase of creating surfaces that express complex styling shapes such as corners and aggregates, the ability to create high-quality surfaces with a high level of system responsiveness.

- (C) The achievement of greater efficiency in time-consuming work processes when attempting to ensure quality based on clear shape targets, for example when styling orientations are adjusted or a design is changed after Class A surfaces have been created.

In the absence of a technology that satisfied Honda's demands in terms of surface quality and computational performance, the development of technologies that would satisfy the functional conditions described above was necessary for achieving the targets set for the new system.

CAD functions satisfying the functional conditions outlined above were realized in the new system through a process of effective technological development. The new technologies employed in the system are described in (1) to (3) below. This paper will provide an introduction to these technologies, which helped to enable the realization of CAD functions.

- (1) Polygon computation was increased in speed in order to help enable interactive manipulation and referencing of

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high-volume scanned data. In addition, a technology was developed that helps to enable the creation of surfaces to be fitted on the scanned data based on hints provided by the stiffness of physical plates. This technology creates high-quality fitting surfaces by solving optimization model that define the behavior and quality of the surfaces.

- (2) The expertise incorporated in the Class A surfacing functions of the previous system were carried over and increased in speed and accuracy, while a command set extending to 185 functions was also developed. In particular, the calculation algorithms for fillet surfaces, which represent approximately 10% of the total number of Class A surfaces, were thoroughly revised to increase both speed and accuracy, helping to enable an increase in the speed of calculation for complex fillet shapes of 90% or more against the previous system.
- (3) For correction of Class A surface shapes, the time-consuming work process until completion of the shapes even if the shape targets are clear was increased in efficiency. Target-driven modeling (TDM), a new concept for CAD operations, was introduced and technologies were developed to realize the concept. The realization of TDM as a CAD function has simplified the modeling work involved in the process of correction of shapes.

2. Overview of Polygon Kernel Functions

Pre-processing and shape analysis of the polygon mesh is necessary in the process of creating Class A surface data from polygon mesh scanned data. Because it was necessary to manipulate data interactively despite the fact that the new system works with high-volume data containing several million vertices, it was necessary to create algorithms that would maximize the characteristics of the polygon kernel data structure and memory management mechanism⁽¹⁾. The main functional requirements and the issues to be solved to achieve them were as follows:

- (1) It was necessary to formulate sectional linear shapes in the mesh as polyline sections in order to help enable the model shapes formed from the scanned data to be checked visually, but the achievement of increased efficiency both in the calculation of points of intersection with cutting planes and in the creation of polylines for groups of points of intersection represented an issue.
- (2) It was necessary to project 3D curves on the polygon mesh in order to help enable selection of specific mesh regions and region-setting for surface generation. However, this represented a challenge when there were holes in the polygon mesh or multiple projected curves.
- (3) A function for filling holes in the polygon mesh due to defects in the scanned data or some other causes was necessary, but the high-speed generation of isotropic hole-filling patches irrespective of the shape or the size of the hole represented an issue.
- (4) It was necessary to incorporate a function that would

calculate the degree of divergence between the polygon mesh and the Class A surfaces in order to help enable users to identify differences between the shapes of the Class A surfaces and the scanned data. High-speed calculation of intersections between the lines and the polygon mesh would be needed for calculation of the degree of divergence.

This chapter will discuss the concrete methods employed to resolve issues (1) through (4) in order to help enable the desired polygon kernel functions to be realized.

2.1. Method of Generation of Polyline Sections

For this function, a two-stage processing method was developed in order to resolve technical issues to help enable efficient treatment of interconnection between contiguous points of intersection, as follows:

- (1) Determination of intersection between sectional plane and polygon mesh topological elements (edges and vertices); and
- (2) Search for contiguous points of intersection and connection processing.

For determining intersection, first the position relationships between the sectional plane and all the vertices are determined, following which intersection is judged for the sectional planes and all the edges. When the vertices are on the obverse side of the sectional planes, they are allocated a positive sign, and when they are on the reverse side they are allocated a negative sign. If the distance between a vertex and a sectional plane is within tolerance values, it is defined as an intersecting vertex. Next, intersecting edges are judged. Edges with vertices that have been allocated different signs are defined as intersecting with sectional planes, and the positions of the points of intersection on the edges are calculated. In Fig. 1, the dotted line represents a sectional plane, the black and white circles respectively represent the vertices on the obverse and reverse sides of the sectional plane, and the red circles represent the vertex intersecting with the sectional plane and the points of intersection on the edge.

During connection processing, an arbitrary intersecting edge or intersecting vertex is defined as the starting point,

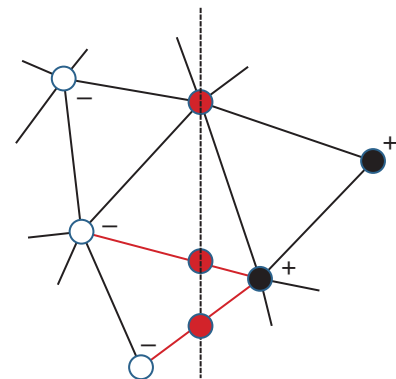


Fig. 1 Polyline-section algorithm

and contiguous intersecting topological elements are successively searched for. These are formed into a single array of points of intersection. In the case of intersecting edges, a search for contiguous intersecting topological elements is commenced from among all the topological elements classified with the faces on both sides, and in the case of intersecting vertices, a search is commenced from among all the topological elements classified with the faces contiguous to the intersecting vertices. One polyline section is completed when all the contiguous topological elements have been determined. The remaining intersecting topological elements are restructured in the same way, and the generation of polyline sections is completed.

2.2. Method of Projecting Curves

It is necessary to correctly output projection calculations even in cases in which the polylines that represent the results of the projections cross holes or the solution consists of multiple polylines. The method used in the new system, which employs the polyline sections discussed in the preceding section, is composed of the following five processing steps. Figure 2 shows an overview of the processing method.

- (1) The curve to be projected is divided into a tessellated polyline.
- (2) The sectional plane is defined from a segment of the tessellated polyline and the direction of projection.
- (3) The polyline of intersection between the sectional plane defined in step (2) and the polygon mesh is determined.
- (4) The projected part of the polyline of intersection that corresponds to the segment is extracted and defined as projected segment.
- (5) A search is conducted for contiguous section fragments by referencing the half-edge data structure; when all the projected segments found in step (4) have been interconnected, the isolation of the curve shape is complete.

2.3. Automatic Hole-filling Processing Method

It was vital to use an algorithm for hole-filling in order

to help ensure polygon quality in the hole-filling patches employed in hole profile shapes. However, performance declined in the case of large hole shapes. The following processing steps were adopted in order to resolve this issue.

- (1) Half-edges with no contralateral elements are detected, and the hole profile is obtained by clustering processing. Figure 3(a) shows the stage of extraction of the hole profile.
- (2) When the hole profile has less than 30 vertices, the Liepa hole-filling algorithm⁽²⁾ is run with conditions set to minimize the total area of the patch, and a hole-filling patch is obtained. Meanwhile, when the hole profile has 30 vertices or more, remeshing is conducted after an initial patch has been generated using the newly developed algorithms in steps (3) to (8).
- (3) The coordinates of the center of the hole profile are determined.
- (4) From among the vertices of the hole profile, new vertices are generated at the point close to the center by the average edge length of the profile polyline, and new edges are extended. At this time, when the distance between contiguous new vertices is lower than a threshold value, the vertices are fused into a single vertex.
- (5) Each contiguous half-edge is tessellated into either a strip shape or a fan shape, generating a band-shaped patch to fill the outer periphery of the hole.
- (6) Steps (1) to (5) are conducted until the small hole in the interior of this belt-shaped patch is smaller than 10 vertices. Figure 3(b) shows a second-generation belt-shaped patch.
- (7) When the size of the hole profile becomes less than 10 vertices, the initial patch is completed by filling the hole in the fan shape with an arbitrary vertex as the starting point. Figure 3(c) shows this stage.
- (8) Remesh processing is conducted for the initial hole-filling patch. In order to obtain an equivalent level of polygon accuracy for the hole-filling patch as for the surrounding shapes, in the remeshing operation, topological rearrangement is repeatedly applied so that the isotropic polygons, the target edge length of which corresponds to

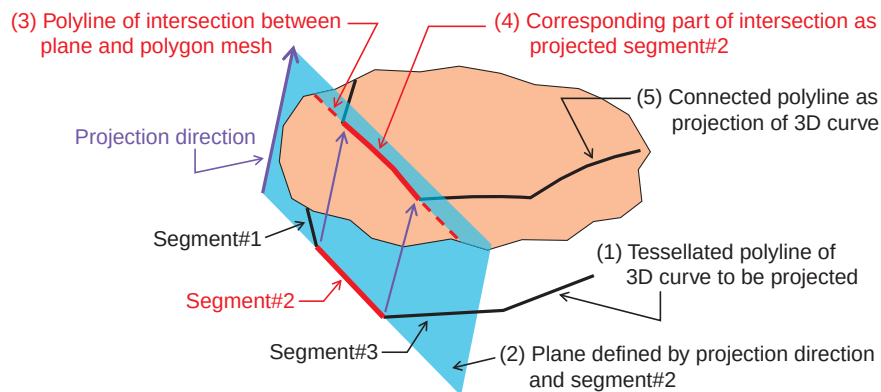


Fig. 2 Algorithm for 3D curve projection on polygon mesh

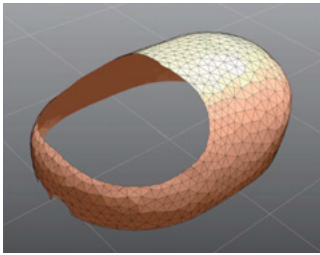


Fig. 3 (a) Hole detection

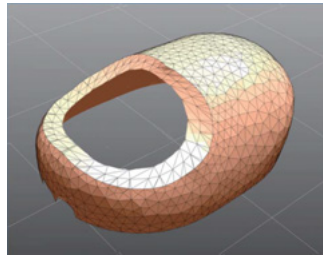


Fig. 3 (b) Initial patch creation

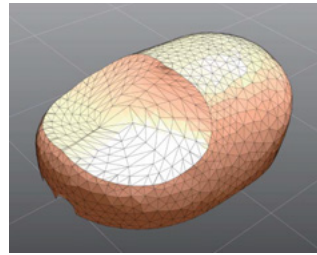


Fig. 3 (c) Completed initial patch

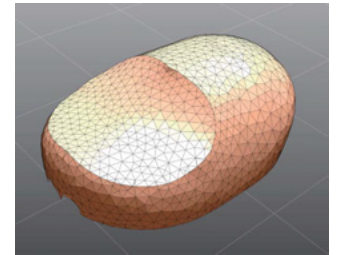


Fig. 3 (d) Hole filled with final patch

the average edge length of the hole profile, become dominant⁽³⁾. The initial patch created in steps (1) to (7) is formed from the initial stage using a set of polygons for which the target edge length has been taken into consideration. This has been of benefit in increasing the speed of remeshing. Figure 3(d) shows the result of completion of hole-filling processing.

2.4. Function for Measurement of Distance between Mesh and Geometric Elements

In order to increase the speed of calculation of intersections between rays and meshes, it was necessary to efficiently create space divisions. This function was realized by means of the following processing steps.

- (1) High-efficiency, high-speed space division
- (2) High-speed intersection calculation using the space divisions created in step (1)
 - Space divisions are created using the following processing steps.
 - (i) Taking an arbitrary polygon as the seed, a cubic region is defined within a fixed distance ($1/40^{\text{th}}$ of the longitudinal diameter of the target mesh) from the center of gravity to each axis.
 - (ii) All the polygons in this region, including the seed, are defined as a component polygon set that forms a grid.
 - (iii) An axis-aligned bounding box (AABB) is calculated for the component polygon set, and this is defined as a grid shape.

- (iv) Targeting polygons that are not classified as part of any grid, steps (i) to (iii) are repeated until all the polygons are classified into either of the grids.

Each grid has overlaps in terms of spatial arrangement, but there is no duplication of the polygon sets incorporated in the grids. That is, any particular polygon will only be assigned to one grid.

Intersection calculations are conducted in two stages. First, intersections with the grids are calculated, following which intersection is calculated for all the polygons inside the grids for which intersection has been determined.

Because all of the grids are created as AABB, judgment of intersections between rays and grids can be conducted at high speed. Figure 4(a) shows a grid aggregate for which intersections with a ray aggregate have been determined in intersection calculations for the target mesh and 10 rays from the exterior. The polygon aggregate inside the grid is shown as pink.

The fact that only this polygon set shown as pink is the focus for judgment of intersections between the rays and the polygons has contributed to reducing the number of necessary calculations, thus boosting performance. The following steps are used to judge intersections between rays and polygons.

- (i) Coordinate transformation is applied to the entire set of vertices of the mesh that is being processed in such a way that the ray and the X axis overlap each other.
- (ii) The polygons are excluded as candidates for intersection

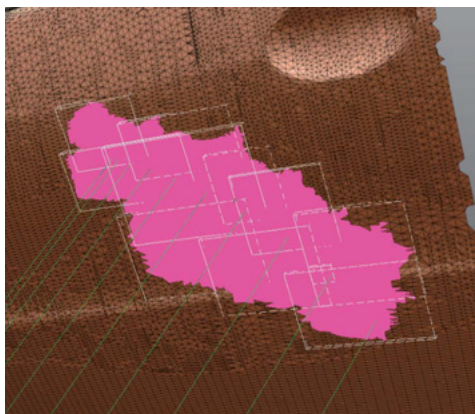


Fig. 4 (a) Ray-grid intersection

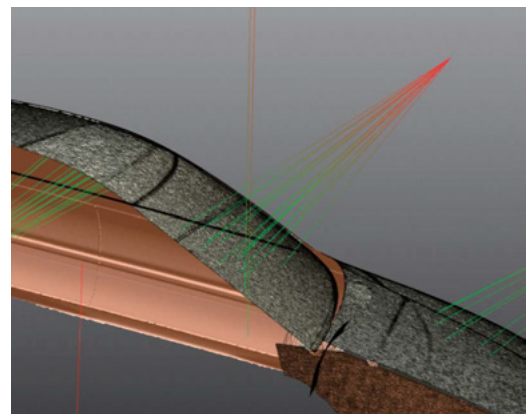


Fig. 4 (b) Ray-mesh intersection

if the signs of all the Y coordinates for three vertices are the same, or if the signs of all the Z coordinates for three vertices are the same.

- (iii) From among the polygons that pass the screening in (ii), all the polygons which intersect with the X axis are defined as intersecting polygons. Since the intersection determination has already been conducted, the calculation of the 3D coordinates of the points of intersection is a simple matter.

Figure 4(b) shows the results of judgment of intersections with multiple rays for the generation of the target mesh.

3. Fitting Technology

Class A data is generated with 3D scanned data from clay models as the input. Formerly, during this process the deviation or styling tendency between the Class A data and the scanned data was judged, and a process of manual correction of the surfaces was repeated. Because this process necessitated an enormous expenditure of man-hours, numerous systems for the automatic generation of surface shapes that reflect styling shapes from scanned data have been considered. This will be defined as “scanned data fitting technology.”

Because undulations in clay models and measurement errors were directly reflected in the output, the previous fitting technology was unable to ensure the necessary surface quality for Class A data. For the new system, the following technologies were therefore developed to remove clay model undulations and measurement errors while maintaining the styling shapes, thus generating surfaces that fulfill Class A data quality demands.

- (1) An intelligent fitting technology that helps to ensure the quality of individual surfaces
- (2) A merge technology that helps to ensure continuity between multiple surfaces

This chapter will discuss the methods employed to realize these technologies.

3.1. Intelligent Fitting Technology

In order to maintain both surface quality and continuity

between surfaces, it is necessary to control the tangency (G1) and curvature (G2) of the surfaces. In addition, as indicated above, undulations in the curvature affect quality, and because of this it is also necessary to control the rate of change of G2, termed G3.

In order to control these parameters, the concept of stiffness has been introduced in the new system. Stiffness here refers to the situation, for example, when a metal ruler is pressed over an actual model: the ruler follows the tendency of the surface, but the minute concavities and convexities are ignored. This is because the ruler has a specific degree of hardness, and will not bend more than a specific amount.

Figure 5 shows an image of the behavior of stiffness. The flesh-colored scanned data incorporates minute differences in level. Commencing with an arbitrary initial surface (shown as blue), surfaces are fitted to minimize the deviation from the scanned data, producing the surfaces shown as purple, which incorporate undulations. In the new system, constraint conditions (shown as the red arrows) are incorporated as the scope of variation in curvature (G3) within the range within which fitting is able to operate. This helps to enable the generation of an intelligent fitting surface (shown as red) which follows the scanned data while undulations are controlled.

Based on this idea, in addition to an approach using the least squares method, the new system employs conditional convergence calculations with the properties of the desired shapes as the conditions. The following objective function and constraint condition are defined for the convergence calculations.

Objective function: $(\text{Deviation from scanned data})^2 + (\text{Value of G3 at each point})$

Constraint condition: $(\text{Value of G3}) < (\text{Upper limit of stiffness})$

The value of G3 was used as a constraint condition because it was considered that it could function to control undulations in the same way as the stiffness of the ruler. An excessive G3 constraint would reduce the ability to trace scanned data, while an insufficient G3 constraint would be prone to produce undulations. Considering this as the hardness of a metal plate, the level of adjustment

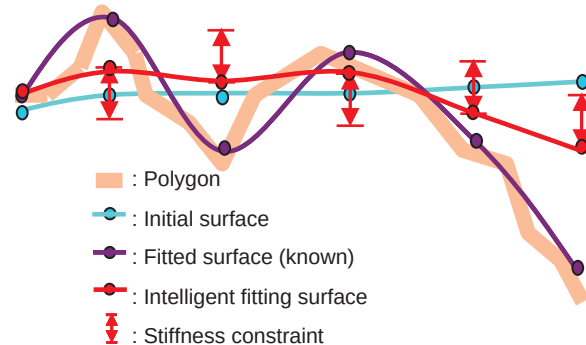
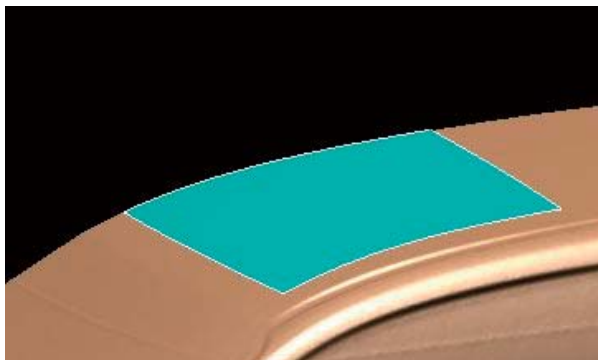


Fig. 5 Stiffness as constraint

of stiffness was applied as a constraint condition, which can be interpreted as altering the thickness of the metal plate. In addition, because undulations are promoted across the entire surface if the value of G3 is increased, this was incorporated in the objective function as an energy function for the convergence calculations.

The concrete methods of incorporating these considerations in the system will be described below.

The input and output of the system can be summarized as follows:

Inputs: 3D scanned data

Four boundary lines defining the area of a surface

Number of control points (CP) for NURBS surfaces

Magnitude of stiffness

Output: Surfaces satisfying shape quality demands

Based on the input data, an initial surface that serves as a standard is generated and fitted. An NURBS surface is used for the mathematical expression of the surface. Within the u and v spaces that can be defined from the input boundary lines, control points are spaced at regular intervals and an initial surface is defined. On the NURBS surface, the surface is expressed by multiplying the basis function $B(u, v)$ by the coordinate values of the control points (CP). The basis function is determined uniquely using the parameter space u, v and the junctures in the space (the knots on the NURBS surface). Now, the equation for the surface, $S(u, v)$ in the parameter space u, v can be expressed as follows using the n control points $a_0, a_1, \dots, a_n$:

$$S(u, v) = (B_0(u, v), B_1(u, v), \dots, B_n(u, v)) \begin{pmatrix} a_0 \\ a_1 \\ \vdots \\ a_n \end{pmatrix} = [B][a] \quad (1)$$

Here, the matrix of the basis function is given as $[B]$.

As Eq. (1) shows, the surfaces can be expressed as linear calculations. Because the number of regions in which surfaces are being fitted and the number of control points are known in advance, the unknown quantity to be determined for the generation of intelligent fitting surfaces is the control point sequence $a_0, a_1, \dots, a_n$.

Because $[B_0(u, v), B_1(u, v), \dots, B_n(u, v)]$ is given, finding the differential of the direction of u, v can be defined as an independent operation to $a_0, a_1, \dots, a_n$. This is defined as the n th-stage differentiation matrix in the u direction, $[D_u^n]$, and used in the calculation of G2 and G3 in Eqs. (9) and (19) below.

$$\partial S / \partial u = [D_u^n][B][a], \quad \partial^2 S / \partial u^2 = [D_u^n][D_u^n][B][a], \quad \partial^3 S / \partial u^3 = [D_u^n][D_u^n][D_u^n][B][a] \quad (2)$$

First, the objective function for the deviation from the measurement points, C_{Dist} is determined. The closest parameter point on the benchmark surface, $(u_0, v_0) \dots (u_m, v_m)$, is defined for the m measurement points $p_0, p_1, \dots, p_m$.

Then, because C_{Dist} is the square of the divergence between P and the parameter point,

$$C_{Dist} = \left[\begin{pmatrix} P_0 \\ P_1 \\ \vdots \\ P_m \end{pmatrix} - \begin{pmatrix} B_0(u_0, v_0) & B_1(u_0, v_0) & \dots & B_n(u_0, v_0) \\ B_0(u_1, v_1) & B_1(u_1, v_1) & \dots & B_n(u_1, v_1) \\ \vdots & \vdots & \ddots & \vdots \\ B_0(u_m, v_m) & B_1(u_m, v_m) & \dots & B_n(u_m, v_m) \end{pmatrix} \begin{pmatrix} a_0 \\ a_1 \\ \vdots \\ a_n \end{pmatrix} \right]^2 \quad (3)$$

$$= [P]^T[P] - 2[P][B]^T[a] + [a]^T[B]^T[B][a]$$

$$\therefore C_{Dist} = [f]^T[a] + \frac{1}{2}[a]^T[H][a] \quad (4)$$

$$[f] = -[B]^T[P]$$

$$\text{Linear component: } 1 \times n \text{ (dependent on } X) \quad (5)$$

$$[H] = [B]^T[B]$$

$$\text{Quadratic component: } n \times n \text{ (not dependent on } X) \quad (6)$$

Here, the $[P]^T[P]$ term is excluded because it is a constant, and C_{Dist} is the term used in convergence calculations.

The number of elements for f and H is not dependent on the number of scanned data, m , and is only dependent on the number of control points, n . Because of this, when Eqs. (5) and (6) have been run as initial calculations, optimization calculations can be conducted without the necessity for an enormous amount of scanned data, thus reducing the amount of computation.

In addition, the objective function for the value of G3, C_{Energy} can be defined as follows using the differentiation matrix.

$$C_{Energy} = \frac{\partial^3 S}{\partial u^3} + \frac{\partial^3 S}{\partial v^3} + \frac{\partial^3 S}{\partial u^2 \partial v} + \frac{\partial^3 S}{\partial u \partial v^2} \quad (7)$$

$$= [D_u^3][D_u^3][D_u^3][B][a] + [D_v^3][D_v^3][D_v^3][B][a] + [D_u^2][D_u^2][D_v][B][a] + [D_u][D_u][D_v^2][D_v^2][B][a]$$

The final objective function, C , is

$$C = C_{Dist} + \lambda C_{Energy} \quad \lambda: \text{Constant} \quad (8)$$

Next, the constraint conditions are determined. Now, the value of G3 in the uv direction is assumed to be greater than b_L and less than b_H . The constraint conditions for each point at this time can be expressed as follows, using the differentiation matrix:

$$b_L^{u3} < \frac{\partial^3 S}{\partial u^3} < b_H^{u3}, \quad b_L^{v3} < \frac{\partial^3 S}{\partial v^3} < b_H^{v3}, \quad b_L^{u2v1} < \frac{\partial^3 S}{\partial u^2 \partial v} < b_H^{u2v1}, \quad b_L^{u1v2} < \frac{\partial^3 S}{\partial u \partial v^2} < b_H^{u1v2} \quad (9)$$

Here, b^{u3} , b^{v3} , b^{u2v1} , and b^{u1v2} at both ends are the partially differentiated G3 values for each rank in the u, v space, and vary with the degree of stiffness specified. In concrete terms, the upper limit value of the G3 constraint is absolutely and quantitatively defined, and the lower

limit value of the G3 constraint is determined on the basis of how stiff the surface will be made in comparison to the results obtained using the upper limit value.

Linear programming is often used in convergence calculations for conditional linear matrices. However, in this case, because quadratic terms appeared in the objective function, extended quadratic programming was used for the quadratic expression. OOQP⁽⁴⁾ was used for the algorithms to solve the quadratic programming tasks.

As Eqs. (4) and (9) show, the objective function, C , and the constraint condition, A , can be expressed as a general expression as follows. Below, in accordance with the sign system used in OOQP⁽⁴⁾, the matrix expression $[X]$ has been simplified to X .

$$C = \frac{1}{2} a^T H a + f^T a \quad (10)$$

$$A a > b \quad (11)$$

a : Solution

H : Objective function (quadratic component)

f : Objective function (linear component)

A : Constraint condition (computational expression)

b : Constraint condition (value)

In addition, taking the G2 connection constraint conditions expressed in Eq. (19) below and other factors into consideration, constraint conditions with equal signs are defined as follows:

$$D a = d \quad (12)$$

D : Constraint condition (computational expression)

d : Constraint condition (value)

Satisfying the conditions of Eqs. (11) and (12), the minimal solution of Eq. (10), a , is found. When there are no constraint conditions,

$$dC/d\mathbf{a} = 0 \quad \therefore H\mathbf{a} = -f \quad (13)$$

for the optimal solution, $\mathbf{a}$. When constraint conditions exist, this is not always satisfied. Therefore, taking into consideration new unknowns for the respective constraint conditions, y , z , and s , L is defined using the equation below. Here, because s and z form an inequality, they are given positive values.

$$L = \frac{1}{2} a^T H a + f^T a - y(Da - d) - z(Aa - s - b) \quad (14)$$

$$s \geq 0 \perp z \geq 0$$

Here, the first and second terms express the objective functions, the third term expresses the equality constraint condition, and the fourth term expresses the inequality

constraint condition. At this time, the following conditional expressions are formed using the optimal solution, $\mathbf{a}$.

$$dL/d\mathbf{a} = 0 \quad \therefore H\mathbf{a} - D^T y - A^T z = -f \quad (15)$$

$$dL/dy = 0 \quad \therefore A^T \mathbf{a} = b \quad (16)$$

$$dL/dz = 0 \quad \therefore C^T z - s = d \quad (17)$$

$$dL/ds = 0 \quad \therefore sz = 0 \quad (18)$$

In this method, these conditional expressions are solved using the Newton method in order to derive $\mathbf{a}$.

3.2. Merge Technology

Completed Class A data is made up of multiple surfaces which display continuity of position, G0, tangency, G1, and curvature, G2. The target for the new system was to produce this continuity automatically.

Using the NURBS equation, the unique definition of numbers in the terminal section helps to enable continuity to be obtained as desired. However, this method produces prominent disturbance in the interior of the surface, and is unsuitable for Class A surfaces. In the new system, therefore, constraint conditions for continuity between surfaces were set in addition to constraint conditions for intelligent fitting surfaces.

The objective function defined in Eq. (8) is not limited to single surfaces, but can also be used in calculations for multiple surfaces.

The simultaneous calculation of the control points for surface 1, $a_0, a_1 \dots a_n$, and the control points for surface 2, $b_0, b_1 \dots b_n$, will now be considered. To help ensure G2 continuity in the u direction for the points defined by (u_0, v_0) on surface 1 and the control points (u_1, v_1) on surface 2, the following should hold:

$$\frac{\partial^2 S_1}{\partial u_1^2} - \frac{\partial^2 S_2}{\partial u_2^2} = 0 \quad (19)$$

Here, by adding the conditional expression Eq. (19) to the intelligent fitting surfaces obtained from Eq. (8) with $a_0, a_1 \dots a_n, b_0, b_1 \dots b_n$ as the unknowns, it is possible to obtain two surfaces with continuity of curvature at their boundaries.

4. Technologies for Increased Speed and Accuracy of Fillet Surfacing

The shapes used in recent automotive styling have increased in complexity, and the number of surfaces

employed has approximately doubled against five years previously, making an increase in the number of man-hours involved in the creation of Class A data an issue. Because the accuracy of surface data is important in Class A data in addition to the reproduction of styling shapes, it is necessary to balance increased speed and increased accuracy in order to reduce the number of man-hours involved in its creation.

As Fig. 6 shows, a fillet is a shape that rounds the angle formed by two sets of surfaces. In automotive styling, fillet surfaces represent approximately 10% of the total number of Class A surfaces. The new system set out to increase the speed of computational processing of fillet surfaces and the accuracy of fillet surface quality.

At Honda, the fillets that obtain G2 continuity with support surfaces are mainly used. To create such fillets, first a group of curves called ribs that obtain G2 continuity with the support surface is calculated, as shown in Fig. 7. And then the fillet surface is generated using these ribs. In the new system, increased speed was achieved through the modification of the ribs generation method, and accuracy was increased through the modification of the surfacing method.

4.1. Technology for Increased Speed

As Fig. 8 shows, the ribs that obtain G2 continuity with the support surfaces are generated as fifth-degree 1-segment B-spline curves with first-order and second-order

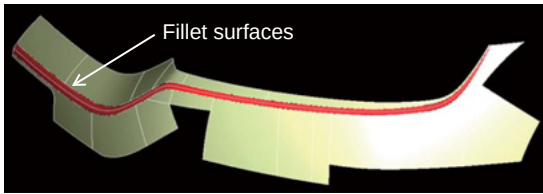


Fig. 6 Sample of fillet

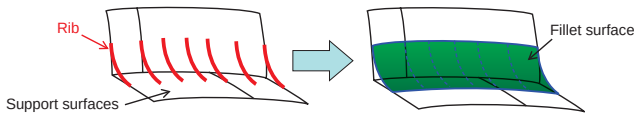


Fig. 7 Sample of rib

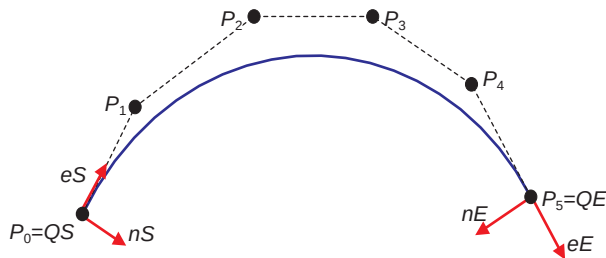


Fig. 8 Control points of 5 degree rib curve

differential vectors as input conditions at both ends. The first-order differential vectors express tangency, and the second-order differential vectors express curvature.

In the previous method, differentials were obtained from the intersection curves between the normal sectional planes of spine curve and the support surfaces in order to find the tangency and curvature of the end points. Because the calculations for the intersections were convergence calculations, this presented the issue of a high calculation time for high-quality fillet surfaces. For the new system, a method was developed for high-speed calculation of the curvature of the rib end points without calculation of intersection, using the surface unit normal vectors and the sectional plane unit normal vectors of the rib end points on the support surface and the normal curvature of the support surface.

For the curve $C(t)$, the unit tangent vector is termed $e(t)$, the unit principal normal vector is termed $n(t)$, and the curvature function is termed $\kappa(t)$.

With $0 \leq t \leq 1$ as a parameter condition and P_0, P_1, P_2, P_3, P_4 , and P_5 as the control points, a fifth-degree Bezier curve, $C(t)$, is expressed as follows:

$$C(t) = (1-t)^5 P_0 + 5(1-t)^4 t P_1 + 10(1-t)^3 t^2 P_2 + 10(1-t)^2 t^3 P_3 + 5(1-t) t^4 P_4 + t^5 P_5 \quad (20)$$

The first- and second-order differential vectors for the curve are expressed as:

$$\Delta P_i = P_{i+1} - P_i \quad (21)$$

$$\Delta^2 P_i = \Delta(\Delta P_i) = \Delta P_{i+1} - \Delta P_i = P_{i+2} - 2P_{i+1} + P_i \quad (22)$$

Using Eqs. (21) and (22), the following expressions can be obtained:

$$\frac{dC}{dt} = 5 \{ (1-t)^4 \Delta P_0 + 4(1-t)^3 t \Delta P_1 + 6(1-t)^2 t^2 \Delta P_2 + 4(1-t) t^3 \Delta P_3 + t^4 \Delta P_4 \} \quad (23)$$

$$\frac{d^2 C}{dt^2} = 20 \{ (1-t)^3 \Delta^2 P_0 + 3(1-t)^2 t \Delta^2 P_1 + 3(1-t) t^2 \Delta^2 P_2 + t^3 \Delta^2 P_3 \} \quad (24)$$

The first- and second-order differential vectors of the start and end points are:

$$\left. \frac{dC}{dt} \right|_{t=0} = 5(\Delta P_0), \quad \left. \frac{dC}{dt} \right|_{t=1} = 5(\Delta P_4) \quad (25)$$

$$\left. \frac{d^2 C}{dt^2} \right|_{t=0} = 20(\Delta P_1 - \Delta P_0), \quad \left. \frac{d^2 C}{dt^2} \right|_{t=1} = 20(\Delta P_4 - \Delta P_3) \quad (26)$$

The first-order differential vector is calculated from the tangent planes at the end points of the ribs on the surface and the normal sectional plane of the spine curve, and P_0 and P_4 in Eq. (25) are determinant.

A Frenet equation⁽⁵⁾ calculated using the arc length parameter expresses the fact that the second-order

differential vector and the first-order differential vector are orthogonal to each other⁽⁶⁾. The magnitude of the second-order vector is determined using the method discussed below. P_2 and P_3 in Eq. (26) are determinant.

Figure 9 shows the method of calculation of the curvature at the end point of a rib.

Using the following definitions,

n : Unit normal vector of support surface at end point of rib

m : Unit normal vector of sectional plane

e : Rib unit tangent vector at end point of rib (calculated from intersection of tangent plane and sectional plane)

α : Angle formed by n and m

κ_n : Normal curvature in e direction at end point of rib

κ : Curvature of cross-section at end point of rib

κ can be calculated using the following equation without explicit determination of the intersection curves:

$$\kappa = \left| \frac{\kappa_n}{\sin \alpha} \right| \quad (27)$$

However, when $\alpha \equiv 0$, the rib cannot be determined because the sectional plane and the tangent plane are identical.

The normal curvature at points on the surface (Fig. 10) is calculated using the following method.

The surface equation is defined as $S(u, v)$ and the unit normal vector is defined as $n(u, v)$. The introduction of e , the unit tangent vector at point p on the surface, determines the sectional plane that includes

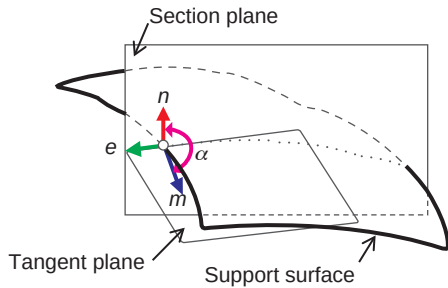


Fig. 9 Method of calculation of curvature at end point of rib

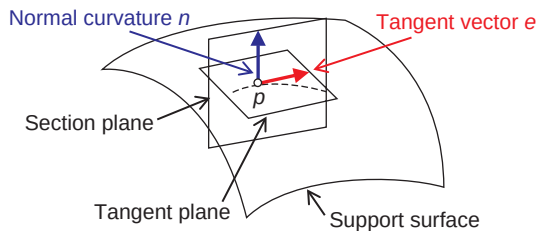


Fig. 10 Normal curvature of surface

e and n . The surface section in this plane is found, and the curvature of the section at point p (considered to be allocated a positive sign when the center of the curvature is on the n side, and a negative sign when the center of the curvature is on the side opposite the n side) is termed the normal curvature at point p on the surface in the e direction. If the tangent vector, e , is made to describe a circuit on the tangent plane around, the normal curvature generally also changes. The maximum and minimum values of the normal curvature are termed the principal curvature. Because the tangent vectors that impart principal curvature are orthogonal to each other, they are termed the principal direction. These calculations are conducted as follows. $\langle \cdot, \cdot \rangle$ represents the inner product of the vectors, and the partial differential at the parameters u, v is expressed as $S_u, S_v, S_{uu}, S_{uv}, S_{vv}$. First, coefficients of the first fundamental form

$$E = \langle S_u, S_u \rangle, F = \langle S_u, S_v \rangle, G = \langle S_v, S_v \rangle \quad (28)$$

and coefficients of the second fundamental form

$$L = \langle S_{uu}, n \rangle, M = \langle S_{uv}, n \rangle, N = \langle S_{vv}, n \rangle \quad (29)$$

are calculated for the position of the point on the surface at which normal curvature will be calculated.

The unit tangent vector, e , is expressed as

$$e = \xi S_u + \eta S_v \quad (30)$$

However,

$$\langle e, e \rangle = \langle \xi S_u + \eta S_v, \xi S_u + \eta S_v \rangle = E\xi^2 + 2F\xi\eta + G\eta^2 = 1 \quad (31)$$

If the principal curvature is termed κ_j , and the corresponding principal direction is termed $e_j = \xi_j S_u + \eta_j S_v$ ($j = 1, 2$), then the principal curvature and principal direction are resolved into the following equation for the determination of eigenvalues and eigenvectors.

$$\frac{1}{EG-F^2} \begin{bmatrix} G & -F \\ -F & E \end{bmatrix} \begin{bmatrix} L & M \\ M & N \end{bmatrix} \begin{bmatrix} \xi_j \\ \eta_j \end{bmatrix} = \kappa_j \begin{bmatrix} \xi_j \\ \eta_j \end{bmatrix} \quad (32)$$

Solving this eigenvalue equation is taken as giving the principal directions and the principal curvature. Because the two principal directions are the orthonormal base of the tangent plane, they can be expressed as

$$e = (\cos \theta) e_1 + (\sin \theta) e_2 \quad (33)$$

At this time, the following Euler formula is generated between the normal curvature in the e direction, κ_n , and the principal curvature.

$$\kappa_n = (\cos \theta)^2 \kappa_1 + (\sin \theta)^2 \kappa_2 \quad (34)$$

Using this formula, the normal curvature in the e direction, κ_n , can be calculated. Finding the sectional curvature at the end points of the ribs from Eq. (27) and finding the fifth-degree 1-segment B-spline curve makes it possible to determine the ribs at high speeds without intersection calculations.

4.2. Technology for Achievement of High Level of Accuracy

Using the previous technology, continuity between the intersection curves of normal sectional planes and support surfaces and intersection curves of normal sectional planes and fillets was sought in order to achieve continuity between support surfaces and fillet surfaces. Because the intersection calculations were convergence calculations, despite the fact that considerable time was expended on the calculations, it was not possible to ensure continuity other than at the position of intersections. In the new system, G2 continuity between support surfaces and fillet surfaces has been realized through the employment of phased Gordon surfacing using cross-boundary derivatives (CBD) (Fig. 11), which provide information concerning the differential values of peripheral surfaces.

Fillet surfaces are generated in the following phases.

- (1) In order to achieve G1 continuity with support surfaces, the local surface produced by the end rib $U_{0,0}(u)$ and two proximate ribs, $U_{0,1}(u)$ and $U_{0,2}(u)$, is focused on; the first-order CBD curve is found using the partial differential in the v direction, and twist vector matching termed global deformation⁽⁷⁾ is applied based on the tangent vector of the support surface.
- (2) In order to achieve G2 continuity with support surfaces, the local surface produced by the end rib, $U_{0,0}(u)$, one reference rib on the inner part of the surface, $U_{0,1}(u)$, and the previously determined first-order CBD curve is focused on, and the second-order CBD curve is determined from the second-order partial differential.
- (3) A G0 Gordon surface is generated based on the first-order CBD curve and second-order CBD curve, without consideration of continuity⁽⁷⁾.

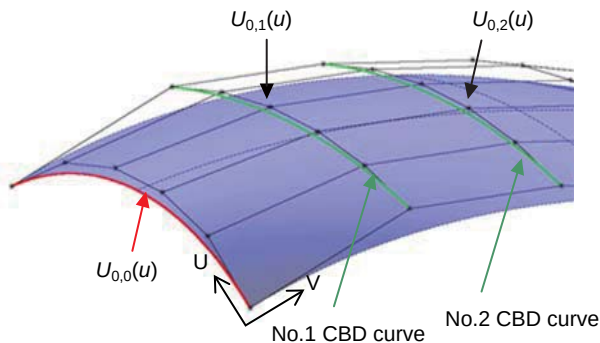


Fig. 11 Cross-boundary derivative curves No.1 and No.2 at $U_0(u)$

- (4) A first-order CBD curve is again obtained from the G0 Gordon surface, and global deformation is again applied in order to achieve G2 continuity with the support surface.
- (5) A Gordon surface is generated from the first-order CBD curve, the second-order CBD curve, the first-order CBD curve for the G0 Gordon surface, the R tangent curve, and the ribs, then the Gordon surface becomes the output.

4.2.1. Determination of first-order CBD following end rib

In the case of the starting end rib, the second-order Lagrange interpolating polynomial $\xi_{0,s}(v)$, $\xi_{1,s}(v)$, $\xi_{2,s}(v)$, which passes a total of three ribs, the starting end rib, $U_{0,0}(u)$, and the two proximate end ribs for reference, $U_{0,1}(u)$ and $U_{0,2}(u)$, is focused on.

From the local surface derived from

$$T(u,v) = U_{0,0}(u)\xi_{0,s}(v) + U_{0,1}(u)\xi_{1,s}(v) + U_{0,2}(u)\xi_{2,s}(v) \quad (35)$$

the starting end CBD is calculated:

$$\frac{\partial T}{\partial v}(u, b_0) = U_{0,0}(u)\xi'_{0,s}(b_0) + U_{0,1}(u)\xi'_{1,s}(b_0) + U_{0,2}(u)\xi'_{2,s}(b_0) \quad (36)$$

At this time, the corner twist vector is determined from the support surface, and twist matching of the tangent vector of the end point of the first-order CBD control curve is necessary. This is termed global deformation of the CBD control curve. Because G2 twist matching with the support surface at the end point of the end rib is considered in the new system, global deformation is applied for twist matching of the first-order CBD control curves and to transform it into a twist vector⁽⁷⁾.

A first-order CBD control curve is calculated in the same way for the finishing end rib, but in this case it is differentiated from the direction of the finishing end.

4.2.2. Determination of second-order CBD following end rib

A method of interpolation with 1 X 2 degree Coons surface will be formulated using the end rib, an inside rib of the end rib, and the first-order CBD function along the end rib. A second-order CBD function is calculated from this Coons surface, $T(u, v)$. The Coons surface, $T(u, v)$, can be expressed as follows:

$$T(u,v) = T_1(u,v) + T_2(u,v) - T_3(u,v) \quad (37)$$

Defining these terms as follows

$$T_1(u,v) = [U_0(u), D_s(u), U_1(u)] \begin{bmatrix} \lambda_0(v) \\ \lambda_1(v) \\ \lambda_2(v) \end{bmatrix} \quad (38)$$

$$T_2(u,v) = [\mu_0(u), \mu_1(u)] \begin{bmatrix} V_0(v) \\ V_1(v) \end{bmatrix} \quad (39)$$

$$T_3(u,v)=[\mu_0(u), \mu_1(u)] \begin{bmatrix} U_0(a_0), D_s(a_0), U_1(a_0) \\ U_0(a_m), D_s(a_m), U_1(a_m) \end{bmatrix} \begin{bmatrix} \lambda_0(v) \\ \lambda_1(v) \\ \lambda_2(v) \end{bmatrix} \quad (40)$$

with $\mu_j(u)$ as the interpolation function in the u direction, $\lambda_j(v)$ as the interpolation function in the v direction, $D_s(u)$ as the first-order CBD control curve on the starting end side, u as the interval $[a_0, a_m]$, and v as the interval $[b_0, b_n]$, and determining $\frac{\partial^2 T}{\partial v^2}(u, b_0)$, produces a second-order CBD function.

A second-order CBD control curve is calculated in the same way for the finishing end rib, but in this case it is differentiated from the direction of the finishing end.

4.2.3. Generation of G0 Gordon surfaces

Gordon surfaces are generated from the R tangent curve, the ribs, and the first-order and second-order CBD functions following the end rib without consideration of continuity with the support surfaces⁽⁷⁾. At this time, the interpolation function in the v direction is considered as a fifth-degree C4 spline function.

$$\tilde{T}(u,v) = \tilde{T}_1(u,v) + \tilde{T}_2(u,v) - \tilde{T}_3(u,v) \quad (41)$$

However, with $D_e(u)$ as the first-order CBD control curve on the finishing end side, $E_s(u)$ as the second-order CBD control curve on the starting end side, and $E_e(u)$ as the second-order CBD control curve for the finishing end,

$$\tilde{T}_1(u,v) = [U_0(u), D_s(u), E_s(u), U_1(u), U_2(u), \dots, U_{n-2}(u), U_{n-1}(u), E_s(u), D_s(u), U_n(u)] \begin{bmatrix} \lambda_0(v) \\ \lambda_1(v) \\ \vdots \\ \lambda_{n+1}(v) \end{bmatrix} \quad (42)$$

$$\tilde{T}_2(u,b_0) = [\mu_0(u), \mu_1(u)] \begin{bmatrix} V_0(v) \\ V_1(v) \end{bmatrix} \quad (43)$$

$$\tilde{T}_3(u,v) = [\mu_0(u), \mu_1(u)] \begin{bmatrix} U_0(a_0), D(a_0), E(a_0), U_1(a_0), U_2(a_0), \dots, U_{n-2}(a_0), U_{n-1}(a_0), E(a_0), D(a_0), U_n(a_0) \\ U_0(a_m), D(a_m), E(a_m), U_1(a_m), U_2(a_m), \dots, U_{n-2}(a_m), U_{n-1}(a_m), E(a_m), D(a_m), U_n(a_m) \end{bmatrix} \begin{bmatrix} \lambda_0(v) \\ \lambda_1(v) \\ \vdots \\ \lambda_{n+1}(v) \end{bmatrix} \quad (44)$$

4.2.4. Determination of first-order CBD following each internal rib from the G0 Gordon surface

First-order CBD control curves for the internal ribs are determined from the G0 Gordon surface, and global deformation is applied in order to achieve G2 continuity with the support surfaces.

4.2.5. Generation of G1 Gordon surfaces

Using fifth-degree C3 interpolation, Gordon surfaces are generated from the first-order CBD functions following the internal ribs which have undergone correction as described above, the R tangent curve, the ribs, and the first- and second-order CBD functions following the end rib. Expressed as $\tilde{T}(u, v)$, these surfaces

are termed G1 Gordon surfaces, and are treated as output surfaces.

$$\tilde{T}(u,v) = \tilde{T}_1(u,v) + \tilde{T}_2(u,v) - \tilde{T}_3(u,v) \quad (45)$$

However, with $D_i(u)$ as the first-order CBD control curves following the internal ribs,

$$\tilde{T}_1(u,v) = [U_0(u), D_0(u), E_0(u), U_1(u), D_1(u), U_2(u), D_2(u), \dots, U_{n-2}(u), D_{n-2}(u), U_{n-1}(u), E_n(u), D_n(u), U_n(u)] \begin{bmatrix} \lambda_0(v) \\ \lambda_1(v) \\ \vdots \\ \lambda_{n+1}(v) \end{bmatrix} \quad (46)$$

$$\tilde{T}_2(u,b_0) = [\mu_0(u), \mu_1(u)] \begin{bmatrix} V_{10}(v) \\ V_{11}(v) \end{bmatrix} \quad (47)$$

$$\tilde{T}_3(u,v) = [\mu_0(u), \mu_1(u)] \begin{bmatrix} U_0(a_0), D(a_0), E(a_0), U_1(a_0), U_2(a_0), D(a_0), \dots, U_{n-2}(a_0), D_{n-2}(a_0), U_{n-1}(a_0), E(a_0), D(a_0), U_n(a_0) \\ U_0(a_m), D(a_m), E(a_m), U_1(a_m), U_2(a_m), D(a_m), \dots, U_{n-2}(a_m), D_{n-2}(a_m), U_{n-1}(a_m), E(a_m), D(a_m), U_n(a_m) \end{bmatrix} \begin{bmatrix} \lambda_0(v) \\ \lambda_1(v) \\ \vdots \\ \lambda_{n+1}(v) \end{bmatrix} \quad (48)$$

5. Global Deformation Technology

The previous method of recreating Class A surface data from the beginning, when clay models were updated frequently in the quest for high-quality styling designs, meant that changes to the data could not keep pace with design changes. In order to increase the efficiency of the generation of Class A data, the existing approach to deforming and correcting Class A surfaces was focused on.

In the previous method of correcting surfaces, obtaining a target shape involved, first, a repeated process of adjusting the mathematical parameters of all the individual surfaces and expressing the desired shapes, following which it was necessary to make fine adjustments to the surfaces in order to connect them with continuity of curvature. These were time-consuming procedures. In the new system, the use of a method of correction of surface shapes that makes it possible to directly input shape targets based on the aesthetic judgment of styling designers has helped to enable the achievement of a more intuitive and efficient data correction process. This is termed “target-driven modeling.” Technologies were developed that make it possible to input a larger number of surfaces at the same time while preserving existing continuity, thus outputting a group of surfaces that displayed continuity. Highlight control and curvature control will be discussed here as examples of these technologies.

5.1. Highlight Control

Highlight control applies the concept of target-driven modeling to two-dimensional highlights, one of the standards employed in the evaluation of the quality of

Class A data. This function makes it possible, by specifying a target highlight for an existing group of surfaces, to deform the surfaces into a shape that will produce the target highlight. The upper panel in Fig. 13 shows the input surface group and the target highlight intended by the styling designer, and the lower panel shows the output result. The blue lines in the images are the highlights on the surface rendered visible. In this example, the surfaces were deformed in order to make the highlight shown by the yellow line move into the position specified by the white line.

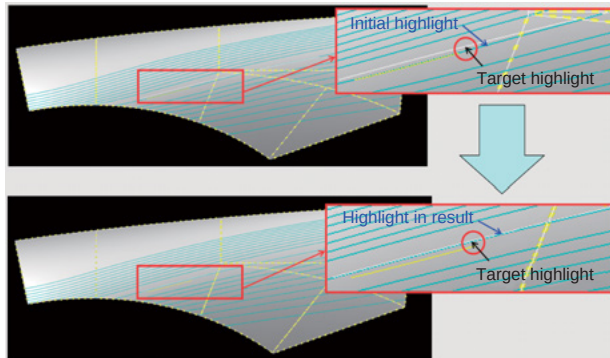


Fig. 13 Input and result of highlight control

5.1.1. Two-dimensional highlights

Figure 14 shows a representation of the definition of a two-dimensional highlight. A two-dimensional highlight is a highlight generated on a surface for evaluation of the styling shape of the surface. A group of planes in a fixed direction are defined for the surface to be evaluated, and a line intersecting these planes is generated. The points at which the tangent vectors of the intersecting line display the same orientation are isolated and connected. This is defined as a curve on the surface to be obtained.

5.1.2. Highlight control method

As indicated in the preceding section, highlights are generated by the connection of points on a surface that have the same tangent direction. Given this, controlling a highlight on a surface involves moving the highlight from the existing highlight line to the target highlight line while preserving the tangent vector. However, as Fig. 15 shows, when the tangent vector of the highlight line is preserved with the existing continuity at the edges of the surface fixed, the surface shapes around the highlight represent the opposite tendency due to over-constraints on the surface operations.

In order to resolve this issue, the following degrees of freedom of surface deformation were introduced in addition to the basic concept of preserving the tangent

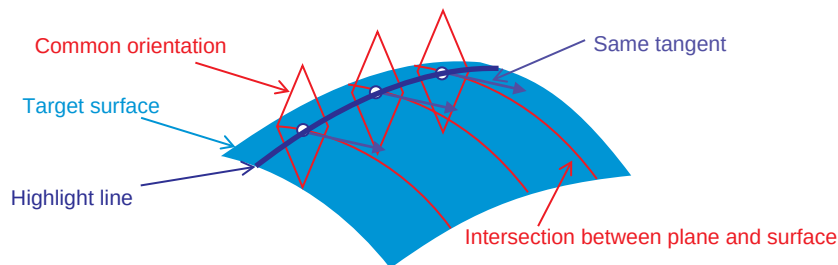


Fig. 14 Definition of 2D highlight

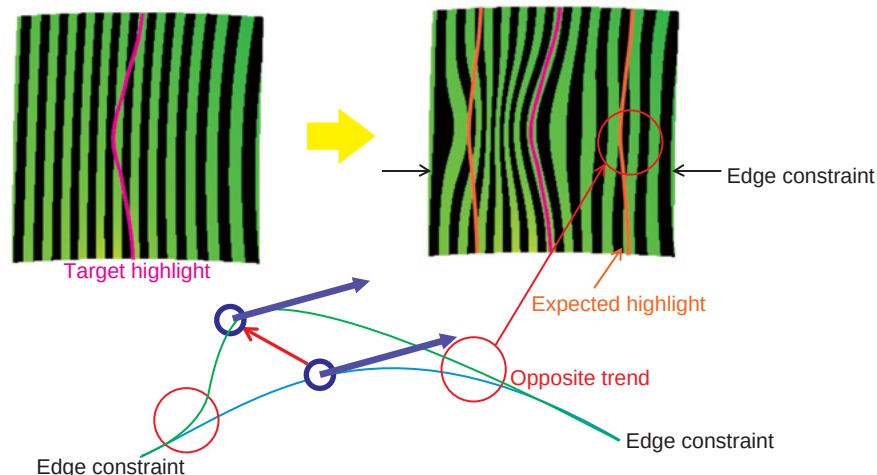


Fig. 15 Issue from over-constraints

vector of the highlight line:

- (1) Provision of degree of freedom to normal direction of surface while preserving tangent vector
- (2) Provision of degree of freedom of position in relation to direction of view in highlight operations

This is shown in Fig. 16. Because the input target highlight line in this function specifies the transformation of the trend of the surface shape, the degrees of freedom described above make it possible to realize the target highlight without any negative effects. The use of this method has resolved the technological issues associated with highlight control.

5.2. Curvature Control

Curvature control is a function that makes it possible to deform an existing curve shape by inputting a target line for a curvature plot to modify an existing curve. In the previous system, the control points and pass points of the curve were adjusted one by one in a repeated process in order to obtain the target curve. As Fig. 17 shows, the purpose of the curvature control function is, based on the concept of target-driven modeling, to help enable the direct deformation of the curve shape to obtain the target curvature plot by inputting the existing curve, C , and the target curvature plot, TF .

5.2.1. Curvature control method

Assuming, for an arbitrary point P on curve C in Fig. 17, that the curvature rate $\underline{\kappa}$ that satisfies the target curvature plot TF occurs at the point of intersection between the normal vector of point P and TF , termed Pf ,

$$\underline{\kappa} = (P - Pf) / \text{Drawing scale of curvature plot} \quad (49)$$

Expressing this for the entire curve using parameter s for curve C ,

$$\underline{\kappa}(s) = (C(s) - TF) / \text{Drawing scale of curvature plot} \quad (50)$$

Terming the starting point of the initial curve C P_0 and the tangent vector of the starting point T_0 , the curve CT , which satisfies the target curvature plot, can be found using

$$\theta(s) = \text{Angle}[T_0] + \int_{[0,s]} \underline{\kappa}(u) \frac{du}{ds} ds \quad (51)$$

$$CT(s) = P_0 + \int_{[0,s]} \{\cos[\theta(u)], \sin[\theta(u)]\} \frac{du}{ds} ds \quad (52)$$

5.2.2. Constraint conditions at start and end points

However, using Eq. (52), it is not possible to specify the position of the end point of the resulting curve CT . In Class A modeling, it is important to preserve not only the position of start and end points of an existing curve, but also their tangent vectors and curvatures.

For example, terming the curvature of the resulting curve CT κ_c , in order to preserve the tangent vectors of the start and end points of the curve it is necessary to satisfy the condition

$$\int_0^u \kappa_c(u) du = \int_0^u \underline{\kappa}(u) du \quad (53)$$

In addition, in order to preserve the curvature of the start and end points of the curve it is necessary to satisfy the conditions

$$\kappa_c(u_a) = \underline{\kappa}(u_a) \quad (54)$$

$$\kappa_c(u_b) = \underline{\kappa}(u_b) \quad (55)$$

Figure 18 conceptualizes Eqs. (53) and (55). In order to preserve the tangent vectors of the start and end points

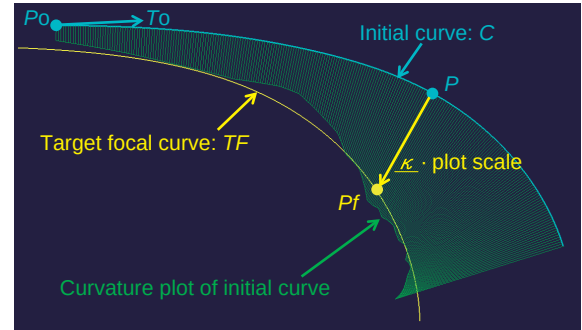


Fig. 17 Input for curvature control

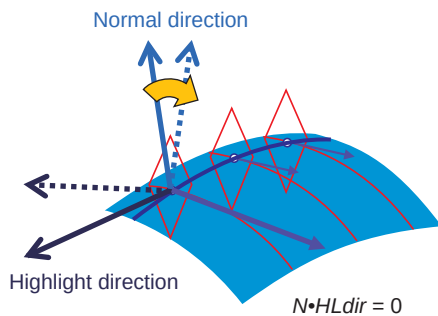
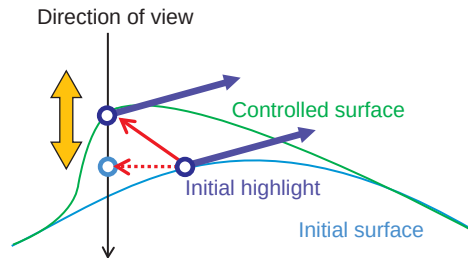


Fig. 16 With given freedom



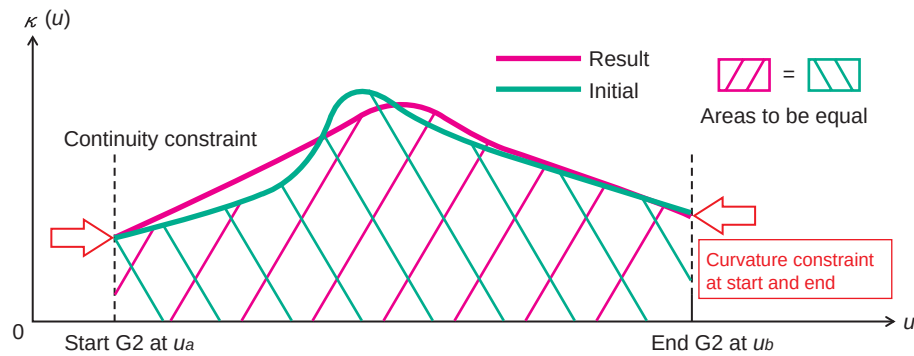


Fig. 18 Constraint of integrated area

of the existing curve, the integrated areas of the initial curve and the resulting curve are to be equal, with the position parameters for the curves specified as $u [u_a, u_b]$ and the curvature at those positions as $\kappa(u)$.

However, because the target curvature plot, TF , is drawn and input by the user, it is challenging to completely satisfy the constraint conditions discussed above. Correction of the curvature $\kappa(u)$, obtained from Eq. (49), is therefore necessary in order to satisfy the constraint conditions expressed by Eqs. (53) to (55). Using a dynamically defined scalar function, $\varphi(u)$, the new system conducts this correction as follows:

$$\kappa_m(u) = \kappa(u) \times \varphi(u) \quad (56)$$

$\kappa_m(u)$ is substituted for the curvature obtained from Eq. (49). By this means, the constraint conditions can be satisfied at the start and end points of the curve, and curvature control is realized with the continuity of shape necessary in Class A modeling.

6. Effects of New Technologies

6.1. Polygon Processing Technologies

The following results have been obtained in the polygon processing technologies in the new system.

- (1) Efficient polyline section calculation using half-edge data structure has been realized. Figure 19 shows calculation results.
- (2) The division of projected curves into segments has helped to enable more accurate projection calculations. Figure 20 shows calculation results.
- (3) The use of an original method for the efficient generation of initial patches has helped to enable the realization of high-speed and robust automatic hole-filling. Figure 3 shows the calculation process and calculation results.
- (4) The use of an original intersection calculation method for high-speed space division of mesh face sets has helped to enable the realization of more efficient calculation of distance between polygon mesh and geometric shapes. Figure 21 shows a visualization of calculation results.

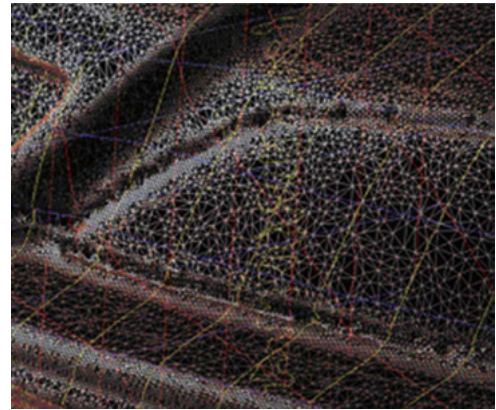


Fig. 19 Creation of polyline-section on scanned data

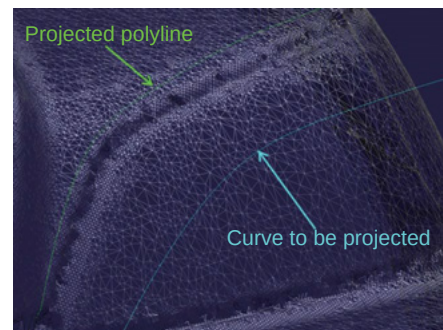


Fig. 20 Projection curve on scanned data

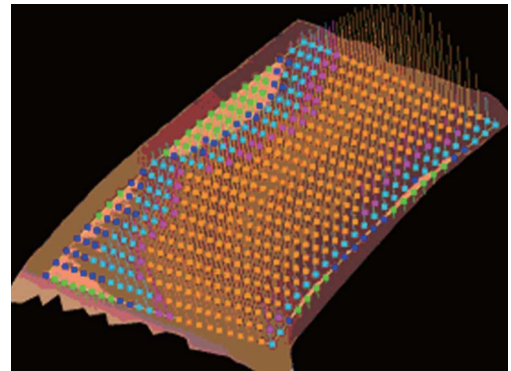


Fig. 21 Mesh-surface distance map

Table 1 Fillet performance

	Previous system	New system
Sample of fillet 1	27 min.	15 sec.
Sample of fillet 2 (Fig. 22)	52 sec.	5 sec.
Sample of fillet 3	34 sec.	3 sec.

6.2. Fitting Technologies

The following results have been obtained through the use of fitting technologies in the new system⁽¹⁾.

- (1) The use of intelligent fitting technologies has helped to ensure shape quality for individual fitting surfaces.
- (2) The application of a merge technology using fitting technologies has helped to enable surface connection with continuity between multiple surfaces.

6.3. Fillet Technologies

The use of fillet technologies in the new system has simultaneously increased the speed of computation and the accuracy of the output shapes⁽¹⁾.

- (1) Extensive revision of the rib calculation algorithm has increased the speed of computation. Table 1 compares times for the generation of fillets in the previous and the developed systems. For all samples, times have been reduced by 90% or more in the new system. Figure 22 shows the most complex shape sample.
- (2) The use of Gordon surfacing in the final stage of generation of fillet surfaces has enabled the creation of fillets with G2 continuity with support surfaces. In addition, the use of phased Gordon surfacing has increased the shape quality of the fillet surfaces themselves. Figure 23 shows a visualization of the Gauss curvature of fillet shapes produced by the previous and the developed system on

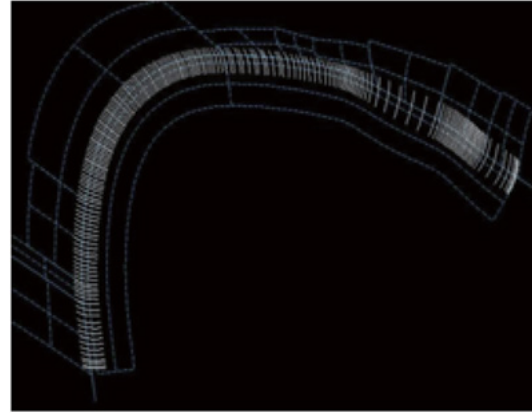


Fig. 22 Sample of fillet 2

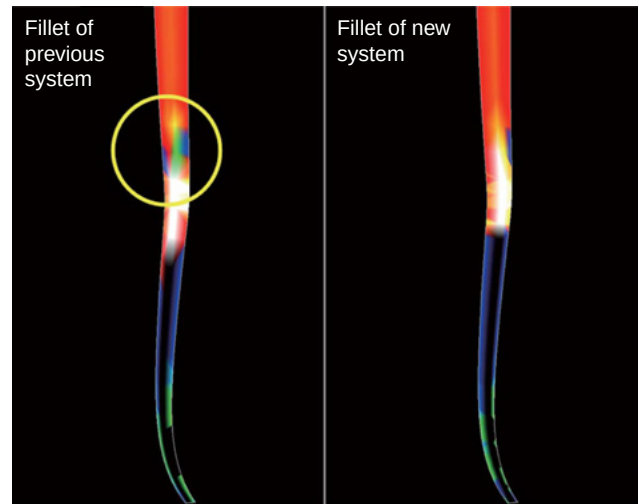


Fig. 23 Visualization of gauss curvature

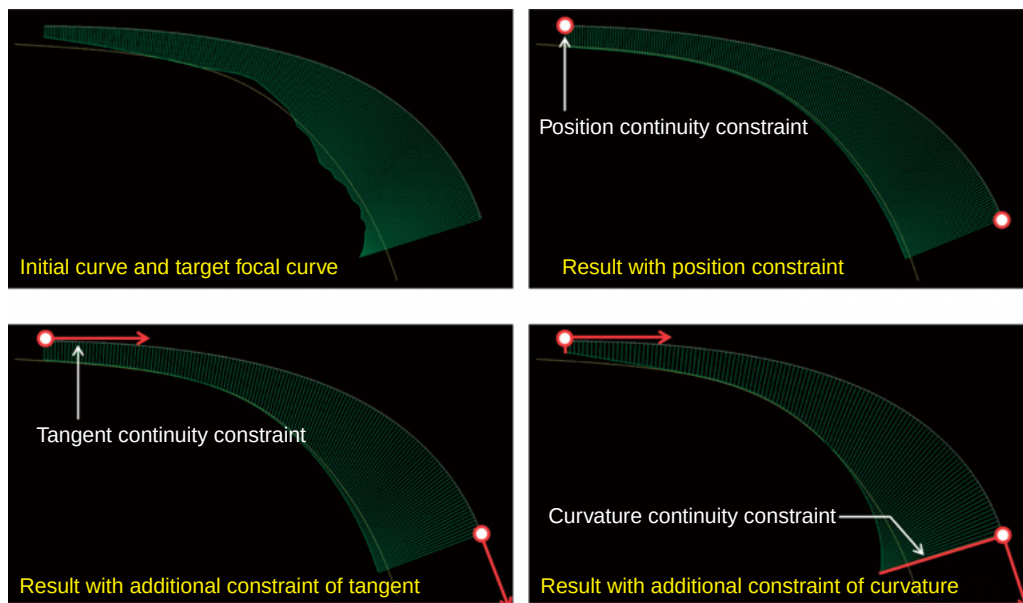


Fig. 24 Input and results of curvature control with various continuity constraints

the basis of identical input. While the fillet surface output by the previous system displays sudden changes in Gauss curvature, a high level of shape quality is achieved in the fillet surface generated by the new system.

6.4. Global Deformation Technologies

Highlight control and curvature control are examples of the realization of target-driven modeling through the application of global deformation technologies⁽¹⁾.

- (1) The highlight control function makes it possible to quickly modify a styling shape composed of multiple surfaces to a target shape simply by specifying a target highlight. The example shown in Fig. 13 would have taken 60 minutes using the previous system, but was conducted by the new system in just 43 seconds.
- (2) The curvature control function makes it possible to deform an existing curve simply by specifying a target focal curve for a curvature plot and continuity conditions for the start and end points of the curve. Figure 24 shows the results output for the same input curve and target curvature plot when only the continuity conditions of the start and end points of the curve are varied. The use of this technology has reduced the time employed in this procedure to 17% in the best cases against the previous method.

7. Conclusion

In order to realize the target of reducing the man-hours necessary for the creation of Class A surface data by 50%, the project discussed in this paper focused on three phases of the modeling process up to the stage of completion of the Class A surfaces, and sought to develop CAD functions which would considerably increase work efficiency in each phase.

- (1) In the phase of generation of fitting surfaces from scanned data from clay models, the use of an original polygon work bench has helped to enable high-speed and robust processing of high-volume polygon data, creating a foundation for the process of fitting surfaces to scanned data. In addition, the surface fitting function realizes surface generation that maintains accuracy in relation to the scanned data while satisfying the quality conditions for basic Class A surfaces, and has increased efficiency at the initial stage of Class A data generation.
- (2) With regard to achieving increased speed and accuracy in Class A surfacing functions, efficiency and data quality have been increased for central functions in the process of generation of Class A data. For example, the application of an original algorithm in the frequently used fillet generation function has increased the speed and accuracy of computation.
- (3) In the phase of correction of Class A data, the introduction of the new concept of target-driven modeling has increased the efficiency of the complex process of responding to adjustments of styling orientation and changes in designs.

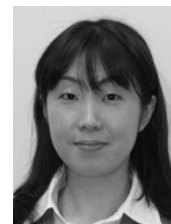
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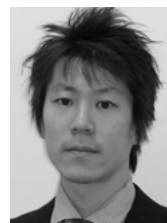
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Coordination of Information between Design and Manufacture Domains by 3D Annotated Model

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ABSTRACT

In order to maintain product quality while shortening development time, it is necessary to conduct front-loaded development. This involves digital verification conducted at an early stage of development using a 3D model of the vehicle instead of working with an actual vehicle.

What can become bottlenecks in this process are drawings that have design information presented in 2D form. Therefore, a consistent process utilizing 3D from design to manufacturing was built, taking as its core the conversion of design information to a 3D annotated model (3DA), which consists of drawings that aggregate all that information in 3D form. A notable point about the adoption of 3DA is the degree of freedom it provides for storage methods and data structure of the shape elements and attributes possessed by 3DA, according to the particular characteristics of the drawings, so as to enable smooth coordination between divisions and systems.

By these means, the early verification of 3D data using 3DA and the enhancement of efficiency in 3DA drawing creation were realized.

1. Introduction

Automobile development in recent years has had to shorten development time while maintaining vehicle quality in order to provide a high-quality product in a timely manner. As this occurs, it is becoming increasingly commonplace to consider and verify product specifications using 3D data in every stage from design to manufacture.

The drawings that show the product shapes for stamped body parts and other such elements (parts drawings) have the product shapes in the form of 3D data. The manufacturing divisions work with those drawings in verification and manufacture of the products. In contrast, there are drawings that show instructions for joining products to each other, sealing them, assembling them, and the like. These are the illustration drawings, whose purpose is combining the product shapes into the finished product. The locations of joins, the areas for coating sealing, and other instructions are also maintained in the form of 3D data during the stage of consideration in the design offices, where their feasibility is examined. When this data was converted into conventional illustration

drawings, however, the items of 3D data were projected into 2D drawings, with the design instruction information that was relevant to the purpose of the illustration drawings then added to the 2D drawings in the form of annotations.

Consequently, those in the manufacturing divisions who are responsible for using the illustration drawings would look at the 2D drawings and read the information from them. They would then perform the following operations as necessary.

- (1) Create 3D shapes for use in robot teaching
- (2) Transfer the design instructions into the form of the respective manufacturing manuals
- (3) Consider specifications that envision the way the parts are utilized in the actual car

The issues that arise in the above operations include the labor required to read, transfer, and create the data, as well as the possibility of human error during transfer. Regarding operation (3), above, since it could be performed more easily through confirmation on an actual car, this operation could only be carried out when an actual car existed, from the middle stages of development

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onward. Due to issues such as these, verification using only 2D drawings was time-consuming, and verification of specifications in the manufacturing divisions at an early stage by means of illustration drawings only took place in some of the divisions.

In light of the fact that the design offices make use of 3D data at early stages of the development process, it was determined that, by converting illustration drawings into 3D annotated models (hereafter, 3DA) and creating a consistent process for utilization of 3D data, digital verification could be carried out efficiently using highly accurate 3D data equivalent to the use of an actual car at early stages of development before there is an actual car.

This article introduces the structure of 3DA and its effects.

2. The Aim of Conversion to 3DA

The use of 3DA in product model development had the following three main aims:

- (1) To create processes for utilizing 3D data by converting illustration drawings to 3DA;
- (2) To carry out early verification using 3DA illustration drawings; and
- (3) To enhance the efficiency of the process of creating 3DA illustration drawings.

The 3D+2D drawing format that used to be the primary method in use would represent shapes using 3D data, and record attributes on 2D drawings in the form of design instruction information. When making the conversion to 3DA, the 3D data and the attributes had to be integrated efficiently in the form of 3D data, which then had to be seamlessly linked to subsequent processes. It was necessary to consider drawing formats that could be handled efficiently in such a way that, whether the 3D data or the attributes were edited, no mismatching would occur.

As shown in Fig. 1, therefore, design instruction information (marked with ※ in the figure) is stored inside the 3D data as attributes and treated as parameters. This reduces mismatching between design intentions and annotations, and controls the stored attribute information, which enables greater efficiency in drawing creation and utilization of drawings.

As shown in Fig. 2, those responsible for making use of illustration drawings through utilization of conventional 2D drawings formerly made suppositions about the specifications as they conducted their examination. This resulted in inconsistent accuracy, and did not lead to early consideration. The measures introduced here provide for free storage of attributes and enabling of control so that the 3D data can be used in all processes and coordination can seamlessly take place. This enabled the creation of processes for the consistent utilization of 3D data.

The use of 3D data to convert drawings to 3DA further allows actions to coordinate utilization between divisions

and systems in ways that could not have been managed with conventional drawings.

When converting illustration drawings to 3DA, the ways in which drawings were being used were sorted out. The usage items that were thus identified were assigned to new processes that use 3D data, and structures for retaining 3D data and locations for storing design instructions in 3D data in the form of attributes were defined in order to make the processes operative.

When designers stored the design instructions by the conventional method for creating drawings, manually entering the design instructions as text required labor. Moreover, this process involved repetitive work, and therefore tended to result in human error. With the conversion to 3DA, however, systems that allow for automated entry of 3D data were developed, and this enhanced the efficiency of design work and the accuracy of design instruction entry.

Carrying out these measures achieved the aims numbered (1) to (3) enumerated at the beginning of this section.

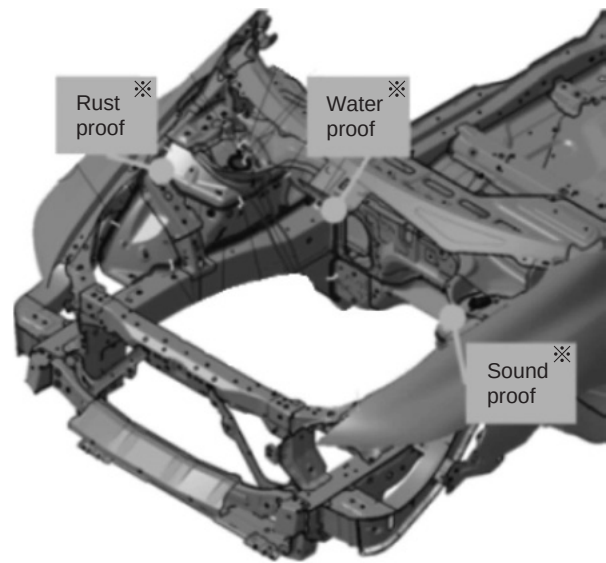


Fig. 1 Keeping attributes in 3D

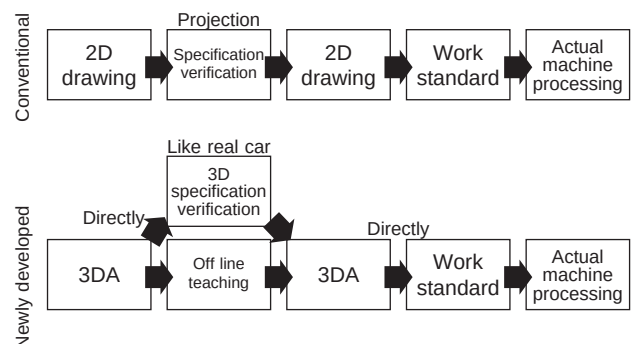


Fig. 2 3D process

3. Basic 3DA Structure

The following explains the basic 3DA structure.

Specific points to observe in creating 3DA drawings in order to conduct smooth coordination between divisions and systems include:

- (1) Classification of shape elements;
- (2) Method for storage of attributes; and
- (3) Structure of the data.

Establishing the above three items to allow a degree of freedom with respect to each particular characteristic of a drawing can enable a variety of different uses, as described in the next section.

3.1. Classification of Shape Elements

Shape elements are classified, as shown in Table 1, into product shape and supplementary shape.

The product shapes that belong in 3DA should be 3D data that enable digital verification close to that with an actual car. Examples include body shapes like those shown in Fig. 3 and shapes of combination elements like those shown in Fig. 4.

Supplementary shapes should be 3D data that are in accordance with the product shape and in a form suitable for use in simulations, manufacturing, and the like. Examples include the centerline that indicates normal direction such as shown in Fig. 5 and the edge curve (marked with ※ in the figure) such as shown in Fig. 6.

3.2. Method for Storage of Attributes

The method for storage of attributes was devised to enable the use, as needed, of the attributes shown in Table 2. These are attributes that reflect drawing information and design instructions (design attributes) and

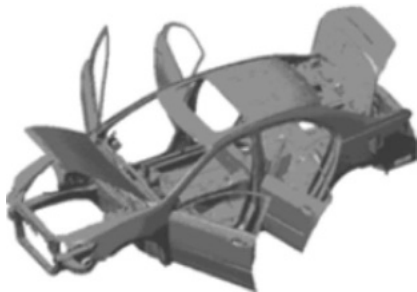


Fig. 3 Body shape

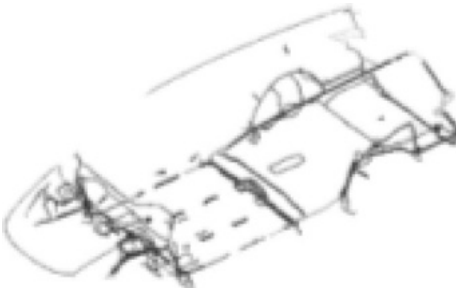


Fig. 4 Shape of combination element

Table 1 Shape element

Shape element	3D shape
Product shape	Body shape, Shape of combination element (adhesives, sealer, MIG weld, etc...), Indication symbol ...
Supplementary shape	Center point, Centerline, Center curve, Edge curve, Area surface ...

Table 2 Attribute list

Attribute	Example
Design attribute	Part number, Part name, Length, Tolerance, Angle, Surface roughness, Mass, Variation, Material, Thickness, Position, Normal direction, Destination ...
System attribute	Management number, System ID, Sequential number, Flag, ...

attributes that serve as identification for the system (system attributes).

A system that automatically acquires 3D data information (part numbers and other such attributes that are already present in the 3D data) and stores it as attributes of design instructions can also be developed in order to enhance the efficiency of drawing creation. By means of a system such as that shown in Fig. 7, designers

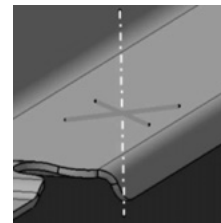


Fig. 5 Centerline

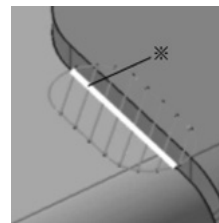


Fig. 6 Edge curve

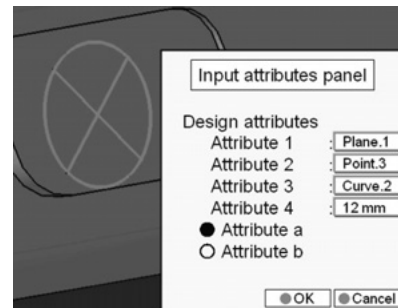


Fig. 7 Input attributes on 3D

will be able, during the course of drawing creation, to store the attributes of design instructions in storage locations controlled by system attributes without having to consider those locations.

3.3. Data Structure

The use of 3DA requires that 3D data be stored in a data structure such as that in Fig. 8. This involves putting the various shape elements and attributes discussed above into systematic order and storing them in 3D data according to defined naming rules. This enables the system to output and display drawing information in accordance with the rules for a well-ordered data structure.

In order to observe the data structure rules when creating data for a design, it is necessary to have files of drawing templates to serve as a base for the 3DA data structure. A system to enable automatic sorting and ordering of the data is also required.

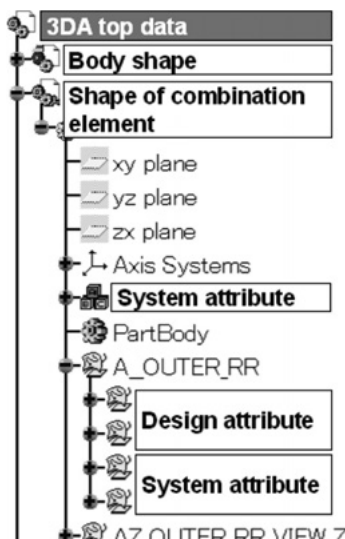


Fig. 8 Data structure

4. Case Examples of 3DA Use

With the aim of front-loading development, 3DA were created at design offices and consistent processes for the use of 3D data in the manufacturing divisions were developed. It was stated in the first section that the following operations would be performed for the purpose of smooth coordination between divisions and systems. Case examples of the use of such operations will be given.

- (1) Create 3D shapes for use in robot teaching
- (2) Transfer the design instructions into the form of the respective manufacturing manuals
- (3) Consider specifications that envision the way the parts are utilized in the actual car

4.1. Use in Robot Teaching

Supplementary shapes are included in 3DA out of

consideration for system coordination with subsequent processes. The center curve, which is a supplementary shape, was included within the 3DA as shown in Fig. 9 so that it could be used as a guide for the robot teaching path.

4.2. Use in Manufacturing Manuals

The method devised for storing design instruction information in the form of attributes realized coordination with a manufacturing manual as described in the following.

In order to make use of the information or to store it at the manufacturing site under the conventional system, those responsible for it in the manufacturing divisions should copy it from the 2D drawings to something resembling a manufacturing manual that they create on paper media. Accelerating conversion to 3DA has created sudden increases in demand for use of 3D data, making it necessary to output the groups of information stored as attributes in the 3D data onto a paper medium in order to create a manufacturing manual. With 3DA, the input and output of attributes is controlled and storage methods are devised whereby that information can be freely used. As a result, the information can be exported in a variety of formats.

As shown in Fig. 10, the system attributes included

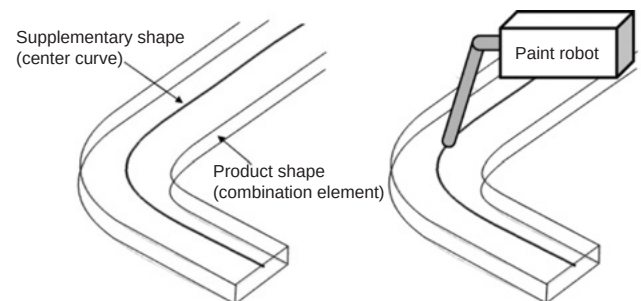


Fig. 9 Utilization of supplementary shape

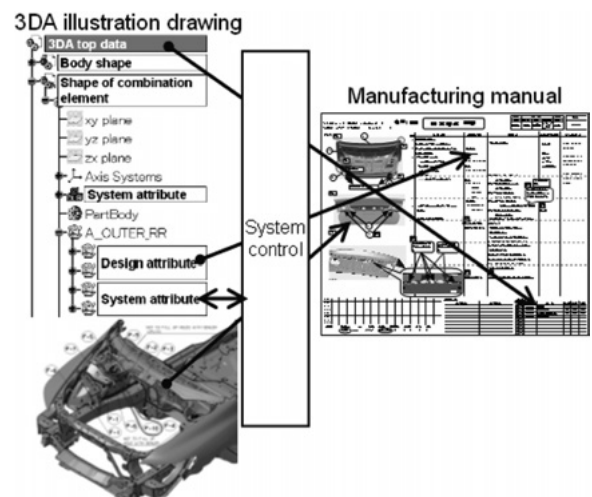


Fig. 10 Utilization of system attributes

in 3DA illustration drawings can be used as keys for automatic control by the system of attributes that are stored in accordance with the data structure rules. The necessary information can be output as PC data in Excel or another such format. Due to the automatic creation of manufacturing manuals from 3DA, transcription errors have been excluded from the creation process and the labor required has been reduced.

4.3. Use in Specification Verification

Those shapes that are necessary for each particular process in the manufacturing divisions can be isolated from the configuration of the bill of material and extracted for provision where needed. In this way, the data structure can be made to link seamlessly with the verification of each process. Where the conventional approach employed verification by projection, those items of data and the design instructions within the 3DA are combined to realize a digital verification that is equivalent to verification using an actual car.

As shown in Fig. 11, the consideration of locations for coating sealer formerly involved the estimation of joining areas from conventional 2D drawings. With the conversion to 3DA, however, joining areas can be confirmed from 3D data that is close to that from an actual car. Moreover, including shapes as design instructions has enabled consideration with greater precision in the manufacturing divisions.

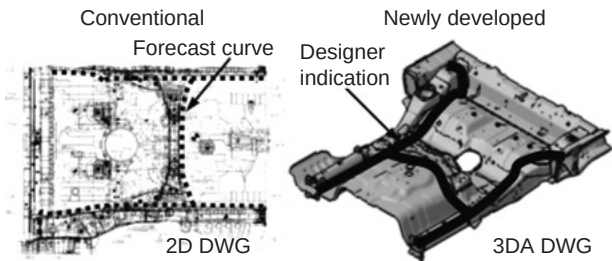


Fig. 11 Utilization of data structure

5. Effects of Conversion to 3DA

Practical implementation of the aims and measures described above realized the creation of consistent processes for utilizing 3D data. This enabled early verification of 3D data using 3DA illustration drawings and enhancement of efficiency in 3DA illustration drawing creation.

Figure 12 shows variations in the number of reports from the manufacturing divisions at each development stage regarding issues with illustration drawings. With a conventional model under development (Car-A in the figure), there were more reports of issues in the middle stage of development. With a model to which processes

utilizing 3DA were applied (Car-B in the figure), verification at an early stage of development enabled reports to be made one event in advance of issues that used to be discovered later. Deeply rooted issues can therefore be addressed earlier.

The necessary processes can be created by implementing related measures in parallel and combining the various measures, realizing verification at an early stage of development. This makes it necessary to issue 3DA illustration drawings early, and the key to this is achieving higher efficiency in creation of drawings in the design offices.

Enhanced efficiency in the creation of 3DA illustration drawings requires reevaluation directed primarily at the issues involved in conventional 2D drawing, the content of repetitive tasks, and the like.

Figure 13 shows an example of heightened efficiency in drawing creation. When creating conventional 2D drawings, the views marked 3, 4, and 5 in the upper part of the figure are created, design instructions are added in the form of annotations, and sections are created. These tasks are carried out repeatedly so that they make up a

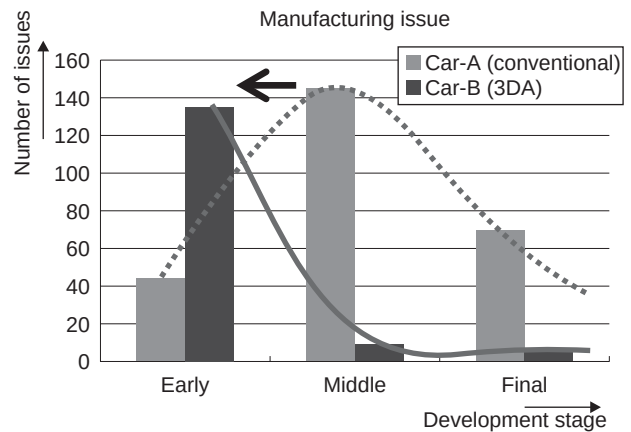


Fig. 12 Number of issues identified by manufacturing domain

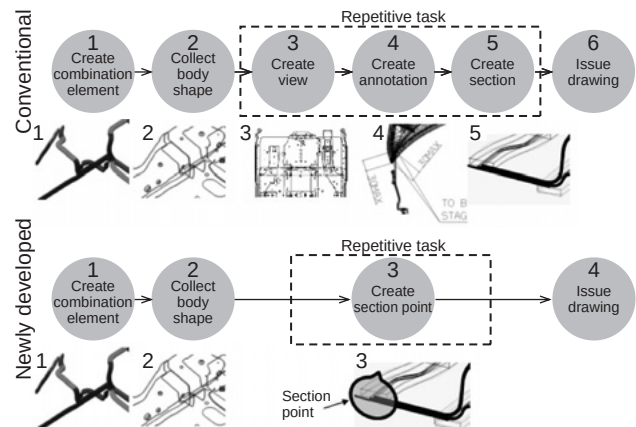


Fig. 13 Enhancing process of making drawings

large proportion of the total labor of drawing creation. When creating 3DA drawings, those tasks are replaced by the sole task of storing attributes at section points as shown in view 3 in the lower part of the figure. Thus, the work became easier and the labor required to create drawings was reduced.

The enhancements made in usability together with the creation of a work process that is free of overloads and inefficiency can realize a reduction in the labor required to create drawings without sacrificing former levels of quality.

6. Conclusion

Illustration drawings were converted to 3DA in order to create consistent processes for utilizing 3D data. This enabled the use of highly accurate 3D data to perform digital verification equivalent to that performed using an actual car at the early stages of development before there is an actual car.

The important point in converting to 3DA is not the replacement of conventional drawings with new ones. Rather, it is necessary to analyze the vehicle development process in order to clearly identify duplicated jobs, repetitive tasks, transfer work and other such bottlenecks. The process can then be recreated in an optimized form that resolves those issues. In doing so, a data structure is adopted that allows free input and output of the attributes, support by CAD functions is added so that the design offices and manufacturing divisions can handle the attributes more easily, and drawings that match the recreated processes should be made.

These measures were adopted and consequently realized front-loading with heightened efficiency in drawing creation and easier coordination of information in design and manufacturing.

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Development of CAD Tool for Layout with 2D-3D Interactive Functionality

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ABSTRACT

In the process of designing industrial products, it is important to be able to review designs while referring simultaneously to 2D and 3D forms, and to be able to mutually reflect the results of these reviews in both 2D and 3D, in particular during layout work when the shapes of parts are formulated before specifications have been established. Functionality helping to enable these procedures to be conducted efficiently is demanded of the design tool, computer-aided design (CAD). Based on design procedures and the concrete methods employed in them, the operations that can be considered as necessary in CAD were outlined with a focus on the relationship between operations in 2D and 3D. A concept and functional requirements were formulated for new functions enabling these operations to be conducted, and these functions were realized as standard CATIA V5 functions. The use of these new functions, combining layout functions and 2D-3D association functions, has increased the efficiency of the design process from the concept stage to the layout stage.

1. Introduction

With recent marked advances in computer performance, 3D models have come to be widely used in the design and manufacture of large-scale assemblies such as automobiles and motorcycles. 3D CAD systems can be seen as playing an increasingly important role in enhancing the efficiency of the design process, a key factor in the competitiveness of product development.

3D CAD advanced significantly with the introduction of the concept of feature-based parametric modeling. Parametric modeling is a method that treats not only completed forms, but also incorporates conditions that produce and constrain the forms (not limited to dimensions, but also including dimensionless numbers and non-numerical conditions such as design intentions). Responses to various design changes are affected by varying these conditions. "Feature-based" means that, rather than editing forms using topological units such as edges and faces, 3D forms able to be categorized semantically, such as bosses and ribs, are treated as individual units. Saving these units as a history in the order of their generation increases workability when forms are modified. The use of these technologies in 3D CAD has helped to enable design intentions, expertise, data employed in the production process and other elements to

be incorporated in CAD models. In addition, the automatic reconstruction of forms when the design is modified, based on the saved constraint conditions, has reduced the man-hours necessary for the modeling of 3D forms. However, in many cases whether or not these technologies will be used efficiently is affected by the level to which the 3D modeling procedures developed by the user have anticipated design modifications. In addition, the challenge of determining suitable 3D modeling procedures concurrently with design work represents an issue in terms of increasing the efficiency of design using the technologies⁽¹⁾.

Honda employs CATIA as its core CAD system. In order to achieve further front-loading of product development and a higher level of workflow integration by means of the use of 3D data, the company has made the transition from Version 4 ("V4" below) to Version 5 ("V5" below), and has worked to formulate high-efficiency 3D design methods using the feature-based parametric modeling functions in V5.

As these efforts proceeded, it was recognized that V5 displayed insufficient functionality to help enable the efficient operationalization of methods in which 3D spaces where the forms of parts were expressed and 2D spaces where cross-sectional forms and projected forms were expressed to be referred to simultaneously without mutual

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contradiction. This represented an issue in terms of increasing the efficiency of product design using V5.

This paper will first describe the design work involved in product development. It will then consider the necessary CAD procedures for each of the processes conducted during conceptual design and layout, when the forms of parts are shaped prior to the determination of specifications. Following this, the paper will introduce the concepts of the new functions projected as being necessary to conducting these procedures efficiently, and the developed functions themselves.

2. Status of Product Design and CAD Work

To provide an example of procedure, this section will describe the design work and the specific CAD procedures involved in a body-in-white development.

2.1. Main Product Design Procedures

Considered in sequence from the commencement of the development process, design procedures can be divided into conceptual design and layout.

Conceptual design involves collecting the data that forms the precondition for the conceptual design, outlining the demands for the design based on this data, and establishing a plan that will help enable their realization. The data that forms the precondition for the conceptual design includes styling sketches, clay model measurement data, packaging, parts from previous models, and new technologies to be introduced. The conceptual design process results in the formulation of diagrams that provide a visual representation of the structure and arrangement of parts, and rough definitions of forms in representative planes of projections and cross-sections. As an example of an outcome of the conceptual design process, Fig. 1 shows a diagram of the parts structure for a tailgate.

In the layout process, positionings and forms of various part which become a standard for the parts are

examined (termed “layout” below), helping to enable the results of the conceptual design process to satisfy various requirements. The form of the product is then determined based on the layouts. The first part of the process is sometimes termed the rough layout, and the second part the detail layout. The various requirements cover a broad range, taking in clearance between parts and with tools, and production considerations such as formability, assemblability, cost, weight, and laws and regulations. The necessary data includes design data, the results of reviews of the conceptual design, and the rough layout of peripheral parts. Figure 2(a) shows an example of a cross-section of the hinge section of a tailgate at the rough layout stage, and Fig. 2(b) shows an example of the formulation of an internal tailgate frame at the detail layout stage.

The process of determining the layout involves a repeated sequence of establishing temporary settings for the form definitions (“temporary settings” below), verifying whether or not the provisional form fulfills the various requirements (“verification” below), and reflecting

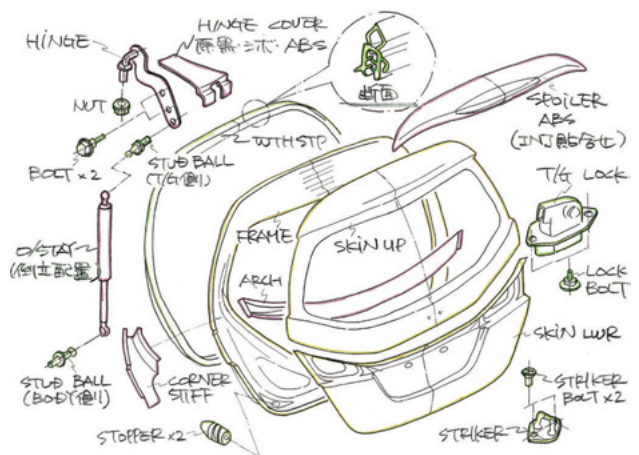
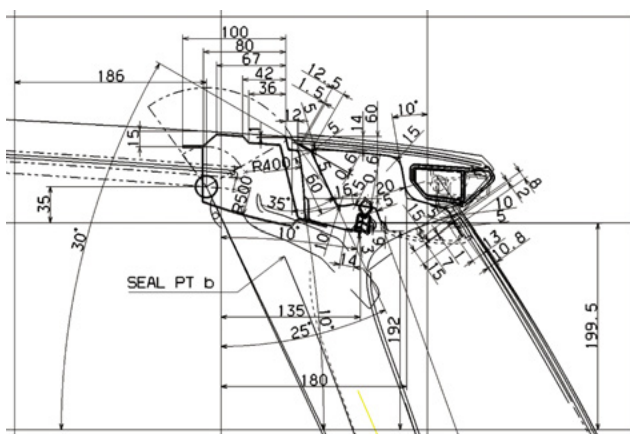
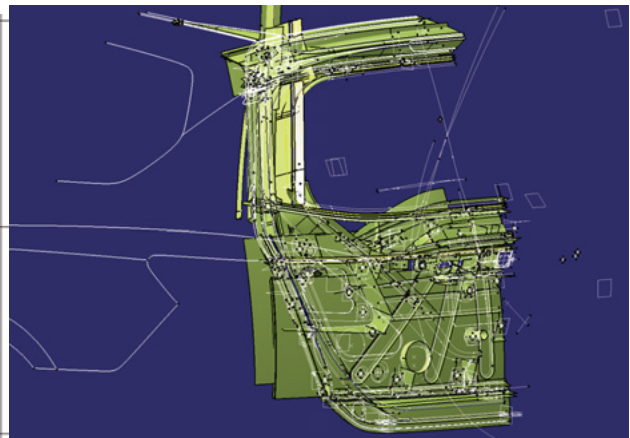


Fig. 1 Concept of parts structure



(a) Rough layout (hinge section)



(b) Detail layout (frame)

Fig. 2 Layout

the results of the verification in the temporary settings. Depending on the circumstances, this repeated process may be conducted simultaneously at multiple sites of a part, or may involve multiple proposals at a single site of a part.

At the layout stage, it is frequently necessary to respond to modifications of design data or the rough layout of peripheral parts, and to redefine forms in order to fulfill the necessary requirements. Designers should seek to determine the optimum layout and create the forms of parts under conditions in which frequent changes are made by themselves and others, and time is limited.

Because of this, it is important that designers be able to rapidly examine the parts they are working on in response to changes in their own situation or changes around them, revise the temporary settings, and conduct another verification.

2.2. Operations in 2D and 3D and their Association

This section will discuss specific methods of examination and CAD operations used in the design procedures described in Section 2.1.

2.2.1. Conceptual design

The main process at the conceptual design stage is the examination of forms in terms of representative planes of projections and cross-sections. During this process, it is important for designers to overlay 2D rough form definitions and 3D forms for previous model data and peripheral parts, and to refer to these while examining their own part. By this means they are able to recognize at an early stage the necessity for adjustments in relation to peripheral parts, and to formulate proposals for form definitions based on the existing expertise. Figure 3 shows an example of the process of overlay with parts designed by other designers. In this case, a layout of the

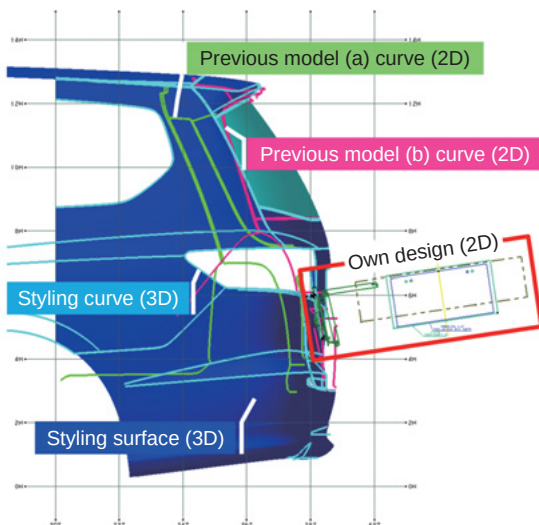


Fig. 3 Overlay

designer's own license plate part was examined, with overlays of 2D forms of the previous model and the 3D forms of the exterior design.

The inclusion of dimensions, comments, extension lines and the like ("annotations" below) and the use of printouts in order to make the concept and orientation of the rough form definition easily graspable by the other people involved in the project is indispensable for the sharing and coordination of information, and it is preferable to employ 2D data for these purposes. Figure 4 shows an example of a 2D layout with various annotations.

2.2.2. Rough layout

As in the case of the conceptual design stage, during the rough layout process, numerous operations for the formulation of standard forms and the determining of positions are carried out in 2D. With design data as a benchmark, trigonometric methods are employed in the formulation of projected views including side, plane, and back views ("views" below), and cross-sectional views ("sections" below). These are laid out, and, as previously at the conceptual design stage, the views and sections are overlayed on other forms to provide a reference as the process of temporary settings and verification is repeated.

However, work in 3D is also added at this stage, because the results of the process should be correctly reflected as positions and forms in 3D data. Figure 5(a) shows an arrangement of views with design data as a benchmark, with the rough forms of the seal sections formulated in each view. Figure 5(b) shows the formulation of synthesized curves in 3D based on the 2D data, used for the examination of the form of the frames.

2.2.3. Detail layout

When the detail layout stage is reached, procedures to generate the detailed 3D forms are added to the process. The main workspace therefore changes, and the majority of formulation and verification of temporary settings is conducted in 3D. However, operations in 2D are also

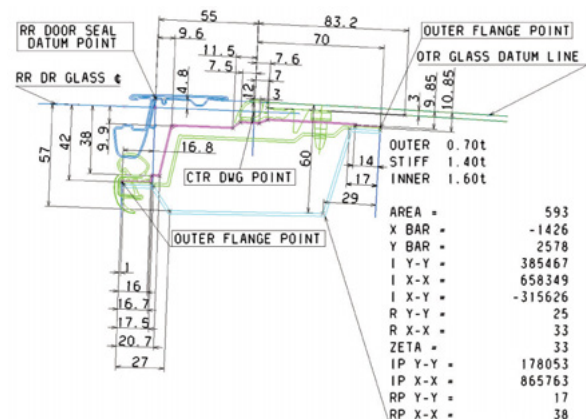


Fig. 4 Annotated 2D layout

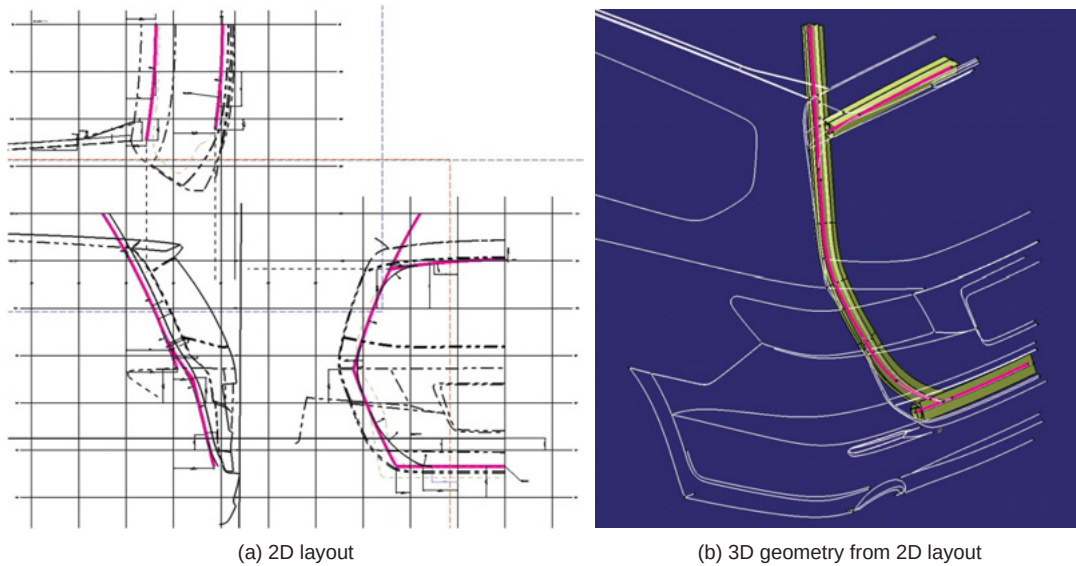


Fig. 5 3D geometry creation from 2D layout

necessary, in order to check whether the generated 3D forms satisfy the various requirements on the standard views and sections.

At this stage also, changes can be made to the design data that forms the standard or to peripheral parts and other areas, and it becomes necessary to revise settings in 2D after the changes in the 3D data have been accurately reflected in the views and sections in 2D. Figure 6 shows an example of the correction of definitions in 2D due to changes in design data. In this case, changes in the form of a glass section are projected on 2D data, and the form of the internal structure is modified.

Given these examination procedures and work functions, it is clear that work in 2D and work in 3D are closely linked, and that as the design process proceeds, the main design workspace shifts from 2D to 3D.

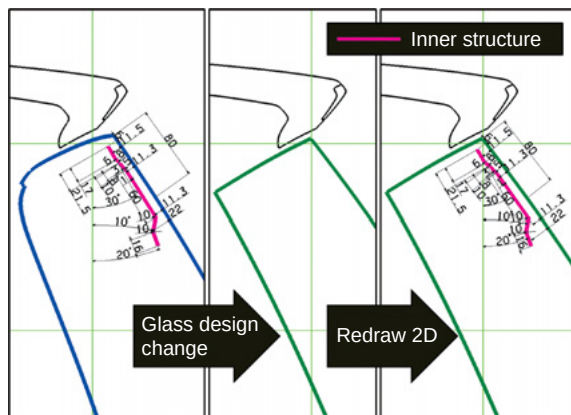


Fig. 6 Modification of 2D data due to styling design change

2.3. Necessary Operations in Conceptual Design and Layout

The necessary operations at the conceptual design and layout stages, as described in Section 2.2, can be summarized as follows.

- (1) The arrangement of the various views and sections based on graphics procedures [Fig. 5(a)]
- (2) The recording of annotations on the 2D forms, and printing as necessary (Fig. 4)
- (3) Overlays on views and sections irrespective of whether the data is 2D or 3D, and use of these as a reference (Fig. 3)
- (4) The creation of 3D forms based on the 2D forms, and the changes in the 2D forms reflected in the 3D forms (Fig. 5)
- (5) The creation of 2D forms based on the 3D forms, and the changes in the 3D forms reflected in the 2D forms (Fig. 6)

At the conceptual design stage, the majority of operations fall into categories (1) to (3), while the layout stage involves operations from (1) through (5). In other words, all of these operations are demanded of CAD, and if designers were able to perform these operations efficiently using CAD, the efficiency of the design process itself could be increased.

3. Existing CAD Systems: Characteristics and Issues

The CAD systems employed to date have been unable to efficiently conduct all of the work procedures described in Section 2.3. The specific issues displayed by the systems are discussed below.

3.1. Insufficient Functionality for Layout: V4

V4 offers a space mode (3D) and a draw mode (2D). Draw mode provides sufficient functionality to conduct the work operations in categories (1) through (3) in Section 2.3, and layout work could be conducted efficiently in 2D. In addition, the use of the AUXVIEW2 function helped to enable 2D forms to be created based on 3D data, and changes in 3D forms to be reflected in 2D (category (5) in Section 2.3).

However, while the 2D forms created in draw mode could be output to space mode (DRW->SPC), there was no update function and it was therefore necessary to regenerate the 3D forms in the event that changes were made in the 2D forms. In addition, the 3D form generation function was insufficient – for example, it was challenging to modify a 3D form after it had been created. These issues prevented the system from being used to efficiently conduct the work in category (4) in Section 2.3 and the subsequent layout work in 3D. As a result, the creation of 2D forms that are directly related to the drawings was prioritized, and it was not possible to create 3D forms concurrently, in a time-efficient manner.

3.2. Insufficient Functionality for Layout: V5

V5 offers feature-based parametric modeling functions displaying a high level of completeness. The basis for 3D forms with dimensions and geometric constraints (“2D profiles” below) can be created mainly using the sketcher function, following which layout work in 3D can proceed using various workbenches. The sketcher is able to create 2D profiles while providing transparent displays of 3D forms and other 2D forms, and can thus be used to conduct the operations in category (3) in Section 2.3. In addition, movement in both directions between 2D profiles and 3D forms can be automatically synchronized. Specifically, changes in 2D forms are automatically reflected in 3D forms simply by shifting from the sketcher workbench to another workbench, and changes in 3D forms can be reflected in 2D forms by means of the “Use-edge” function. These functions make it possible to efficiently conduct the work operations in categories (4) and (5) in Section 2.3.

However, the sketcher function is a function for the creation of 2D profiles in a single plane, and multiple 2D profiles created in different positions cannot be simultaneously aligned for reference. In addition, despite the fact that dimensions can be applied to the generated profiles, text, tables, and various symbols cannot be interpolated. These deficiencies in functionality prevented the work operations in categories (1) and (2) from being conducted using the sketcher.

On the other hand, while the drafting workbench, a function for the creation of 2D drawings, was provided with functions helping to enable the arrangement of views and sections [category (1)], the interpolation of annotations and printing [category (2)], and the generation

of 2D forms from 3D forms [category (5)], the fact that it possessed no functions making it possible to generate or update 3D forms from 2D forms [category (4)] or to overlay 3D forms [category (3)], and that 2D forms were saved in separate files to 3D forms, made it also unsuitable for layout.

Ultimately, V5 was also unable to efficiently conduct the layout work in categories (1) through (5).

4. Concept of New Functions and Functional Requirements

In Section 2.1, it was indicated that a repeated cycle of establishment of temporary settings and verification of their suitability was important in establishing the optimum layout under conditions of limited time and numerous changes to the designer’s own part and other parts. Section 2.2 pointed out that work in 2D and work in 3D are closely linked, and that as the design process proceeds, there is a transition from 2D to 3D as the main design workspace. This means that it is vital to conduct work operations that span 2D and 3D as efficiently as possible from the conceptual design stage, the first stage of the design process, to the detail layout stage, the later stage of the process.

In addition, the operations that are necessary for the conceptual design and the layout, as outlined in Section 2.3, demonstrate the functionality that is necessary in order to conduct work operations that span 2D and 3D.

Based on the above, the concept for the development of the new functions was formulated as the realization of CAD functions that helped to enable work procedures with mutual linkages between 2D and 3D from conceptual design to detail layout. The achievement of the mutual linkage between 2D and 3D can be summed up in the following functions:

- (a) 2D function equivalent to drawing function
- (b) Function for transparent background display of 2D and 3D forms
- (c) Associativity from 2D forms to 3D forms
- (d) Associativity from 3D forms to 2D forms

Functions that satisfied these conditions from (a) to (d) would help to enable all of the operations described in Section 2.3 to be conducted. The development program was therefore conducted on the basis of these conditions.

5. Development of 2D Layout for 3D Design

The ability to be operated on the V5 platform was a prerequisite in the development of functions that would satisfy conditions (a) through (d) in Chapter 4. For this reason, the development was conducted in cooperation with Dassault Systèmes, the developer of V5. The use of existing V5 functions to the greatest extent possible was established as a development guideline in order to reduce the time necessary for development. On the 2D side, the

drafting function was used as a base in order to obtain the drawing function (a), and the sketcher function was incorporated in the 2D-3D association functions [(c) and (d)]. However, a new development was necessary in the case of (b), because no existing function realized a simultaneous drawing function and transparent background display. On the 3D side, it was decided that switching with the existing 3D workbench would be utilized. These functions were incorporated in a standard workbench that could be operated on the V5 platform. The functions were termed “2D layout for 3D design” (2DLO). Table 1 compares existing and 2DLO functions, and Fig. 7 shows the 2DLO screen structure.

When the 2DLO workbench is activated, a 2D layout sheet is formulated in the parts data file (CATPart), and views and sections related to positions in 3D can be generated and arranged using trigonometric methods as shown in Fig. 8(a). Annotations can be added, and each sheet can be printed [condition (a) in Chapter 4]. 2D forms can be generated on views and sections, and at the same time a “Display Backgrounds” function helps to enable transparent display of other views, sections, and 3D forms in the CATPart file [condition (b) in Chapter 4]. Figure 9 shows an example of a study involving the overlay of data for two previous models and related 3D design data on the sections for the designer’s own part.

In addition, a “3D Profile” function makes it possible to use 2D forms as 3D forms, and an update function

Table 1 2DLO functional advantages

Category	Item	V4	V5 SKT	V5 DRW	V5 2DLO
Layout functionality	Drawing functionality	Yes	N/A	Yes	Yes
	2D overlay	Yes	N/A	Yes	Yes
	3D overlay	Yes	Yes	N/A	Yes
2D and 3D associativity	2D->3D	N/A	Yes	N/A	Yes
	3D->2D	N/A	Yes	Yes	Yes

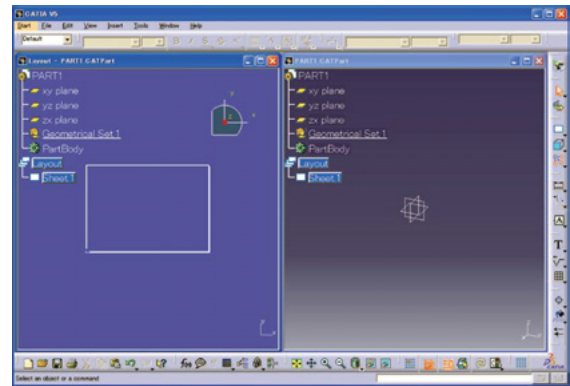
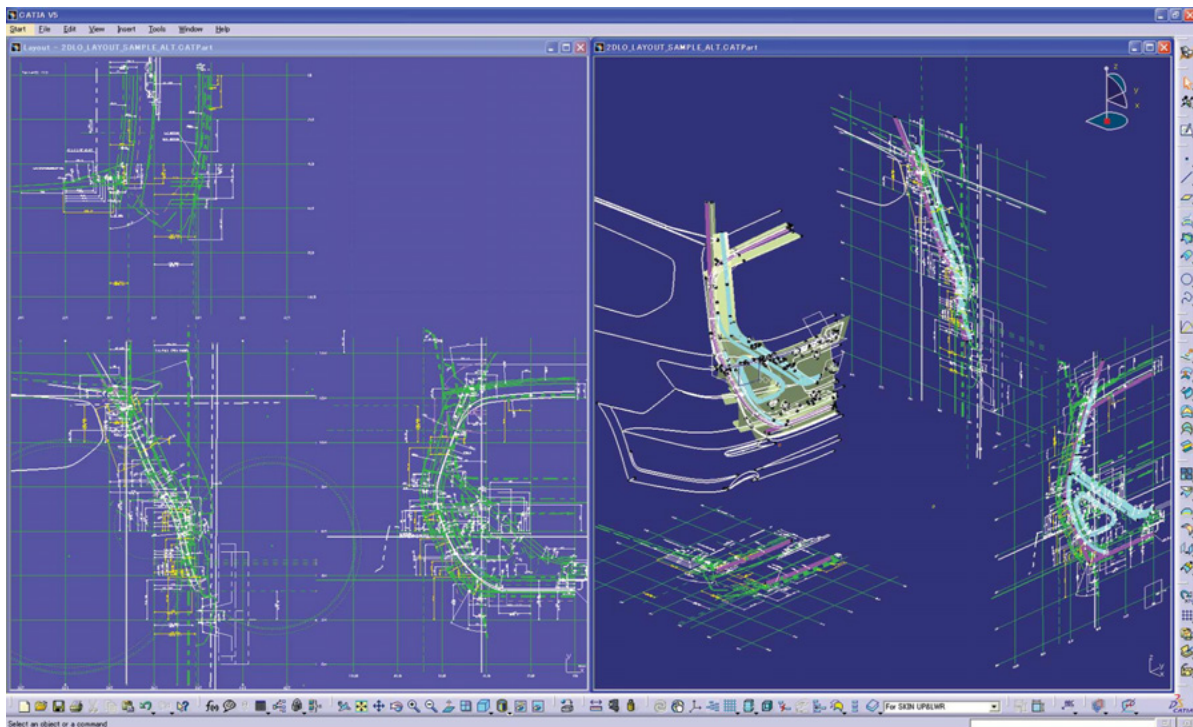


Fig. 7 2DLO screen structure



(a) View alignment

(b) 3D geometry from 2D layout

Fig. 8 2DLO 3D geometry creation from 2D layout

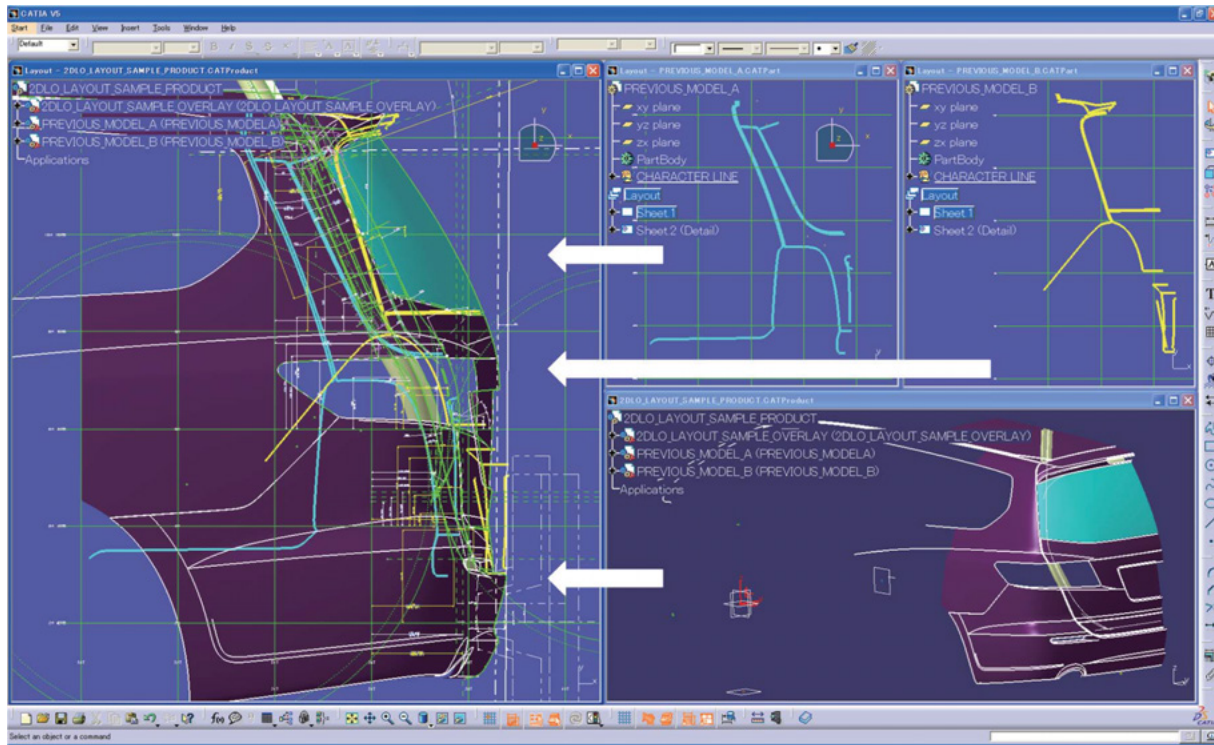


Fig. 9 2DLO overlay

helps to enable the results of editing 2D forms to be instantly reflected in 3D [condition (c) in Chapter 4]. As an example, in Fig. 8(b), 3D curves have been synthesized based on the 2D forms generated using the views in Fig. 8(a), and a frame form has been generated in 3D from this.

Further, when 3D forms have been modified using the “Use-edge” function, the update function is able to instantly reflect the latest status in the projected forms and the section forms generated on the views and sections on the basis of the 3D forms [condition (d) in Chapter 4].

In the example shown in Fig. 10, the design of a glass section has been changed, and the sectional definitions of the frame form generated based on the glass section are automatically updated without any necessity for regeneration.

These abilities confirm 2DLO as a CAD tool that helps to enable work operations to be conducted while referring simultaneously to 2D and 3D forms, and, by means of the update function, changes to be mutually reflected in 2D and 3D forms. It is a tool that satisfies the necessary functional requirements for conceptual design and layout.

6. Effects and Status of Use

6.1. Effects in Product Development Process

The effect of the use of the CAD tool in product development can be summarized in the following four points:

- (1) Work efficiency has been increased because the tool helps to enable procedures from conceptual design to the 3D formulation of the product to be conducted using identical functions as part of an integrated process.
- (2) Fewer man-hours are necessary for data correction due to discrepancies between 2D and 3D and correction of standards.
- (3) Greater sharing of concepts and design intentions has been promoted.
- (4) With the diversification of applications of the tool, the use of the same format has simplified data exchange.

With regard to (1), the new tool has helped to enable

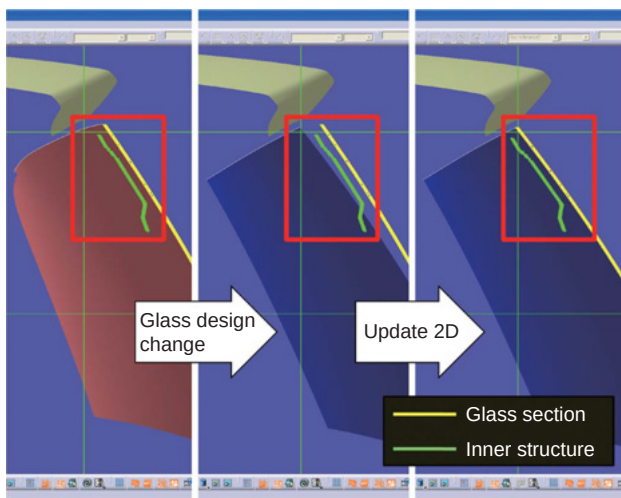


Fig. 10 2DLO Use-edge

efficient layout work with cooperation between 2D and 3D work in concept formulation, rough layout, and detail layout. This covers a wide range of product design procedures, from the stage before the specifications and surrounding forms have been determined to the formulation of 3D product data.

With regard to (2), the ability to save 2D form definitions and 3D forms in the same file has made mutual reflection of changes a simple matter, reducing the potential for unintended contradictions or discrepancies between 2D and 3D, and thus reducing the necessity to re-perform layout work due to failures to respond to modifications. In addition, fewer man-hours are employed in regenerating forms due to changes in internal and external standards such as modification of design data or revision of 2D settings.

Concerning (3), because the system makes it possible, in addition to saving 2D form definitions and 3D forms in the same file, to refer to both simultaneously, the readability of layouts in which 2D and 3D data formulated

by different designers are mixed has been increased. This is also useful in helping designers to accumulate expertise in methods of reviewing layouts. In addition, the ability to print reviews that overlay 3D cross-sections, layouts formulated by other designers, and layouts formulated by the user, has reduced the man-hours expended on preparation when checks are conducted within the department or details are coordinated with other departments.

Concerning (4), in design processes other than the body-in-white development used as an example in this paper, the system's 2D functions can be prioritized in areas in which definitions in 2D are relatively important factor; in areas in which operations in 3D are more important, studies can be conducted using a transparent display of the 3D layout. This ability to realize a variety of uses while employing a single format has made data conversion unnecessary for data exchange between different departments.

6.2. Status of Use

In addition to being used as a method of generating 3D forms with initial sketches in 2D as a base – one of the aims of its development – the new system is also finding extensive application as a tool for verification of temporary 3D settings in 2D, with layout reviews being conducted in 3D from a comparatively early stage involving transparent display on views and sections. Printouts are also being used in review meetings and to assist in coordinating details with other departments.

In addition to the layout applications described above, 2DLO is also being widely used in applications such as the generation of 2D data for printing 3D annotated model (3DA) of individual parts (Fig. 11), and the formulation of blank layouts for cost calculations (Fig. 12).

7. Conclusions

- (1) The CAD procedures necessary for design work were elucidated, and a concept and necessary requirements were established for functions helping to enable the production of layouts. These functions combined layout functions including arrangement of views and sections and the overlay of data and 2D-3D association functions.
- (2) The use of the 2DLO developed on the basis of this concept and these functional requirements has helped to enable operations to be conducted consistently using a single integrated function set from the initial stage of design to the creation of the product form, with designers able to refer simultaneously to 2D and 3D data and mutually reflect changes in each.
- (3) In addition to increasing the efficiency of work procedures conducted while referring simultaneously to 2D and 3D data, the new CAD tool has also reduced the man-hours previously expended on regenerating forms, thus increasing the efficiency of the design process as a whole.

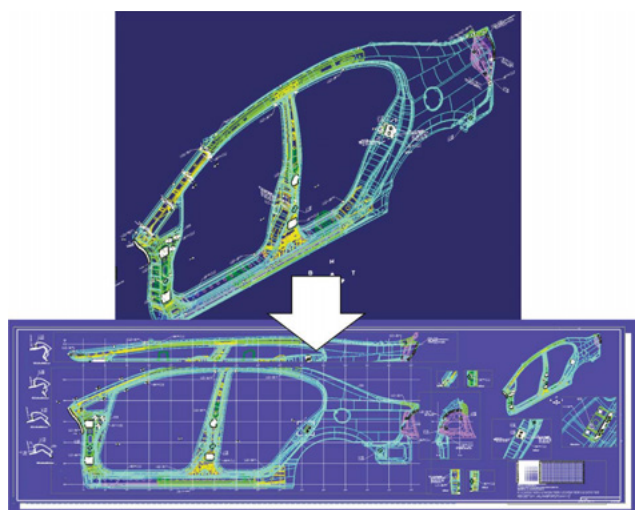


Fig. 11 2D viewing of 3DA for print

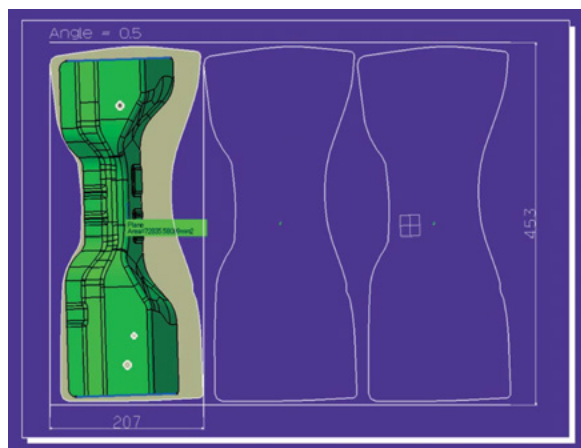


Fig. 12 Blank layout

Reference

- (1) Nikkei CG: Shin CAD no kisochishiki, Nikkei BP, p. 127-136 (1996) (in Japanese)

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Development of DPM System for CATIA V5

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ABSTRACT

With the increasing use of Data Proto Model, the number of model types and their variations for which the system is employed has increased. Coordination with Bill of Materials systems is now demanded to increase the speed of response of the system from units of weeks to days and to increase the part structure accuracy. In addition, it is also necessary to deal CATIA V5 data. A system for the construction of DPM was therefore developed to help enable the functions of CATIA V5 to be maximally exploited and the data used in drawing issues to be made available as DPM as quickly as possible. During the development process, the accuracy of the shape in DPM was verified, including the accuracy of CATIA V4 data formulated using the previous system, and the data was imported to the new system after conversion for V5. As a result, the period from a drawing issue to the completion of the DPM has been reduced by a maximum of four days, and other departments are able to review specifications ahead of schedule.

1. Introduction

The process of product development in the manufacturing industry today uses 3D computerized models called Digital Mock-ups (DMU) to help enable specifications to be reviewed at the initial stage of the design process by the design, manufacturing, and other departments⁽¹⁾. DMUs called Data Proto Models (DPM) have also proven useful to Honda in product development⁽²⁾. In the construction of a DPM, 3D data for the parts making up a full car are collected, and the data is recorded to enable the completed vehicle to be displayed on a CAD system, with mounting positions being defined using a global coordinate system in the case of 3D data formulated using a local coordinate system. Formerly, a system employing ENOVIA VPM ("VPM" below), manufactured by Dassault Systèmes, S.A., was employed to function as a Product Data Management (PDM) system in order to manage the recorded data and support the construction of the model ("former system" below). The product configuration output from the Bill of Material (BOM) system was recorded, and the DPM was constructed.

However, the following issues arose as a result of expanded application of DPM by users and the transition of the CAD system from CATIA V4 ("V4" below) to CATIA V5 ("V5" below):

(1) System processing speed declined as a result of an increase

in the volume of data due to the construction of DPM for an increased number of models and variations;

(2) An increased frequency of coordination with BOM system data became necessary in order to construct DPM using the most recent parts possible; and

(3) Compliance with V5 and Windows environments became necessary.

This paper will discuss a system development conducted to respond to these issues, which helped to enable the realization of an increased range of applications for DPM.

2. Orientation of System Development

The following development goals were established based on the issues listed above:

(1) The development of a data storage method helping to enable the number of models and variations for which DPM were constructed to be increased;

(2) The realization of an increase in the frequency of coordination with BOM system data from units of weeks to units of days;

(3) The development of a PDM system able to function in Windows environments; and

(4) The realization of scalability in the DPM construction system.

This paper will discuss the results of studies conducted in relation to these four development goals.

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3. Focus Areas in System Development

3.1. Changing the Method of Storage of DPM Data

The product configuration can be considered as being divided into technological BOM data (“BOM information” below), which corresponds to data concerning the applications of the parts, and data concerning the structure of the parts themselves (“parts structure data” below) (Fig. 1).

The BOM information is classified, from the highest level in relation to the model of vehicle downwards, into family, section, and sub-section categories. The application of the parts at the level of helping to ensure functions (“Level 1 parts” below) is managed at this stage. In the parts structure data, the component parts of each of the Level 1 parts are arranged in parent-child relationships from Level 2 to Level N.

In the former system, all of both the BOM information and parts structure data were exploded for each model and connected using the parent-child relationships in each level as link data to form a tree structure; the data was managed using a PDM system.

The only CAD data associated in the parts structure data was data for individual parts that expressed the three-dimensional form. However, because the CAD data for the same individual part could be used multiple times in the same vehicle model, or used in another model, it was necessary to set position data separately to the three-dimensional-form data. Accordingly, the variation in which the part would be applied and the date of its application were set in the link data with the next highest part, in order to function as a coordinate transformation matrix for the local coordinate system of the parent parts of each part (Fig. 2). This tree structure will be termed the “exploded mode” below.

In the exploded mode, when the history of the design changes for the variations and the parts structure for each vehicle model increase, the link data that expresses the tree structure increases as well. Because of this, the responsiveness of the system declines when search conditions are set for the link data and filtering processing is conducted using the date of application and the variation, and this was a factor that impeded the expansion

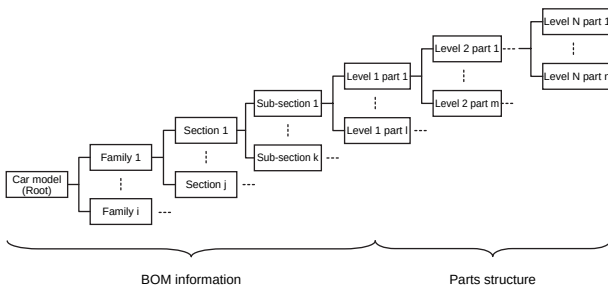


Fig. 1 Car structure in BOM system

of the range of models for which DPM could be constructed.

By contrast, in the system discussed in this paper, parts structure data is stored as V5 data called CATProduct (assembly structure-preserving file) data, and parts structure data and parts positioning data at Level 2 and below is thus not managed in the PDM system as a tree structure (Fig. 3).

Below, this tree structure will be called Work Package Mode.

The use of Work Package Mode helps to control the increase in link data when the history of design changes for variations and parts structure increases, and helps to ensure that system responsiveness is maintained when filtering processing is applied.

3.2. Coordination with BOM System Data in Units of Days

Using the former system, the latest parts structure data for a full car was recorded in units of weeks according to the model name and data instructions for the variations.

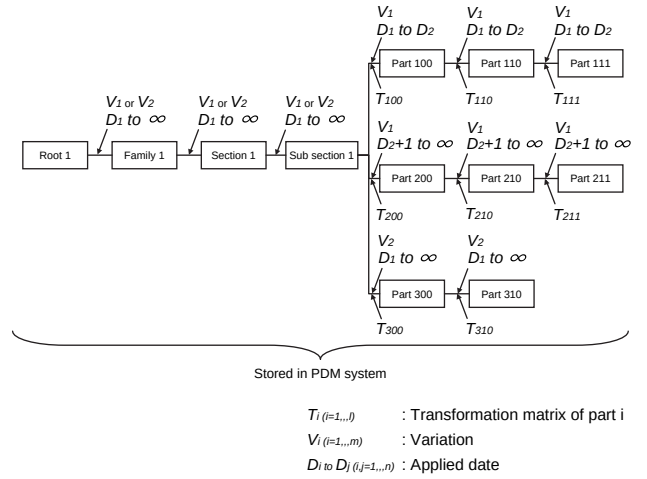


Fig. 2 Structure tree definition of Exploded Mode

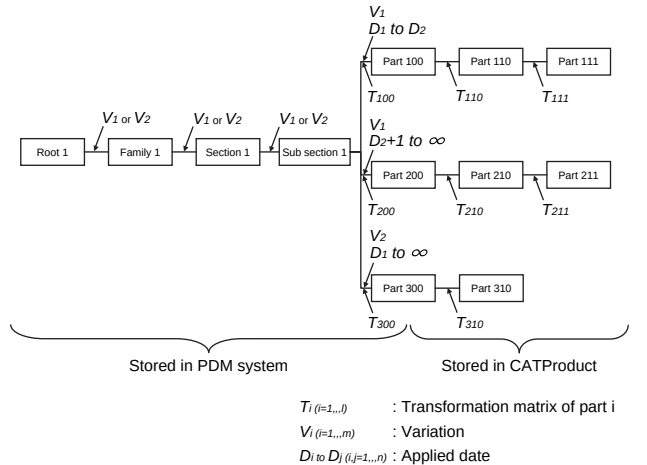


Fig. 3 Structure tree definition of Work Package Mode

Because of this, an increase in the number of models and variations for which DPM were constructed would result in an increase in time for output from the BOM system and time for recording in the DPM construction system. In addition, because updates were conducted in units of weeks, inconsistencies occurred with the details of design changes that were processed on a daily basis in the BOM system, and this became an issue impeding the expansion of the application of DPM.

Accordingly, in the new system, data concerning models and variations for which DPM will be constructed is recorded in advance in the BOM system, and is output to external files in units of design changes of the parts applied in the recorded variations (Fig. 4).

This has helped to reduce the time for extraction of BOM data processed by the BOM system.

In addition, the daily recording of the external files in the system has reduced the time for recording, and has helped to prevent inconsistencies with the BOM system data.

3.3. Changes to the PDM System

In the former system, DPM were constructed with VPM operating in a UNIX environment employed for 3D data formulated using V4. The introduction and maintenance of UNIX machines entailed high costs, and this was a further issue impeding the expansion of the system by means of increasing the number of models and variations for which DPM were constructed.

With the transition of the CAD system from V4 to V5, the new system adopted ENOVIA V5 VPM ("LCA" below) in order to obtain the following advantages:

- (1) High level of compatibility with V5 (Fig. 5);
- (2) Ease of access to parts structure data and 3D data from V5 (Fig. 6); and
- (3) Low-cost and high-speed processing using Windows machines.

This helped to enable the reduction of modeling time following recording of parts structure data.

3.4. Realization of Scalability via Distributed Processing

When the new system conducted distributed processing, errors sometimes occurred in the various

processing steps. A function helping to enable the system to easily detect processing errors and respond rapidly was therefore necessary for system operation. A distributed processing function developed in-house for application in another system was used in the new system to help satisfy these conditions.

Figure 7 shows an overview of the configuration of the job control system.

Data from the DPM construction server (the submit machine) is recorded as a job in the job queue (database).

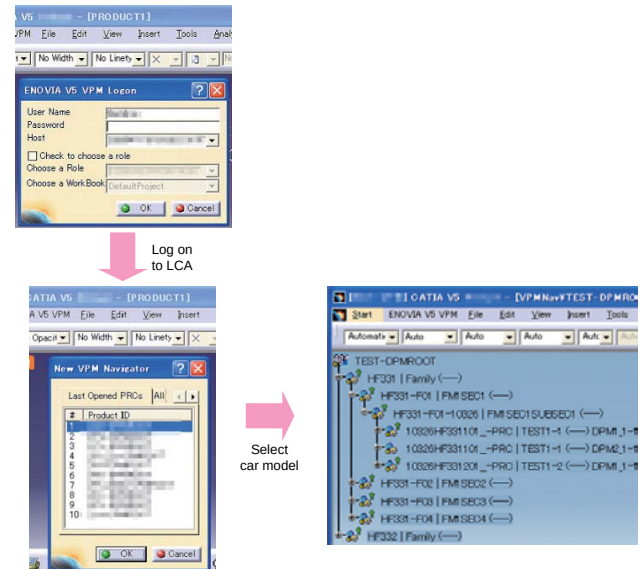


Fig. 5 Operation to access to structure tree

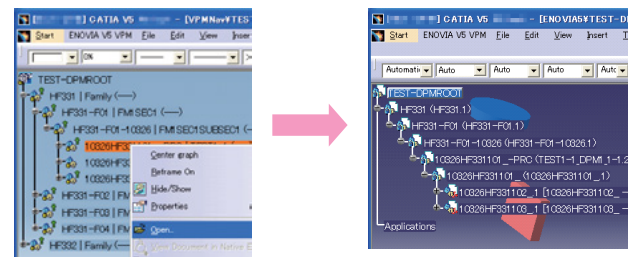


Fig. 6 Operation to access 3D data from structure tree

	Date						
	Mon.	Tue.	Wed.	Thu.	Fri.	Sat.	Sun.
Previous system							
Drawing issues	✓	✓	✓	✓	✓		
Structure lists							
: Parts structure of full car							
New system							
	Mon.	Tue.	Wed.	Thu.	Fri.	Sat.	Sun.
Drawing issues	✓	✓	✓	✓	✓		
Structure lists	✓	✓	✓	✓	✓		
: Parts structure of 1 drawing issue							

Fig. 4 Timing of drawing issue and structure output

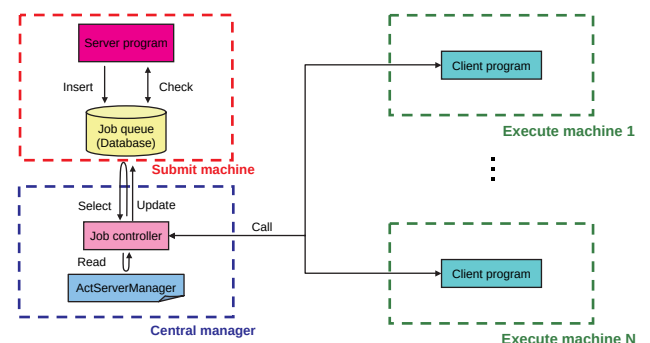


Fig. 7 System configuration of job control system

The job controller periodically monitors the database, and allocates the job to the CATIA processing machines (the execute machines). In the CATIA processing machines to which the job has been allocated, the data is written in a view format in the processing management file (the ACT server manager); the number of processing treatments that can be conducted simultaneously can be set here. The processing management file can be modified even during job allocation, and scalability can be realized via the number of processing treatments.

In addition, the status of the job that has been executed is reflected in the database, and when an error occurs, processing can easily be conducted again by modifying the status. This has helped to increase the efficiency of the operation of the DPM construction system.

4. Data Transfer from Former System

During the development of the new system, it was vital to transfer data in order to effectively utilize the DPM data resources of the former system. Data for models that would be usable in the new system was selected for transfer. Table 1 shows the numbers of data transferred.

Based on these data numbers and the period for which they would be used in the new system, data transfer was conducted with the following service assurances set for the departments that would utilize the data in-house:

- (1) Conversion of Exploded Mode into Work Package Mode, with assurance of no discrepancies in the fitting positions of each part in the global coordinate system;
- (2) Conversion of V4 data to V5 data with assurance of accuracy of three-dimensional form; and
- (3) Completion of data transfer per model and suspension of service within the shortest period (two weeks).

4.1. Conversion of Data from VPM to LCA

To allow coordination with external systems, the data formulated in the former system was output in the EXPRESS data modeling language specified by ISO 10303-11 ("STEP Part 11" below), as external files expressed using the exchange structure format specified in ISO 10303-21 ("STEP Part 21" below)⁽³⁾.

However, the readability of the STEP format is low: not only is there a high volume of standards documents, but the standards are dispersed, and if they are not reconstituted, understanding them represents a challenge. In addition, because the files themselves were expressed using VPM data management information, it became challenging to grasp the data in-house as parts and vehicle configuration data.

It was therefore determined that for future DPM data conversion, the STEP Part 21 format would be replaced by the XML format, based on consideration of the following points:

- (1) Increased in-house usability;
- (2) Increased readability when data is converted; and

Table 1 Amount of data convert

Item	Number of models	Number of variations	Number of V4 data	Number of V5 data	Total number of V4/V5 data
Total	24	69	31888	14750	46638

- (3) Reduction of system development costs by using widely available XML tools.

The conversion from Exploded Mode to Work Package Mode was conducted simultaneously. The process was conducted such that the coordinate transformation matrices for the global coordinate structure of the lead parts for each product structure were also written in the XML file, helping to enable their positions in the full vehicle to be recovered from the XML file.

In addition, the XML was made effective in the specific namespace of the new system, and development was conducted to help enable it to also be used as an input file format. At the same time, the system was configured to help enable DPM parts structure data formulated in the new system to be output in XML in the same format. This helped to enable the following:

- (1) Shared use of data for other models when DPM are being newly constructed;
- (2) Copying of position data when parts numbers are switched from prototype form to mass-production form; and
- (3) Provision of data to other systems.

4.2. Conversion from Exploded Mode to Work Package Mode

As discussed above, in Exploded Mode, position data for each part in the parts structure data is expressed as a coordinate transformation matrix in relation to the local coordinate system of that part's parent part. Because of this, when the coordinates of a part in the global coordinate system are calculated, a product of the coordinate transformation matrices for all the higher levels in relation to the part in question is necessary (Fig. 8).

By contrast, in the Work Package Mode employed in the new system, parts at a lower structural level (assembly parts) are expressed using CATProduct. The positions of 3D data included in the CATProduct are expressed as coordinate transformation matrices from the parent

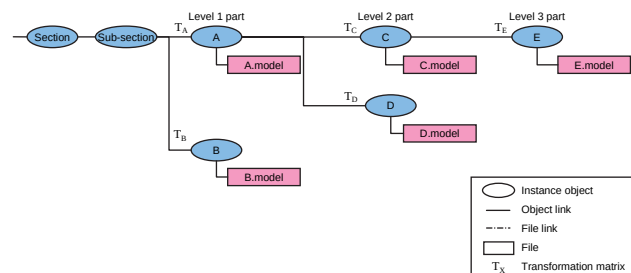


Fig. 8 Position matrices of Exploded Mode

CATProduct, and stored in CATProduct (Fig. 9).

This helps to enable the expression of position data for assembly parts as local coordinate systems, and when assembly parts are utilized in another model, they can be arranged without being affected by the specific position data for that model.

When the unique XML data for this system is formulated as discussed in the preceding section, coordinate transformation matrices that provide position data for each part are calculated, and coordinate transformation is conducted from the global coordinate system to local coordinate systems. The use of the XML developed for the system has helped to enable automatic formulation of CATProduct data that expresses the parts structure, and offered the ability to set relative positions within the assembly.

4.3. Assurance of Form when Converting to V5

The format for storing 3D data in V5 differs in part from that used in V4, and it is therefore necessary to check for any discrepancies in form after conversion. A three-dimensional form error detection system that was developed in-house was therefore linked to the system and the three-dimensional data following conversion was checked to help enable study of differences in form occurring when V4 data formulated was converted to V5. This process of detection of differences in form was conducted in the following two stages for CAD data before and after conversion:

- (1) Comparison of solid numbers, surface numbers, volumes, centers of gravity, and surface areas; and
- (2) Comparison of surfaces and edges.

However, it was decided that in order to reduce processing time during data transfer, (2) would only be conducted when a discrepancy had been detected by (1).

The results of these form comparisons were as follows:

- (1) No discrepancies of form were observed for approximately 90% of the models for which three-dimensional data formulated in V4 was transferred;
- (2) In the case of models for which form discrepancies were observed, the three-dimensional models were manually corrected; and
- (3) In the case of data for which the process of comparison was not conducted correctly, forms were compared in V5.

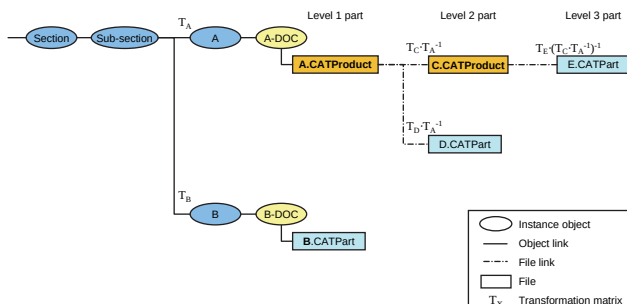


Fig. 9 Position matrices of Work Package Mode

As a result, it was possible to convert all of the 31888 pieces of V4 data shown in Table 1 to V5 data.

5. Effects of the New System

- (1) The time from a drawing issue to the completion of the DPM has been reduced by four days from a maximum of 10 days to six days.
- (2) Position data in the global coordinate system from the output product configuration data can be used in the focus model or in other models, and the time for setting positions has been reduced by approximately five minutes per Level 1 part.
- (3) Models can be constructed on Windows machines.
- (4) The ability to extract changes in units of design modifications in coordination with the BOM system has reduced the man-hours necessary for extraction of changes.

6. Conclusion

The development of the system discussed above produced the following results:

- (1) The reduction of tree data managed in the PDM system in comparison to the former system has helped to enable the models and variations for which DPM are constructed to be increased;
- (2) Data can be coordinated with BOM system data in units of days, helping to enable data from drawing issues to be formulated as a DPM in the shortest possible time-frame;
- (3) The use of V5 as the system for the construction of DPM has helped to enable DPM to be constructed in Windows environments; and
- (4) The introduction of a distributed processing function has helped to realize scalability of the DPM system.

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Development of DSS (Drawing Support System)

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ABSTRACT

The DSS system operates on CAD software (hereafter, “CATIA V5”). It inputs the drawing history, components and other management information (hereafter, “parts management information”), compares this information with the information of the technical parts table system (hereafter, “NEWS”) that comprehensively manages basic automobile product information, and checks for compatibility between the two.

An interface was developed to efficiently input parts management information, and the series of tasks to feed back the results of the compatibility check with NEWS to the drawings and correct the drawings was automated. In addition, security measures, when connecting with external networks, were strengthened, enabling the DSS system to be used not only by Honda R&D Co., Ltd, but also with suppliers’ system environments via the internet.

This effectively reduced the drawing creation and issue processing time needed for designers to create and correct drawings by approximately 35% compared to the CATIA V4 drawing support system.

1. Introduction

The CATIA V4 drawing support system (hereafter, “old system”) extracts parts management information from the title blocks and composition columns in the drawing data, and checks this information. This old system was developed and deployed in-house, but creating drawing frames and composition columns and correcting check results required much processing time, and there were issues with ensuring security and external database connections. For these reasons the old system was not provided to suppliers, and there were the following issues and limitations when creating and issuing drawings.

(1) Complicated drawing creation procedures

When designers created drawings, they had to create title blocks and composition columns while manually positioning standardized symbol shapes (i.e., table frame lines used to express composition columns, and symbols and text inside blocks and columns) that conform to the drawing creation rules, including technical standards designated as drawing creation rules. For example, when there are multiple components, composition columns

should be created by selecting from the symbol shapes for each component.

(2) Parts table information check function

The series of check tasks was not automated, so corrections required much processing time. Part names and quantities applied in the composition column were checked manually, which took much processing time.

(3) Lack of drawing creation environments at suppliers

When each supplier issues drawings, drawings created according to in-house rules should be submitted to the designers in charge. However, the old system had security, communication performance and other issues, so it was not provided to suppliers. Designers had to check drawings created by suppliers and correct the title blocks, composition columns and other drawing information using the old system, which imposed a heavy workload.

To address these issues with the old system, a drawing support system (DSS) was developed in conjunction with the shift to the CATIA V5 system. This new DSS system supports the creation of parts management information and compares it with NEWS to check for compatibility between the two.

* Automobile R&D Center

2. Development Goals

In CATIA V5, files are divided into drawing (2D) and product/part (3D) files. A function that links and manages these files enables the reuse of existing data and makes it easy to divide work between multiple designers. The V5 technical characteristics were used to address the aforementioned issues and develop a new drawing support system with the following goals.

- (1) Developing a system that supports the creation of drawing and parts management information for CATIA V5, and increasing the work efficiency of designers.
- (2) Developing functions linked with NEWS that check and automatically correct parts management information, and enhancing the drawing quality check automation functions.
- (3) Strengthening security measures, providing drawing support tools to suppliers, and increasing convenience.

Figure 1 shows the system overview. The system aimed to enhance drawing quality and simplify drawing creation procedures by centralizing drawing support functions (drawing support commands) such as automatic title block and composition column placement, drawing information input, and input results checking.

Additional development goals included linking with Web Help to enable designers to make corrections while referencing Help corresponding to the error factors, maintaining security, and enabling suppliers to also use the DSS.

Strengthening the link with NEWS, increasing the drawing check function accuracy, minimizing drawing rejections, and reducing the lead time until drawing issue by 35% were also set as goals.

In addition, system construction aimed to facilitate links between each system to enable easy function expansion in the future, such as 3D attribute information.

3. Functions Developed for the DSS

New functions were implemented in accordance with the development goals. The developed functions are described below.

3.1. Parts Management Information Creation Support Function

In the data structure of the old system, attribute information required for drawing and parts management was held within text and other symbol shapes. Designers used multiple drawing commands to first locate symbol shapes on the drawing, and then entered the attribute information. This created issues such as low work efficiency and the inability to flexibly relocate symbol shapes.

The new system uses a data structure that separates and enables individual handling of symbol shapes and attribute information. The link between symbol shapes and attribute information is managed inside the system, enabling designers to input parts management information without the need to pay conscious attention to symbol shapes. Title blocks and composition columns can be efficiently created and flexibly laid out on drawings by inputting the parts management information from a special user interface window used to create title blocks and composition columns, and designating the location coordinates on the drawing (Fig. 2).

In addition, Excel import and export functions were also developed. The import function inputs title block and composition column information described in the dedicated Excel format to the individual title blocks and composition columns on the DSS. The export function outputs a list of the individual title blocks and composition columns on the DSS in the dedicated Excel format.

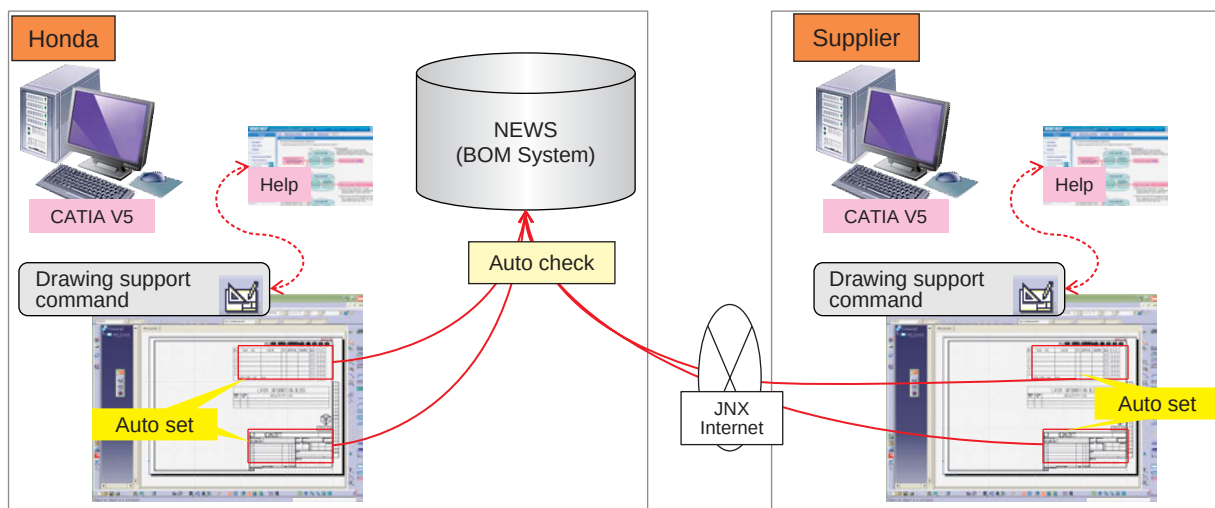


Fig. 1 Image of DSS

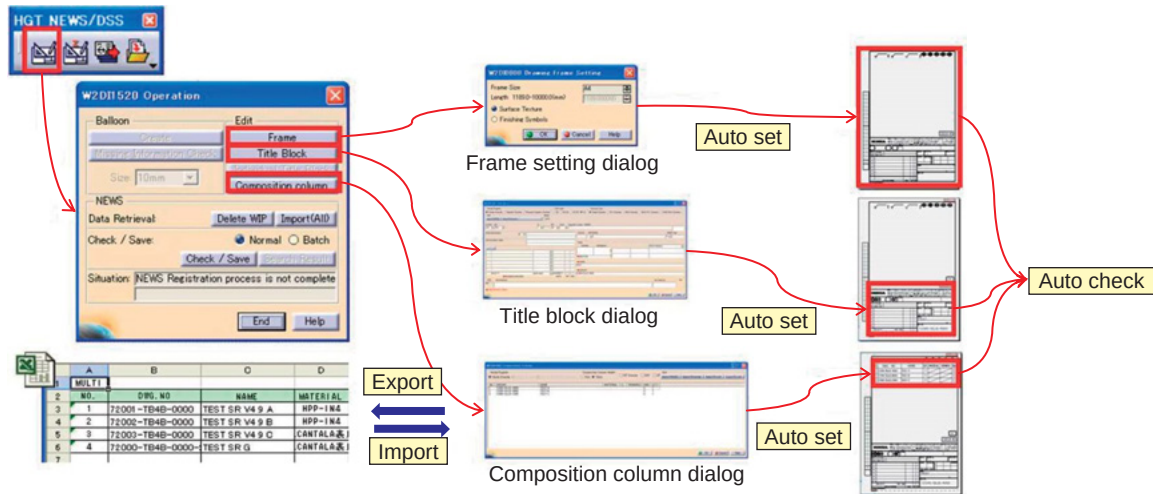


Fig. 2 2D drawing support

Input errors may occur when designers manually input part numbers and part names on the DSS, but this information can be input to the DSS without input errors by copying the composition information from NEWS, pasting it to the dedicated Excel format, and thereby importing the information.

3.2. Parts Management Information Check and Correction Function

A function was developed that links the two DSS and NEWS systems by socket communication, and automatically checks the parts management information (Fig. 3). This function outputs the parts management information created by the DSS as a text file, connects with the NEWS distributor server ((A) in the figure), and issues a processing request to a NEWS calculation server that is not currently performing processing ((B) in the

figure). The NEWS calculation server then checks the compatibility with the basic product information in the NEWS database. The overall processing speed is increased by connecting multiple NEWS calculation servers in parallel and distributing the processing.

Compatibility is checked by comparing the part number history (design change history), part names, and applied part quantities, and the check results are stored on the NAS ((C) in the figure). When the drawing check results in an error, the check results are fed back to the DSS as a text file ((D) in the figure) so that the parts information is automatically kept up to date. In addition, when contents require confirmation by the designer, correction confirmation locations are highlighted on the drawing, providing the flexibility for designers to judge the need for the corrections. A Help window showing error messages and remedies at the same time is also displayed, enabling drawings to be corrected while checking these contents. Drawings with large numbers of components may have 300 or more composition columns. Compatibility was checked by tracing the design change history for these components, which resulted in a heavy workload on designers. Formerly all parts management information was checked manually, but the DSS enables automatic checking.

3.3. Supplier DSS Functions

Figure 4 shows the flow when the drawing check is performed using the supplier DSS. First, a text file containing the parts table information is created locally within the supplier's system environment from the drawing data. A connection is made with the certification server located in the DMZ area, and SSL certification and login certification are received. Installation of a reverse proxy server enables a proxy response to the certification server, which further enhances security. A firewall is installed in the DMZ to help prevent illegal access from the outside.

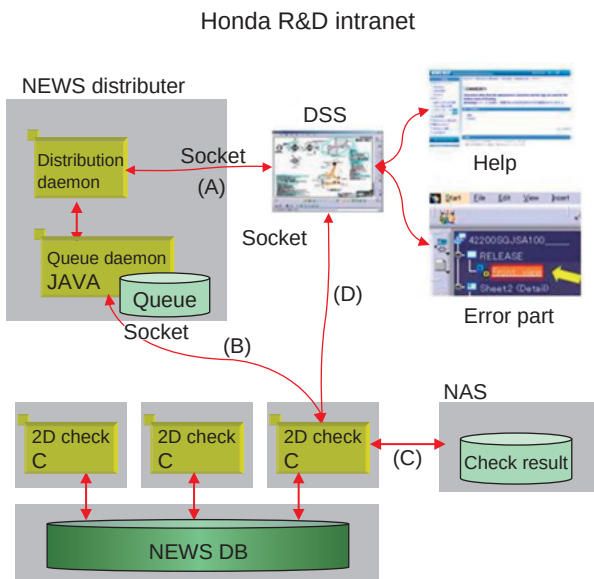


Fig. 3 DSS check flow

After that, the parts table information file is uploaded from the Servlet to a server located within the Honda intranet. The check is performed in the same manner as that of the in-house DSS, and the check results file is output via the server to the supplier's local terminal. Finally, the check results are displayed in the supplier DSS dialog.

The Remote Method Invocation over Internet Inter-ORB Protocol (RMI-IIOP), which is the industry standard for inter-system links, was used as the connection protocol from the external DMZ area to the server located in the Honda R&D intranet.

Each supplier is given a unique manufacturer ID tag as a measure to strengthen software security. This manufacturer ID tag contains the manufacturer code, manufacturer name, and a site name that notes the Honda

facilities abbreviation. When an attempt is made to perform the drawing check using the supplier DSS without this manufacturer ID tag included in the module, control is performed to disable the drawing check execution.

An installer was created for introducing the DSS in order to increase supplier system environment construction and operation efficiency. As shown in Fig. 5, the supplier DSS main program and Help can be installed separately, enabling the software to be provided with the minimum difference module size.

4. Effects

4.1. Increased Drawing Creation Work Efficiency

The DSS unified the drawing functions into a single function, enabling automatic creation of drawing frames, title blocks, composition columns and other items using a single unified command (Fig. 6).

4.2. Enhanced Drawing Quality Check Function

The drawing quality check function was enhanced by linking with NEWS and introducing a mutual check system for parts table information and drawing information when creating drawings.

Automation of checks both simplified and increased the speed of designer work. This increased drawing creation efficiency and the enhanced drawing quality check function reduced the designer work time by 35% or more compared to the old system (Fig. 7).

4.3. Provision of a Drawing Creation Environment to Suppliers

The supplier DSS provides the same functions as those in the in-house version, and contributes to enhanced drawing creation efficiency and drawing quality check functions at suppliers. As of August 2009, the DSS is being used by 210 suppliers both domestically and overseas.

4.4. Enhanced System Capability

The DSS integrated part of the drawing check function with the drawing creation function, which shortened the

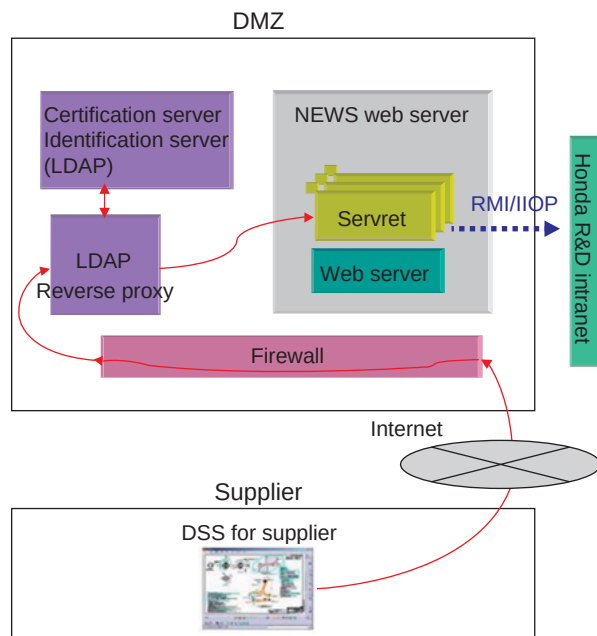


Fig. 4 Supplier DSS check flow

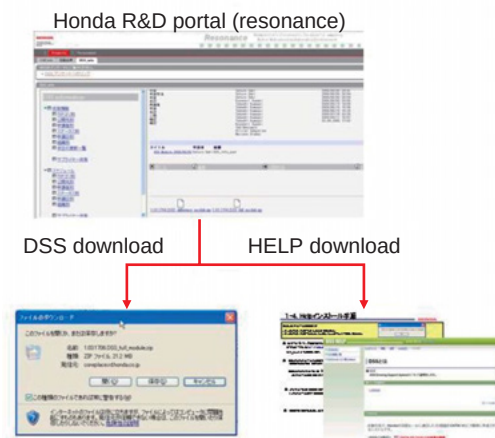


Fig. 5 Supplier DSS difference installer

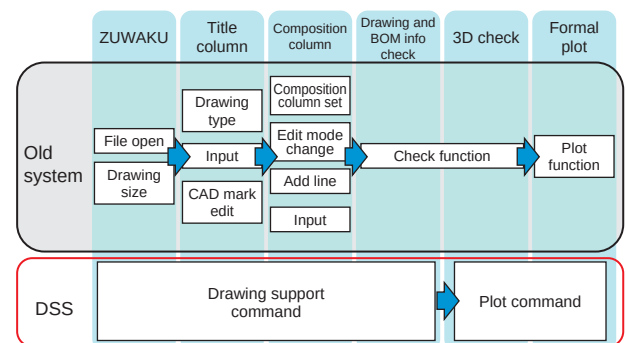


Fig. 6 Unification of command

time required for drawing checks while at the same time increasing the check accuracy. Parallel connection of multiple processing servers and other measures were also taken, which enhanced system stability, increased the overall processing speed, and otherwise secured performance equal to or better than that of the old system.

In addition, strengthening the link with NEWS increased system convenience and facilitated links with other systems.

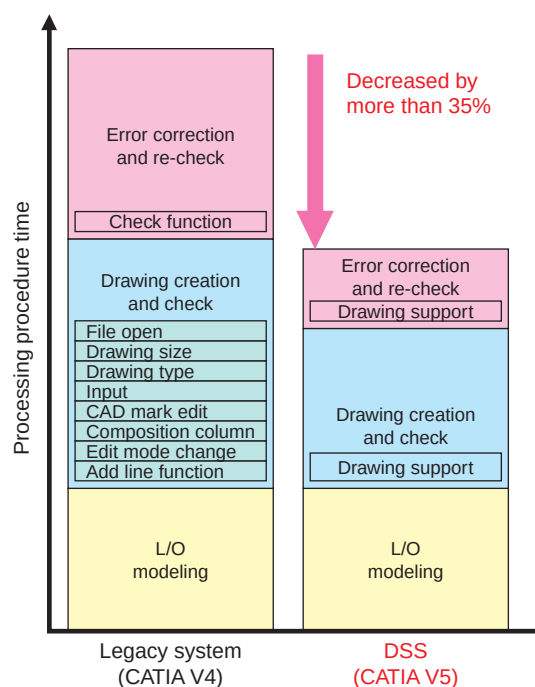


Fig. 7 Processing procedure time

5. Conclusion

Development of the DSS increased designer work efficiency, enhanced the drawing quality check automation function, and reduced the drawing creation and issue processing time by approximately 35%. In addition, security measures were strengthened, and the DSS was also provided to suppliers, which increased convenience.

Marked advances are being made in 3D modeling technology, and in the future the drawing data format will shift from 2D to 3D, and expand to a data format that holds 3D attribute information.

Future DSS developments include the ability to load and check 3D attribute information written on models, and enhancements enabling the exchange of accurate data with downstream processes in the same manner as the current drawing check. In addition, efforts will be made to construct a system that can more easily realize links with NEWS and other systems, and to deploy a system that can contribute to model development.

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Development of NOx Reduction System Based on Exhaust Valve Controls for Use in Small Motorcycle Engine

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ABSTRACT

An internal EGR system was devised that opens and closes an exhaust valve slightly at the beginning of the compression stroke, or between the end of the intake stroke and the beginning of the compression stroke. When this valve timing is used, the fresh air-fuel mixture charging rate shows practically no decrease because the action occurs when the intake stroke is more or less over. Sufficient burnt gas for NOx reduction rapidly flows into the cylinder because the in-cylinder pressure is negative. The thermal energy and turbulence kinetic energy of the burnt gas activate the air-fuel mixture in the cylinder. Therefore, this system does not require variable valve timing, and it was designed to use a solenoid to operate an EGR cam that is located on the same shaft as the exhaust camshaft. As a result, the system achieved favorable drivability while reducing NOx by approximately 50% and reducing CO by approximately 20% in the Euro 3 driving mode. Significant reduction of NOx also enabled a reduction of approximately 70% in the catalyst volume needed to satisfy Euro 3 exhaust emissions standards.

1. Introduction

As motorcycle exhaust emission standards have become more stringent every year, the reality of current exhaust emission reduction systems is that they are dependent on the conversion efficiency performance of the catalyst. It is necessary to increase the conversion efficiency of the catalyst in particular when greater NOx reduction is sought. Therefore, it is necessary to consider stoichiometric control of the air-fuel ratio, increasing the catalyst volume, the location of the catalyst inside the muffler, the temperature of the catalyst, and other such factors. As the result, the system often becomes expensive. Moreover, when the air-fuel ratio is increased in order to reduce fuel consumption, the NOx conversion efficiency of the catalyst decreases significantly. Consequently, lean-burn measures cannot be taken. Therefore, lower-cost technology to reduce NOx without depending on the catalyst becomes necessary⁽¹⁾⁻⁽³⁾.

In the present study, an internal exhaust gas recirculation (EGR) system with a two-stage opening of the exhaust valve has been designed and is of reasonable

cost. It opens and closes the exhaust valve slightly at the beginning of the compression stroke, or between the end of the intake stroke and the beginning of the compression stroke. This paper describes the NOx reduction effect of that system and the method used to achieve the effect.

2. Development Goals

The goals in development of this system are to reduce the volume of catalyst used to decrease exhaust emissions by means of a reduction in NOx emissions, and to satisfy both environmental performance and economic efficiency in small motorcycles. Therefore, the following items were set as requirements:

- (1) Activating the air-fuel mixture in the cylinder by the thermal and turbulence kinetic energy of the burnt gas, and reducing NOx significantly;
- (2) Not decreasing significantly the charging rate of the fresh air-fuel mixture by the inflow of the burnt gas by the EGR in order to obtain favorable drivability;
- (3) Establishing the low-cost internal EGR system with lightness and compactness.

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Table 1 Engine specifications

Items		Specifications
Engine type		4-stroke Single-cylinder
Cooling type		Liquid cooled
Bore × stroke	(mm)	50.0 × 55.0
Displacement	(cm ³)	108
Compression ratio		11.0
Valve train		SOHC
Valve diameter (mm)	Inlet	φ25.5 × 1
	Exhaust	φ21.0 × 1
Max. power	(kW/r/min)	6.6/7 500
Max. torque	(Nm/r/min)	9.3/6 000

3. Outline of Developed System

3.1. System Actuation Mechanism

Table 1 shows the main specifications of the present engine. Figure 1 shows a general view of the EGR cam actuation mechanism in this system. As shown in Fig. 1, simplicity of the valve train mechanism is realized by placing the EGR cam on the same shaft as the camshaft and causing it to slide in the shaft direction. Switching between the deactivated EGR cam (EGR OFF) and the activated EGR cam (EGR ON) is conducted by a solenoid through a link. The main items considered in this system layout are shown below.

- (1) By adding a link mechanism and giving it leverage, an appropriate EGR cam slide stroke could be obtained by the smaller stroke of the solenoid.
- (2) The EGR cam and the ordinary exhaust valve cam actuate the exhaust valve by means of the same rocker arm.

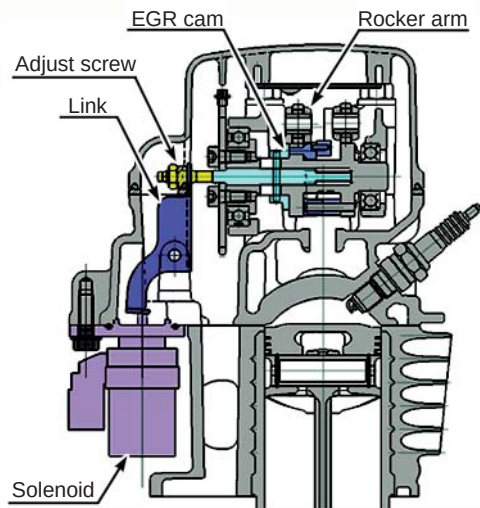


Fig. 1 EGR cam actuation mechanism

- (3) The cumulative clearance from the tolerances of each part is absorbed by adding an adjustment screw to the link mechanism.

Figure 2 shows the lift curves for the exhaust valve when the EGR cam is actuated.

3.2. Confirmation that System Enhances Combustion

3.2.1. Results of combustion analysis on a present engine

Figure 3 shows the in-cylinder pressure, the exhaust pipe pressure, and the intake pipe pressure in the case of EGR OFF and EGR ON.

This graph shows that when the EGR cam is actuated, the exhaust pipe pressure simultaneously decreases and the in-cylinder pressure increases. In other words, this makes it clear that the in-cylinder pressure is negative during actuation of the EGR cam, so that when the exhaust valve is opened by the EGR cam, burnt gas simultaneously flows into the cylinder from the exhaust port and causes the in-cylinder pressure to rise.

There is practically no increase in the intake manifold pressure, so essentially no burnt gas flows into the intake manifold. Consequently, it can be considered that the energy of the burnt gas is being used effectively to activate the air-fuel mixture in the cylinder.

The P-V diagram and the rate of heat release ($dQ/d\theta$) are shown in Figs. 4 and 5 in the case of EGR OFF and EGR ON. As is clear from these graphs, the in-cylinder pressure rises higher in the EGR ON state than in EGR OFF, and $dQ/d\theta$ also rises higher. Figure 6 shows the in-cylinder temperature in the EGR OFF and ON states. In EGR ON, even though the temperature rises during the compression stroke, the maximum temperature during the combustion stroke has been lowered.

As indicated by the above, the valve-open timing of the EGR cam in this system is set to the beginning of the compression stroke or the end of the intake stroke. The valve-close timing is set to the compression stroke, and it became clear that making this valve-close timing coincide with a state of negative pressure inside the cylinder causes combustion to be enhanced while the maximum temperature in the cylinder is reduced.

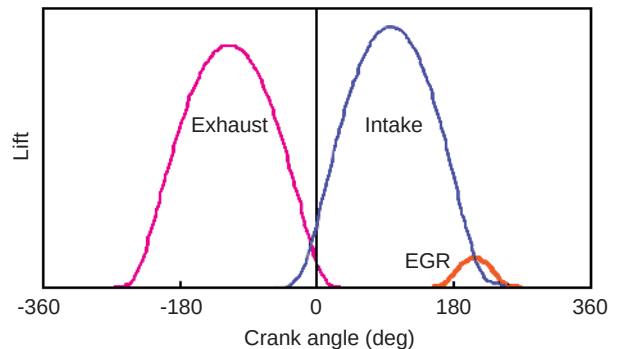


Fig. 2 Exhaust valve lift curves during actuation of EGR cam

3.2.2. Results of Computational Fluid Dynamics (CFD) Analysis of In-Cylinder Flow and Combustion

(1) Conditions for computation

In order to confirm the effects of the inflow of burnt gas and turbulence kinetic energy based on the test results in Sub-section 3.2.1, verification of the combustion analysis including in-cylinder flow was performed by computational fluid dynamics (CFD) analysis.

CFD computation was performed using commercially available code for 3D thermo-fluid analysis called FIRE (from AVL List GmbH). The standard $k-\epsilon$ model was applied for the turbulence model and the extended coherent flame model (ECFM) was applied for the combustion model. The initial conditions for computation and the boundary conditions were set using pressure and temperature measurement values for the various parts as obtained from present engine measurements. The fuel

sprayed from the injector was assumed to have completely vaporized in the intake port during the interval before the intake valve opens, and the process whereby fresh air-fuel that is homogeneous and single-phase (equivalence ratio=1) flows into the cylinder, mixes with the burnt gas,

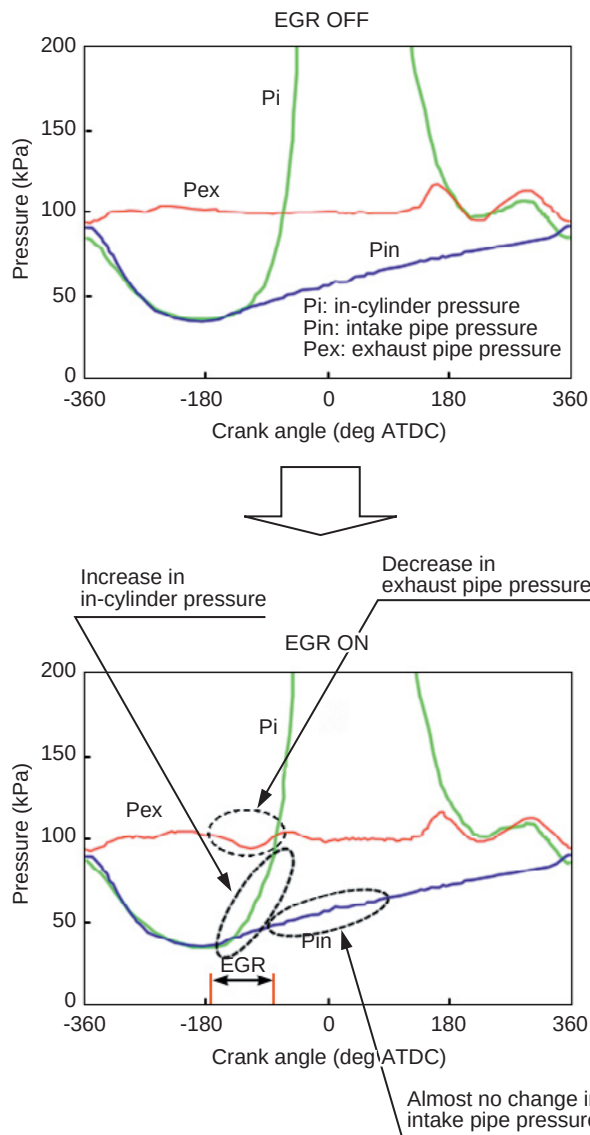


Fig. 3 Pressure applied to each part when EGR cam is in OFF and ON state

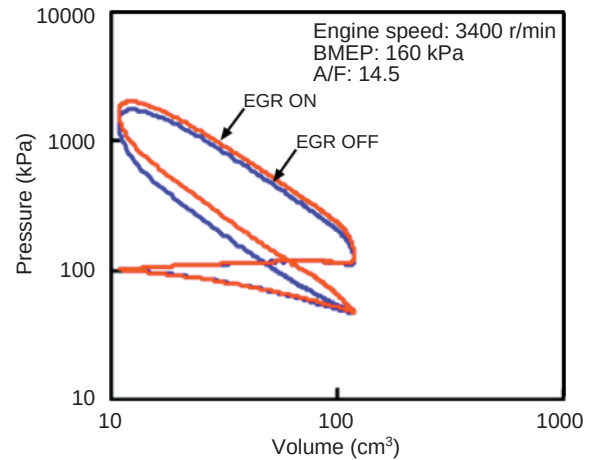


Fig. 4 P-V diagram when EGR cam is in OFF and ON state

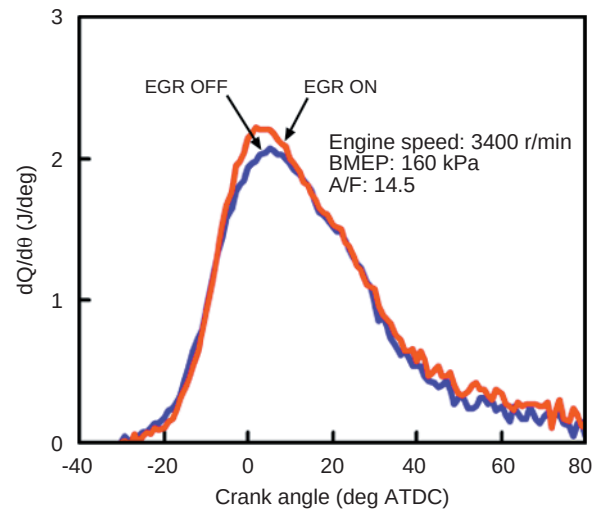


Fig. 5 $dQ/d\theta$ when EGR cam is in OFF and ON state

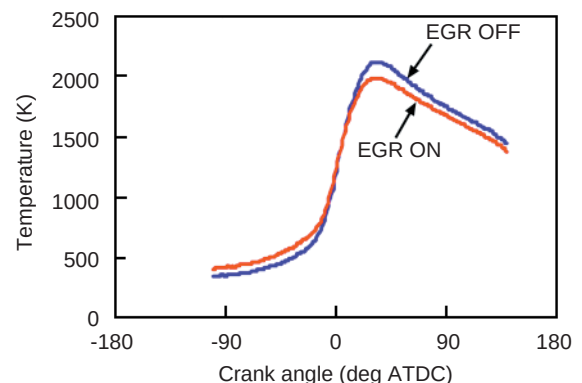


Fig. 6 In-cylinder temperature when EGR cam is in OFF and ON state

and undergoes combustion was computed.

(2) EGR gas distribution

Figure 7 presents views of burnt gas from the exhaust pipe flowing into the cylinder and EGR gas distribution with EGR OFF and ON. When EGR is ON, the exhaust valve is opened again by the EGR cam with timing at the beginning of the compression stroke (from -160 to -100 deg ATDC). It is clear that the burnt gas in the exhaust pipe flows into the cylinder along the cylinder wall and the EGR ratio in the cylinder increases. This enabled confirmation in greater detail of the phenomenon of burnt gas inflow to the cylinder that can be extrapolated from the various pressure measurement values obtained when EGR is actuated.

(3) Turbulence kinetic energy in the cylinder

Figure 8 shows the turbulence kinetic energy around the spark plug with EGR OFF and ON. Figure 9 shows

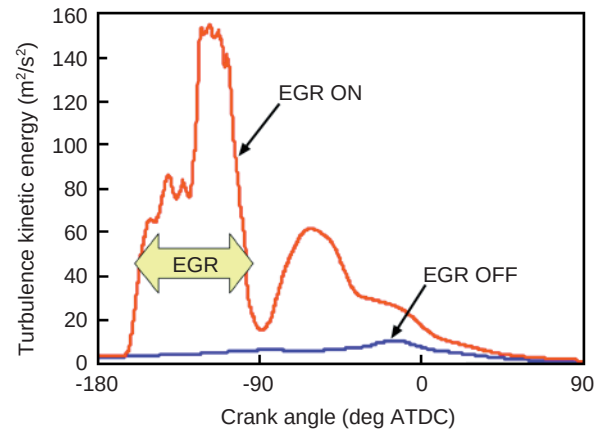


Fig. 8 CFD analysis of turbulence kinetic energy around spark plug

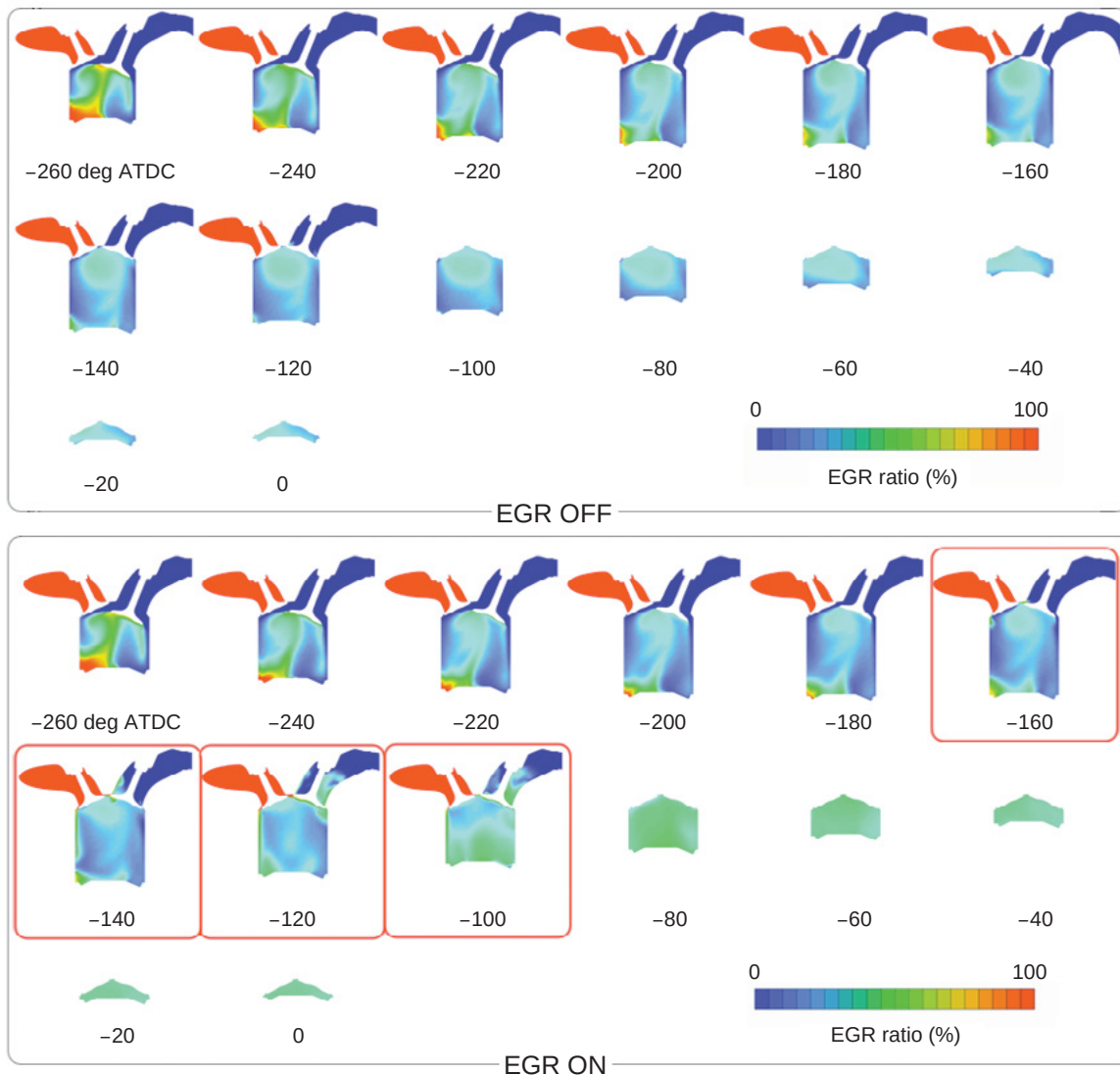


Fig. 7 CFD analysis of EGR

the average turbulence kinetic energy in the cylinder. It is clear that when EGR is ON, the turbulence kinetic energy in the cylinder increases. When EGR is actuated, the burnt gas that flows in due to the difference between the pressure in the exhaust pipe and in the cylinder becomes the source of the increase in turbulence kinetic energy. Thus, the changes in turbulence kinetic energy, which is considered one factor of enhanced combustion in this system, could be confirmed in greater detail.

(4) Flame propagation

Figure 10 shows flame propagation with EGR OFF and ON. It is clear that although some ignition delay occurs when EGR is ON, flame propagation occurs rapidly after ignition. This is thought to be due to the turbulence kinetic energy in the cylinder from EGR gas inflow and the increased thermal energy from the internal

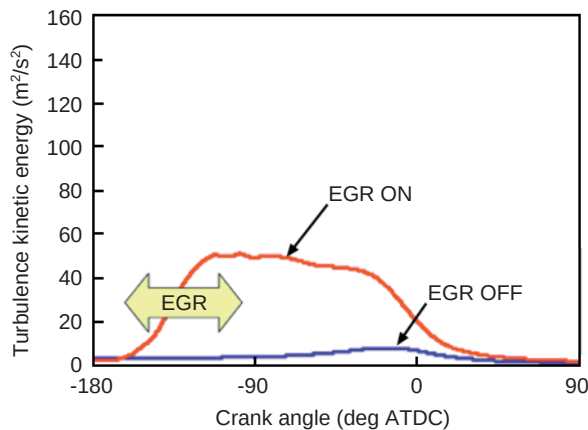


Fig. 9 CFD analysis of average turbulence kinetic energy in cylinder

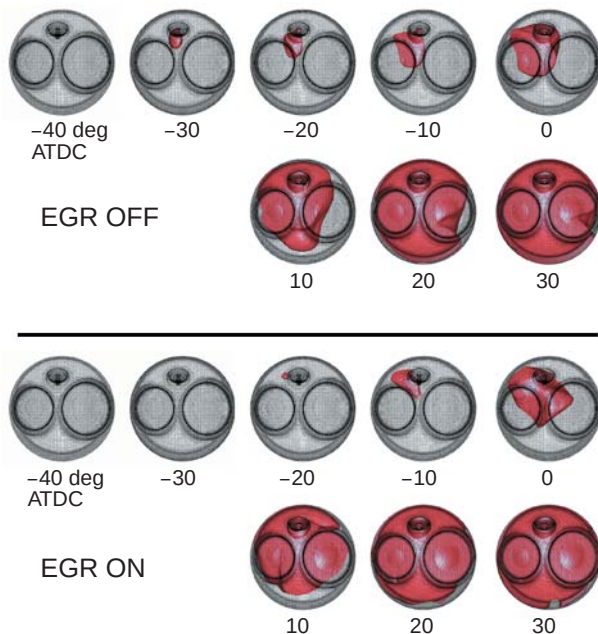


Fig. 10 CFD analysis of flame propagation

EGR. It is clear that combustion is enhanced despite the higher EGR ratio, just as in the test results.

(5) In-cylinder temperature

Figure 11 shows changes in in-cylinder temperature during the combustion stroke with EGR OFF and ON.

It is clear that when EGR is ON, the in-cylinder temperature during the combustion stroke is reduced.

These results confirmed that the system can enhance combustion while reducing the combustion temperature even in CFD analysis.

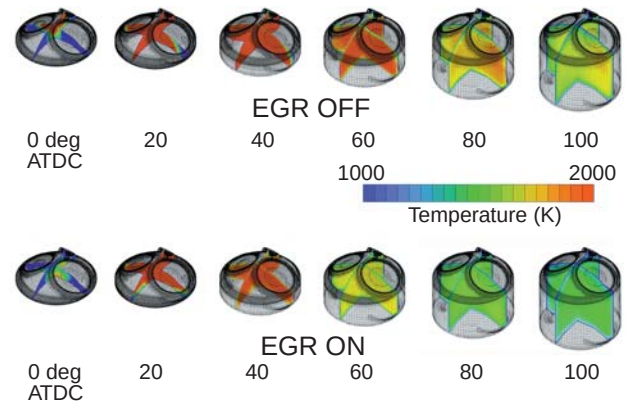


Fig. 11 CFD analysis of in-cylinder temperature

4. Configuring the EGR Cam Actuation Characteristics

4.1. Actuation Timing

Figure 12 shows the exhaust valve lift curves for each specification (1, 2, 3) in which the EGR cam lift is kept constant and the actuating timing is varied. The engine speed and load were also changed in each specification and the resulting NOx was measured. Those measurement points (a, b, c, d, e, and f) are shown in Fig. 13, and the results for the NOx reduction rates obtained due to EGR at each measurement point are shown in Fig. 14.

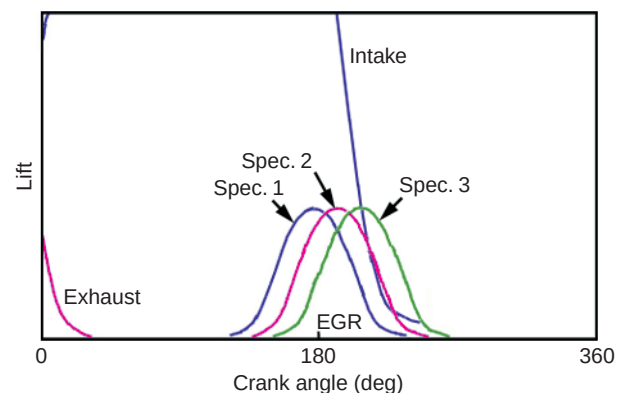


Fig. 12 Exhaust valve lift curves by difference in actuating timings of EGR cam

All the specifications are alike in obtaining an NOx reduction rate that is higher at lower speeds and lower loads. This is because the magnitude of the negative pressure inside the cylinder is greater when the EGR cam is actuated, thereby increasing the volume of burnt gas that flows into the cylinder. On the other hand, the NOx reduction rate is lower when the speed and load are higher. This is because the magnitude of the negative pressure inside the cylinder is lower when the EGR cam is actuated, decreasing the volume of burnt gas that flows into the cylinder.

In specifications 2 and 3, however, NOx was reduced in the high-speed, high-load region (measurement point f). This is because, with the EGR cam timing for closing the valve, the in-cylinder pressure was greater than the exhaust pipe pressure, so that the air-fuel mixture in the cylinder escaped out of the cylinder to the exhaust port. As a result, the reduction in compression pressure caused combustion to deteriorate. Figure 15 shows the in-cylinder pressure, exhaust pipe pressure, and intake manifold pressure at that time. Figure 16 shows $dQ/d\theta$ as derived from the in-cylinder pressure. It is also clear from these graphs that combustion had deteriorated.

It is apparent from these results that specification 1

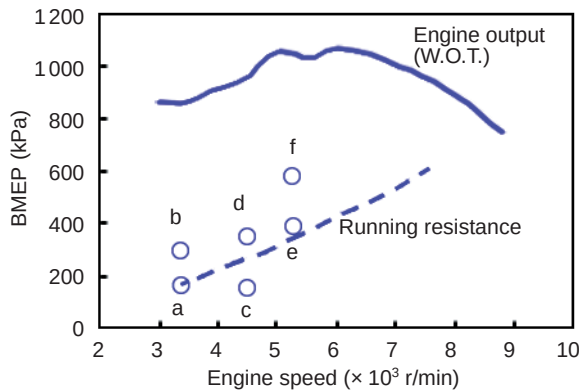


Fig. 13 EGR effect verification points

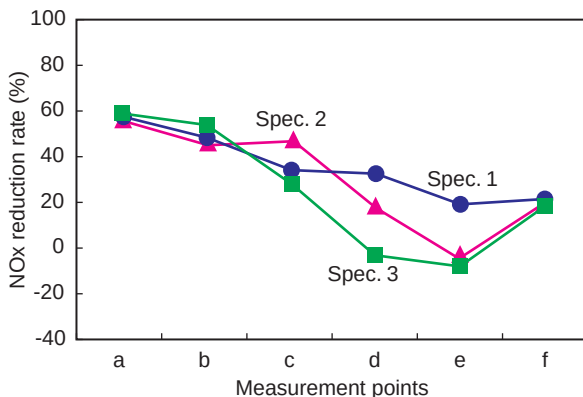


Fig. 14 NOx reduction rate by difference in actuating timings of EGR cam

offers better EGR cam actuating timing for obtaining the NOx reduction effect from low load to high load without causing combustion to deteriorate.

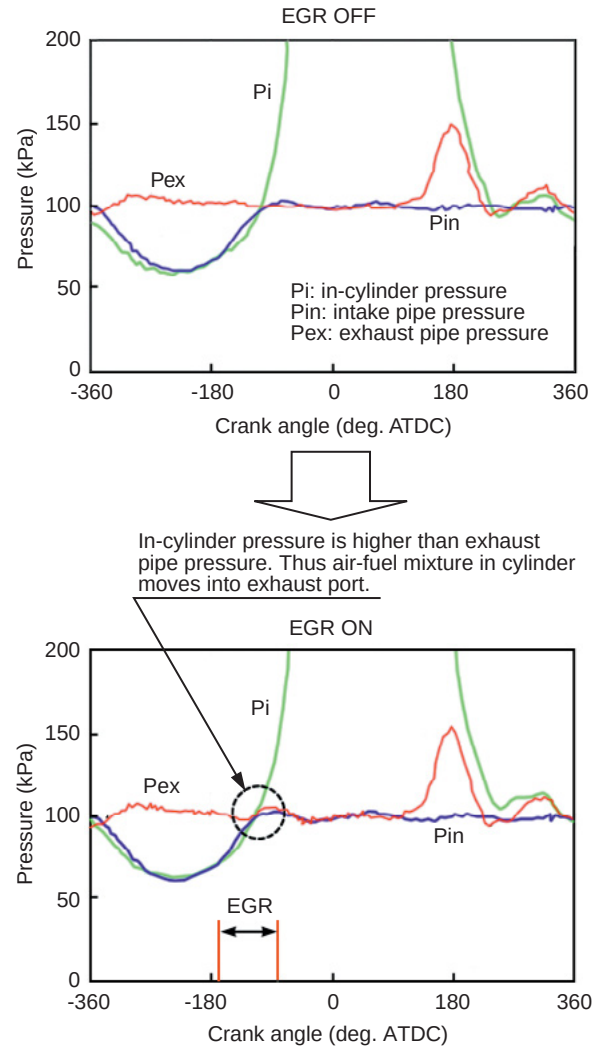


Fig. 15 Pressure applied to each part at measurement point "f" in specification 3

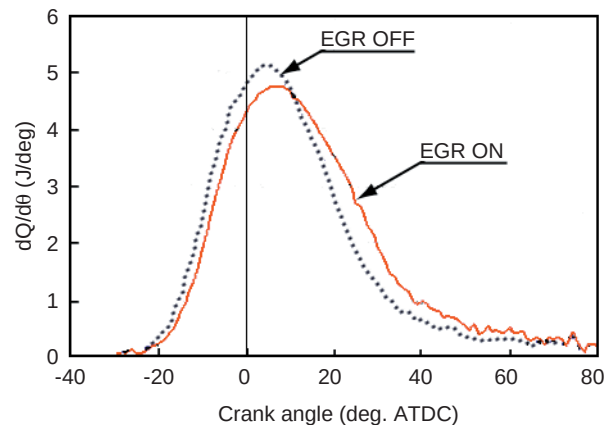


Fig. 16 $dQ/d\theta$ at measurement point "f" in specification 3

4.2. Actuation Lift

Figure 17 shows the exhaust valve lift curves for specifications (A, B, C, and D) in which the EGR cam closing timing was fixed (specification 1 in section 4.1) and the lift height was changed. For each of these specifications, the engine speed and load were also changed and the resulting NOx was measured. The measurement points were those points shown in Fig. 13, and the NOx reduction rate results due to EGR are shown in Fig. 18.

With low lift (specification A), the opening area of the exhaust valve was smaller so that the amount of burnt gas that flowed into the cylinder was smaller. Consequently, the NOx reduction rate was also smaller. On the other hand, with high lift (specification D), the NOx reduction rate became high over all of the measurement points used. This was due to the opening area of the exhaust valve being larger so that the amount of burnt gas that flowed into the cylinder became large. In the low-speed and low-load region (measurement points a and c), however, combustion deteriorated and the amount of HC increased, as shown in Fig. 19, because an excessive amount of burnt gas flowed into the cylinder.

It became clear from the above that the optimum EGR cam specification for the purpose of reducing NOx in the present test engine was specification 1 in terms of EGR cam actuating timing and specification C in terms of lift. It was decided to provide ON/OFF control of EGR cam actuation based on engine speed (N_e) and throttle position (θ_{th}).

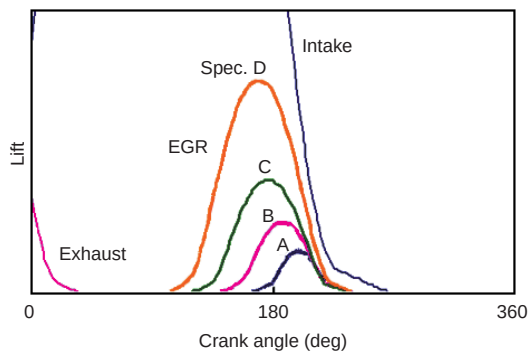


Fig. 17 Exhaust valve lift curves by difference in EGR cam lifts

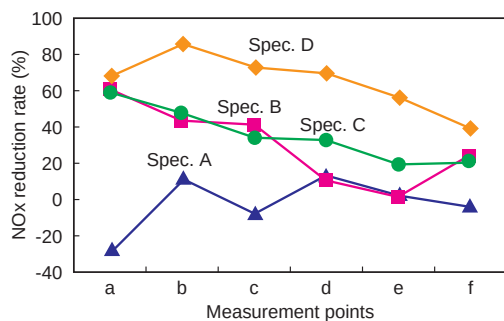


Fig. 18 NOx reduction rate by difference in EGR cam lifts

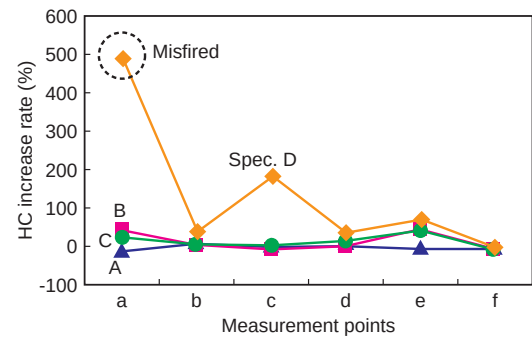


Fig. 19 HC increase rate by difference in EGR cam lifts

5. Exhaust Emission Performance

This system was used to confirm exhaust emissions on a vehicle. Figure 20 shows the test results for driving mode under exhaust emission standards (Euro 3) without the catalyst. It is clear from this graph that NOx and CO are reduced, not only when running at steady speed but also when accelerating and decelerating.

Figure 21 shows the total exhaust emissions in Euro 3 driving mode without the catalyst. As the graph indicates, NOx is reduced by approximately 50% with this

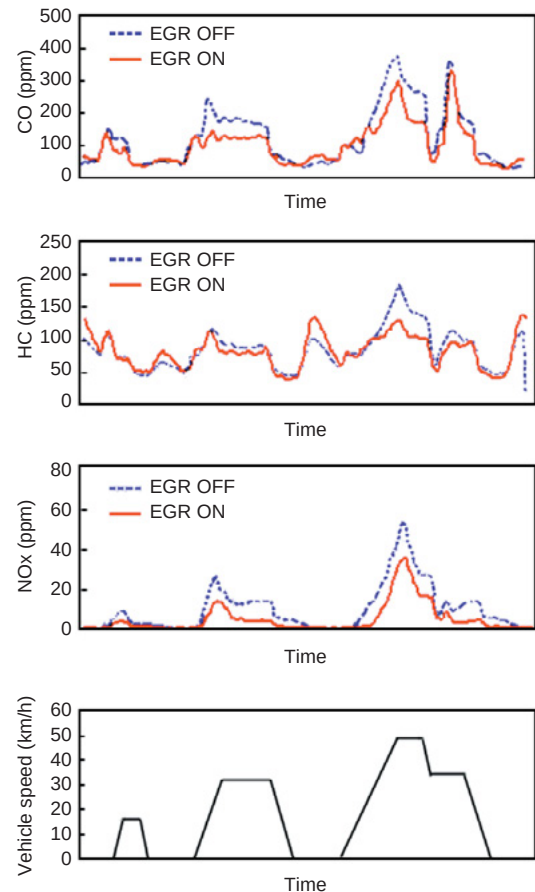


Fig. 20 Time history of exhaust emission in Euro 3 driving mode

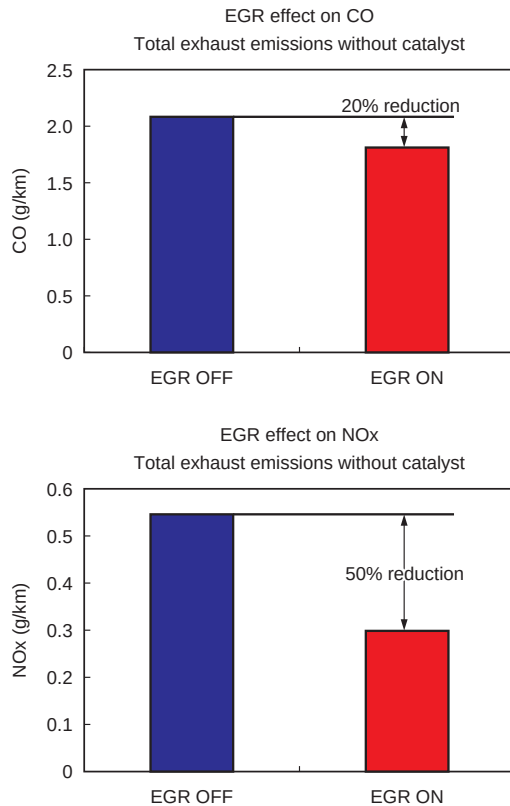


Fig. 21 Exhaust emission reduction effect by developed system in Euro 3 driving mode

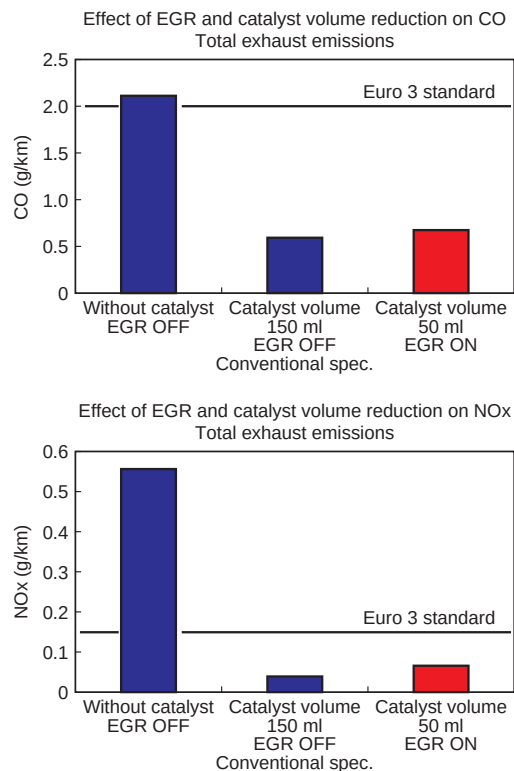


Fig. 22 Catalyst volume reduction effect by developed system in Euro 3 driving mode

system and CO is reduced by approximately 20%. Exhaust emissions were then measured in Euro 3 driving mode with the catalyst volume reduced. Figure 22 shows that the catalyst volume needed to satisfy Euro 3 exhaust emissions standards can be reduced by approximately 70% from the volume that was formerly needed.

6. Conclusion

The internal EGR system that was devised as described here yielded the following findings and results:

- (1) It is preferable to delay actuation by the EGR cam as much as possible so that the valve will open and close when the in-cylinder pressure is negative;
- (2) Exhaust emissions values in Euro 3 driving mode showed that NOx is reduced by approximately 50% and CO is reduced by approximately 20%;
- (3) The significant reduction in NOx enabled the catalyst volume needed to satisfy Euro 3 exhaust emissions standards to be reduced by approximately 70%.

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Teardown Analysis for Piston Assembly Friction with Optimization Controls

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ABSTRACT

This research involved creating a method for analyzing the contribution of the skirt and rings to the friction force of the piston assembly, and implementing such analysis. The contribution analysis was carried out by identifying the respective Stribeck curves for the skirt and the rings. The results showed that the skirt is in the fluid-lubricated region and the rings are in the mixed-lubricated region of the curve. The results also made clear that the friction force of the skirt is the dominant component under high-speed, low-load conditions, in particular.

1. Introduction

There is demand for further enhanced fuel efficiency in automobile engines in order to contain global warming. The enhancement of fuel efficiency necessitates a reduction in friction loss together with an enhancement of indicated thermal efficiency. It has been reported that friction loss in the pistons and connecting rods accounts for about one-third of total friction loss⁽¹⁾, and the reduction of piston friction loss, in particular, has been treated as an important issue and subjected to measurement and analysis many times.

A number of methods for measuring piston assembly friction have been implemented by holding a sleeve in a floating mounting and measuring the load that is applied^{(2), (3)}. However, the friction is measured as the resultant forces on the skirt and the rings, so that the contribution of each part is not clearly identified. By contrast, there are examples of individual analysis of the separate parts, such as friction measurement of rings by bench testing⁽⁴⁾, calculation using CAE⁽⁴⁾⁻⁽⁶⁾, and the like, but there have not been any reports of analysis made during actual operation. These issues can be resolved by performing direct teardown analysis of actual measurements made of piston assembly friction.

This research presents a method for identifying Stribeck curves to tear down the measured piston assembly friction into the portion that is generated in the skirt and the portion generated in the rings. This was used to calculate the friction distribution during actual

operation. The curves obtained in this way were then used to examine the influence of engine speed and load on friction loss separately in the skirt and rings and analyze their contribution to piston friction loss. This aids in reducing friction loss.

2. Test Device

In order to measure piston assembly friction, a single-piston engine with a sleeve held in a floating mounting was fabricated. Table 1 and Fig. 1 show the specifications and a sectional view of the fabricated engine.

A triaxial force sensor was used as the sleeve support structure, with reference to the research of Itoh et al.⁽³⁾, because it possesses adequate rigidity and allows measurement up to the high speed range. A total of six triaxial force sensors were disposed [(A) in Fig. 1] on both intake and exhaust sides in order to assure rigidity and prevent the sleeve from tilting. This applies a tensile load on both sides of the sleeve, so that secondary deformation could occur in the bore. In order to prevent this, the sensor holder [(B) in Fig. 1] was made independent of the block.

The combustion chamber seal was formed in the space between the sleeve and the head stuck into the cylinder. This enables prevention of the direct application of cylinder pressure against the upper surface of the sleeve. In this case, however, the use of O rings for the seal has been reported⁽⁷⁾ to result in load transmission to the sleeve

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Table 1 Engine specifications

Engine type	4-stroke
Number of cylinders	1
Bore (mm) × Stroke (mm)	86 × 86
Compression ratio	9.8
Crank offset (mm)	0.0
Cooling system	Water - cooled
Camshaft	SOHC
Valves (IN/EX)	1/1
Lubrication oil	SAE 0W-20 (80±1°C)

due to the head deformation caused by combustion pressure. Therefore, a seal ring with a shape corresponding to the piston rings was used for the seal so as to realize a balance between providing a cylinder pressure seal and preventing load transmission.

The sleeve was given a structure with two layers, inner and outer, to assure the circularity of the bore as well as to reduce weight and enhance maintainability. A flange added on the top surface was used to fasten the sleeve, and the gaps were made into cooling water passages [(C) in Fig. 1].

Figure 2 shows the measuring system.

The friction was calculated by subtracting the inertial

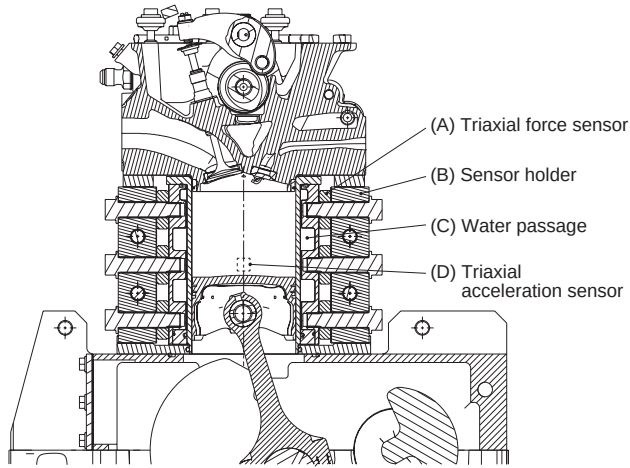


Fig. 1 Engine cross section

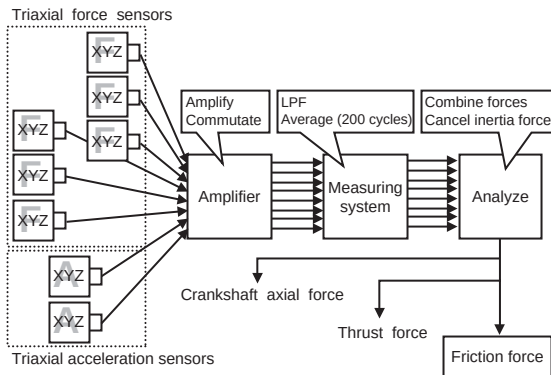


Fig. 2 Friction measuring system

force of the sleeve from the sum of the triaxial force sensor output values. That inertial force was calculated by the values from the two acceleration sensors [(D) in Fig. 1], which were placed so as to pass through the sleeve center of gravity.

3. Identification of the Stribeck Curve

Piston assembly friction F is expanded as shown below, as the sum of the friction of the skirt (hereafter subscript s) and the friction of the rings (hereafter subscript r).

$$F = F_s + \sum F_r \quad (1)$$

The various forces of friction can be represented as products of the coefficient of friction μ and the normal force N . Clarifying these characteristics of μ enables teardown of the friction forces as shown in Eq. (1).

3.1. Stribeck Curve Identification Method

It is known that the coefficient of friction μ on the one hand, together with the viscosity η , speed v , and surface pressure p on the other, can be used to generate a Stribeck curve with characteristics as shown in Fig. 3. In this research, the following three items were used as the control factors for this function.

- (1) The coefficient of friction gradient in the fluid-lubricated region: a
- (2) The minimum coefficient of friction in the mixed-lubricated region: b
- (3) The boundary coefficient of friction: c

The lubricant was taken to be a Newtonian fluid, the diametric clearance during the stroke was taken to be fixed, and the fluid-lubricated region was postulated to be a first-order function that passes through the origin. The mixed-lubricated region was postulated to be a second-order function that passes through minimum coefficient of friction b and boundary coefficient of friction c and links them smoothly.

If the skirt and rings are considered to have their own respective curves, they can be expressed as shown below.

$$(\text{Skirt}) \quad \mu_s(\eta, v, p) = \mu_{a_s, b_s, c_s}(\eta, v, p) \quad (2)$$

$$(\text{Rings}) \quad \mu_r(\eta, v, p) = \mu_{a_r, b_r, c_r}(\eta, v, p) \quad (3)$$

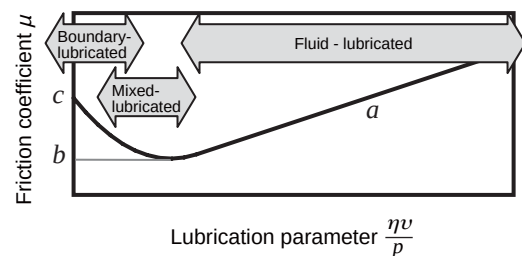


Fig. 3 Stribeck curve

If it is postulated that the normal force that acts on the skirt surface is equal to the thrust load that acts on the piston, and that the normal force that acts on the surface of the outer circumference of the rings is determined by the tensile force and pressure on the reverse surface of the rings, then surface pressure p and the normal force N that are necessary when calculating μ can be expressed as in the following.

$$\text{(Skirt)} \quad p_s = \frac{N_s}{A_s} \quad (4)$$

$$N_s = F_{th} \quad (5)$$

$$\text{(Rings)} \quad p_{r_i} = \frac{N_{r_i}}{\pi D h_{1_i}} \quad (6)$$

$$N_{r_i} = 2\pi F_{t_i} + \pi D h_{1_i} p_{g_i} \quad (7)$$

A_s is the area of one skirt side, F_{th} is the thrust load, D is the bore diameter, h_1 is a ring width, F_t is the ring tension, and p_g is the ring groove pressure, where subscript i (1 to 3) identify the rings. These together with piston speed v and oil viscosity η , are used to express friction F as in the following.

$$F = \mu_s(\eta, v, p_s)N_s + \sum_{i=1}^3 (\mu_r(\eta, v, p_{r_i})N_{r_i}) \quad (8)$$

As is apparent from Eqs. (5) and (7), N_s and N_r here have different factors. Using these to make F match the actual measurement value F_{measured} for freely selected conditions and crank angles enables determination of $\mu_s(\cdot)$ and $\mu_r(\cdot)$. The mean square error, ε_f , with respect to the measured value, with the number of actual measurement points as m , will be:

$$\varepsilon_f = \sqrt{\frac{1}{m} \sum_{j=1}^m (F_j - F_{j \text{ measured}})^2} \quad (9)$$

In order to minimize this error ε_f , optimization analysis using variables a_s - c_r was performed and the Stribeck curve functions $\mu_s(\cdot)$ and $\mu_r(\cdot)$ were identified. Figure 4 shows the above system.

3.2. Stribeck Curve Identification and Results

Results under the following two conditions were used to identify the curves.

- (1) Motored at 1500 rpm, intake manifold absolute pressure 25 kPa ($P_{\text{max}} = 0.4$ MPa)
- (2) Motored at 2500 rpm, intake manifold absolute pressure 100 kPa ($P_{\text{max}} = 2.1$ MPa)

In order to inhibit the effects of vibrational components, the components were discretized at every 30 deg of crank angle, as shown in Fig. 5. The data from all 23 points (a total of 46 points) were used. This

excluded the point immediately after compression top dead center, where vibration is great.

The parameters a_s - c_r were subjected to optimization analysis in order to satisfy these discretized empirical values. Simulated annealing⁽⁸⁾ was used for the optimization.

Figure 6 shows the Stribeck curves obtained.

The coefficients of boundary friction were approximately 0.09 for the skirt and approximately 0.04 for the rings. The coefficient of friction for the rings was smaller, about half the figure for the skirt. Gradient a_r in the fluid-lubricated region was smaller, at about one-tenth the figure for the skirt. This is thought to be because the real contact width of the rings was smaller than the nominal width, h_1 , so that the real surface pressure became greater than the nominal value.

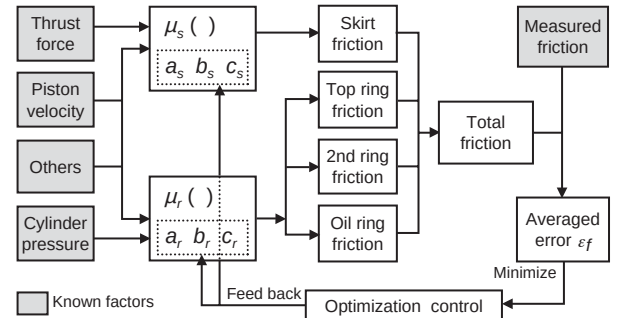


Fig. 4 System for identifying Stribeck curves

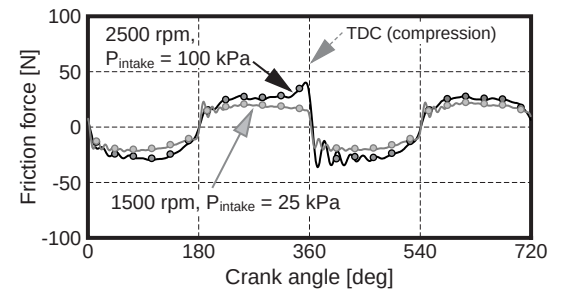


Fig. 5 Discretized empirical data

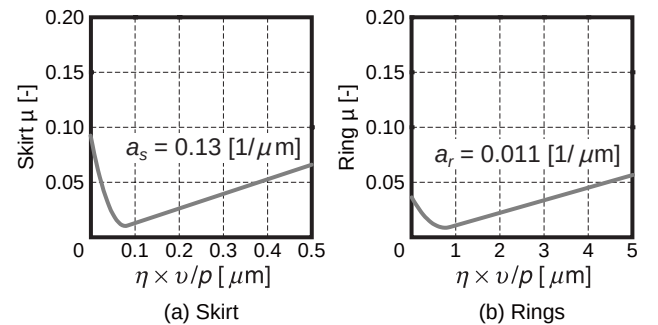
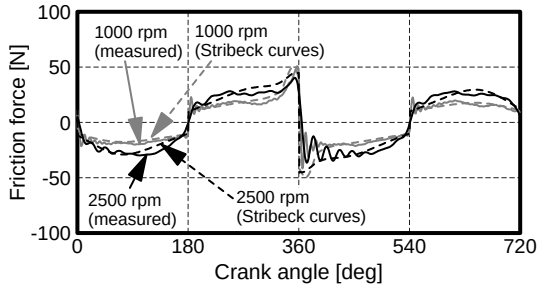


Fig. 6 Stribeck curves for skirt and rings

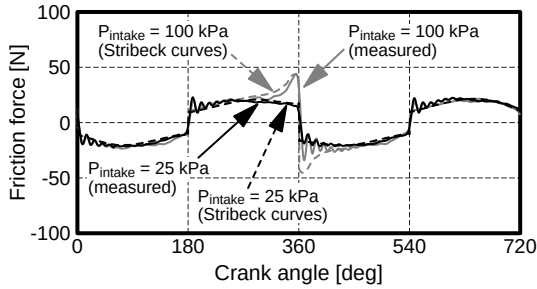
3.3. Verification of the Validity of Stribeck Curves

The friction forces calculated from Stribeck curves for multiple conditions were compared with the empirical results to verify the validity of the curves. Figure 7 shows the results.

This clearly illustrates the way in which friction force in the piston high speed region changes with engine speed [Fig. 7(a)] and the distinctive phenomena whereby friction force increases near compression top dead center due to differences in cylinder pressure [Fig. 7(b)]. The validity of the Stribeck curves obtained could therefore be confirmed.



(a) Influence of speed (motored, intake pressure = 100 kPa)



(b) Influence of intake pressure (1500 rpm motored)

Fig. 7 Validation of Stribeck curves

4. Contribution Analysis of Piston Assembly Friction

Stribeck curves $\mu_s()$ and $\mu_r()$, obtained using Eq. (8), were applied to perform contribution analysis of the friction forces.

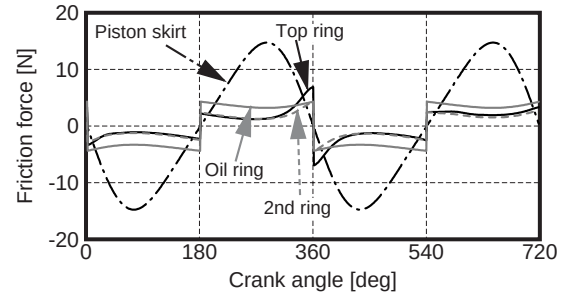
4.1. Contribution Analysis of Actual Friction Measurements

An example of the results obtained through teardown analysis of actual friction measurements together with the application areas of the Stribeck curves at that time are shown in Figs. 8 and 9. The friction loss distribution is shown in Fig. 10.

It is apparent from Fig. 8 that friction in the skirt occurs in proportion to speed, and from the application

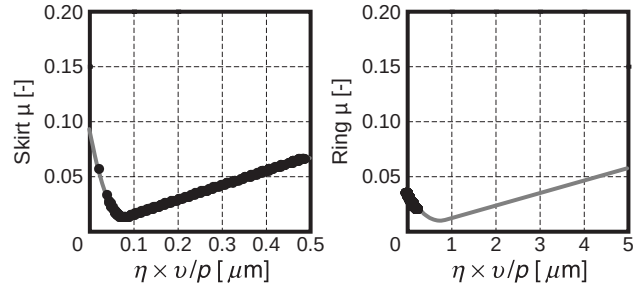
area of the curve in Fig. 9(a) that the friction is occurring under adequately fluid-lubricated conditions. In Fig. 8, by contrast, the friction waveforms for the rings show largely fixed values, so that it can be predicted that the boundary-lubricated condition will be dominant. This can also be confirmed from the results in Fig. 9(b).

It was also apparent from the friction loss distribution in Fig. 10 that the skirt accounts for approximately half of the piston friction loss under these conditions. This was followed by the large influence from the top ring, which is susceptible to the influence of cylinder pressure, and from the oil ring, which experiences high tensile force.



(1500 rpm motored, intake pressure = 25 kPa)

Fig. 8 Friction distribution

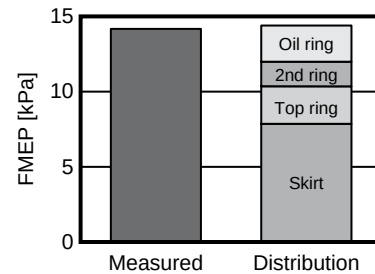


(a) Skirt

(b) Rings

(1500 rpm motored, intake pressure = 25 kPa)

Fig. 9 Application area of Stribeck curves



(1500 rpm motored, intake pressure = 25 kPa)

Fig. 10 FMEP distribution

4.2. Changes in Friction Loss and Contribution due to Engine Speed and Load

In order to confirm how the contribution to friction loss changes due to engine speed and load, calculations were performed to determine friction loss when engine speed and intake pressure were changed. Figure 11 shows the results.

Examination of Fig. 11 (a) shows that, because the skirt is in the fluid-lubricated region, friction loss increases in proportion to engine speed. It is also apparent, from Fig. 11(b), that there is less friction loss in the rings than in the skirt, and that it diminishes still further as engine speed increases because it more closely approaches the fluid-lubricated region. Friction loss increases when the load increases because of the increase in the load on the reverse surface of the rings due to cylinder pressure.

Figure 12 shows friction loss in the piston as a whole and its percentages in the skirt.

Friction loss in the whole increases as engine speed becomes higher and load becomes greater. Moreover, it was ascertained that the percentage of friction loss occurring in the skirt is at least 50% more or less across the entire range. Under conditions of high engine speed and low load, in particular, it accounts for 90%.

In this research, the actual measurement results were used to examine the piston assembly friction and ring friction separately. The following are considered necessary in order to further enhance accuracy:

- (1) Identification of curves using results from fired conditions
- (2) Methods to make curve shapes more closely resemble actual operating conditions

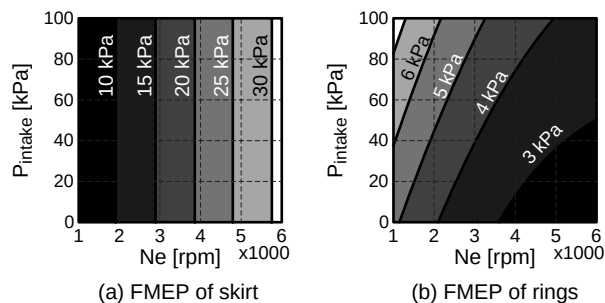


Fig. 11 FMEP maps of skirt and rings

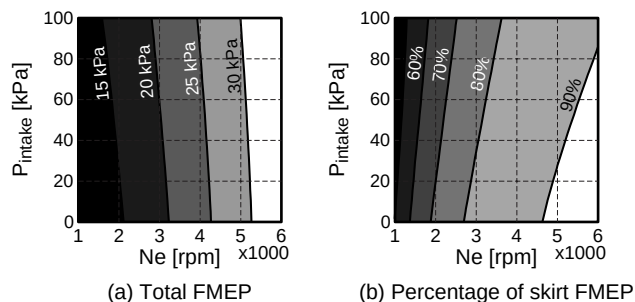


Fig. 12 Total FMEP and skirt percentage

- (3) Introduction of independent curves for each ring

It is also thought that this method could be applied, for example, so that friction measurement results for an entire engine could be used to analyze the contribution of each individual component without making use of motored teardown methods.

5. Conclusion

Analysis of friction was performed using empirical results with the aim of reducing piston friction loss, and the following findings were obtained:

- (1) A method of identifying Stribeck curves on the basis of empirical results was devised, and it yielded good matching results with empirical data under a variety of conditions.
- (2) Use of the above method succeeded in tearing down the actual measured friction into the portion contributed by the skirt and the portion contributed by the rings. As a result, it was confirmed that in this engine, the fluid-lubricated condition was dominant in the skirt while the rings were mainly under boundary-lubricated or mixed-lubricated conditions.
- (3) Analysis of the influence on skirt friction and ring friction related to engine speed and load confirmed that skirt friction is strongly influenced by engine speed and that ring friction is strongly influenced by engine speed and load. The percentage of friction loss in the skirt was found to be about 50-90%, and the percentage of skirt friction loss was found to increase as the engine speed increased and the load diminished.

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Static Torque Sharing Distribution of Planetary Gear Sets Generated by Manufacturing Error Distribution

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ABSTRACT

Creating an appropriate design for gear strength in a planetary gear set requires that manufacturing errors be grasped statistically and that the distribution of torque to the planet gears due to manufacturing errors be projected. This article describes the derivation of an expression of relation between manufacturing errors and the torque applied to planet gears in a common n-planet planetary gear set. As a result, a formula expressing the distribution of torque to the planet gears could be obtained. Tests were also conducted to verify the relational expression, and its validity was confirmed. In order to project the distribution of torque, it was necessary to estimate the stiffness of the planetary gear set in the relational expression. Those stiffness values were calculated by numerical analysis using a 3D finite element model (FEM) to which the plan values and material property values were entered. The accuracy of the estimated stiffness values was verified by testing, and it was confirmed that the stiffness values could be estimated with a high degree of accuracy. Given the above, the distribution of torque to planet gears due to manufacturing error in a common n-planet planetary gear set could be estimated with a high degree of accuracy using only estimated data obtained by desktop methods.

1. Introduction

Because the loads input to planetary gear sets are distributed between multiple planet gears before being output, the use of these gear sets can reduce the size and weight of the automatic transmission. Ideally, input torque should be distributed equally among the planet gears and transmitted. However, in actual practice planetary gears may be unable to effectively demonstrate their advantages because manufacturing errors would result in an uneven distribution of torque to each gear.

Because manufacturing errors differ depending on the product, the ratio of torque distributed to each planet gear in a planetary gear set also differs depending on the product. Given this, when the entire set of manufactured products is considered statistically, the collection of the torque of a planet gear, which shares a given arbitrary torque input to a planetary gear set with other planet gears, shows a distribution in response to the distribution of manufacturing errors. It is therefore necessary to take manufacturing error into consideration when designing the strength of the gears used in planetary gear sets.

Conventionally, in strength design for gears used in planetary gear sets, the effect of production errors has

been considered based on experiential evidence of the safety factor, and the gears have tended to be manufactured with excessive quality. If it was possible to predict the distribution of torque to the planet gears from the distribution of production errors, the safety factor could be optimized, and it would be possible to design gears that possessed the necessary and sufficient strength.

Dimensional error in machining is known to generally display a normal distribution, and the dimensions of the gears are managed using blueprint values based on this fact. Because of this, the distribution of torque to the planet gears in relation to the distribution of production errors could be known if the relationship between any arbitrary production error that affects the distribution of torque between the planet gears and the torque on each of the gears was known theoretically. In addition, if the theoretical relationship between production errors and the torque on each planet gear was known, it would be possible to predict the distribution of torque to the planet gears without conducting strength analyses for a number of arbitrarily-selected combinations of production errors.

An equation expressing the relationship between an arbitrary production error and the torque on the planet gears has been derived for 4-planet planetary gear sets⁽¹⁾,

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but not for normal n -planet planetary gear sets. In addition, the validity of the relational expression has not been verified.

Because the torque capacity of planetary gear sets can be increased by increasing the number of planet gears, planetary gear sets using five or more gears are now in use in vehicles in order to achieve increased torque capacity. Given this, an equation expressing the relationship between any arbitrary production error and the torque on the planet gears is necessary for planetary gear sets containing five planet gears or more.

This paper therefore discusses the derivation of a relational expression between any arbitrary production error and the torque on the planet gears for a normal n -planet planetary gear set. The validity of this equation was verified in tests.

2. Relationship between Eccentric Error and Load

Hayashi et al.⁽¹⁾ and Hidaka et al.^{(2),(3)} formulated linear models of planetary gear sets with multiple degrees of freedom in the transverse plane, and conducted analyses of the relationship between various types of error and the torque distribution between the planet gears by defining the non-uniformity of the torque distribution between the gears resulting from the various types of error as differences in the timing of the commencement of meshing between the gears produced by the errors, and converting the different production errors to transmission errors between the gears. Following this methodology, in the project discussed here the author formulated a linear model of a planetary gear set in the transverse plane, assuming an proper design in which the distribution of the tooth-contact pattern for each gear in the set was close to the centers of the teeth. Converting production errors into equivalent transmission errors and focusing on the self-centering action in the planetary gear set resulted in the derivation of an expression of relation between production errors and the torque on each planetary gear in a planetary gear set containing n normal planet gears. When production errors were converted to equivalent transmission errors, the production errors were sufficiently small in relation to the distance between the axes of the gears so that the change in the line of action due to the production errors could be ignored. The planetary gear set studied in the project was assumed to display sufficient clearance in the radial direction, and its self-centering action was assumed to function perfectly. Because the production errors were converted to equivalent transmission errors using the same scale of measurement, they are treated below as a single category of production error. Eccentric errors in the positioning of the holes for the pins that support the planet gears are known to be the errors that have the greatest effect on the distribution of torque between the gears⁽⁴⁾, and production errors are

therefore treated here as eccentric errors in the positioning of the pin holes.

2.1. Eccentric Errors and Equivalent Transmission Errors

For a planetary gear set with n planet gears, the equivalent transmission error for the meshing between the j th planet gear and the sun gear, ε_{SPj} , and the equivalent transmission error for the meshing between the j th planet gear and the internal gear, ε_{IPj} , both produced by an eccentric error in the positioning of the hole for the support pin for the j th planet gear, can be expressed as follows:

$$\varepsilon_{SPj} = -\vec{d}_{pj} \cdot \vec{i}_{SPj} \quad (1)$$

$$\varepsilon_{IPj} = -\vec{d}_{pj} \cdot \vec{i}_{IPj} \quad (2)$$

where

$\vec{d}_{pj}$: Eccentric error in positioning of hole for support pin for j th planet gear

$\vec{i}_{SPj}$: Unit vector in line of action of sun gear and j th planet gear (from sun gear towards planet gear = positive)

$\vec{i}_{IPj}$: Unit vector in direction of line of action of internal gear and j th planet gear (from internal gear towards planet gear = positive)

The symbol $\cdot$ in the equations represents the inner product of the vector.

The equivalent transmission errors ε_{SPj} and ε_{IPj} take on positive values when the eccentric error of the pin hole position is located in the direction of separation of the gear teeth. The sum of the equivalent transmission errors for the meshing of the j th planet gear, the sun gear, and the internal gear is expressed as ε_{pj} .

$$\varepsilon_{pj} = -a_p \vec{d}_{pj} \cdot \vec{i}_{pj} \quad (3)$$

Here,

$$\vec{a}_{pj} = \vec{i}_{SPj} + \vec{i}_{IPj} \quad (4)$$

$$\vec{a}_p = |\vec{a}_{SP1}| = |\vec{a}_{SP2}| = \cdots = |\vec{a}_{SPn}| \quad (5)$$

$$\vec{i}_{pj} = \vec{a}_{pj} / a_p \quad (6)$$

To take an example, when the position of one of the pin holes deviates from the correct position as shown in Fig. 1, the planet gear supported by that pin (Planet 1) will commence meshing earlier than the other planet gears. The degree of this difference in meshing times, expressed as a distance on the line of action, is the equivalent transmission error.

2.2. Translational Motion and Equivalent Transmission Error

When torque increases in a planetary gear set in which there is an eccentric error in a pin hole position, the

positions of the sun gear, the planet carriers, and the internal gear will settle into a position in which the gear mesh forces are in equilibrium, as long as sufficient clearance exists in the radial direction of the supports.

There are countless combinations of positions of the sun gear, planet carriers, and internal gear in which the gear mesh forces are in equilibrium, but for the sake of convenience, in this paper the sun gear and the internal gear will be considered to be concentric, and only the position of the centers of the planet carriers will differ against these. Because the torque on each planet gear is identical even in differing combinations of positions, generality is not lost despite this approach.

For example, for the pin hole displacement in Fig. 1, the position of equilibrium for the planet carrier is in the direction of the resultant force of the gear mesh forces on the planet gear Planet 1. As Fig. 2 shows, from the state shown in Fig. 1, the planet carriers move in the direction of the resultant force of the gear mesh forces on Planet 1, to the position where Planet 1 and the opposite gear Planet 3 simultaneously transmit torque. Because Planet 1 and Planet 3 transmit an identical volume of torque in this position, the forces on the planet carriers are in equilibrium. With regard to Planet 2 and Planet 4, the distance between the gear teeth of the sun gear and the planet gears will either increase or decrease, and the distance between the gear teeth of the planet gears and the internal gear will consequently either decrease or increase; these effects cancel each other, and there is therefore no change in the fact that these planet gears are delayed against Planet 1 in commencing to mesh, despite the movement of the planet carriers described above.

The equivalent transmission error for the meshing of the j th planet gear and the sun gear, c_{SPj} , and the equivalent transmission error for the meshing of the j th planet gear and the internal gear, c_{IPj} , generated by the eccentricity of the planet carriers, can be expressed as follows:

$$c_{SPj} = -\vec{c} \cdot \vec{i}_{SPj} \quad (7)$$

$$c_{IPj} = -\vec{c} \cdot \vec{i}_{IPj} \quad (8)$$

where

$\vec{c}$: Position of center of planet carrier in relation to centers of sun gear and internal gear in state of equilibrium of forces

The sum of these errors is determined and termed c_{pj} .

$$c_{pj} = -a_p \vec{c} \cdot \vec{i}_{pj} \quad (9)$$

The sufficient clearance in the radial direction assumed in the planetary gear set is given when RCL_c , the measured clearance of the planetary carrier in the radial direction on the condition that the sun gear and the internal gear are concentric, satisfies the following

equation:

$$RCL_c \geq |\vec{c}| \quad (10)$$

2.3. Equations of Gear Mesh Force

A linear spring model of the type shown in Fig. 3 will be considered as a stiffness model of a planetary gear set. Here, the meshing stiffness of the sun gear and the planet gears, the stiffness of the bearings supporting the planet gears, the stiffness of the pins, and the stiffness of the planet carriers are considered linear springs with fixed spring constants; k_{sp} is the synthetic spring constant

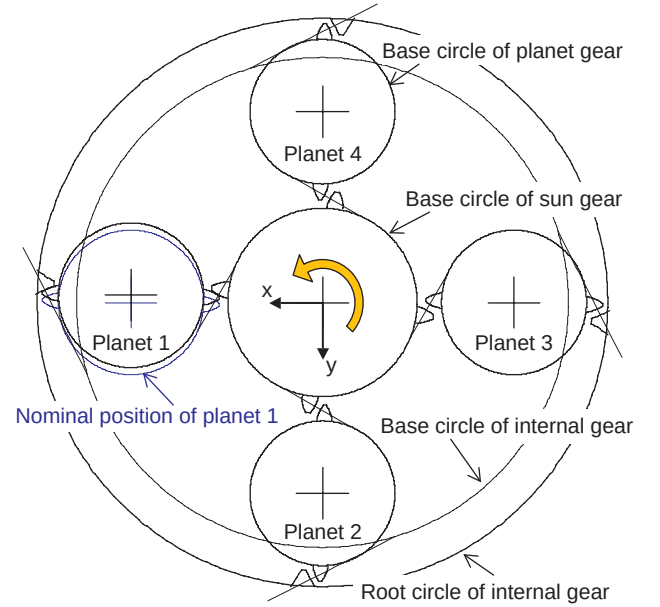


Fig. 1 Pin hole displacement

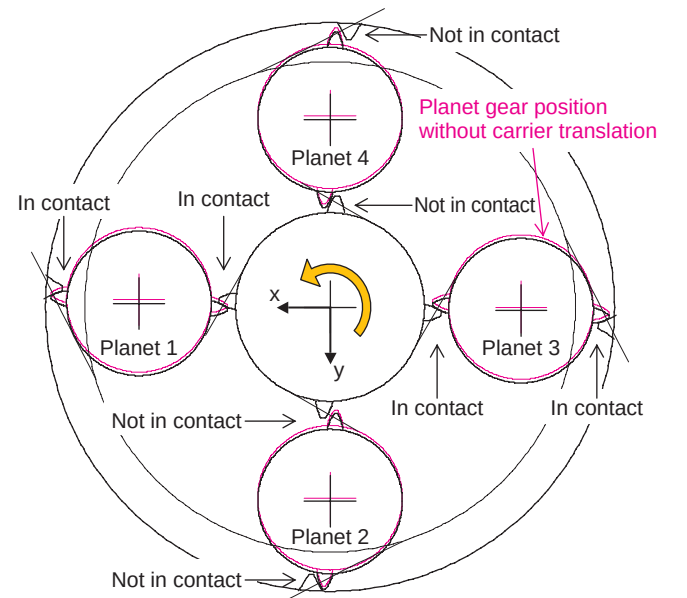


Fig. 2 Carrier translation

formed by synthesizing these terms. k_{IP} is the synthetic spring constant formed by synthesizing the meshing stiffness of the internal gear and the planet gears, the stiffness of the bearings supporting the planet gears, the stiffness of the pins, the stiffness of the planet carriers, and the annular stiffness of the internal gear, considered as linear springs with fixed spring constants.

The degree of progression of the contact points between the j th planet gear and the sun gear on the line of action in the period between the commencement of contact between the j th planet gear and both the sun gear and the internal gear and the completion of deformation of each part of the j th planet gear under the resulting loads is termed x_{SPj} . The degree of progression of points of contact between the j th planet gear and the internal gear on the line of action is termed x_{IPj} . In addition, because of the equilibrium in the direction of rotation of the planet gears, the magnitude of the gear mesh force between the planet gear and the sun gear and the magnitude of the gear mesh force between the planet gear and the internal gear are equivalent, and the magnitude of these forces is termed f_{pj} . The magnitude of the gear mesh force on the j th planet gear, f_{pj} , in relation to meshing with the sun gear and meshing with the internal gear, can be expressed as follows:

$$f_{pj} = k_{SP}x_{SPj} \quad (11)$$

$$f_{pj} = k_{IP}x_{IPj} \quad (12)$$

In addition, the sum of x_{SPj} and x_{IPj} is found and substituted as x_{pj} .

$$x_{pj} = x_{SPj} + x_{IPj} \quad (13)$$

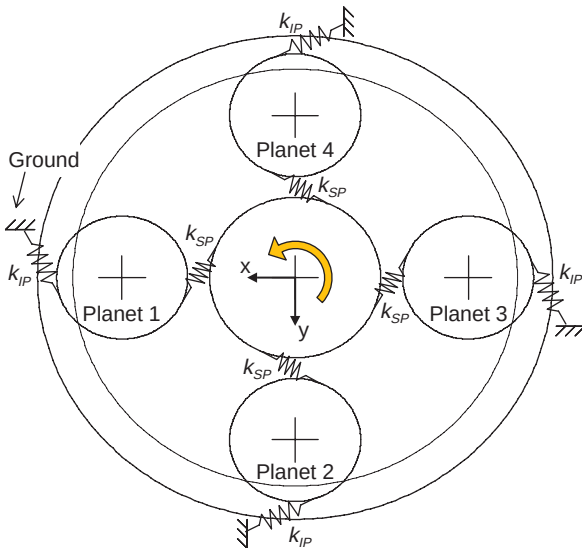


Fig. 3 Stiffness model of planetary gear sets

Given Eqs. (11), (12), and (13), the magnitude of the gear mesh force on the j th planet gear, termed f_{pj} , can be expressed as follows:

$$f_{pj} = K_P x_{pj} \quad (14)$$

Here,

$$K_P = k_{SP}k_{IP}/(k_{SP} + k_{IP}) \quad (15)$$

Because the difference in the degree of progression of the contact points between the planet gears is the equivalent transmission error, values from x_{pj} , excepting ε_{pj} and c_{pj} , are equivalent for each planet gear. Terming this value $\hat{x}_p$,

$$x_{pj} = \hat{x}_p + \varepsilon_{pj} + c_{pj} \quad (16)$$

From Eq. (16), f_{pj} can be expressed as follows:

$$f_{pj} = K_P(\hat{x}_p + \varepsilon_{pj} + c_{pj}) \quad (17)$$

The following equations can be obtained from Eq. (17):

$$\vec{f}_{SPj} = K_P(\hat{x}_p + \varepsilon_{pj} + c_{pj})\vec{l}_{SPj} \quad (18)$$

$$\vec{f}_{IPj} = K_P(\hat{x}_p + \varepsilon_{pj} + c_{pj})\vec{l}_{IPj} \quad (19)$$

Here,

$\vec{f}_{SPj}$: Gear mesh force between j th planet gear and sun gear

$\vec{f}_{IPj}$: Gear mesh force between j th planet gear and internal gear

2.4. Equation of Equilibrium of Forces

The following equation can be obtained from the state of equilibrium between the forces on the planet carriers and Eqs. (18) and (19):

$$\sum_{j=1}^n K_P(\hat{x}_p + \varepsilon_{pj} + c_{pj})a_p\vec{l}_{pj} = \vec{0} \quad (20)$$

Because the sum of the torque transmitted between the planet gears is the torque on the sun gear, T_s , if the radius of the base circle of the sun gear is termed r_{BS} , then

$$\sum_{j=1}^n K_P(\hat{x}_p + \varepsilon_{pj} + c_{pj})r_{BS} = T_s \quad (21)$$

2.5. Relationship between Equivalent Transmission Error and Torque Distribution

A relational expression between equivalent transmission error and torque distribution was derived from the equilibrium between the forces.

Even if, from symmetry,

$$\vec{i}_{p1} = \begin{pmatrix} 1 \\ 0 \end{pmatrix} \quad (22)$$

generality is not lost. At this time,

$$\vec{i}_{pj} = \begin{pmatrix} \cos(2\pi(j-1)/n) \\ \sin(2\pi(j-1)/n) \end{pmatrix} \quad (23)$$

The following equation is obtained from Eqs. (9), (20), and (23):

$$\vec{c} = \begin{pmatrix} 2u/(na_p) \\ 2v/(na_p) \end{pmatrix} \quad (24)$$

Here,

$$u = \sum_{h=1}^n \varepsilon_{ph} \cos(2\pi(h-1)/n) \quad (25)$$

$$v = \sum_{h=1}^n \varepsilon_{ph} \sin(2\pi(h-1)/n) \quad (26)$$

Further, the following equation, expressing the relationship between equivalent transmission error and the gear mesh force between each planet gear, can be obtained from Eqs. (9), (17), (21), and (24):

$$f_{pj} = T_s/(nr_{BS}) + K_p E_{pj} \quad (27)$$

Here,

$$E_{pj} = \varepsilon_{pj} - \left(U_{pj} + V_{pj} + \sum_{h=1}^n \varepsilon_{ph} \right) / n \quad (28)$$

$$U_{pj} = 2u \cos(2\pi(j-1)/n) \quad (29)$$

$$V_{pj} = 2v \sin(2\pi(j-1)/n) \quad (30)$$

E_{pj} , obtained from Eq. (28), will be termed the projected error for the j th planet gear.

Having termed the radius of the base circle of the planet gear r_{BP} , from Eq. (27), the torque transmitted by the j th planet gear, T_{pj} , is

$$T_{pj} = (T_s r_{BP}) / (nr_{BS}) + K_p r_{BP} E_{pj} \quad (31)$$

The first term on the right-hand side of this equation is the value of torque when the load is equally distributed, and the second term on the right-hand side is the deviation from the torque value when the load is equally distributed. The deviation is proportional to the projected error of the j th planet gear, E_{pj} .

If the positional tolerance of the pin hole is termed ϕb , the following equation for the distribution of torque between the planet gears can be obtained from Eq. (31):

$$f(T_{pj}) = (1/\sqrt{2\pi}\sigma) \exp(-(T_{pj} - m)^2 / (2\sigma^2)) \quad (32)$$

Here,

$$m = (T_s r_{BP}) / (nr_{BS}) \quad (33)$$

$$\sigma = a_p K_p r_{BP} \sigma_{PH} \sqrt{(n-3)/n} \quad (34)$$

$$\sigma_{PH} = b/6 \quad (35)$$

3. Verification in Actual Planetary Gear Set

This chapter will discuss the results of tests conducted to verify the validity of the relational expression between the eccentric error of the pin hole position and the torque on the planet gears. Following this, it will discuss the results of tests conducted to verify the accuracy of estimation of the stiffness value K_p , which is necessary for prediction of the distribution of torque between the planet gears resulting from production errors.

The tests used a 4-planet planetary gear set, and the verification process focused on the relational expression in the case of four planet gears. The process of verification using the actual gear set was divided into two stages. First, a planet carrier with vanishingly small pin hole displacement was used to verify the method of measurement of the torque on the planet gears. Next, planet carriers in which pin hole positions had been intentionally displaced were used to verify the relational expression. Following this, the test results were used to verify the accuracy of estimation of the stiffness value, K_p .

In the case of a 4-planet planetary gear set, the relational expression between pin hole displacement and the torque on the j th planet gear, T_{pj} , is as follows:

$$T_{pj} = (T_s r_{BP}) / (4r_{BS}) + K_p r_{BP} E_{pj} \quad (36)$$

Here,

$$E_{pj} = (\varepsilon_{p1} - \varepsilon_{p2} + \varepsilon_{p3} - \varepsilon_{p4}) / 4 \quad (j = 1, 3) \quad (37)$$

$$E_{pj} = -(\varepsilon_{p1} - \varepsilon_{p2} + \varepsilon_{p3} - \varepsilon_{p4}) / 4 \quad (j = 2, 4) \quad (38)$$

From Eqs. (37) and (38), it was predicted that the torque on opposite pairs of planet gears in the test would be equivalent.

3.1. Test Rig

Figure 4 shows the layout of the test rig. The test rig contained two planet gears. Two internal gears were fixed to the case, connecting the planet carriers, and two sun gears were connected by a shaft. The planet gear in the left of the diagram was the subject of the measurements. Table 1 shows the specifications of this gear. Direct

Table 1 Planetary gear set specifications

Term		Sign	Value
Number of teeth	Sun gear	z_s	30
	Planet gear	z_p	23
	Internal gear	z_i	78
Normal module		M	1.3
Normal pressure angle		α	20 deg
Helix angle		β	20 deg
Center distance		a	37.1 mm

measurement of the torque on each planet gear represented a challenge. Strain gauges were therefore attached to the outer circumference of the internal gear, and the strain on the circular section of the internal gear when the planet gear passed under the section to which the strain gauges were attached was measured. Torque was estimated on this basis. Figure 5 shows the internal gear with a strain gauge attached. Two strain gauges were attached in the axial direction at intervals of approximately 90° in the circumferential direction, for a total of eight strain gauges.

3.2. Test Conditions

The tests to verify the method of measurement of the torque on the planet gears were conducted using a planet carrier with vanishingly small pin hole displacement. The tests to verify the validity of the relational expression were conducted using planet carriers in which pin hole positions

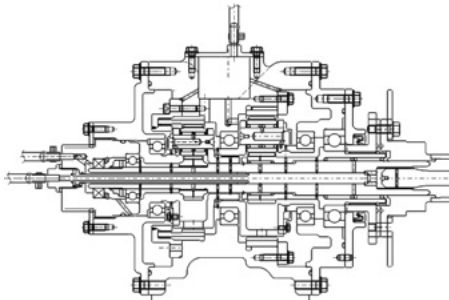


Fig. 4 Layout of test rig



Fig. 5 Position of strain gauge

had been intentionally displaced. Table 2 shows the deviation from the correct pin hole position in the four planet carriers, A to D. Table 3 shows the projected error of each planet gear in the planet carriers. Tests were conducted on each of the planet carriers at input torques of 118, 157, 196, and 235 Nm.

3.3. Results of Verification Test of Torque Measurement Method

Figure 6 shows the results of strain measurements at the different levels of input torque in the planet carrier with vanishingly small pin hole displacement. Because the speed of rotation was fixed, the horizontal axis that represents time is virtually corresponding to the angles of rotation of the planet carriers. The vertical axis shows strain on the internal gear. Strain was generated cyclically in response to the rotation of the planet carriers. Tensile strain was generated when the planet gears passed directly underneath the strain gauge, and compressional strain was generated at the median position between the two adjacent planet gears. The peak of tensile strain in particular can be seen as displaying a correlation with the torque on the planet gears. The four peaks shown in Fig. 6 correspond to each of the four planet gears in the planetary gear set. The values of the four peaks are virtually identical, and this indicates that the torque is evenly divided between the planet gears.

The peak values for tensile strain were isolated from measurement results for each input torque, and the arithmetic mean for each input torque was determined. The arithmetic means were plotted against the torque on the planet gears. Considering the load as having been distributed equally to each planet gear, the values of torque on the planet gears were estimated from the torque sharing ratio, the input torque on the sun gear, and gear ratio. The results are shown in Fig. 7. The horizontal axis shows the value of torque on the planet gear, while the vertical axis shows the arithmetic mean of the peak values of strain on the internal gear.

The strain on the outer circumference of the circular section of the internal gear is proportional to the torque

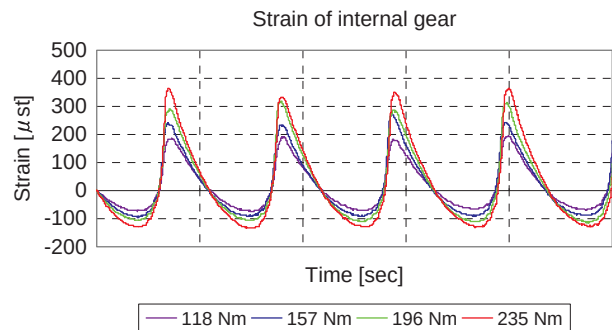


Fig. 6 Strain of internal gear for carrier with vanishingly small pin hole displacement

Table 2 Pin hole displacement of test carriers

Carrier	Pin hole displacement							
	μm							
	Planet 1		Planet 2		Planet 3		Planet 4	
	dx	dy	dx	dy	dx	dy	dx	dy
A	-1.5	40.2	0.5	-9.2	1.7	-3.3	-2.5	-5.4
B	-25.9	-2.0	-30.5	-8.6	-24.6	-16.5	-17.9	91.1
C	-11.6	2.5	-23.6	89.9	89.5	-20.9	1.0	-108.8
D	-20.4	3.0	-19.8	3.1	-19.1	54.4	-20.1	4.2

Table 3 Projected error of test carriers

Carrier	Projected error			
	μm			
	Planet 1	Planet 2	Planet 3	Planet 4
	E_{P1}	E_{P2}	E_{P3}	E_{P4}
A	-21.8	21.8	-21.8	21.8
B	-3.3	3.3	-3.3	3.3
C	7.7	-7.7	7.7	-7.7
D	24.0	-24.0	24.0	-24.0

on the planet gear. Therefore, the ratio of strain for each planet gear can be considered to be the ratio of torque carried by each planet gear. In the verification test for the relational expression, the ratio of torque carried by each planet gear to the total torque applied was determined from the ratio of strain, and the torque carried by each planet gear was estimated from the determined ratio of torque, the input torque to the sun gear, and gear ratio.

3.4. Results of Measurements of Torque on Planet Gears at Different Pin Hole Displacements

Figure 8 shows the results of measurements of strain at differing input torques for test piece A. The peak value of tensile strain was focused on, and it was seen to differ

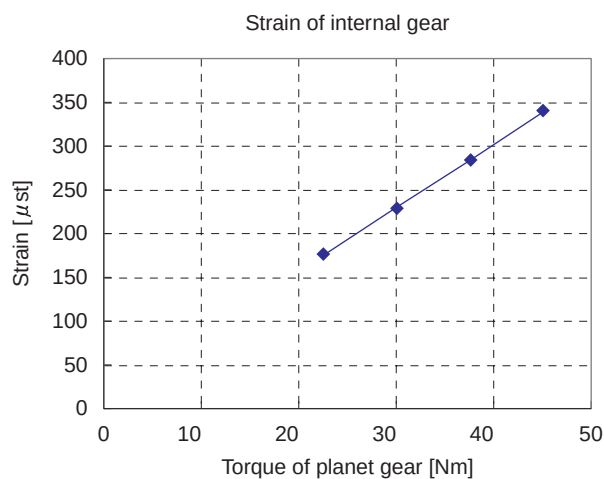


Fig. 7 Peak value of strain of internal gear with planet gear torque for carrier with vanishingly small pin hole displacement

depending on the planet gear. This result can be interpreted as reflecting the unevenness in the distribution of torque between the planet gears. The level of strain on pairs of opposing planet gears was virtually identical. This can be interpreted as resulting from the fact that the torque on opposing planet gears is identical, and accords with the result predicted from Eqs. (37) and (38).

Tests were conducted for test pieces A to D, and the peak values of tensile strain corresponding to each planet gear were isolated from the measurement results. The arithmetic means of the peak values for each input torque were determined, and the torque carried by each planet gear was estimated from these mean values. The torque estimated for each planet gear from strain was plotted against the projected error calculated from the degree of displacement of the pin hole positions in each test piece. Figure 9 shows the results for one of the four planet gears. The horizontal axis shows the projected error, and the vertical axis shows the torque on the planet gear. The

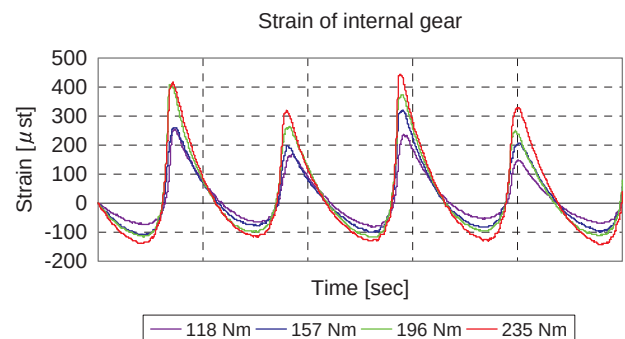


Fig. 8 Strain of internal gear for test carrier A

lines on the graph show the results of a least mean square approximation of test data using linear functions. The different colors show the different input torques.

As Fig. 9 shows, the value of the torque on the planet gear is proportional to the projected error. In addition, the y-intercept of the fitted lines showing the approximations are virtually identical to the torque value when the input torque is distributed equally between the planet gears. These results demonstrate that the proportional relationship expressed in Eq. (36) holds true in an actual planetary gear set.

3.5. Verification of Accuracy of Prediction of Stiffness Value K_p

Test results demonstrated that the proportional relationship expressed in Eq. (36) held true, but the accurate prediction of the distribution of torque between the planet gears would necessitate the accurate estimation of the stiffness value K_p . The value of K_p can be determined from a numerical analysis conducted using 3D FEM. The accuracy of estimation of the stiffness value K_p was verified for the planetary gear set used in the tests.

A 3D FEM model of the planetary gear set was formulated, and a numerical analysis was conducted. The only data employed was blueprint data such as shape and layout data, and physical values for the materials, such as Young's modulus. Using the stiffness value K_p obtained from calculation results, the torque on the planet gears was calculated for test pieces A to D. Figure 10 shows the results of these calculations with test results. The horizontal axis shows torque on the sun gear, i.e., input torque, and the vertical axis shows torque on the planet gears. The colored circles in the figure show test results, while the blank squares show calculation results. The differences in color represent the different planet carriers used in the tests.

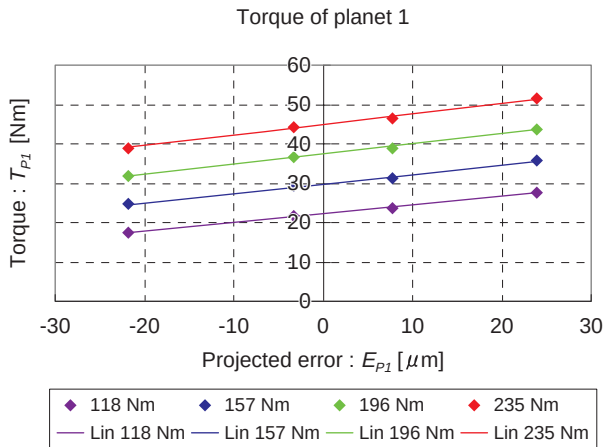


Fig. 9 Variation of torque of planet gear No.1 T_{P1} with projected error E_{P1} for different input torque

A comparison of test and calculation results reveals a good match between the two. This demonstrates that the stiffness value K_p can be accurately predicted by means of a 3D FEM analysis using only computer data.

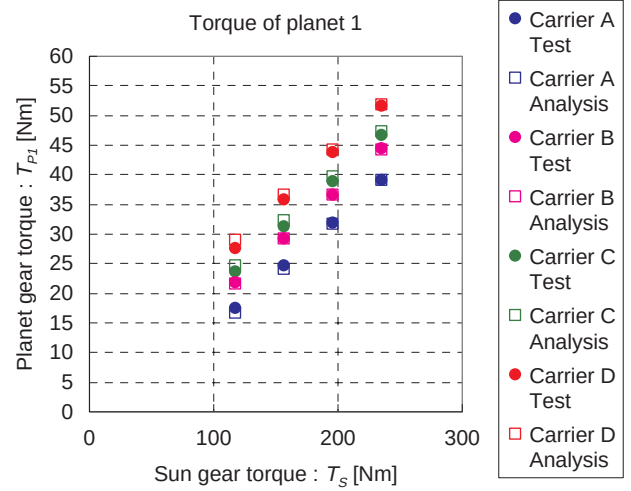


Fig. 10 Variation of torque of planet gear No.1 T_{P1} with sun gear torque T_s for different carrier

5. Conclusion

The project discussed in this paper focused on equivalent transmission error for planetary gear sets, with the following results:

- (1) A relational expression between eccentric error of the pin hole positions and the torque on the planet gears in a planetary gear set containing n normal planet gears was derived.
- (2) This relational expression made it possible to derive an equation expressing the distribution of torque to the planet gears in a planetary gear set due to production errors in relation to the pin hole positions.
- (3) The relational expression held true in an actual planetary gear set.
- (4) It was possible to accurately predict the stiffness value, K_p , necessary to the prediction of the distribution of torque to planet gears due to production errors, using a 3D FEM numerical analysis.

Based on these results, it is now possible to accurately predict the distribution of torque to the planet gears in a planetary gear set containing normal n planet gears using only computer data.

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■ Author ■



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Optimization of V-surface Angle for Metal Pushing V-belt CVT

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ABSTRACT

The V-surface angles of the elements of metal-pushing V-belts were focused on, and an optimum angle was studied, in order to increase both torque transfer capacity and torque transfer efficiency in a CVT. The resulting reduction of the element V-surface angles from 11° to 9° produced an increase of approximately 20% in torque transfer capacity, and an increase of approximately 0.5% in unit efficiency in the OD ratio (close to a speed ratio of 0.45). With regard to the fatigue strength of the belt, measurements of the force between the shafts and durability tests showed that the ring tension is reduced and torque transfer capacity is increased in proportion to the degree of narrowing of the V-surface angles of the elements. The torque transfer efficiency of a metal-pushing V-belt is affected by the maximum level of variation in belt trajectory, as represented by the difference between the maximum and minimum values of the movement of the belt in the radial direction on the pulley. The research discussed in this paper indicated that control of the degree of variation in belt trajectory reduced friction loss due to the slip-motion of the belt in the radial direction, thus increasing torque transfer efficiency.

1. Introduction

The metal-pushing V-belt continuously variable transmission (CVT), invented in 1970 by Dr. Hub van Doorne, is today widely used in order to increase fuel efficiency and reduce exhaust emissions. Increases in torque transfer capacity and efficiency are constantly demanded from the metal-pushing V-belts that are framework components of the CVTs. However, it has been indicated that increasing transmission efficiency during high-speed cruising represents an issue in metal-pushing V-belt CVTs⁽¹⁾, and a variety of arguments has developed concerning the form of CVTs. A recent report has indicated that torque transfer efficiency can be increased if CVT pulley thrust is optimized⁽²⁾, and this suggests that there is still potential for increasing torque transfer efficiency. In addition, to date little research has focused on the relationship between the V-surface angles of the elements and torque transfer capacity and efficiency.

The research discussed in this paper focused on optimizing the element V-surface angles of a metal-

pushing V-belt CVT, and determined the optimum V-surface angle for increasing torque transfer capacity and efficiency. The results of the research have indicated that torque transfer capacity and efficiency can be increased against a standard metal-pushing V-belt if the V-surface angles are modified from the conventional 11° to 9° . This paper will present an overview of the research.

2. Development Goals and Targets

A variety of methods of increasing the torque transfer capacity and efficiency of a metal-pushing V-belt CVT can be considered: The metal-pushing V-belt itself can be modified, the pulleys and other components that clamp the belt can be modified, and the pulley control pressure can be optimized. The development discussed here set out to increase the torque transfer capacity of the metal-pushing V-belt by optimizing the V-surface angles of the elements, while achieving a level of torque transfer efficiency equivalent to or greater than that of a conventional belt. The torque transfer capacity of a metal-pushing V-belt

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CVT is defined as the limit time in which torque to be transmitted with conditions set for the number of input rotations, input torque, speed ratio, and the thrust of the pulley on the low-thrust side. The parameter used to evaluate torque transfer capacity in this research was the rate of increase in the durability time for which the unit was operated in the driving mode determined by the TOP ratio (the speed ratio for maximum vehicle speed).

The following development targets were set:

- (1) Metal-pushing V-belt torque transfer capacity: +20% against conventional belt; and
- (2) Metal-pushing V-belt torque transfer efficiency: Equivalent to conventional belt.

3. Development Concept

In order to increase the torque transfer capacity of a metal-pushing V-belt, it is necessary to reduce the stress generated on the belt. Table 1 shows a list of possible methods of reducing ring stress in order to increase torque transfer capacity by 20%. Increasing the width of the rings, increasing the number of laminated rings, and increasing the distance between the shafts of the CVT are available as methods of reducing ring stress, but they all result in an increase in the size and the weight of the belt.

The present development narrowed the standard V-surface angles of the elements to 9°. The development concept formulated was that by this means, torque transfer capacity could be increased by 20% with no increase in the size or weight of the CVT, and a torque transfer capacity equivalent to or greater than that of a standard belt could be achieved.

Table 1 Technique for reduction of ring stress

	Item changed in metal pushing V-belt			Item changed in transmission
Spec.	V-surface angle 11 → 9 [deg]	Width of rings 9.2 → 18.2 [mm]	Number of laminated rings 12 → 17 [Number]	Distance between shafts 156 → 170 [mm]
Weight	Same	Increase	Increase	Increase

4. Configuration of Metal-pushing V-belt and Names of Parts

The metal-pushing V-belts used in CVTs are made up of several hundred elements and two sets of layered rings (Fig. 1).

Figure 2 shows the configuration of a belt and the names of the individual parts.

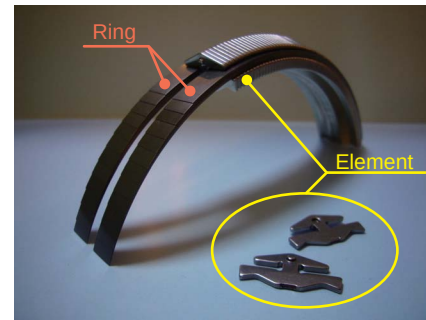


Fig. 1 Metal pushing V-belt

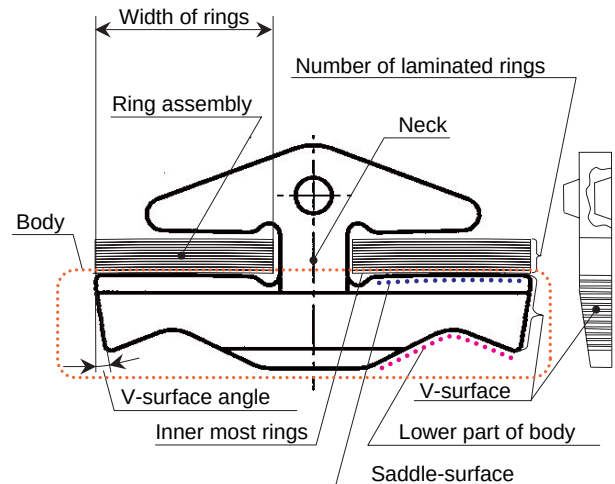


Fig. 2 Configuration of element and ring assembly

5. Specifications of Metal-pushing V-belt

When the V-surface angles of the elements of the metal-pushing V-belt were modified to 9° in this project, the form of the lower part of the element bodies was also modified, with consideration of the stiffness of the elements. Other specifications remained identical to those employed in elements with a V-surface angle of 11°.

5.1. Study of Element V-surface Stiffness

The V-surface stiffness is designed to maintain a constant theoretical V-surface angle irrespective of the speed ratio or the input torque. When the V-surface angle of the elements was modified, the vector of the thrust from the pulleys on the elements also changed (Fig. 3). Because of this, it was necessary to modify the form of the lower sections of the element bodies in accordance with the V-surface angle that was set (Fig. 4).

Figure 5 shows the method used to evaluate V-surface stiffness. In the figure, the uppermost section of the element V-surface is termed A, and the lowermost section is termed B. The form of the lower section of the element body was designed to help to ensure that the displacement of the V-surface given by A-B would be the minimum figure irrespective of the speed ratio.

5.2. Effect of Element V-surface Stiffness

The effect of the element V-surface stiffness on the maximum torque transfer capacity was studied. Figure 6 shows deformation diagrams for an element when the V-surface angle is constant and when it is not. In Fig. 6(a), the displacement of the V-surface is low, and the V-surface angle is constant, while in Fig. 6(b) the V-surface displacement is high, and the V-surface angle is not constant. Figure 7 shows the relationship between V-surface displacement (A-B) and maximum torque transfer capacity. The graph shows the percentage of the measured maximum torque transfer capacity when the V-surface angle is not held constant against the maximum torque transfer capacity when the angle is constant. When the V-surface angle is not held constant, the maximum transfer capacity is reduced by 3.2% against the case in which the

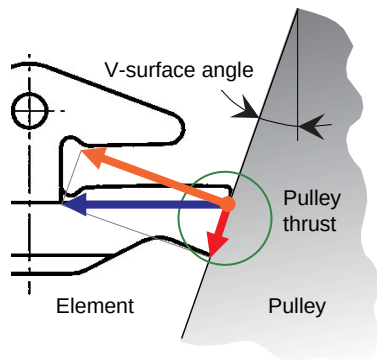


Fig. 3 Pulley thrust vector

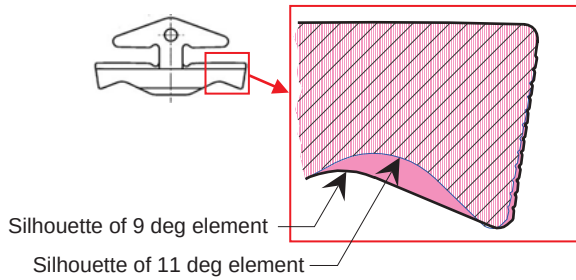


Fig. 4 Difference of outline of lower part of body

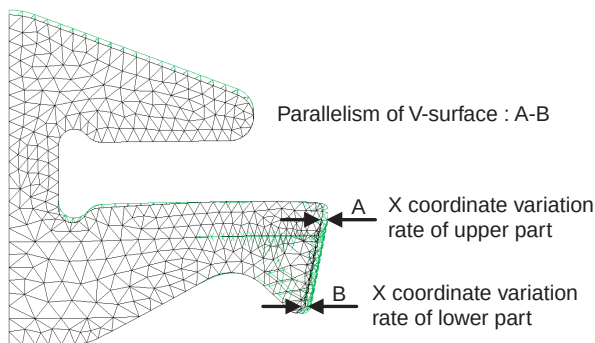


Fig. 5 Parallelism of V-surface

angle was constant.

A metal-pushing V-belt transfers torque through shearing of the boundary lubrication oil film formed between the element V-surfaces and the pulley V-surfaces. It is thought that in the case shown in Fig. 6(b), in which the V-surface displacement is high due to its stiffness, the element V-surfaces make localized contact with the pulley V-surfaces, and a boundary lubrication oil film cannot be formed effectively; because of this, the shear force declines⁽³⁾ and there is a reduction in the maximum transmitted torque. Therefore, the maintenance of the stiffness of the element V-surfaces is related to the maximum torque transfer that can be achieved.

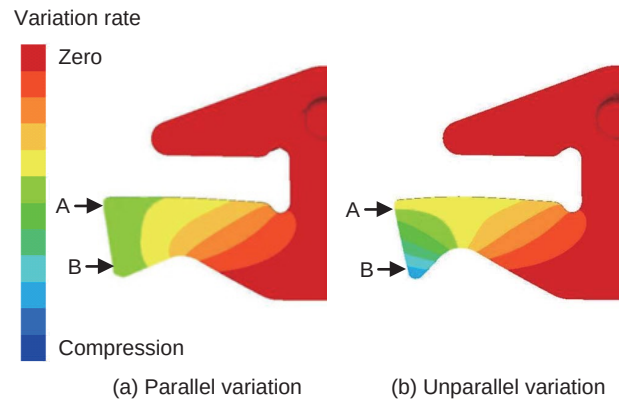


Fig. 6 Variation rate of element

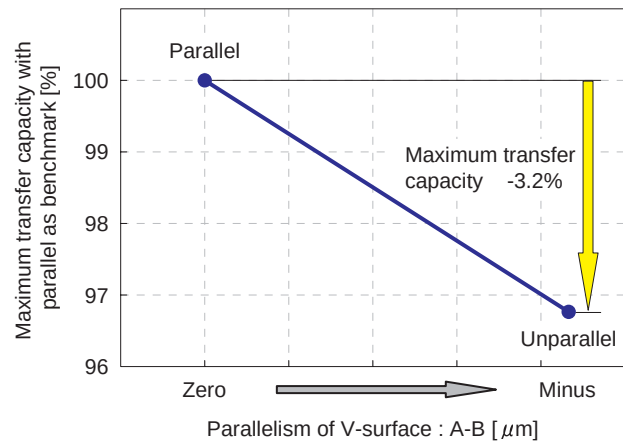


Fig. 7 Relation between rigidity of V-surface and maximum transfer capacity

6. Verification of Effect in Increasing Torque Transfer Capacity

The effect of the modification of the element V-surface angles to 9° in increasing torque transfer capacity was verified by checking the force between the shafts in simulations and actual measurements, and by conducting durability tests.

6.1. Estimation of Force between Shafts

Employing a simulation, it was estimated that the modification of the element V-surface angles from 11° to 9° would reduce the force between the shafts by approximately 10%, and that, as a result, the amplitude of stress on the rings, σ_a , would be reduced by approximately 4% and the mean stress on the rings, σ_m , would be reduced by approximately 7%, and that transfer capacity would increase.

Figure 8 shows results for the stress generated on the innermost rings of a metal-pushing V-belt for different phases of rotation at a speed ratio of 0.61. Figure 8(a) shows the positions of the rings and the phases of rotation used in the analysis.

Figure 8(b) shows results for a conventional metal-pushing V-belt using V-surface angles of 11° , while Fig. 8(c) shows results for a belt in which the V-surface angles had been set at 9° . Table 2 shows the stress amplitude (σ_a) and the mean stress (σ_m) determined from Figs. 8(b) and 8(c).

6.2. Measurement of Force between Shafts

The force between the shafts, produced by pulley thrust and torque transfer, acts between the pulleys and is proportional to the tension of the metal-pushing V-belt. The reduction of the element V-surface angles from 11° to 9° reduced the force between the shafts, and also reduced the contact pressure of the element saddle surfaces. It was therefore possible to compare the magnitude of ring tension by measuring the magnitude of the force between the shafts.

6.2.1. Measurement method

Figure 9 shows the method employed to measure the force between the shafts. A driven pulley stand was fixed to a base, and a drive pulley stand was fixed to a slide rail that could be freely moved in a direction orthogonal to the axis of rotation. One end of the drive pulley stand was connected to a fixed surface via a load cell. By this means, the load cell was able to measure the force generated between the shafts of the drive and driven pulleys by the tension of the metal-pushing V-belt.

6.2.2. Measurement results

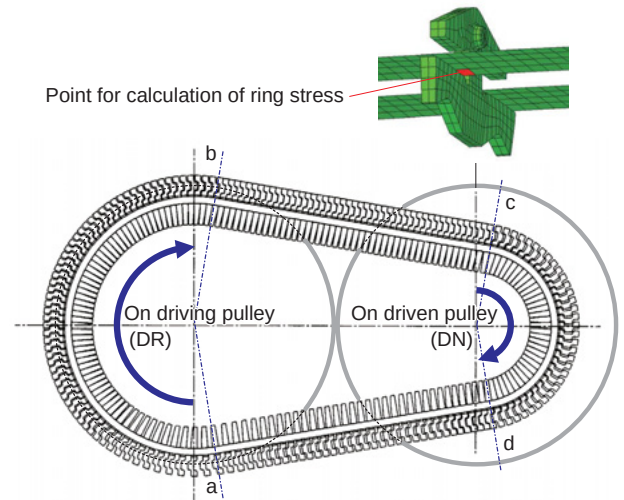
The results of the measurements showed that the reduction of the element V-surface angles to 9° reduced the force between the shafts by 16% in an operating mode corresponding to the top speed in an actual vehicle. This result displays the same tendency as the simulation results (Fig. 10).

6.3. Verification of Torque Transfer Capacity

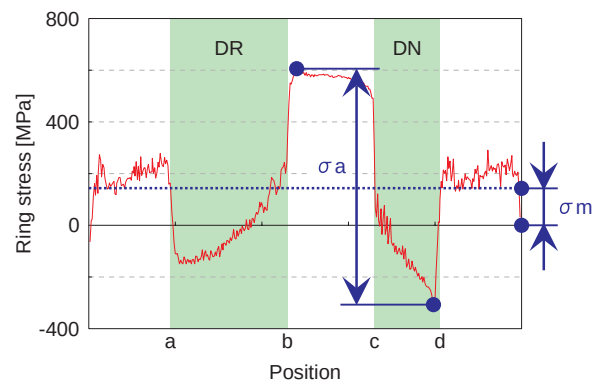
In order to verify the increase in torque transfer capacity in a metal-pushing V-belt in which the element V-surface angles had been reduced to 9° , the belt was subjected to a durability test extended beyond the limit

Table 2 List of calculated ring stresses

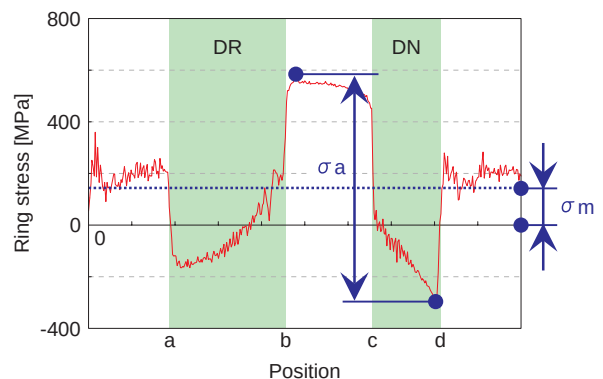
	V-surface angle		Rate of easing of stress
	11 deg	9 deg	
Stress amplitude σ_a	461 MPa	441 MPa	4%
Mean stress σ_m	144 MPa	134 MPa	7%



(a) Position on metal pushing V-belt in operation



(b) Ring stress diagram for 11 deg V-surface angle



(c) Ring stress diagram for 9 deg V-surface angle

Fig. 8 Calculated ring stress

time for a metal-pushing V-belt with V-surface angles of 11° . Figure 11 shows a comparison of the force between the shafts against the number of rotation cycles for belts with V-surface angles of 9° and 11° . The results show that the reduction in the force between the shafts when the V-surface angles are reduced to 9° produces a 23% increase in torque transfer capacity.

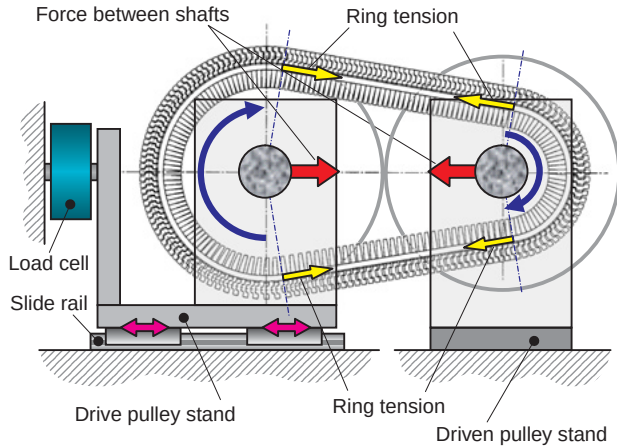


Fig. 9 Method for measurement of force between shafts

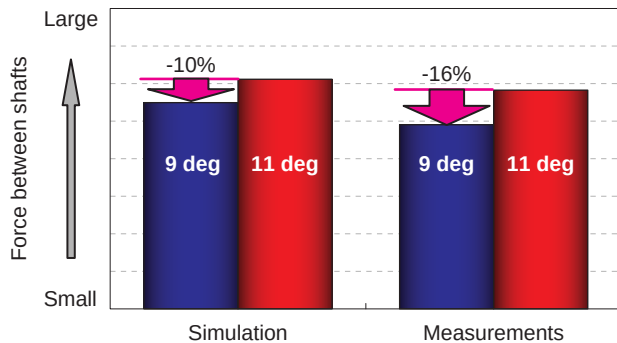


Fig. 10 Relation between V-surface angle and force between shafts

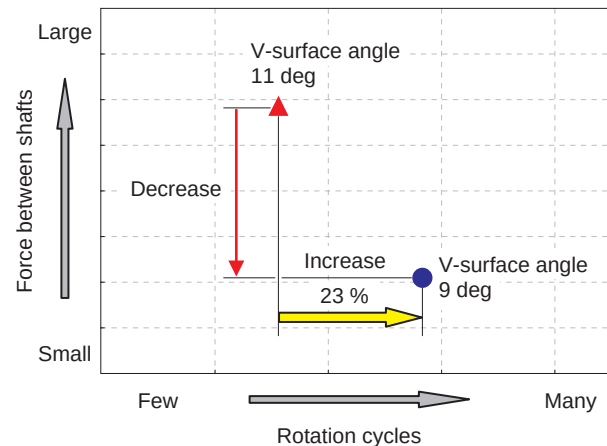


Fig. 11 Relation between rotation cycles of metal pushing V-belt and force between shafts

7. Verification of Torque Transfer Efficiency

During the verification of torque transfer efficiency, the degree of variation in the trajectory of the belt on the pulleys was measured and checked against measurements of efficiency, demonstrating the fact that the degree of variation in the belt trajectory affects efficiency.

7.1. Necessity for Measurement of Belt Trajectory on Pulleys

It was previously thought that due to the speed ratio, the trajectory of the belt on the pulleys would take on a pitch radius that represented the geometric radius. However, due to a variety of factors including the inclination of the pulleys under hydraulic pressure, the trajectory of the belt can diverge slightly from the winding radius. It is thought that these variations in belt trajectory result in a decline in torque transfer efficiency because the belt shifts in a direction orthogonal to the direction of rotation, in which torque is transferred. When the V-surface angles of the belt elements were modified, a different maintenance thrust force was necessary to maintain the speed ratio under the same operating conditions, and the belt trajectory therefore changed. It was therefore necessary to measure the trajectory of the metal-pushing V-belt around the pulleys for the different V-surface angles.

7.2. Method of Measurement of Variation in Trajectory

Measurement equipment was employed to measure the trajectories of the belts from the pulley entrance to the pulley exit. Figure 12 shows the equipment used to measure the trajectories of the belts on the drive pulley. Eddy current displacement sensors were positioned on the outer circumference of the belts, and the distance to the

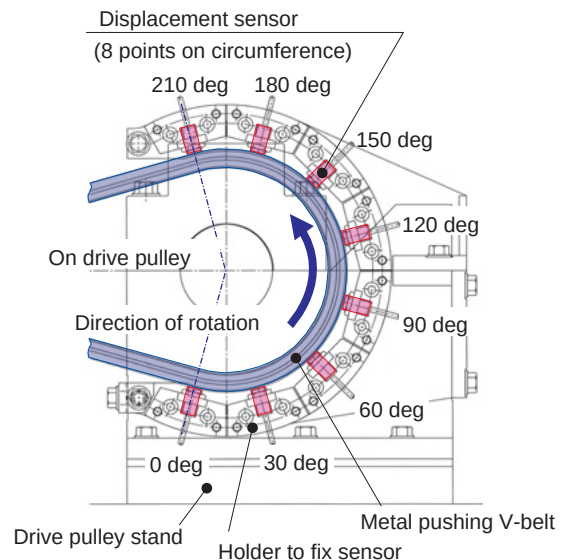


Fig. 12 Equipment for measurement of trajectory of metal pushing V-belt on pulley

outermost surfaces of the elements was measured. The measurement pitch was set at 30° , and eight sensors were used to cover a range of 210° from the entrance to the exit of the drive pulley. The winding radius (pitch radius) determined geometrically from the speed ratio was used as a benchmark, and the displacement data for each measured rotation phase was expressed as relative degrees of displacement from the pitch radius.

7.3. Maximum Variation of Belt Trajectory and Torque Transfer Efficiency

Figure 13 shows the results of measurements of variations in the trajectory of a metal-pushing V-belt. The maximum degree of variation in the trajectory of the belt is defined as the difference between the maximum radius and the minimum radius, with the pitch radius as the benchmark figure. Figure 14 shows a diagram of the state of the belt when wound around the pulley, based on the trajectory shown in Fig.13.

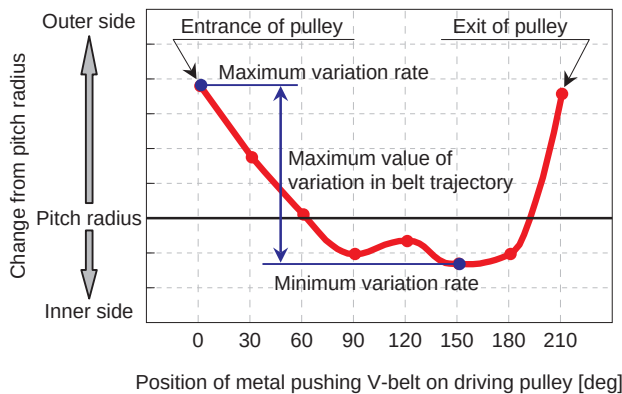


Fig. 13 Results of measurement of trajectory of metal pushing V-belt

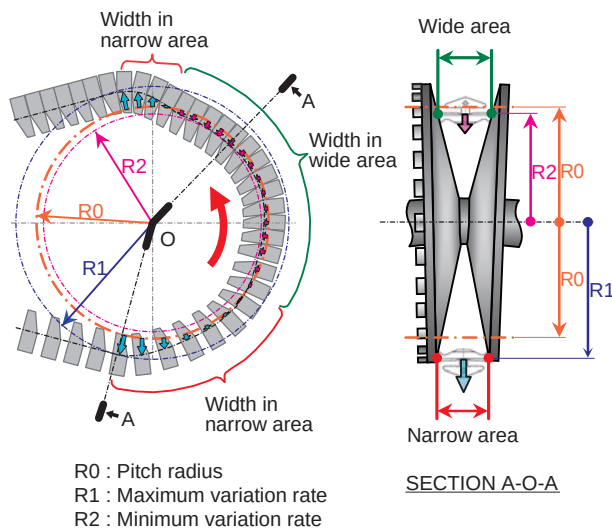


Fig. 14 Mimetic diagram of metal pushing V-belt trajectory on pulley

Because there are no elements present between the pulley V-surfaces in the area in which the belt is not wound around the pulley (the rotation phase between 210° and 360° from the pulley entrance), the gap between the pulley V-surfaces is narrow due to pulley deflection and other factors. Due to the effect of this phenomenon, the trajectory of the belt at the entrance of the pulley (the rotation phase between the entrance and 60°) and the exit of the pulley (the rotation phase between 190° and 210° from the entrance) is outside the pitch radius, and these are the highest radii among any of the measured points of the trajectory. In the area in which the belt is wound around the pulley (the rotation phase between the entrance and 210°), there are elements present between the pulley V-surfaces, and the gap between the V-surfaces is therefore wide. It is thought that because of this the belt trajectory approaches the inside of the pitch radius in the rotation phase between 60° and 190° from the pulley entrance.

It can be conjectured that these types of variation in the trajectory of the metal-pushing V-belt, producing a slip-motion in the radial direction, orthogonal to the direction of pulley rotation, act as a torque loss factor in torque transfer by the belt.

Next, the effect of variations in the belt trajectory on torque transfer efficiency was studied. The maximum value of trajectory variation was employed as the parameter for the variation in trajectory, and its relationship with torque transfer efficiency was examined (Fig. 15).

Using a metal-pushing V-belt with element V-surface angles of 9° , torque transfer efficiency was measured while varying the trajectory of the belt by varying the points of action of axial thrust force applied to the pulleys. A correlation was observed between the maximum variation in belt trajectory and torque transfer efficiency, with results displaying a tendency for efficiency to increase in proportion to the degree to which the variation in belt trajectory was controlled to a lower value.

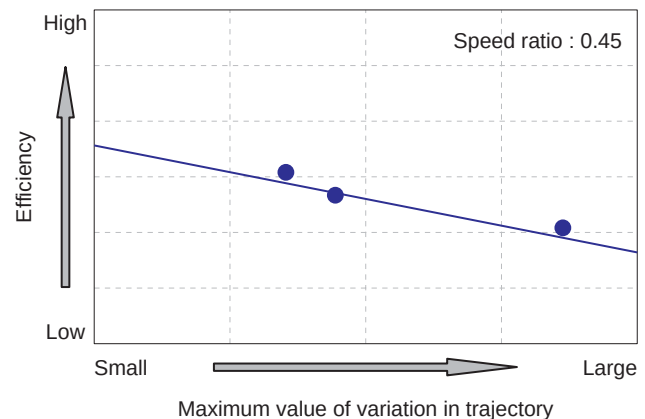


Fig. 15 Relation between maximum value of variation in belt trajectory and torque transfer efficiency

7.4. V-surface Angle and Maximum Variation in Trajectory

The relationship between the element V-surface angles and the maximum variation in belt trajectory was studied. Figure 16 shows a comparison of changes in the degree of variation in the belt trajectory against input torque for belts with element V-surface angles of 11° , 9° , and 7° , with 11° as the benchmark. The degree of variation in the trajectory of the belt was at its maximum in the belt with element V-surface angles of 7° , followed by the belt with element V-surface angles of 11° . The belt with element V-surface angles of 9° displayed the lowest level of variation in trajectory. This trend in the relationship between the V-surface angles of the elements and the maximum variation in belt trajectory remained unchanged when input torque was varied.

7.5. V-surface Angle and Torque Transfer Efficiency

Figure 17 shows the torque transfer efficiency for belts using elements with each of the V-surface angles as

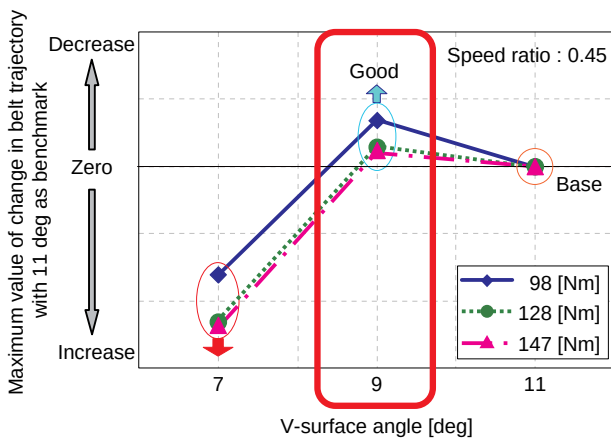


Fig. 16 Relation between V-surface angle and maximum value of change in belt trajectory

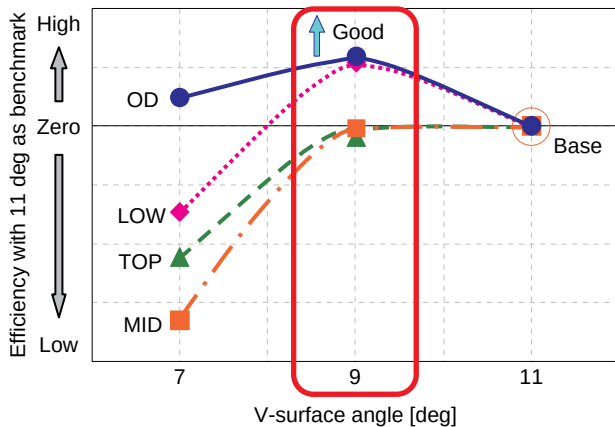


Fig. 17 Relation between V-surface angle and torque transfer efficiency

a difference with the torque transfer efficiency of a belt using elements with V-surface angles of 11° as the benchmark, for each speed ratio.

The belt using elements with V-surface angles of 9° displays identical torque transfer efficiency as the belt using elements with 11° V-surface angles at the MID and TOP ratios, and its efficiency reaches a maximum at the LOW and OD ratios. With the exception of the OD ratio, belts using elements with narrower V-surface angles than 9° displayed lower torque transfer efficiency than the belt using elements with 11° V-surface angles at all speed ratios. It can therefore be concluded that 9° is the optimum setting for the V-surface angle of the belt elements in order to realize maximum torque transfer efficiency at all speed ratios.

8. Vehicle Shift Characteristic

The shift characteristic of the conventional belt and the developed belt were compared in an actual vehicle.

8.1. V-surface Angle and Pulley Stroke

When the V-surface angles of the elements are made narrower, the pulley stroke is reduced in the same ratio range. Because of this, the stroke volume is also reduced, and the shift time during rapid acceleration and deceleration can therefore be reduced.

8.2. Results of Measurements of Shift Time

The belt with element V-surface angles of 9° displayed a faster hydraulic pressure rise time on the driven side than the belt with element V-surface angles of 11° , and its change in speed ratio towards the LOW side was faster to the same degree.

Figure 18 shows the shift characteristics in an actual vehicle under rapid deceleration (-4.9 m/s^2) from 60 km/h when the speed ratio returns to the LOW ratio side.

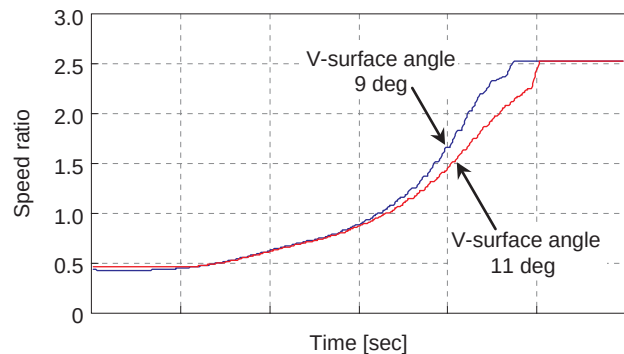


Fig. 18 Comparison of change in speed ratio with V-surface angle

9. Conclusion

The project discussed in this paper focused on the V-surface angles of the elements used in metal-pushing V-belts, and attempted to determine the optimum angle. The study showed that 9° was the optimum V-surface angle, and produced the following conclusions:

- (1) The use of element V-surface angles of 9° resulted in a 23% increase in torque transfer capacity against a belt using element V-surface angles of 11°;
- (2) A belt with element V-surface angles of 9° displayed equivalent or greater torque transfer efficiency to a conventional belt at all speed ratios, and displayed the highest torque transfer efficiency among belts with element V-surface angles of 7°, 9°, and 11°;
- (3) Narrowing the V-surface angles of the elements reduced the pulley stroke volume, enabling shift time during rapid acceleration and deceleration to be reduced; and
- (4) The maximum degree of variation in the trajectory of the belt when it is wound around the pulley affects torque transfer efficiency.

10. Afterword

Because natural selection goes into action if evolution stops, the development process is always a serious issue in the case of parts that are the focus of fierce competition for performance gains. Precisely because of this seriousness, it is necessary to clarify targets more precisely, and to pursue small details even in a specification that has until that point seemed to be perfected. The development discussed here has shown that if small changes are not overlooked, further potential for enhancement of performance becomes clear, and if this enhancement is pursued untiringly, the next stage of evolution can emerge.

In future developments, the authors wish to focus more closely on detailed points while engaging in further theoretical research on transmission mechanisms, in order to realize performance increases in CVTs.

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Prediction of Tooth Root Bending Stress in Hypoid Gears for Drivetrain

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ABSTRACT

A technology for the prediction of tooth root bending stress that considers changes in the gear meshing position has been developed to help enable accurate evaluation of the tooth root strength of hypoid gears used in drivetrains. The gear sets in actual vehicles are supported by a number of other components, and deformation of these components in operation produces different gear meshing positions to those observed when the gears are unloaded. These changes in meshing position increase the stress on the tooth roots. A hypoid gear tooth root stress simulation that considers the stiffness of the supporting components was therefore developed. This technology displayed a good correlation with measurement results for stress distribution along the tooth bottom in the direction of the tooth width and for maximum stress values.

Strength evaluations based on tooth root stress obtained in this way also increased the accuracy of prediction of the lifespans of hypoid gears.

1. Introduction

From considerations of environmental protection, the reduction of weight is today demanded in automotive development in order to reduce CO₂ emissions and increase fuel efficiency. At the same time, the transmission torque load on automotive drivetrain parts is increasing, as a result of increases in engine power and response to market needs (for example, in the manufacture of SUVs). In the area of 4WD, high-performance 4WD systems are being developed to boost the driving stability, extending to systems other than those whose main purpose is handling on snowy roads, and the frequency of operation of these systems during vehicle use is also increasing. Against this background, the ability to accurately predict loads on individual parts in order to realize designs that balance low weight with strength is also becoming increasingly important in 4WD development.

This paper will focus on the hypoid gears that determine the drivetrain framework of 4WD systems.

The contact position of the tooth surfaces of hypoid

gears changes under torque load. In addition, because the thickness of the teeth is not constant in the tooth-width direction, the stress generated on the tooth roots fluctuates. The magnitude of the change in meshing position varies with the configuration, arrangement, and shape of the parts that support the gears, and with the amount of transmitted torque, among other factors. The accurate prediction of the stress generated on the tooth roots by these changes in meshing position is an important factor in evaluating the bending strength of hypoid gear tooth roots in actual vehicles.

A method of analyzing bending stress on individual gears has previously been reported as research on tooth root stress in hypoid gears⁽¹⁾.

The Gleason Corporation software^{(2), (3)} generally employed to calculate tooth root bending stress in hypoid gears is an effective tool in conducting calculations for individual gears, but does not enable calculation of the stiffness of the gear supports.

A prediction technology was therefore developed that, using structural analysis, helps to enable the realization of a seamless process from reproduction of the degree of

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change in the meshing position in a real vehicle to the calculation of the bending stress on the tooth roots. The use of this method has increased the accuracy of calculation of tooth root bending stress in hypoid gears.

2. Development Goals

In a torque-splitting 4WD system using a standard FF vehicle as a base, engine torque is transmitted to the rear axle from the transmission positioned on the front axle and the transfer unit attached to its final reduction stage via the propeller shaft and the rear differential unit (Fig. 1).

The function of the transfer unit is to transfer the torque and the rotation split from the final reduction stage in a right angular direction via the hypoid gears, and to output it to the propeller shaft.

The transfer unit has a significant effect on the packaging layout of the engine housing. Given the relationship of positioning between the transfer unit and chassis parts (such as the steering gearbox) and exhaust system parts, if the transfer unit is too large, it may produce a change in the vehicle height or the positioning of the engine. Because of this, it is desirable to reduce the size of the transfer unit.

The use of a cassette method to achieve separation from the transmission and two-stage multiplication using helical and hypoid gears can be identified as specific characteristics of Honda transfer units.

The advantage of the Honda system is that by concentrating functions in the transfer unit, a unified transmission framework can be employed, without consideration of the differences between 2WD and 4WD. In addition, the use of two-stage multiplication offers freedom in the positioning of the transfer unit.

However, as Fig. 2 shows, in addition to reaction force from the hypoid gears, the transfer unit housing is subject to reaction force from the helical driven gears positioned at the exterior of the bearing support points. In addition, because the transfer unit housing is made up of housing

A, which supports the ring gears, and housing B, which supports the pinion gears, a minute degree of slip occurs at the junctions. This slip is also a factor in altering the meshing position of the gears and increasing tooth root bending stress.

It is necessary to achieve sufficient strength in the hypoid gears even under these conditions. Because no method of accurately predicting changes in the meshing position has been available, gear strength has conventionally been evaluated based on experience. This made it necessary to develop a method of analyzing tooth root bending stress capable of reproducing the state of gear-meshing in actual operation.

Given this, development goals were formulated as follows:

- (1) The development of a technology for the prediction of hypoid gear tooth root bending stress that considers the state of meshing in actual operation; and
- (2) The achievement of increased accuracy in predicting the lifespan of hypoid gears.

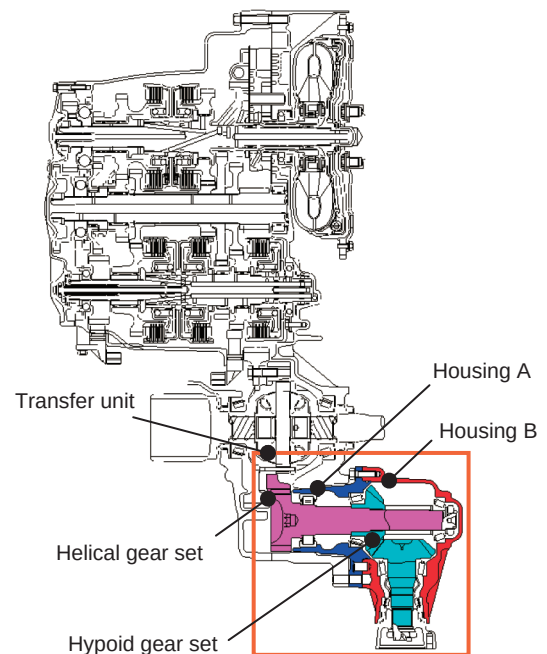


Fig. 2 4WD main section

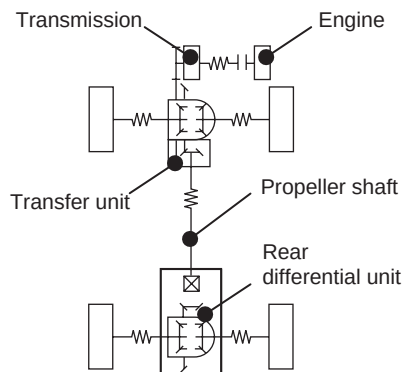


Fig. 1 4WD vehicle model

3. Technology for the Prediction of Hypoid Gear Tooth Root Bending Stress

In order to reproduce the state of meshing in the transfer unit during actual operation, a structural analysis model was formulated that incorporated the stiffness, fitting, and other characteristics of the helical gears, shafts, and hypoid gear set that form the torque transmission path; and of the housing, bearings, and other parts that support them.

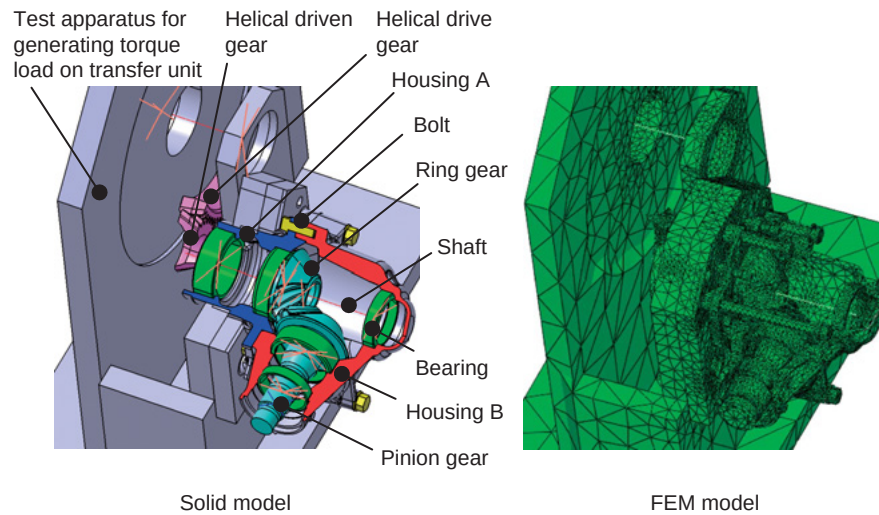


Fig. 3 Components of transfer unit for analysis

Because the formulation of a simulation method was the main focus of the project, the hypoid gear tooth surfaces were modeled after measurement of their actual form using an optical non-contact measurement device⁽⁴⁾.

3.1. Simulation Model

Figure 3 shows the configuration of the transfer unit assembly model used in the simulation of hypoid gear tooth root bending stress.

In order to take into consideration the deformation and the minute degree of slip of the connected surfaces of the housing, as discussed above, a coefficient of friction was defined as a condition for the connected surfaces of the housing, and clearances were set for the fitted sections. In addition, shaft diameter, materials, and axial force were set as parameters for the coupling bolts⁽⁵⁾.

The next part of this section will discuss the modeling of the internal structural parts.

Because the bearings are subject to reaction forces in both radial and thrust directions, it is important to set appropriate stiffness values in relation to both forms of reaction force. Given this, the inner surface of the outer ring was connected to the central point of the bearing

using a rigid bar, and this central point was connected to the shaft, and values were set for the stiffness of the bearings in the radial, axial, and inclined directions (Fig. 4). In addition, defining the external face of the outer ring and the housing as contact surfaces helped to enable the reproduction of the compressive force acting in the direction of load on the bearings and the resulting separation in the opposite direction⁽⁶⁾.

An elastic bar was used for the shaft axle, and the diameter and materials of the axle were defined in order to reproduce deflection and torsion. Finally, the hypoid gears were connected to the shaft axle using a rigid bar, and their fitting with the spline gears was reproduced.

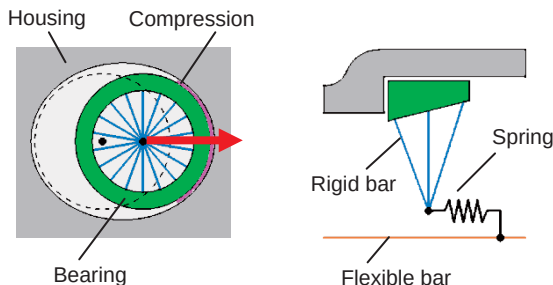


Fig. 4 Bearing model



Fig. 5 3D model making process

3.2. 3D Modeling of Hypoid Gears

Figure 5 shows the flow of gear modeling.

Reverse engineering software was used to generate polygon data in which the tooth surfaces and tooth bases were integrated based on the point group data obtained from the optical non-contact measurement device. At this stage, the polygon data also reproduced the concavities and convexities at the level of tooth surface roughness, including marks left by the tooth cutting and abrasion processes. If this data was employed in the structural analysis, the number of meshes would increase significantly, making it necessary to remove this detailed form data. A section of the measured shape within a range of 5 μm was therefore isolated from the polygon data, and smooth surface data that expressed the form of the grooves on the teeth was generated.

Finally, the gear model was completed by removing this surface form from a solid model of the base material form created from a diagram.

Next, an FEM model was formulated based on this solid model. To help enable the reproduction of torque transmission by gear meshing in the structural analysis, contact surfaces were defined between the ring gear and pinion gear tooth surfaces, and a coefficient of friction was set. Secondary elements were selected to help enable the curved surface form to be traced to the maximum extent possible. A mesh size of 0.4 mm or less was employed for the tooth bases, since it would affect the accuracy of the simulated stress. In addition, in order to reduce the mesh number, six teeth were used, with all teeth that would not affect the results of the analysis being omitted from the model.

3.3. Analysis Procedure

Figure 6 shows the analysis procedure used in this method.

In Step 1, the state of the transfer unit immediately following fitting in a jig was reproduced. First, the tightening axial force on the bolts was input and the housings connected, and the preset load on the bearings was input. At this stage, the hypoid gears and the helical gears were constrained to help prevent rotary motion.

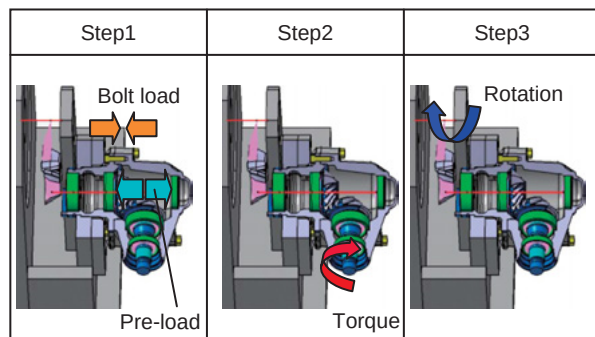


Fig. 6 Analysis procedure

In Step 2, the torque was loaded on the spline section of the pinion gears fitted to the propeller shaft, in order to reproduce the state of the static torque load.

In Step 3, forced rotation was applied to the helical drive gears at the side of the jig in the opposite direction to torque input in order to reproduce the state of meshing during hypoid gear drive. Here, the tooth root bending stress was simulated at 0.5° increments of the angle of rotation of the helical drive gears from the commencement to the completion of hypoid gear meshing.

Finally, the rest of this section will discuss the method employed to process the results.

Tensile stress is generated on the tooth roots of the meshing teeth as shown in Fig. 7.

Figure 8 shows stress contours for three meshing phases from the commencement to the completion of meshing, and examples of the results of calculation of the distribution of tooth root bending stress in the direction

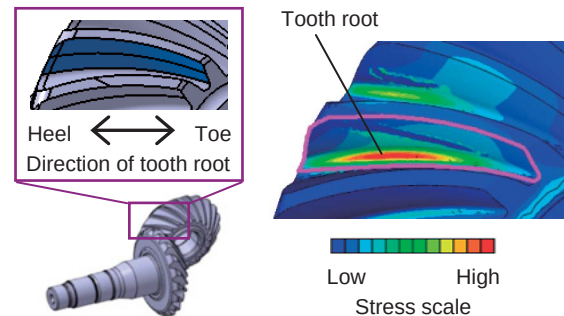


Fig. 7 Tooth root bending stress

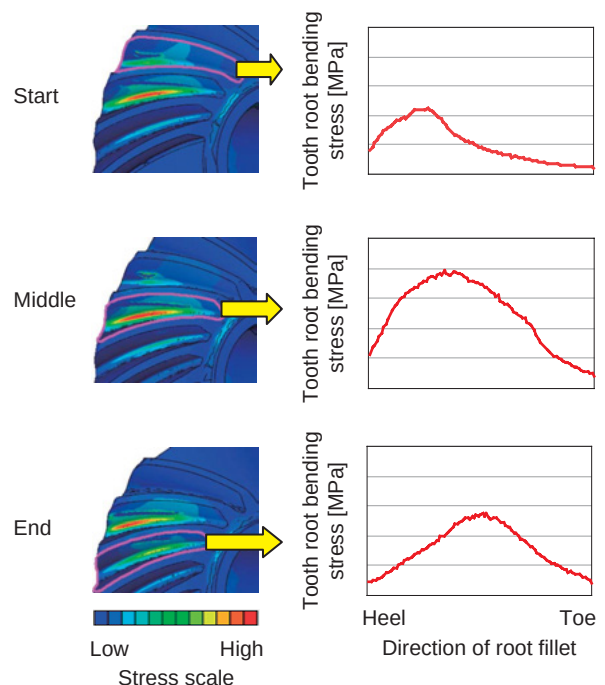


Fig. 8 Tooth root bending stress in different phase

of tooth width. As these examples show, tooth root bending stress varies depending on the phase of meshing. Since strength must be evaluated at the maximum level of stress generated, the results were processed to find the highest stress values in the meshing phases from the commencement to the completion of meshing.

3.4. Simulation Results

The distribution of tooth root bending stress was compared when the stiffness of the gear supports was considered and when it was not considered (Fig. 9) in order to determine the effect of the degree of change in the meshing position on tooth root bending stress. All the support components were defined as rigid bodies in the analysis model that did not consider the stiffness of the supports.

When the stiffness of the supports was considered, the position at which the peak stress was generated shifted in the direction of the heel, and the peak stress value increased. At the same time, the stress in the direction of the toe decreased.

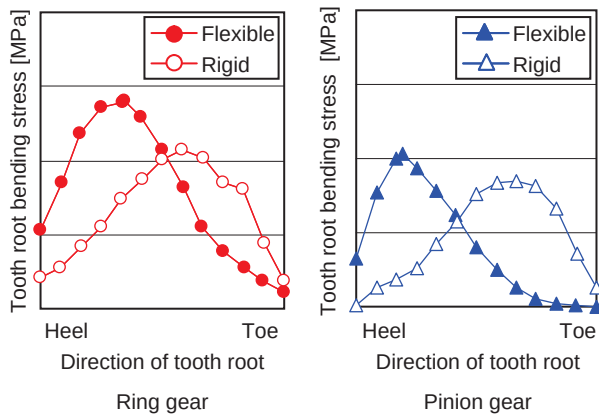


Fig. 9 Influence of stiffness of gear supports

4. Verification of Accuracy of Analysis

4.1. Measurement of Tooth Root Bending Stress

In order to verify the accuracy of the analysis, the tensile stress on the tooth roots was measured with the gears meshed under a static load in the bench-top torsion tester shown in Fig. 10.

The torsion tester has a configuration in which, as in the simulation model, torque is applied from above the pinion gear shaft, and rotation is held steady via the helical gear.

As Fig. 11 shows, for the measurements five strain gauges were attached to one gear tooth. Tooth root bending stress was measured for three gears, with the position of the gauges staggered in the direction of tooth width, giving stress values for 15 points in the direction

of tooth width. The stress values used for verification were, as in the case of the simulation model, the highest values in the phases of meshing of the hypoid gears from commencement to completion.

In past durability tests using actual gears, the hypoid gear set used in the measurements had displayed a tendency for the ring gears to fracture first. In the measurements discussed here also, stress values were highest in the ring gears. Given this background, the ring gears were focused on in the verification process.

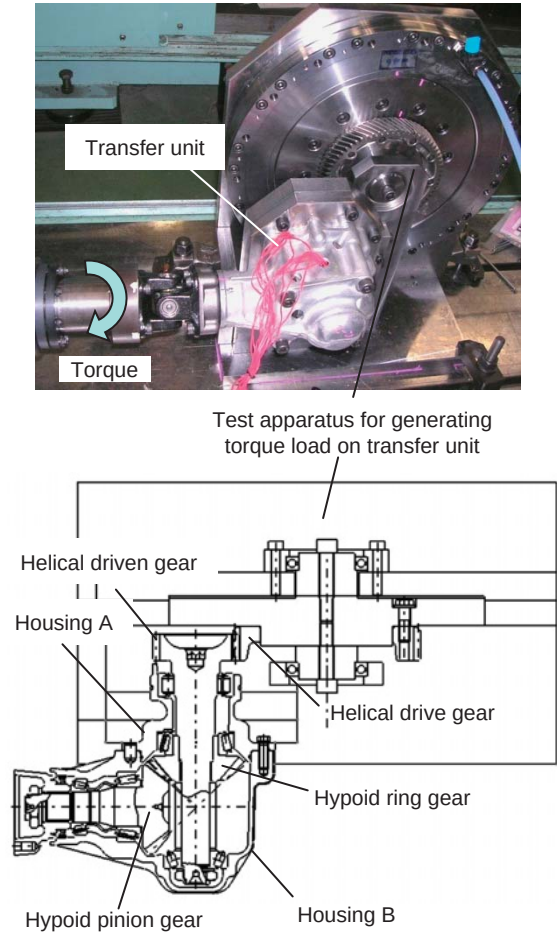


Fig. 10 Torsion tester

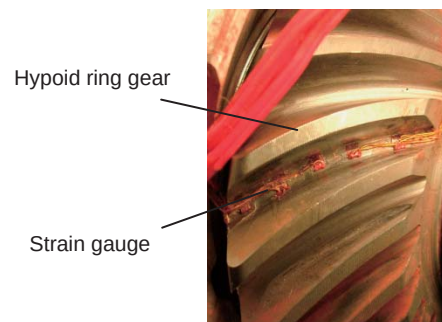


Fig. 11 Gear stress measurement set-up

4.2. Comparison of Calculated Values and Measured Values

Figure 12 shows a comparison of stress distribution for the calculated values and the measured values. A good correlation was obtained between the distributions. In the actual vehicle, the position at which peak stress was

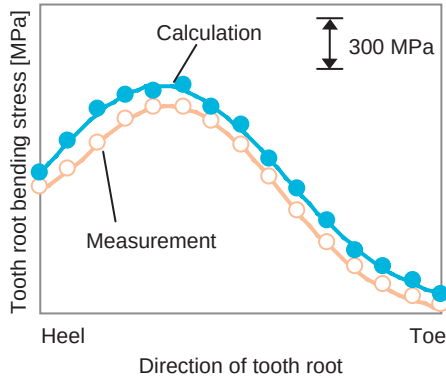


Fig. 12 Distribution of tooth root bending stress for ring gear

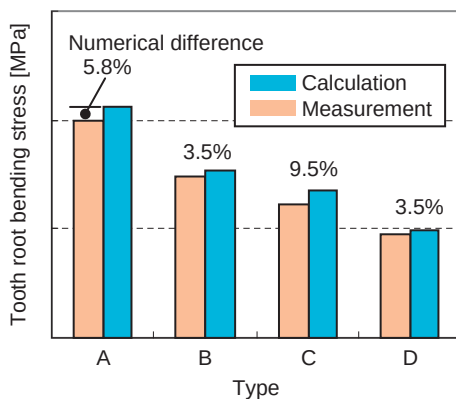


Fig. 13 Comparison of actual measurements and analysis values for tooth root bending stress

generated shifted towards the heel, and the load on the tooth root was not distributed across the entire tooth width. This shows, as in the case of the above-mentioned simulation model that considered the stiffness of the supports, that the degree of change in the meshing position of the gears affects the bending stress on the tooth roots.

Next, the accuracy of the peak values is discussed. Figure 13 shows a comparison between the calculated values and values measured in four hypoid gear sets using different gear sizes. Values were lowest for gear size A, and highest for gear size D. The simulation values tended to be high in comparison to the measured values. This is thought to be due to the fact that the mesh width in the simulation model was low in relation to the width measured by each strain gauge. The difference between the figures is 10% or less.

5. Evaluation of Hypoid Gear Tooth Root Bending Strength

Fatigue tests were conducted on hypoid gears in order to determine the correlation between stress values in the simulation and the number of meshing cycles until fracture.

5.1. Hypoid Gear Fatigue Test

In order to formulate a life diagram, four types of transfer unit were created by assembling combinations of three different gear types of differing sizes and four types of housing of different shapes, and durability tests were conducted (Fig. 14). The materials and carburizing conditions were identical for all the hypoid gears.

Figure 15 shows the durability tester, which was able to apply a continuous high load. The tester detected the vibration generated when a gear broke using a G sensor, and was set to stop automatically when this occurred. In

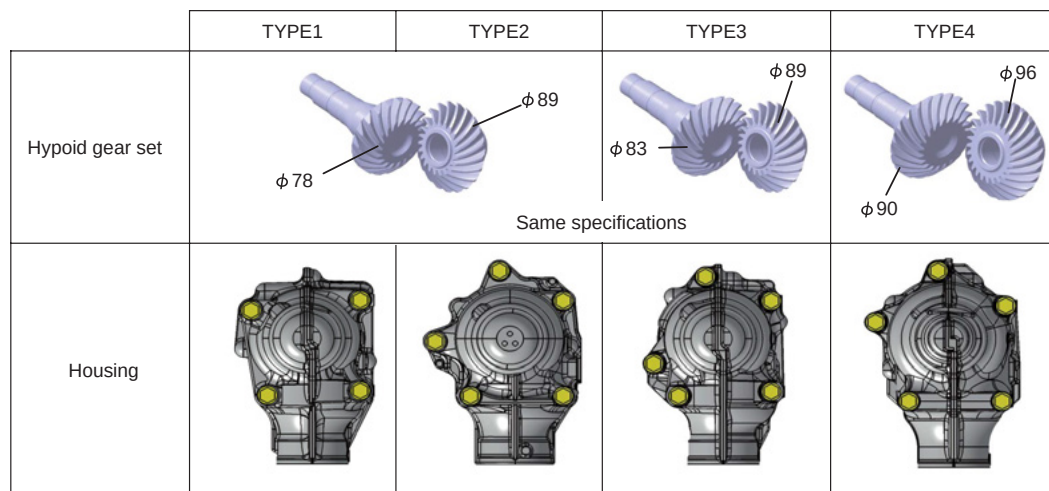


Fig. 14 Combination of hypoid gear sets used for examination and housing

the tests, torque was input from the side of the pinion gears and the unit was operated until a hypoid gear broke. The number of meshing cycles that had elapsed up to that point was then determined.

As operating conditions, input torque values of 500, 750, 900, and 1100 Nm were employed, and the number of input rotations was constant.

Figure 16 shows a gear after the completion of the

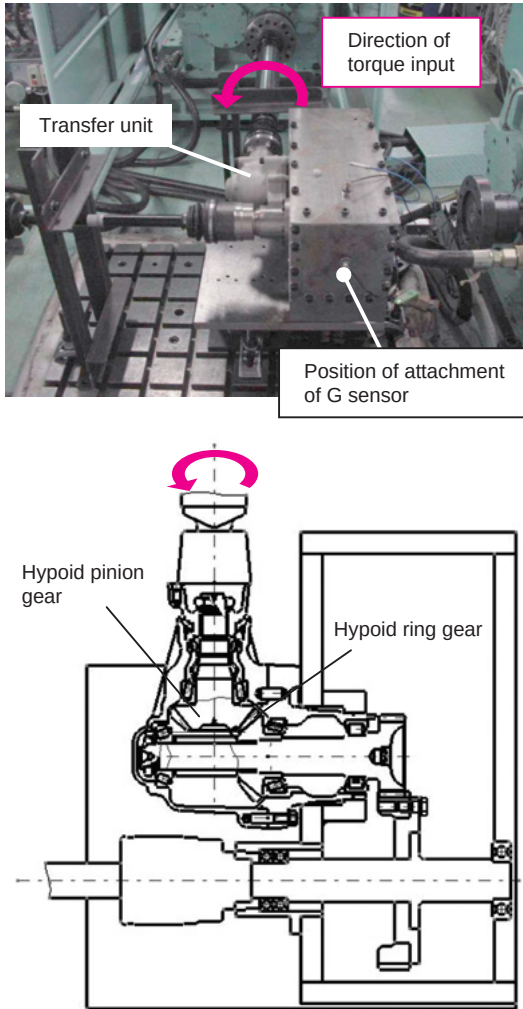


Fig. 15 Equipment used for durability test



Fig. 16 Status of gear damaged in durability test

test. The tooth broke away from the tooth root in the direction of the heel, where tooth root stress is high.

5.2. Relationship between Tooth Root Bending Stress and Durability Life

Figure 17 shows a plot of the gear fatigue test results, with simulation stress values, S , on the vertical axis, and the number of meshing cycles at the point of breakage in the fatigue tests, N , on the horizontal axis.

The coefficient of correlation between stress values obtained from the simulation and gear fatigue life was 0.94. This indicates that the method helps to enable accurate prediction of gear life, and that the size of hypoid gears can be set without relying on experience.

In addition, the usefulness of the method will not be limited to the optimal design of gear size, but can also be used in relation to parameters such as the shape of the housing and the arrangement of the bearings when the transfer unit framework is under consideration at the initial stage of design.

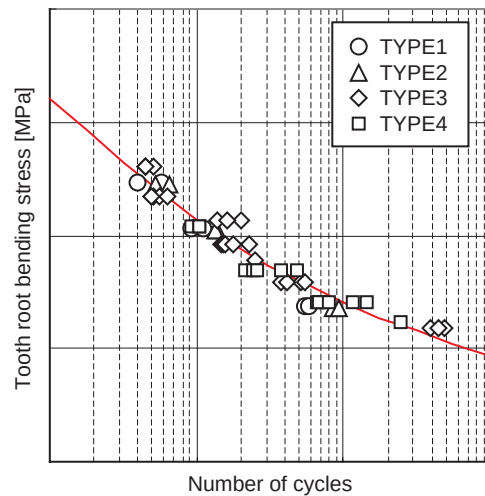


Fig. 17 Stress number diagram

6. Conclusion

- (1) A hypoid gear tooth root bending stress simulation was developed that reproduces the stiffness of the gear supports and the state of gear-meshing, in order to accurately predict tooth root bending stress in drivetrain hypoid gears.
- (2) Simulation stress values predicted measured values to an accuracy of within 10%.
- (3) A high correlation was obtained between simulation stress values and the number of meshing cycles at breakage, indicating that the method helps to enable accurate prediction of fatigue life.

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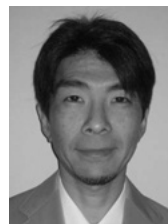
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Development of Active Control Engine Mount System which Reduces Higher Harmonic Vibration

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ABSTRACT

The range of three-cylinder drive can be increased in order to boost the fuel efficiency performance of V6 cylinder-on-demand engines. The use of three-cylinder drive increases the combustion pressure per cylinder, and also increases engine vibration not only at the fundamental order of combustion, but also at higher harmonic frequencies. A new active control engine mount (ACM) system was developed to respond to this higher harmonic vibration in order to satisfy commercial demands for noise and vibration (NV) performance. The operation of an ACM generates higher harmonic vibration other than the vibration produced by the engine. The new ACM system applies sine-wave drive control and multi-order frequency control in order to reduce these two forms of higher harmonic vibration. In addition, the use of a regenerative circuit has reduced heat production, helping to enable a reduction in size of the electronic control unit to be reconciled with reliability. Tests of the system in an actual vehicle showed that NV performance was increased even when an expanded range of three-cylinder drive was employed.

1. Introduction

ACM systems utilizing map control of vacuum actuators⁽¹⁾ and adaptive control of electromagnetic actuators⁽²⁾ as the drive method have been put into practical application. Honda's ACM, developed for use with V6 engines fitted with cylinder-on-demand systems, use a drive method that applies feed-forward control to a highly responsive linear solenoid without the use of specialized sensors⁽³⁾, realizing a level of NV performance that belies three-cylinder drive. The ACM has been employed on the market since the 2004 model year INSPIRE.

The cylinder-on-demand system is a technology that closes one bank of valves during periods of low-load operation, such as at constant speed or during deceleration, in order to reduce pumping loss and increase fuel efficiency. However, due to the reduction in the number of combustion cylinders, three-cylinder operation reduces the 3rd order of engine speed, the engine's fundamental order, to the 1.5th order and increases combustion pressure per cylinder, resulting in increased engine vibration⁽³⁾.

Conventional ACM systems mainly reduce the 1.5th-order component of engine vibration during three-cylinder

operation, and there has been no necessity for ACM to control the 3rd-order component, which is the higher harmonic component.

The development of a new type of ACM system discussed in this paper was conducted to provide a new control system able to respond to an increase in the amplitude of 1.5th-order and 3rd-order engine vibrations consequent upon an expansion of the range of three-cylinder drive in order to increase the fuel efficiency performance of the cylinder-on-demand system. The aim of the development was to achieve a level of NV performance equivalent to that of the previous system, at which users are unaware that three-cylinder drive is in operation.

2. Development Goals

The cylinder-on-demand system with an expanded range of three-cylinder drive will be termed the new cylinder-on-demand system, and the ACM that realized a degree of displacement that cancelled the vibration of the new cylinder-on-demand system will be termed the new form of ACM.

The power of the new form of ACM was increased by

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reducing coil resistance against the previous ACM in order to increase current responsiveness; this increased electromagnetic energy, thus boosting ACM output.

Figure 1 shows the load transmitted to the body from an engine bracket during three-cylinder drive in vehicles fitted with the conventional and new cylinder-on-demand systems. In the vehicle fitted with the new cylinder-on-demand system, there is greater engagement of three-cylinder drive in the range in which torque is necessary than in the vehicle fitted with the previous system. Because of this, the 1.5th-order and 3rd-order loads are both greater, and depending on the operating range, the 3rd-order load can reach a level at which NV targets are not satisfied. It was therefore necessary to drive the ACM using a form of control that was able to respond to 3rd-order vibration in addition to 1.5th-order vibration, in order to reduce the transmitted loads.

Figure 2 shows the force generated by the previous ACM and the new form of ACM in isolation when the units are driven by a current-impressing device under

identical conditions and a load corresponding to the weight of the engine is applied. Both ACMs produce a higher harmonic component, but the higher harmonic component is greater in the new form of ACM in proportion to the degree to which the 1st-order component increases in response to the fundamental order of engine vibration. If the new form of ACM was used to cancel engine vibration in this state, the fundamental order of engine vibration would be reduced, but the vibration of the ACM would produce vibration at higher harmonic frequencies. The reason for this is that the ACM could not be operated to render the force generated by the ACM into a sine wave.

Based on the above, the targets established for the development of the new form of ACM were, in addition to increasing ACM power in order to respond to an increased amplitude of engine vibration, to reduce the higher harmonic vibration from the engine and ACM, the new and conspicuous factor generating a decline in NV performance, to a level acceptable in a marketed vehicle.

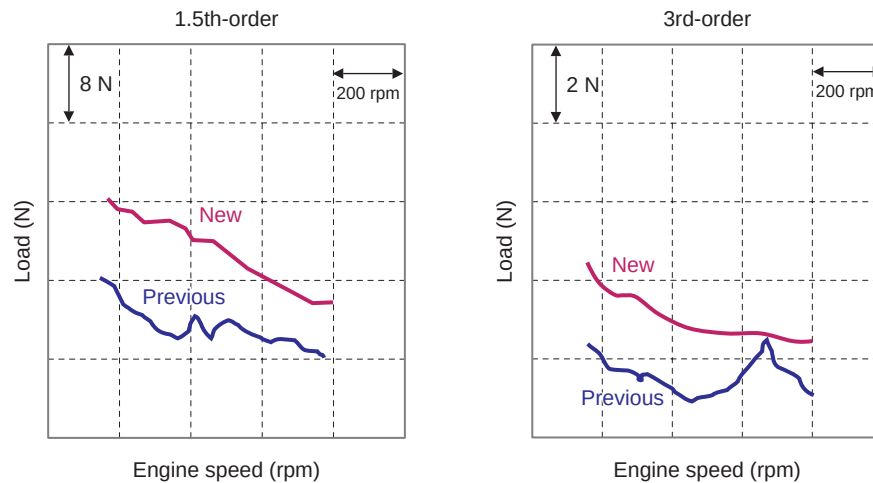


Fig. 1 Load transmitted to body from engine bracket during 3-cylinder stop mode

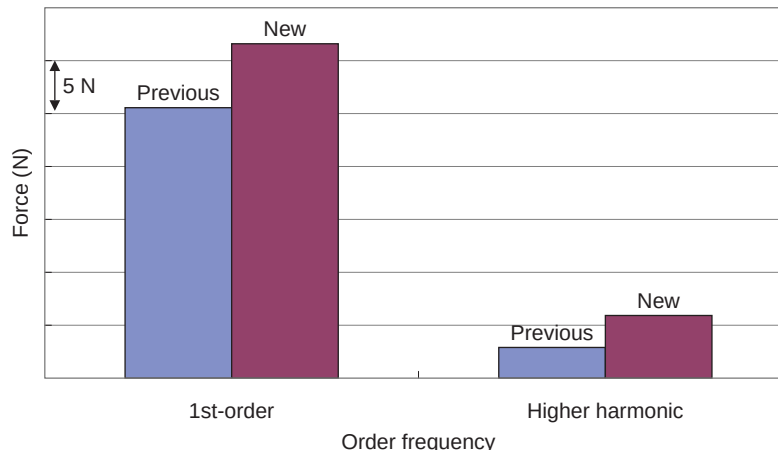


Fig. 2 Order frequency of output force from developed and previous ACM

3. Configuration of ACM System

3.1. System Overview

Figure 3 shows an example of the ACM system when fitted in a vehicle. ACMs are fitted at both the front and rear of a V6 engine provided with the cylinder-on-demand system, and the electronic control unit (ECU) that controls the ACM is located in the vehicle cabin.

Figure 4 shows the system configuration. The ACM ECU is connected to the engine control unit (PGM-FI), to help ensure that it receives the necessary signals to track engine vibration.

Figure 5 shows the configuration of the ACM. The ACM is made up of a liquid seal section and a linear solenoid section. Hydraulic pressure is varied through the use of an armature in the linear solenoid section to induce vibration in an exciter plate in the liquid seal section. This results in vibration of the mounting rubber. By this means, the dynamic spring constant of the ACM can be varied, helping to enable the load transmitted from the engine brackets to the body (sub-frame) to be reduced.

3.2. Generation of Higher Harmonic Component from ACM Output

When a sinusoidal current was impressed on the ACM, a higher harmonic component was generated from the ACM output. The ACM output is determined by the movement of the exciter plate. The displacement of the exciter plate, X , can be found using Eq.(1). Figure 6 shows a simple model of the exciter plate and the linear

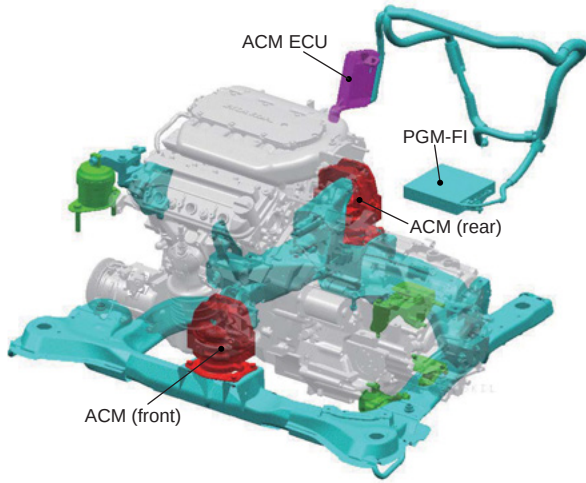


Fig. 3 ACM system configuration

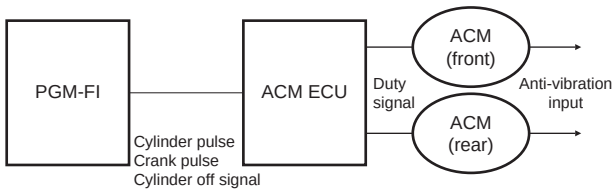


Fig. 4 Block diagram of ACM system

solenoid, used for deriving Eq. (1).

$$F_{act}(i, X) = M\ddot{X} + \{Ce + Cj(i)\}\dot{X} + \{Ke + Kj(i)\}X + \mu(i, \dot{X})F_n(i, X) + F_{BH}(i) \quad (1)$$

F_{act} : Magnetic force

F_n : Horizontal component of magnetic force

F_{BH} : B-H hysteresis

M : Total moving mass

Ce : Attenuation coefficient of exciter plate

Cj : Attenuation coefficient which changes with current

Ke : Spring constant of exciter plate

Kj : Spring constant which changes with current

μ : Coefficient of operation friction on bearing

i : Drive current

X : Displacement of exciter plate

The parameters displaying nonlinear characteristics in Eq. (1) are (1) the attenuation characteristics Ce and Cj , (2) the sliding resistance of the bearings, μF_n , and (3) the B-H hysteresis, F_{BH} , and these parameters are factors in the generation of the higher harmonic component.

The developed system allows this nonlinearity of the hardware, and applies a new method of control in order to reduce the higher harmonic component.

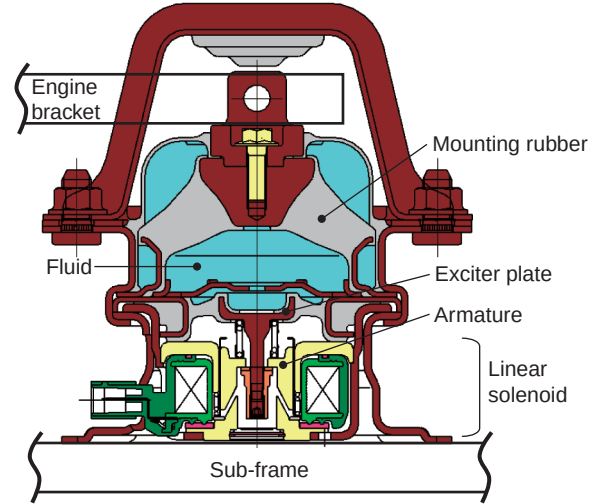


Fig. 5 Cutaway view of ACM

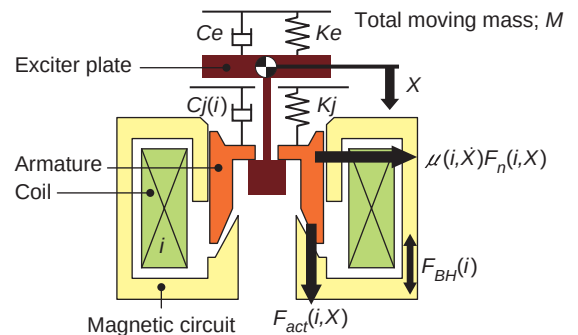


Fig. 6 Exciter plate displacement model

4. Control for Reduction of Higher Harmonic Component

4.1. Sine-wave Drive Control

Figures 7 and 8, respectively, show the magnitude of the frequency order components of the higher harmonic output and the phase difference of the higher harmonic wave with the 1st-order component when the ACM is driven by impressing a sinusoidal current. The main components of the higher harmonic output are the 2nd and 3rd orders, and its phase difference is reproducible as linear or nonlinear against the frequency. Therefore, if a current was synthesized so that the 2nd- and 3rd-order components of the impressed current were reduced, sine-wave drive of the ACM would be possible, and the higher harmonic output could be reduced.

If I_1 , I_2 , I_3 , θ_2 , and θ_3 represent, respectively, the magnitude of the 1st-, 2nd- and 3rd-order currents, and the phase of the 2nd- and 3rd-order currents, f represents the frequency of the 1st-order current, and t represents the period of ACM drive, $ICMD$ in Eq. (2) gives the impressed current necessary for sine-wave drive of the ACM.

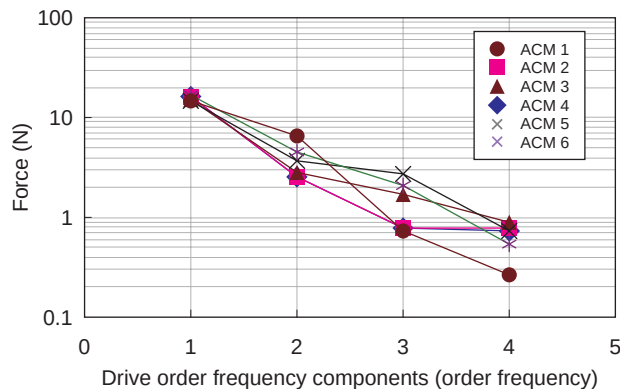


Fig. 7 Components of higher harmonic wave

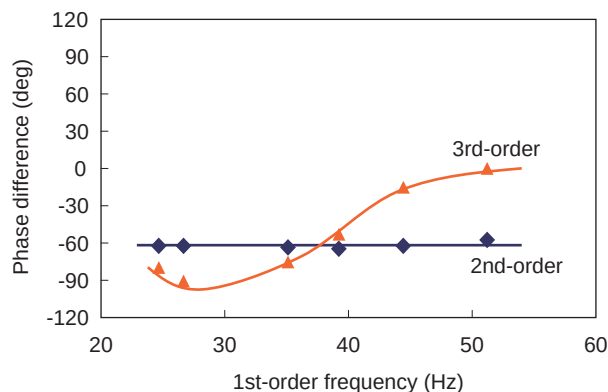


Fig. 8 Phase difference of higher harmonic wave with 1st-order

$$ICMD = I_1 \sin \omega t + I_2 \sin(2\omega t + \theta_2) + I_3 \sin(3\omega t + \theta_3) \quad (2)$$

Here,

$$\omega = 2\pi f \quad (3)$$

I_1 is implemented in the ECU as a map of engine speed and estimates of engine vibration. Because the values of I_2 , I_3 , θ_2 and θ_3 vary in response to the magnitude and frequency of the 1st-order current, they are implemented as a map (ICMD map) of the magnitude and frequency of the 1st-order current.

It was necessary to develop a high-precision current control to drive the ACM following Eq. (2). Equations (4) to (7) show the method used for current control. I_{ACT} is the actual current value for each ACM drive cycle. The power to be supplied to the linear solenoid, $DUTY$, is calculated to render the values of I_{ACT} and $ICMD$ identical. Current feedback control is the basic framework of this method of current control, but because $ICMD$ varies with each ACM drive cycle, traceability could not be ensured using only the feedback terms $DSOLP$ (the proportion term) and $DSOLI$ (the integration term). Therefore, using the fact that Eq. (2) is already known, the feed-forward term $DSOLFF$ is added.

In addition, the ACM drive circuits providing feed-forward and feedback control of current should be sufficiently responsive to be able to reduce the current to 0 A within the frequency of the 1st-order current even under conditions in which inductance impedes the ability to reduce the current. This is because, due to the structure of the ACM, if a direct current component remains, the armature will be drawn up and the vibrating center of the exciter plate will be offset, reducing the level of cancellation displacement against the impressed current.

Therefore, situations when it is necessary to increase the current and when it is necessary to decrease the current are responded to by switching the drive circuit.

Figure 9 shows the ACM drive circuit. The drive circuit is made up of a FET (T1, T2) half-bridge circuit, a FET (T3) that switches responsiveness when the

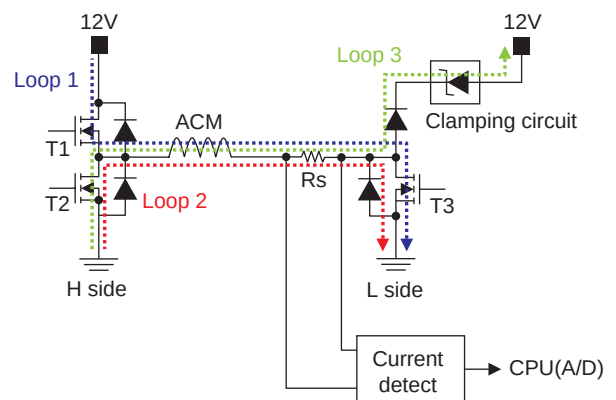


Fig. 9 ACM drive circuit

current is reduced, and a circuit that detects feedback current (shunt resistor R_s). Figure 10 shows the current waveform during ACM drive. When the current is increased, PWM control is applied to T1 and T2 to switch to the current pathways shown as Loop 1 and Loop 2. When the current is decreased, PWM control is applied to T3 to switch to the current pathways Loop 2 and Loop 3.

Because current responsiveness is determined by the magnitude of the voltage impressed on the ACM, current responsiveness in the current pathway Loop 3 was increased by increasing the voltage impressed on the ACM to a level higher than the source voltage, and adding a clamping circuit that regenerates energy and supplies it to the power source.

The command to switch the drive circuit is provided by the *DUTY* code. When it is positive, Loop 1 and Loop 2 are selected, and when it is negative, Loop 2 and Loop 3 are selected. In addition, because responsiveness varies in relation to deviation ($ICMD - IACT$) when the drive circuit is switched, the proportional gain, K_p , and the integration gain, K_i , are switched in Loops 1, 2, and 3.

$$DUTY = DSOLFF + DSOLP + DSOLI_{(n)} \quad (4)$$

$$DSOLFF = (K_p + K_i) \times \{ICMD_{(n+1)} - IACT_{(n)}\} \quad (5)$$

$$DSOLP = K_p \times \{ICMD_{(n)} - IACT_{(n)}\} \quad (6)$$

$$DSOLI_{(n)} = K_i \times \{ICMD_{(n)} - IACT_{(n)}\} + DSOLI_{(n-1)} \quad (7)$$

4.2. Unit Tests

Figure 11 shows the reduction of the higher harmonic wave during ACM operation in unit tests, and Fig. 12

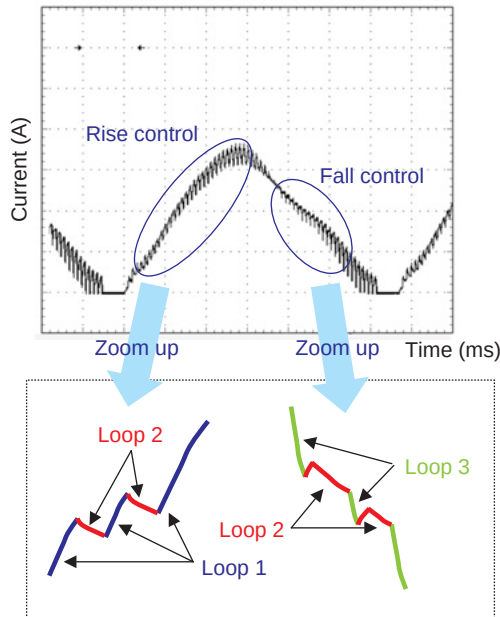


Fig. 10 Driving current waveform

shows the force waveform generated by the ACM at this time. The use of the new current control method has reduced the higher harmonic components (2nd- and 3rd-order) against drive using a sinusoidal current.

4.3. Reduction of Higher Harmonic Wave and Multi-order Frequency Control as Related to NV in an Actual Vehicle

Using sine-wave drive control, it was possible to apply sine-wave drive to the ACM.

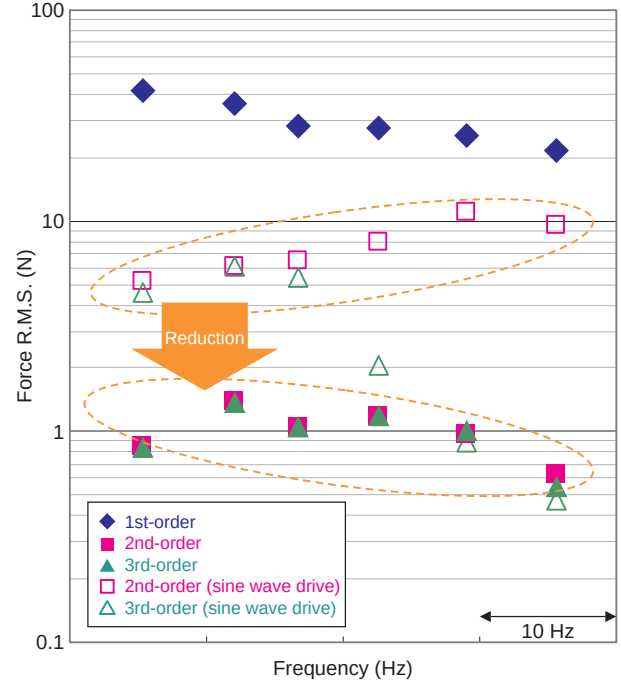


Fig. 11 Reduction of higher harmonic wave

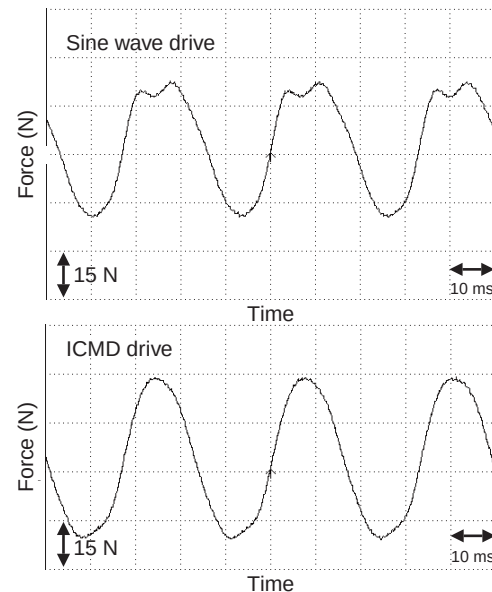


Fig. 12 Force waveform

Engine vibration can be considered to be a synthesis of the fundamental order and the higher harmonic components. Given this, if the ACM could be driven using a synthesis of the ACM sine-wave drive current form effective against the fundamental order and the ACM sine-wave drive current form effective against the major higher harmonic components, it would be possible to reduce engine vibration at multiple frequency orders.

When Eq. (2) is substituted for $ICMD_m$, the ACM sine-wave drive current form effective against the fundamental order of engine vibration, because the frequency of the current of each order is double $ICMD_m$, $ICMD_h$, the ACM sine-wave drive current form effective against the major higher harmonic components, can be expressed as Eqs. (9) and (10), in which f in Eq. (3) has been transformed into $2f$, and $ICMD$ in Eq. (8) is the current form for input to feed-forward and feedback control when reducing engine vibration at multiple frequency orders. I''_1 , I''_2 , I''_3 , θ''_2 , and θ''_3 represent, respectively, the magnitude of the 1st-, 2nd- and 3rd-order currents, and the 2nd- and 3rd-order current phases, all in relation to the higher harmonic components. θ represents the phase of $ICMD_h$ in relation to $ICMD_m$. I''_1 and θ are implemented in the ECU as a map of engine speed and estimated engine vibration. I''_2 ,

I''_3 , θ''_2 , and θ''_3 are calculated from an ICMD map into which I''_1 and $2f$ have been input.

In addition, because the current form is calculated by the ECU for each ACM drive cycle, the lowest point for $ICMD$ in Eq. (8) can be offset, and the offset is corrected to make the lowest point 0 A.

Figure 13 shows the control block.

$$ICMD = ICMD_m + ICMD_h \quad (8)$$

$$ICMD_h = I''_1 \sin(\omega''t + \theta) + I''_2 \sin(2\omega''t + 2\theta + \theta''_2) + I''_3 \sin(3\omega''t + 3\theta + \theta''_3) \quad (9)$$

Here,

$$\omega'' = 2\pi(2f) \quad (10)$$

5. Effect on NV in an Actual Vehicle

Figure 14 shows the effect of the new system in reducing the transmitted load during three-cylinder drive when the ACM is in operation. The new form of ACM reduces the transmitted load for the 3rd-order component of engine vibration to less than half the figure when the ACM is not in operation, and is also effective in reducing the higher harmonic wave during three-cylinder drive.

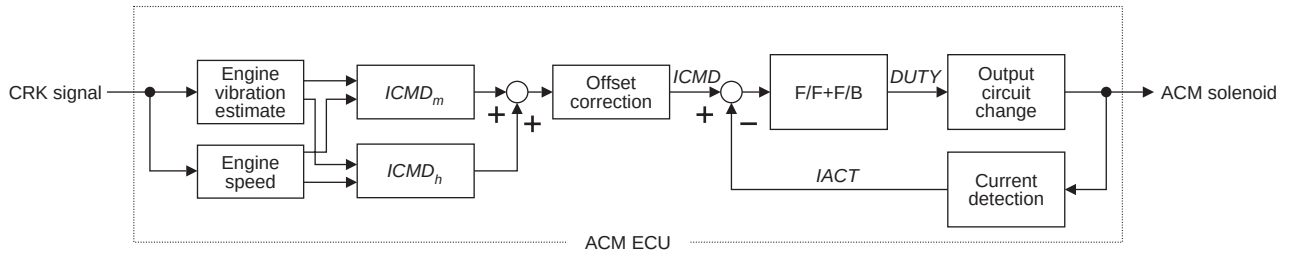


Fig. 13 Control block diagram

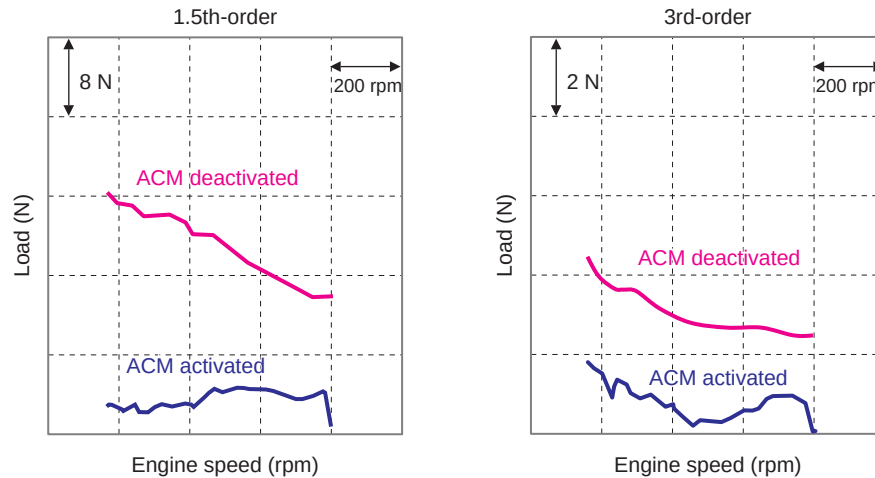


Fig. 14 Effect of new ACM system

Table 1 Specifications of developed ECU

Items	Developed ECU
PCB and frame	
Power (max)	72 W (12 V × 6 A)
Circuit loss	6 W
Size	86 × 98 × 35 mm
Weight	295 g

References

- (1) Ozaki, H., Tsukamoto, T., Ichikawa, A., Yamazoe, H., Shibata, A., Maeno, T., Tajima, H.: Development of an Active Control Engine Mount, 1998 JSAE Annual Congress Proceedings, No. 983, p. 141-144 (1998)
- (2) Aoki, K., Hirade, T., Hyoudou, Y., Aihara, T., Shikata, T.: Noise and Vibration Reduction of Diesel Engine Vehicle Making Use of the Active Control Engine Mount (ACM) System, 1998 JSAE Annual Congress Proceedings, No. 983, p. 133-136 (1998)
- (3) Matsuoka, H., Mikasa, T., Nemoto, H.: Development of Active Control Mount, Honda R&D Technical Review, Vol. 15, No. 2, p. 209-214

6. Reduction of ECU Loss and Achievement of Size Reductions

Increasing the power of the ACM involved an increase in the maximum output of the ECU from the 60 W of the previous specification to 72 W, while the optimization of the drive circuits through the use of regenerative circuits and other measures reduced circuit loss to approximately 6 W against approximately 15 W in the previous specification. This helped to enable the weight of the heat dissipating frame to be reduced by 44% and the size of the ECU by 35%, reconciling increased power with reduced size (Table 1).

7. Conclusion

An ACM system that employs feed-forward and feedback current control was developed in order to respond to an increase in the amplitude of engine vibrations resulting from the use of a new cylinder-on-demand system. The following results were obtained:

- (1) A level of NV performance was achieved such that users are unaware of the operation of three-cylinder drive in the new cylinder-on-demand system.
- (2) A control method making possible sine-wave drive of the ACM was developed, and the higher harmonic vibration generated by the ACM was reduced.
- (3) A multi-order frequency control method was developed that allowed the synthesis of sine-wave drive current forms to be effective against the fundamental order and the higher harmonic components of the engine vibration cycles, reducing the higher harmonic vibration transmitted from the engine brackets to the vehicle body.
- (4) Optimization of the drive circuits reduced loss, helping to enable increased ACM power to be reconciled with the reduction in size of the ECU.

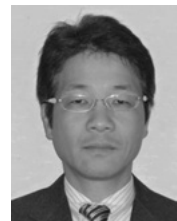
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Proposal of Sensorless Control Method for Small Brushless DC Motor

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ABSTRACT

On-car motor quantities and applications have tended to increase in recent years. This is increasing the total size and weight of motors per vehicle, and operating environments are also becoming more severe, including mounting inside the engine room, resulting in an increasing trend towards replacement with compact, lightweight brushless DC motors that have better durability under harsh environments. However, as phase current and rotor position sensors are generally needed to control brushless DC motors, the cost tends to rise and reducing cost is an issue. An algorithm was therefore constructed with the aim of reducing cost while maintaining brushless DC motor performance and reliability. This algorithm estimates both the phase current and the rotor position, without using phase current or rotor position sensors, by using the DC current sensor for fail detection that was originally present in the controller, and the motor drive voltage information. The newly constructed method provided prospects for reducing brushless DC motor cost.

1. Introduction

Recently, attention is being given to safety and environmental measures in addition to the basic automobile performance dynamics of running, turning and stopping, and functions are being enhanced through the introduction of electronic control to various systems. Typical drive actuators converted to electronic control include motors, and with application increases come demands for compact size, high performance, high reliability, low cost, and low power consumption. Motor usage originally expanded from DC motors with brushes, but the switchover to brushless DC (BLDC) motors is accelerating from the standpoints of enhancing performance and reliability and reducing power consumption. However, although the motor unit of a BLDC motor is less expensive than that of a DC motor, there is the issue of overall cost increase due to factors such as the addition of the rotor position and phase current sensors required for motor control, and increasingly complex drive circuits and control logic.

This study aimed to make BLDC motor cost competitive with that of DC motors. To achieve this, the authors focused on the fact that the two sensors required for motor control are the main factors increasing the cost, and proposed a sensorless control method that targets power-use motors that mainly rotate in a single direction,

and which are often used in cars.

Much research has been done on sensorless control methods that involve rotor position estimation⁽¹⁾⁻⁽⁴⁾. However, there is little research on methods that simultaneously estimate both the phase current and the rotor position⁽⁵⁾. The cited study used past estimation values in areas where phase current estimation is a challenge, and applied these values to simplified vector control without a speed controller or a current controller, limiting such motor control performance as responsiveness. Therefore, this study proposes a sensorless control algorithm that simultaneously estimates the phase current and the rotor position, and can also be applied to generally used vector control.

The hardware comprises the conventional BLDC motor control system minus the two sensors and their respective interface circuits.

Regarding phase current estimation, intermittent phase current information is obtained from the DC current sensor for fail detection, in accordance with the FET switching pattern obtained by performing pulse-width modulation (PWM) on the motor drive voltage. Focusing on the symmetry of the FET switching pattern, phase current estimation is performed at the timing appropriate for rotor position estimation by detecting the DC current.

On the other hand, rotor position estimation uses the motor drive voltage and the phase current, which have an

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Table 1 Motor classification by use

	Power use		Servo use	
Control	Torque control		Speed control; Position control	
Specific examples	Household electrical appliance; Factory automation	Car application	Household electrical appliance; Factory automation	Car application
	Vacuum cleaner; Washing machine	Oil pump; Water pump; Fan	Robot	Variable lift system

input-output relationship relative to the motor. A motor model that is not dependent on the rotor position and rotational speed is configured within the CPU used for motor control, with the voltage applied to the motor used as the input, and the model phase current obtained as the output. The rotor position is estimated by comparing the model phase current with the phase current estimated from the DC current that has been detected according to the FET switching pattern, and extracting the rotor position and rotational speed information.

2. Current Situation and Issues for Compact BLDC Motors

Table 1 shows BLDC motor classification.

BLDC motor applications are broadly divided into power use and servo use, and a significant difference

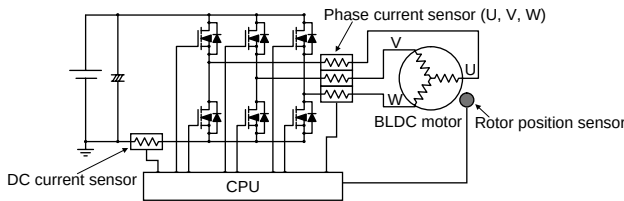


Fig. 1 BLDC motor drive electric circuit

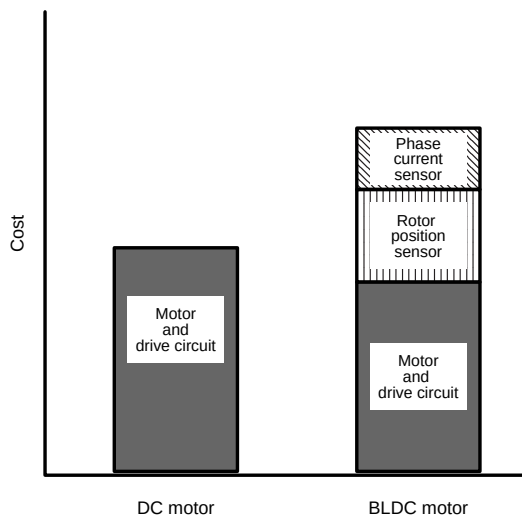


Fig. 2 DC motor and BLDC motor costs

between the two is the different motor control methods used. This study targeted power-use motors that mainly rotate in a single direction, and which are often used in cars.

Figure 1 shows a typical BLDC motor drive circuit. The circuit comprises a smoothing capacitor, FET bridge, a DC current sensor for fail detection located between the smoothing capacitor and FET bridge, a CPU, and the two types of sensors conventionally required for BLDC motor driving: a phase current sensor (U, V, W), and a rotor position sensor mounted on the motor rotor.

Figure 2 compares the costs of a typical DC motor and BLDC motor. This shows that BLDC motors tend to be more expensive than DC motors, and that the factors increasing the cost are the phase current and rotor position sensors. Therefore, of the three types of sensors, this study simultaneously eliminated the phase current and rotor position sensors that are the factors increasing the cost.

3. Control Algorithm

The following control methods used to eliminate the phase current and rotor position sensors, accurately estimate the phase current and rotor position, and achieve the target performances are described below.

- (1) Phase current estimation algorithm
- (2) Rotor position estimation algorithm
- (3) Coordination between rotor position estimation and phase current estimation

3.1. Phase Current Estimation Algorithm

The conventional phase current estimation method detects and uses the phase current information at the timing according to the FET switching pattern using a DC current sensor. The phase current information for each phase is not the same timing, which made it challenging to accurately estimate phase current. In addition, rotor position estimation requires phase current information that matches the command voltage update timing, but the phase current information obtained by the conventional phase current estimation method does not match the command voltage update timing, and is instead the information at the timing according to the FET switching pattern. This made it challenging to accurately estimate the rotor position.

Therefore, focusing on the symmetry of the FET switching pattern, an algorithm is proposed that accurately estimates the phase current at the command voltage update timing by detecting the DC current.

Figure 3 shows the proposed phase current estimation algorithm. The controller outputs the command voltage according to the command torque and the phase current. The command voltage is compared with the carrier wave and converted to the FET switching pattern by the PWM modulator. The phase current is generated and the motor

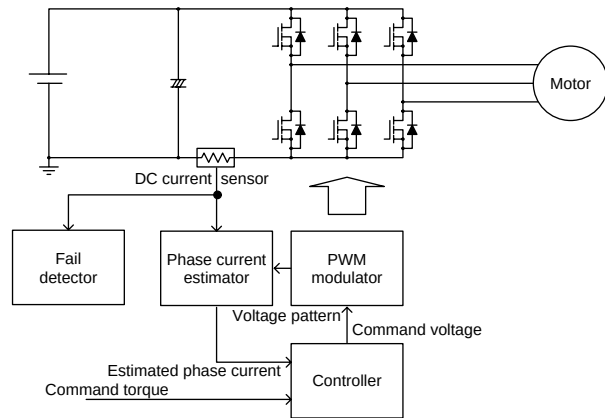


Fig. 3 Block diagram of phase current estimation system

Table 2 Relationship between basic voltage vector and detected current (I_{dc})

Voltage vector	Phase U	Phase V	Phase W	Detected current (I_{dc})
V0-	Low	Low	Low	0
Vu+	High	Low	Low	I_u
Vv+	Low	High	Low	I_v
Vw-	High	High	Low	$-I_w$
Vw+	Low	Low	High	I_w
Vv-	High	Low	High	$-I_v$
Vu-	Low	High	High	$-I_u$
V0+	High	High	High	0

is driven by switching the FET in accordance with this pattern. On the other hand, an intermittent phase current flows to the DC current sensor for fail detection in accordance with the FET switching pattern. Therefore, the phase current can be estimated by obtaining the DC current from the DC current sensor detection in accordance with the FET switching pattern.

The FET switching pattern is determined by the ON/OFF combination (8 patterns) of the 6 FET in the motor drive circuit. Figure 4 shows each FET operation pattern. Upper FET ON is indicated by orange, and lower FET ON is indicated by blue. The zero voltage vectors (V0-, V0+) and voltage vectors offset by a 60 deg phase from each other (Vu+, Vu-, Vv+, Vv-, Vw+, Vw-) can be generated according to the FET ON/OFF combinations. Table 2 describes the relationship between each voltage vector and

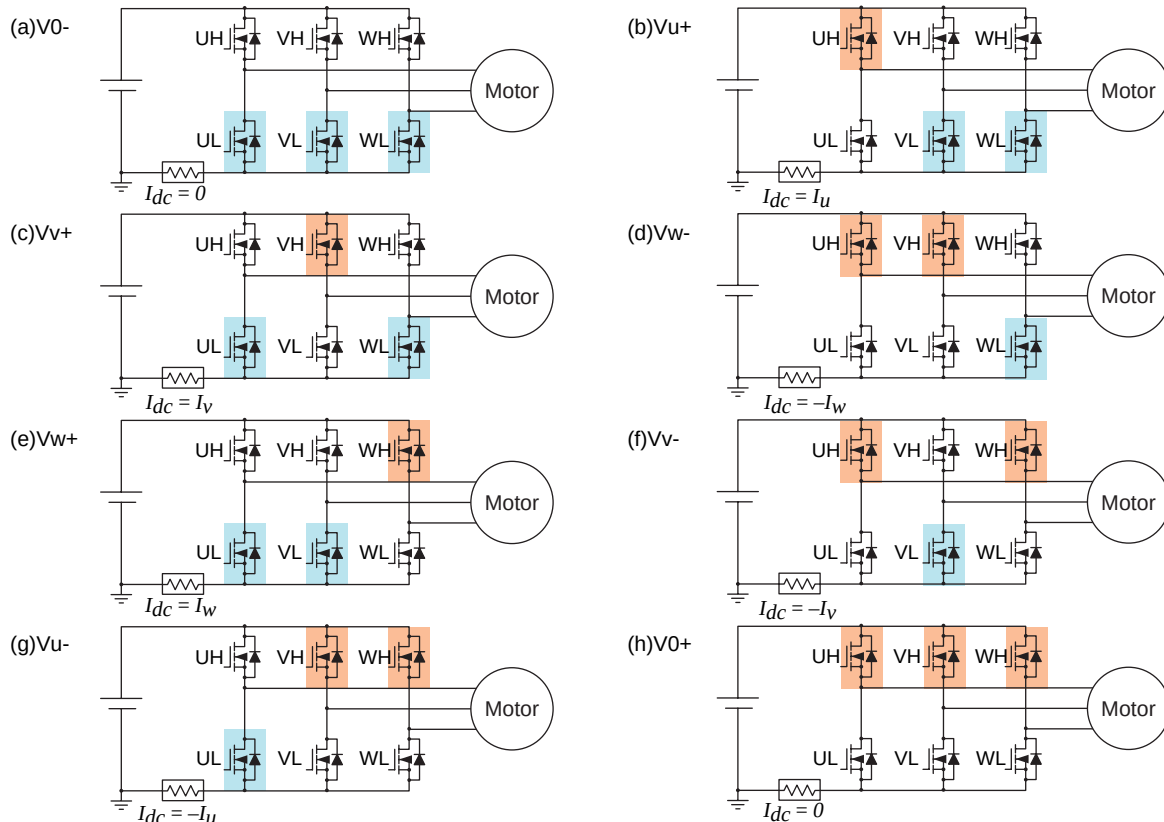


Fig. 4 Relationship between basic voltage vector and detected current

the phase currents that can be detected by the DC current sensor.

Figure 5 shows the FET switching pattern obtained by performing PWM on the command voltages using a triangular carrier wave. The command voltages are generated according to the FET switching pattern within one carrier wave cycle. When the command voltages (V_u^* , V_v^* , V_w^*) are larger than the carrier wave, the respective upper FET turn ON (orange). Conversely, when the command voltages are smaller than the carrier wave, the respective lower FET turn ON (blue). Using the voltage vectors shown in Fig. 4, the FET switching patterns per carrier wave cycle are V_0^- , V_u^+ , V_w^- , V_0^+ , V_w^- , V_u^+ and V_0^- .

Figure 6 shows the relationship between the FET switching pattern and the phase current information that can be detected by the DC current sensor. As shown in Table 2, when the voltage pattern is V_u^+ , the U phase

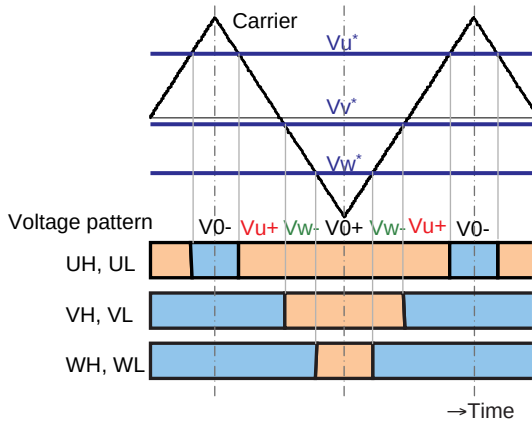


Fig. 5 Relationship between command voltage and voltage pattern

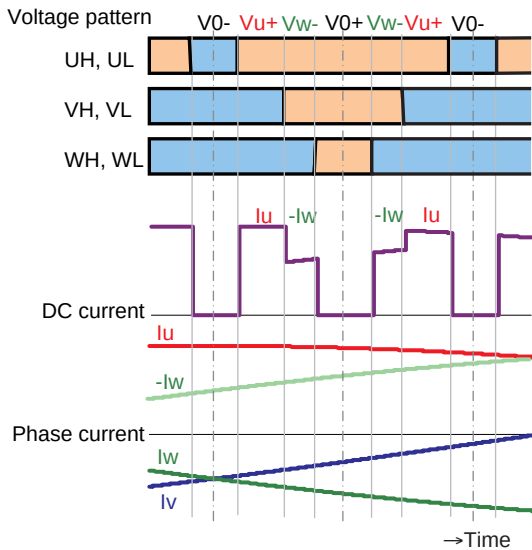


Fig. 6 Relationship between voltage pattern and phase current

current (I_u) can be detected, and when the voltage pattern is V_w^- , the value with the W phase current (I_w) polarity inverted can be detected. The phase current information for two phases can be detected during one cycle from the carrier wave peak to the next peak, and the remaining phase currents can then also be estimated using the principle that the sum of the phase currents of the three BLDC motor phases is zero.

However, the estimated phase currents are not the phase current information aligned with the command voltage update timing, and the timing of the phase current information for each phase also differs. This makes it challenging to accurately estimate the phase current.

Therefore, Fig. 7 below shows an average calculation method that uses the symmetry of the carrier wave peaks as a means of reducing the phase current estimation error. This shows that the FET switching pattern per carrier wave cycle is V_0^- , V_u^+ , V_w^- , V_0^+ , V_w^- , V_u^+ and V_0^- , and that the pattern is symmetrical relative to the carrier wave trough ($T_{c[m-1]}$). Focusing on the symmetry of the FET switching timing, the phase current values at the trough of the carrier wave (■ marks in the figure) are estimated by obtaining the U phase currents (I_{u1} , I_{u2}) and W phase currents (I_{w1} , I_{w2}) at two symmetrical timings relative to the carrier wave trough (● marks in the figure), and averaging those values (I_u , I_w). The remaining V phase current (I_v) is then calculated using the estimated U phase current (I_u), the estimated W phase current (I_w),

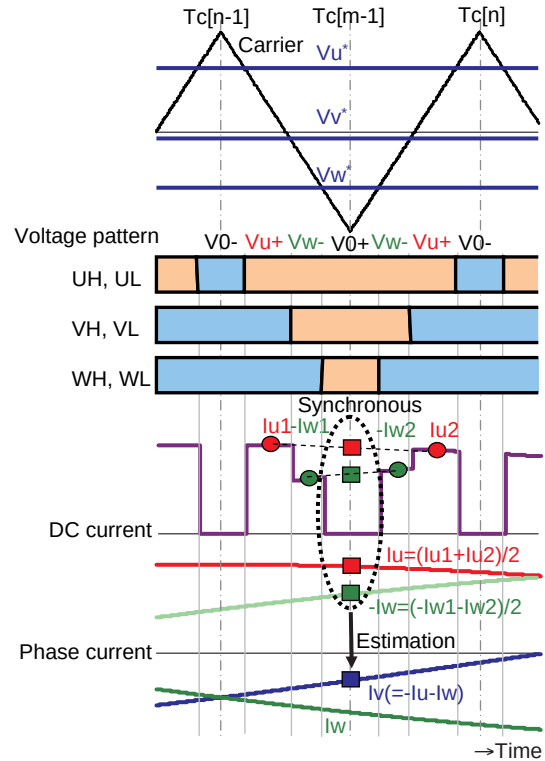


Fig. 7 Measures for reducing phase current estimation error

and the fact that the sum of the three phase currents is zero. All three estimated phase current values are the values at the same timing of the carrier wave trough. In addition, the command voltage update timing is generally synchronized with the carrier wave peak, so the estimated values at the trough timing are suitable as the phase current information used for rotor position estimation.

As shown in Fig. 8, it is challenging to estimate the phase currents when the command voltages are approximately equal ($V_u^* \approx V_v^* \approx V_w^*$). In this case, the only voltage pattern that enables phase current detection during one carrier wave cycle is V_u+ , so only the U phase currents (i_{u1} , i_{u2}) can be obtained, and only the U phase current value (i_u) can be estimated at the carrier wave trough. Phase current estimation values for a minimum of two phases are required to estimate the phase currents for all three phases, which makes it challenging to estimate

the phase currents for all three phases.

To avoid this situation, the minimum necessary voltage is superimposed onto the command voltages so that the command voltage level is different for each phase, thus enabling estimation of all three phase currents. Since rotor position estimation requires the command voltage information, the superimposed voltages should be quantitatively understood. Figure 9 shows the voltage superimposed to avoid areas where estimation is challenging. Gray areas indicate areas where the command voltages for two phases are approximately the same level, making it challenging to estimate the phase currents for three phases. In Fig. 9(a), the command voltage (V^*) is a gray area, making it challenging to estimate the phase currents for three phases. Therefore, as shown in Fig. 9(b), voltages ($V^*h[m]$, $V^*h[m+1]$) with the same level but different vectors to avoid the gray area are

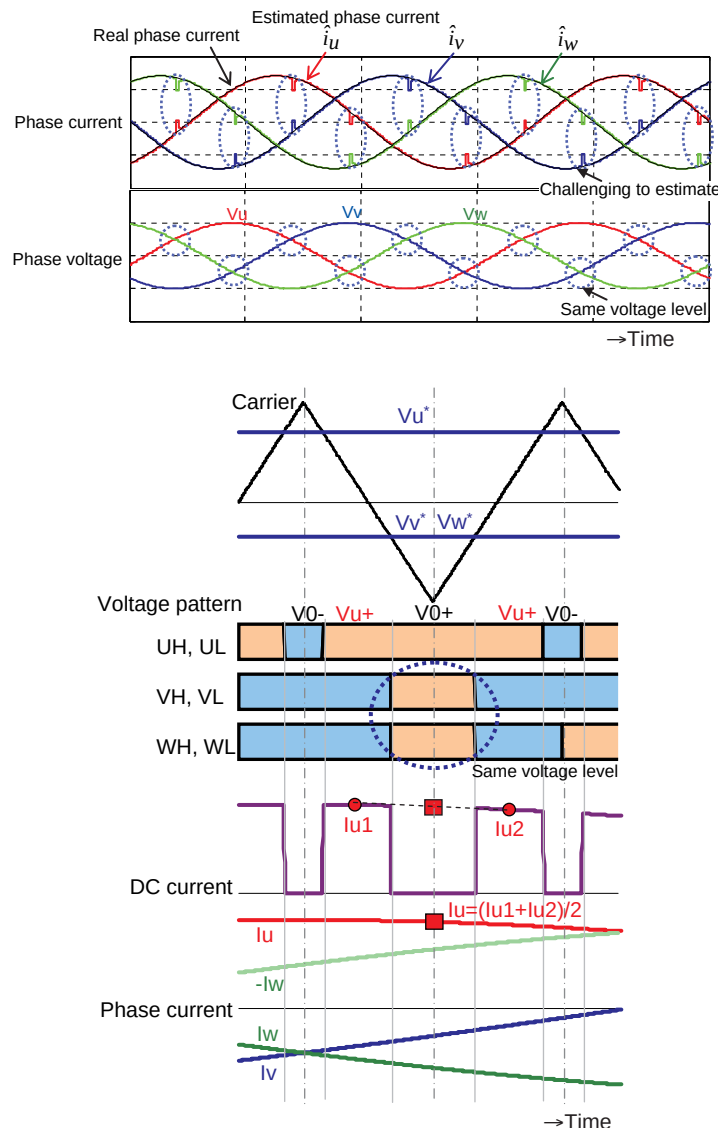


Fig. 8 Challenging cases in phase current estimation

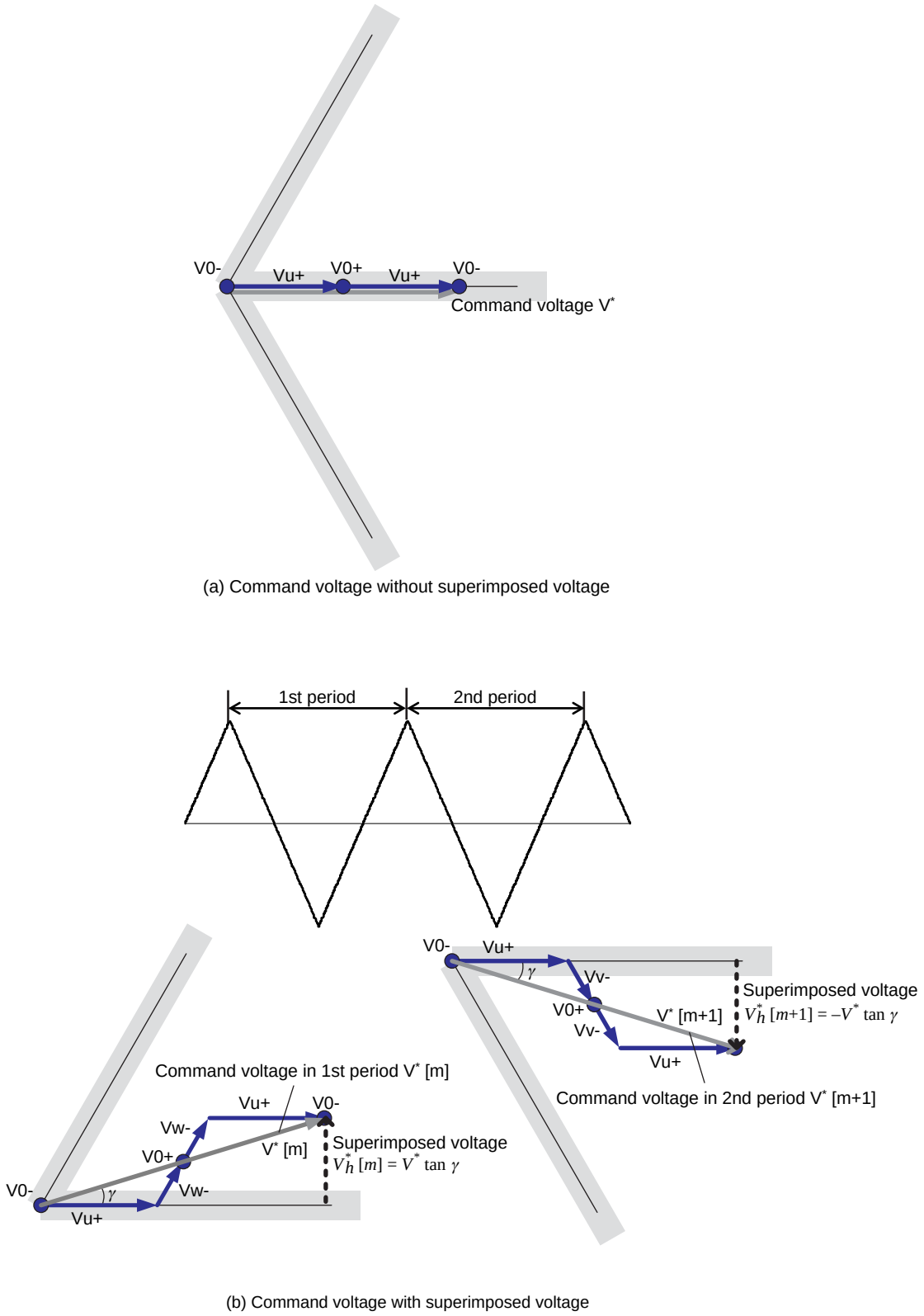


Fig. 9 Relationship between command voltage and superimposed voltage

superimposed onto the command voltage (V^*) in two consecutive carrier wave cycles (first cycle and second cycle). The command voltage with superimposed voltage ($V^*[m]$, $V^*[m+1]$) has a phase difference γ relative to the original command voltage (V^*). Equation (1) shows the superimposed command voltage. The phase currents for all three phases can be estimated by using the values of the command voltage with the superimposed voltages ($V^*[m]$, $V^*[m+1]$) for rotor position estimation.

$$V_h^* = \pm V^* \tan \gamma \quad (1)$$

3.2. Rotor Position Estimation Algorithm

For power use, startup control in which a rotational magnetic field unrelated to the rotor position is generated is used to drive the motor up to a rotational speed at which the electromotive force (EMF) information can be detected. When the rotational speed range is reached at which the EMF information can be detected, the motor is controlled in accordance with the rotor position estimated using the EMF information. Figure 10 shows the proposed rotor position estimation algorithm that uses EMF.

The model installed in parallel with the actual motor does not include EMF correlated with the rotor position, and comprises only resistance and inductance. EMF information is included in the command voltages applied to the real motor, and this information is also input to the model. Rotor position information is therefore included in the current deviation calculated from the model phase current output from the model and the real motor phase current. As a result, the estimated rotor position error is calculated from the current deviation, and the rotor position and rotational speed are estimated using phase-locked loop control (PLL) to minimize the estimated rotor position error.

The automatic current regulator (ACR) uses the estimated phase current and rotor position to calculate the command voltage and perform motor control in accordance with the command torque.

3.3. Coordination between Rotor Position Estimation and Phase Current Estimation

Figure 11 shows the coordination between rotor position estimation and phase current estimation. As previously mentioned, the command voltage and phase current values are required for rotor position estimation. In addition, the command voltages ($V^*[n-1]$, $V^*[n]$) are generated by the combination of voltage vectors during the cycle from one carrier wave peak to the next, so the phase current used for rotor position estimation should also be obtained in accordance with the carrier wave. The phase current estimation method used in this study enables estimation of the motor phase currents ($I_u[m-1]$, $I_u[m]$, $I_w[m-1]$, $I_w[m]$) at the carrier wave trough. The command voltages ($V^*[m]$) between adjacent carrier wave troughs, which are the phase current estimation timing, are the average values of $V^*[n-1]$ and $V^*[n]$. Rotor position estimation is performed using the values of the phase currents ($I_u[m-1]$, $I_w[m-1]$) estimated at the carrier wave trough, the phase currents ($I_u[m]$, $I_w[m]$) estimated at the next trough, and the command voltages ($V^*[m]$) during this period.

The phase current and rotor position can be accurately estimated simultaneously by performing phase current estimation at the timing of the carrier wave trough, which is suitable for rotor position estimation, and at the same time correcting the command voltages used for rotor position estimation in accordance with the phase current estimation timing.

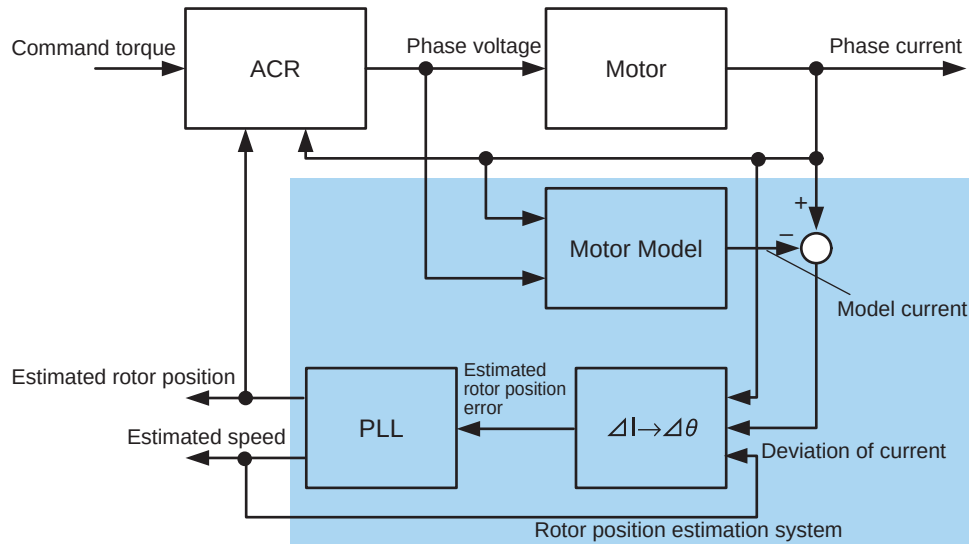


Fig. 10 Block diagram of rotor position estimation system

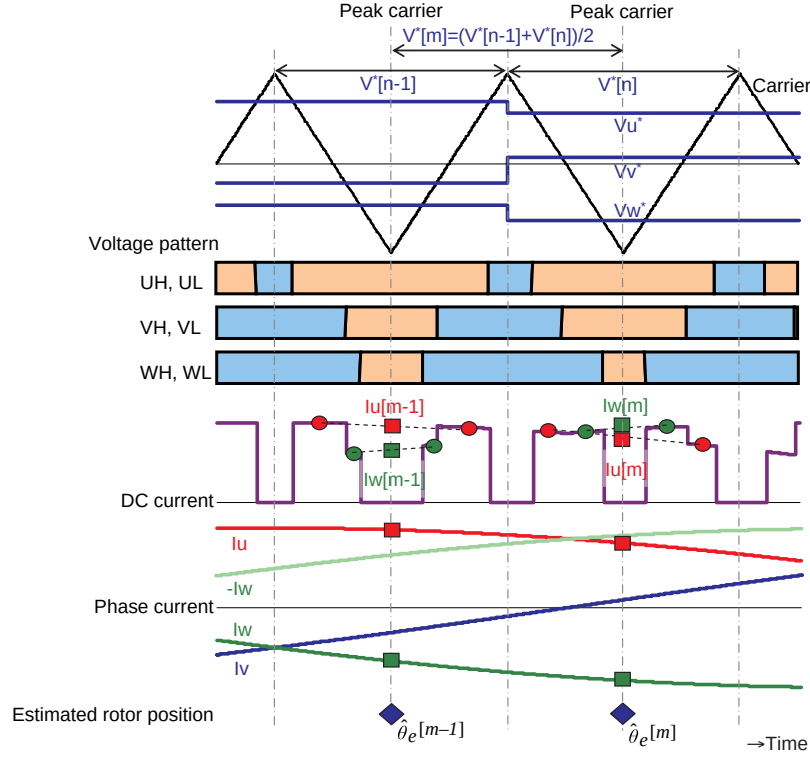


Fig. 11 Coordination between rotor position estimation and phase current estimation

4. Simulation Analysis

4.1. Analysis Methods

The feasibility of the proposed rotor position estimation and phase current estimation was verified by simulation analysis. MATLAB was applied as the analysis simulator. In the simulation analysis, the motor was a 100 W interior permanent magnet synchronous motor (IPMSM), the maximum motor phase current was 10 Arms, the supply voltage was 12 V, and the generally used vector control was used as the control method. In addition, the switching elements were assumed to be in the ideal state (switching time = 0 s).

4.2. Rotor Position Estimation Characteristics and Phase Current Estimation Characteristics

Figures 12 and 13 show the estimated rotor position error and estimated phase current error in each rotational speed range. These errors indicate the maximum value when the motor phase current is analyzed from 0 to the maximum motor current. It was found that estimation for both the phase current and the rotor position can be performed within the range that does not hinder practical application. In addition, the estimation error increases at a rotational speed of around 500 rpm. For power use, startup control in which a rotational magnetic field unrelated to the rotor position is generated is used to drive the motor up to a rotational speed at which the electromotive force (EMF) information can be detected.

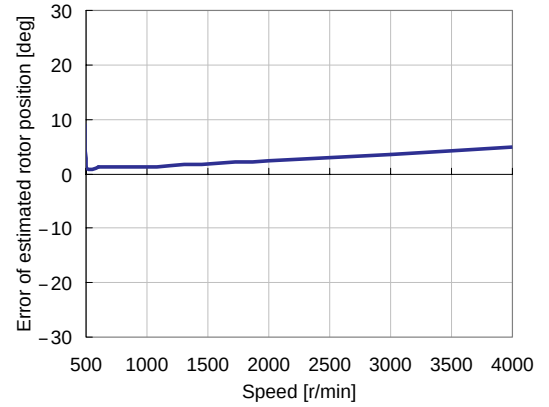


Fig. 12 Simulation results of estimated rotor position

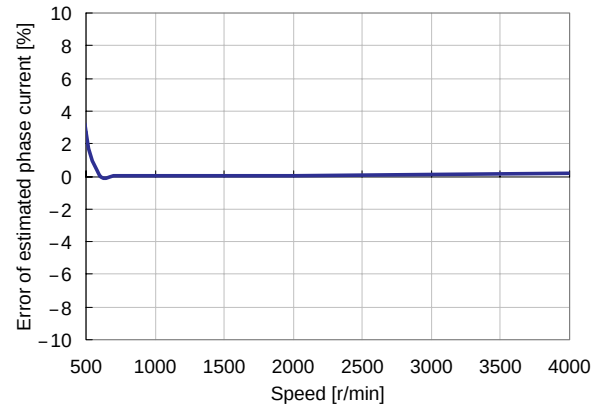


Fig. 13 Simulation results of estimated phase current

Therefore, by setting the startup control range exceeding 500 rpm, this does not hinder practical application for power use.

5. Evaluation Using a Real Motor

The feasibility of the proposed rotor position estimation and phase current estimation was verified by evaluation using a real motor. Like the simulation analysis, the motor was a 100 W IPMSM, the supply voltage was 12 V, and vector control was used as the control method.

Figure 14 shows the estimated and real rotor position at a rotational speed of 1000 rpm. The real rotor position was measured using a generally used resolver sensor. The results show that the rotor position was accurately estimated.

Figure 15 shows the estimated and real phase current at a rotational speed of 1000 rpm. The real phase current was measured using a Hall effect current sensor. The sine wave is slightly deformed due to the effects of the

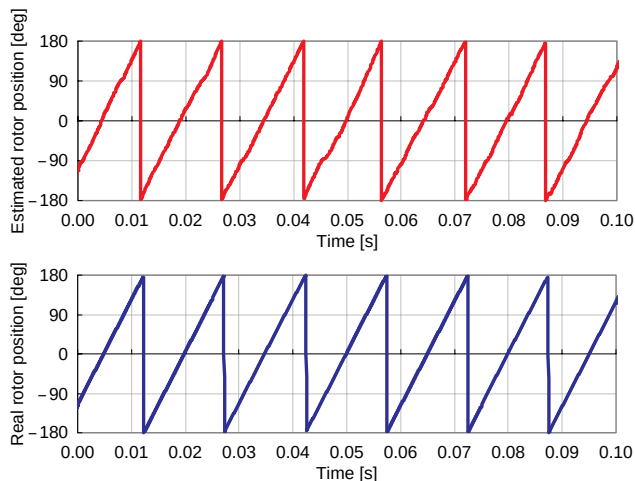


Fig. 14 Results of estimated rotor position

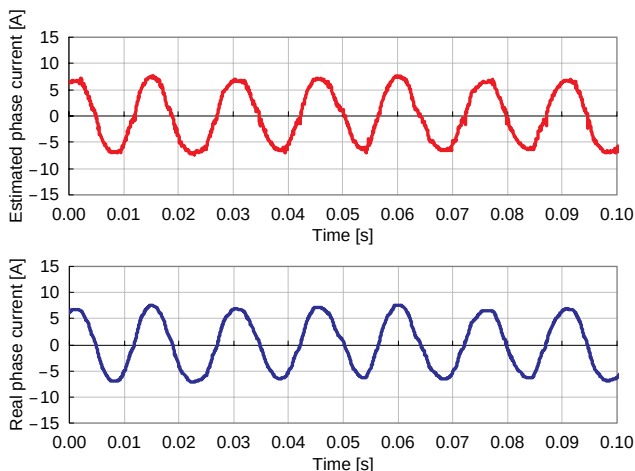


Fig. 15 Results of estimated phase current

superimposed voltage used to avoid areas in which phase current estimation is challenging, but these results show that the phase current was also estimated at a level that does not hinder practical application.

6. Conclusion

This study constructed a sensorless control algorithm with the aim of reducing cost by enabling sensorless control of BLDC motors. This algorithm simultaneously estimates the rotor position and phase current without using the rotor position and phase current sensors generally considered necessary for BLDC motors.

Phase current estimation focused on the symmetry of the FET switching pattern, and a method was presented for estimating the phase current at the timing suitable for rotor position estimation, by detecting the DC current.

For rotor position estimation, a motor model was constructed that does not depend on the rotor position and rotational speed, and a method was presented for estimating the rotor position by comparing the model phase current and the phase current value of a real motor estimated using phase current estimation.

Regarding coordination between rotor position estimation and phase current estimation, a method was presented for estimating the phase current and rotor position simultaneously by estimating the phase current at the timing of the carrier wave trough, and at the same time correcting the command voltage used for rotor position estimation in accordance with the phase current estimation timing.

The feasibility of the proposed algorithm was verified by simulation and evaluation using a real motor, and the results confirmed that the algorithm accurately estimates both the phase current and rotor position.

The proposed algorithm can be applied to generally used vector control, and is effective for on-car power-use BLDC motors. This proposed a sensorless control method for power-use BLDC motors, and established prospects for reducing the cost of brushless DC motors.

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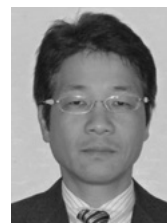
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MAG-Laser Hybrid Welding that Increases Amount of Axial Collapse Energy Absorption in Collisions

Shigeru KOBAYASHI* Tetsuya GOMI*

ABSTRACT

A technology for the application of MAG-laser hybrid welding that enhances the axial collapse column energy absorption during a collision was developed. Taking advantage of the gap weldability that is a characteristic of MAG welding and of the penetrability characteristic of laser welding, this MAG-laser hybrid welding technology resolved issues that conventional laser welding could not readily address, such as lap welding Zn-coated materials together, and welding areas where there are gaps.

The axial collapse column energy absorption during collisions obtained in drop weight tests using hat-shaped specimens on which this MAG laser hybrid welding technology had been applied was approximately 40% greater than that absorption when spot welding had been applied.

1. Introduction

Demands continue to increase for greater fuel economy with a view toward global environmental protection as well as the demands for reduction in chassis weight and for enhanced collision safety and related matters. It is known that the use of laser welding and other such continuous welding instead of conventional spot welding, for members that absorb energy during a collision (such as the front side frames) can enhance energy absorption during collapse. According to Furusako⁽¹⁾ and others, the application of continuous welding employing lasers has been reported to increase energy absorption by about 15% relative to the spot welding. However, the source does not discuss the verification of optimal locations for laser welding nor measures to counter the joint strength decrease, which becomes an issue when welding gaps and when lap welding Zn-coated materials. The application of laser welding will require the resolution of multiple such issues that are unique to laser welding and that have been the main factors impeding its wider adoption.

At present, members that absorb energy during collisions (such as the front side frames) include multiple stamped parts that employ spot welds and other such point welding. To help protect vehicle occupants from the impact of collisions, the function of absorbing impact energy is demanded of these frame parts. With point

welding, however, there are intervals between the joints so that openings form during collapse, and this loss reduces the amount of energy that can be absorbed. Thicker sheets could be used for such materials, or higher strength steel could be used as a countermeasure, but such countermeasures result in increased weight and greater costs. Therefore, MAG-laser hybrid welding that effectively uses the respective strengths of both MAG welding and laser welding was developed together with the technology for its application, which enabled continuous welding that is capable of creating joints in parts of collision members where there are gaps as well as creating joints that experience little loss of joint strength even when lap welding Zn-coated steel. The development also increased the energy absorption during collapse to a higher level than conventional spot welding. This article will give an overview of an application technology for this MAG-laser hybrid welding and report on the results of drop weight tests conducted on hat-shaped specimens using this joining technology.

2. MAG-laser Hybrid Welding Technology

2.1. Issues in Laser Welding

As a countermeasure to the increase in weight that accompanies greater sheet thickness and the increases in material costs and processing costs due to the use of high

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strength steel, the use of laser welding or other such continuous welding instead of conventional spot welding as a method to join hat-shaped members and increase their collapse energy absorption was studied. In order to apply laser welding instead of conventional spot welding as the method for joining current side frame members, however, it was necessary to resolve issues such as the following that are unique to laser welding.

- (1) The side frames are members used in a rust environment, and therefore they are ordinarily made corrosion-resistant through the use of Zn-coated steel. When laser welding the areas of lapped sheets, portions of vaporized Zn coating remain on the joint and form blow holes^{(2), (3)} (below, “vapor defect”).
- (2) Laser welding forms a joint with molten metal produced by melting the parent metal. In gap welding (where there is a space between the sheets), the bondability is diminished. Even a slight gap (0.2 mm or more) can result in an inability to maintain joint strength^{(4), (5)}.

In addition, even when conventional spot welding is replaced by laser welding of the same part, energy absorption by axial collapse only increases by about 15%⁽¹⁾.

2.2. The Mechanism of Vapor Defects

Figure 1 shows the condition of two lapping sheets of Zn-coated steel that were subjected to laser welding and resulted in defects. When laser welding two tightly overlapping sheets of Zn-coated steel, the heat of the laser can cause the Zn coating on the surfaces of the steel sheets to vaporize and form blow holes that remain inside the joint boundary, so that the strength of the joint is reduced. Numerous pits, which are holes formed in the surface of the joint at points where the vaporized Zn coating has been released into the atmosphere, are also visible. Figure 2 shows blow holes identified using X-rays, and Fig. 3 shows pits that have erupted on the surface.

Typical methods used to reduce the occurrence of blow holes include creating a small gap between the two sheets to discharge the vapor from the Zn coating, for example,

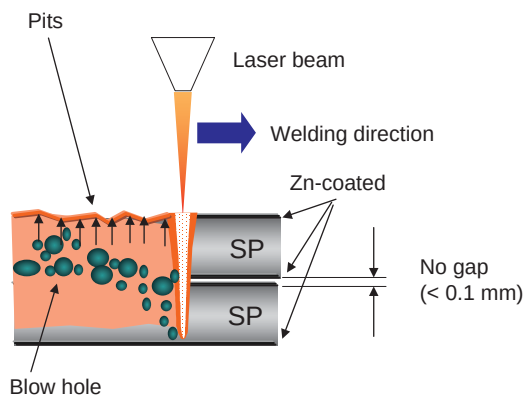


Fig. 1 Schematic defect generation model

by preventing the two sheets from adhering to each other by means of embossing or other such forming applied in advance, and by exposing just the top sheet of the two sheets to a low-intensity laser in advance so that the resulting heat deformation will create minute gaps that allow vapor to be discharged. However, either of these methods involves a front-end process that requires machine processing or other such additional work to be done. Consequently, these cannot be considered simple and certain measures against Zn coating vapor.

2.3. Testing of Zn Coating Vapor Defect Prevention

Attention was therefore focused on joints made at the root of a flange (at the termination of the curve), that can efficiently discharge Zn coating vapor to the outside using gaps and can be expected to provide increased energy absorption during axial collapse. In order to form joints in which defects due to the gap do not occur, MAG-laser hybrid welding technology^{(6), (7)} that combines MAG and a YAG laser was used to make welds on hat-shaped test pieces, and the effects were verified. Figure 4 shows the configuration of equipment used for this testing, and Table 1 shows the welding conditions. The configuration

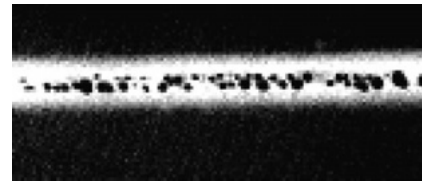


Fig. 2 X-ray inspection of blow hole

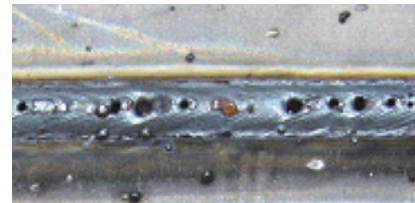


Fig. 3 Surface appearance of pit

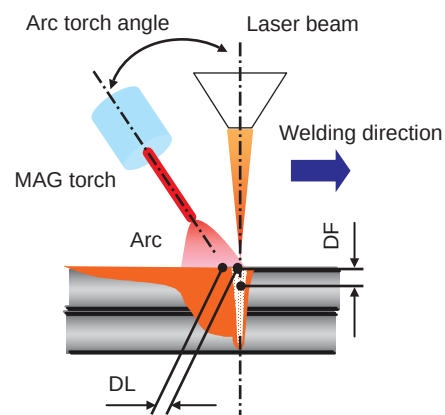
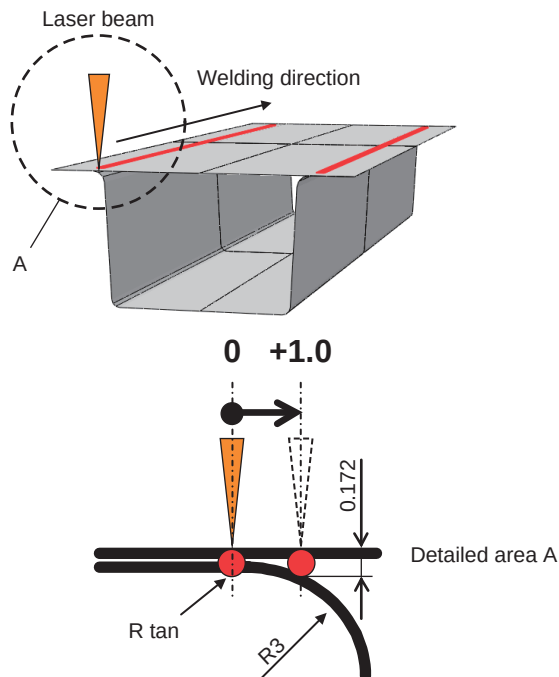


Fig. 4 MAG-Laser hybrid welding configuration

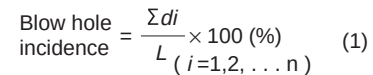
YAG-Laser power	3.5–4.0 kW
Focus radius	Φ0.6 mm
Defocus distance (DF)	–2 mm
Travelling speed	250 cm/min
Arc current (MODE)	250 A (CMT)
Arc torch angle	25°
Shielding gas	Ar+20%CO ₂
Distance of laser and arc (DL)	2.0–3.0 mm

(*CMT: a low-heat arc welding power source developed by Fronius International GmbH)

Figure 5 shows an overall view of the hat-shaped test piece used in this testing and an expanded view of the area in the vicinity of the termination of the curve indicating the laser aim points. Equation (1) shows the method and the formula for calculating the blow hole incidence per unit length, while Fig. 6 shows X-ray images of each weld made with varying aim points centered on the termination of the curve together with the



incidence of blow holes formed there. A blow hole incidence of 30% or more, for which the joint strength would be lower than that with spot welding, was taken as a joint defect. When the laser aim point moved from the termination of the curve (± 0) toward the side in the direction of the flange (-), the number of blow holes increased suddenly, and the incidence rate already exceeded 30% at a distance of 0.5 mm. When the laser aim point moved from the termination of the curve toward the side in the opposite direction from the flange (+), the gap increased in conformity with the position along the curve so that the number of blow holes decreased suddenly, and the incidence rate became zero at a distance of 1 mm. When the aim point moved 1 mm or more, however, the gap opened up too wide. At a distance of 1.5 mm, a hole was made in the upper sheet due to burn through. The laser aim point positional tolerance on the hat-shaped test piece was, therefore, within the range of 0.0 mm to 1.0 mm from the termination of the curve, which was taken as the origin (Fig. 7). The maximum gap



Welding position (mm)	X-ray inspection of blow hole	Blow hole incidence (%)	Evaluation
-1.5		55.6	Defect
-1.0		60.9	Defect
-0.5		39.7	Defect
±0		15.4	Good
+0.5		6.3	Good
+1.0		0	Good
+1.5		0 *Burn through	Defect

Welding area (mm)	Blow hole incidence (%)
-1.5	60
-1.0	45
-0.5	35
0.0	15
0.5	5
1.0	0
1.5	0

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size was 0.172 mm with the laser aim point of 1.0 mm (+). The above results clarified welding conditions for MAG-laser hybrid welding that do not produce vapor defects and the range for appropriate joins. Figure 8 shows an exterior view of the MAG-laser hybrid welding system used in the present testing.

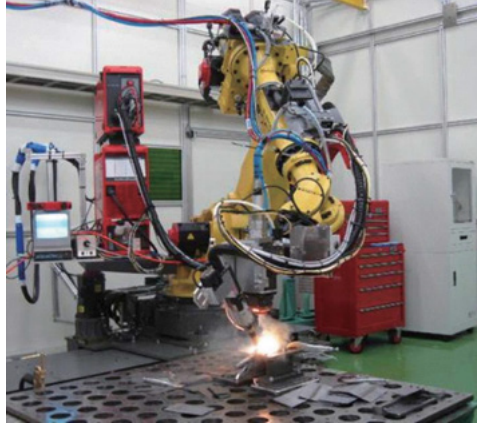


Fig. 8 MAG-Laser hybrid welding equipment

3. Verification of the Effect using Hat-shaped Test Piece

In order to verify the effect from the application of continuous joining by means of MAG-laser hybrid welding, CAE and drop weight testing were used to verify the differences in energy absorption that result from different welding locations and welding methods. The CAE simulation made use of finite element analysis with the general-purpose explicit solver RADIOSS Ver. 4.4.

3.1. Verification by Means of CAE

Figure 9 shows the finite element model for spot welding and for continuous welding as modeled for drop weight tests. The finite element model was 50 mm in height, 70 mm in width, and 300 mm in length with a mesh pitch of 3 mm. The following items (1) to (4) show the welding conditions (welding method and welding location) used for verification:

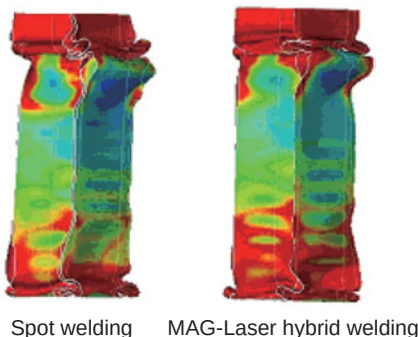


Fig. 9 Axial collapse FEM model

- (1) Spot welding at the flange center (weld interval of 30 mm)
- (2) MAG-laser continuous welding at the flange edge
- (3) MAG-laser continuous welding at the flange center
- (4) MAG-laser continuous welding at the flange root (the termination of the curve)

Using the finite element model described above, the energy absorption when the member collapsed with a stroke of up to 150 mm at a speed of 50 km/h at the time of contact was calculated.

Figure 10 shows results for the comparison of energy absorption with continuous joining at the different welding locations when the energy absorption of spot welding is 100%.

The above results from CAE analysis of hat-shaped test pieces to which the MAG-laser hybrid welding in question had been applied showed that continuous welding applied at the root (the area of termination of the curve) of a hat-shaped vertical wall yielded the greatest energy absorption.

3.2. Verification by Drop Weight Testing

Figure 11 shows the configuration of the hat-shaped

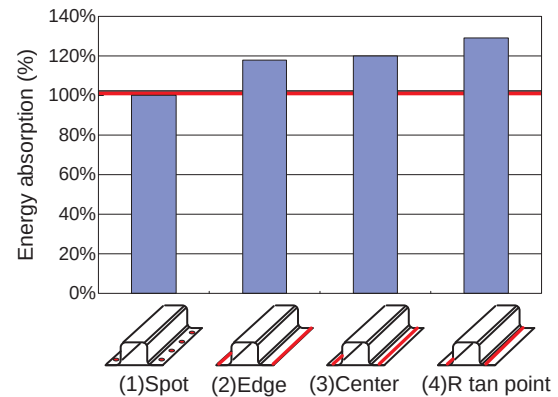


Fig. 10 Comparison of energy absorption between spot welding and MAG-Laser hybrid welding

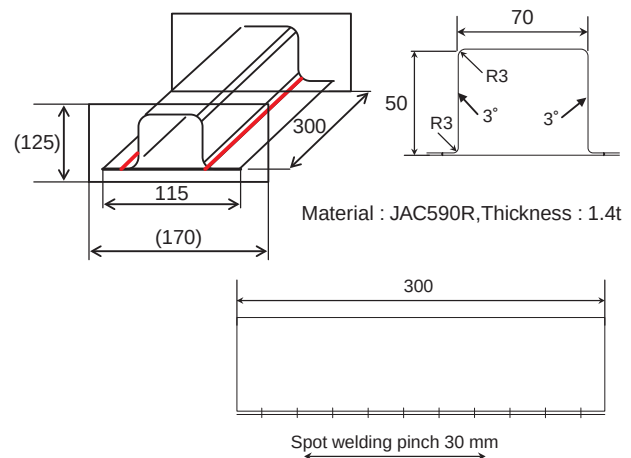


Fig. 11 Configurations of hat-shaped specimen for drop weight tests

Table 2 Drop weight test conditions

Item	Conditions
Drop weight	324 kg
Drop height	9.84 m
Crash speed	50 km/h

test piece used for this testing. The test piece configuration and welding conditions (welding method and welding location) were as in items (1) to (4) describing the finite element model used for verification by CAE. In order to conduct the axial collapse test, end plates with 10 mm thick were attached to each end of the hat-shaped test piece by MIG welding all around the periphery of the plates. The drop weight test conditions including the indenter weight and drop height were specified as shown in Table 2 in order to obtain a stroke of 150 mm or more and a crash speed of 50 km/h against the test piece.

3.3. Drop Weight Test Results

Figure 12 shows the load (kN)-stroke (mm) curves for a stroke of up to 150 mm after the drop weight test of hat-shaped test pieces. One curve is for pieces joined by spot welding, and the other curve is for pieces joined by MAG-laser hybrid welding at the base area of the vertical wall, which has the greatest energy absorption. The peak load immediately after the weight was dropped and the load for each stroke amount were basically greater with the specimen subjected to continuous MAG-laser welding than with the specimen subjected to spot welding. Figure 13 shows the side and rear surface conditions of the hat-shaped specimen after drop weight testing. Inspection of the images after the weight was dropped shows that the flange side surface of the specimen subjected to spot welding had formed openings and had bent in complex ways, while the specimen subjected to continuous MAG-laser welding had not formed openings and had bent evenly and regularly. No ruptures were found in either the spot-welded joint or the MAG-laser hybrid-welded joint, and the condition of both welds was good.

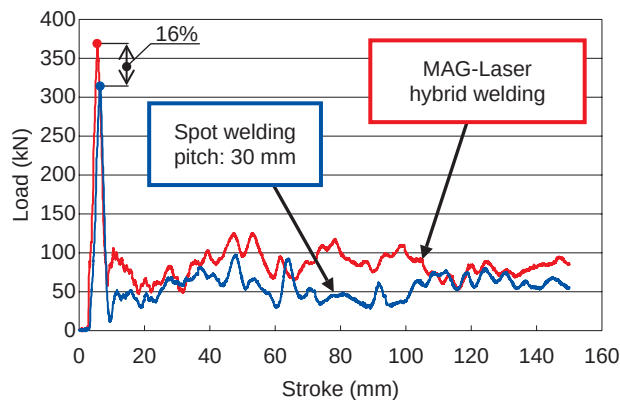


Fig. 12 Comparison of load-stroke curve in axial collapse column between spot welding and MAG-Laser hybrid welding

3.4. Examination of Drop Weight Test Results

The drop weight testing described here showed an increase in energy absorption, and the causes of this increase were examined. The spot welding test piece had openings along the flange portion between the welded spots, and it is very likely that these openings reduced the collapse load.

This is explained by the von Karman effective width theory. Karman's effective width theory is expressed in Eq. (2):

$$P_{cr} = b_e \cdot t \cdot \sigma_r \quad (2)$$

$$b_e = (\pi^2 E k / (12(1-\nu^2)\sigma_r))^{1/2} \quad (3)$$

When $b_e > b$, then:

$$P_{cr} = b \cdot t \cdot \sigma_r \quad (4)$$

P_{cr} is the maximum compressive load; b_e is the effective width; t is the sheet thickness; σ_r is the yield stress; b is the plate width; E is the modulus of longitudinal elasticity; ν is the Poisson ratio; and k is the constant determined by boundary constraint conditions.

When there are angular edges at both ends, $k = 4.00$. When one end has angular edges and the other end is spot welded, $k = 3.15$. When both ends are spot welded, $k = 2.40$. When one end has angular edges and the other end is free, $k = 0.61$. This indicates that the performance of continuous welding results in an increase in the constraint force of the angular edges, which is a boundary constraint condition, so that the collapse load increases⁽⁸⁾. Assuming that the MAG-laser hybrid welded joint is an angular edge and applying that to this hat-shaped test piece, then the maximum collapse load for a MAG-laser hybrid weld can be predicted to rise by 16% relative to a spot weld. The result of the actual drop weight test shows a 16% increase in the maximum collapse load, which correlates closely with Karman's effective width theory. However, comparison with the maximum load from the second wave and thereafter shows an increase of 28.7%, and an

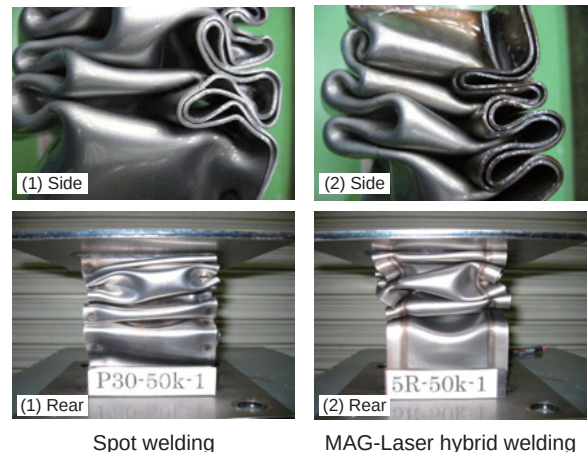


Fig. 13 Appearance of column after collapse test

increase in the average load of 46.6%. This exceeds the efficiency under Karman's effective width theory. This is the range that most of the energy absorption depends on. The formation of openings in this interval in hat-shaped test pieces that have been spot welded accounts for the load difference with MAG-laser hybrid welding, and this difference is thought to have manifested as the difference in overall energy absorption.

Figure 14 shows a comparison of the energy absorption. When the energy absorption with spot welding with welds at the conventional pitch of 30 mm is 100%, then the energy absorption obtained with continuous joining by MAG-laser hybrid welding at the termination of the curve was increased by 38.5% (2.6 times greater than the gain of about 15%⁽¹⁾ obtained through continuous welding by laser only) relative to the spot welding. The use of continuous joining to constrain the angular edge of the flange portion, which contributes the most to energy absorption in the hat-shaped specimen, enables a higher level of energy absorption that could not be obtained using conventional welding methods.

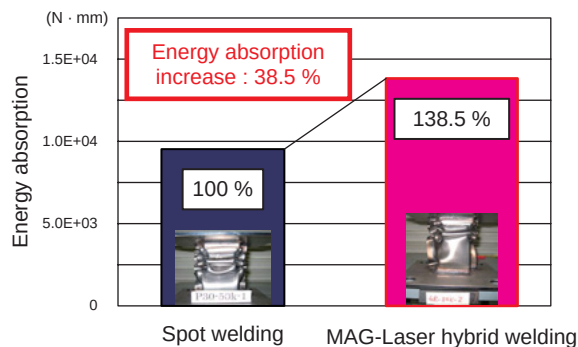


Fig. 14 Comparison of energy absorption between spot welding and MAG-Laser hybrid welding

4. Conclusion

MAG-laser hybrid welding technology and methods for application of that technology were developed with a view towards use on the side frames, which are members that absorb energy during collisions. As a result, the following performance enhancements were achieved:

- (1) MAG-laser hybrid welding technology and applications for that technology were developed to make effective use of gap weldability, which is a characteristic of MAG welding, and of penetrability, which is a characteristic of laser welding. This development enabled continuous welding at the termination of the curve, where energy absorption is highest, and where conventional welding could not be used.
- (2) By enabling continuous welding at the termination of the curve, which laser welding alone had not been capable of joining, the absorption of axial collapse column energy

during collisions was increased a maximum of 38.5% (2.6 times greater than continuous welding by laser only) compared to spot welding.

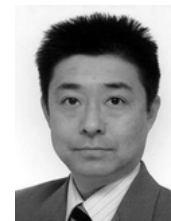
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Editorial Afterword

When I was a kid, I used to chase cars down dirt roads, and daydreamed about traveling all over the world on my motorbike. Around that time, people yearned for affluence and convenience with full of dreams that they wished to make them come true. For the past 50 years, how many technologies have been introduced to the world, and how many products created by Honda R&D Center?

Our predecessors have overcome trials and tribulations to realize their dreams, such as motorbikes, cars and tillers, and even airplanes and robots. This proves that Honda has constantly striven for its original vision to meet the current customers' needs. The "engine" that transformed Honda's dreams into reality was the R&D Center.

There is no doubt that the dominant sense of value in the 20th century were those summarized by advancement and expansion powered by the development of science and technologies. However, it led to unintentional consequences that environmental issues have rapidly escalated. The prerequisite for satisfying our present customers and even for our future customers is "high quality," which has been changing from "expansion" to "sustainability" nowadays. In order to fix the inadvertent consequences of the quest for progress, and to prevent their recurrences, it is essential for the society as a whole to share the development of science and technologies, and to hand on the outcomes to future generations. Although Soichiro-san's biographies and legends are too numerous to mention, do they sufficiently convey his scientific and technological knowledge to be shared and accumulated? I think he must have still had precious knowledge to impart.

On the occasion of the publication of the inaugural issue of "Honda R&D Technical Review", Kawamoto-san wrote about his sincere hope, "this review will be established not as a tool for publicity or a routine forum, but as an instrument for vindicating unmatched ingenuity." It means, we not only provide tangible materials, such as products and technologies, but also introduce, present, and discuss new knowledge that allowed us to create them; thus, we can accomplish our mission, "Vindication of Unmatched Ingenuity," and hand them on to future generations. I believe Honda's engineers will continue to publish their papers based on their outstanding researches that will satisfy even our future customers for the next fifty years.

(Yasuhiro Sugimoto)

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ISSN 1881-6223

Published by:
Honda Motor Co., Ltd.
2-1-1, Minami-Aoyama, Minato-ku,
Tokyo 107-8556, Japan

Edited by:
Honda R&D Co., Ltd.
Honda R&D Technical Paper Committee

Printed:
Mar. 1, 2010

