

Honda R&D

Technical Review



April 2009

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Comment on the Cover

Environmental Synergy

Life is a precious gift and protecting it should be an inextricable part of any human activity.

To protect life we all collectively need to care about preserving nature.

This notion is always a vector around which Honda products are developed.

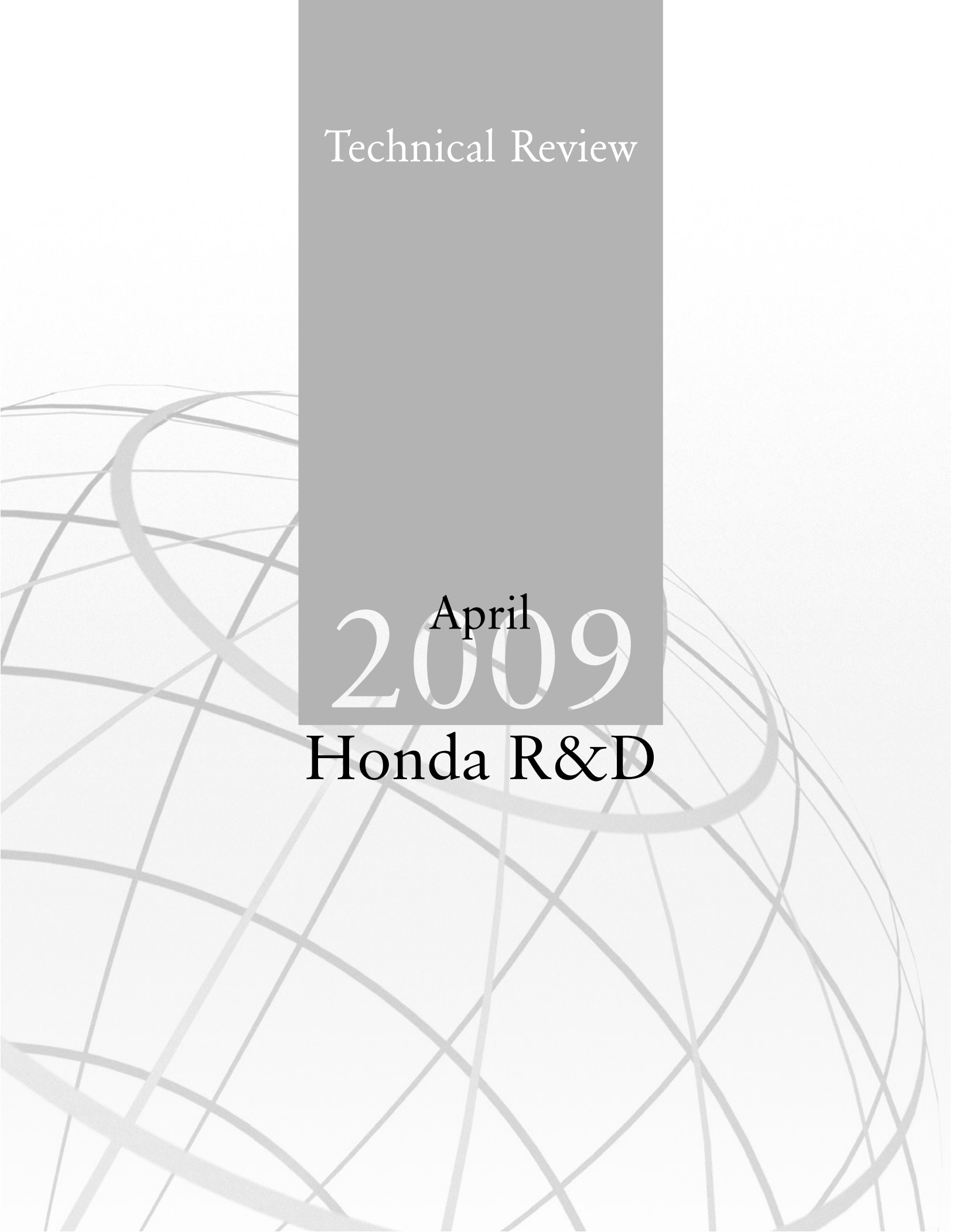
I am proud that I work for a company that always strives for the perfect balance between nature and technology.

The inspiration for the cover came from the 'marriage' between technology and environment:

Many different Honda products, all combined together as a synergy to sustain the environment.



HONDA R&D
EUROPE (DEUTSCHLAND) GMBH
Dimitrios Darkoudis

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

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Style and Design of FCX CLARITY

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ABSTRACT

The FCX CLARITY, a new fuel cell vehicle (FCV), has a package layout and design that make full use of one of the FCV's great advantages, that of a freedom of package layout with distributed powerplant components. On top of being a zero emissions eco-friendlier vehicle, the FCX CLARITY was developed with the goal of creating a next-generation sedan offering new value and appeal, joy of driving, and pride of ownership. The eye-catching design was meant to give a glimpse of the future in a way that makes it instantly recognizable as different from that of an internal combustion engine vehicle. This includes the exterior design that achieves new proportions with short nose and long cabin with its configuration of dynamic surfaces. The resulting new form is completely unlike that of any conventional sedan. The interior was designed ahead of its time, optimizing its wide open cabin space to create spacious and comfortable personal space, and a cockpit which gives a new sense of fun with futuristic features.

1. Introduction

In 2002, Honda introduced the 2003 model year FCX, the first fuel cell vehicle (below, FCV) certified by the US EPA, and thereby demonstrated the strong basic performance of this vehicle. Since then, Honda has performed research and development with the goal of further enhancing the performance of this new technology and thus promoting it⁽¹⁾.

The FCV has an FC stack which generates electricity, as well as an electric drive motor, instead of the traditional internal combustion engine. Because these components can be distributed, it gives the automobile the potential for a radically changed package layout and design. However, the FCX through the 2005 model had a number of limitations that were due to, a large powerplant as well

as a chassis borrowed from the Honda EV Plus⁽¹⁾. Because of these limitations, nothing about the exterior identified the FCX as an automobile created with leading edge technology.

For the new FCV, the FCX CLARITY, Honda developed a new powerplant that includes a newly developed, smaller and higher-output FC stack, and a platform specifically for the FCV which enabled a new package layout and design taking full advantage of the freedom afforded to this package. The goal of this development was to convey to a larger audience the potential of the new fuel cell technology.

This paper reports on the development of the design of the FCX CLARITY (below, CLARITY) (Fig. 1).

2. Design Goals

The design concept was to design an automobile that would be instantly recognizable as offering new value and attractiveness impossible to achieve with an internal combustion engine vehicle. Honda's goal was to create a futuristic, eye-catching design that would steal the heart of the viewer at first glance, provide a glimpse into the future the very instant the door was opened, and inspire a new generation of drivers with dreams about the automobile society and the global environment (Fig. 2).



Fig. 1 2008 model year FCX CLARITY

* Automobile R&D Center

Image not displayed for copyright reasons.

Fig. 2 Design concept

Additionally, the designers felt that the unique features of this FCV, namely its strong environmental performance (with zero emissions), interior spaciousness enabled by the new package, and the high-torque yet quiet ride afforded by its electric motor, made this package layout appropriate to a premium car of the future, and so Honda has considered the CLARITY to be a “premium sedan for the next generation.” Other design goals were to achieve a level of presence, class, and comfort appropriate to such a vehicle.

3. Package Layout

In order to achieve a package layout optimizing FCV's biggest advantage — the potential for a distributed power plant — all the powerplant components were made more

efficient and compact, and a platform was developed exclusively for the FCV that promotes their efficient layout (Fig. 3).

Up to now, large FC stacks have been located, for example, under the floor of a high-standing vehicle. The new FC stack was given an innovative new structure that made it efficient and compact^{(2), (3)} enough to be placed in the center tunnel. As a result, the sedan package has a low floor, low center of gravity and low overall height that would have been impossible in a conventional FCV (Fig. 4).

Additionally, thanks to a more compact front-wheel-drive motor⁽²⁾, it was possible to design a short nose and achieve a long cabin that increases interior space for all occupants while limiting overall length. The result is the generous cabin space of a large-class body in a medium-class body.

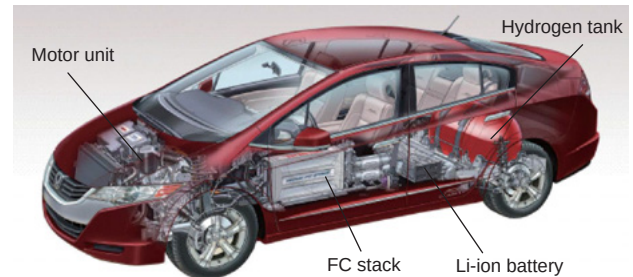


Fig. 3 Main powerplant components

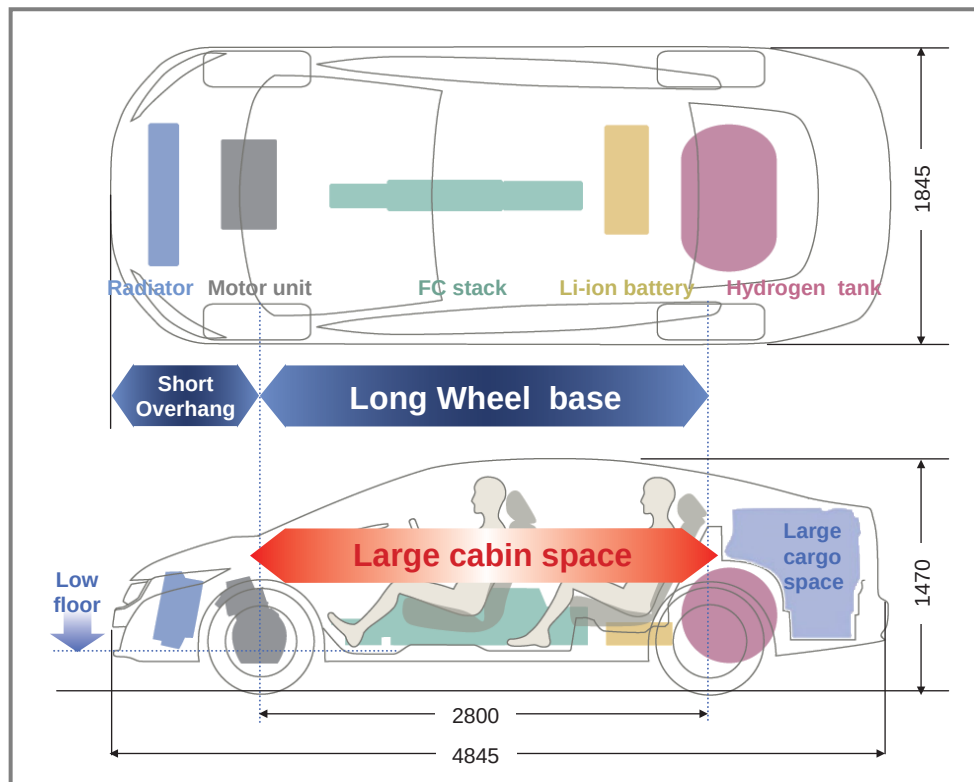


Fig. 4 Package layout

Furthermore, by reducing the size of the hydrogen tank and the auxiliary power source⁽⁴⁾, by making their layout more efficient, and by providing a high-deck trunk, the design enabled enough trunk space to fit four golf bags or one folded-up wheelchair.

This FCV package layout is achieved because there is no longer an internal combustion engine. This provides not only more cabin space but also a high degree of design flexibility. It is also possible as a result to achieve an “eye-catching design” with new proportions (namely, a short nose and long cabin) and a newly designed interior space.

4. Exterior Design

The concept of the exterior design was the “Dynamic Full Cabin Sedan,” characterized by new, long cabin proportions and the bold, tapering cabin afforded by the innovative platform. The concept also features wider fenders and a lower center of gravity, which together invoke a dynamic cruising form (Fig. 5). The package aims to create a vehicle instantly recognizable as the CLARITY from whatever angle it is viewed (Fig. 6).

The front expresses a strong graphic (Fig. 7) from the bumper to the headlights, stressing strong presence and width, while the front pillars, which jut out far to the front, evoke the dynamic, progressive image of the CLARITY.

From the side, the expansive shoulder section, made possible by greatly tapering the mono-form cabin in front and back, symbolizes the hood and trunk of a new type of sedan. The door mirror takes up the plated molding coming from the side glass, making the mirror appear to be of a single piece with the body. The exterior door handle is designed to accentuate the workmanship of the individual components. The handle’s sharpness and length are used to draw out the elegance and length of the cabin (Fig. 8).

The back of the vehicle is characterized by rear fenders that blend strongly into the tapering of the cabin and by the contoured rear panel. These features give CLARITY a strong individuality unseen in sedans up to now. The



Fig. 6 Final renderings



Fig. 7 Front view



Fig. 8 Cabin, door mirror, and exterior handle



Fig. 5 Dynamic full cabin sedan image



Fig. 9 Rear view

rear combination lamps, integrated into the extra window that traverses the area, accentuate a thoroughly wide-looking stance (Fig. 9).

5. Interior Design

The concept of the interior design is a “Futuristic Cocoon,” which evokes a sense of the future the moment the door is opened, and a comfortable area that uses the ample cabin space to envelop each passenger (Fig. 10).

As Figs. 11 and 12 show, the area around the cockpit gives a constant sense of driving a car of the future. The innovative three-dimensional FC meter and newly developed compact electric shifter, among other

unprecedented points of interest, are built around the meter visor. The new H₂ ball-meter located in the center of the FC meter is both functional and entertaining as it encourages the driver to monitor hydrogen consumption with sensory clues - the color and size of the ball.

In order to highlight the refreshing and open nature of the forward space made possible by eliminating the internal combustion engine in front, the CLARITY has a “floating layered instrument panel” whose components appear to float in layers, suggesting open space where air might pass between the panels (Fig. 13).

In addition, the four seats are each independent spaces which enable four passengers to sit in comfort. The door linings have a dynamic, round form that looks as if they were boldly scooped out with a spoon in the center to enfold each passenger. This provides a wider and more enveloping feel creating a personal space in which each passenger can be comfortable (Fig. 14).

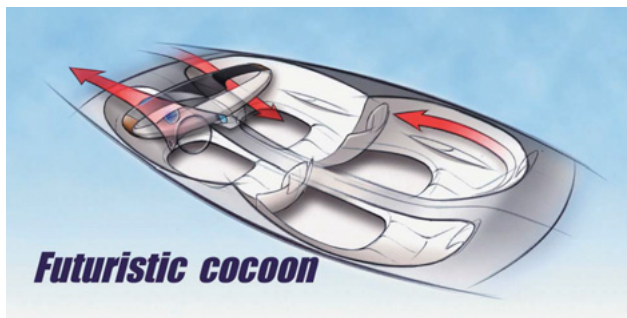


Fig. 10 Futuristic cocoon

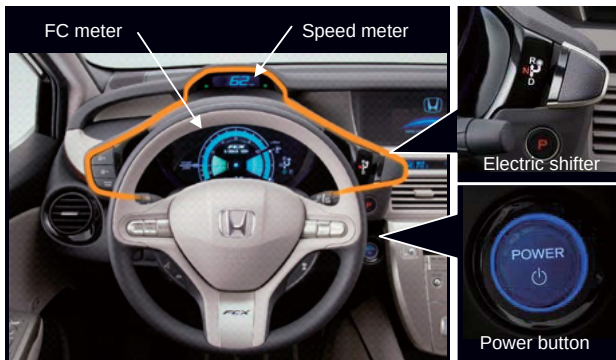


Fig. 11 Futuristic design around meter-visor



Fig. 13 Floating layered instrument panel



Fig. 14 Dynamic round form

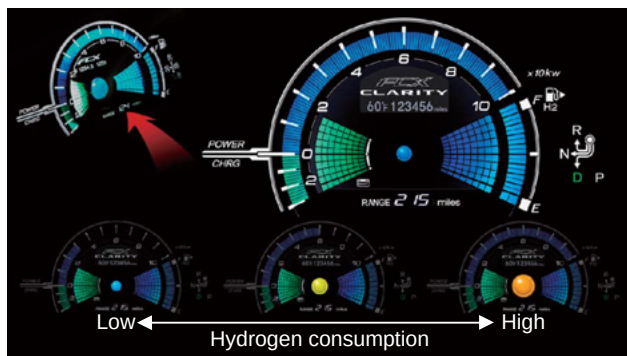


Fig. 12 FC performance meter

6. Color Design

6.1. Exterior

The exterior color was chosen to be both elegant and high-class for a next-generation premium sedan, but also to be brilliant and evoke a progressive feel. The objective was to project all the composure and dignity of a premium sedan while also expressing shadows on the contoured

surfaces such as the fenders and C-pillars. The color chosen with an image of the star garnet gemstone has a strong contrast of deep black and bright red highlights. A newly developed high-contrast paint was used to express a progressive feel (Fig. 15). Furthermore, with the intention of giving the CLARITY an iconic presence, star garnet metallic was selected as the only color.



Fig. 15 Star garnet image

6.2. Interior

The theme of the interior is “Future Craft,” with a futuristic look as well as a warmth that envelops the passenger in comfort. The colors and materials were coordinated to project these ideals (Fig. 16).

The seats, armrests, console trays, and other components which come in direct contact with the skin are made with Honda Biofabric⁽⁵⁾, a new plant-based material that is eco-friendlier and has a high-quality feel to the touch. A pale-pink color scheme gives an air of warmth. The instrument panel, door lining, and the like are made with a combination of traditional materials like wood-grain panels, and new materials such as silver accents and clear acrylic. By coordinating new materials different from anything found in conventional high-end cars, the CLARITY projects a next-generation premium image.



Fig. 16 Interior material coordination

7. Emblem Design

The emblem design also features the CLARITY’s clean and premium nature.

The letters in “FCX” use a design that symbolizes the environmental performance and progressiveness of an FCV with clear material that evokes a clean image from which the sharp, plated lettering floats (Fig. 17).

In addition, the logo of the new name “CLARITY” stretches out longer than the lettering of “FCX.” With a separate-type lettering and satin-plated finish, this component too projects a premium feel and is designed to establish name recognition.



Fig. 17 Emblem design

8. Conclusion

The FCX CLARITY was designed specially as an FCV taking full advantage of the characteristics of a new technology - the fuel cell - to replace internal combustion engines. Thus, it is the first FCV to create new value and attractiveness in addition to the strong environmental performance of zero emissions.

Going far beyond the idea of the eco-friendlier car, the CLARITY is the first FCV to achieve not only the universal attractions of a car, which include the pleasure of driving and joy of ownership. It was also built as a “next-generation premium sedan” that surpasses the existing idea of a high-end automobile.

For the design, a package was used that made full use of the potential of the exclusive platform, providing new sedan proportions (a short nose and the mono-form of the long cabin, both completely different from conventional sedans) and a cabin space that combines an exciting futuristic feel with enveloping comfort. Thus, the CLARITY provides new direction for future car package layout and designs.

It is Honda’s hope that the FCX CLARITY will showcase the possibilities of the new technology of fuel cells to more and more people and inspire a new generation with dreams about the future of the global environment and the automobile society.

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Fuel Cell Powertrain for FCX Clarity

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ABSTRACT

Honda has developed a new fuel cell vehicle in order to respond to global warming and energy issues. The new vehicle, the FCX Clarity, displays enhancement in driving performance and fuel efficiency against previous FCX, and embodies a new appeal not available in a reciprocating engine vehicle. The key to the Clarity's development was the achievement of size and weight reductions and increased efficiency in the fuel cell powertrain.

The FCX Clarity's fuel cell powertrain has Honda's proprietary V Flow FC stack at its core, and displays a weight power density 2 times higher and a volume power density 2.2 times higher than the powertrain of the previous FCX. In addition to increasing maximum power to 100 kW, the V Flow FC stack achieves a 50% increase in volume power density and 67% increase in weight power density against the previous FC stack. Startability at low temperatures has also increased, with the vehicle able to start at -30°C.

The maximum power of the Clarity's drive motor has been increased to 100 kW. In combination with the provision of power assist using a lithium-ion battery, this has enabled the realization of a smooth, powerful and continuously extendable acceleration feel. The increased efficiency of the powertrain and superior energy management have resulted in the achievement of a 60% operating energy efficiency, and the vehicle displays a fuel economy 2.1 times higher than that of a compact gasoline vehicle, and 1.4 times higher than that of a compact hybrid vehicle.

1. Introduction

Honda first introduced the environmentally-friendlier FCX fuel cell vehicle to the Japanese and U.S. markets in 2002⁽¹⁾. In 2004, the company developed an FCX vehicle fitted with a fuel cell (FC) stack able to be started at -20 °C, enabling the vehicle to be used in cold climates^{(2), (3)}.

The enhancements in driving performance and environmental performance, which have been sought in the development of the FCX, mean that the vehicle is now able to compete with a reciprocating engine vehicle in terms of performance.

The FCX fuel cell powertrain was subject to a thoroughgoing reexamination in order to achieve a further evolution of the vehicle's performance and to increase its commercial appeal. This reexamination indicated that the reduction of the size of the powertrain would be the key to enabling maximal exploitation of the high degree of freedom in layout, which is one of the major characteristics of the fuel cell system.

The result of the development of an all-new powertrain for the FCX has been the debut of the FCX Clarity, the

new model fuel cell vehicle that is introduced in this paper. The Clarity's unique design gives it an interior and controls with a futuristic feel, and it features a roomy cabin for a compact vehicle. Boasting a driving range equaling that of a reciprocating engine vehicle, the Clarity offers a drive feel unavailable from a reciprocating engine.

Driving distance and refuel time are issues that fuel cell vehicles share with vehicles using electric drives, and this paper will also compare the Clarity's performance in these areas with that of an electric vehicle, based on the latest battery technology.

2. Development Goals

The following development goals were established in order to manifest the new commercial appeal and new value that is able to be realized by the FCX Clarity, because it employs a fuel cell powertrain:

- (1) Realization of an appealing vehicle design
 - An all-new exterior and interior design making use of the degree of freedom in layout afforded by the use of a fuel cell powertrain
 - A cockpit offering a new sensory experience – like

* Automobile R&D Center

driving a vehicle of the future

- (2) Realization of the full cabin space available only in a fuel cell vehicle
 - Short and slanted nose
 - Spacious cabin
 - Low floor, low overall cabin height
 - Functional trunk space
- (3) Enhanced basic vehicle performance
 - Smooth, powerful, continuously extendable acceleration feel
 - Vehicle range equal to that of a reciprocating engine vehicle
 - Enhanced low-temperature start-up performance

3. Vehicle Design and Interior Equipments

3.1. Vehicle Design

Figure 1 shows the exterior of the FCX Clarity. Fuel cell vehicles up to the present have mainly employed an SUV platform, with the electric generating system positioned under the floor. The design of the FCX Clarity sought to take advantage of the freedom in layout that is a major feature of fuel cell vehicle packages, and to realize the type of layout that is only possible in fuel cell vehicle. The result is a revolutionary low-floor sedan platform incorporating unique interior and exterior designs.

A high deck design has been employed for the rear of the vehicle. The world-first extra window for vehicles has a visibility control function enabling the achievement of good rear visibility for the driver to be reconciled with the restriction of visibility from outside the vehicle into the trunk space. This has enabled the realization of both good rear visibility and a roomy trunk. The color of the vehicle exterior is a special tint based on the semi-precious gemstone star garnet.

Table 1 shows the major specifications of the FCX Clarity.

3.2. Main Vehicle Equipments

3.2.1. FCX multiplex meters

Newly designed multiplex meters have been employed in the Clarity to give the driver the feeling of being behind



Fig. 1 FCX Clarity

Table 1 Major specifications

Model	FCX Clarity	2005 model year FCX
Length/Width/Height	4845 x 1845 x 1470 mm	4165 x 1760 x 1645 mm
Wheel base	2800 mm	2530 mm
Vehicle weight	1625 kg	1670 kg
Number of passengers	4	4
Maximum speed	160 km/h	150 km/h
Range (EPA combined label)	280 mile	210 mile
Maximum motor power	100 kW [136 PS]	80 kW [109 PS]
Maximum motor torque	256 Nm [26.1 kgm]	272 Nm [27.7 kgm]
Fuel cell stack	PEMFC	PEMFC
Power assist	Li-ion battery	Ultra-capacitor
Hydrogen supply system	High-pressure hydrogen tank	High-pressure hydrogen tank
Hydrogen pressure	35 MPa	34.4 MPa
Hydrogen storage volume	171 L	156.6 L

the wheel of a car of the future. In the center of the meters is a hydrogen ball meter which displays hydrogen consumption. To the left and right are a hydrogen fuel meter and an Li-ion battery meter, which displays the level of battery assist and the status of power regeneration. At the top of the meter visor, in a position that necessitates the driver make minimal adjustments to their line of vision, is the speedometer (Fig. 2).

As Fig. 3 shows, the hydrogen fuel meter and Li-ion battery meter feature fan-shaped displays, and have ten levels corresponding to the amount of fuel or battery charge. The final level is further subdivided into another ten levels, increasing ease of reading and accuracy. The hydrogen ball meter in the center of the meters changes size and color in response to operating conditions in order



Fig. 2 FCX multiplex meters(multi-layered instrument panel)

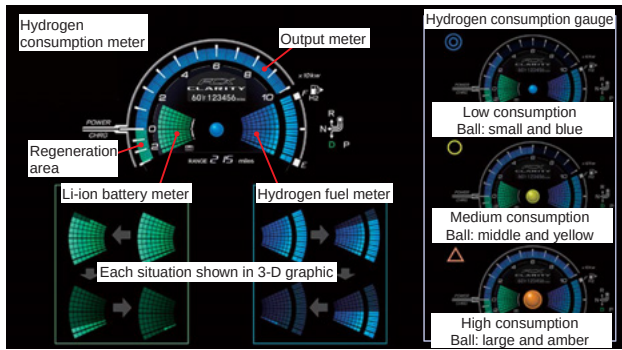


Fig. 3 Hydrogen consumption meter



Fig. 4 Shift-by-wire

to enable the driver to easily identify the status of hydrogen consumption. While enabling the driver to quickly identify vehicle operating status, the multiplex meters also offer the enjoyment of the sensation of driving a car of the future.

3.2.2. Shift-by-wire

The use of electrical control has enabled the realization of a compact shift unit. The shift lever is positioned at the side of the FCX multiplex meters. Only a light operating load and short operating stroke are needed to operate the shift (Fig. 4).

4. Powertrain Technologies

4.1. Powertrain System

As Fig. 5 shows, the Clarity's powertrain is made up of an electric drive system, fuel cell system, Li-ion battery system, and high-pressure hydrogen supply system. The components necessary to generate electricity have been dispersed throughout the vehicle. Each component has been reduced in size, and the layout takes maximum advantage of this fact. Beginning with the fuel cell stack, each component required for electricity generation has been reduced in size. The result has been a reduction in the size and weight of the powertrain, achievement of a weight output density two times greater and volume output

Table 2 Comparison of drive motor performance

	FCX Clarity	2005 model year FCX
Max. output	100 kW	80 kW
Max. torque	256 Nm	272 Nm
Max. rpm	12500 min ⁻¹	11000 min ⁻¹
Max. efficiency	97 %	97%

density 2.2 times greater than the powertrain of the previous FCX (Fig. 6), and realization of a spacious full cabin that has previously been unavailable in a conventional reciprocating engine vehicle.

The high power produced by the fuel cell stacks and the power assist provided by the Li-ion battery enable the new powertrain to output a powerful drive force via a high-power motor. Motor torque is continuous, with no gear shift, and the smooth, powerful and continuously extendable acceleration across the entire driving range, from takeoff to high speeds, realizes a drive feel in a class of its own.

4.1.1. Electric drive system

An electric drive system in a new configuration, comprising a drive motor, reduction box, and Power Drive Unit (PDU), was developed in order to realize powerful acceleration, increased maximum speed and greater compactness for vehicle packaging (Table 2).

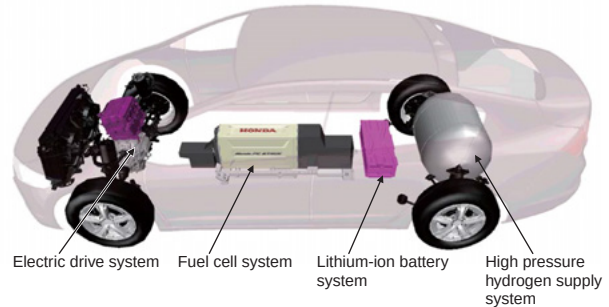


Fig. 5 FCX Clarity powertrain layout

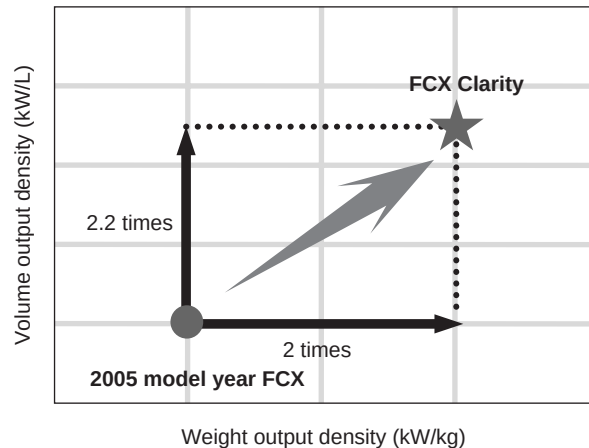


Fig. 6 Evolution of Honda powertrain

A unique shape and positioning has been employed for the magnets in the drive motor rotor. The application of low core loss magnetic steel plates in the stator has reduced loss at high speeds, and optimization of the shape of the slots has increased their lamination factor, reducing resistance in the magnet wire. These innovations have enabled the realization of high power, high torque, high speed, and high efficiency. The maximum power of the motor has been increased from 80 to 100 kW, and this level of output can be maintained up to high speeds, offering comfortable passing acceleration⁽⁴⁾ (Fig. 7).

In addition, a coaxial configuration has been used for the drive motor and reduction box and the motor has been integrated with the PDU, enabling the motor to be reduced

by 162 mm in length and 240 mm in height (Fig. 8). These changes have enabled the realization of the vehicle's bold and stylish short- and slanted-nose design and spacious cabin.

4.1.2. Fuel cell system

A fuel cell system with a new configuration was

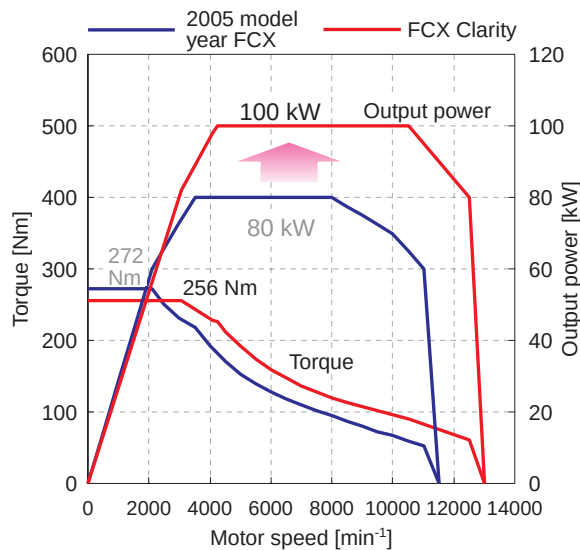


Fig. 7 Output and torque characteristics

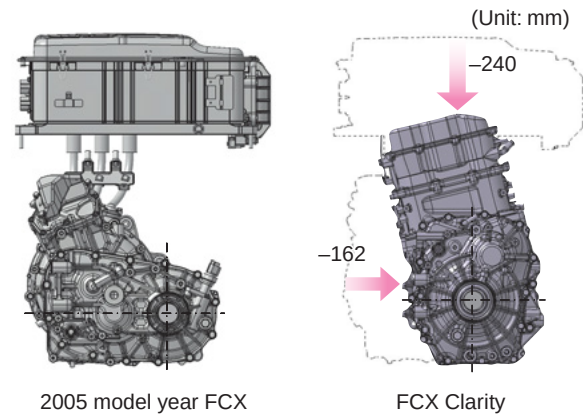


Fig. 8 Evolution of drive motor and power drive unit

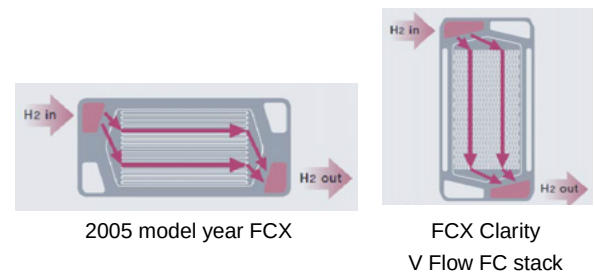


Fig. 9 Comparison of hydrogen flow

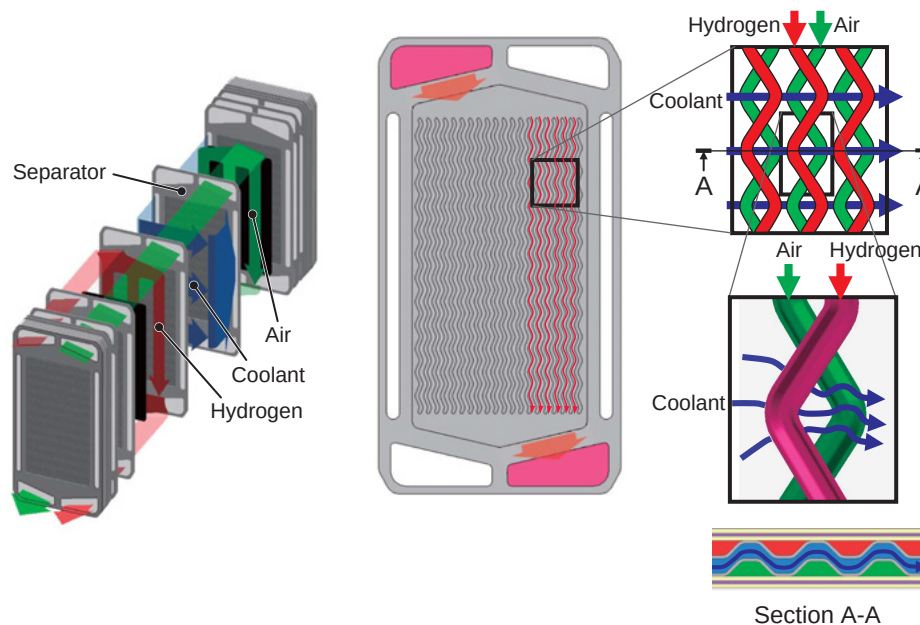


Fig. 10 Detail of wave flow channel separator

developed to enable the realization of the type of full cabin only available in a fuel cell vehicle.

The previous Honda FC stack employed a configuration in which hydrogen and oxygen flowed into the cells horizontally. In the new configuration used in the V flow FC stack, hydrogen and oxygen flow vertically (Fig. 9). This enables the system to smoothly drain the water that is produced on the generating surfaces using the force of gravity. This enhancement in drainage capacity has enabled the height of the flow channels to be reduced by 17%, contributing to weight savings and greater compactness.

The V Flow FC stack employs wave channel flow separators to help ensure a uniform supply of hydrogen, oxygen, and coolant to the interior of the generating surfaces. The coolant flows horizontally, across the vertical flow of the hydrogen and oxygen through the flow channels (Fig. 10).

The uniform cooling of the generating surfaces achieved by this method has enabled the number of cooling layers required for each cell to be reduced by half. The evolution of the internal structure of the system has enabled the previous two-box configuration to be replaced by one box. As a result, the size and weight of the stack have been reduced against the previous stack, enabling the achievement of a maximum power of 100 kW. The volume output density of the new FC stack is 50% higher and its weight output density 67% higher than the previous stack, in addition to which it is lighter and more compact⁽⁵⁾⁻⁽⁷⁾ (Fig. 11).

Low-temperature startability has also been increased. The abovementioned reduction of the size and weight of the unit has reduced heat capacity by approximately 40%. This has resulted in enhanced warm-up performance. The time required for the Clarity to reach 50% output when started at -20°C is one-quarter that required by the previous FCX, and the vehicle is also able to start at -30°C (Fig. 12).

The conventional two-box Honda FC stack configuration has been concentrated into a one-box

configuration in the Clarity, enabling the components that connected the stacks to be simplified or eliminated. In addition, the use of a one-box configuration has enabled the hydrogen supply system, humidification system, and contactors to be integrated longitudinally, resulting in a 65% reduction in the area of the fuel cell system box against that of the previous model. This has enabled the fuel cell system to be positioned in the center tunnel, contributing to the realization of a spacious cabin, low floor, and low overall cabin height (Fig. 13).

4.1.3. Li-ion battery system

A Li-ion battery system has been used as the assist power source in order to enhance assist performance, increase the system's rate of energy recovery, and contribute to the achievement of weight savings and increased compactness⁽⁸⁾.

The use of a hydraulic-regenerative cooperative brake system that controls the allocation of regenerative and

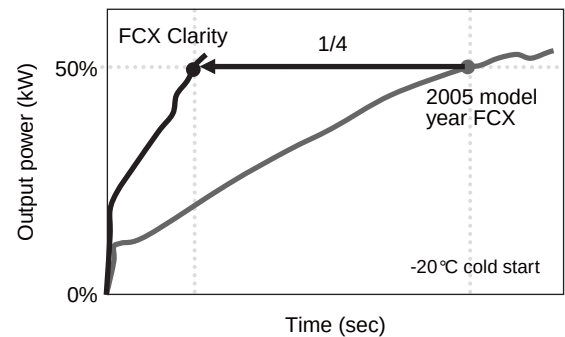


Fig. 12 Cold start performance

2005 model year FCX

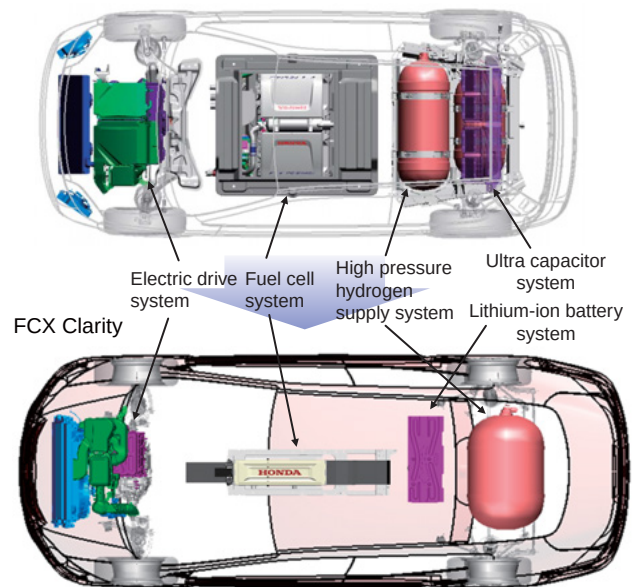


Fig. 13 Comparison of Honda powertrains

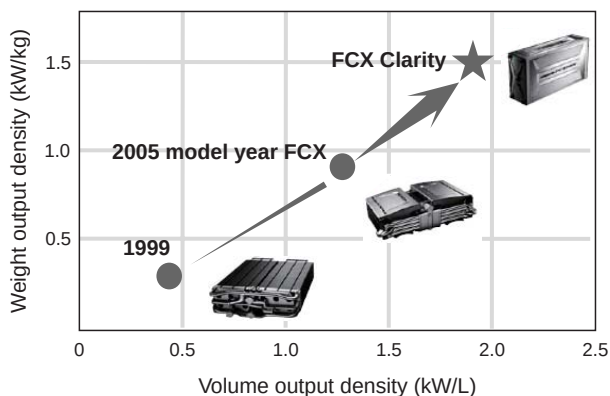


Fig. 11 Evolution of Honda fuel cell stacks

hydraulic braking has increased the Clarity's rate of recovery of energy regenerated during braking by 11% against the previous FCX, enabling recovery of 57% of total braking energy (LA-4 mode). The Li-ion battery system enables assist to be provided to the output of the fuel cell stack at maximum power for an extended period than the ultracapacitor employed in the previous FCX (Fig. 14).

The use of a side-flow system to cool the batteries and employment of a one-box configuration for both the battery pack and ECU have resulted in a 40% weight

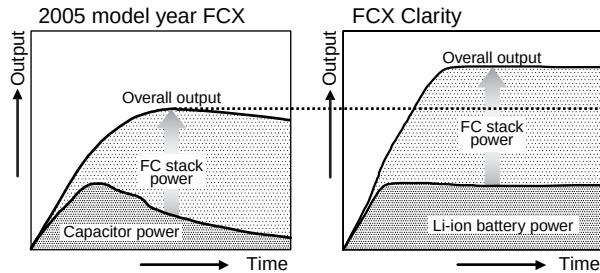


Fig. 14 Assist power characteristic

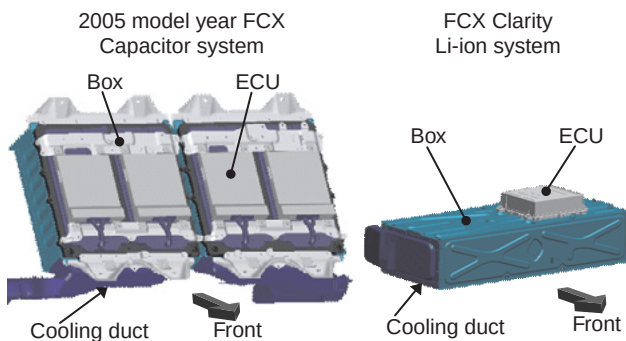


Fig. 15 Comparison of assist batteries

reduction and 50% volume reduction in the unit against the ultracapacitor employed in the previous FCX. As a result, it has been possible to fit the Li-ion battery system underneath the rear seats. This has contributed to the realization of spacious and comfortable rear seats, a low floor and low overall cabin height, as well as a functional trunk space (Fig. 15).

4.1.4. High-pressure hydrogen supply system

One high-pressure hydrogen tank has been used in place of the previous two tanks in order to increase rear seat comfort and realize a roomy trunk space. In addition, the filling and supply system, comprising a cutoff valve, regulator, pressure sensor, and other components, has been redesigned and condensed to be used as an in-tank module. This has enabled the number of parts employed in the high-pressure hydrogen supply system to be reduced by 74%. As a result, the volume of the tank has been increased while its space efficiency has been increased by 24% (Fig. 16).

In combination with the Li-ion battery system discussed in section 4.1.3, this has enabled the realization of a 457-liter trunk space (VDA method), 3.4 times the trunk space of the previous model.

5. Vehicle Performance

5.1. Acceleration Performance

The use of a Li-ion battery able to supply assist power for an extended period in addition to the achievement of increased output from the fuel cell stacks has resulted in increased total power, and has enabled the system to maintain a high level of power output for an extended period. The newly designed drive motor is able to output a maximum power of 100 kW, and can maintain this level of output up to high speeds, enabling a high level of drive power to be realized across the entire range of vehicle

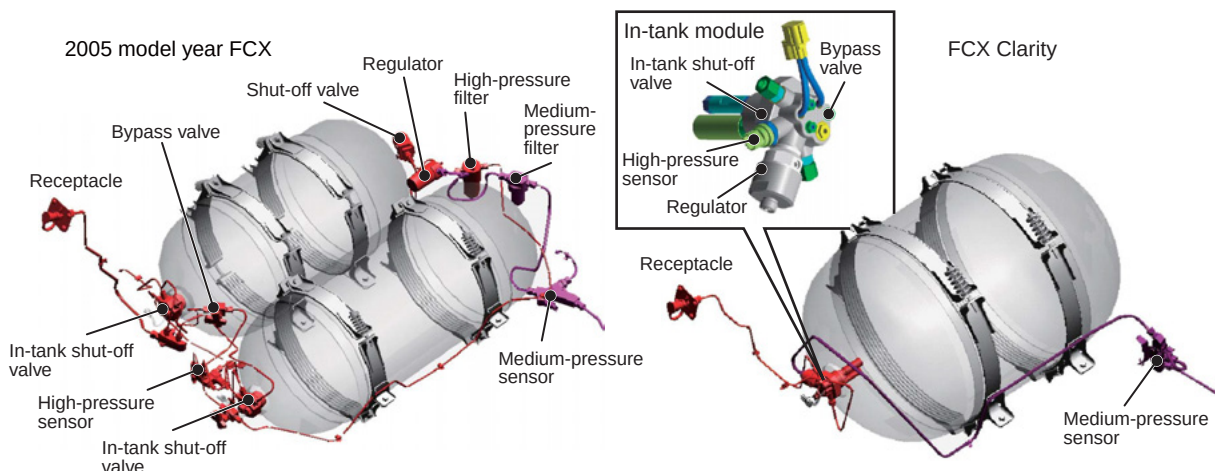


Fig. 16 Hydrogen supply system

speeds, from directly after takeoff to the high speed range.

Figure 17 shows the Clarity's acceleration characteristic. One of the features of the electric motor is its ability to supply maximum torque from low speeds, resulting in rapid acceleration pickup. In addition, the motor's ability to supply continuous torque without shifting gears offers smooth and uninterrupted acceleration. Take-off acceleration time has been increased by 34% (0-60 mph, 9.7 sec) and passing acceleration time by 75% (50-70 mph, 5.9 sec) against the previous FCX, resulting in the realization of a smooth, powerful and continuously extendable acceleration feel (Fig. 18).

5.2. Efficiency and Fuel Economy

The increased generating efficiency of the FCX Clarity fuel cell system and the increased rate of recovery of regenerated energy achieved through the use of a Li-ion battery and hydraulic-regenerative cooperative braking system has increased the rate of capture of regenerated energy, enabling the achievement of an energy efficiency of 60% (LA-4 mode). This figure is more than three times higher than the figure for a gasoline-engine vehicle and more than twice as high as

that for a hybrid vehicle (Fig. 19).

This increased energy efficiency, the reduction of running resistance, and the achievement of weight savings have enhanced fuel consumption performance in the Clarity by 26% against the previous FCX (EPA combined label). The increased compactness of the powertrain incorporating the small-size, high-power V Flow FC stack has enabled the realization of a low-floor sedan platform, which has contributed to the reduction of running resistance.

The energy of 1 kg of hydrogen can be regarded as basically equivalent to the energy of 1 gallon of gasoline. The fuel economy of the FCX Clarity and an equivalently-sized compact gasoline vehicle and compact hybrid vehicle was compared (EPA combined label). The FCX Clarity recorded a fuel economy of 72 miles/kg- H_2 , corresponding to a level 2.1 times higher than that of the gasoline-engine vehicle and 1.4 times higher than that of the hybrid vehicle (Fig. 20).

5.3. Vehicle Range and Refuel Time

In addition to the Clarity's increased fuel consumption performance, the volume of the vehicle's hydrogen tank

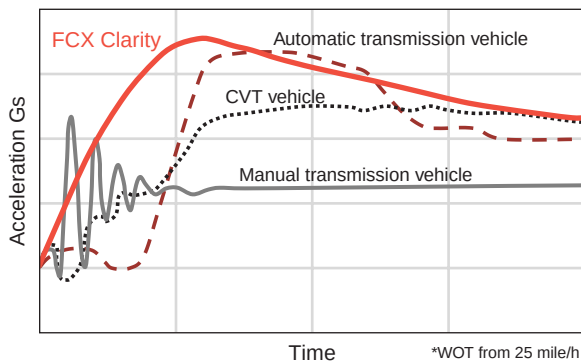


Fig. 17 Takeoff acceleration

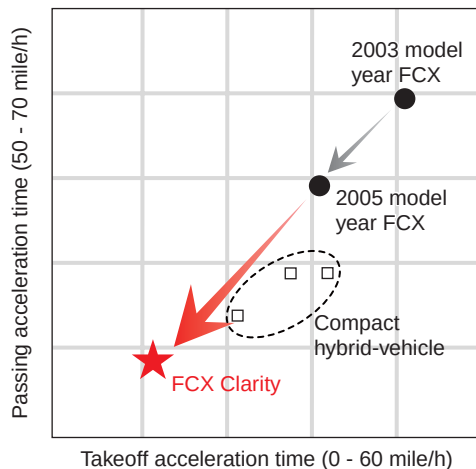


Fig. 18 Acceleration performance

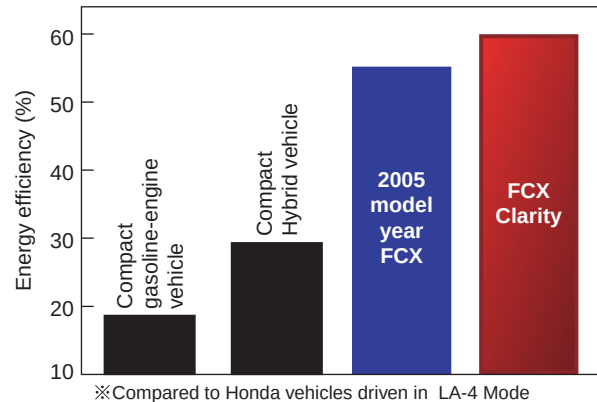


Fig. 19 Vehicle energy efficiency

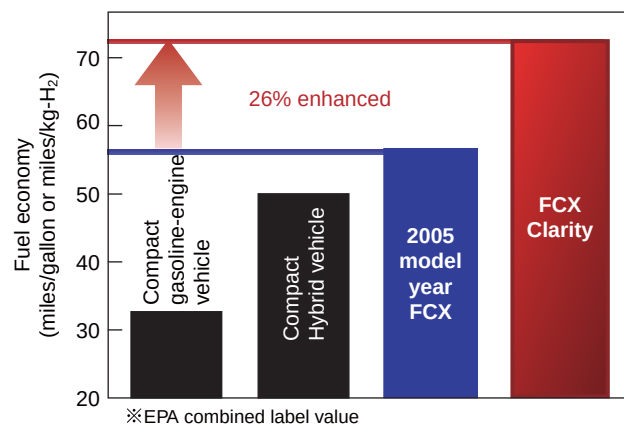


Fig. 20 Vehicle fuel economy

has been increased by 14 L against the previous FCX, enabling the achievement of a 30% increase in vehicle range to 280 miles (EPA combined label) (Fig. 21). This vehicle range is not inferior to that of a reciprocating engine vehicle.

Vehicle range represents an issue not just for fuel cell vehicles but also for electric vehicles. In the mid-1990s, automotive manufacturers made advances in the development of pure electric vehicles (pure EVs). However, the low range and long refueling times of the vehicles were issues, and they did not sell in large numbers. The more recent enhancement in battery performance achieved by the development of the Li-ion battery has seen attention beginning to be focused once again on the potential of electric vehicles. Estimates were therefore made of the vehicle range and refuel time of the EV PLUS, developed by Honda in 1997, in the case its Ni-MH battery were to be replaced by the latest Li-ion battery.

The results of these calculations showed that the vehicle's range would increase to approximately 160 miles. The charging time for the assumed electric vehicle is approximately 12 hours in the case of home charge, and approximately 1.5 hours in the case of quick charging.

Even when contemporary enhancements in battery performance are taken into consideration, there is still a considerable gulf between electric vehicles and reciprocating engine vehicles in terms of vehicle range and refuel time. Electric vehicles are suited to commuters to whom these shortcomings do not represent issues. By contrast, vehicle range and refuel time do not represent restrictions in the case of the FCX Clarity. Able to refuel with approximately 4 kg of hydrogen in 4 minutes, the vehicle offers a level of convenience equivalent to that of a reciprocating engine vehicle.

Powered by hydrogen rather than fossil fuels, the FCX Clarity produces no emissions of the greenhouse gas CO₂, but in terms of vehicle performance competes with a reciprocating engine vehicle.

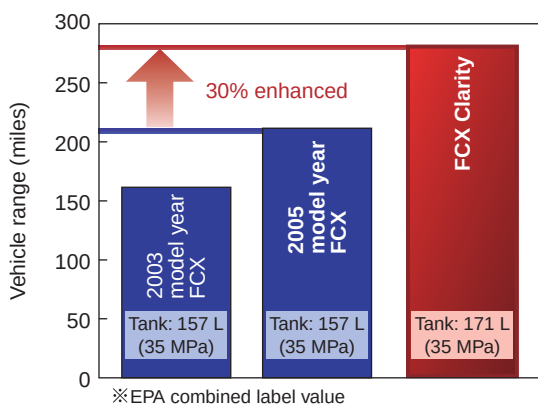


Fig. 21 Vehicle range

6. Conclusion

The following results have been achieved by reducing the size and weight and increasing the efficiency of each component of the FCX Clarity fuel cell powertrain, increasing the power of the vehicle's electric drive system, and increasing power regeneration and assist performance through the use of a Li-ion battery:

- (1) A low-floor platform has been realized by means of positioning the fuel cell system in the center tunnel, and a futuristic cabin has been realized through the use of multiplex meters and other design features.
- (2) The reduction of the longitudinal length of the electric drive system, the reduction of the size of the fuel cell system and high-pressure hydrogen supply system, and the positioning of the Li-ion battery system under the rear seats have enabled the realization of a spacious full cabin.
- (3) The Clarity has achieved enhanced acceleration performance against the previous model, with a 34% faster take-off acceleration time and 75% faster passing acceleration time.
- (4) The Clarity has achieved a 280 mile vehicle range (EPA combined label), 30% higher than that of the previous model.
- (5) The range of temperatures at which the Clarity can start is wider than the previous model, with the vehicle able to be started at -30°C .

7. Afterword

The FCX Clarity has eclipsed the performance of previous fuel cell vehicles. At present it is one of the highest-performing zero emission vehicles (ZEV) known to date. The appeal of the Clarity's performance and design has reached a new dimension, transcending the framework of appeal of the reciprocating engine vehicle.

The FCX Clarity has been leased in California from July 2008. The issue involved in achieving mass market acceptance of the vehicle in the future is the development of technologies that will reduce costs. In order to contribute towards combating global warming and providing solutions to energy issues, the authors will continue to pursue technological innovation in order to achieve reduced costs and mass sales of the FCX Clarity, thus enabling more customers in a wider range of regions to experience the appeal of the fuel cell vehicle.

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New Fuel Cell Stack for FCX Clarity

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ABSTRACT

In order to achieve weight savings, increased compactness, and higher power output from the fuel cell stack that functions as the core technology for a fuel cell vehicle, Honda has employed a unique configuration in the development of its new V Flow FC stack. The development of the new FC stack has enabled the realization of a V Flow FC platform in which the FC stack is positioned in the center tunnel. This has in turn enabled the use of a full sedan layout.

Based on a V Flow cell structure and wave flow channel separators, and employing a one-box stack configuration, the V Flow FC stack achieves a 50% increase in volume output density and 67% increase in weight output density against the previous FC stack. In addition, the new configuration has increased water drainage capacity, which has reduced heat mass and resulted in enhanced low-temperature startability. Continuous production and automated layering of the MEA and separators that comprise the cells, in addition to a reduction in the number of parts, have enabled the achievement of increased productivity.

1. Introduction

Emitting no CO₂ or environmentally-harmful substances, fuel cells are viewed as a clean, resource-recycling powerplant, able to assist in responding to global warming and energy issues. Fuel cell vehicles employ a system in which electricity generated by fuel cells is used to drive a motor, and Honda has pushed ahead with R&D on fuel cells in the belief that a vehicle embodying a new sense of value could be created by means of the design of the powerplant layout and adjustment of its power characteristic.

Honda first fitted a fuel cell stack developed in-house in an experimental fuel cell vehicle in September 1999. In October 2003, the company announced the development of the Honda FC stack, employing stamped metal separators with unitized seals and aromatic electrolytic membranes. The Honda FC stack achieved a high power density, and was able to start in cold climates⁽¹⁾.

As a result of continuous efforts to introduce further innovations to the configuration of its fuel cell stacks, Honda has been able to develop a one-box stack, using a proprietary V Flow cell structure and wave flow channel separators, that achieves a 50% increase in volume output density and 67% increase in weight output density. In September 2006, the company announced an FCX concept based on its proprietary V Flow FC stack and a V Flow

FC platform in which the V Flow FC stack was fitted in the center tunnel. In November 2007, the FCX Clarity, a vehicle embodying the FCX concept, was announced. Production of the FCX Clarity commenced in June 2008.

This paper will provide an overview of the V Flow FC stack configuration, which has enabled the realization of weight savings and increased compactness, higher power, enhanced low-temperature startability, and increased productivity. Figure 1 shows a cut-away view of the V Flow FC stack. In this image, hydrogen and its flow path is shown in red, and air and its flow path is shown in blue.



Fig. 1 Cut-away model of V Flow FC stack

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2. What is a Fuel Cell?

As Fig. 2 shows, a proton exchange membrane fuel cell is a cell that utilizes a polymer electrolytic membrane to directly convert chemical energy into electrical energy via an electrochemical reaction between hydrogen and oxygen.

Hydrogen electrode (-): $\text{H}_2 \rightarrow 2\text{H}^+ + 2\text{e}^-$

Oxygen electrode (+): $\frac{1}{2}\text{O}_2 + 2\text{H}^+ + 2\text{e}^- \rightarrow \text{H}_2\text{O}$

The polymer electrolytic membrane that allows the passage of hydrogen ions is sandwiched between two electrode layers (a hydrogen electrode and an oxygen electrode) and two diffusion layers, which together form a Membrane Electrode Assembly (MEA)⁽²⁾. This MEA is sandwiched between two separators, forming a single cell. Because the voltage of a single cell cannot exceed a theoretical voltage of 1.23 V even when the circuit is open, the cells are stacked as shown in Fig. 3, and a high voltage is produced by the generation of electricity by

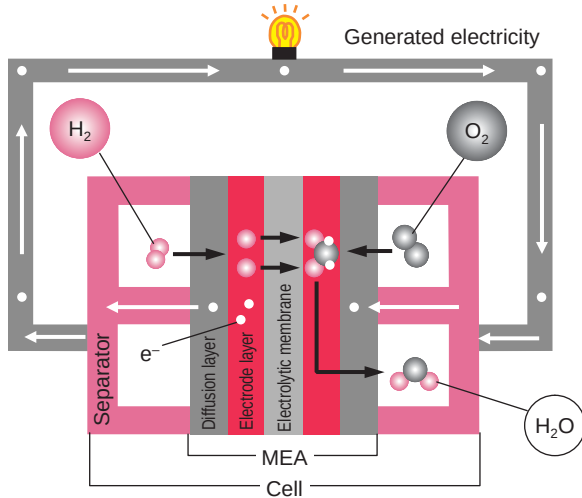


Fig. 2 Mechanism of electricity generation in fuel cell stack

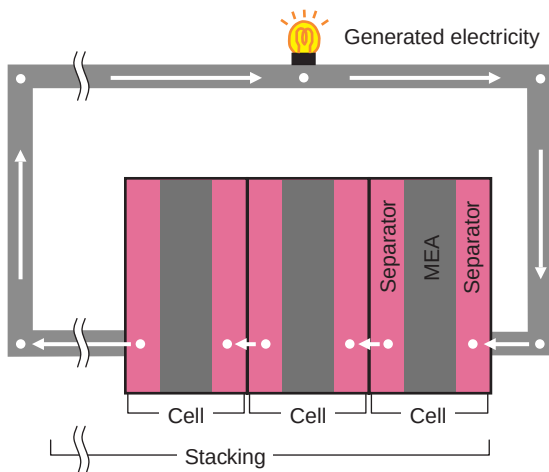


Fig. 3 Fuel cell stack configuration

each individual cell. Voltage is therefore determined by the number of cells in the stack, and current is determined by the area of the electrodes. Because of this, it is necessary to increase either the number of cells in the stack or the area of the electrodes in order to increase power.

3. Development Goals

The following goals were set for the development of a fuel cell vehicle:

- (1) Contribution to combating global warming and addressing energy issues.
- (2) Embodiment of new values that change the future of the automobile.
- (3) Development of mass production technologies looking towards widespread diffusion.

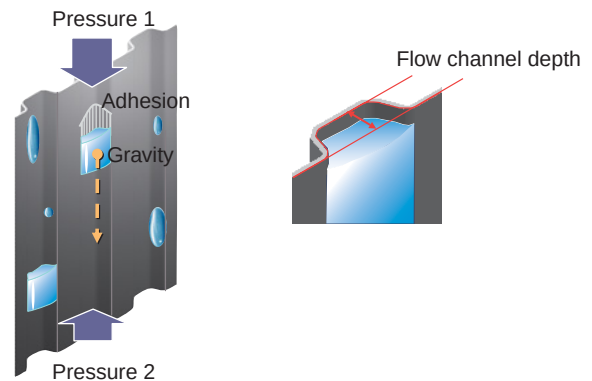
The following four points were therefore established as the main targets in the development of the new fuel cell stack:

- (a) Achievement of weight savings and increased compactness
- (b) Increased power and stability of electrical generation
- (c) Enhanced low-temperature startability
- (d) Increased productivity

4. V Flow Cell Structure

In order to generate electricity by the fuel cells, it is necessary to provide hydrogen and air continuously over the generating surfaces. However, if the water produced from the reaction between the hydrogen and oxygen accumulates in the gas channels, it can impede the flow of gas, preventing hydrogen and air from being supplied uniformly to the generating surfaces and making generation unstable.

In order to make the fuel cell stack more compact, it



$$\text{Water drainage capacity} = \text{Pressure 1} - \text{Pressure 2} + \text{Gravity}$$

$$\text{Flow channel depth} = \frac{\text{Water adhesion force}}{\text{Water drainage capacity}}$$

Fig. 4 V Flow cell structure

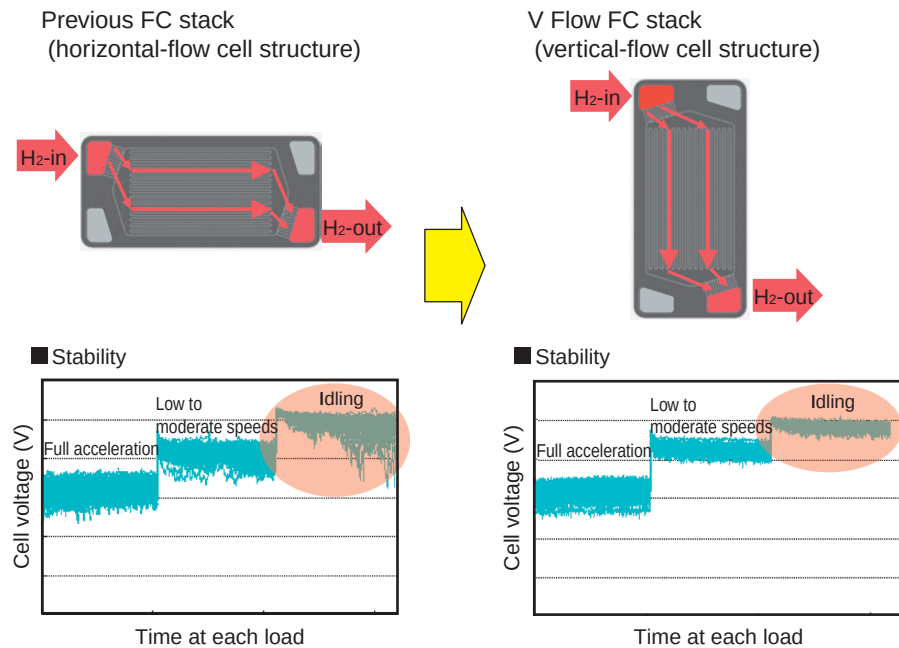


Fig. 5 Comparison of cell structure

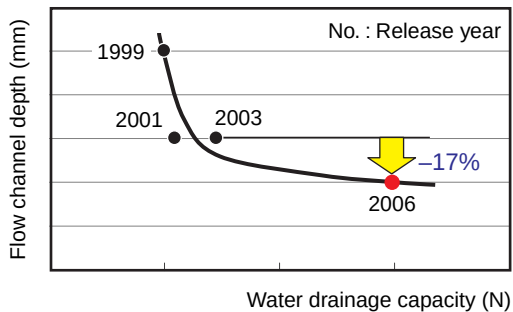


Fig. 6 Comparison of flow channel depth

is necessary to reduce the thickness of the cells by reducing the depth of the gas flow channels for hydrogen and air. However, reducing the depth of the gas channels would increase the force producing adhesion of the water to the walls of the channels, thereby promoting water accumulation. The relationship between the depth of the channels, the water adhesion force and the water drainage capacity can be expressed as follows (Fig. 4):

Flow channel depth = Water adhesion force / Water drainage capacity

In order to reconcile the achievement of stable generation performance with reduction of the depth of the gas channels, it is therefore necessary to increase the water drainage capacity to the same degree that the water adhesion force increases as a result of the reduction of the depth of the channels. This led to the development of a V Flow cell structure, in which hydrogen and air flow vertically, rather than horizontally, as in the previous FC stack. In the previous horizontal-flow cell structure, the

difference in pressure between the inlet and the outlet of the cells functioned as the only force promoting drainage of the water in the flow channels. The new vertical-flow cell structure adds gravity to this pressure difference, thus increasing the water drainage capacity. This has enabled stable generation to be maintained despite the reduction in depth of the flow channels (Fig. 5). As a result, the depth of the flow channels has been reduced by 17% against that of the previous cells (Fig. 6).

5. Wave Flow Channel Separators

Generation performance can be maximized by achieving a uniform flow of hydrogen, air, and coolant to the generating surfaces. If the coolant in particular does not flow uniformly, localized areas of high temperature will form in the MEA, and changes will occur in the MEA in those areas.

The use of stamped metal separators and straight flow channels creates a configuration in which hydrogen, air, and coolant all flow in the same direction. This necessitated the use of three manifolds positioned in a row in each cell to introduce the substances, reducing the width of their supply path to the generating surfaces and impeding uniform supply.

It was judged that positioning the gas manifolds vertically and widening the gas supply paths, in addition to bifurcating the short sides of the generating surfaces, would be an effective means of achieving a uniform supply of hydrogen and air to the generating surfaces. Wave flow channel separators were therefore used to achieve a horizontal flow of coolant (Fig. 7). The development of wave flow channel separators that turn the

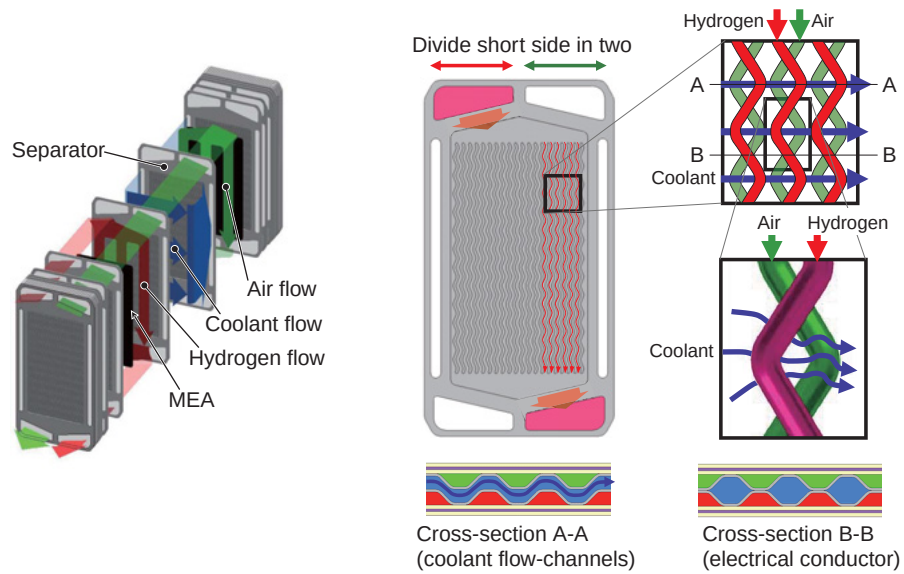


Fig. 7 Wave flow channel separator structure

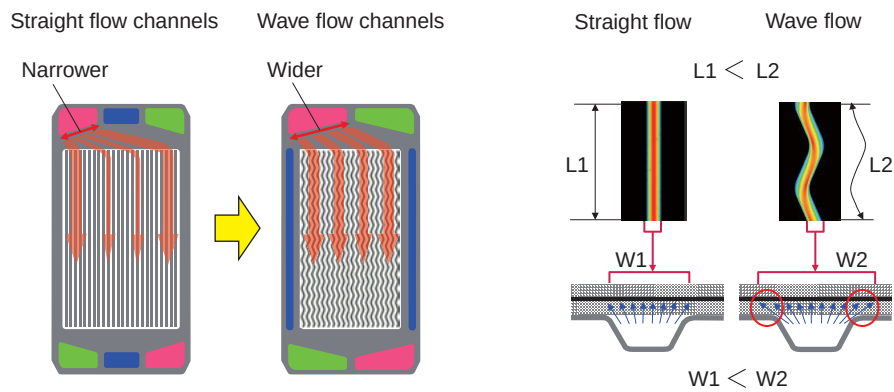


Fig. 8 Enhanced hydrogen-air distribution

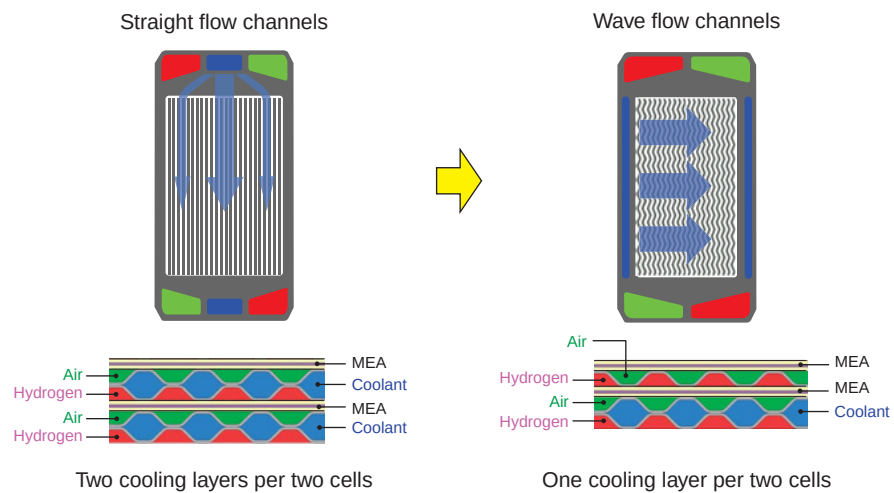


Fig. 9 Enhanced coolant distribution

coolant flow in a horizontal direction, across the flow of hydrogen and air, enabled the width of the supply paths for both gases and coolant to be increased, enabling the achievement of uniform supply to the generating surfaces. In addition, because each wave flow channel is longer than a corresponding linear flow channel, and varies the flow, the use of the wave flow channel separators has increased the dispersion of hydrogen and air (Fig. 8).

As a result, hydrogen and air pass over the entire surface of the electrode layer. More effective use is made

of the generating surfaces, resulting in a 10% increase in generating performance against that of the linear flow channels. In addition, the horizontal coolant flow has helped to achieve uniform cooling of the generating surfaces, enabling the cooling layers that were previously required for each cell to be halved in number, and the number of separators required for every two MEA to be reduced from four to three (Fig. 9). As a result, the thickness of the stacks has been reduced by 20%, and their mass by 30% (Fig. 10).

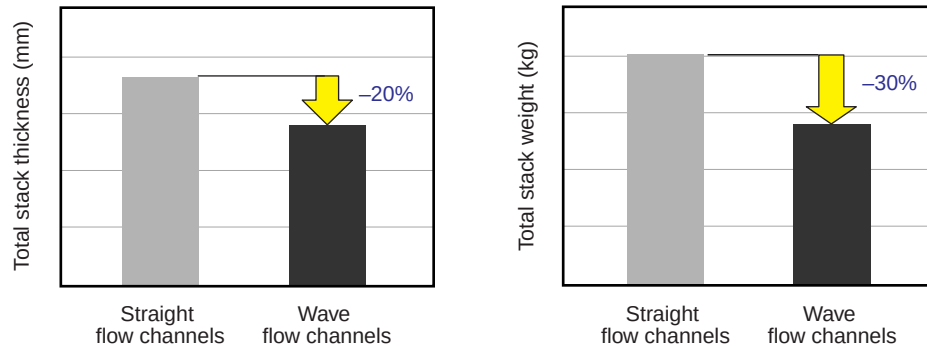


Fig. 10 Reduction of stack thickness and weight

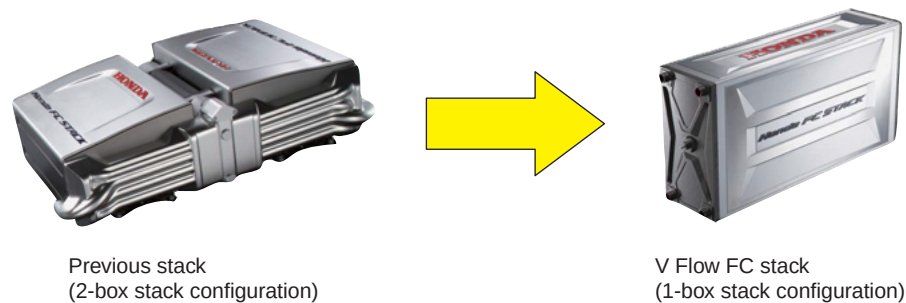


Fig. 11 Evolution of stack configuration

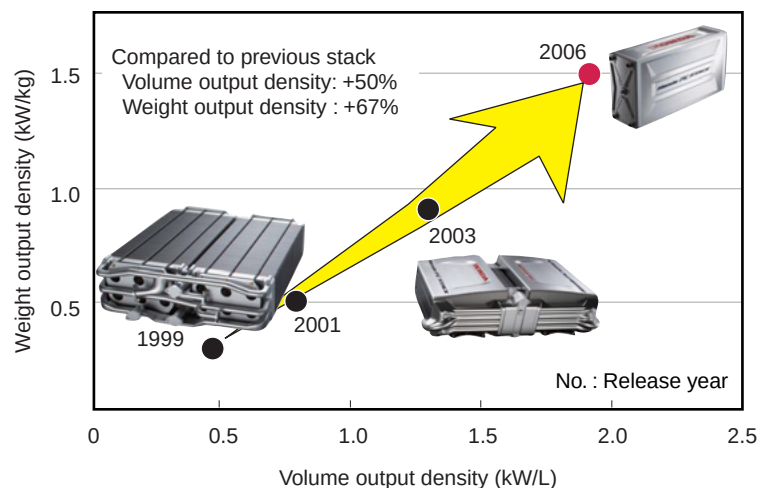


Fig. 12 Comparison of stack output density

Table 1 Evolution of Honda fuel cell stack

Release year	1999	2003	2006
Appearance			
Output	60 kW	86 kW	100 kW
Volume	134 L	66 L	52 L
Weight	202 kg	96 kg	67 kg
Electrolytic membrane	Fluorine electrolytic membrane Operating temperature: 0 °C - 80 °C	Aromatic electrolytic membrane Operating temperature: – 20 °C - 95 °C	Aromatic electrolytic membrane Operating temperature: – 30 °C - 95 °C
Stack configuration	Bolt-fastened configuration	Panel box configuration	1-box stack configuration
Cell structure	Machined carbon separators Separate seals	Stamped metal separators with unitized seals	V Flow cell structure Wave flow channel separators

6. One-box Configuration

The reduction in the thickness of the cells achieved through the use of a V Flow cell structure and wave flow channel separators has helped to enable a one-box FC stack configuration to be employed in place of the previous two-box configuration (Fig. 11). As a result, volume output density has been increased by 50% and weight output density by 67% against the previous stack, and packageability has been enhanced (Fig. 12).

Table 1 shows the evolution of the Honda FC stack. The new one-box stack, achieving a volume output density of 1.92 kW/L, has helped to enable the realization of the full sedan layout used in the FCX Clarity, a V Flow FC platform in which the fuel cell stacks are fitted in the center tunnel.

7. Enhanced Low-temperature Startability

The previous Honda FC stack employed aromatic electrolytic membranes in which membrane resistance had been reduced by half and stamped metal separators with high heat conduction, enabling generation to be commenced at -20°C.

One of the goals established for the present FC stack was to enable its use in a wider range of areas. The use of the V Flow cell structure has increased water drainage

capacity and reduced heat mass by 40% (Fig. 13), resulting in increased output at start-up. Heat mass was reduced as a result of the reduction of the volume of coolant and the number of separators employed, enabled by the use of the wave flow channel separators, and by the reduction of the number of box components resulting from the use of a one-box stack configuration. The lower heat mass has enabled the time required to reach 50% output at start-up at -20°C to be reduced by three-quarters against that in the previous stack (Fig. 14), and has also enabled start-up at -30°C.

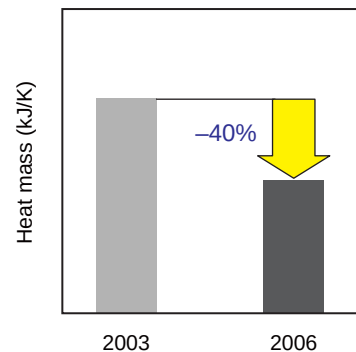


Fig. 13 Comparison of heat mass

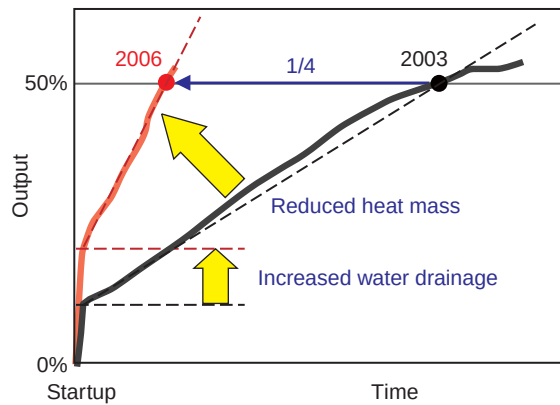


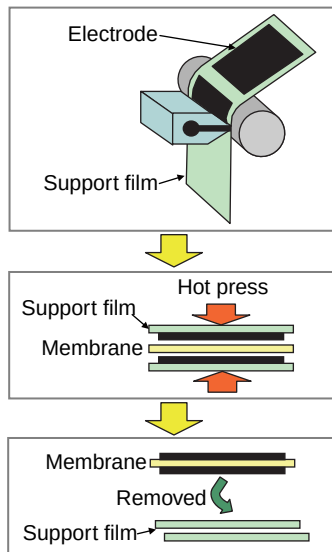
Fig. 14 Comparison of low-temperature performance (-20°C operation)

8. Increased Productivity

Fuel cell stacks are made up of some hundreds of layered cells, making the achievement of increased productivity for the cell components an issue. Production technologies that increased productivity for the MEA and separators that make up the cells and cell layers, in addition to specifications that were appropriate for these production technologies, were therefore developed.

In the case of the previous MEA, in order to produce a Catalyst Coated Membrane (CCM) prior to sandwiching between the diffusion layers, the electrode was hot-pressed onto the membrane after the electrode had been coated on a support film. For the new cells, a technique of shape-controlled electrode coating on the membrane has been

Conventional process



New process

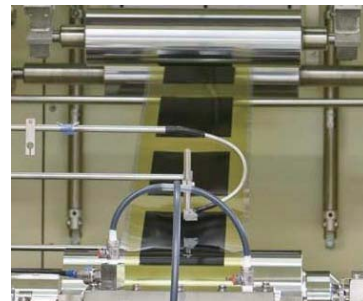
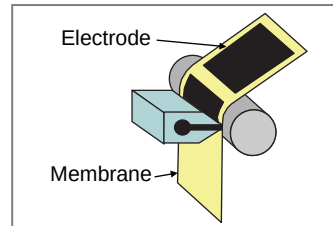
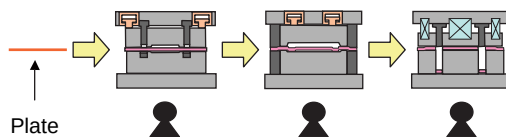


Fig. 15 Electrode coating process

Conventional process



New process

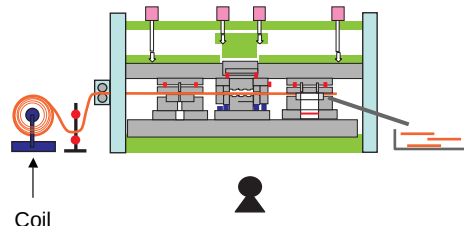


Fig. 16 Separator stamping process

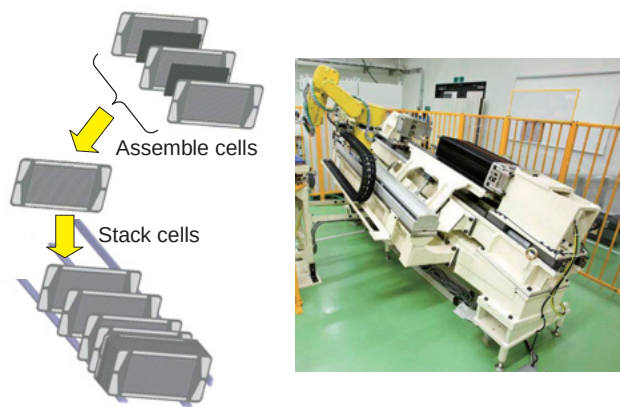


Fig. 17 Cell stacking process

developed to enable direct formation of the electrodes in a predetermined shape on the roll-shaped electrolytic membrane (Fig. 15). In addition, progressive stamping using metal in coils has been employed to produce the stamped metal separators, in place of the previous method of stamping individual plates in successive batches, in order to enable continuous production (Fig. 16).

The previous MEA were easily broken, making their handling during layering a considerable challenge. In the new configuration, two cells formed from two MEA and three separators are treated as a unit, enabling layering to be carried out automatically by robots as if stacking books on a bookshelf (Fig. 17).

In addition, the reduction of the number of cooling layers and use of a one-box configuration has enabled the number of components to be reduced.

The development of these production technologies and appropriate specifications has enabled the achievement of continuous production, resulting in increased productivity.

9. Conclusion

The V Flow FC stack forms the core of the V Flow FC platform employed in the FCX Clarity, a fuel cell vehicle that embodies an advanced design. The following major technologies were applied to the V Flow FC stack:

- (1) The development of a V Flow cell structure enabling vertical flow of hydrogen and air has increased water drainage capacity. This has enabled compactness to be increased without compromising the stability of electricity generation and has enhanced low-temperature startability.
- (2) The development of wave flow channel separators that produce a uniform supply of hydrogen, air, and coolant to the generating surfaces and enable the number of cooling layers to be reduced has resulted in the achievement of weight savings, increased compactness, higher power, and enhanced low-temperature startability.
- (3) The reduction of the thickness of the cells and the achievement of weight savings and increased

compactness have enabled the use of a one-box stack configuration in place of the previous two-box stack configuration, enabling the FC stack to be positioned in the center tunnel of the vehicle.

- (4) Continuous production of the MEA and separators that make up the cells, automated layering, and the reduced number of cell components has resulted in increased productivity.

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Electric Shift System for FCX Clarity

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ABSTRACT

An electric shift system has been developed for the FCX Clarity, a vehicle incorporating numerous Honda advanced technologies. Optimizing the size of the shift-by-wire (SBW) shifter and setting an operation load and stroke that make the shift easy to use has enabled the realization of an advanced design. A parking actuator has been used to lock and release the transmission parking gear, and control by means of an electronic control unit (ECU) has enabled the achievement of fast responsiveness. The engineering of redundancy into the SBW shifter and parking actuator control unit has helped to ensure reliability. In addition, an alternative means of operating the parking gear has been provided in the event that the battery stops supplying power.

1. Introduction

The development and diffusion of electric and fuel cell vehicles is an important means to help the reduction of the environmental impact of automobiles. Honda commenced leasing of the production fuel cell vehicle FCX in Japan and the U.S. in 2002, and since then has worked to make further strides in the evolution and enhancement of the performance of its fuel cell vehicles. Shifters for automatic transmissions have conventionally been positioned on the instrument panel or the center console. The recent appearance of electric shift systems in the market has enabled the location of shifters in easy-to-use positions and the realization of advanced design. The employment of electric shift system increases the degree of freedom of positioning and modes of operation used, enabling a balance between advanced design and operability at a higher level.

This paper discusses the development of an electric shift system to enable a realization of the enjoyment offered by a futuristic style of operation, which was one of the development goals for the FCX Clarity, in addition to the element technologies employed in order to reconcile advanced design with operability.

2. Development Goals

The main goals set for the development of the electric shift system were as follows:

- (1) Realization of a high level of shifter operability
- (2) Realization of an advanced design by integrating the shifter with the meter visor
- (3) Realization of rapid responsiveness to driver input

3. System Configuration

3.1. Overall Configuration

Figure 1 shows the overall configuration of the electric shift system. The shift-by-wire (SBW) shifter, which enables the driver to control the shift position, is positioned in the meter visor.

In addition to input and output to and from the electric shift system (shifter signal and position sensor signal input, parking actuator control unit signal input and output, and the like), electric shift control required a variety of other parameters, including brake signal, accelerator pedal signal, and vehicle speed signal. Processing is therefore conducted in an electric control unit (ECU) that controls all powerplant systems.

* Automobile R&D Center

Figure 2 shows a diagram of the electric shift control system.

The ECU and the parking actuator control unit, which controls the parking actuator, are positioned in the instrument panel. The parking actuator, which locks and releases the transmission parking gear, is located on the electric motor case.

3.2. SBW Shifter

Figure 3 shows an external view of the SBW shifter and the parking switch. The SBW shifter knob is positioned to enable the driver to reach it by a slight extension of the hand from the steering wheel. Together

with designers, the project team created a number of possible knob forms, eventually deciding on the final form as the one that enabled a simple fingertip operation based on tests of operability. The size of the SBW shifter has been optimized in order to enable realization of a design that integrated the shifter with the meter visor at the side of the meters.

Figure 4 shows the configuration of the SBW shifter. The sensor section, which converts the driver's operations into electrical signals, is separated from the mechanical section. This configuration helps to ensure that the mechanical section bears the load generated by the driver's operation of the shifter, without affecting the

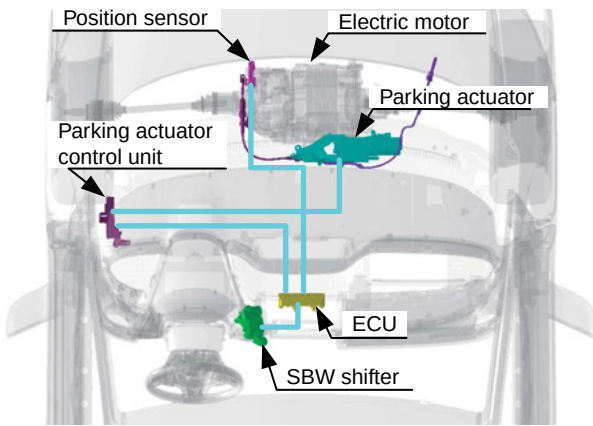


Fig. 1 SBW system layout

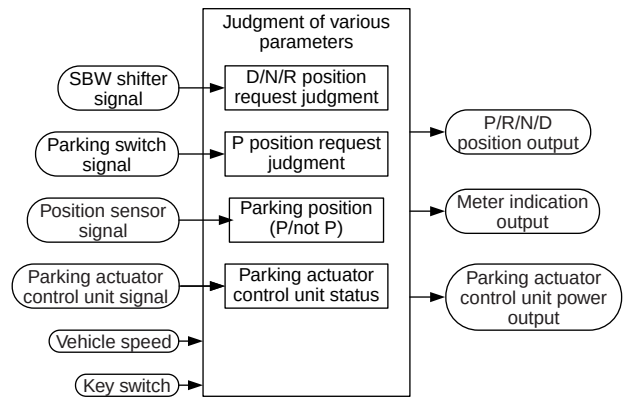


Fig. 2 Control system block diagram



Fig. 3 SBW shifter

sensor section.

The operation of the SBW shifter is converted into electric signals by multiple sensors and directly input to the ECU. In order to help prevent mistaken shift selection due to a sensor malfunction, the sensor output patterns for each position have been configured by combining sensor signals.

Figure 5 shows the shift pattern of the SBW shifter. In order to enable operation from the same starting position and in the same pattern for each position⁽¹⁾, the SBW shifter has been provided with a home position to which it will return if the driver removes his or her hands from the SBW shifter knob. The D position can be smoothly selected by pushing the SBW shifter knob to the lower left, while the R position requires a U-shaped movement, a pattern that helps prevent mistakes in operation. In addition, a parking switch that enables the driver to shift to P position from any other position has been fitted below the SBW shifter, thus reducing the time required for operation. Tests were conducted to determine the shift pattern design that was easy to comprehend visually from among a number of proposed designs, leading to the selection of a design in which the direction of motion of the SBW shifter knob was marked by arrows.

3.3. Parking Actuator

Figure 6 shows the configuration of the parking

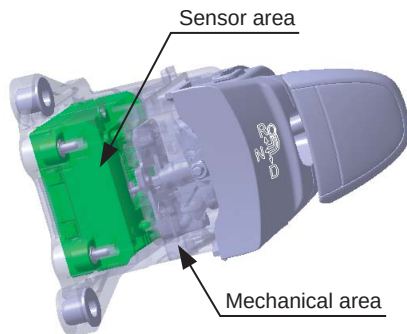


Fig. 4 Configuration of SBW shifter



Fig. 5 Shift pattern

actuator. The parking actuator locks and releases the parking gear by means of a control wire.

Figure 7 shows the mechanical section of the parking actuator. The rotation of the motor is converted to the stroke of a linear shaft by a ball screw. The high efficiency of the ball screw enables manual stroke of the linear shaft.

Locking and release of the parking gear is set by the positioning of a detent. To enable coordination of the control of parking actuator and the positioning of detent, there is some play in the linear shaft stroke in relation to the contact surface of the ball screw, which helps to ensure that there is no effect from the positioning of the ball screw or variations in the dimensions of the control wires, and the like. Figures 8 and 9 show an example.

The parking actuator is also provided with a manual control wire that enables it to be moved by hand in order to release the parking gear when the battery is not supplying power. Manual release of the parking gear is shown in Fig. 10. The pulling load required for release of the lock has been set to enable the user to release the lock using only a small tool like a Phillips screwdriver.

3.4. Parking Actuator Control Unit

By contrast with conventional mechanical shift systems that lock and release the parking gear mechanically, locking and release is electrically controlled in the electric

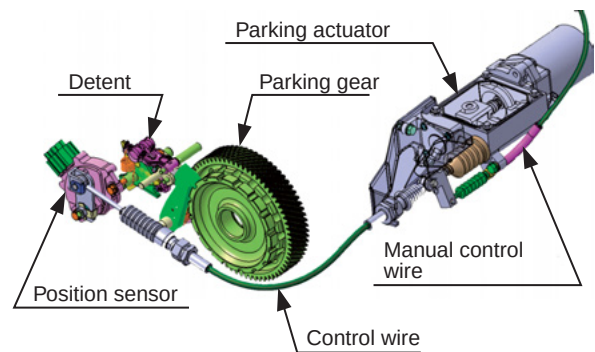


Fig. 6 Parking actuator layout

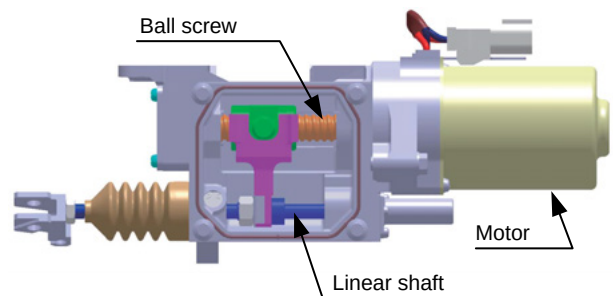


Fig. 7 Parking actuator

shift system. Redundant circuits have therefore been employed in the system to help to ensure that mistaken operation does not occur and the parking gear can be locked and released even if an electrical malfunction occurs in one part of the system.

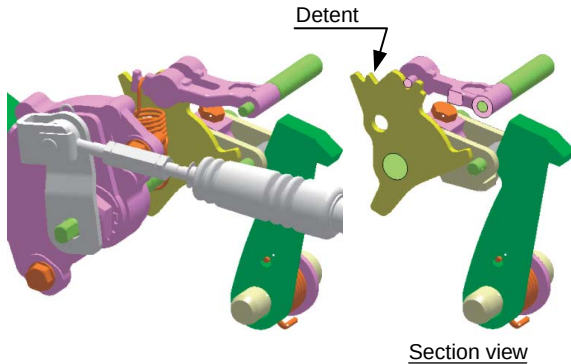


Fig. 8 Transmission detent

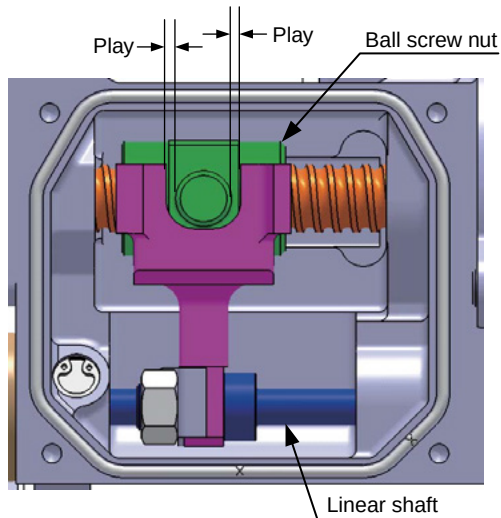


Fig. 9 Play in ball screw nut



Fig. 10 Parking lock release

4. Electric Shift Control

4.1. Non-mechanical Shift Lock Control

Automatic transmissions are provided with a shift lock function to prevent shifting from P position unless the brake pedal is depressed. The shifter has conventionally been directly connected to the lock mechanism of parking gear, and the shift lever has therefore been prevented from moving by mechanical means. Because the SBW shifter lever is not fixed in a selected position, the use of a control method in which the results of operation of the shifter are not reflected in engagement of the shift when the conditions for release of the parking gear are not met (non-mechanical shift lock) has enabled realization of a shift lock function in the SBW shifter without employing additional devices such as solenoids.

4.2. Automatic Parking Gear Control

Automatic transmissions are provided with a key interlock function preventing the key from being removed if the shifter is not in P position. In the electric shift system, P position is automatically selected when the key switch is switched to I (accessory), enabling the driver to remove the key without considering the position of the shifter.



Fig. 11 Shift lock in operation



Fig. 12 R inhibitor in operation



Fig. 13 P inhibitor in operation

4.3. P/D/R Inhibition Control

Automatic transmissions require control systems to help ensure that mistaken operation (for example shifting to P position or the reverse position while driving) does not result in breakdown or vehicle instability. In the electric shift system, shifting to opposite position at a high vehicle speed is judged to be an invalid operation, and the system will automatically select neutral position. Switching to P position while the vehicle is in motion is also judged to be an invalid operation, and the control system will function to maintain the position prior to the operation of the P switch by the driver.

4.4. Electric Shift Operation Assist

The mode of operation of the electric shift system differs from that of a conventional automatic transmission. A multi-information display has therefore been used to inform the driver of the correct method of operation when the operation and results of the operation differ. Figures 11 through 13 show examples of displays.

4.5. Fail-safe Control

In order to safely control the vehicle in the event of an incorrect shift setting selection or an incorrect operation of the system due to a malfunction, malfunctions in various elements are accurately detected and a warning is displayed. Control is applied to help prevent operation of the SBW shifter from being reflected in shift settings if the malfunction continues.

5. Performance

5.1. SBW Shifter Characteristic

Figure 14 shows the results of subjective evaluation of the operation load and stroke of the SBW shifter. Evaluation tests were conducted while changing the operation load and stroke. A load of 4.9 N and stroke of

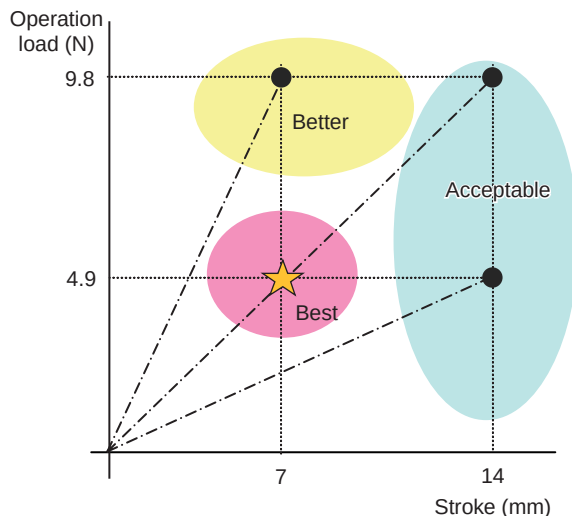


Fig. 14 Rating of SBW shifter operation

7 mm were evaluated most highly, and these settings were therefore selected. In addition, the load that automatically returns the shift knob to home position was optimized for balance with the operation load.

5.2. Parking Actuator Characteristic

During operation of the parking actuator, the present position of the actuator is judged on the basis of the transmission position sensor signal, and the actuator is moved to the position indicated by the ECU. This position switching is detected based on the position sensor signal and then the motor is stopped. Only the linear shaft continues moving, and is stopped by the transmission detent. Position switching displays rapid responsiveness even when shifted from parking to another position. Figure 15 shows an example of parking actuator control.

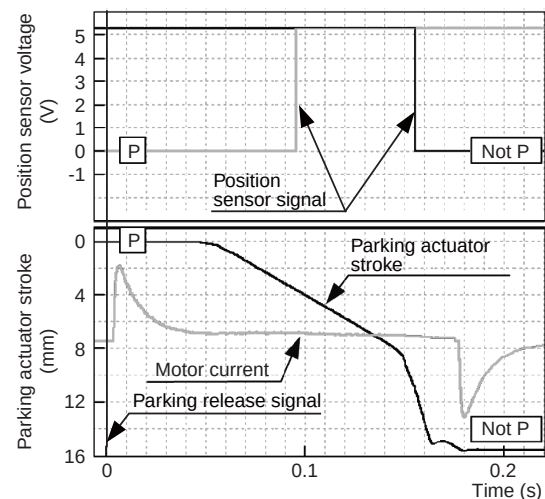


Fig. 15 Example of parking actuator control

6. Conclusion

An electric shift system has been newly developed for the FCX Clarity, which was released in November 2007. Optimized operability and advanced design have been achieved by the development project.

- (1) The use of a new shift pattern and setting the optimized operation load and stroke have increased ease of use.
- (2) Appropriate size of the SBW shifter enabled the realization of an advanced design in which the shifter was integrated with the meter visor. The SBW shifter was positioned at the side of the meters, within easy reach from the steering wheel, which contributed to enhanced readability.
- (3) Rapid responsiveness in switching shift positions has been achieved.

In the future, additional enhancements will be incorporated with the goal of increasing the number of vehicles equipped with electric shift system.

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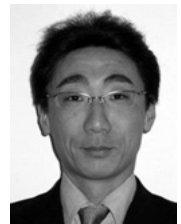
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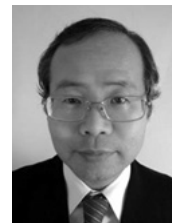
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Heated and Cooled Seat for FCX Clarity

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ABSTRACT

Heated and cooled seats have been developed using Honda's new "bio-fabric" material as a covering in order to enable adjustment of the vehicle cabin temperature for occupant comfort in an environmentally-friendlier manner that is appropriate for the FCX Clarity fuel cell vehicle, a vehicle that realizes the ultimate in clean performance. Thermo-control devices have been positioned in the cushions and backs of front seats (both U.S. and domestic specifications) and rear seats. These devices heat or cool air taken in by a blower, and the air is then directed towards the covering structure of the seat, which has been rendered permeable.

1. Introduction

In response to the environmental issues generated by the consumption of petroleum by automobiles, increased attention is being focused on environmental technologies centering on the use of next-generation energy sources that produce zero emissions and do not utilize fossil fuel resources.

Honda has been pushing ahead with the development of a fuel cell vehicle that is highly energy-efficient and utilizes hydrogen that can be created using natural energy sources, and the development of new bio-fabrics containing materials derived from plants. The company has also been working to increase the comfort of the vehicle interior by increasing the comfort of the temperature of the seat surfaces, part of the vehicle with which users are in direct contact and are therefore highly aware of. Seat heaters have been introduced, mainly in leather seats; a high-permeability covering ("space fabric") was employed in the 2000 model year INSIGHT, and "ventilation seats" were employed in the 2005 model year ACURA RL. Honda has also been involved in research on heated and cooled seats, in which the seat surface is directly heated and cooled, since 1995. This method of heating and cooling is fast-acting and reduces the operating load on the air-conditioner. For these reasons, heated and cooled seats were developed, in combination with the use of the Honda bio-fabric, for the FCX Clarity. This paper will discuss this technology.

2. Development Goals

The following three development goals were set in order to achieve a level of environmental performance appropriate for a fuel cell vehicle and comfortable heating and cooling performance, in addition to the basic functions of the seats and elements in the appeal of the vehicle.

- (1) Use of Honda bio-fabric as a covering
Use of the Honda bio-fabric, made by adding materials derived from plants. This would be the first time that a cloth covering had been used for a heated and cooled seat, offering a natural feel and texture. The use of the bio-fabric would also reduce CO₂ emissions due to the reduced reliance on petroleum in its production.
- (2) Achievement of a more comfortable heating and cooling function than previous systems
More rapid cooling than a ventilation seat and more rapid heating than a heated seat. Achievement of an equivalent level of fan noise and ease of operation to a ventilation seat.
- (3) Maintenance of functions and customer appeal of conventional seats
Employment of additional thermo-control devices, while maintaining the existing seat structures, original functions, and customer appeal of the seats.

3. System Configuration

3.1. Overall Configuration

Figure 1 shows an exterior view of the FCX Clarity

* Automobile R&D Center

in which the new heated and cooled seats are fitted, and Fig. 2 shows an image of one of the seats⁽¹⁾. Heated and cooled seats generally employ a configuration in which a single blower positioned in the seat cushion blows air over thermo-electric modules positioned in the seat back and the seat cushion. In the new seats, the seat cushion and seat back are each provided with a blower and a thermo-electric module, in addition to a seat surface temperature sensor. This enables independent control of air flow in the seat back and the seat cushion, increasing heating and cooling efficiency.

In addition, the fact that an air duct between the seat back and seat cushion was no longer required in order to connect the blower and thermo-electric module meant that no special seat frame was necessary, and that the heating and cooling function could be incorporated without modification of the basic seat structure. The thermo-electric modules in the seat backs have also been provided with exhaust ducts to guide waste heat outside the seats, helping prevent any increase in intake temperature due to heat being trapped in the seat backs.



Fig. 1 FCX Clarity



Fig. 2 Heated and cooled seat

As Fig. 3 shows, the operating switches have been positioned to the left and right of the air-conditioner operating panel, increasing user awareness that the switches operate a heating and cooling system.

3.2. Thermo-control Devices

As shown in Fig. 4, the blower and thermo-electric module that send air to the seat surface are connected into a single unit by a plastic duct. This unit is fixed by screws to brackets attached to the seat frame.

3.2.1. Blower

A thin, low-noise DC brushless motor and a sirocco fan have been used in the blower in order to increase customer appeal. In addition, as Fig. 5 shows, a removable filter has been fitted to the inlet, helping prevent infiltration of dust and foreign matter, and providing for easy replacement.

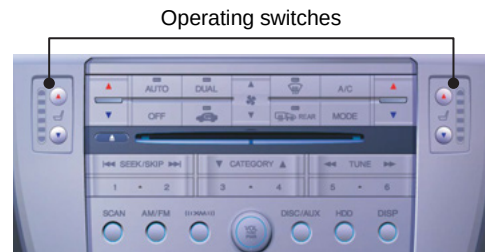


Fig. 3 Location of switches

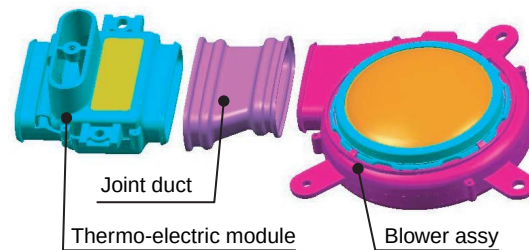


Fig. 4 Thermo-control device

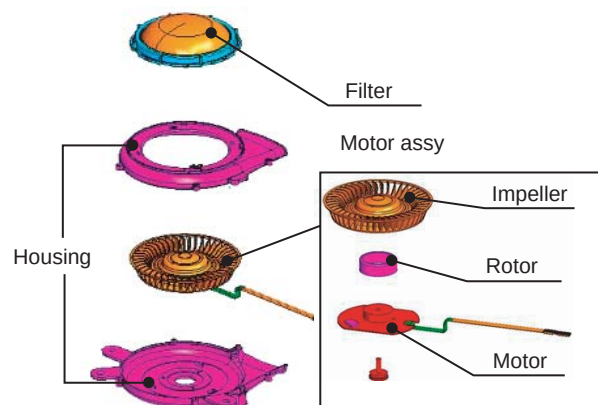


Fig. 5 Blower assy

3.2.2. Thermo-electric module

As shown in Fig. 6, the thermo-electric module is composed of a Peltier module, in which switching between heating and cooling is determined by the direction of the current, sandwiched between radiators above and below. This unit is fitted within a plastic housing. Air from the blower passes across the radiator above the Peltier module and its temperature is adjusted, after which it turns 90° and passes out the top of the unit. Unnecessary heat produced by heat exchange is exhausted at the side. Temperature sensors are positioned on the electrodes of the Peltier module, enabling the temperature of the seat surface to be monitored.

3.3. Control Unit

Relays are generally employed to drive a Peltier module, enabling the current that flows during heating and during cooling to be reversed. On the other hand, the modules employed in the new seats are composed of semiconductor relays using Field Effect Transistor (FET), which has contributed to increased durability and

reliability due to contactless configuration, and enhanced customer appeal due to silent switching.

3.4. Air Channels

As Fig. 7 shows, ditches have been formed in the surfaces of the pads used inside the seat backs and seat cushions in order to direct the air that has been heated or cooled by the thermo-electric modules positioned behind them to the covering structure of the seats. Because cloth has been used as the seat covering for the first time in these seats rather than the leather previously used for heated and cooled seats, plastic plates have been positioned above these ditches (Fig. 8) in order to help prevent a reduction in the air flow due to the ditches constricting when the occupant sits in the seat, reduce the appearance of dents in the surface of the seat, and to minimize occupant discomfort. In addition, adjustment of the position, size, and shape of holes in the plastic plates has increased the efficiency of air supply. Figure 9 shows the results of an analysis of the effects of the plates conducted with and without the plates using computational

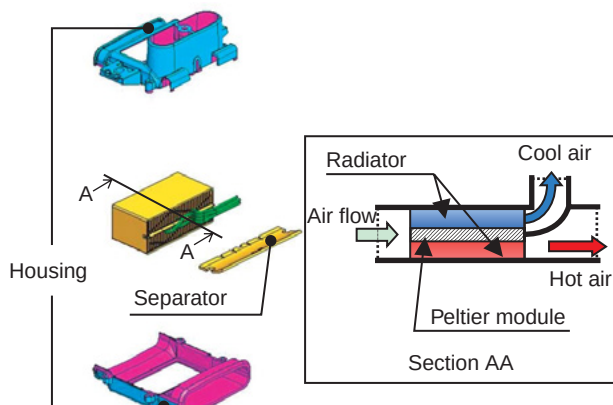


Fig. 6 Thermo-electric module



Fig. 7 Ventilation ditches

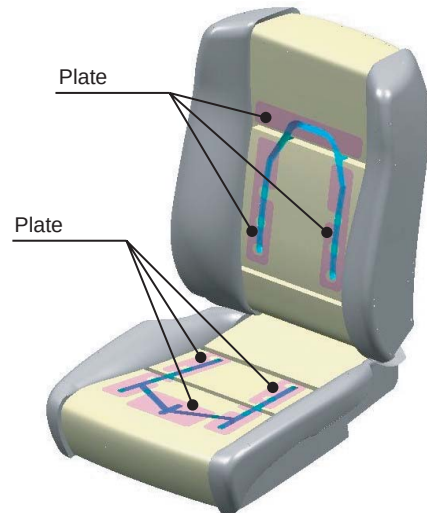


Fig. 8 Plates employed on ditches

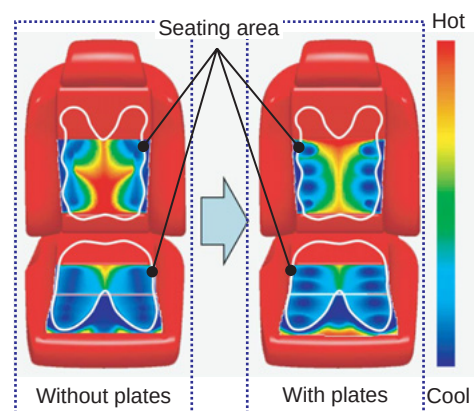


Fig. 9 Results of CFD analysis

fluid dynamics (CFD). These results indicate that adding the plates increased cooling efficiency around the center part of seating area.

3.5. Covering Structure

Figure 10 shows a cross-section of the structure of the seat covering. The use of a permeable foam for the wadding underneath the covering has helped to ensure high permeability and made the surface pleasant to the touch. Figure 11 shows a view of the appearance of the Honda bio-fabric used as the covering material⁽²⁾. The surface has been covered in tiny holes in the same way as the leather previously used in heated and cooled seats, offering an image of high permeability.

3.6. Operating Switches

The operating switches are the same type previously used for the ventilation seats in the ACURA RL. The LED indicator shows that the system is functioning, and lights in red when the seat is being heated or in blue when being cooled.

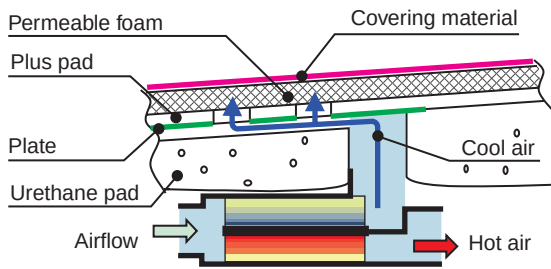


Fig. 10 Structure of seat covering



Fig. 11 Seat covering material (Honda bio-fabric)

4. Heating and Cooling Control

Figure 12 shows the configuration of the heating and cooling control system. Heating and cooling each have three operating modes, High, Mid, and Low. The seat surface target temperature is determined based on the

operating mode selected via the switch, and the amount of heat output by the Peltier module is controlled. Upper and lower limits have been set for the High modes in order to help prevent discomfort due to excessive heat when heating the seat and excessive cold when cooling the seat. The target temperatures for Mid and Low modes have been set to enable users to be aware of a difference between the modes.

4.1. Peltier Control

To achieve the heat output from the Peltier module, the seat surface target temperature, per the selected operating mode, is determined by information from temperature sensors in the seat surfaces and Peltier sensors, and is fed-back to the Peltier output. This helps to prevent overshooting or undershooting the seat surface temperature, and has reduced the time required to reach the target temperature.

4.2. Blower Control

The air flow is controlled using tables of motor voltage derived from operating mode and elapsed time. The control also compensates for the difference between the target temperature and seat surface temperature.

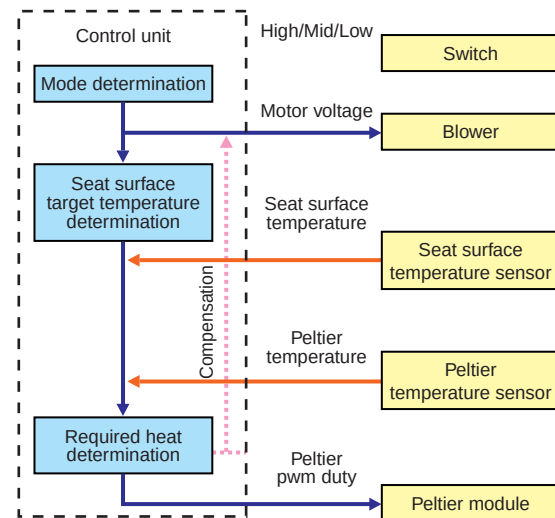


Fig. 12 Heating and cooling control system

5. Performance

The cool-down and heat-up characteristics of the new seats are shown in Figs. 13 and 14 respectively. These results show clearly that the new heated and cooled seats increase occupant comfort more rapidly than the previous ventilation seats or heated seats.

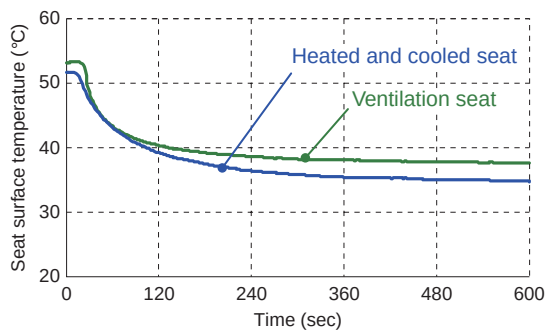


Fig. 13 Cool-down characteristic

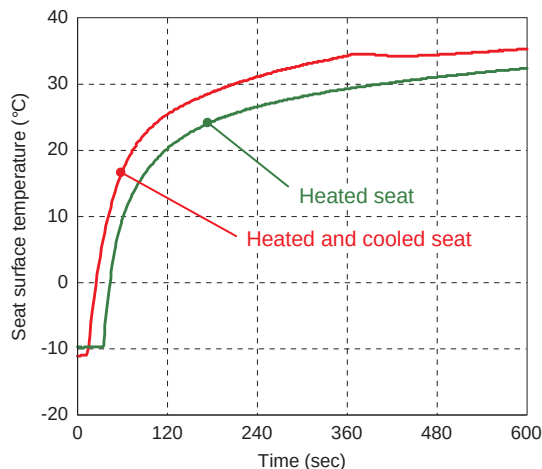


Fig. 14 Heat-up characteristic

6. Conclusion

Heated and cooled seats have been developed as a new means of increasing occupant comfort. The following results were obtained from this development project:

- (1) Use of the new seat heating and cooling system in combination with Honda's new bio-fabric material has enabled the realization in an environmentally-friendly manner of a comfortable texture that is appropriate for a fuel cell vehicle.
- (2) The use of Honda-developed control methods has enabled the achievement of more rapid heating and cooling performance than the previous ventilation seat or heated seat.
- (3) Efficient positioning of thin and lightweight thermo-control devices has enabled the seats to be provided with heating and cooling functions without compromising their basic functions or sacrificing customer appeal.

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PDU for FCX Clarity

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ABSTRACT

A Power Drive Unit (PDU) has been developed for the 2009 model year FCX Clarity that conforms to the “Dynamic Full-cabin Sedan” concept established for the vehicle.

In previous models, the PDU and the driving motor were mounted separately and were connected by exposed, waterproof high-voltage shielded cables for the distribution of electric power. The newly developed PDU has been integrated with the driving motor, and the three-phase power lines are connected internally. The new configuration was enabled by the use of FEM simulations in order to develop measures to counteract wobbling, the use of vibration tracking analysis to counteract motor vibration, the employment of cooling terminals to reduce heat transfer, and the adjustment of the level of electrostatic coupling between the control circuits and the PDU case in order to reduce electrical noise.

In addition, the size of the PDU itself has been reduced by the miniaturization of components and the use of a high-density layout.

As a result of the measures indicated above, the height of the PDU and the driving motor in the motor room has been reduced by 240 mm. This has contributed to the realization of the body design concept of the 2009 model year FCX Clarity.

1. Introduction

The global rise in environmental awareness today has brought further demand for vehicles with lower CO₂ as a measure against warming. Honda sought to meet these expectations by developing the Honda FCX fuel cell car as the ultimate environmentally-compliant vehicle. Leading the rest of the world, Honda delivered fuel cell vehicles to the Japanese government and to the city of Los Angeles in the United States in 2002⁽¹⁾. The New Honda FCX (hereafter referred to as the 2005 model year FCX)⁽²⁾ was then developed. In order to promote even more widespread use of fuel cell vehicles, this model was equipped with a stack manufactured by Honda for enhanced low-temperature performance. Delivered to the State of New York in 2004 and to Hokkaido in 2005, the 2005 model year FCX was the first practical fuel cell vehicle in the world to enable operation in temperatures below freezing⁽³⁾.

In July 2008, Honda brought to the market the 2009 model year FCX Clarity (hereafter the FCX Clarity) using its original fuel cell stack, the V Flow FC Stack, as the core technology⁽⁴⁾. This model was given an eye-catching design and revolutionary packaging to make it instantly

recognizable as a fuel cell vehicle because only a fuel cell vehicle could possibly convey that vivid a sense of the future. This model also realized a whole new dimension of drive feel, and the FCX Clarity delivered not only the ultimate in clean performance, but also the embodiment of new value and appeal as a car.

The FCX Clarity realized the body design concept of a Dynamic Full-Cabin Sedan that presents innovative packaging and powerful driving performance. This was accomplished by making the entire power plant more compact and placing it in an appropriate layout. This paper will describe the development of the PDU for the FCX Clarity.

2. Development Goals

Based on the concept of the Dynamic Full-Cabin Sedan, which is founded in Honda’s philosophy of Man Maximum and Mecha Minimum (M/M), the FCX Clarity adopted a short-nosed design with a compact motor room. The components inside the motor room had to be made more compact than in the 2005 model year FCX in order to realize this design. Limiting the height of the hood, in particular, was a crucial point on which the possibility of

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this design depended.

Therefore, the PDU for the FCX Clarity was developed with its size, together with the driving motor in the motor room reduced in the vertical dimension, in order to conform to the body design concept.

3. Integrated PDU and Driving Motor

The volume occupied by components in the motor room was reduced by adopting a structure that integrates the PDU with the driving motor and connects the three-phase power lines internally. The components were made compact and a high-density layout was adopted in order to reduce the size of the PDU itself. The purpose of this was to limit the increase in the space necessary to help prevent interference when wobbling occurs due to the integration of both components. These measures enabled the size of the combined PDU plus driving motor in the FCX Clarity to be reduced, as shown in Fig. 1, by 240 mm in the vertical dimension relative to the 2005 model year FCX. This contributed to realization of the short nose style.

Adoption of an integrated PDU and motor structure is effective as a method for achieving greater compactness. In order to realize this structure, however, measures had to be taken to address the following kinds of technical issues in the PDU:

- (1) Limiting deformation of the PDU case due to wobbling and absorbing the relative displacement of the PDU case from the motor case
- (2) Assuring the vibration durability of the PDU with respect to vibration originating in the motor
- (3) Limiting the rise of temperature inside the PDU due to heat transfer from the motor side
- (4) Enhancing the noise resistance of control systems with respect to the high-density layout of the internal components and the greater electrical coupling with noise sources

The technologies used to address these technical issues are introduced in the following.

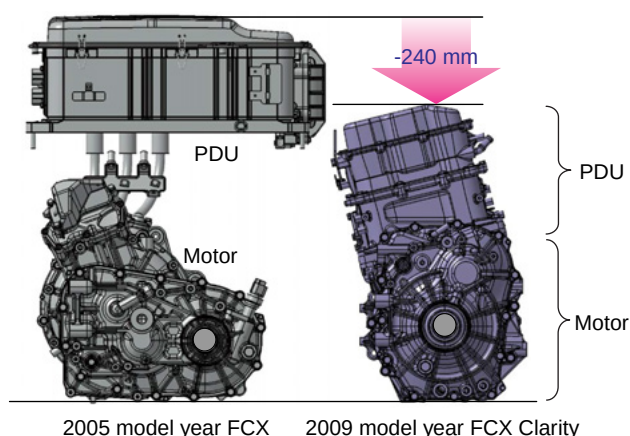


Fig. 1 Comparison of size (PDU and driving motor)

3.1. Measures to Counter Wobbling

The structure of the PDU integrated with the driving motor has the PDU and motor cases joined directly to each other with an O-ring sealing the join. The three-phase power lines are wired inside the cases. These measures enable the greater compactness. When the motor produces wobbling, a large moment is generated at the PDU case and the surfaces joined by the seal are deformed so that gaps open up. The PDU case is therefore made rigid in order to help prevent loss of the seal function.

The PDU and motor cases are further fastened together with bolts to form an integrated structure. The motor case and PDU case cannot, strictly speaking, be viewed as a single rigid body, and the connecting member for the three-phase power lines that link the two cases calls for a structure to absorb the relative displacement resulting from wobbling of the motor and PDU.

3.1.1. Rigid case design

The operating conditions under which the seal area of the PDU case undergoes maximum deformation when the PDU case is mounted in an actual vehicle were determined. An FEM analysis of strength and vibration was then carried out, as shown in Fig. 2, to help ensure that the size of the gap that opens up at the seal comes within the amount by which the O-ring is compressed even under those conditions. The figures obtained in the FEM analysis were used in deciding the position of the ribs on the case and the mounting boss, the thickness of the walls, and the location of the fastening, thus achieving increased rigidity in the case while limiting the increase in its weight.

3.1.2. Relative displacement absorbing structure of three-phase power lines

When wobbling occurs, differences in modes of

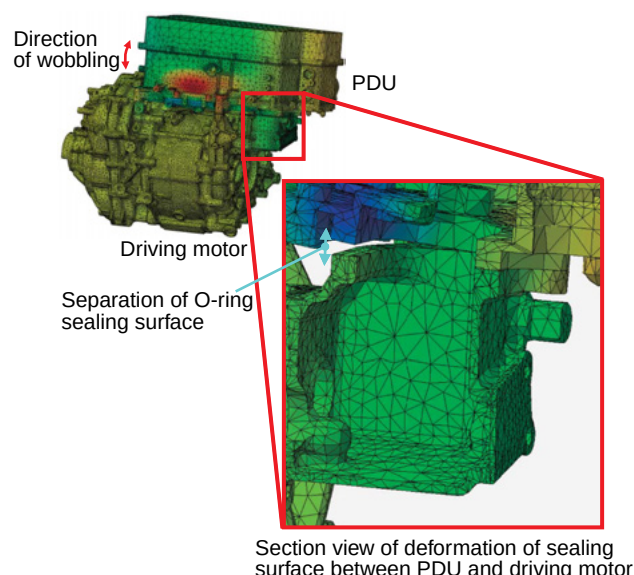


Fig. 2 Evaluation of deformation of PDU case

deformation and phase lag of response result in relative displacement, however slight, between the PDU and motor. Consequently, the three-phase power lines that form the connection between the power module mounted on the PDU and the windings affixed to the motor should have a mechanism to absorb the relative displacement between these systems produced by wobbling.

A structure as shown in Fig. 3 was adopted, therefore, in which terminals are used to fasten the bus bar from the power module to the PDU case and to fasten the motor windings to the motor case, respectively. These terminals are then joined to each other by a short high-voltage cable. This structure enables the relative displacement between the PDU and motor to be absorbed without causing the bus bar and motor windings to undergo deformation stress from wobbling.

The integration of the PDU with the motor brings the terminals on the motor and PDU sides closer together. As a result, the cables connecting the two components need to be configured into a space-saving layout. With the conventional cable structure, however, the cable has a larger diameter because of the limits on the cable's heat resistance temperature, and so a larger layout space is necessary. The following measures were therefore taken to reduce the amount of temperature increase in the cable and to raise the heat resistance temperature of the cable so that the wire diameter could be reduced and a space-saving layout realized.

- (1) Active cooling structure on the PDU terminal and motor terminal
- (2) Active cooling structure for the bus bar inside the power module
- (3) Increased cable heat resistance (heat resistance temperature raised from 120° C to 150° C)

3.2. Measures to Counter Motor Vibration

When the PDU is to be mounted on the vehicle body, as it was in the 2005 model year FCX, the frequency band

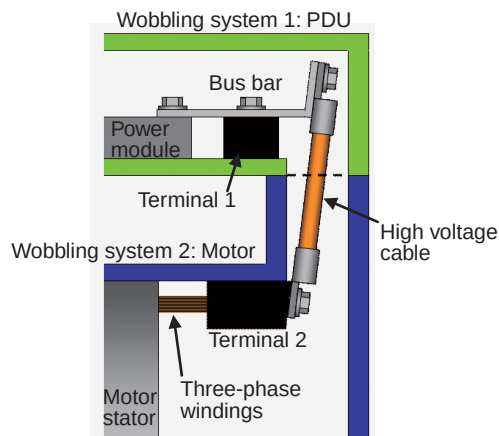


Fig. 3 Configuration of three-phase power line connection

of the vibration applied to the PDU is the same as for ordinary vehicle components. The vibration durability can be verified by the same methods used for those components.

The PDU for the FCX Clarity, however, is rigidly mounted on the driving motor. Consequently, the vibration generated by the rotation of the motor is applied directly to the PDU by means of the connecting member.

3.2.1. Motor vibration characteristics

In the case of the FCX Clarity driving motor, the most conspicuous source of motor vibration occurs when the stator deforms due to magnetic excitation of the windings. There are pluralities of modes (revolution orders) in stator deformation, depending on the slot shape and the number of electrode pairs. These states are always generated simultaneously at frequencies that are in proportion to the motor speed, so that the motor vibration can be measured as their compositional vibration.

The vibration acceleration of each mode is in proportion to the product of the vibromotive force (the magnitude of stator deformation produced by magnetic excitation) and the vibration response gain of the motor structure in the excitation frequency. Therefore, the characteristics of motor vibration, which are the sum of the vibration modes, vary in a complex manner with the changes in the motor speed and generated torque.

Figure 4 shows the results of tracking analysis of vibration characteristics in the FCX Clarity driving motor at maximum generated torque.

This graph suggests the following:

- (1) There is a specific frequency band in which the motor vibration reaches higher levels.
- (2) There is a specific range of motor speeds in which the motor vibration reaches higher levels.
- (3) When a specific mode (revolution order) line impinges on the above range, the magnitude of motor vibration becomes conspicuous.

The PDU was designed in light of these characteristics relating to motor vibration.

3.2.2. Vibration-proofing design

The design was created taking into account the

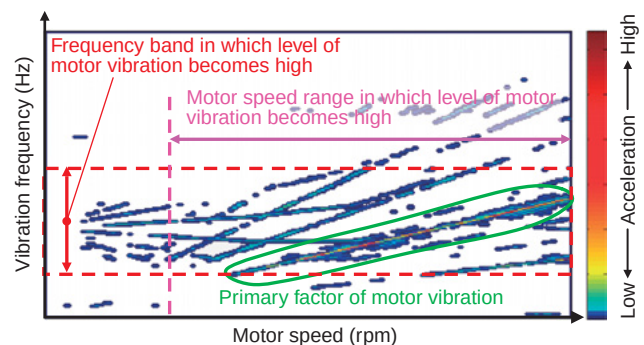


Fig. 4 Result of tracking analysis of motor vibration

methods used to constrain the various components and the configurations defining their resonance characteristics. The purpose was to help prevent any major increase in the relative displacement of the component parts due to vibration in the frequency band where motor vibration reaches a high level, based on the results from the analysis of motor vibration.

- (1) Limit relative displacement by regulating resonance frequencies: Determine the shapes of components and select the materials used in them so as to remove resonance frequencies from the frequency band at which motor vibration reaches a high level.
- (2) Fix in place with multipoint constraints: The interval between fastening points is made small for components that have a large surface area, such as circuit boards, in order to limit displacement of the component.
- (3) Vibration isolation using rubber mounts: Use rubber mounts for components for which the measures in items (1) and (2) are insufficient, in order to isolate them from motor vibration.

Figure 5 shows the distribution of resonance frequencies for the PDU parts. These frequencies were determined for every part by examining the resonance points of those parts for which vibration acceleration can be measured directly, and by calculating them from the materials and shape of other, smaller parts.

As this graph indicates, chips and other such small parts are at the high range of resonance frequencies and so are not directly affected by motor vibration. The resonance points on other parts were successfully kept out of the frequency band at which motor vibration reaches high levels.

3.2.3. Verification of vibration durability

Although the vibromotive force is small, there are components that are affected by vibratory excitation at resonance points on the order lines with the changes in the motor speed. The vibrational durability of these components was therefore verified using the procedure shown in Fig. 6.

(1) Vibration characteristics: A frequency sweep was performed on the reference sine wave input vibration and the vibration response characteristics of the PDU

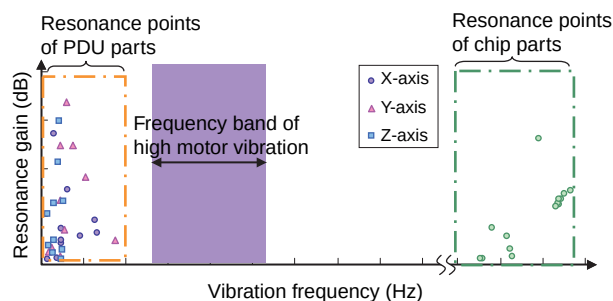


Fig. 5 Distribution of resonance frequencies of PDU parts

components were measured. The resonance frequencies for each component as well as the resonance amplification were thus identified. Further frequency analysis of the measurement results was conducted to determine the transfer function of vibration (the frequency characteristics of gain and phase) from the point of application of the vibration to the PDU to each component.

(2) Allowable level of relative displacement: The vibration characteristics for each component determined in (1) were used to conduct vibration durability tests at a high level of input vibration to determine the allowable level of relative displacement⁽⁵⁾ at which the component would not be damaged. In order to selectively excite the object component, the test frequency was set at the resonance frequency of that component, but the allowable level of relative displacement is not dependent on the frequency (Fig. 7).

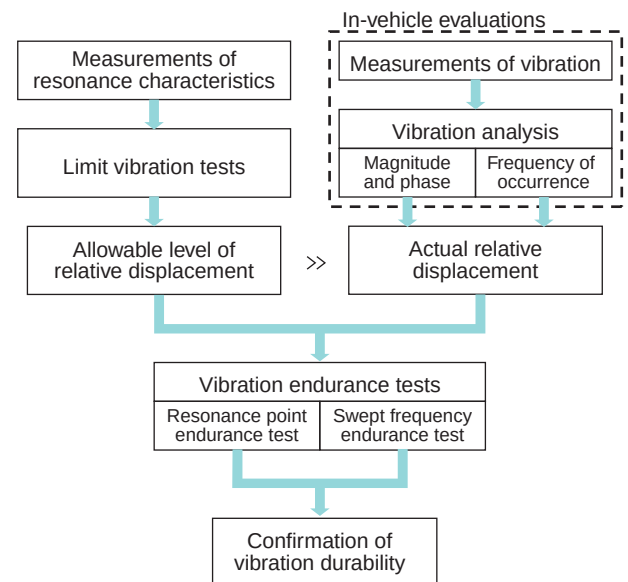


Fig. 6 Procedure for confirmation of vibration durability

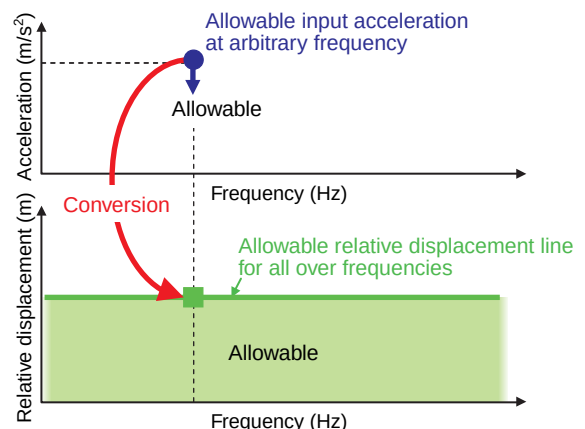


Fig. 7 Allowable relative displacement

The fact that the relative displacement of a component is at or below the allowable level means that the component is being used at a range of stress that is lower than the fatigue limit of the component material.

(3) Vibration when vehicle is in motion: Frequency analysis was conducted of the motor vibration that is applied to the PDU when a completed vehicle is in motion. This was done to confirm the vibrational state of each component (the magnitude of vibration acceleration and the frequency of occurrence for each frequency) by the vibration input in a completed vehicle, as well as the conditions of vibration under which relative displacement rises to its maximum in connected components. In this way, it was confirmed that the obtained levels were lower than the allowable level of relative displacement determined in item (2). Figure 8 shows examples of the confirmation results.

The conditions under which relative displacement reaches its maximum level are determined from the frequency characteristics of gain and phase for the vibration of each component (response) with respect to the motor vibration (input). The vibration of a given component in a certain direction can be viewed, when simplified, as a single-degree-of-freedom forced vibration by displacement such as shown in Fig. 9.

When the base (the component that is the source of excitation) vibrates in a vertical direction such that the relative displacement with respect to the base is $x(t)$, then

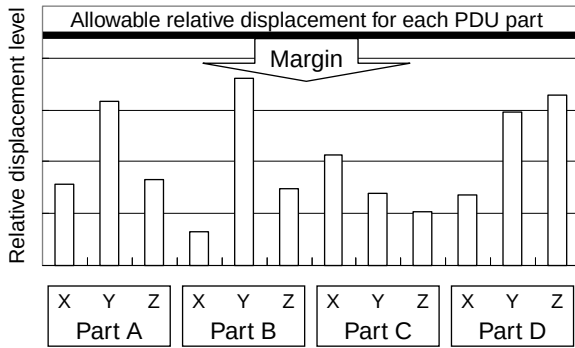


Fig. 8 Maximum relative displacement levels of PDU parts measured in vehicle

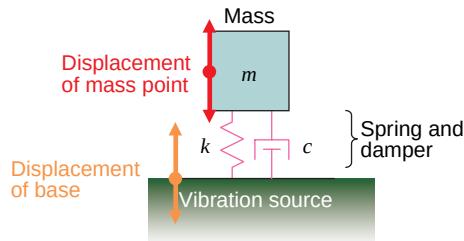


Fig. 9 Single-degree-of-freedom vibration by displacement

the equation of motion for the mass point (the component that is subject to vibration) will be (1) below, where m is the mass of the mass point, a is the amplitude of displacement of the vibration source (the component that is the source of excitation), k is the spring constant, c is the attenuation factor, and ω is the angular velocity:

$$m\ddot{x} + c\dot{x} + kx = ma\omega^2 \sin\omega t \quad (1)$$

The steady-state vibration can be expressed by:

$$x = x_{\max} \sin(\omega t - \gamma) \quad (2)$$

In this case, the magnification of the relative displacement of the vibration source and mass point and the phase lag angle can be expressed by the following equations:

$$\frac{x_{\max}}{a} = \frac{(\omega/\omega_0)^2}{\sqrt{\{1-(\omega/\omega_0)^2\}^2 + (2\xi \cdot \omega/\omega_0)^2}} \quad (3)$$

$$\gamma = \tan^{-1} \frac{2\xi \cdot \omega/\omega_0}{1-(\omega/\omega_0)^2} \quad 0 \leq \gamma \leq \pi \quad (4)$$

Figure 10 shows examples of the frequency characteristics of relative displacement and phase lag angle derived from Eqs. (3) and (4) when the attenuation ratio $\xi = 0.015^{(6), (7)}$.

The magnification of the relative displacement with respect to the input displacement reaches a maximum at the resonance condition of the component. The magnitude of the relative displacement is determined by the response gain, the phase difference, and the magnitude of the input vibration. When a large input vibration is generated at a frequency other than the resonance frequency, therefore, the conditions under which the relative displacement reaches its maximum will not necessarily coincide with the resonance conditions. As shown by the example in

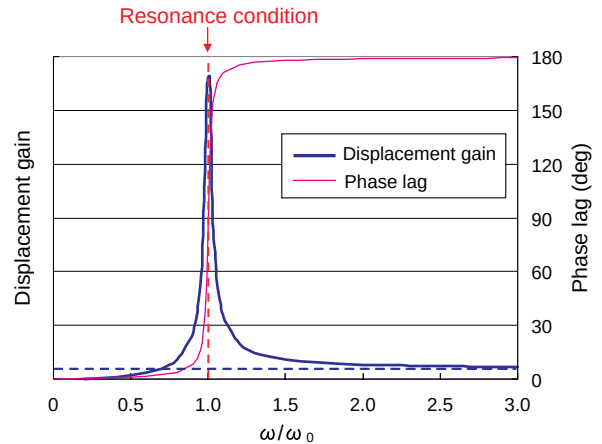


Fig. 10 Example of frequency characteristics of displacement gain and phase lag

Fig. 11, the results from measurement of relative displacement made with this PDU indicate that these conditions do coincide with the resonance conditions of the components that undergo excitation by motor vibration.

(4)Durability testing: In order to demonstrate the vibration durability of each component, the following durability testing was performed, thereby confirming that the components were undamaged:

(a) Resonance endurance test

The relative displacement of every part was analyzed from the vibration acceleration that occurred with a completed vehicle in motion and vibration conditions (the frequency and acceleration of vibration input to the resonance point) under which the relative displacement with respect to connected components reaches a maximum were identified. Resonance endurance testing was performed for every part based on the results of the above, and it was confirmed that no damage occurs.

(b) Sweep vibration endurance test

An analysis was made of the frequency of occurrence of vibration when a completed vehicle was driven in various modes. The results were used to configure a sweep vibration test profile that simulates operating life in terms of the kind of frequency, acceleration, and duration of excitation for each component shown in Fig. 12. A vibration testing device was used to conduct sweep

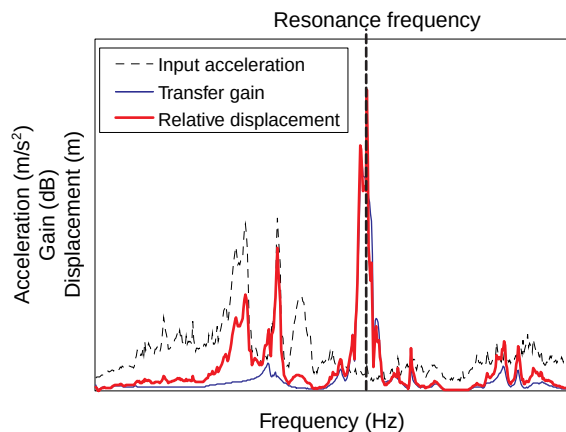


Fig. 11 Example of relative displacement characteristic

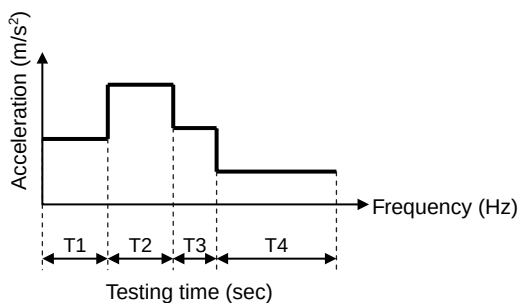


Fig. 12 Vibration profile for sweep vibration endurance test

vibration endurance tests accordingly, and it was confirmed that the components were not damaged.

3.3. Measures to Counter Heat Transfer

The three-phase windings in the driving motor reach higher temperatures than the internal components of the PDU. When the PDU and the driving motor were separate structures, the heat in the windings was released into the outside air while being conducted through the power cable connecting the two structures. Consequently, there was little impact on the PDU from heat receiving. When the PDU and the motor are integrated in a single structure, however, the three-phase power lines are housed inside the case and the wiring is reduced to the shortest length. With this layout, the heat of the windings flows into the PDU side so that the temperature of internal PDU components rises. The FCX Clarity is designed to reduce this transfer of heat from the windings while also reducing the cables to a smaller diameter, as explained earlier, by the installation in the PDU of a junction terminal incorporating a heat transfer structure as shown in Fig. 13.

The heat coming from the three-phase power lines is passed through the insulating resin and aluminum block and released on the heat sink side. As a result, the temperature rise in the block portion of the terminal due to heat receiving from the windings could be reduced by approximately 11°C, as shown in Fig. 14.

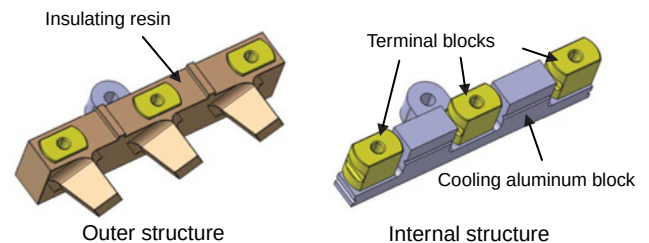


Fig. 13 Structure of cooling terminal for PDU

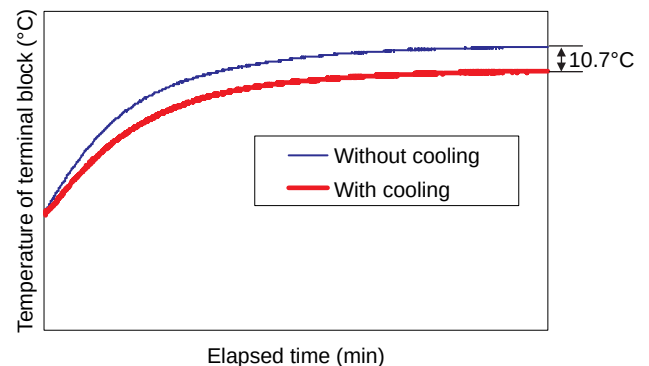


Fig. 14 Effect of cooling terminal for PDU

As shown in Fig. 15, the bus bar inside the power module and the terminal on the motor side also have cooling devices built in. This reduces the heat transfer from the power elements and motor.

3.4. Measures to Counter Noise

Electrically floating configuration is used to isolate the high-voltage power lines in the FCX Clarity from the body, and the interface between the PDU and outside circuits is entirely a low-voltage system, with the exception of the high-voltage power cables. The PDU consists of a power circuit section, which performs high-power switching using semiconductors, and a control circuit section, which functions as an interface between the inverter switching control and external devices. These are located inside the same metal case (Fig. 16).

In order to help prevent malfunctions of the inverter circuits and control circuits due to noise and to strengthen tolerance against EMI, the high- and low-voltage circuits inside the same case should be kept separated as far as possible. This goal, however, is incompatible with the use of a high-density layout. Consequently, the geometry of the high-voltage wiring assembly, its distance to the control board, and its location were optimized in order to limit the effects on the PDU control board produced by conductive and radiation noise from the high-voltage wiring, as well as to enhance the high-density layout. This

achieved a balance between the use of a high-density layout and the measures against noise.

In the motor, a stray impedance is present between the motor windings and housing, and, for structural reasons, a flow of noise currents is generated during inverter switching. These currents conduct noise throughout the entire vehicle and may sometimes affect the quality of the vehicle as a commodity because of radio noise, malfunctions in other components, and the like. Optimization was therefore performed with a line bypass capacitor used inside the PDU to feed back the noise currents and limit the influence of noise on the vehicle.

Another consideration is that when these feedback currents travel through the PDU case, they may sometimes cause malfunctions in the control boards through stray capacitance between the frame around the printed circuit board (PCB) that serves as the control board inside the PDU case and the copper foil patterns in the PCB's inner layers. One example of the means used to address this issue, as shown in Fig. 17, is to maintain the distance between the wiring patterns inside the circuit board and the rim of the metal frame that holds the board by keeping the two from overlapping. This structure keeps the stray capacitance between the two at or below a certain level, and thereby optimizes the noise level.

These measures succeeded in reducing noise levels. Figure 18 shows the results.

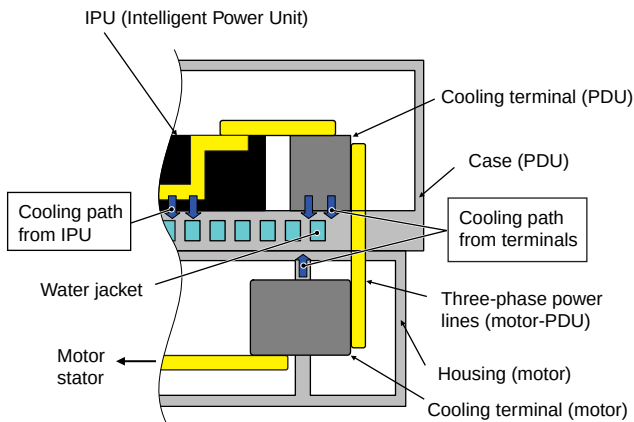


Fig. 15 Structure of cooling system for three-phase power line

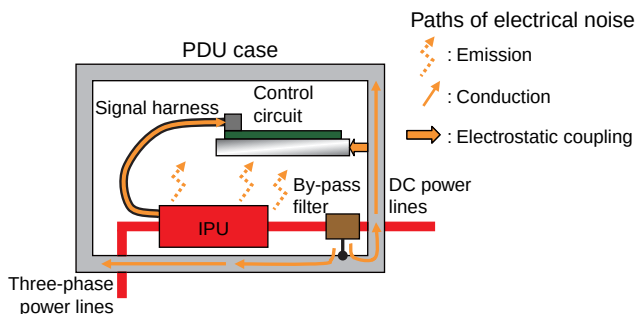


Fig. 16 Noise environment of PDU

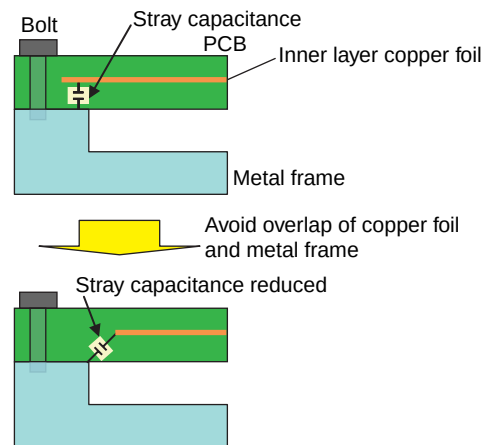


Fig. 17 Reduction of stray capacitance between inner layer copper foil and metal frame

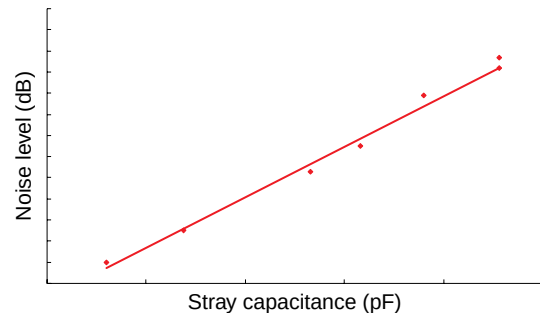


Fig. 18 Effect of reduction of stray capacitance

4. Conclusion

A PDU integrated with the driving motor was developed in order to limit the size of the integrated structure in the vertical direction inside the motor room.

- (1) The high-voltage cables are laid out to make the shortest possible connections inside the PDU case, thereby realizing a compact connecting structure for the three-phase power lines that also includes a mechanism to absorb the relative displacement between the motor and PDU when wobbling occurs.
- (2) Vibration-proofing design was carried out according to the results from analysis of motor vibration characteristics, thereby assuring the durability of the PDU against motor vibration.
- (3) Terminals incorporating a cooling structure enable heat from the three-phase power lines to escape to the heat sink side, thereby reducing the increase in temperature by components inside the PDU due to heat receiving from the motor windings.
- (4) The geometry of the high-voltage wiring assembly, its installed position, the position of the control board, the capacity of the line bypass capacitor, and the stray capacitance between the pattern wiring of the control circuit PCB and the frame, were optimized. These measures limited the effects on the control board and vehicle body from noise due to conduction, radiation, and electrostatic coupling, thereby enabling realization of a high-density component layout.

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Development of Lithium Ion Battery System for Fuel Cell Vehicle

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ABSTRACT

A lithium ion battery system was developed for the fuel cell vehicle Honda FCX Clarity.

This system achieves output power better than the ultra-capacitor mounted in the previous Honda FCX, and enhances both starting acceleration and fuel economy performance by increasing the energy recovery rate. In addition, durability and reliability requirements are also satisfied by performing appropriate battery cooling and control.

Regarding mounting in a vehicle, this system is 40% lighter and 50% smaller by volume than the previous Honda FCX system, and mounting the system under the vehicle floor results in expanded cabin space, which increases vehicle appeal. In addition, the new system also realizes increased resistance to environmental factors such as water and dust.

1. Introduction

Increased attention is being placed on environmental measures, and there are high expectations for fuel cell vehicles as eco-friendlier vehicles. As such, energy storage with the same high performance as fuel cells is demanded to enhance the fuel economy and acceleration performance of fuel cell vehicles. The Honda FCX⁽¹⁾, the world's first fuel cell vehicle certified for commercial use by the U.S. government, was delivered to its customers in December 2002. It employed an ultra-capacitor^{(2), (3)} that realized acceleration performance exceeding that of hybrid vehicles. At the same time it was also understood that increasing the energy capacity and energy recovery rate would lead to further enhancements in fuel economy.

The FCX Clarity is the first Honda vehicle to use a lithium ion (hereafter, "Li-ion") battery to enhance fuel economy without impairing the conventional acceleration performance. A Li-ion battery system was mounted in the vehicle including cooling and control that aims to maximize battery performance.

When mounting the battery system in the vehicle, placement under the vehicle floor expanded the cabin and trunk room space, which increased product appeal. On the other hand, it increased its exposure to environmental factors outside the cabin, such that protecting the high voltage parts from water and dust needed to be addressed.

This paper reports the Li-ion battery performance and

durability, and describes the technologies used to address environmental factors outside the cabin, when mounting the Li-ion battery system in an actual vehicle.

2. Adoption of Lithium Ion Batteries

The two main performance features required of the energy storage mounted in the FCX Clarity were output power density, which is related to acceleration performance, and energy density, which is related to fuel economy. Figure 1 shows a chart comparing the energy density and output power density of various energy storage types.

The Li-ion batteries used in consumer products drive those products for long times, so they have a high energy density. On the other hand, ultra-capacitors have low internal resistance, which gives them reliable output power density. This development used a Li-ion battery with low internal resistance that features both reliable output power density and energy density as the energy storage for realizing the required performance. This Li-ion battery uses different positive and negative-electrode materials and a different collector structure than those of Li-ion batteries for consumer products. This achieves an output power density greater than that of an ultra-capacitor and an energy density greater than that of the nickel-metal-hydride batteries used in hybrid vehicles. As a result, the battery output power is approximately three times that of

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a hybrid vehicle. In addition, the new battery has higher energy than the ultra-capacitor used in the previous model, which extends the time that the acceleration state can be maintained, increasing the acceleration time from vehicle launch by 40%. The energy level has been increased and a cooperative regenerative braking system that controls the division between generation and braking is also used, which increases the amount of regenerative energy recovered during deceleration by 13% in 10-15 mode, leading to enhanced fuel economy. Furthermore, the new Li-ion battery has a high rated voltage per battery unit approximately three times that of a nickel-metal-hydrate battery, and approximately 1.5 times that of an ultra-capacitor. This means that fewer Li-ion batteries are needed to produce the same system voltage, which enables a more compact size. In addition, there is no memory accumulation like with a nickel-metal-hydrate battery, and the charging status can be understood from the voltage, so the control system can also be simplified.

For these reasons, a Li-ion battery with both reliable energy density and high output power density was adopted as the energy storage for the FCX Clarity.

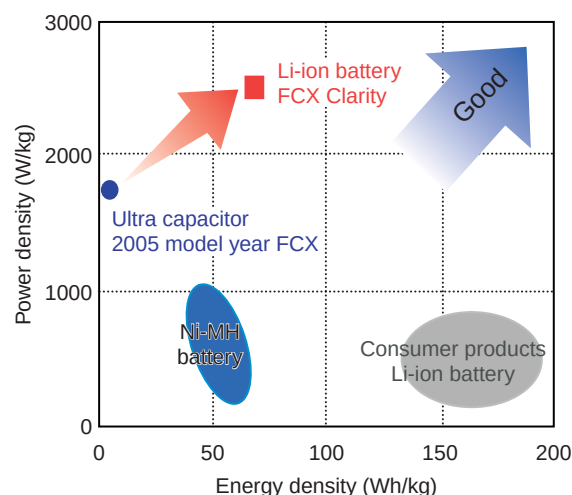


Fig. 1 Battery performance comparison

3. System Goals

Development of the Li-ion battery system focused on the following two goals:

- (1) Establishing battery durability
- (2) Achieving a full-cabin sedan style

Establishing battery durability is important for maintaining vehicle performance over long periods using multiple batteries. This makes it necessary to construct a system that includes cooling so that the multiple batteries are always placed under the same environment in order to increase durability.

In addition, development emphasized a compact and

lightweight Li-ion battery system to enable mounting in a manner that achieved the full-cabin sedan style that is a feature of the FCX Clarity. The goal was to enhance both comfort and mounting performance, so the Li-ion battery system was mounted under the floor of the rear seat, allowing even more cabin and trunk space.

4. Battery Durability

4.1. Factors and Estimation of Drops in Performance

The effects of the state of charge (hereafter, "SOC") and temperature produce a drop in the Li-ion battery capacity and a rise in resistance, leading to a drop in performance. In particular, this drop in performance progresses faster when used continuously under high SOC and high temperature conditions. Figure 2 shows the change in capacity at different environmental temperatures in a charge-discharge cycle test. (The horizontal axes in Figs. 2 through 7 and Fig. 9 hereafter show the time with the durable life of the FCX Clarity defined as "1".) These results show that the drop in capacity increases when the test is performed under high environmental temperature conditions.

Next, Fig. 3 shows the change in capacity in a shelf test. As seen in the charge-discharge cycle test, the drop

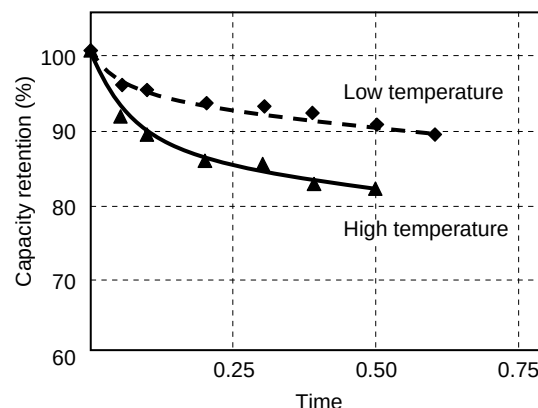


Fig. 2 Capacity retention in charge-discharge test

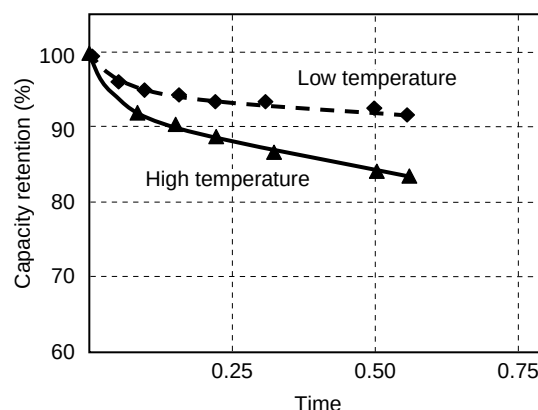


Fig. 3 Capacity retention in shelf test

in capacity increases when left under high environmental temperature conditions. This difference in the progression of performance drop due to differences in the environmental temperature occurs not only for capacity; the same phenomenon also occurs for resistance (Fig. 4). In addition, Fig. 5 shows that even at the same environmental temperature, the resistance changes according to the SOC.

In this manner, the change in Li-ion battery performance is influenced by the battery state, including the temperature and SOC, and the rate of change can be arranged using the 1/2 power-law of time⁽⁴⁾. Figures 6 and 7 show examples of shelf tests. A performance drop coefficient can be calculated from the results of tests performed under different temperature and SOC conditions, and that the coefficient can be used to estimate the change in Li-ion battery performance during the vehicle operation period.

The drop in performance progresses regardless of whether the Li-ion battery is performing charge-discharge or in the shelved state, so acceleration tests can be performed by calculating the performance drop coefficient and setting various conditions. This ability to understand changes in performance occurring over long-term vehicle operation in a short time is useful for development.

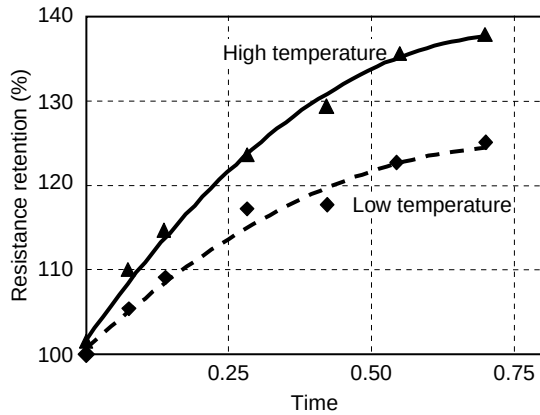


Fig. 4 Resistance retention in charge-discharge test

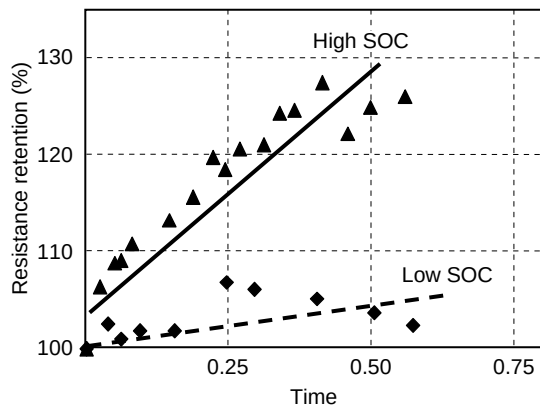


Fig. 5 Resistance retention in shelf test

4.2. Operating Temperature and SOC Range Settings

The battery operating temperature range and SOC operating range should be determined in consideration of various thermal effects, such as the environmental temperature of the region where the vehicle is operated, radiated heat from the road surface, battery heat generation due to current input/output while running. Environmental temperatures were assumed to range from an outside air temperature of 45°C in regions with intense heat to -30°C in regions with extreme cold. Battery heat generation due to current input/output varies according to the running conditions, so past running data for gasoline and hybrid vehicles as well as running data for the predecessor model were taken into account as operational data. These data were combined and evaluation was repeatedly performed to determine the battery operating temperature and SOC ranges over which the required performance could be satisfied for the vehicle operation period.

4.3. Cooling System Structure and Performance

Simulations were repeatedly performed to design a cooling structure that could efficiently cool all the

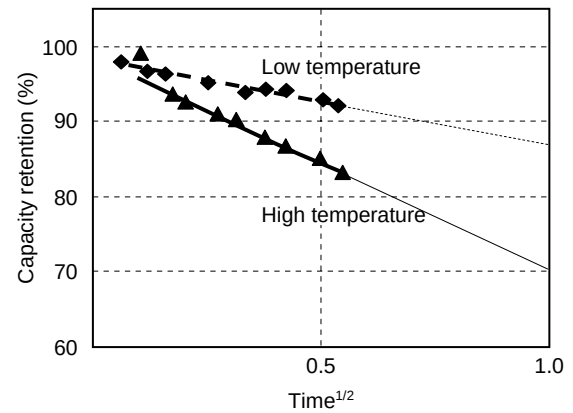


Fig. 6 Capacity retention in shelf test (acceleration test)

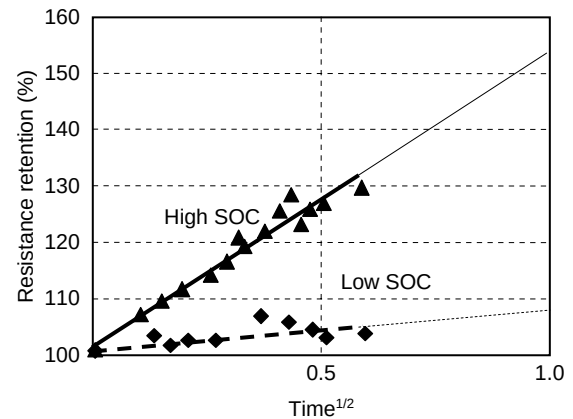


Fig. 7 Resistance retention in shelf test (acceleration test)

batteries. In contrast to the capacitor system of the previous model that was divided into two boxes and performed cooling for each box, the new system places all batteries into a single box for cooling, which enables a more compact size. Figure 8 shows the cooling structure. An air-cooled side flow format was selected, where external air sucked in from the top of the intake duct is divided inside the duct to cool from the right and left sides of the battery box. In addition, an exhaust duct was installed inside the battery frame. Side plates that serve as structural members were installed on the sides of the box, and openings were made in these plates as entrances for outside air. An opening was also set in the center of the battery frame as an exhaust port. Battery temperature variance is reduced by adjusting the area of these openings to adjust the air flow. In addition, the design divided the inside of the box into front and rear

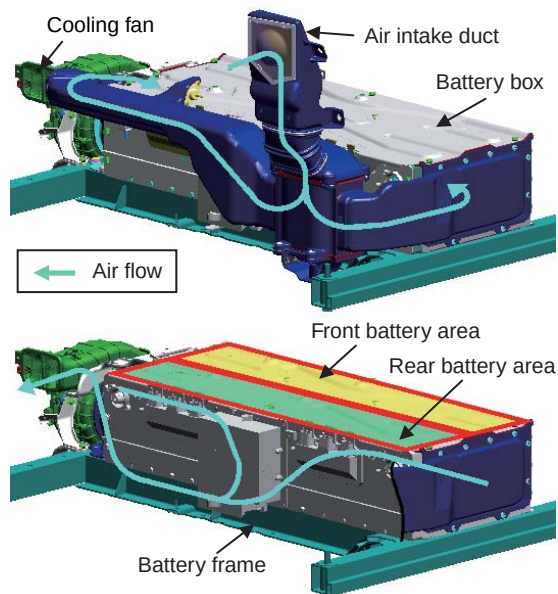


Fig. 8 Battery cooling air flow

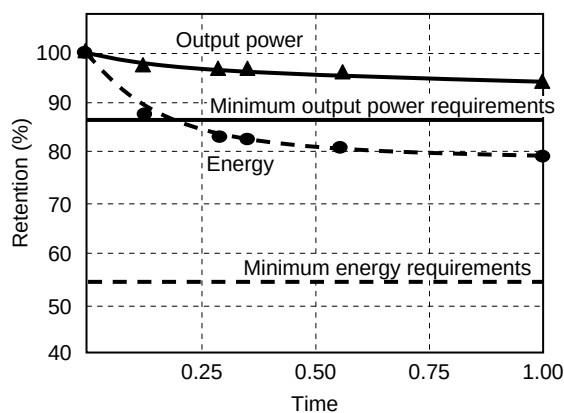


Fig. 9 Life test of Li-ion battery system

portions, and the front battery temperature was set a few degrees higher than the rear battery temperature. This enabled measurement with emphasis on the front battery temperature, which helped to prevent the increase of thermistors for detecting the battery temperature. Figure 9 shows the life test results using the set SOC operating range and new cooling structure. This confirmed that both the energy and output power satisfy the required performance levels for the assumed operation method over the number of operation years.

5. Features of the Vehicle Package

Figure 10 shows the mounting structure of the Li-ion battery system. This system is mounted under the floor of the rear seat, and consists of a box enclosing the battery units in the center, a battery ECU on top, a cooling fan on the side, and a contactor box in the rear.

As mentioned above, air cooling was selected as the cooling method, and a side flow format that reduces the structural height was used to secure a wide cabin space in order to achieve both high performance and light weight. This realized a weight reduction of 40% and a volume reduction of 50% compared to the previous model.

On the other hand, placing the energy storage outside the cabin made it important that consideration be given to external factors such as water and dust, and this became a technical issue in this development.

5.1. Structure with the Battery Module Placed outside the Cabin

An important point for realizing a structure that places the battery module outside the cabin with external air intake and exhaust is measures to help prevent the entry of water. This is because the batteries are constantly high voltage, and water entering may eventually lead to a

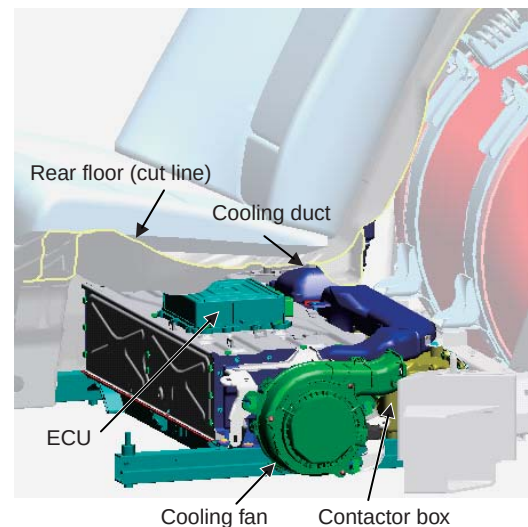


Fig. 10 Battery module placement

battery short circuit. The selection of external air intake and exhaust means that the cooling air itself contains moisture, so each individual battery was waterproofed. In this development, the batteries were protected by a resin covering that not only secured waterproofing and insulation performance, but at the same time enabled direct exposure to the cooling air. The terminal structure combines surface sealing using synthetic rubber with outstanding weather resistance, and adhesive sealing, to secure reliability. The battery ECU and cooling fan electrical parts are waterproofed by applying the adhesive sealing technology used for the ECU of mass production gasoline vehicles. This achieved the high reliability of being able to provide insulation performance even when the entire battery module is submerged. However, for these structures to maintain stable quality over long periods, consideration should be given to foreign matter and dust in addition to water.

Dust and foreign matter countermeasures are basically divided into two parts: a structure that helps prevent the entry of dead leaves and other large foreign matter, and a structure that catches dust and other fine particles. The former uses a mesh over the air intake, and the latter catches and helps prevent the entry of foreign matter and dust using an air filter. The air intake to which the mesh is attached faces the rear, or the direction opposite of vehicle travel, and is located at a position above the battery module. This is approximately the same height as the air intake of mass production vehicles, and helps prevent the entry of foreign matter and water flung up when running.

Figure 11 shows the intake structure.

The air filter catching ability was set in consideration

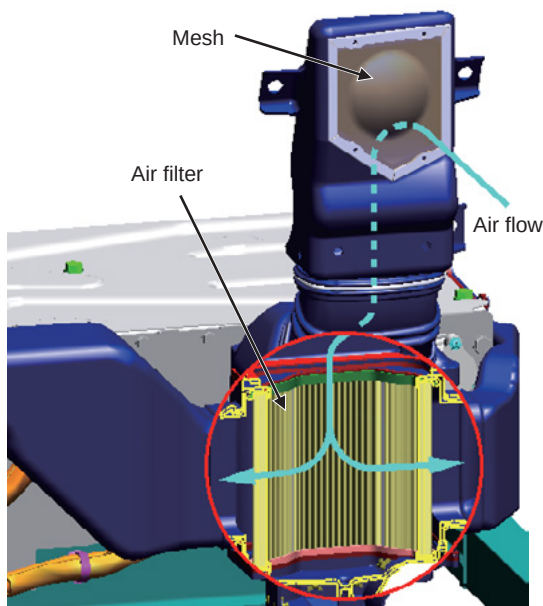


Fig. 11 Cooling duct structure

of the balance with the cooling capacity. As mentioned above, the battery's waterproof structure also gives it a certain degree of dust resistance. However, attention was given to the following three points as the effects of dust.

- (1) Short circuits resulting from potential differences and moisture-laden dust (tracking)
- (2) Structural erosion resulting from collisions between dust and the batteries (erosion)
- (3) Structural corrosion resulting from corrosive products (corrosion)

Figure 12 shows the relationship between dust catching ability and the effects of these three points for various dust particle sizes.

When addressing tracking which may lead to dielectric breakdown, pollution level 3 specified in JIS C0704⁽⁵⁾ – an environment in which conductive pollution exists – was assumed in consideration of placement outside the cabin. Tracking resistance was determined to be a potential difference of 2 000 V or more to take measures against voltage spike noise. In order to achieve this tracking resistance, a minimum distance between terminals of 16.5 mm was provided, and a comparative tracking index, which indicates tracking resistance performance, of 600 or more was maintained by examining resin materials.

Erosion was determined by calculating the impact value from the duct length, wind flow and other factors based on the Kanto loam formation (JIS Z8901 TYPE 8)⁽⁶⁾, and the results showed there to be no problem with a particle diameter of 30 μm or less.

From a broad group of potentially corrosive dusts, ranging from mainly inorganic dusts such as Kanto loam formation to organic dusts such as pollen and spores, emphasis was placed on corrosion due to the effects of organic dust and moisture, and the like.

Pollen, spores and other small particles generally have a diameter of approximately 10 μm , and a particle diameter of 30 μm or less presents no issues in terms of erosion, so an air filter with dust catching ability of approximately 10 μm was selected. Corrosion resistance

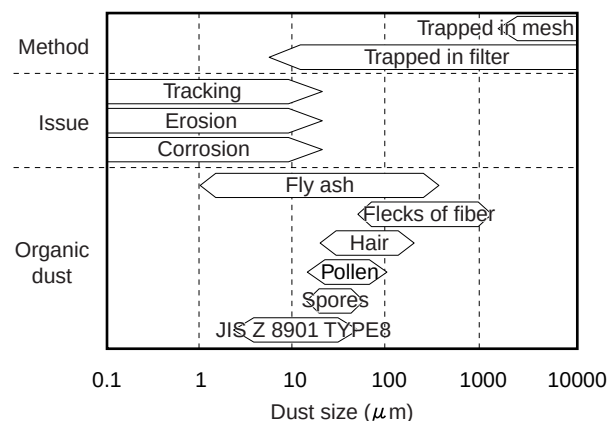


Fig. 12 Dust catching ability and issues

with respect to the entry of dust particles 10 μm and smaller was secured by employing resin materials with outstanding environmental resistance, which are used on the exterior of the vehicle and inside the engine room, in the cooling channel. This waterproof, dustproof structure enabled placement outside the cabin.

6. Conclusion

This paper introduced the Li-ion battery system developed for the FCX Clarity. The conclusions reached as a result of this system development are as follows.

- Li-ion batteries have high energy density and high output power compared to other forms of energy storage.
- Changes in battery performance can be estimated, and it was confirmed that this new system can supply stable energy over the vehicle operation period by setting durability modes and constructing a cooling system based on market usage.
- Addressing environmental factors in consideration of placement outside the cabin gives this Li-ion battery system increased mounting freedom and makes it possible to enhance cabin comfort.

In this manner the Li-ion battery system mounted in the FCX Clarity features outstanding performance and reliability. This contributes to continuously extendable acceleration with powerful torque and a spacious cabin interior, which increases the appeal of the FCX Clarity.

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Development of Friction Stir Welding Technology for Subframe for Fuel Cell System

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ABSTRACT

In order to save weight and increase productivity, the FC subframe developed for vehicle mounting of the fuel cell system employed in the FCX Clarity fuel cell vehicle is constructed from both extruded and cut parts produced from 6000 series aluminum alloy. Friction stir welding (FSW) was used to weld the main structural parts of the subframe. FSW was applied to 12 types of lap joints and one type of butt joint. Five tools were developed for the lap joints and one tool for the butt joints. The strength of lap joints formed using FSW can be diminished by the infiltration of the material in the bottom sheet into the material in the top sheet, a phenomenon referred to as “hook.” The project discussed here clarified the mechanism of this phenomenon, and reduced its occurrence by forming grooves on the probes of the welding tools. In addition, in the case of butt welds of forms with radii, the occurrence of elongated cavity was controlled without the use of filler metal by having the center of the tool approach the thickest side of the material.

The use of these methods has enabled the achievement of a 17% weight saving against a conventional subframe using stamped parts.

1. Introduction

Friction stir welding (FSW) involves a lower heat input than arc welding (such as MIG welding and TIG welding) and other types of fusion welding, and enables reduced distortion. Although the method has been applied mainly to railway rolling stock, instances of use in

automotive parts⁽¹⁾ have been limited to suspension arms, hoods, tailgates, doors, and other components that open and close. The application of FSW on components with a hollow cross-section, such as the subframe, has not advanced. This is because the heavy load applied during joining can deform the workpiece and end up producing joining imperfections.

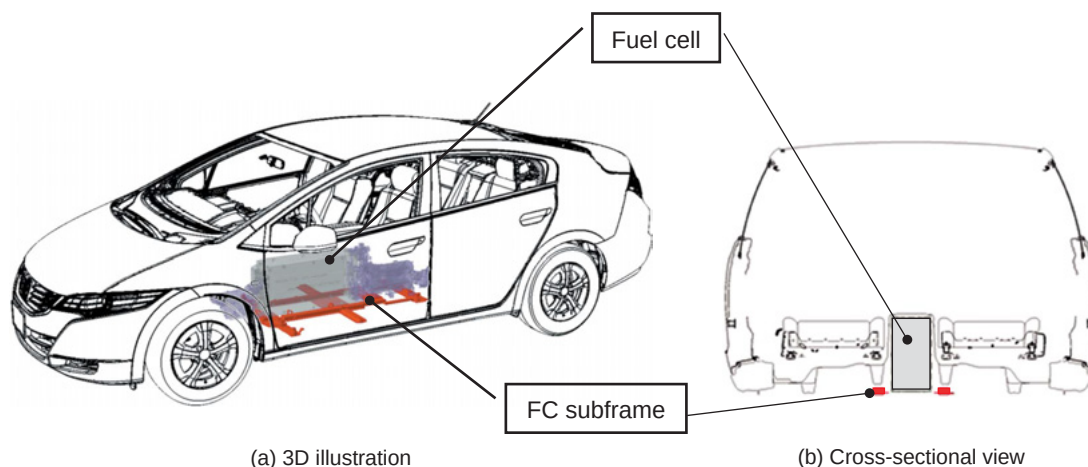


Fig. 1 Positioning of FC subframe in Honda FCX Clarity

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The FC subframe used in the FCX Clarity, as shown in Fig. 1, is installed under the floor to hold and protect the fuel cell system. Since saving weight was a priority, aluminum alloy was used for all component parts. With conventional aluminum parts, however, there was an issue of deformation as a result of arc welding. FSW was therefore adopted as the joining method for the main structural parts. This was intended to reduce deformation due to joining and was also expected to save weight by simplifying joint structures and optimizing wall thicknesses.

Lap joints were used for the joint structure so that dimensional errors in the parts being joined, from when the parts were manufactured, could be absorbed. In order to obtain the necessary joint strength, linear welding was used. There are few cases, however, of linear welding in lap joints using FSW.

Here, therefore, linear welding techniques for FSW lap joints and butt joints were developed for that purpose. Suitable structures for lap joints were also designed and implemented.

This paper provides an overview of the development of those FSW techniques.

2. FSW and Circumstances of FSW Application

The FSW technology was developed by The Welding Institute Ltd. (TWI) in the United Kingdom in 1991⁽²⁾. Figure 2 shows the welding tool (hereafter the tool) with the pointed tip known as a probe. In order to make a join, this tool is rotated at high speed and pressed against the material being worked on until the shoulder portion touches the material. The material around the probe and the shoulder are softened by the frictional heat, and the plastic flow produced by the rotation of the tool leads to the material being joined in a solid state.

This technology is said to be suited mainly to use with aluminum alloys, copper alloys, magnesium alloys, and other metals with low melting points.

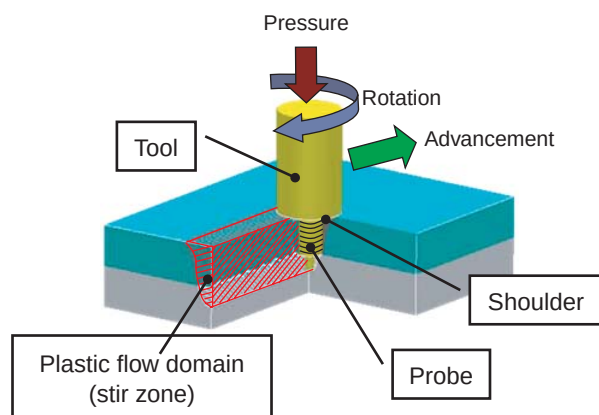


Fig. 2 Schematic illustration of FSW

Characteristics of FSW include:

- (1) It is a solid phase join that does not melt the materials being joined.
- (2) It produces little joint deformation.
- (3) Blowholes and other internal quality imperfections produced by melting and solidification do not occur.
- (4) In the case of aluminum alloys, shielding gas is not needed.
- (5) Filler metal is not needed.
- (6) Fumes, spattering, and the like do not occur.
- (7) A large amount of electric power is not needed.

Drawbacks of FSW include the following⁽³⁾:

- (1) Rigid clamping is necessary.
- (2) The allowable range of gaps between workpieces is narrow.
- (3) It is challenging to make fillet joints and other such joins of members with complex shapes.
- (4) A hole remains at the end of the weld.
- (5) It is limited for use with metals that have low melting points.

Making the most of the above characteristics, Japan has seen an increase in cases of FSW being used in particular⁽⁴⁾ with railway rolling stock made from aluminum alloy. This involves many long parts as much as 20 m in length, which necessitates measures against deformation during welding when undergoing conventional fusion welding. The application of FSW, which results in little deformation, not only increases productivity, it is also able to save energy and provide a clean work environment.

The automotive field also shows an increase in the number of cases⁽⁵⁾ of FSW being used, such as resistance spot welding for spot joining on doors, hoods, and other aluminum alloy components.

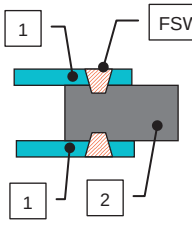
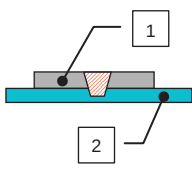
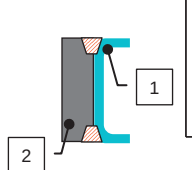
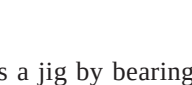
3. FC Subframe

3.1. FC Subframe Structure

Figure 3 shows the FC subframe materials and components, and Table 1 shows FSW weld structures. The components basically consist of combinations of aluminum alloy extruded parts and cut solid parts (hereafter solid parts). Table 2 shows the cross-sectional shape of the extruded parts. Many cross-sections are hollow and complex. The cross-section circumscribed diameters range from 36 to 214 mm. The parts are made thin, with thickness ranging from 1.3 to 3.5 mm. In order to achieve material quality that balances extrudability and strength, A6N01S-T5⁽⁶⁾ was used. For the solid parts, the priority was placed on strength and A6061P-T651⁽⁷⁾ was used. For the stamped parts used in attaching components, A5052P-O⁽⁷⁾ was used for strength and formability, while A6063S-T5⁽⁶⁾ was used for other attaching parts.

The main structural parts were constructed by inserting solid parts into extruded parts and joining them using FSW. This construction was adopted so that the solid parts

Table 1 FSW weld structures in FC subframe

	Part 1		Part 2	
	Material 1	Thickness (mm)	Material 2	Thickness (mm)
	A6N01S-T5	1.3	A6061P-T651	
		1.5		
		2.0		
		3.0		
		3.5		
	A6N01S-T5	2.0	A6N01S-T5	2.0
			A6063S-T5	2.0
			A5052P-O	1.2
			A5052P-O	2.0
	A6N01S-T5	3.0	A5052P-O	1.0
			A6063S-T5	2.0
	A6N01S-T5	2.0	A6961P-T651	7.0

would function as a jig by bearing the load from the tool during joining. Another purpose is to absorb the error from lengthwise processing of extruded parts.

The joints in conventional FC subframes were formed using resistance spot welding and MIG welding. These methods had limits on the thickness of the plates they could join, however, and they could not be optimized. They also resulted in more complex joint structures that were incompatible with weight reduction.

The present structures were designed with FSW in

mind. They were intended to save 10% or more weight by such means as simplifying the joints, eliminating joint flanges, and optimizing the wall thicknesses of extruded parts. Another goal was to discontinue corrective work after joining.

The present FC subframe has 48 lap joints with the structure of thick solid parts inserted into extruded parts, 14 lap joints joining two thin plates, and four butt joints with solid parts joined to the side surfaces of extruded parts.

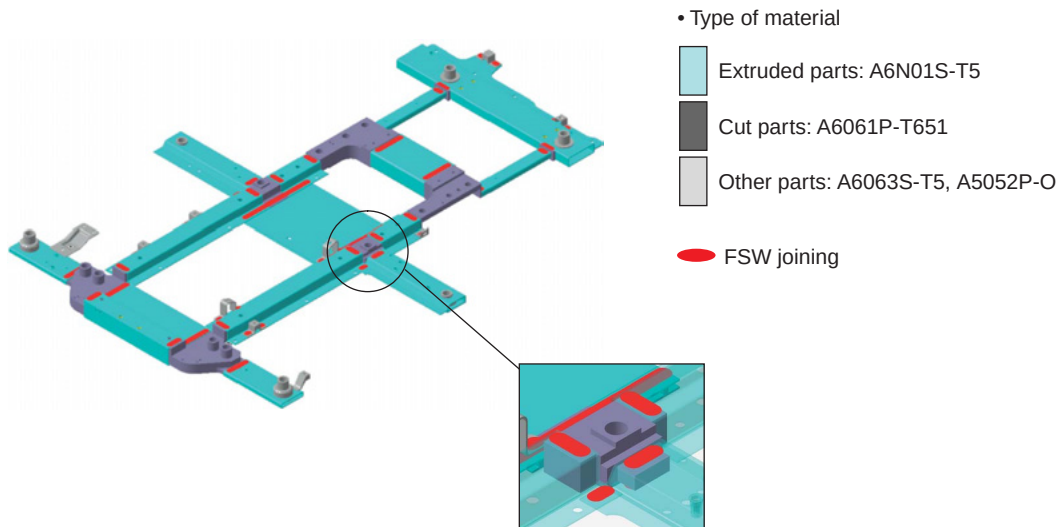


Fig. 3 Components of FC subframe

Table 2 Cross-sectional shape of extruded frames

Cross-sectional view	Circumcircle diameter (mm)	Minimum thickness (mm)	Thickness of FSW weld (mm)
	36	1.5	1.5
	73	2.0	2.0
	99	1.3	1.3
			2.0
	111	2.0	3.0
	111	2.0	2.0
			3.5
	156	2.0	2.0
	190	2.0	2.0
			3.0
	214	2.0	2.0
			3.0

3.2. Issues in Joint Structures

It was found that when the FSW tool used for butt joints is used for a lap joint, the tensile strength is reduced. This mechanism is illustrated in Fig. 4. The rotation of the tool generates frictional heat that softens the material, and the reverse screw effect of the probe generates a downward plastic flow. The material that flows downward along the probe is blocked by the bottom sheet so that it flows outward and up. The upward flow of material around the outer margin of the zone of plastic flow lifts up the softened material in the periphery as though dragging that material along with it. The result is the hook phenomenon, in which material from the bottom sheet infiltrates the top sheet in a wedge-like manner. The boundary of unjoined material is moved up into the top

sheet just as it is by infiltration, so that the thickness of the bottom sheet is reduced by the amount of material removed from the original plate thickness by the hook, thus lowering the tensile strength.

Figure 5 shows a cross-sectional photograph of a lap joint. It can be seen that the plastic flow zone has spread outward from the tool center to form a trapezoidal shape. Figure 6 is a cross-sectional photograph showing the top sheet forced upwards in order to illustrate clearly how the hook process has induced infiltration of the boundary of unjoined material into the top sheet. In this case, the hook height accounts for 1.60 mm of the 2.89 mm of the top sheet. The remaining thickness of the top sheet has been reduced to half or less that of the original plate, so that the tensile strength of the joint portion is also one-half or

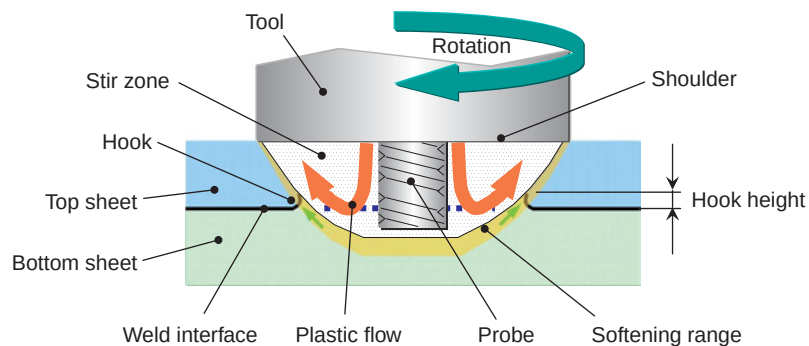


Fig. 4 Mechanism of FSW for lap joint

less of the strength of the parent material in the original plate.

Figure 7 presents a schematic view of a butt joint. It shows how the curvature of the angular edge areas of the extruded part is joined just as it is. Figure 8 is a cross-sectional photograph showing that an elongated cavity has formed. A hole like this is generally thought to form because of insufficient heat input⁽⁸⁾, but in this case the

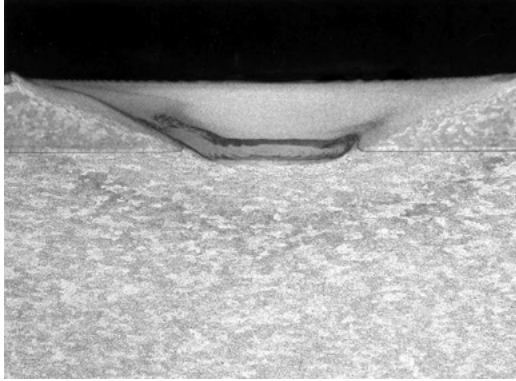


Fig. 5 Cross-sectional micrograph of FSW at lap joint

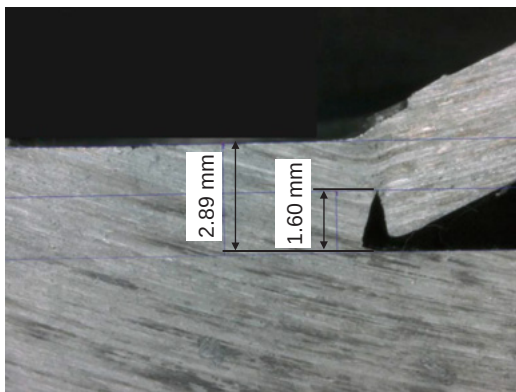


Fig. 6 Cross-sectional micrograph of FSW at lap joint when top sheet is forced upwards

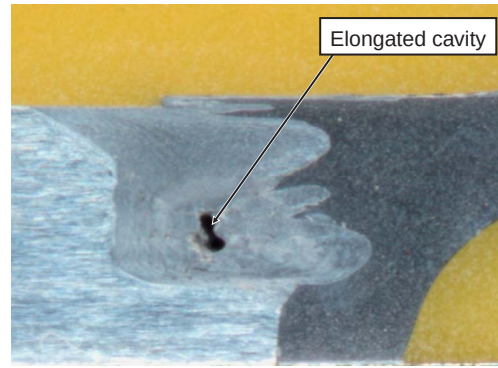


Fig. 8 Cross-sectional micrograph of elongated cavity

origin is probably the cavity formed because of the curvature of the angular edge area of the extruded part. Since FSW does not allow weld reinforcement, such holes are thought to form because there is insufficient material to fill the cavity resulting from the curvature of the angular edge area of the extruded part. This imperfection can lead to reduced tensile strength, and it therefore had to be addressed for the development of practical applications.

4. Joining Technology

4.1. Addressing the Lap Joint Issue

In order to prevent a reduction in the tensile strength of the lap joint, it is crucial that the hook process be contained. One conceivable method for containing hook is to weaken the plastic flow in a longitudinal direction.

First of all, reducing the rotational speed of the tool appears likely to weaken plastic flow in the longitudinal direction. If the tool rotates slowly, however, it will not generate the heat required to form the join, which will therefore result in unstable joints. Figure 9 shows the cases of lap joints formed in 6000-series aluminum alloy with varying tool rotation speeds and welding speeds.

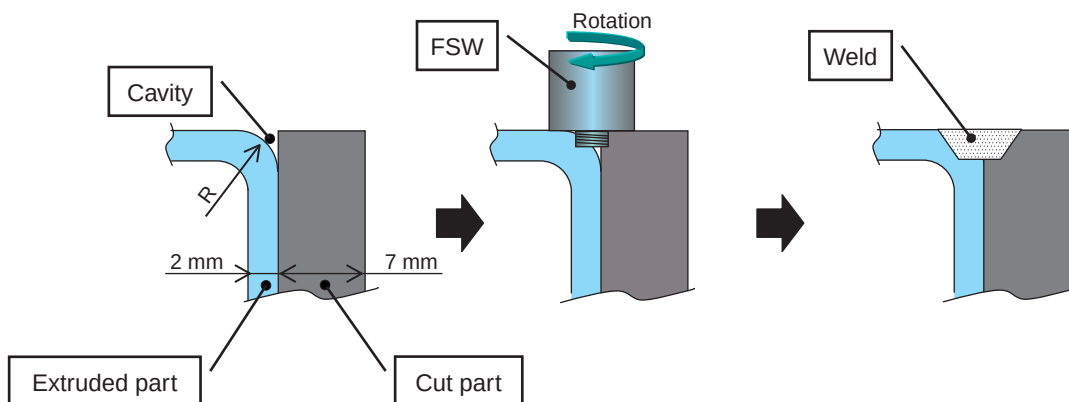


Fig. 7 Cross-sectional image of FSW butt joint

When the tool rotation speed is 1000 rpm, good join quality was obtained with the welding speed of 100–150 mm/min. When the tool rotation speed was decreased to 800 rpm or less, however, the heat input was inadequate and the joints were unstable. When the tool rotation speed was 1100–1200 rpm and the welding speed 100 mm/min, good quality joints were not obtained. It is likely that the excess heat generated at high revolutions results in imperfections from surface gouging and excessive winding up.

The shape of the probe on the developed tools is shown in Fig. 10. It was decided to add grooves like those of a thread-cutting tap to the probe in order to maintain the rotational speed required to form a join and weaken the plastic flow. Grooves in the probe would ordinarily promote plastic flow, but these grooves were devised to divide up and weaken the plastic flow by retaining the softened metal inside the grooves. When the downward plastic flow produced as a result of the screw effect is weakened, the plastic flow that moves upward while spreading outward, thereby generating the hook, is also weakened. The tip of the probe was also hollowed out in a concave form that is linked to the grooves, so that the solid volume of the tool at the probe tip is reduced. This can also be expected to have the effect of lowering the pressure exerted on the material from the periphery of the probe. This was intended to lower the pressure from the probe tip and reduce the force exerted on the periphery to spread it outward in order to contain the plastic flow. By incorporating these design purposes into the probe shape, it was expected to contain the hook phenomenon, even at tool rotating speeds, that generate the frictional heat required to form a join.

4.2. Results of Lap Joint Study by Simulation

A CAE analysis was carried out utilizing general-purpose thermal flow modeling and analysis software

(FLUENT from ANSYS, Inc.) to confirm the effects of tool shape on plastic flow.

CATIA V5 was used to prepare a model of conditions when the tool is inserted in a region of the parent material with a diameter approximately 1.3 times that of the tool shoulder. The automatic mesh generation function of the thermal flow modeling and analysis tool (FLUENT for CATIA V5 from ANSYS, Inc.) was used to generate a mesh for analysis. With the density and coefficient of viscosity set as constant physical properties, the tool was assigned rotating speeds and analyzed in a steady state.

Figure 11 presents typical cross-sections showing the differences in flow direction and flow rate distribution obtained from the simulation of tools with and without grooves. Here, the maximum flow rate, defined as 1, is a dimensionless number.

It can be seen from Fig. 11 that when a groove is placed in the probe, the flow rate is reduced even at the same tool rotational speed. This is thought to occur because the groove induced the flow around the outer margin of the probe to be divided. It is apparent that the

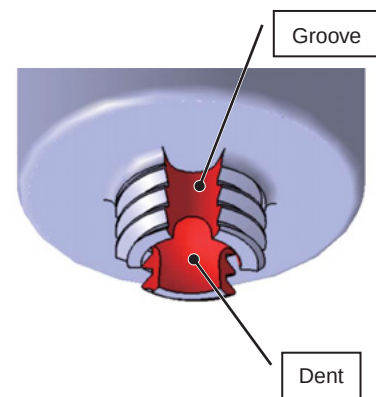


Fig. 10 Probe for developed tool

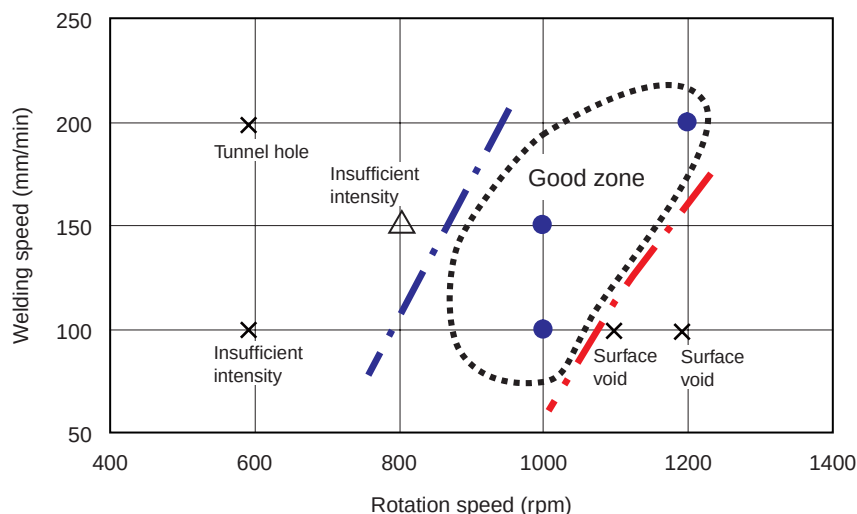
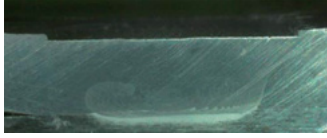
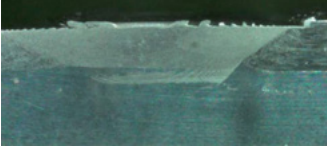
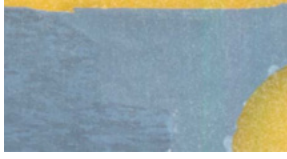
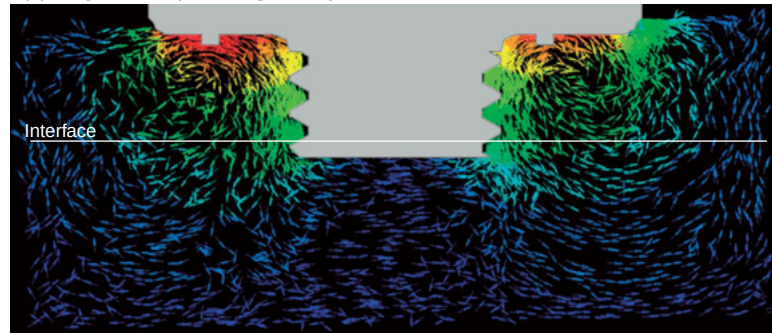


Fig. 9 Example of conditions for lap joint connection of 6000 series aluminium alloys

Table 3 Results of FSW using developed tools

No.	Joint	Type of material		Thickness (mm)	Cross-sectional micrograph	Joint efficiency
1	Lap joint	Top	A6N01S-T5	1.3		78%
		Bottom	A6061P-T6	28.0		
2	Lap joint	Top	A6N01S-T5	2.0		88%
		Bottom	A6061P-T6	28.0		
3	Lap joint	Top	A6N01S-T5	3.0		78%
		Bottom	A6061P-T6	14.0		
4	Lap joint	Top	A6061P-T6	2.0		71%
		Bottom	A6N01S-T5	3.0		
5	Butt joint	Right side	A6N01S-T5	2.0		91%
		Left side	A6061P-T6	7.0		

(a) Original tool (without grooves)



(b) Developed tool (with grooves)

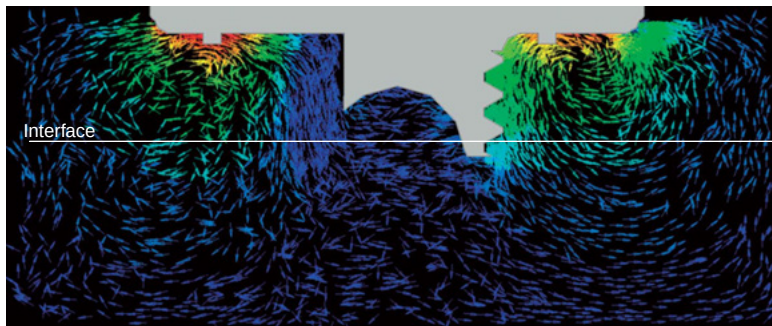


Fig. 11 Calculation of effectiveness of developed tool using CFD

flow is smaller inside the grooves, in particular. This suggests that the flow rate in the longitudinal direction will also be reduced, and the plastic flow weakened.

4.3. Addressing the Butt Joint Issue

The present butt joints had cavities produced by the curvature of the angular edge area of the material. In order to fill in those cavities, it was decided to displace the center of the tool approximately 0.5 mm toward the solid part. The reasoning was that this would cause the material to flow away from the solid-part side so that it could fill in the cavity. The joining conditions were also set so that the tool indentation amount would be deeper than usual on the reasoning that this would be effective in filling the cavity.

4.4. Results

The present FC subframe for use in the FCX Clarity has plates combined using 12 types of lap joints. Five types of tools were developed to match the upper plate thickness in those various combinations. These tools have grooves set into the probe. One tool for use with butt joints was developed.

Table 3 shows the verification results of the quality of joints made using the developed tools. Given the ability to contain the hook phenomenon, the ratio of the tensile strength of the joint to that of the parent material (hereafter joint efficiency) was 60% or more in every part of the 13 types of plate combination. The results from observation of cross-sections also confirmed that the hook phenomenon was being contained.

Figure 12 shows the finished FC subframe. The length of the FSW welds amounted to 2075 mm at a total of 66 locations. Distortion produced by the welding was very slight, enabling the attachment of welding jigs, without having to add an allowance for anticipated distortion. Parts accuracy requirements could also be satisfied without having to perform any corrective work after welding. This

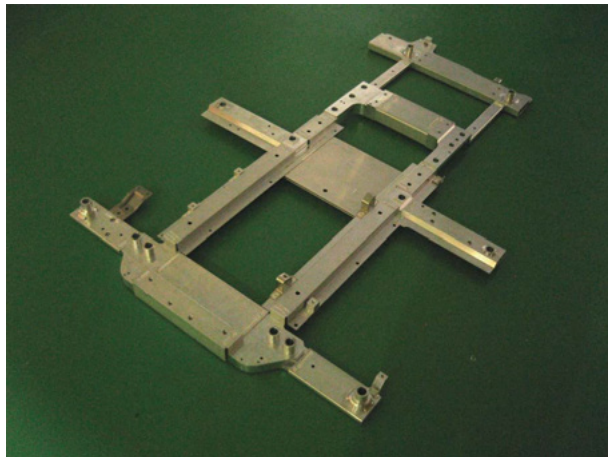


Fig. 12 Finished FC subframe

has curtailed corrective work processes, thus enabling greater work efficiency.

5. Mass Production Equipment

A brief introduction of the mass production equipment will be given here. Figure 13 presents a photograph showing an exterior view of the equipment. This system uses a general-purpose multi-axis robot for the FSW tool⁽⁹⁾. The FC subframe is set on a welding jig that is supported by two robots. The stationary FSW welding device clamps onto the workpiece and jig together and makes the join. The two robots simultaneously move the jig with the workpiece placed on it to form the welds. The device in this system is able to freely control the load on the FSW tool, and by accurately synchronizing its action with the movement of the robots, the system is able to produce stable joints. The FSW tool is attached to a highly rigid frame and on the jig side it bears the load from joining by means of rollers. This system is capable of forming joints easily in two dimensions by means of robot teaching, which also enables it to do three-dimensional welding work.

Quality management is implemented by monitoring the welding conditions, including the load on the FSW tool, its rotational speed, and the welding speed. Six types of tool are color-coded by their particular areas of application. The length of the probe and the shape of the tip are checked regularly.

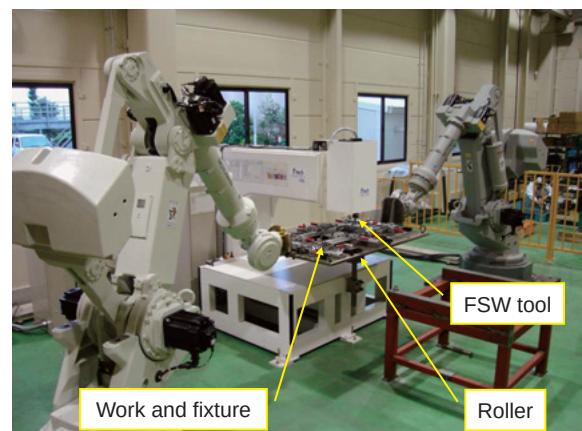


Fig. 13 Components of FSW machine

6. Conclusion

This FC subframe for use in the FCX Clarity achieves a 17% weight saving relative to the conventional subframe made from stamped aluminum alloy using resistance spot welding and MIG welding. It also achieved the reduction of corrective work. Productivity increased as a result.

- (1) The mechanism by which the hook phenomenon, generated by plastic flow in the FSW lap joint, reduces

the strength of the join was clarified. Flow modeling and analysis software were used to develop tools that contain the winding up phenomenon by means of grooves.

- (2) Using these tools, extruded parts made of 6000-series alloy and parts of various kinds of aluminum alloy were joined by FSW in lap joints and butt joints. Joint efficiency reached levels of 60% or higher.

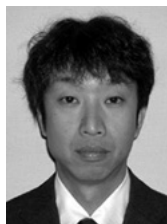
The robot welding system constituting a new mass production device is capable of controlling the welding load to perform stable welds. At the same time, this equipment is also capable of carrying out welding work in three dimensions. The system is producing FC subframes of stable quality.

Weight saving is a vital technology for the motor vehicle development of the future. Promoting widespread adoption of the present techniques can be expected to contribute to weight savings.

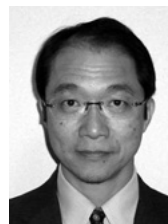
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Development of Fuel Cell Coolant

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ABSTRACT

Since fuel cell coolant (FC coolant) should have insulating properties for the purpose of electric leakage prevention, ion-exchange resin is used in the system to remove ions from the coolant. An antioxidant appropriate for FC coolant was formulated for the developed coolant, thereby enhancing oxidation resistance and reducing the generated ion content. This enabled a 40% reduction in the amount of ion-exchange resin.

1. Introduction

Fuel cell vehicles are attracting attention as zero emission automobiles because they do not generate CO₂ when they run. However, the fuel cell (FC) stacks that generate their power also generate heat together with electricity, so they require cooling with coolant. The coolant requires insulating properties in order to help prevent electric leakage, and while the FC generates power, those insulating properties are maintained by ion-exchange resin that is built into the system for the purpose of removing ions from the coolant. Figure 1 shows the layout of the ion-exchange resin in the FC cooling system.

When ethylene glycol (EG), which is generally used as a coolant base material in internal combustion engines, is used as the FC coolant base material, the non-freezing property of EG can be assured, but oxidation results in ionic material being produced. Therefore, the system needs large amounts of ion-exchange resin to be loaded

in order to remove the ionic material from the coolant.

The present development project addressed the need for lighter weight components in fuel cell vehicle. An FC coolant that minimizes the production of oxidants generated from the coolant was developed with the goal of reducing the amount of ion-exchange resin installed in the system.

2. Development Goals

The development of FC coolant that would enable reduction of the amount of ion-exchange resin first required enhanced oxidation resistance and reduced production of ionic material. It was also necessary that the FC coolant not promote the release of ions from the material composing the cooling system parts nor alter the physical properties of that material to any great extent. It is further necessary to assure its cooling performance as well as its non-freezing property at low temperatures for use in cold regions. The objectives of this development project were therefore set forth as follows:

- (1) Enhanced oxidation resistance
Enhance oxidation resistance with regard to EG and reduce the production of ionic material due to oxidation by 90% or more.
- (2) Limited elution of ions from the cooling system material
Limit ion release from material used in the cooling system to the same level as that from EG in an aqueous solution.
- (3) Compatibility with cooling material
Limit changes in volume and other physical properties with respect to material used in the cooling system to the same level as EG in aqueous solution.
- (4) Cooling performance as well as non-freezing property in cold regions

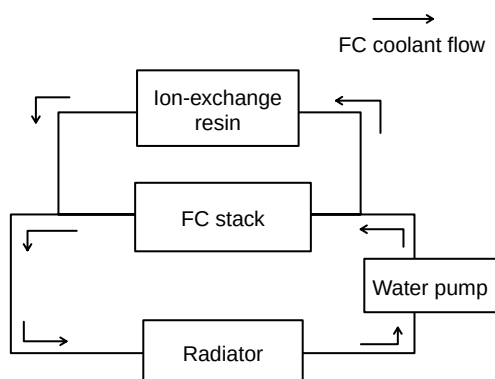


Fig. 1 Ion-exchange resin layout in FC cooling system

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The cooling performance and non-freezing property are physical properties of the mixed solvent composed of the base materials and water, and these properties are to be equivalent to EG in an aqueous solution.

3. Method of Addressing Goals

In order to obtain the needed cooling performance, the base materials were studied. In order to enhance the oxidation resistance, a study of additives was conducted.

3.1. Selection of Base Materials

Table 1 shows the main properties of commonly used base materials, in this case, EG, water, alcohols, glycols (other than EG), glycol ethers, and hydrocarbon. These base materials are compared with EG in terms of specific heat, viscosity, and melting point. Water is superior in specific heat and viscosity but has a high melting point that makes it inferior in non-freezing properties. Although the alcohols, ethers, and hydrocarbon are superior in viscosity and melting point, they have low specific heat so their cooling properties are inferior. Consequently, there are no base materials with physical properties that make them superior to EG in terms of specific heat, viscosity, and melting point. EG was therefore selected as a base material that would enable a balance between cooling and non-freezing properties.

3.2. Investigation of Additives

3.2.1. EG oxidation reaction

Figure 2 shows the oxidation reaction undergone by EG in aqueous solution containing dissolved oxygen. In general, primary alcohols possessing hydroxy groups undergo oxidation reactions that proceed through radical reactions to aldehyde and then to carboxylic acid. The oxidation reaction proceeds similarly in EG, ultimately producing acids such as glycolic acid and formic acid⁽¹⁾.

Figure 3 shows the adsorption of glycolic acid and formic acid from among the ions adsorbed by ion-exchange resin in the FC cooling system during power generation. Extending the duration of power generation by the FC tends to cause more formic acid and glycolic acid ions to be adsorbed by the ion-exchange resin. This indicates that the glycolic acid and formic acid produced by the oxidation of EG in the FC cooling system has been adsorbed when passing the ion-exchange resin.

3.2.2. Criteria for selection of antioxidant

Methods for inhibiting the oxidation of EG include:

- (1) Inhibit progression of the reaction by radical-scavenging
- (2) Reduce the environmental temperature
- (3) Remove dissolved oxygen

Of these, items (2) regarding temperature and (3) regarding removal of oxygen are determined by the system environment. This makes it challenging to address these factors from the coolant side. Investigation was therefore

Table 1 Performance of base materials

Base material	Specific heat	Viscosity	-- -- + ++ Good
			Melting point
EG	+	+	+
Water	++	++	-
Alcohol	-	++	++
Glycol (except EG)	-	-	-
Glycol ether	-	++	++
Hydrocarbon	--	++	+

directed to antioxidants with radical-scavenging and deoxygenation properties.

The selection of an antioxidant also involves two required criteria on the system side in addition to prevention of oxidation. One is heat resistance. The antioxidant should perform its role even in high temperature regions. The other is that it should have nonionic properties so that it will not be removed by the ion-exchange resin installed inside the passages.

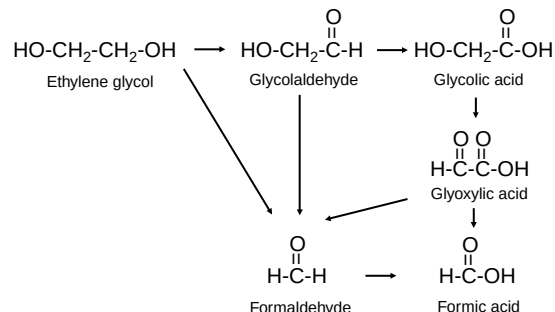


Fig. 2 Oxidation reaction of ethylene glycol

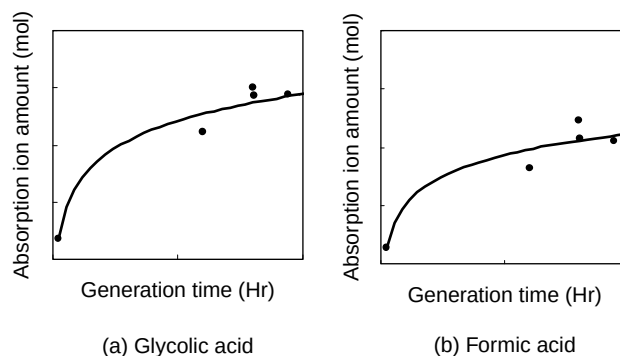


Fig. 3 Adsorption ion on ion-exchange resin (glycolic acid and formic acid)

Table 2 Performance of antioxidants

	Antioxidant	Heat resistance	Ionic
Nitrogen-containing compounds	#1	++	Ionic
	#2	++	Ionic
	#3	++	Ionic
Hydroxy compounds	#4	-	Non ionic
	#5	++	Non ionic
	#6	++	Ionic
	#7	++	Non ionic
	#8	+	Ionic

-- - + ++
Good

3.2.3. Results of antioxidant selection

Various antioxidants with the nitrogen-containing compounds and hydroxy compounds like those numbered 1 to 8 in Table 2 were selected according to the above criteria. Numbers 5 and 7 in the hydroxy group were selected as satisfying the conditions of heat resistance and nonionic property. These were designated antioxidants A and B.

3.2.4. Predicting the antioxidant reaction mechanism

In general, the following types of reaction provide anti-oxidation performance in antioxidants that possess hydroxy groups⁽²⁾:



As in Eq. (1), the antioxidant AO-H provides the hydroxy group hydrogen atom $H^{\cdot}$, and the antioxidant radical $AO^{\cdot}$ captures the various radicals generated by the reaction system. Figure 4 shows one example of the antioxidant reaction mechanism that is conjectured to operate in EG oxidation⁽³⁾. First, one of the EG hydroxy groups is attacked by hydroxyl radicals or other such radicals that pull away the hydrogen atom $H^{\cdot}$, turning the group into an alcoxyl radical. That then reacts with oxygen in the passageway to form glycoaldehyde. When that oxidation reaction occurs, the hydrogen atom provided by the antioxidant captures the hydroxyl radicals, alcoxyl

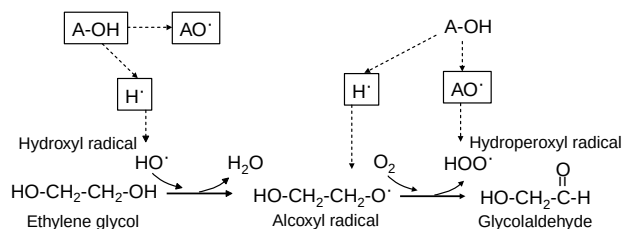


Fig. 4 Oxidation inhibition mechanism of EG by antioxidant

radicals, hydroperoxyl radicals, and the like to stabilize them, and the remainder antioxidant radical $AO^{\cdot}$ captures them as well.

There are also antioxidants that possess deoxygenation properties. An example of such a reaction is shown in Eq. (2).



The rate of the oxidation reaction is also affected to a large extent by the oxygen concentration. The deoxygenation function can therefore slow down the oxidation reaction rate.

4. Results of Performance Evaluation

4.1. Results of Oxidation Resistance Evaluation

The FC coolant was made from the selected base materials and antioxidant, and its oxidation resistance performance was evaluated by means of component heat-resistance testing. The dilution concentration of the base materials, EG, was set at 50%. Antioxidant A or B was added to make FC coolant A and FC coolant B. Their evaluation temperature was set at 100°C and their oxidation resistance was compared to that of the EG aqueous solution. Figure 5 shows the results.

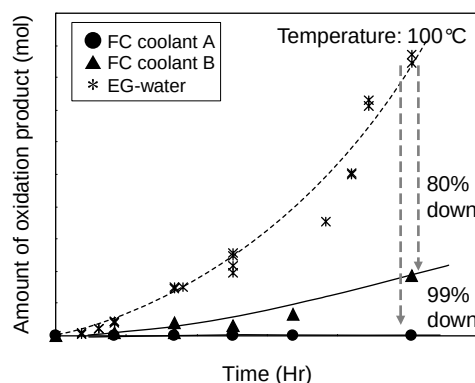


Fig. 5 Oxidation resistance test result

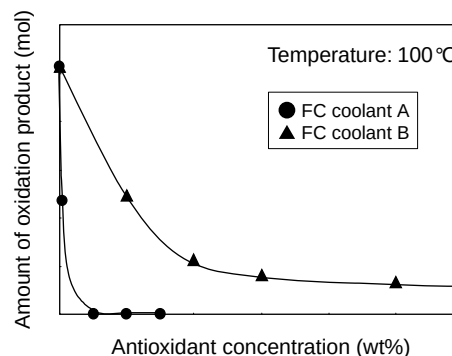


Fig. 6 Effect of antioxidant concentration for oxidation resistance

The EG aqueous solution to which they were compared showed an increase in oxidation product as the test time increased. However, the amount of oxidation product was held down in the FC coolant formulated with antioxidant, with FC coolant A showing a reduction in oxidation production of 99% or more, and FC coolant B a reduction of approximately 80%.

Figure 6 shows the amount of oxidation product with varying concentrations of antioxidant. The amount of oxidation product can be reduced by increasing the antioxidant concentration. In the case of FC coolant A, in particular, adding antioxidant to a certain concentration or greater brought the amount of oxidation product more or less to zero, providing sufficient anti-oxidation performance.

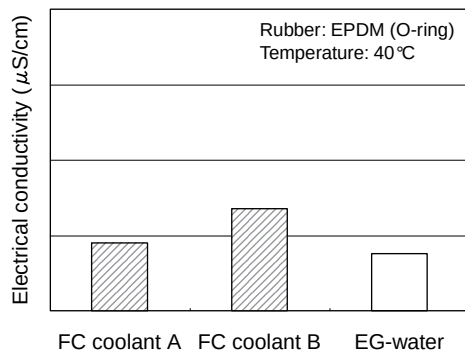


Fig. 7 Coolant extraction performance for rubber

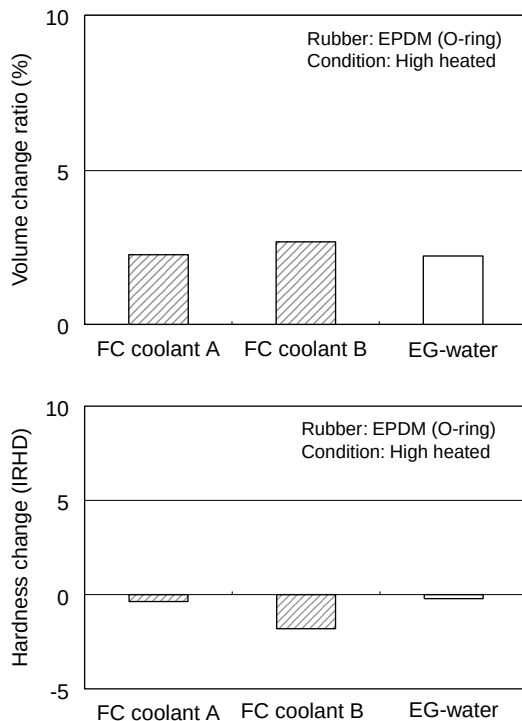


Fig. 8 Coolant application performance for rubber

4.2. Results from Verification of Material Elution

As explained above, it is important that the insulating properties of the FC coolant be maintained. Therefore, the elution of ionic material from the material composing the passages should not be promoted. An evaluation was therefore carried out of elution in the rubber and resin materials from which cooling system parts are made. The quantity of ions eluted by immersion of the material was evaluated in terms of changes in the conductivity of the coolant. It was confirmed that with both the rubber and resin materials, the conductivity of the FC coolant was equivalent to that of the EG aqueous solution. Figure 7 shows ion elution results from rubber material as an example of this.

4.3. Results from Confirmation of Material Compatibility

In order to confirm the influence of the physical properties of the material from which the passageways are made, the rubber and resin materials were subjected to testing for material compatibility similar to the testing for elution. The rubber material was measured for changes in volume, hardness, tensile strength, and elongation before and after immersion. The resin material was measured for changes in flexural strength and flexural modulus. Both the rubber and resin materials showed rates of change with FC coolant that were equivalent to those with EG aqueous solution, confirming that their material compatibility is equivalent to the EG aqueous solution. Figure 8 shows an example of the material compatibility results for rubber material.

5. Ion-exchange Resin Reduction Effect

The effect of reduction in the amount of ion-exchange resin by enhancing oxidation resistance in the FC coolant was estimated with regard to FC coolant A, in particular, which has superior oxidation resistance. The results are shown in Fig. 9. Enhancement of the FC coolant oxidation resistance reduced the adsorption of ionic material due to oxidation of the FC coolant. As a result,

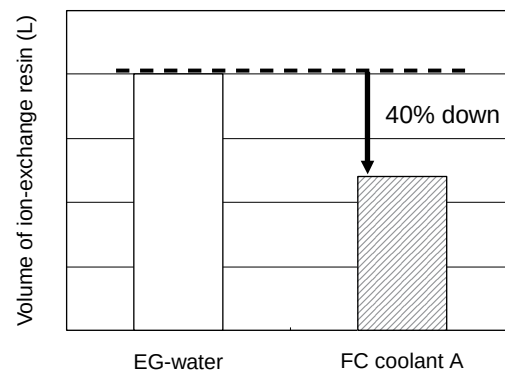


Fig. 9 Reduction of ion-exchange resin through FC coolant

the amount of ion-exchange resin installed in the vehicle could be reduced by 40%, for a saving of weight in parts.

6. Conclusion

- (1) As an effect of adding antioxidant, an FC coolant was developed that reduced the ionic product due to oxidation 90% or more relative to EG.
- (2) It was confirmed that this coolant has compatibility with cooling system parts equivalent to that of EG.
- (3) Adoption of this coolant for use enabled a 40% reduction in the amount of ion-exchange resin.

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Tsuyumi TAKASHIBA



Shiro YAGAWA

Electric Drive Motor for Fuel Cell Vehicle FCX Clarity

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ABSTRACT

A new DC brushless motor has been developed as the electric drive motor for the Honda FCX Clarity fuel cell vehicle.

Due to the demands of vehicle packaging, a coaxial configuration was employed for the drive motor and gear box, and the power drive unit has been integrated with the drive motor, enabling the size of the overall drive unit to be reduced. The integration of the drive motor and gear box has reduced friction, contributing to increased operating efficiency.

An IPM rotor has been employed to increase motor speed in order to increase maximum vehicle speed, and to increase power output in order to enhance vehicle performance. The use of a limit value for the inductance setting that enables greater power to be achieved at maximum motor speed has reconciled high motor speed with high power.

These innovations have enabled the length of the drive unit to be reduced by 162 mm and its height to be reduced by 240 mm. The electric drive motor achieves a maximum power of 100 kW, maximum torque of 256 Nm, and maximum speed of 12 500 min⁻¹. In addition, a maximum motor drive efficiency of 97% has been achieved.

1. Introduction

Against a background of increasing global environmental awareness, further reductions in CO₂ emissions are demanded in order to reduce automotive emissions and to combat global warming. Responding to these demands, Honda moved forward with the development of the Honda FCX fuel cell vehicle (2003 model year FCX) as the ultimate environmentally-friendly automobile. In 2002, Honda led the world by leasing fuel cell vehicles to the Japanese government and the City of Los Angeles in the United States⁽¹⁾. In order to increase the diffusion of fuel cell vehicles, the company went on to develop the New Honda FCX (2005 model year FCX), which was fitted with a proprietary fuel cell stack that displayed enhanced performance at low temperatures. This was the world's first fuel cell vehicle able to be used at temperatures below freezing, and the vehicles were leased to the New York State administration in 2004 and the Hokkaido prefectural government in 2005⁽²⁾.

In July 2008, Honda commenced marketing the FCX Clarity, which has as its core Honda's proprietary V Flow FC (Vertical Gas Flow Fuel Cell) stack, a cell configuration in which hydrogen and air flow vertically⁽³⁾. The FCX Clarity has been developed not only to realize the ultimate in clean automotive performance, but to

create new value and offer new appeal as an automobile. The Clarity's short nose and long wheel base, enabled precisely by the use of fuel cells in the vehicle, give it a spacious cabin, and it offers continuous smooth and powerful acceleration from takeoff to top speed. The drive motor employed in the FCX Clarity (the MCF31) has been reduced in size and increased in efficiency, power and speed against the MCF21 motor used in the 2005 model year FCX.

This paper will discuss the technologies employed in the MCF31 and the achieved performance of the motor.

2. Development Goals

The two main goals established for the development of the MCF31 motor were the achievement of:

- (1) Increased compactness and higher efficiency in the drive unit
- (2) Increased power and speed

In order to enable the realization of the design – featuring a short nose and low overall height – that had been made possible by the use of fuel cells, the motor magnetic circuits would be reduced in size and the motor, gearbox, and power drive unit would be integrated to make the drive unit more compact.

In order to realize a 60% level of energy efficiency,

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the achievement of a maximum drive efficiency of 97%, equivalent to that of the MCF21, was established as a target for the motor, to be balanced with the power performance targets described below.

Power performance targets were established from the vehicle performance requirements as follows.

Equation (1) shows the relationship between maximum drive power ($F_{\max}$) and maximum vehicle speed ($V_{\max}$), and maximum motor torque ($T_{\max}$) and maximum motor speed ($N_{\max}$)⁽⁴⁾.

$$F_{\max} \cdot V_{\max} = 2\pi \cdot T_{\max} \cdot N_{\max} \quad (1)$$

In order to achieve the maximum speed of 100 mile/h (160 km/h) that had been established as vehicle targets and maximum drive power required during acceleration, a figure of 12 500 min⁻¹ was established for the maximum motor speed, $N_{\max}$, taking into consideration the range for the reduction ratio of the coaxial gearbox and a balance with increased friction, and the like. From this figure, a figure of at least 256 Nm was established for the maximum motor torque, $T_{\max}$.

A figure for minimum motor torque (bottom value) at 0 min⁻¹ was established from the minimum drive power required for acceleration up a slope from a stationary state. At the same time, the torque constant was increased in order to help ensure that the phase current value at 0 min⁻¹ did not exceed the maximum allowable inverter current.

The target for maximum motor power was established as 100 kW to enable the realization of an equivalent level of passing acceleration performance to a 2.4 L-class gasoline vehicle. In addition, a target for continuous power output was established from the torque conditions required for continuous operation at maximum speed and to enable the vehicle to be stopped while climbing a hill using only accelerator operation, and the achievement of reduced loss and enhanced cooling performance was planned.

In order to increase energy efficiency, the minimum input voltage to enable maximum motor power to be achieved was set as the output voltage of the fuel cell stack with no voltage booster employed. With this as the input voltage, it would be necessary to reconcile the antagonistic requirements of a high torque constant and the achievement of the necessary power at maximum motor speeds.

3. Technologies Employed

Table 1 shows the main specifications of the MCF31 motor. Figure 1 provides an overview of the external appearance of the integrated drive unit formed from the motor, gearbox, and power drive unit.

The basic motor type is a permanent magnet (PM) motor that can use reluctance torque, as employed for the MCF21. An eight-pole rotor was selected in order to control loss (in particular high-frequency core loss), noise, and vibration.

Table 1 Comparison of motor specifications

	MCF21 (2005 model year FCX)	MCF31 (FCX Clarity)
Type	PM synchronous motor	PM synchronous motor
Phase	3	3
Number of poles	12	8
Rotor type	Salient SPM	IPM
Magnet	Nd-Fe-B	Nd-Fe-B
Cooling system	Water-cooled	Water-cooled

3.1. Drive Unit Configuration

The configuration discussed below was employed to reduce the size of the drive unit in order to enable the vehicle packaging to be realized. The transmission efficiency of the drive unit was also increased in order to increase the energy efficiency of the vehicle.

3.1.1. Integration of motor and power drive unit

The total height of the drive unit was reduced by integrating the power drive unit and the motor, to enable the total vehicle height to be reduced and to help protect pedestrians in the event of a collision.

To enable integration of the motor and power drive unit, motor vibration was reduced and vibration resistance of the power drive unit was increased. This has contributed to enabling the height of the drive unit from the bottom of the motor to be reduced by 240 mm against the 2005 model year FCX, as shown in Fig. 2, resulting in reduced space needed to fit the motor.

3.1.2. Coaxial configuration of motor and gearbox

The total length of the drive unit has been reduced in order to realize the short nose design and help ensure occupant safety in a collision.

The rotor shaft of the motor is hollow, and the drive shaft passes through it, enabling the employment of a coaxial configuration for the motor and the gearbox. The

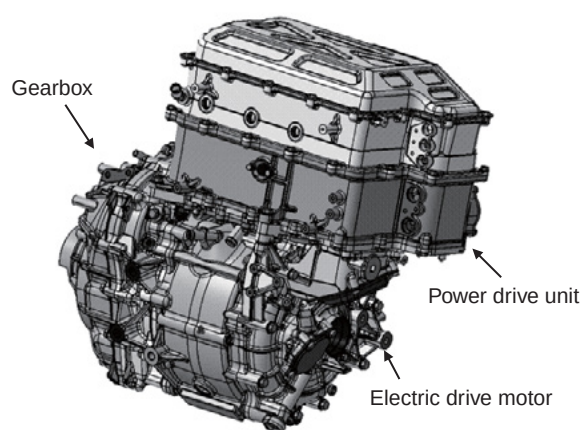


Fig. 1 Appearance of drive unit assembly

groundbreaking new configuration achieved in this motor, together with the reduction of the external diameter of the motor by 4%, has enabled the length from the center of the differential shaft to the front of the motor to be reduced by 162 mm against the 2005 model year FCX, as shown in Fig. 3. This has also reduced the space needed to fit the motor.

3.1.3. Motor oil lubrication

Friction has been reduced in the drive unit for increased transmission efficiency, in order to increase the energy efficiency of the vehicle.

In the 2005 model year FCX [Fig. 4(a)], the motor and the gearbox were housed separately, and their bearings were lubricated using grease and gear oil respectively. In the FCX Clarity [Fig. 4(b)], the oil resistance of the internal components of the motor has been increased, and the same housing has been employed for the motor and the gearbox. The number of oil seals

has been reduced, and newly developed bearings lubricated by gear oil have been employed, enabling the achievement of a maximum reduction of 36% in friction against the 2005 model year FCX (Fig. 5).

In addition, the temperature of the stator windings has been rendered uniform by the use of gear oil in the motor, enabling both power and the range of continuous operation to be increased.

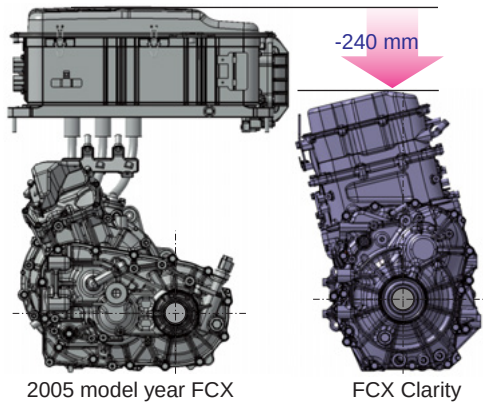


Fig. 2 Comparison of total height of drive unit

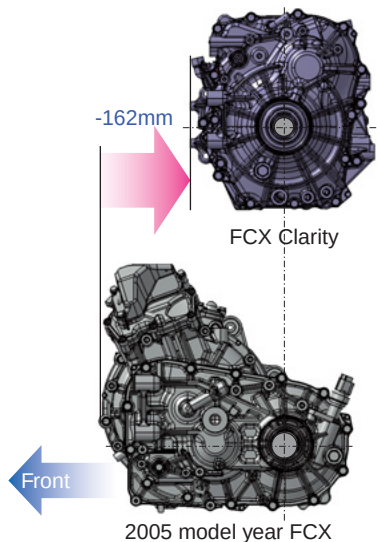
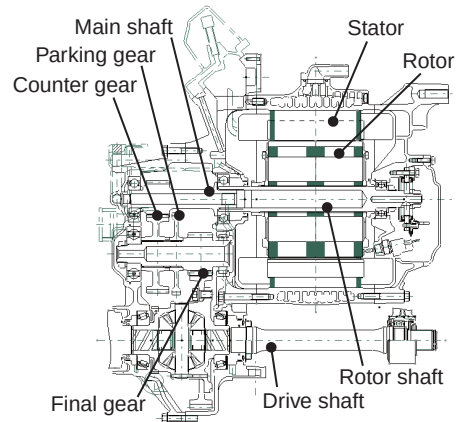
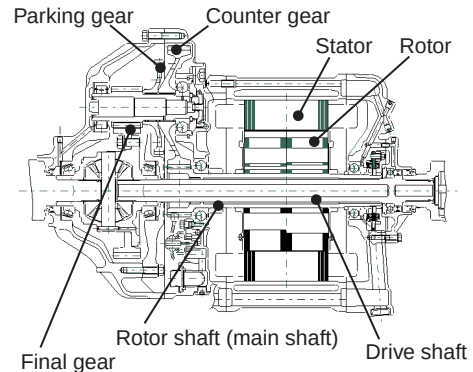


Fig. 3 Comparison of total length of drive unit



(a) 2005 model year FCX



(b) FCX Clarity

Fig. 4 Section of drive unit

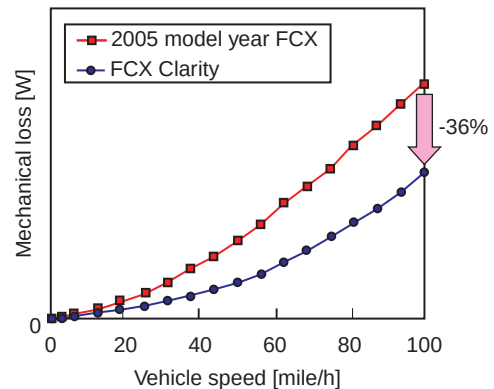


Fig. 5 Comparison of mechanical loss

3.2. PM Motor Able to Use Reluctance Torque

A PM motor that can use reluctance torque was selected as the basic motor type, as in the case of the MCF21, in order to realize high efficiency across the entire driving range.

As Fig. 6 shows, in a PM motor that can use reluctance torque, the magnetic torque generated by the rotor magnets and the reluctance torque generated by the salient poles of the electrical steel sheets between the magnets can be combined, enabling the degree of use of the magnets to be reduced.

The torque generated by a motor that can use reluctance torque, T_q , is expressed by Eq. (2). With the torque axis expressed as the q -axis and the flux field axis expressed as the d -axis, ϕa is the magnetic flux linkage of the magnet, Pn is the number of pole pairs, i_d and i_q are the currents at the d - and the q -axes respectively, and L_d and L_q are the inductances at the d - and q -axes respectively. The first term on the right-hand side of the equation expresses magnetic torque, and the second term expresses reluctance torque⁽⁵⁾.

$$T_q = Pn \cdot \phi a \cdot i_q + Pn(L_d - L_q)i_d \cdot i_q \quad (2)$$

From the second term, it is clear that reluctance torque is proportional to $(L_d - L_q)$.

The current required for field-weakening drive, i_d , is expressed by Eq. (3), where ω is the speed of the electrical angle and V_b is the input voltage.

$$i_d = -\frac{\phi a}{L_d} + \frac{1}{L_d} \sqrt{\frac{V_b^2}{\omega^2} - (L_q \cdot i_q)^2} \quad (3)$$

Achieving a high inductance, which increases L_d , would have the effect of reducing the current required for field-weakening drive.

However, the power, P , can be approximated by Eq. (4), showing that increasing inductance reduces the maximum power at the voltage limit.

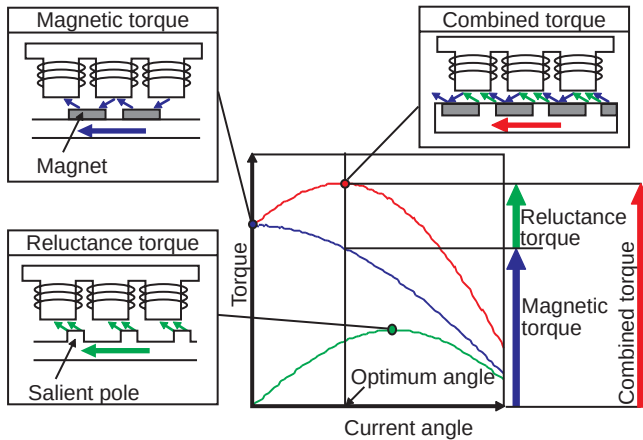


Fig. 6 Conceptual diagram of torque characteristic and current angle

$$P \propto \frac{V_b \cdot \phi a}{L_d} \quad (4)$$

3.3. Selection of Rotor Type

The type of rotor to be used in the MCF31 was selected in order to enable power and speed to be increased and size to be reduced against the MCF21.

It was necessary to optimize the inductance value in order to increase power and reduce size. L_d was set as the limit value (maximum) enabling the achievement of the target power when the motor was operating at maximum speed within the input voltage range. L_q was set in order to maximize the result of $(L_d - L_q)$, given the above condition. This would minimize the current required for field-weakening drive at high motor speed and enable reluctance torque to be used effectively, thus enabling the achievement of increased power and reduced size.

Achieving sufficient yoke strength is one issue in increasing rotor speed. At maximum speed, the weight of the magnets results in a strong centrifugal force being exerted on the yoke. It is necessary to achieve sufficient strength in the yoke to enable it to resist this centrifugal force.

With the goals set as optimization of the inductance value and the achievement of sufficient yoke strength, magnetic field analyses and strength analyses were conducted in order to compare the following two types of permanent magnet that have been successfully mass-produced:

Type A: Salient pole-type surface permanent magnet (SPM) used in the MCF21 motor [Fig. 7(a)]

Type B: Rectangular interior permanent magnet (IPM) with center rib used in the ACCORD Hybrid⁽⁶⁾ [Fig. 7(b)]

In type A, because the magnets can be positioned close to the stator core, the magnetic resistance on the d -axis

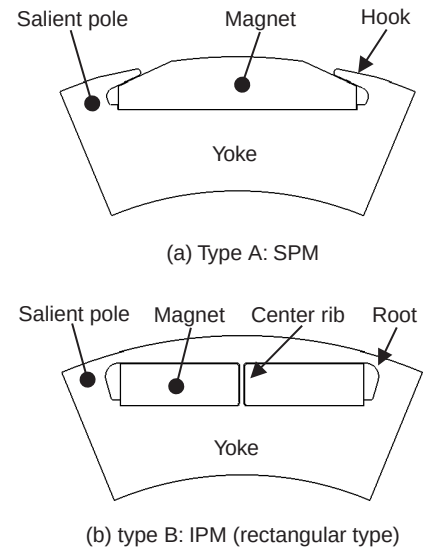


Fig. 7 Structure of rotor

can be increased, enabling L_d to be set low. In addition, (L_d-L_q) can be set high, and this type is therefore beneficial from the perspective of increased power. The results of the magnetic field analysis showed that the use of this type of rotor would enable the achievement of torque and power targets.

However, in addition to its higher speed, the reduction of the number of poles in the rotor in the MCF31 against the MCF21 would make it necessary to increase the amount of magnets per pole, resulting in a strong centrifugal force being exerted on the rotor yoke at maximum speed. The results of the strength analysis showed that stress would concentrate on the yoke hook sections that support the magnets, making it challenging to achieve sufficient strength at high speeds.

Type B divides the magnets in two by means of center ribs. The center ribs and the base of the yoke bear the centrifugal force that is generated, thus enabling excessive stress to be negated. The results of the strength analysis showed that optimum allocation of stress would enable sufficient strength to be achieved up to maximum speeds.

However, this type also reduces the magnetic resistance on the d -axis, thus increasing L_d , and the results of the magnetic field analysis showed that it would not enable the target power to be achieved at high speeds.

Based on the results discussed above, a type C was sought that would fuse the benefits of type A for high power and of type B for high speed.

Type C employed half-moon-shaped magnets to enable L_d to be set low, and used a center rib IPM configuration to enable the achievement of sufficient strength. A magnetic field analysis and strength analysis showed that type C would enable the target of a balance between high power and high speed to be achieved (Fig. 8), and an IPM rotor using half-moon-shaped magnets with center ribs was therefore adopted for use in the MCF31 (Fig. 9). Figure 10 shows the stress distribution on the magnets and the rotor yoke during rotor operation, as calculated in a nonlinear structural analysis.

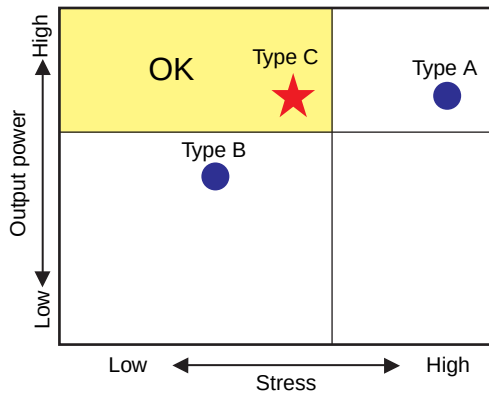


Fig. 8 Conceptual diagram of output power and stress and current angle

As detailed analysis of the half-moon-type IPM with center ribs proceeded, a new issue became apparent: Variations within the dimensional tolerance and variations in the mounting positions in the yoke slots resulted in considerable changes in the stress on the yoke at high speeds. This issue resulted from the fact that the employment of a unique half-moon shape as an IPM meant that the positions of contact between the magnets and yoke slots were unstable. The shapes of the magnets and yoke slots were therefore optimized to help ensure that the positions of contact would be stable, irrespective of the permutation of the dimensional tolerances and the mounting positions. As a result, fracture strengths of 23 000 min⁻¹ or more were achieved in fracture tests using actual rotors.

3.4. Responses to Magnet Heat Production and Surface Treatment

One of the characteristics of the Nd-Fe-B magnets used in the rotor is that their flux density declines as the temperature of the magnet increases. Because of this, as the temperature of the magnets increases with an increase in eddy currents at high speeds, the flux linkage of the magnets, ϕ_a , declines and maximum power also declines, following the relationship shown in Eq. (4)⁽⁷⁾. The upper limit for allowable magnet temperature can be determined from the motor power required in order to achieve maximum vehicle speed and the magnet temperature characteristic.

Heat generated by eddy currents produced in the interior of the magnets by changes in the magnetic flux in the rotor result in a marked increase in magnet temperature. In the MCF31, the magnets were divided in

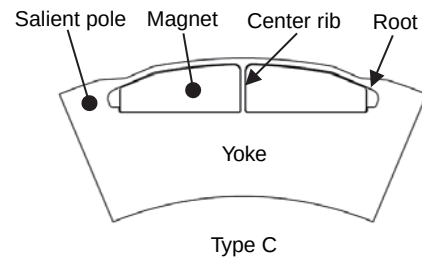


Fig. 9 Structure of IPM rotor (half moon type)

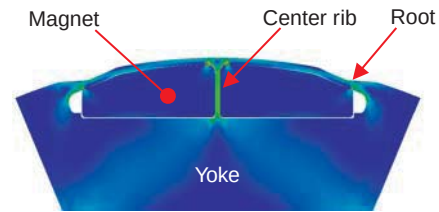


Fig. 10 Nonlinear analysis of IPM rotor (half moon type)

the axial direction and a plastic coating applied to their surfaces in order to reduce eddy currents. This has controlled the production of heat and kept the temperature of the magnets from exceeding the upper limit temperature, enabling the achievement of the power target.

Because the motor and gearbox have been integrated in the MCF31 in order to reduce the size of the drive unit, gear oil, used to lubricate the bearings of the primary shaft and drive shaft, is present in the motor. For this reason, in addition to insulation enabling eddy currents to be reduced and anti-corrosion performance, oil resistance was one of the qualities sought from the surface treatment applied to the magnets. An inorganic surface reforming treatment with superb oil resistance properties was therefore applied to the surfaces of the magnets, and the insulating surface treatment was applied over this treatment. Figure 11 shows the relationship between the number of divisions in the magnet and the target temperature, and Fig. 12 shows part of the divided magnet configuration inside the rotor.

3.5. Stator Cooling Technology

In order to help ensure a broad range of continuous operation, from hill-climbing to top speed, copper loss was reduced, and thermal resistance was reduced in the stator. Thermal resistance in the pathways for heat flow from the coils to the water jacket positioned on the outer circumference of the stator is known to be almost entirely dependent on the area of contact between the coils and the stator⁽⁷⁾. Eight poles have been employed in the rotor of the MCF31 in place of the 12 employed in the MCF21 in order to reduce motor loss. This reduced the area of contact between the coils and the stator, which increased thermal resistance and resulted in a decline in cooling performance. The ratio of the number of slots to the number of poles (n) was therefore increased from $3n$, as

in the MCF21, to $6n$ (Fig. 13) in order to reconcile the reduction of phase resistance (loss) with an increase in the contact area between the coils and the stator, enabling target cooling performance to be achieved.

This has enabled the temperature of the coils to be controlled to within the range of allowable temperatures

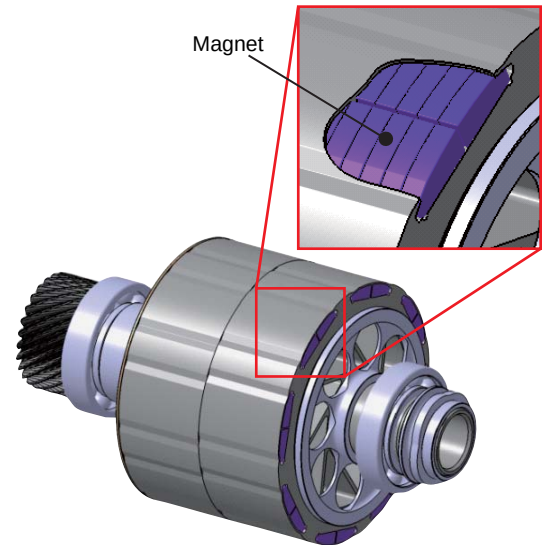


Fig. 12 Part of magnet division structure in rotor

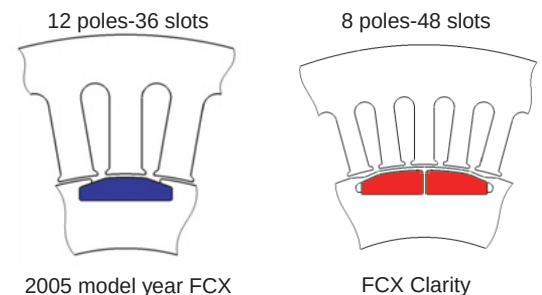


Fig. 13 Ratio of slots and poles

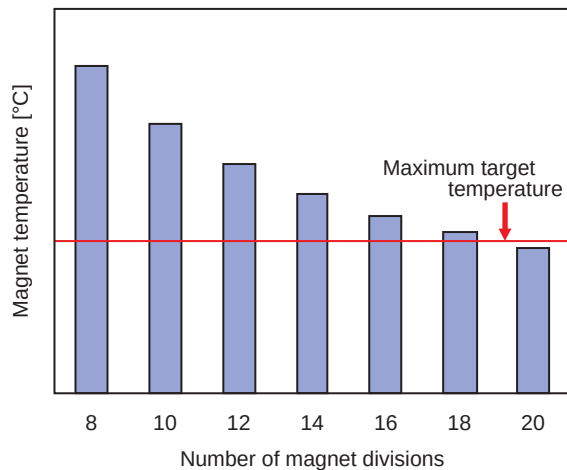


Fig. 11 Relation between number of magnet divisions and temperature

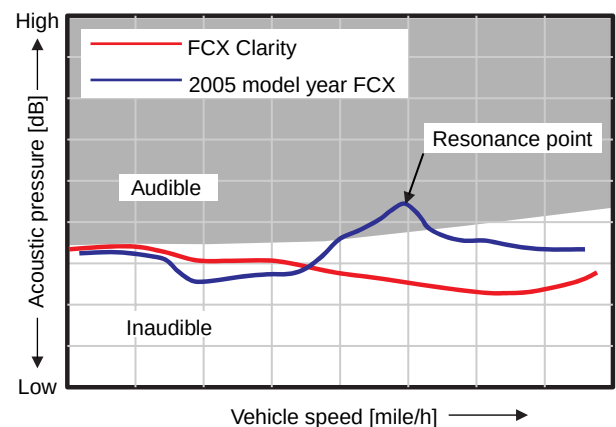


Fig. 14 Noise and vibration performance

for heat resistance in each section even when current is being continuously supplied when the vehicle is climbing a hill or the motor is being operated at maximum speed, conditions which represent a challenge for cooling.

3.6. Motor Noise and Vibration Performance

A level of noise and vibration performance was sought from the new motor that would enable the realization of the quietness only available from a motor drive – which features none of the vibration generated by reciprocating pistons or the combustion noise that characterize internal combustion engines - up to maximum speeds.

The use of eight poles in the MCF31 rotor in place of the 12 used in the MCF21 helped to eliminate resonance points in the use range of the motor (Fig. 14), enabling quietness to be reconciled with increased power.

4. Motor Performance

Table 2 shows the achieved performance of the MCF31. The motor's maximum efficiency is identical to that of the MCF21, but its torque density is more than 15% higher and its power density is more than 20% higher, while its weight is more than 20% lower.

4.1. Power

Test results indicated that the MCF31 motor achieves its target performance of a maximum power of 100 kW, maximum torque of 256 Nm, and maximum speed of 12 500 min^{-1} . The results are shown in Fig. 15. The range at which the motor's maximum power of 100 kW can be output extends from the mid-speed- (approximately 30 mile/h) to high-speed-range (approximately 80 mile/h), enabling the realization of take-off acceleration and passing acceleration performance equivalent to or higher

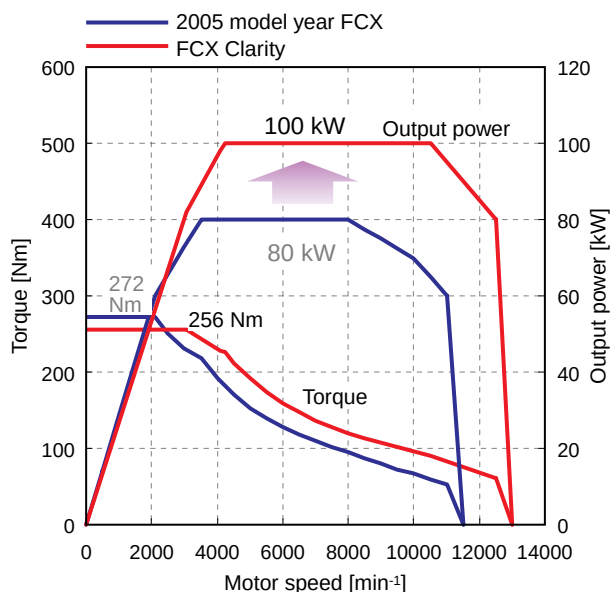


Fig. 15 Motor output performance

Table 2 Motor performance

Type	MCF21 (2005 model year FCX)	MCF31 (FCX Clarity)
Maximum power [kW]	80	100
Maximum torque [Nm]	272	256
Maximum speed [min^{-1}]	11000	12500
Maximum efficiency [%]	97	97

than those of a 2.4 L-class gasoline vehicle as shown in Fig. 16. In addition, as a result of the motor's outstanding cooling performance, the range for continuous rated output extends from hill climbing through city driving to top speed.

4.2. Energy Efficiency

With the achievement of a maximum efficiency of 97% during motor operation and a 36% reduction in drive unit friction loss, high efficiency operation has been realized in the drive unit. In addition, as Fig. 17 shows, optimization of energy management has enabled the

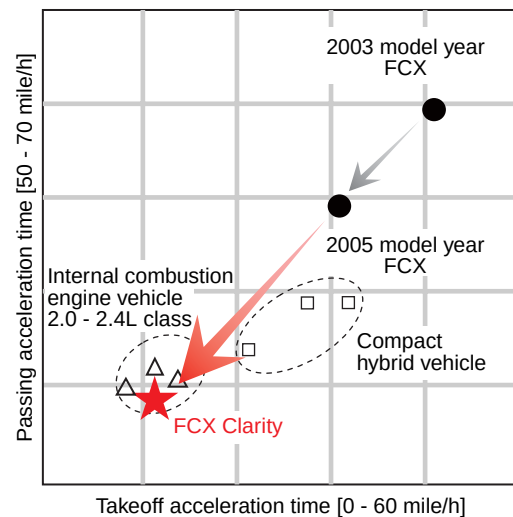


Fig. 16 Acceleration performance

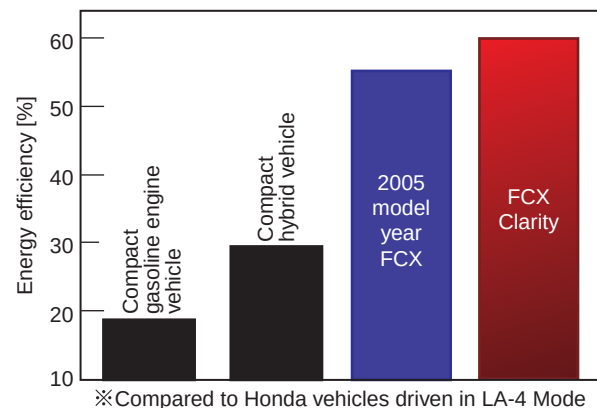


Fig. 17 Energy efficiency, LA-4 Mode

achievement of the target of an energy efficiency of 60%, approximately 10% higher than the 2005 model year FCX.

5. Conclusion

A drive motor was newly developed for the FCX Clarity, and has been employed in the vehicle. This development produced the following results.

- (1) Integrating the motor, gearbox, and power drive unit reduced the size of the drive unit, and enabled the realization of a design featuring a short nose and low overall height.
- (2) Increasing motor efficiency and reducing friction in the drive unit resulted in increased transmission efficiency, contributing to the achievement of an energy efficiency of 60%.
- (3) Reducing inductance in the motor enabled a power output of 100 kW to be maintained up to high speeds, enhancing the dynamic performance of the vehicle.
- (4) Increasing the strength of the rotor yoke enabled the rotor to be operated at higher speeds, resulting in the achievement of increased maximum vehicle speed.

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Hydrogen Visualization Using Multi-photon Ionization

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ABSTRACT

Research has been conducted on a hydrogen visualization technology in order to provide support to the process of research and development on fuel cell vehicles by “making invisible hydrogen visible.” The project eschewed conventional approaches such as indirect measurement of hydrogen using chemical sensors or substitute gases. Instead, it attempted to directly detect hydrogen, and to enable visualization of the state of diffusion of hydrogen in space.

The use of a laser was selected from among a variety of hydrogen detection methods, as its use is considered to be effective in detection at low concentrations. The project focused on a principle by means of which, contrary to conventional fluorescence analysis, fluorescence would occur even if the laser light for excitation did not match the absorption wavelength, and proposed a theory for hydrogen detection.

Results confirmed that the use of the principle of multi-photon ionization enabled direct fluorescence and detection of hydrogen. In addition, results suggested that the technology has the potential to be employed to visualize the changes in the state of diffusion of hydrogen in space as the flow rate changes, enabling analysis and measurement of the state of hydrogen diffusion.

1. Introduction

Honda is conducting R&D to promote more widespread use and enable mass production of vehicles powered by fuel cells, which the company regards as the ultimate next-generation clean powerplant, in order to provide an alternative to fossil fuels, reduce exhaust emissions, and ameliorate the automotive contribution to global warming.

Fuel cell vehicles use hydrogen as fuel, and therefore necessitate technologies enabling monitoring of the flow and motion of hydrogen, which have not previously been necessary in gasoline vehicles. The first item of research has been a hydrogen visualization technology enabling hydrogen flow and motion to be directly monitored.

Conventional fluorescence analysis methods are available for the visualization of gases, but they are ineffective in the case of hydrogen atoms. Research has therefore been conducted on camera imaging following the direct excitation and subsequent light emission of hydrogen atoms. However, an excitation energy of 13.6 eV or higher is necessary for hydrogen atoms to emit light, necessitating the use of large-scale devices.

A hydrogen visualization technology employing a single laser and using multi-photon ionization has therefore been developed to enable direct light emission from hydrogen atoms⁽¹⁾.

2. Development Goals

Looking towards the advent of the coming hydrogen society, the goal of the project was established as the realization of a visualization technology which would “make invisible hydrogen visible.”

The development concept for the hydrogen visualization technology was as follows:

- (1) Able to directly detect hydrogen
- (2) Able to visualize the concentration distribution of hydrogen when it is diffused in space

3. Background to Use of Multi-photon Ionization Method

3.1. Selection of Method for Detection of Hydrogen

Table 1 shows the methods generally used to detect hydrogen⁽²⁾. Hydrogen disperses rapidly in space, and its

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Table 1 Summary of evaluation of technologies

Technique	Distinguishing characteristic	Applicability
Passive sonar	Turbulent airflow (ultrasound)	Demonstrated Low spatial resolution Sensitivity depends on pressure and aperture
Raman-spontaneous	Allowable energy levels	H ₂ : Sensitivity of 2% demonstrated He: None - monatomic, therefore no vibration
Raman-CARS	Allowable energy levels	H ₂ : Sensitivity of 10 ppm demonstrated He: None - monatomic, therefore no vibration
Rayleigh doppler	Molecular/atomic velocities	Theoretically applicable for both H ₂ and He
Indirect thermal	Temperature	Limited by clutter?
Shearography	Index of refraction	Sensitivity limited by clutter

concentration subsequently diminishes. Different methods detect hydrogen on the basis of sound waves, temperature, mass, index of refraction, and band gaps. Among these methods, the CARS method achieves a lower limit concentration of 10 ppm. Given this, it was considered that a measurement method using an optical laser would be effective in enabling detection at low concentrations.

3.2. Calculation of the Lower Limit Concentration for Hydrogen Detection

As Table 2 shows, at normal temperature and pressure, hydrogen is a colorless, transparent, and odorless gas. Its diffusion coefficient in the atmosphere is $7.78 \times 10^{-5} \text{ m}^2/\text{s}^{(2)}$. These characteristics make it challenging to study the flow and motion of the gas. Substitute gases can be employed to enable study of flow and motion, but the fact that their mass differs from that of hydrogen means that accurate values are not obtained. The necessary lower limit of concentration for a hydrogen visualization technology to enable the flow and motion of hydrogen to be studied was calculated.

The percentage concentration of hydrogen when it diffuses in a large area at a diffusion coefficient of $D \text{ m}^2/\text{s}$ over a distance of $r \text{ m}$ for a time of $t \text{ sec}$ is shown by the following differential equation⁽²⁾:

$$\frac{\partial C(r, t)}{\partial t} = D \frac{1}{r^2} \frac{\partial}{\partial r} \left(r^2 \frac{\partial C(r, t)}{\partial r} \right) \quad (1)$$

Equation (2) is a model of the diffusion of hydrogen at a flow rate of $\phi \text{ ml/min}$ from an ideal punctiform hole.

$$C(r, t) = \frac{\phi}{4\pi Dr} \operatorname{erfc} \left[\frac{r}{\sqrt{4Dr}} \right] \quad (2)$$

Equation (3) shows the equilibrium concentration after the elapse of infinite time.

$$C(r) = \frac{\phi}{4\pi Dr} \quad (3)$$

Figure 1 shows equilibrium concentrations for

Table 2 Comparison of physical properties of gasoline and hydrogen

	Gasoline	Hydrogen
State at normal temperature	Liquid	Gas
Color	Orange	Colorless
Odor	Irritating odor	No odor
Diffusion coefficient in atmosphere [cm ² /sec]	0.05	0.778

hydrogen diffusing into the atmosphere from an ideal punctiform hole at various flow rates ($\phi \text{ ml/min}$) over a distance $r \text{ m}$, at a diffusion coefficient of $D_{\text{H}_2\text{-Air}} = 7.78 \times 10^{-5} \text{ m}^2/\text{s}$, a temperature of 25°C and a pressure of 101.3 kPa .

Calculation of the lower limit concentration for hydrogen detection assumed the use of a commercially available laser, and the distance at which it was able to approach an ideal punctiform hole was set at 5 mm , corresponding to the radius of the laser. The lower limit for the controllable flow rate was set at 0.01 ml/min .

The hydrogen concentration at a flow rate of 0.01 ml/min and distance of 5 mm was calculated to be approximately 40 ppm . With this as the lower limit

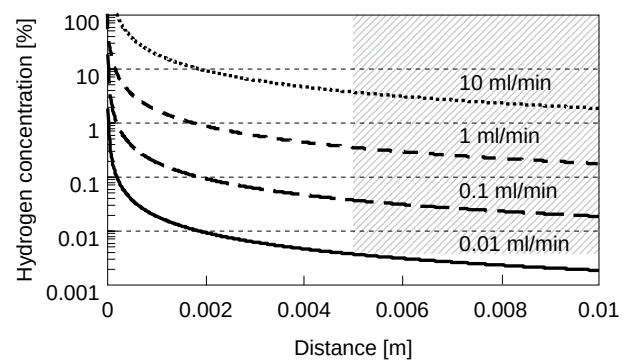


Fig. 1 Concentration of hydrogen at steady state

concentration for detection, it would be possible to detect the diffusion of hydrogen in the shaded section of Fig. 1.

3.3. Issues of Hydrogen Detection by Fluorescence Analysis

In standard fluorescence analysis, it is necessary to use a laser wavelength that matches the absorption wavelength of the band gap of the atoms or plasma being studied. Because of this, the wavelength of the laser light should be changed according to the gas that is to be detected. In the case of hydrogen, even when the laser wavelength corresponds to the absorption wavelength of the band gap, the band gap of a hydrogen atom is 13.6 eV or more, while the absorption wavelength is 120 nm or less, in the extended vacuum ultraviolet range. Light in the extended vacuum ultraviolet range is instantly absorbed and attenuated in the atmosphere, making it challenging to focus the laser on the area of detection.

Multiple lasers are necessary in order to address this issue, making the equipment used in the method very large.

Raman-CARS and spontaneous Raman are also known as methods enabling optical measurements of gases. However, these methods are only useful for gases with molecules that can be made to vibrate or can be rotationally excited by laser beam irradiation. In the case of hydrogen molecules, irradiation by a laser beam with intensities of 13.6 eV or higher at a wavelength that does not match the absorption wavelength will result in the molecules dissociating. As a result, hydrogen will be present in its atomic state, and will not be able to be detected. This made it necessary to develop a method differing from conventional fluorescence analysis methods.

3.4. Resolution of Issues Using Multi-photon Ionization

Figure 2 shows the results of a hydrogen atom ionization test using the second harmonic wave of an Nd-YAG laser⁽³⁾. At intensities of 10^{13} W/cm² and higher, 100% of the hydrogen atoms were ionized. When the ionized hydrogen stabilized, excitation energy was released and fluorescence was produced.

The results shown in Fig. 2 led to a focus on the phenomenon of fluorescence of the atoms occurring even if the wavelength of the incident laser was not equivalent to the absorption frequency of the band gap, and the proposal of a theory concerning the detection of hydrogen concentration by means of this phenomenon. Figure 3 shows a model of this theory.

A laser with an energy density of 10^{13} W/cm² or higher in the perpendicular section of the optical axis irradiates the gas being measured. Even if the wavelength of the irradiating laser does not match the absorption wavelength, the use of a high-directivity, high-photon density laser results in simultaneous absorption of multiple photons, as shown by the multiple arrows, Y1, in Fig. 3. This excites

the atoms from normal energy levels to a continuous state, ionizing them. This is the phenomenon of multi-photon ionization⁽⁴⁾.

After the gas being measured has been excited in this way, the molecules de-excite from an upper energy level to lower energy level 2 or 1, as shown by arrows Y4, Y3, and Y2 in Fig. 3. When this occurs, the gas produces specific types of fluorescence, A and B. It was therefore considered that the quantification of fluorescence A and B would enable the concentration of the gas being measured to be calculated, and the flow and motion of the gas to be visualized by means of imaging.

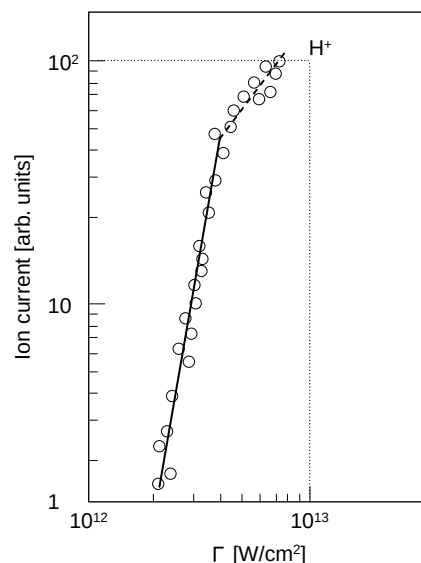


Fig. 2 Log-log plot of H⁺ from H₂ as function of laser intensity I

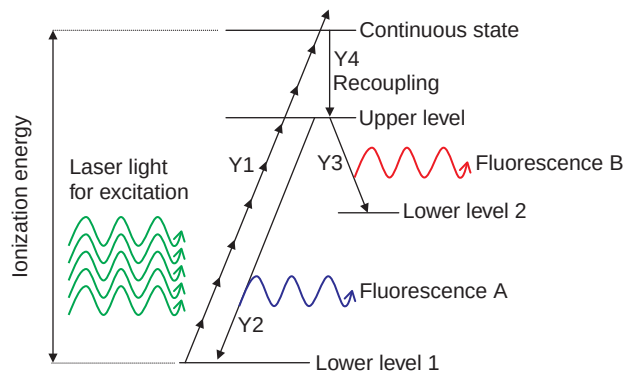


Fig. 3 Model of multi-photon ionization

4. Realization of Hydrogen Visualization Technology

The development of a high-peak laser with an energy density of 10^{13} W/cm² or higher in the perpendicular section of the optical axis was the key issue in the

realization of the principle of multi-photon ionization. With a standard laser energy density of 10^6 W/cm², the necessary laser strength would be from 100 to 1 000 000 times higher than that used in standard fluorescence spectroscopy.

In addition, because the principle indicated that all of the gas at the focal point of the beam would ionize and fluoresce, if a CCD camera was simply used without any extra considerations, the fluorescence produced by all the gas at the focal point would be photographed. This would represent an impediment to visualization, because the results would not clearly indicate where the hydrogen was present. The next sections will discuss the measures used to resolve these issues.

4.1. Achievement of Increased Laser Energy Density

The equation employed to calculate the rate of hydrogen fluorescence per pulse (N) in the development of a high-peak strength laser with an energy density of 10^{13} W/cm² or higher is shown below⁽⁵⁾.

$$N \propto \left(\frac{P}{hvA\Delta t} \right)^\alpha \times (n_{H_2}AL) \times \sigma(\theta) \times (\Omega \times \varepsilon_0 \times \varepsilon_d) \times \Delta t \quad (4)$$

Laser power: P [J]

Energy per photon: hv

Laser focal area: A [m²]

Pulse width: Δt [sec]

Concentration: n_{H_2} [%]

Thickness: L [1/m³]

Probability of fluorescence: $\sigma(\theta)$ [m²]

Solid angle: Ω [sr]

Efficiency of optical system: ε_0

Efficiency of detection equipment: ε_d



Fig. 4 Pulse compressor

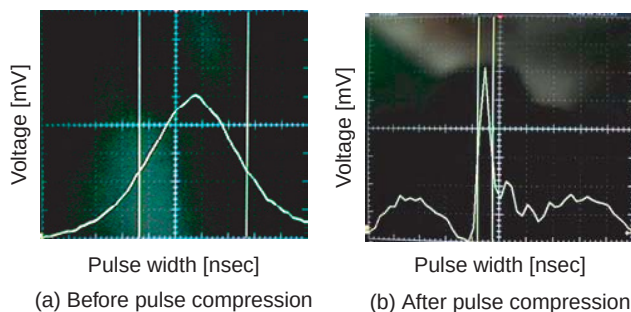


Fig. 5 Test of pulse compression

As Eq. (4) shows, as the laser power is increased, the focal area is restricted and the pulse width is shortened, increasing energy density.

Laser power depends on the wavelength of the irradiating beam, and higher energy per photon is effective. From this perspective, in the case of an Nd-YAG laser with a fundamental wave of 1 064 nm, the 213 nm fifth harmonic wave would be effective. However, conversion using a nonlinear optical crystal results in conversion loss, reducing total energy. The 532 nm second harmonic wave was therefore selected as being more effective from the perspective of total energy.

The pulse compressor shown in Fig. 4, which uses liquid as a medium, was developed and employed to compress the pulse width. Focusing the beam by means of a concave mirror and using the phenomena of backwards Brillouin scattering and stimulated Brillouin scattering enabled the pulse width to be compressed as shown in Figs. 5(a) and (b).

A lens that narrowed the focus area was developed and employed in order to increase energy density.

4.2. Imaging Hydrogen Fluorescence

A bandpass filter was constructed that would filter everything but the wavelength representing the characteristic emission spectrum of hydrogen from the irradiating laser light and all the fluorescence produced by the gas at the focal point of the laser (Fig. 6). This spectroscopic technique enabled only the characteristic emission spectrum of hydrogen to be imaged.

Tests of the wavelength filtering performance of the bandpass filter were conducted. In Fig. 7(a), the fluorescence produced by all the gas at the focal point of the laser is measured, but the application of the bandpass filter enables only the characteristic emission spectrum of hydrogen to be measured, as shown in Fig. 7(b).

4.3. Overview of Hydrogen Visualization System

The developed hydrogen visualization system is made up of three major sub-systems. The sub-sections below provide an overview of these sub-systems.

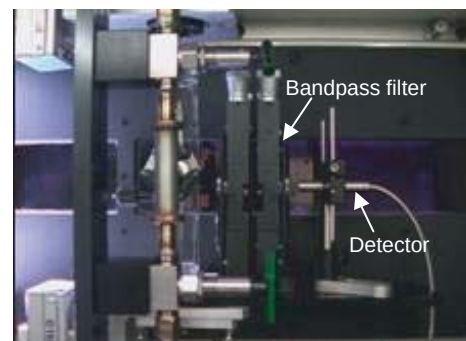
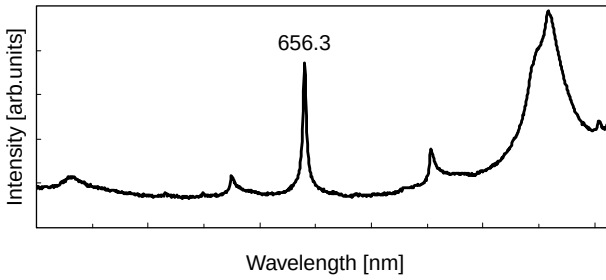


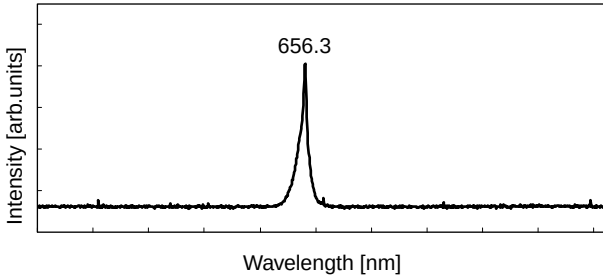
Fig. 6 Bandpass filter

4.3.1. Multi-photon ionization generating system

The multi-photon ionization generating system is shown in Fig. 8(a). A Q switch Nd-YAG laser enabling a top-hat beam profile to be obtained is employed. The laser has a wavelength of 532 nm, an energy of 425 mJ, a repeat cycle of 10 Hz and a pulse width of 6 nsec. An optical phase conjugator is incorporated in the center of the post-oscillation beam path. The optical phase conjugator maintains the energy density distribution of the pulse laser by means of optical inverse transformation of the laser light. The system also employs, as an optical amplifier, a pulse compressor that compresses the pulse width. A focusing lens is positioned at the end of the beam path.



(a) Before application of bandpassfilter: All wavelength



(b) After application of bandpass filter: Only characteristic wavelength of hydrogen

Fig. 7 Test of bandpass filter performance

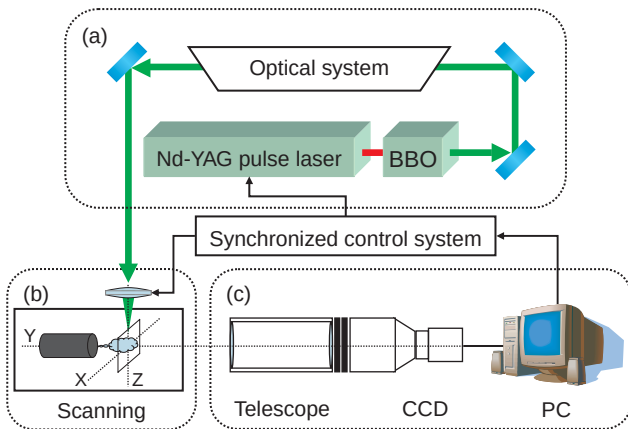


Fig. 8 Outline of hydrogen visualization system

4.3.2. Scanning system

The scanning system is shown in Fig. 8(b). This is composed of a mechanism for shifting the focal point of the laser (a focusing lens and mirrors incorporated into a robot) and a mechanism for adjustment of the measurement environment. Planar measurement is enabled by opening the exposure of the CCD camera in the imaging system and shifting the focal point of the laser without changing the strength of the laser across the range of measurement. This has enabled planar imaging of the flow and motion of hydrogen.

4.3.3. Imaging system

The imaging system is shown in Fig. 8(c). It is made up of an apochromat telescope, a bandpass filter and a CCD camera. The system is shielded from the irradiating laser light and stray light, and the CCD camera therefore images only the characteristic emission spectrum of hydrogen produced at the focal point of the laser. The use of a Peltier device has enabled the CCD sensors to be cooled to below -30°C in order to reduce thermal noise. This has increased the sensitivity of the sensors by approximately 1 000-fold against sensors at normal temperature, enabling imaging of even faint fluorescence. A PC conducts image processing of the data obtained by the camera, and synchronously controls the various systems.

5. Verification and Testing Using Hydrogen Visualization System

5.1. Verification of the Principle of Multi-photon Ionization

The developed system was used to verify the principle of multi-photon ionization. For the tests, hydrogen enclosed at 101.3 kPa within a silica tube was irradiated with an Nd-YAG laser at a wavelength of 532 nm, 100 mJ of energy and pulse width of 2 nsec.

The fluorescence of the hydrogen at the focal point of the laser was studied, as shown in Fig. 9. Figure 10 shows the results of a qualitative analysis conducted using a spectrometer. The wavelength of the fluoresced light is 656.3 nm, the characteristic wavelength of hydrogen.

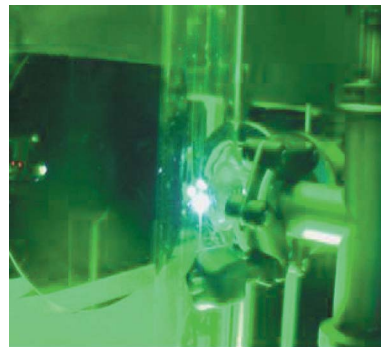


Fig. 9 Moment of hydrogen fluorescence

This result verifies the fact that the multi-photon ionization method is able to directly produce fluorescence of hydrogen atoms.

Next, a test was conducted to determine the lower limit hydrogen concentration that could be detected. Hydrogen was enclosed at known concentrations in a silica tube at 101.3 kPa. The hydrogen was irradiated by the laser under identical oscillating conditions as employed in the previous test. Figure 11 shows results obtained as the concentration of the hydrogen was shifted towards lower concentrations. A sensitivity of up to 30 ppm of hydrogen was verified by the spectrometer.

5.2. Tests of Hydrogen Visualization

Tests of visualization of hydrogen dispersing from a single hole were conducted in order to verify the ability of the system to determine the distribution of hydrogen concentration when dispersed in space. A hole with a diameter of 5 mm was pierced in the center of a stainless steel flange 40 mm in diameter and 5 mm in thickness. The hydrogen was dispersed in a planar direction from the hole, and was irradiated in a perpendicular direction by a laser positioned 5 mm from the end of the flange as shown in Fig. 8(b). The range for imaging was set at a 60 mm square, and the exposure of the CCD camera was open from the first to the last laser pulse. The laser was pulsed at vertical intervals of 5 mm and horizontal intervals of 2 mm. The hydrogen was dispersed at flow rates of 25 ml/min and 50 ml/min.

The results of the tests are shown in Figs. 12(a) and

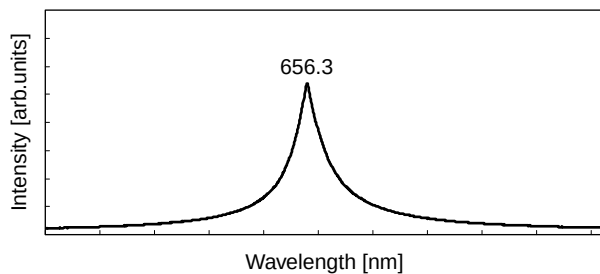


Fig. 10 Characteristic wavelength of hydrogen

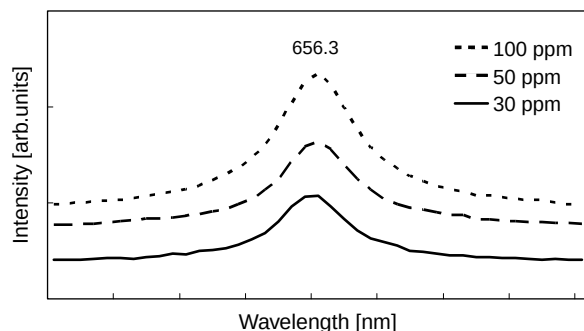


Fig. 11 Minimum limit for detection of hydrogen concentration

(b). These results indicate that it is possible to photograph and visualize hydrogen dispersing upwards from a single hole. The colored portions of the images are hydrogen fluorescence produced by the laser. The color bars in Fig. 12 show the relative hydrogen concentration. Figures 12(a) and (b) also indicate that it is possible to visualize changes in hydrogen dispersion that occur with changes in the flow rate.

5.3. Tests of Application of Hydrogen Visualization System

Next, in order to test the performance of the system in visualizing the flow and motion of hydrogen under conditions more closely approximating reality, tests were conducted of the visualization of hydrogen dispersing from multiple holes.

As shown in Fig. 13(a), 12 holes of approximately 30 μm in diameter were pierced in an H-shaped pattern in a stainless steel vacuum flange of 100 mm in diameter and 10 mm in thickness. Hydrogen was dispersed from each hole at 1 ml/min, for a total of 12 ml/min.

The hydrogen was irradiated by a laser positioned 5 mm from the end of the flange, using identical oscillating conditions to those employed in the previous tests. The results are shown in Fig. 13(b).

The results indicate that the system is able to visualize the dispersion of hydrogen not only from a single hole, but from multiple holes.

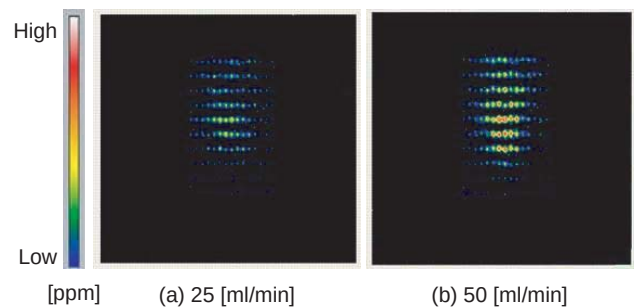
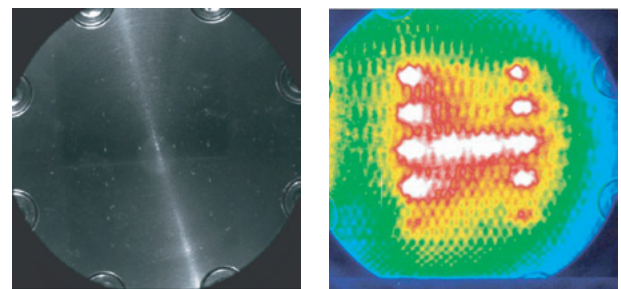


Fig. 12 State of hydrogen diffusion from single point



(a) Stainless steel vacuum flange

(b) Visualization of “H”

Fig. 13 State of hydrogen diffusion from multiple points

6. Conclusion

Using the multi-photon ionization method, the research discussed in this paper has taken the first step towards the important achievement of enabling visualization of hydrogen. The following knowledge was obtained.

- (1) The project demonstrated that the principle of multi-photon ionization could be used to enable direct fluorescence and detection of hydrogen.
- (2) It was possible to image and visualize the concentration distribution of hydrogen dispersed in space using a CCD camera.
- (3) Tests verified that the developed system was able to visualize changes in the state of dispersion of hydrogen occurring as a result of changes in the flow rate.
- (4) The results of the project suggest that the multi-photon ionization method can be employed in technologies for the analysis and measurement of hydrogen dispersion.

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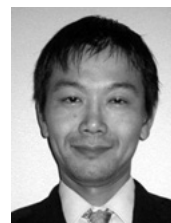
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Development of GLOBAL Hybrid Car Aimed at Popularization

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ABSTRACT

Amidst rising worldwide concern for the environment and the need to conserve energy, a special hybrid model, aimed at popularization, has been developed.

This model features a newly developed Ecological Drive Assist System that enhances fuel economy, an advanced exterior design that takes into account aerodynamics, an emotional interior design, and a package with greater utility achieved by a lightweight and compact hybrid system.

This hybrid car enhances fuel economy, provides refreshing everyday driving performance, and realizes a top level of safety performance, making it a car that can delight customers around the world.

1. Introduction

The world currently faces the issues of global warming and increases in oil prices, and the reduction of CO₂ in particular is an important issue. Cars with even better fuel economy and lower CO₂ emissions are therefore desirable. However, not only are electric vehicles and fuel cell vehicles still expensive, but infrastructure issues still exist for those vehicles, so the era of the hybrid car is predicted to continue for some time. This means that to reduce CO₂, it is important to make hybrid cars available at easier-to-afford prices, so more customers will be able to purchase them.

Therefore, efforts were made to develop a GLOBAL hybrid car aimed at popularization, namely the new model INSIGHT, which can contribute to the global environment by reducing CO₂ emissions.

In addition, this paper also introduces an assist system that enhances fuel economy, one that has newly been developed to enhance fuel economy as much as possible.

2. Development Concept

When establishing the development concept, the intention was to create a next-generation hybrid car that would serve as a benchmark for 10 years. With that in

mind, the grand concept was set as “Next Era Transportation (A Small Car Standard for a New Era)”.

To achieve this, development was promoted based on the following four items:

- (1) Clean and Green (Environmentally friendlier)
Good fuel economy, low exhaust emissions
- (2) Usable
Wide cabin and trunk space for a compact class vehicle
- (3) Fun
Exhilarating everyday performance, exclusive hybrid body
- (4) Affordable

3. Environmentally Friendlier Technology

3.1. Development Goals for Ecological Drive Assist System

Fuel economy is affected by traffic conditions and various driving operations by users. An assist system that can help enhance fuel economy (Ecological Drive Assist System) was developed to reduce the effects of traffic conditions and user driving operations, and to enable users to deepen their understanding about fuel-efficient driving.

3.2. System Overview

The Ecological Drive Assist System (Eco Assist) is comprised of two elements: ECON control that

* Automobile R&D Center

automatically controls the power train and air conditioning system with a priority on fuel economy, and a coaching system that informs users of fuel-efficient driving conditions and deepens understanding of driving that enhances fuel economy.

Figure 1 shows an overview of the hardware configuration.

- (1) ECON switch
- (2) Ambient meter
- (3) Eco guide (displayed on the multi information display)
- (4) Eco guide (displayed on the NAVI screen with Japanese specifications)

3.3. ECON Control

When the ECON switch is ON, control of the engine, motor, CVT, and air conditioning system switches to fuel economy priority mode, and the effects of traffic conditions and user driving operations on fuel economy

are reduced.

An overview of control when the ECON switch is ON is as follows.

- (1) Lowering the target value of the torque for the accelerator pedal pressing amount
- (2) Limiting the throttle valve opening
- (3) Limiting the CVT gear ratio change response
- (4) Increasing the amount of regeneration during deceleration
- (5) Saving energy consumption of the air conditioning system

3.4. Coaching System

The coaching system has a three-step configuration consisting of coaching that informs users of the fuel economy condition in real time, feedback that informs of the resulting eco-driving rate, and teaching that advises each user on points for enhancing fuel economy.

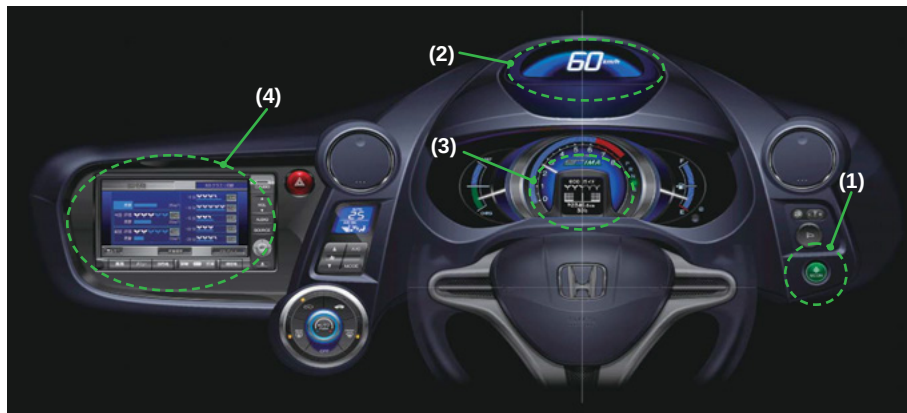


Fig. 1 Overview of ecological drive assist system

Driving condition	Coaching display	
	eco GUIDE (in multi-info. display)	Ambient meter
Efficient cruising Optimal environmental driving		
Optimal environmental deceleration		
Moderate acceleration		
Moderate deceleration		
Aggressive acceleration		
Quick deceleration		

Fig. 2 Ambient meter

(1) Coaching

This function enables users to easily understand the fuel efficiency at any given time by changing the speedometer background color (ambient meter) in three steps (blue, blue-green, green) according to the eco-driving rate of accelerator and brake operations. In addition, an eco guide display is provided as a multi information display screen, and shows the eco-driving rate of accelerator and brake operations by eco drive bar (Fig. 2).

(2) Teaching

The user's eco-driving rate is scored centering on accelerator and brake operations, and the results are displayed as the number of "leaves" using a plant leaf motif in the above-mentioned eco guide display (Fig. 3). In addition, the system works to maintain and increase

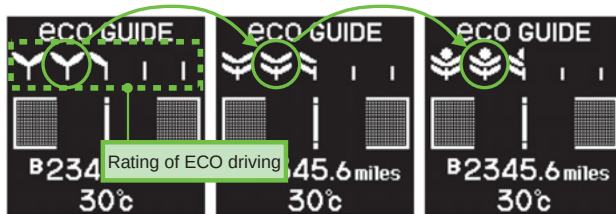


Fig. 3 ECO guide display

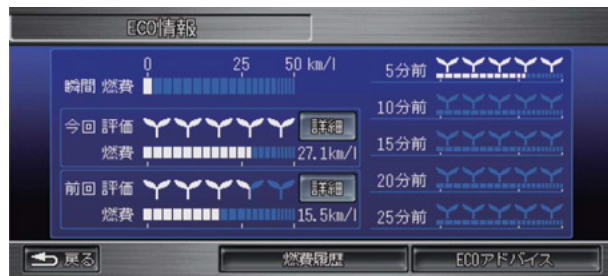


Fig. 4 NAVI display

Table 1 Mode fuel economy

Destination		Mode FE	Remarks
US	CITY	40 mpg	EPA label FE
	HWY	43 mpg	EPA label FE
EU		4.4 L/100km CO ₂ 101 g/km	
J	10 · 15	30 km/L	
	JC08	26 km/L	

motivation towards eco-driving by showing the leaves growing in three steps according to the eco-driving mastery level of the user.

(3) Teaching (Japan version only)

Detailed fuel economy information, a breakdown of the above-mentioned eco-driving rate score, and advice on driving methods that further enhance fuel economy are displayed using a NAVI screen available as a factory-installed option in order to deepen user understanding about driving that enhances fuel economy (Fig. 4).

In addition, a system was constructed that can provide similar information as the above, even to users without a NAVI screen installed, by using the link between the HDS (Honda Diagnostic System) terminal and a PC.

Various information is also provided on a web page via a link with the internavi system, such as fuel economy ranking, history of eco-driving rate mastery, and detailed advice on driving that enhances fuel economy, and the like. This creates an environment that lets users approach eco-driving while having fun.

3.5. Technology for Increasing Mode Fuel Economy

As seen in Honda's other hybrid models, this model is equipped with a simple, compact, and high efficiency IMA (Integrated Motor Assist) system, and efforts were made to enhance fuel economy compared to gasoline engine cars.

Table 1 shows the mode fuel economy for each destination.

The 1.3 L i-VTEC engine mounted in this model incorporates the following technologies for enhancing combustion and reducing both friction and knocking, while increasing standalone fuel economy performance by 2%:

(1) Combustion enhancing technologies

- Exhaust valve timing delay
- Highly ignitable double fine electrode spark plugs

These combustion enhancing technologies provided stable combustion, realized the introduction of high volume EGR (Exhaust Gas Recirculation), and reduced BSFC (Brake Specific Fuel Consumption).

(2) Friction reducing technologies

- Piston skirt MoS₂ pattern coating
- Optimized piston skirt shape

- Low-tension piston rings
- Main bearing MoS₂ shot peening
- Cylinder liner plateau honing
- Reduced cam chain width
- Reduced cam journal surface roughness
- Reduced cam journal contact area

These friction reducing technologies successfully reduced friction by 2% compared to the conventional IMA engine.

(3) Knocking reducing technologies

Compared to a conventional IMA engine, this model uses a main coolant flow circulation route that first flows in a single direction around the cylinder block, and then flows in a single direction inside the cylinder head. In addition, the heat transfer rate was increased by speeding up the coolant flow rate around the combustion chamber. These measures helped to suppress knocking compared to the conventional IMA engine.

3.6. Aerodynamics

Aerodynamic performance was enhanced to aim for an

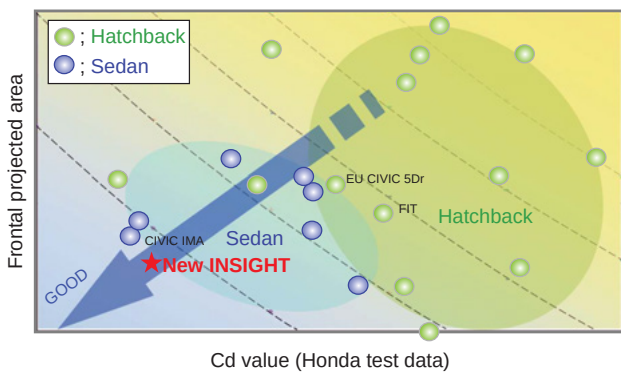


Fig. 5 Aerodynamic performance

environmentally friendlier car (good fuel economy, low exhaust emissions), and achieved an aerodynamic performance that is among the top class in the world (Fig. 5).

3.6.1. Exterior design

In keeping with the grand concept of an environmentally friendlier car (good fuel economy, low exhaust emissions), “Aero Athlete” was selected as the exterior design concept. The aim was to achieve an advanced style that organically integrates the aerodynamic performance developed through wind tunnel tests with a solid feeling sports car chassis (Fig. 6).

The front emphasizes the feeling of the hood extending outward in a single line from the cabin, and the flow from the bumper to the lower parts presents an advanced facial form unique to Honda.

The sides flow directly from hood to rear in an elegant yet strong mono-form that, together with a solid and speedy lower body, presents a sports car chassis with a powerful, solid feeling.

The rear also expresses a sports car chassis with a



Fig. 6 Exterior design concept

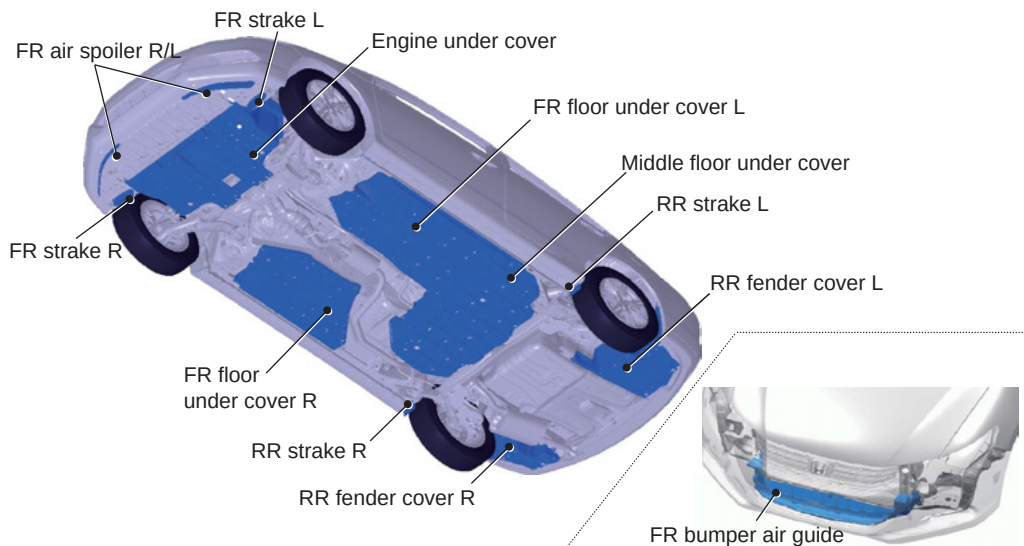


Fig. 7 Aerodynamic parts

narrowed cabin and a wide lower body, and the rear end of the cabin, door mirrors, and bumper lower edge have also been optimized through wind tunnel tests to realize high aerodynamic performance.

3.6.2. Aerodynamic devices

A front chin spoiler and front and rear strakes are used as aerodynamic devices to enhance the straightening effect of the underbody. In addition, the area from the engine room bottom to the floor, fuel tank bottom, and rear bumper is almost fully covered and flattened. Air guides are also set to enhance the straightening effect from the front bumper opening to the radiator (Fig. 7).

3.7. Air Cooling System

Dual layer AC pipes are used inside the engine room for the first time in any Honda car to enhance fuel economy. The dual layer structure uses an inner pipe with a spiral groove, and exchanging heat between the high pressure liquid and low pressure gas pipes increases the cycle efficiency by 4% (Fig. 8).

3.8. Low Emissions Technology

A high level of low emissions performance for each destination (Table 2) is realized by adopting various technologies such as a cylinder head with an integrated exhaust manifold, quick catalyst activation using a close-coupled catalytic converter, a UEGO (Universal Exhaust Gas Oxygen) sensor located in front of the close-coupled

Table 2 Mode emission

Destination		EM rating
US	KL	AT-PZEV
	KA	US BIN 3
EU		EURO 5
J		75% lower than 2005 standards

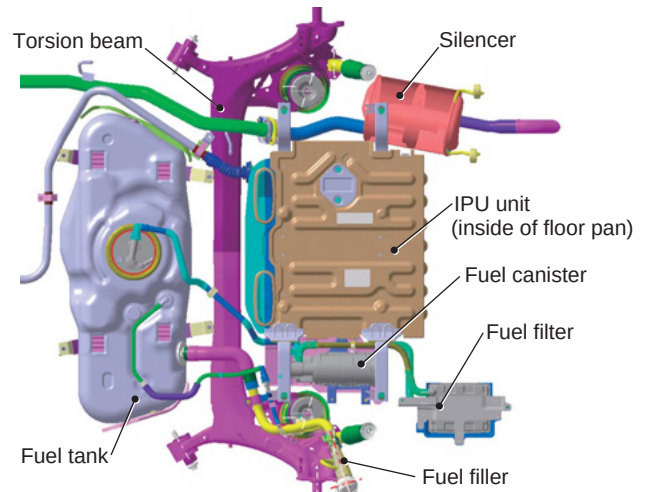


Fig. 9 Layout of fuel system

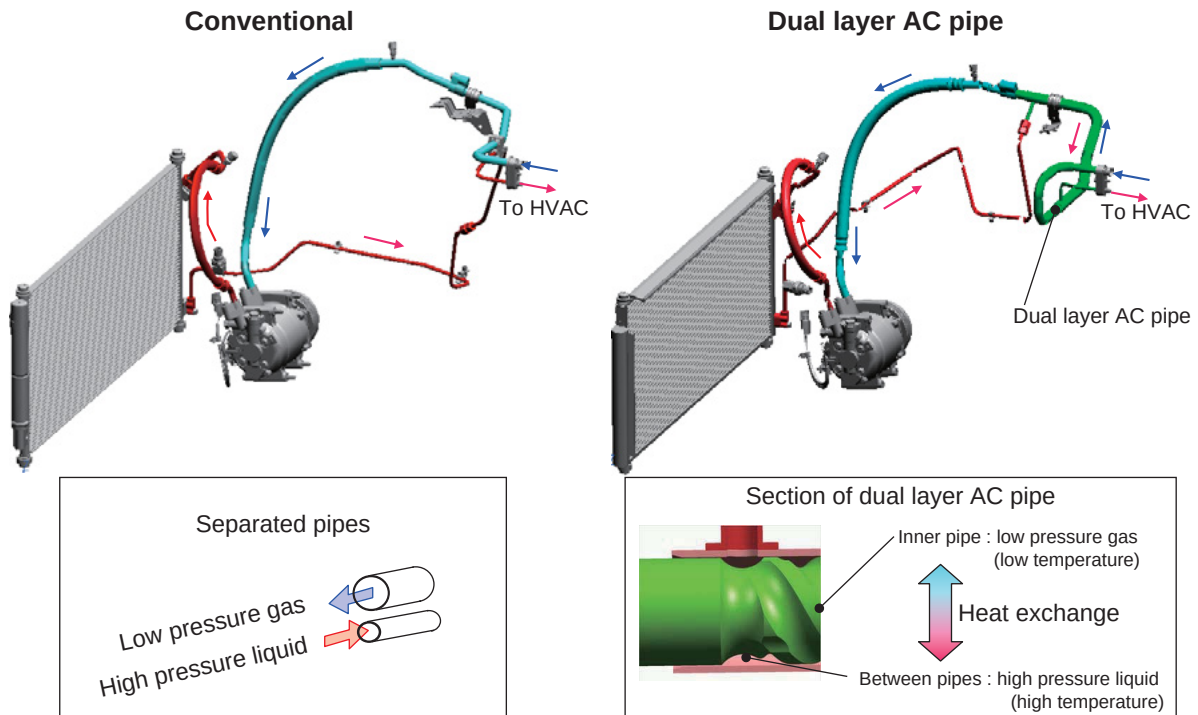


Fig. 8 Dual layer AC pipe

catalytic converter, a HEGO (Heated Exhaust Gas Oxygen) sensor located in front of the under-floor catalytic converter, precision adaptive control of the air-fuel ratio using DBW (Drive By Wire) and an AFM (Air Flow Meter).

3.9. ZERO-EVAP Fuel System

The fuel tank uses a resin tank that has a high degree of shape freedom and contributes to reducing weight. A lower overall height is realized by locating the fuel tank under the rear seat. In addition, a 40 L capacity combined with good fuel economy performance provides a cruising range that establishes the advantage of hybrid vehicles.

A large-sized fuel canister has been developed for the fuel system to meet the AT-PZEV standards in the U.S. A compact layout for the fuel tank, fuel canister, and fuel filler contributes to rear-end crash safety performance. At the same time, using a parallel layout, in which the IPU (Intelligent Power Unit) is located inside the floor pan, silencer and fuel canister under the floor, enables a flat trunk compartment. This contributes to increasing the product appeal of the trunk compartment (Fig. 9).

4. Aiming for Greater Utility

4.1. Package

The aim was to simultaneously achieve a low overall height with a reduced frontal projected area for enhanced fuel economy, and a spacious cabin interior in which adults can sit comfortably.

The platform is newly designed based on the 2008 model year FIT, and has a similar low floor structure to the FIT. However, the fuel tank location is changed from under the front seat to under the rear seat, which enables the seat position to be lowered, thereby securing sufficient headroom and providing a sporty driving position (Fig. 10).

4.2. IPU Package

The IPU is comprised of a battery and the PCU (Power Control Unit), and the overall size and weight have been

reduced by 19% and 28%, respectively, compared to the conventional unit. This enabled the IPU to be mounted under the trunk compartment floor, which achieved a hybrid car with low center of gravity and large trunk compartment equal to that of a normal passenger car.

In order to reduce the IPU size, the nickel-metal-hydride battery⁽¹⁾ performance is enhanced to secure the required output power, and the number of batteries is reduced for a smaller battery box size. The inverter and DC-DC converter of the PCU are positioned immediately next to each other, and the size is reduced so that all parts including the ECU can fit into a single case (Fig. 11).

4.3. Interior

4.3.1. Design

“Emotional Hybrid Interior” was selected as the interior concept to express an advanced feeling as “Next Era Transportation” in line with the development concepts of greater utility (Fig. 12).

The heater controls and other controls use a layout centered on the driver to increase driving pleasure.

The meters use a two-layer arrangement as seen in the 2006 model year CIVIC to present an advanced feeling,

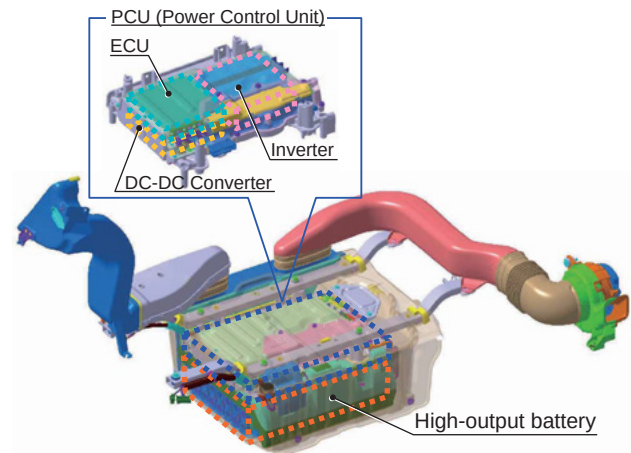


Fig. 11 Overview of IPU assy

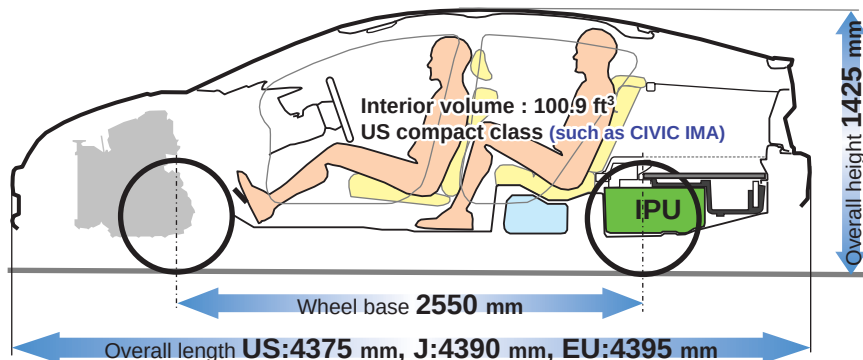


Fig. 10 Package design



Fig. 12 Interior design concept



Fig. 13 Rear seat

and visibility is enhanced by division into an area with information that is particularly important while driving (e.g., speed) and an area with other information. Cost was reduced by integrating the meter circuit board and case, which were separate in the CIVIC, into a single unit.

The instrument panel has a converse round shape with the center bulging outward towards the driver. This provides an expansive and refreshing feeling in addition to an advanced feeling.

The interior uses a bright and clean two-tone color scheme in the image of the environment.

4.3.2. Seats

The front seat shape and use of different hardness pads gives high holding performance, and a combination of stretchable covering and highly flexible seat surface due to deeper surface anchor positions provides softer seating comfort.

The rear seat can be folded down to create a flat and gap-free floor with the trunk compartment. In addition, the embedded headrests and built-in center 3-point seatbelts help maximize rearward visibility (Fig. 13).

4.3.3. Trunk compartment

To further emphasize the development concept of a “useful” wide trunk compartment, versatility is added to the instant mobility flat tire repair system (IMS) specifications. The trunk floor can be moved up and down in two levels to expand the height, as was favorably received in the European CIVIC. In addition, a box with lid is added and other measures are taken to help prevent small items from rolling around (Fig. 14).

For specifications with a spare tire, the under-floor storage is used as a storage space for small items (Fig. 15).

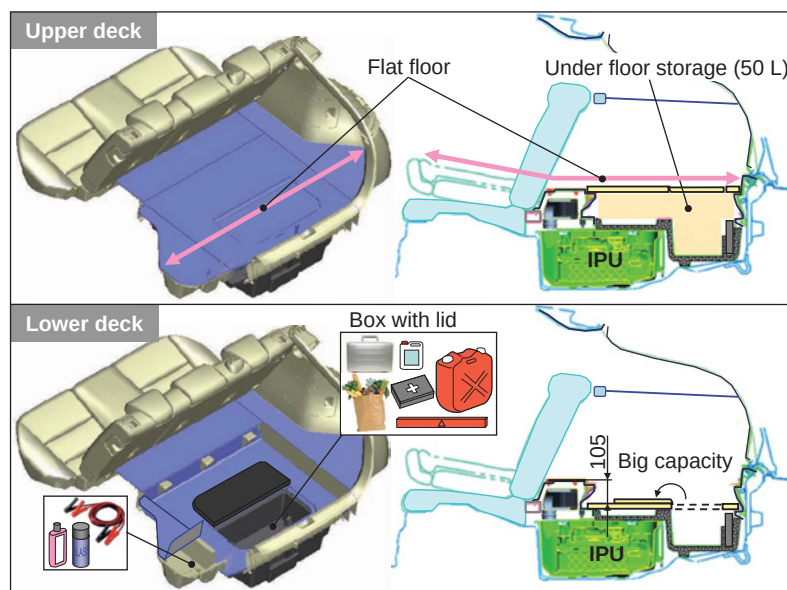


Fig. 14 Trunk with IMS

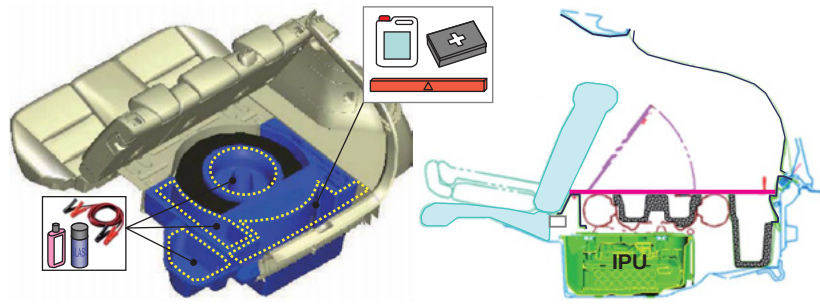


Fig. 15 Trunk with spare tire

5. Fun

5.1. Power Train Drivability

The power train of this model utilizes the high response that is a feature of the IMA system and the direct feeling of the starting clutch type CVT. In addition, the starting clutch contact timing control has been refined by collaboration between the engine, the IMA motor, and the CVT, thus realizing both fuel economy and brisk acceleration performance.

5.2. Chassis

5.2.1. Riding comfort

The front suspension uses a strut type, and the rear suspension uses an H-shaped torsion beam type. Stabilizers are used for both the front and rear suspension. The spring and damper characteristics use special settings, and the steering uses a high efficiency EPS and special settings. This achieved an organic balance between nimble and highly stable handling, smooth riding comfort, and a refreshing steering feel, and realized a pleasantly exhilarating riding feel in everyday driving.

The mount system that supports the power train uses the 3-point principal axis of inertial method with torque rods. Special engine mount and torque rod characteristics are set to optimize the mounting balance due to the use of the IMA system. In particular, a fluid sealing type trans-mount has been developed, which contributes to enhanced riding comfort.

5.2.2. Brakes

The brakes use negative pressure type boosters, and the pedal stroke and pressing force are optimized, realizing a pleasant and linear braking feel. As for the CAS (Creep Aid System), instead of a conventional separate CAS unit, software is added to the VSA unit to provide CAS functions, which reduces cost and weight (Fig. 16).

5.3. Body

5.3.1. Body rigidity

Body rigidity is an important factor that supports fun driving. Simulations were used to increase connection rigidity, which realized a lightweight and highly rigid

structure. Regarding the large cross section tailgate opening, the rear panel is enlarged and connected to the frame end to achieve a big tailgate opening design with good torsional rigidity. Regarding the rear damper mounts, the rear pillar damper stiffener and rear end gusset sizes are increased, which increases the connection efficiency of the opening cross section and the rear frame. In addition, regarding the rear compliance bush mounts, V-shape braces are applied to achieve both higher lateral rigidity of the compliance bush mounting points and the fuel tank layout (Fig. 17). Only the main items are noted

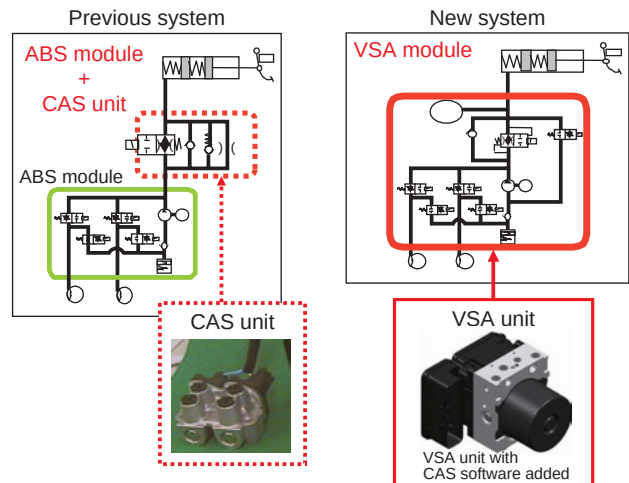


Fig. 16 CAS system

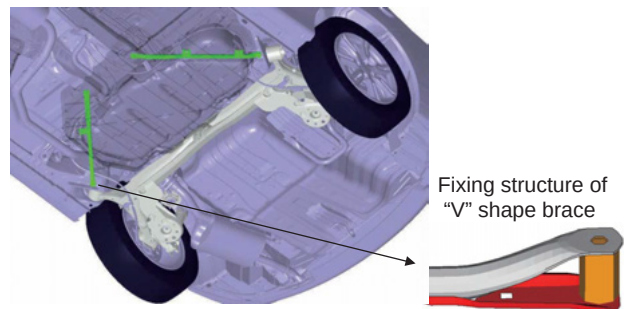


Fig. 17 Reinforcement of rear compliance

above, but as the result of these and other measures, the rigidity of the body mounting points shown in Fig. 18 is realized.

5.3.2. Sound insulation and IR cut

The combination of sound insulation and IR (Infrared Rays) cut glass is used for the first time in the world in a compact car (Fig. 19). Sound insulation and heat shielding functions are added to the intermediate film of the front windshield laminated glass. Thus, sound insulation effects of approximately -3.0 dB for high frequency sound of 1 kHz or higher are obtained. In addition, heat shielding effects equivalent to an increase in fuel economy of approximately 0.3 km/L (10-15 mode driving, outside air temperature 40°C, solar radiation

conditions 930 W/m²) when using the air conditioning during the daytime are obtained.

5.3.3. Lightweight soundproof package

Car body quietness is enhanced and weight is reduced by reviewing the layered structure of the conventional sound insulation materials, and by applying materials with good sound absorption performance. In addition, sound penetration is reduced by adding separators containing foamed filler to the base of each pillar on the bottom of the car body (Fig. 20).

5.3.4. Crash safety performance

Crash safety performance that is among the highest in its class is achieved for a frontal offset deformable barrier crash at 64 km/h, frontal flat barrier crash at 55 km/h, and side crash at 55 km/h with respect to the New Car Assessment Program (NCAP) of each country, including Japan's J-NCAP.

A compatibility support body⁽²⁾ is used for the front of the car to help reduce vertical or lateral misalignment of vehicle structures in a car-to-car crash.

To help protect the fuel system and IPU in a rear-end

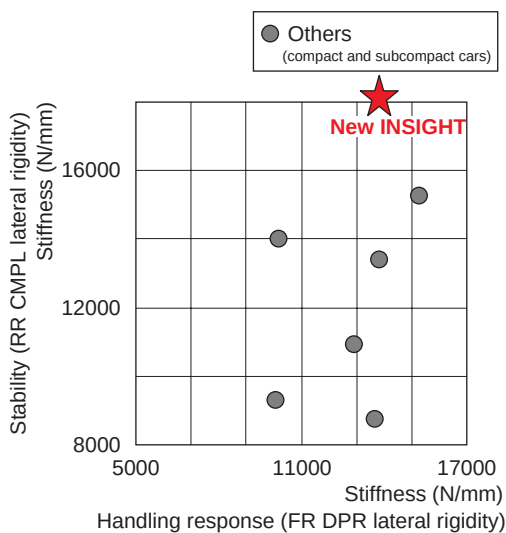


Fig. 18 Body rigidity for rear suspension

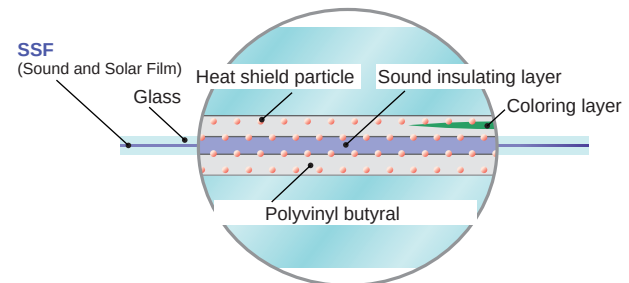


Fig. 19 Sound insulation and IR cut glass

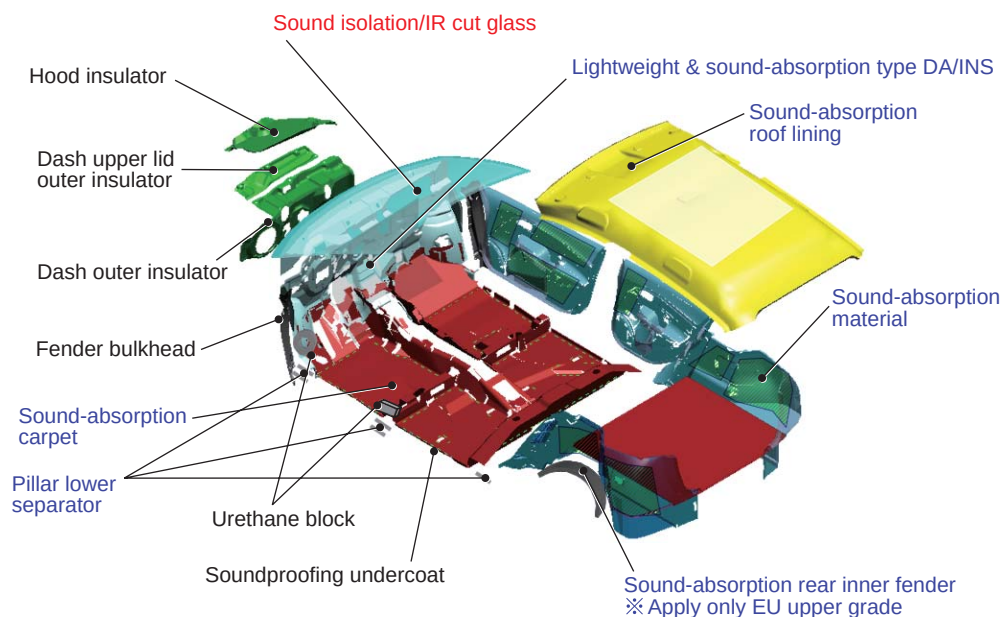


Fig. 20 Sound insulation materials

crash measures are taken to create a structure that can efficiently absorb the crash energy with a short crash stroke, including employing a rear frame with large cross section and adding a rigid cross section structure to the rear panel.

In addition, the front seats are equipped with active head restraints as a standard feature. This helps reduce the backward motion of the head in a rear-end crash, which mitigates the impact to the neck.

6. Affordable

6.1. Hybrid System

6.1.1. Technologies for reducing the IPU cost

In addition to the above-mentioned technologies for reducing the IPU size, the IPU cost has been reduced by measures such as using beam lead bonding for the inverter power module, reducing the number of fuses by optimizing the protective structure of the DC-DC converter circuit, and integrating the high voltage cable and DC-DC converter output cable wiring. This has achieved a cost reduction of approximately 42% compared to the conventional model.

6.1.2. Thin type motor

The motor uses the same basic structure as the conventional IMA motor, but the thickness and weight of the magnet parts have been reduced by 22% and 15%, respectively, by optimizing the output power relative to the vehicle weight. In addition to streamlining the manufacturing process, this has achieved a cost reduction of approximately 35% compared to the conventional model.

6.2. Chassis Commonality

The full use is made of the chassis platform of the 2008 model year FIT, which is well received by the market, and efforts have been made to promote commonality. However, in this new hybrid vehicle the fuel tank and exhaust system use different designs to realize a low overall height (Fig. 21).

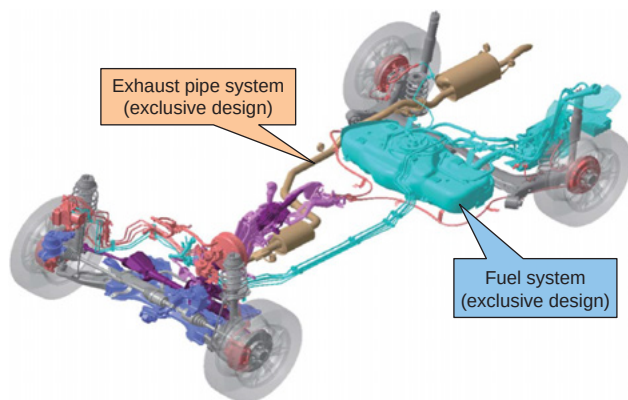


Fig. 21 Chassis platform

7. Conclusion

The first generation INSIGHT was the pioneer vehicle for hybrid technology in Europe and North America. Its naming included the meaning of a car that announced the arrival of a new era. As such, this new model was also named “INSIGHT” as the successor to that first generation INSIGHT.

The concept for the new INSIGHT is that of a popularization type that is easily affordable and will be driven by even more customers. Therefore, the name includes the meaning of a car that announces the arrival of the new era of the full-scale popularization of hybrid cars.

The authors are confident that this “GLOBAL Hybrid Car aimed at Popularization” will be driven by even more customers than previous Hybrid models.

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Technology for Increased Fuel Efficiency in New INSIGHT

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ABSTRACT

The 2009 model year INSIGHT was developed with the goal of achieving a higher level of actual fuel efficiency and driving performance.

An analysis has revealed that the ways of operating a vehicle that result in reduced fuel economy are rough accelerator operation during vehicle cruising, excessive accelerator operation during acceleration, and accessory loads, such as air conditioning.

The developed system employs fuel economy priority control, which uses an ECON mode selector switch to permit high fuel economy driving. In ECON ON mode, the engine and transmission are coordinated with each other. Restricting the engine throttle and optimizing the CVT shift response enable the minimum fuel consumption amount (Gram Fuel minimum; hereafter called GF minimum) to be traced, even during rough accelerator operation, thus facilitating low-fuel-consumption driving. In tests of actual fuel efficiency, these technologies reduced the level of variation between users and enabled the achievement of a 9.5% reduction in average fuel consumption in ECON ON mode.

1. Introduction

Considering the importance of protecting the global environment by reducing CO₂, plus the depletion of crude oil supply, increased fuel economy in automobiles is becoming even more critical. In 1999, Honda developed the Integrated Motor Assist (IMA) system, and released the INSIGHT^{(1), (2)} as a dedicated hybrid vehicle. Subsequently, in 2001, it released a small sedan called the CIVIC Hybrid^{(3), (4)}, and in 2004 a medium sedan called the ACCORD Hybrid⁽⁵⁾. In 2005, the CIVIC Hybrid⁽⁶⁾⁻⁽¹⁰⁾ underwent a full model change involving the adoption of 3-stage VTEC technology including deactivation of all cylinders, a hydraulic booster brake system, and a hybrid air conditioner, aiming at enhanced fuel economy.

Nearly 10 years have passed since the first hybrid vehicle was put on the market, and now a dedicated hybrid car was developed aiming at widespread popularization through reduced cost. This paper describes the technology that has been developed for the enhancement of actual fuel economy in order to meet the needs of various users.

2. Development Goals

An analysis has revealed that the following ways of operating a vehicle are the main factors involved with reduced fuel economy.

- (1) Rough operation of the accelerator during vehicle cruising
- (2) Excessive operation of the accelerator during acceleration
- (3) Accessory loads, such as the air conditioner

A CVT hybrid vehicle enables the engine output, motor output, and transmission ratio to be controlled freely. Accordingly, this development was aimed at obtaining an increase in actual fuel economy of 10% by enhancing engine control and CVT shift control. The same drive feeling was also sought to be achieved by fully utilizing the features of IMA to provide torque assistance from the motor, and to realize CVT shift control in which torque varies linearly with the depression of the accelerator.

3. Outline and Specifications of Power Plant

The engine of the new INSIGHT employs the system

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for deactivation of all cylinders employed in the 2006 model CIVIC Hybrid, plus plateau honing, low friction piston rings, and other friction reduction technologies. As a result, friction was reduced and the fuel economy of the engine on its own was increased by 2.0%.

The motor used was a thin DC brushless type. Along with the use of a lower rated voltage, the rotor and stator were both enhanced, aiming at higher efficiency. Also, the motor was reduced in thickness, the size of the magnet was minimized, and the materials were changed, thereby realizing a price reduction.

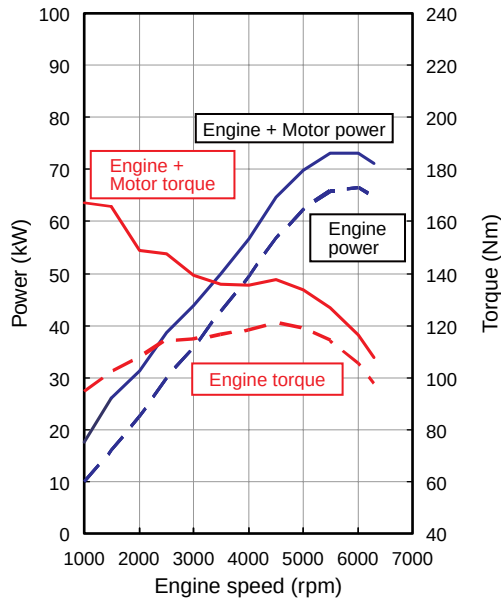


Fig. 1 Output performance of IMA system

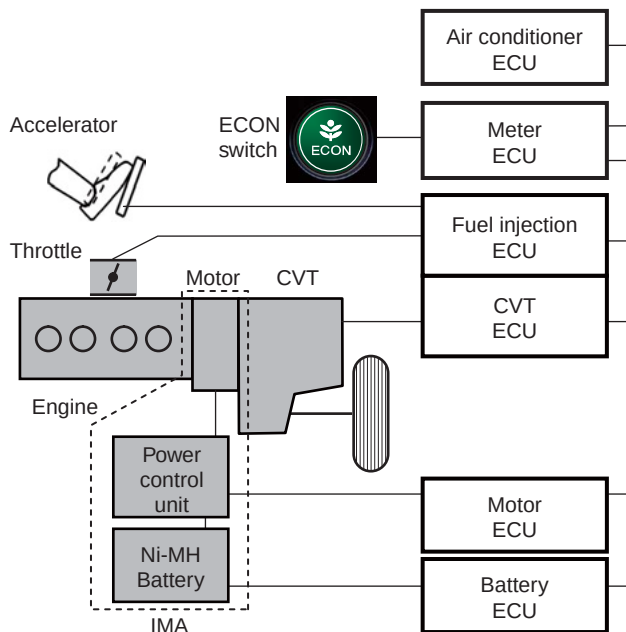


Fig. 2 New INSIGHT system

Regarding the battery, the number of cells was reduced by increasing the output density, and the voltage was reduced to 100 V, realizing reduced size, weight, and price. The abovementioned engine, motor, and battery resulted in a power plant that had a high-torque response in the low-speed range, which is unattainable with a conventional gasoline vehicle. Figure 1 shows the power plant torque and power characteristics versus engine speed.

The CVT used in this development had a low final ratio that was 6.7% less than that of the CVT in the 2006 model year CIVIC Hybrid, with overall development aimed at optimizing fuel economy and drive feeling through the power plant torque, vehicle weight, and tire diameter.

Figure 2 shows the control system of the new INSIGHT. In this system, fuel economy priority control using an ECON selector switch was employed. In ECON OFF mode, the engine is designed to respond well to the accelerator operation, and in ECON ON mode, high fuel economy operation can be easily realized.

The following four items are affecting when the mode is switched between ECON OFF and ECON ON.

- (1) GF minimum control⁽¹⁾
- (2) Driving-force-on-demand control
- (3) Deceleration regeneration control
- (4) Air conditioner control

Details of the above are described below.

4. GF Minimum Control

Rough operation of the accelerator during vehicle cruising is one way of operating a vehicle that results in the reduction of fuel economy. If, during usual cruising, the accelerator is maintained at a fixed position, the CVT is shifted to the optimum ratio and the fuel economy increases. However, if the driver operates the accelerator roughly by repeatedly depressing and releasing the accelerator, the CVT will shift to increased engine speed as soon as the accelerator is depressed, and inertia load occurs. Also, the engine will be subjected to a high load, and it will deviate from the point of minimum fuel economy, resulting in a decrease of the fuel economy.

The new INSIGHT employs GF minimum control in consideration of the above issue. GF minimum control is used to control the engine speed and engine load, utilizing a Brake Specific Fuel Consumption (BSFC) map and the transmission efficiency of the CVT, so that the fuel consumption with respect to the power demand is minimized, thus enabling the GF minimum target to be achieved.

Figure 3 shows the BSFC map, and Fig. 4 indicates the CVT efficiency. In Area 1 of the BSFC minimum target line in Fig. 3, the CVT efficiency increases in inverse proportion to the engine speed. In Areas 2 and 3, the efficiency is roughly constant between 1 000 and 1 500 rpm. In Area 4, there is almost no difference in efficiency

between 1 500 and 2 500 rpm. Consequently, in this system, the GF minimum target can be achieved by controlling the engine speed and engine load in such a way that the BSFC minimum target line shown in Fig. 3 is traced with respect to power demand.

Details of CVT shift control for determining the engine speed and throttle control for determining the engine load - which were employed in order to trace this GF minimum target line - are described below.

4.1. CVT Shift Control

CVT shift control, which achieves the GF minimum target, is described below.

- (1) Area 1 shown in Fig. 3 is the zone in which the driving-force-on-demand accelerator (controlling accelerator for determining driving-force-on-demand) is extremely small. Here, a CVT shift schedule is used in which the lowest possible engine speed at which the vehicle can run is maintained, so the fuel consumption rate versus the power demand is minimized.

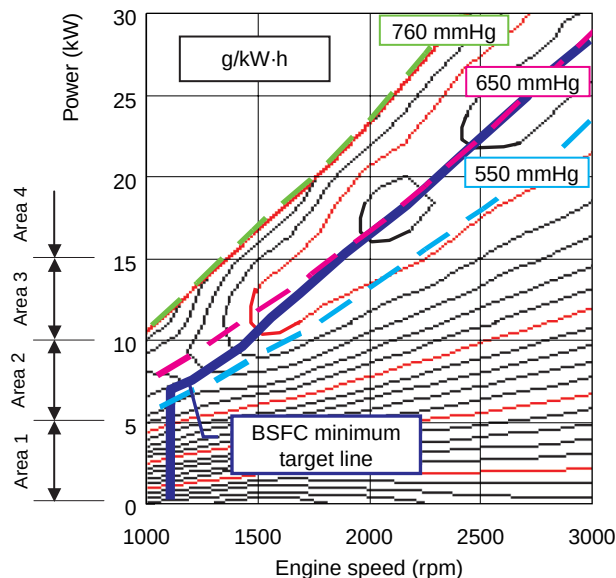


Fig. 3 Brake specific fuel consumption map

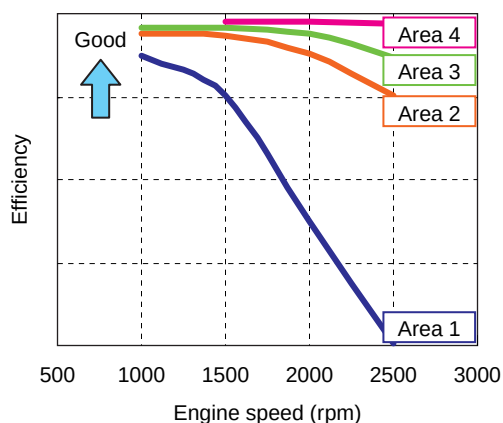


Fig. 4 CVT efficiency

In Areas 2 to 4, in which the accelerator is depressed more than an extremely small driving-force-on-demand accelerator, a shift schedule that takes into account drive feeling is utilized. It is a shift schedule in which the engine speed rises linearly with respect to the increase in vehicle speed, as shown in Fig. 5, while the BSFC minimum target line shown in Fig. 3 is traced.

In total, driving force was generated linearly with respect to the driving-force-on-demand accelerator, without any dip occurring in the generated power, as shown in Fig. 6.

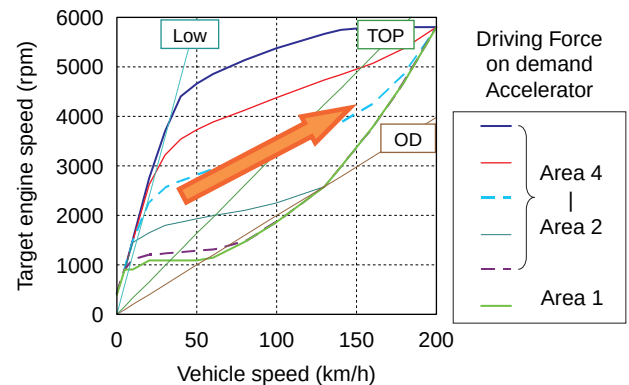


Fig. 5 CVT shift schedule

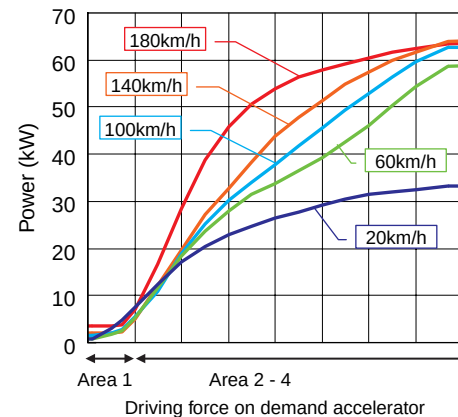


Fig. 6 Power versus Driving - force - on - demand accelerator

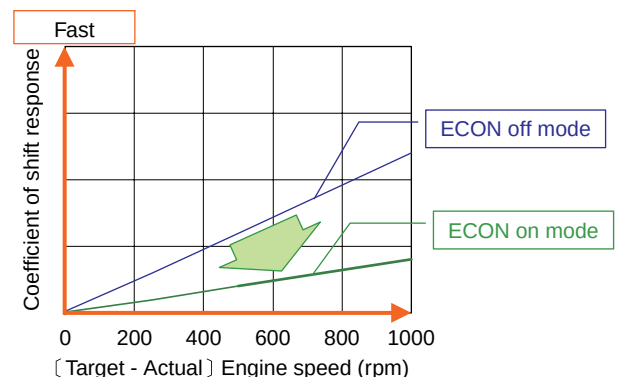


Fig. 7 Coefficient of shift response

- (2) Figure 7 shows the CVT shift response characteristics in the ECON OFF mode and ON mode, respectively. As shown in Fig. 7, the shift response of the CVT is determined by the deviation between the target engine speed and the actual engine speed. In the ECON ON mode, this shift response coefficient is reduced, thus reducing the speed at which a down-shift occurs when the accelerator is depressed. This helps prevent the engine

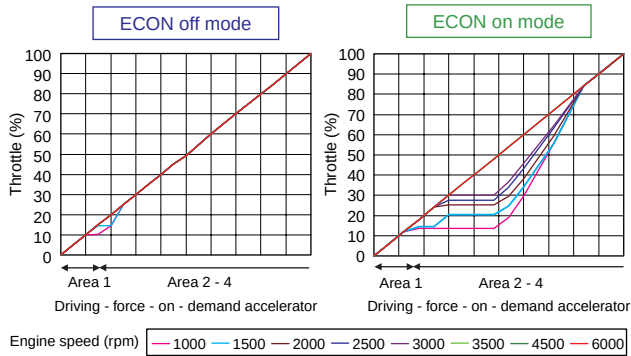


Fig. 8 Restriction of throttle

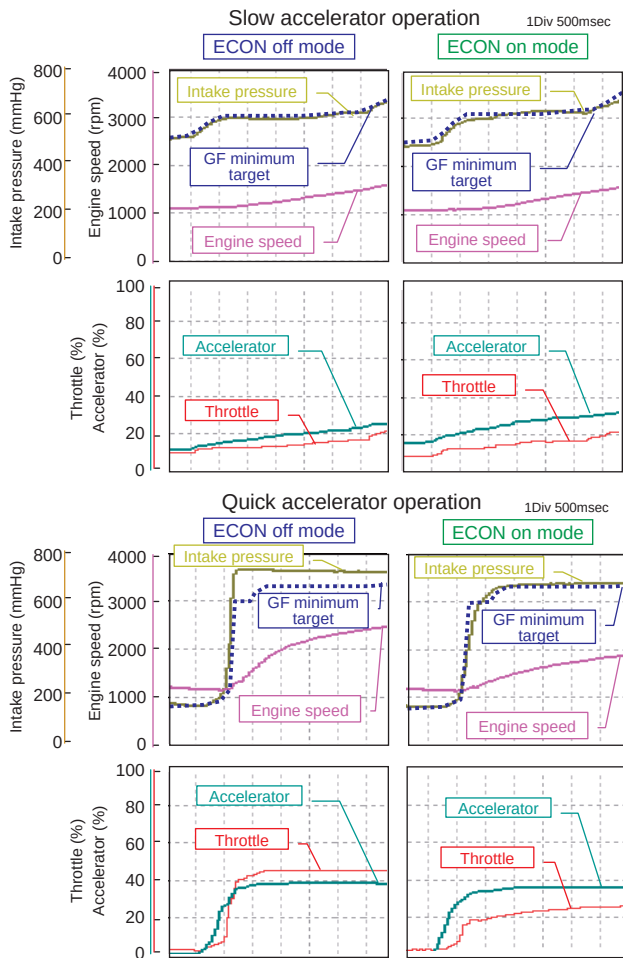


Fig. 9 Comparison using different speed of accelerator operation and control modes

speed from inadvertently rising when the driver depresses the accelerator roughly, thus facilitating high fuel economy driving.

4.2. Throttle Control

Figure 8 shows the throttle limits in ECON OFF mode and ECON ON mode, respectively. By limiting the throttle position with respect to the actual engine speed and driving-force-on-demand accelerator, loading exceeding the best BSFC of the engine is avoided, and the BSFC minimum target line is traced. Also, the throttle limit region is switched depending on whether ECON OFF or ECON ON mode is active.

In ECON OFF mode, the throttle limit region is minimal. Consequently, the GF minimum target line can be traced at low accelerator demand, and when the depression of the accelerator is large the throttle is not

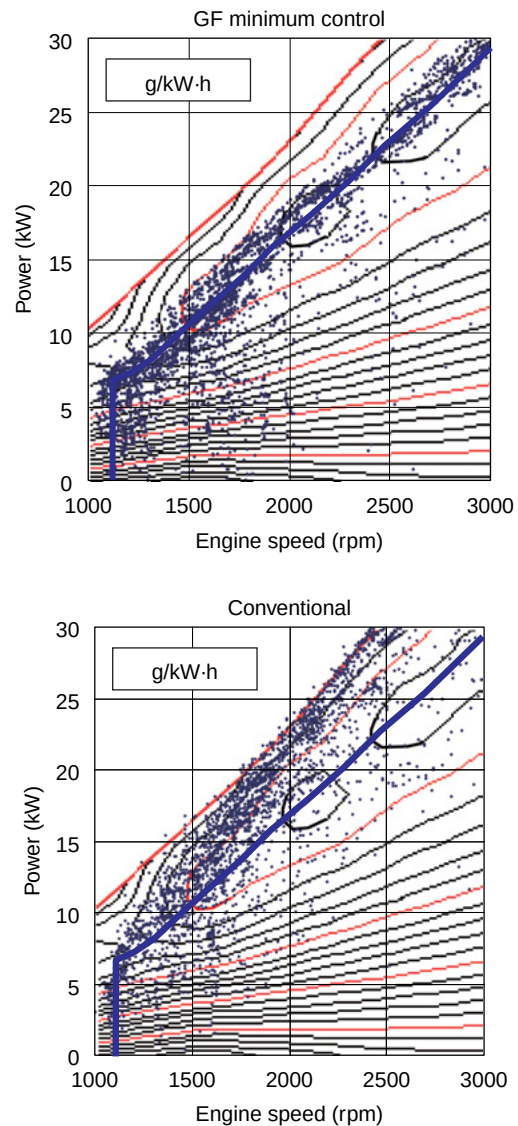


Fig. 10 Comparison with and without GF minimum control

limited, so a rapid torque response is obtained.

In ECON ON mode, the throttle limit zone is increased, enabling the GF minimum line to be traced even when the driver operates the accelerator roughly. Also, even if the throttle is greatly limited, a driving force that is linearly proportional to the depression of the accelerator is realized as a result of control of engine speed by the CVT and torque assistance from the IMA motor.

Figure 9 shows the response characteristics of the vehicle when the accelerator is operated slowly and when it is operated rapidly, with ECON OFF mode active and with ECON ON mode active. In the case of low fuel consumption driving in which the accelerator is operated slowly, there is no major difference in response between with ECON OFF mode active and with ECON ON mode active. In the other case, when the accelerator is operated rapidly, the intake pressure and engine speed rapidly respond to the operation of the accelerator with ECON OFF mode active. When the accelerator is operated rapidly in ECON ON mode, the limit imposed on the throttle results in the intake pressure in the engine increasing gradually. In addition, the shift response characteristics of the CVT for ECON ON mode suppress the rise of the engine speed, thus achieving higher fuel economy.

Figure 10 shows a comparison of the GF minimum target line traced, with and without GF minimum control which utilizes CVT shift control and throttle limit control.

Compared to the conventional control method, this GF minimum control enables the GF minimum line to be traced accurately, thus raising the fuel economy by 0.3 km/L in 10-15 mode.

5. Driving-force-on-demand Control

Conventionally, Honda's CVT has provided good drive feeling by selecting the optimum shift schedule from 16 kinds of maps according to the particular running conditions such as inclination, road surface condition, accelerator pedal operation, and lateral G when cornering.

However, with the new INSIGHT, the mode is switched between ECON OFF and ECON ON using the ECON switch in order to simultaneously realize good fuel economy and drive feeling. In the case of conventional CVT shift control, which is based on the accelerator, when the mode is switched, the torque output characteristics versus the accelerator position change, so it is necessary to have a shift schedule for each mode, resulting in the CVT shift control becoming more complicated. Also, the relationship between the throttle and the engine speed is altered, and the abovementioned GF minimum control can no longer be implemented.

In order to overcome these issues, the new INSIGHT employs driving-force-on-demand control.

As shown in Fig. 11, normal drive-force-on-demand

control directly determines the demand driving force, that is, the demand power, from the accelerator position, and performs engine throttle control and CVT shift control with respect to the demand power. Consequently, the throttle and engine speed are controlled independently with respect to the acceleration operation. As a result, there were cases in which there was a non-linear torque feeling or response with respect to an accelerator operation, or a non-linear increase in the engine speed with respect to vehicle speed.

Details of the driving-force-on-demand control of the new INSIGHT, used to overcome these issues, are explained below.

5.1. New INSIGHT Driving-force-on-demand Control

Figure 12 shows an overview of the driving-force-on-demand control that has been employed in the new INSIGHT.

- (1) The accelerator position is converted into driving-force-on-demand accelerator position for each vehicle speed which determines the driving force on demand.
- (2) Engine throttle control and CVT shift control are performed based on the driving-force-on-demand accelerator position.

The engine output torque is determined by the engine throttle control, and the engine speed is determined by the CVT shift control, hence the power for each vehicle speed

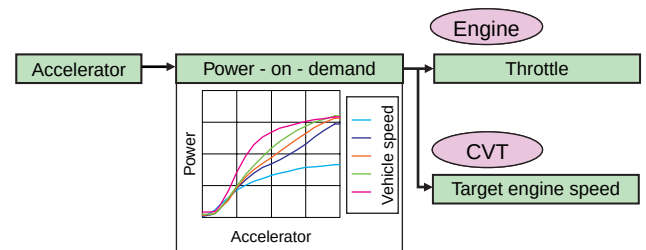


Fig. 11 Normal power - on - demand control

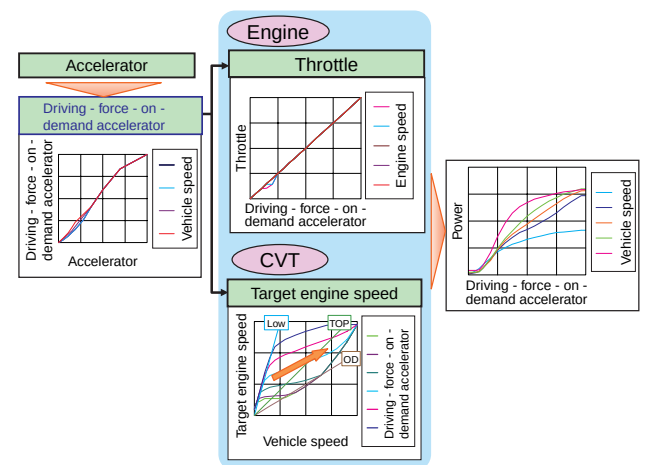


Fig. 12 New INSIGHT control

is determined. In other words, the driving force can be determined.

The driving-force-on-demand control that is employed in the new INSIGHT is directly linked to the accelerator operation, so it provides a linear feeling of torque and response, which was the merit of conventional Honda's vehicles with CVT, and a linear feeling of engine speed versus vehicle speed. In addition, the previous CVT shift control can be used without change, enabling good drive feeling to be sustained.

Also, by optimally setting the shift schedule and the throttle limit versus driving-force-on-demand, GF minimum control is maintained, regardless of whether ECON OFF or ECON ON mode is active. Changing the drive feeling can then be done separately by simply changing the relationship between accelerator and driving-force-on-demand.

5.2. Driving-force-on-demand Accelerator Characteristics

One kind of fuel economy enhancement technology is technology that stops the supply of fuel while the vehicle is moving. In the case of the CIVIC Hybrid, all cylinders are deactivated in the accelerator low position region, and when the accelerator is not depressed, thus stopping the supply of fuel while reducing friction, and realizing motor-only drive (MD). However, because the MD zone is narrow with respect to the accelerator position, it was challenging to keep MD operation when with a rough accelerator work. Accordingly, with the new INSIGHT, the gain of the driving-force-on-demand accelerator versus accelerator characteristics in ECON ON mode has been reduced, resulting in extended MD operation while

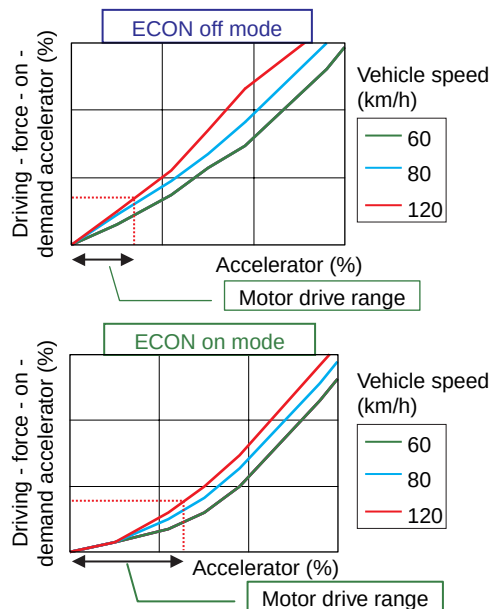


Fig. 13 Driving - force - on - demand accelerator characteristic

gradually decelerating from a cruising condition. The features of this modification are as follows:

- (a) Increased MD frequency
- (b) Better sustainability of FC (Fuel Cut) operation in a situation such as on gentle downhill runs

Figure 13 shows the actual accelerator position versus driving-force-on-demand accelerator position characteristics.

In ECON ON mode, this change in characteristics resulted in a 20% increase in MD frequency. Also, the frequency of use of driving-force-on-demand accelerator moved to the low position side, thus realizing high fuel economy operation.

6. Deceleration Regeneration Control

In the development of the new INSIGHT, efforts were made to enhance fuel economy by carrying out deceleration regeneration according to the brake fluid pressure. Also, in ECON ON mode, efforts were made to increase the regeneration in places where the brake fluid pressure was high, thereby increasing the amount of energy recovered. In places where the brake fluid pressure is low, the regeneration amount was the same in both the ECON ON and ECON OFF modes; therefore, the brake pedal touch feeling was the same in these two modes as well. Figure 14 shows the deceleration regeneration amount when varying break fluid pressure.

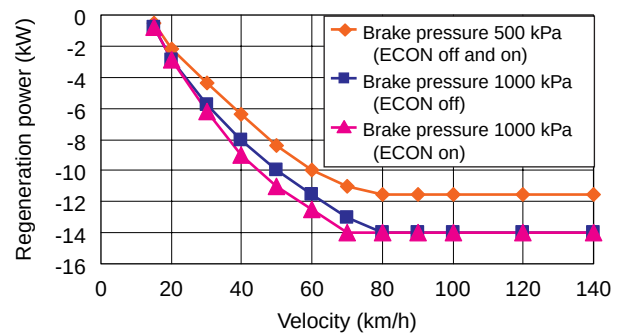


Fig. 14 Regeneration power

7. Air Conditioner Control

Air conditioner control involves using the ECON switch to select the following two items of control, aimed at enhancing the fuel economy in ECON ON mode:

- (1) Enlargement of the "idling stop" zone
 - (a) During ECON OFF mode control, the "idling stop" time is determined by window fogging and comfort.
 - (b) In ECON ON mode, the "idling stop" time is determined by window fogging alone.
- (2) Engine load reduction control at high air conditioner load operation

In ECON ON mode, engine load reduction was achieved by implementing the following three items:

- When the outdoor temperature is high, internal circulation is used, and the intake air load was reduced.
- The electrical load was reduced due to a reduction in the interior fan voltage in the cooling zone.
- The engine load was reduced due to a reduction in the frequency of the air conditioner clutch.

8. Fuel Economy Effect

A fuel economy effect confirmation was performed by 17 males and three females on a test course that took into account practical operation. The results confirmed that the average fuel economy was 9.5% higher in ECON ON mode compared to ECON OFF mode.

Figure 15 shows the results of confirming the fuel economy effect by comparing the fuel consumption when the ECON switch was in the ON and OFF positions, respectively.

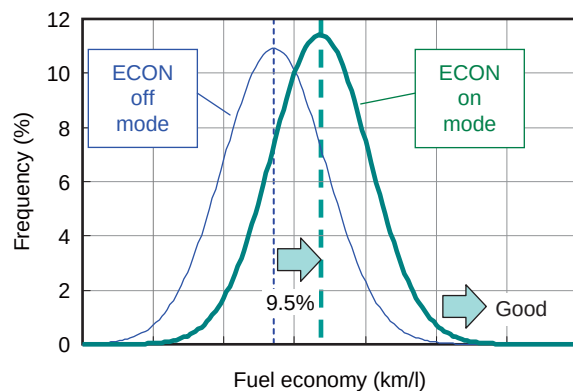


Fig. 15 Fuel economy frequency distribution

9. Conclusion

In summary, it has been shown that the new INSIGHT has both good fuel economy and good drive feeling.

- Fuel economy priority control can be selected with the ECON switch. In ECON OFF mode, the GF minimum target line during high fuel economy driving in which the accelerator operation amount is small can be traced. In ECON ON mode, the GF minimum target line can be traced, even during a rough accelerator operation, thus increasing fuel economy.
- The use of driving-force-on-demand control employing a driving-force-on-demand accelerator parameter resulted in drive feeling achieved by linear driving force and response characteristics with respect to accelerator operation, and linear increase in engine speed with respect to vehicle speed.

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Toshihiko FUKUDA



Takahiro EGUCHI

Development of Power Control Unit for 2009 Model Year INSIGHT

Yasuyuki DANJO* Tomohiro FUKAZU*

ABSTRACT

A Power Control Unit (PCU) has been developed for the Integrated Motor Assist (IMA) system employed in the new model INSIGHT. The development aimed to reduce the overall size, height, and weight of the PCU in order to realize a layout that positioned it under the floor of the rear trunk. Contiguous positioning of the DC-DC converter and the inverter in the PCU has reduced the area required for mounting the unit, in addition to enabling the DC-DC converter input filter to be removed and the number of terminals reduced. This has contributed to the realization of size and cost reductions. The total height of the DC-DC converter has been reduced through the use of copper wire printed on a substrate for part of the main transformer winding wire, contributing to the reduced height of the PCU. The use of a smoothing capacitor with the electrostatic capacity reduced by half enabled reductions in size and cost. In addition, the use of beam leads for the internal wiring of the transfer-molded power module (TPM), together with connections made by direct lead bonding (DLB), enabled the production process to be simplified and costs reduced.

1. Introduction

Today, there are further heightened demands for technology to reduce the amount of CO₂ contained in automobile emissions and enhance fuel economy in order to reduce the greenhouse gases that bring about global warming. Hybrid systems are one method used to address these issues, and the proprietary IMA system developed by Honda has been installed in the first-generation INSIGHT, CIVIC Hybrid, and ACCORD Hybrid. The IMA has lower fuel consumption than conventional gasoline-fueled vehicles, and can be expected to yield a CO₂ reduction effect if its use becomes widespread. This will necessitate a simultaneous reduction in cost and enhancement of marketability as a product, and there is demand, therefore, for a PCU that can satisfy those conditions.

Technologies to enhance product marketability (reduced size, reduced weight) and achieve lower cost are used in the PCU, the unit component of the IMA that handles power conversion, and these technologies are described in the following.

2. Development Goals

Development of the IMA system installed in the first-generation INSIGHT, CIVIC Hybrid, and ACCORD Hybrid was carried out with a focus on fuel economy performance, and the system has evolved accordingly. Apart from the first-generation INSIGHT, these vehicles are built on four-door sedan bodies and the intelligent power unit (IPU) is installed behind the rear-seat backrests. Because of the IPU layout, a hole could not be opened through the IPU to enable configuration of a continuous open space from the trunk through to the rear passenger compartment. Therefore, cars equipped with the conventional IMA did not support loads consisting of long cargo items.

The IPU for the IMA installed in the new model INSIGHT was intended for use in a subcompact-sized five-door hatchback body with split-folding rear seatbacks. In order to realize this layout, development used the previous IMA technology as a base while seeking to reduce size and weight as much as possible, and it was necessary to install the unit under the floor of the trunk

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room (Fig. 1).

In conjunction with smaller size and lower weight, a reduction in the PCU cost was also sought as a development objective.

Reduced size, weight, and cost of the PCU were sought by various means. The internal layout of the PCU was optimized, shared parts were adopted, and new structures were adopted for the inverter and DC-DC converter.

Figure 2 shows the exterior appearance of the IPU. Figure 2(a) shows the former structure, and Fig. 2(b) shows the new structure.

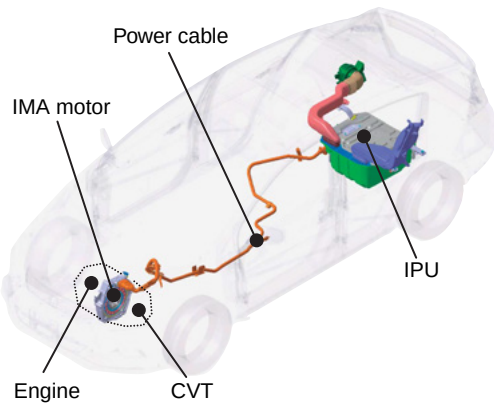


Fig. 1 Configuration of IPU

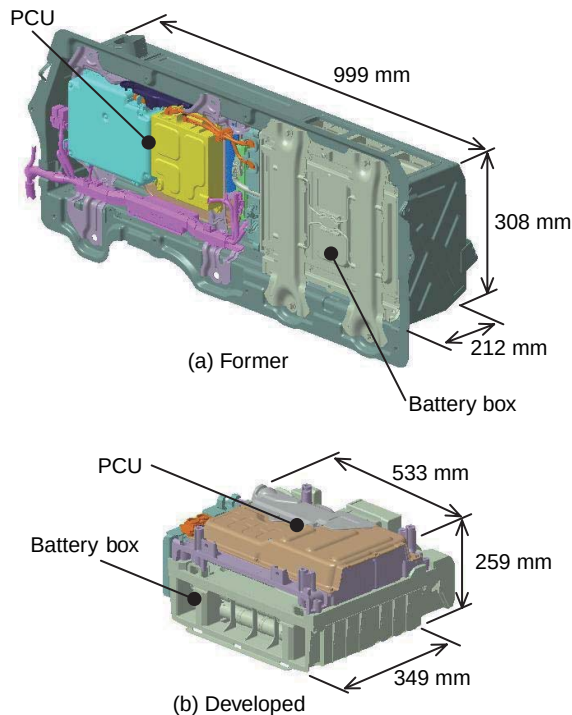


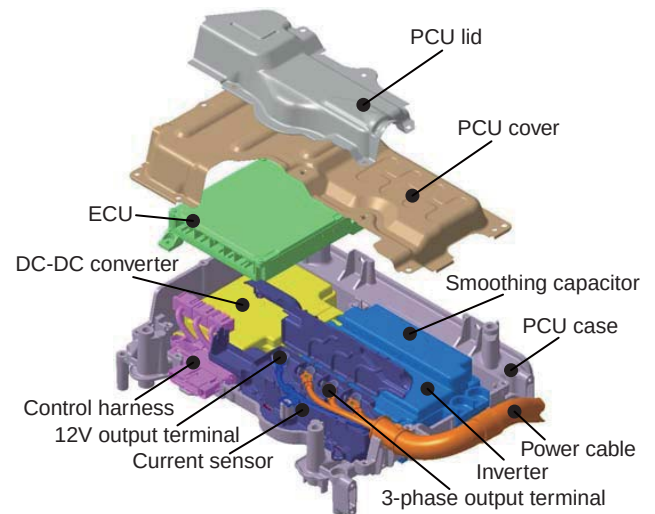
Fig. 2 Comparison of IPU

3. Technology for Reduced Size and Lower Cost

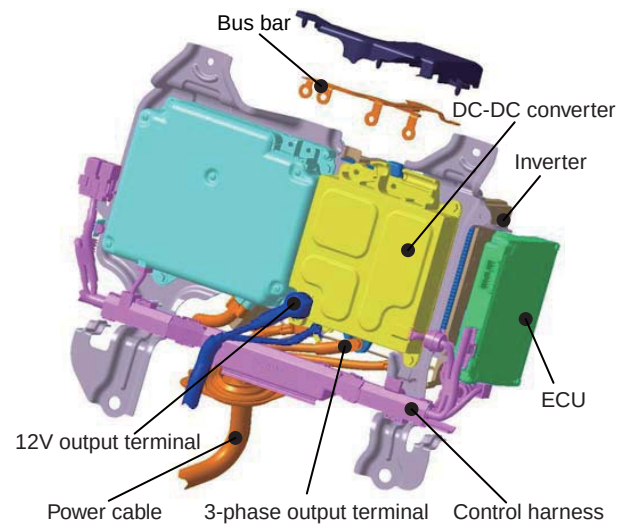
3.1. Structure of the PCU

The PCU consists of the inverter, the DC-DC converter, the electronic control unit (ECU), the current sensor, and the control harness that connects these components. It functions to control power conversion between the IMA system's high-voltage battery and the motor and to supply load power of 12 V. At the same time, electromagnetically shielded construction and a power distribution mechanism are also necessary for the PCU.

Figure 3 shows the exterior appearance of the PCU. Figure 3(a) shows the new structure and Fig. 3(b) shows the former structure.



(a) Developed



(b) Former

Fig. 3 Comparison of PCU

3.1.1. Electromagnetically shielded construction

Since the PCU has an internal high-voltage switching circuit, there is a possibility that switching noise could affect the vehicle. Electromagnetic shielding is therefore necessary. The former PCU had its inverter, DC-DC converter, and ECU enclosed in separate metal cases for that reason [Fig. 3(b)].

The newly developed PCU is simultaneously shielded overall by the PCU case and cover, which is constructed to limit the conduction or radiation of switching noise generated by the inverter, the DC-DC converter, or the ECU. As a result, the covers for the inverter and DC-DC converter could be made from the minimal amount of resin, contributing to the reduction in size and weight.

The new system also adopted a power cable that has the motor cable and 12-V output cable contained inside aluminum piping. This enabled power-cable shields, PCU cover, and PCU case to be electrically connected to provide total shielding for the IMA high-voltage unit in a structure that could be achieved simply and under low cost [Fig. 3(a)].

3.1.2. Power distribution mechanism

The component units of the PCU that are supplied power from the high-voltage battery are the inverter and the DC-DC converter. In the former configuration, the inverter and DC-DC converter were located facing each other so that a bus bar was needed to connect their high-voltage input portions [Fig. 3(b)]. In the new configuration, the PCU has the DC-DC converter located close beside the inverter. Providing power in this configuration enabled the DC-DC converter input bus bar to be removed, together with use of a smoothing capacitor for the inverter, so as to reduce the capacity of the DC-DC converter's filter capacitor. These steps were taken to reduce the PCU size and lower its cost [Fig. 3(a)].

In the former configuration, the inverter and DC-DC converter each had their own output terminal for 3-phase output and 12-V output, respectively. The cable connections and wiring took up a large area [Fig. 3(b)]. The new configuration PCU, by contrast, has added output terminal functionality to the current sensor so that the 3-phase current output from the inverter and the 12-V output from the DC-DC converter can be contained compactly, contributing to the reductions in size as well as in the cost of the shield construction described earlier [Fig. 3(a)].

These technologies enabled the PCU footprint area to be contained within the projected top surface area of the battery box. The IPU could therefore be contained between the car body's side frames, enabling it to be installed beneath the trunk room floor. This approach realized the five-door hatchback body layout.

3.2. Inverter

The IMA system is equipped with an inverter to drive the DC brushless motor. This component performs high-

voltage DC and 3-phase AC power conversion. The basic structure of the inverter includes a power semiconductor module that performs high-voltage, high-current switching, a control board with switching control, protection, and communications functionality, a smoothing capacitor to handle smoothing for the high-voltage input section, a high-voltage wiring case using an insert-molded bus bar, a heat sink, and a cover. The exterior appearance of the former inverter and the developed inverter are shown in Fig. 4.

3.2.1. Reduction of inverter smoothing capacitor capacity

The inverter is equipped with a smoothing capacitor to reduce the DC input voltage and current fluctuations.

The electrostatic capacity of the smoothing capacitor influences the voltage build-up rate during system malfunctions, and to reduce the electrostatic capacity, it is also necessary to shorten the protection response time. This inverter has a response time 27% shorter than before, thus enabling realization of a 50% reduction in electrostatic capacity, relative to the former inverter, as well as a reduction in cost.

Thin polypropylene film is used as a material for the capacitor device. This contributes to the 45% reduction in capacitor size as well as reduction in weight. Figure 5 shows the former capacitor and the newly developed capacitor.

3.2.2. DLB structure of TPM inner wiring

Figure 6 shows the exterior appearance of the inverter's power semiconductor module, with the former structure shown in Fig. 6(a) and the new structure in Fig. 6(b). The power semiconductor module uses a TPM

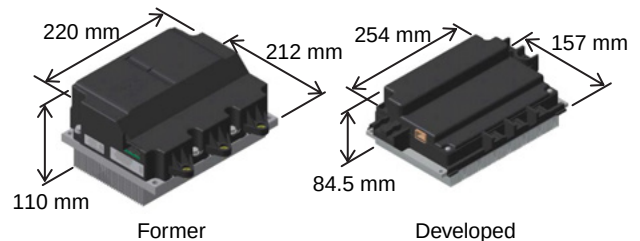


Fig. 4 Comparison of inverter

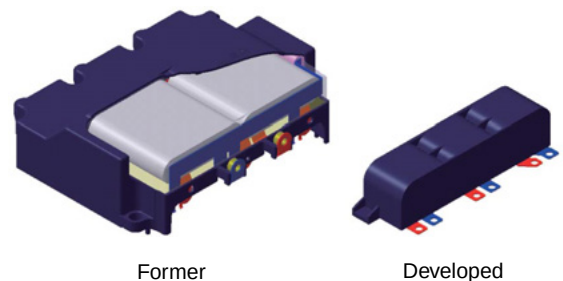


Fig. 5 Comparison of capacitor

structure that packages one phase of the 3-phase current.

The interior of the TPM houses the power devices (IGBT and diodes), the insulated substrate, and the bus bar. A DLB structure has been adopted [Fig. 6(b)] in order to connect the power devices and the bus bar electrically. The former structure used wirebonding [Fig. 6(a)] to connect the power devices and bus bar, and this accounted for a large part of the power semiconductor module manufacturing process. The DLB method simplifies that process and enables a cost reduction^{(1), (2)}.

3.2.3. Increasing the allowable junction temperature in the power devices

The power devices generate heat due to the passage of electric current, and this raises the junction temperature. If the rise exceeds the allowable temperature, damage will result. The rise in temperature can be limited by increasing the area of the power devices, but this method raises the cost. The device area should therefore be kept as small as possible.

The newly developed product has adopted DLB, as described above, and the increased reliability has succeeded in lifting the allowable junction temperature 17%. The devices are now capable of carrying a 38% larger electric current with the same device area as that of the former device.

Use of the above technology has reduced the size of the power semiconductor module to 60% of the former device.

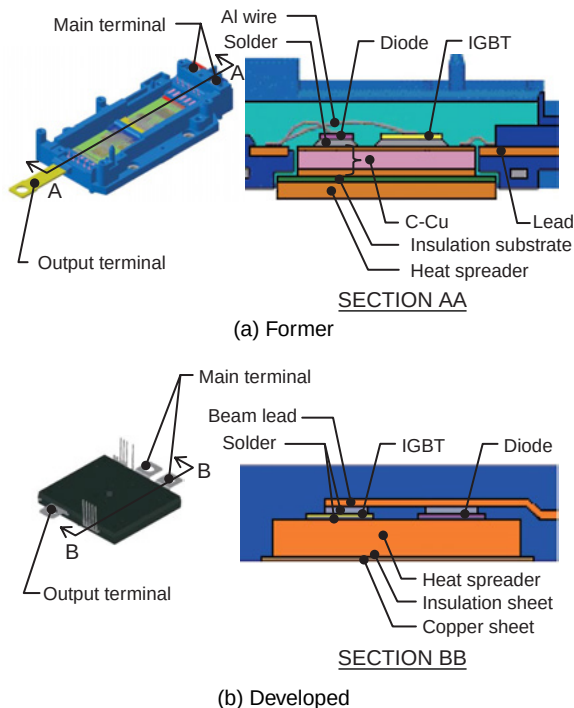


Fig. 6 Comparison of power module

3.3. DC-DC Converter

Vehicles equipped with IMA systems use a DC-DC converter to supply 12-V auxiliary load power. This device is composed of a high-voltage switching device, a transformer, a rectifier diode, a choke coil, a filter, a connecting bus bar, a heat sink, and a control board.

Figure 7 shows a comparison of the exterior appearance of the former converter and the newly developed converter.

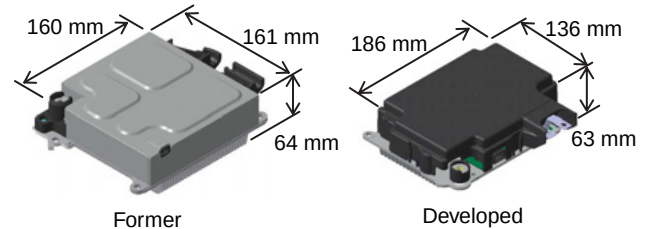


Fig. 7 Comparison of DC-DC converter

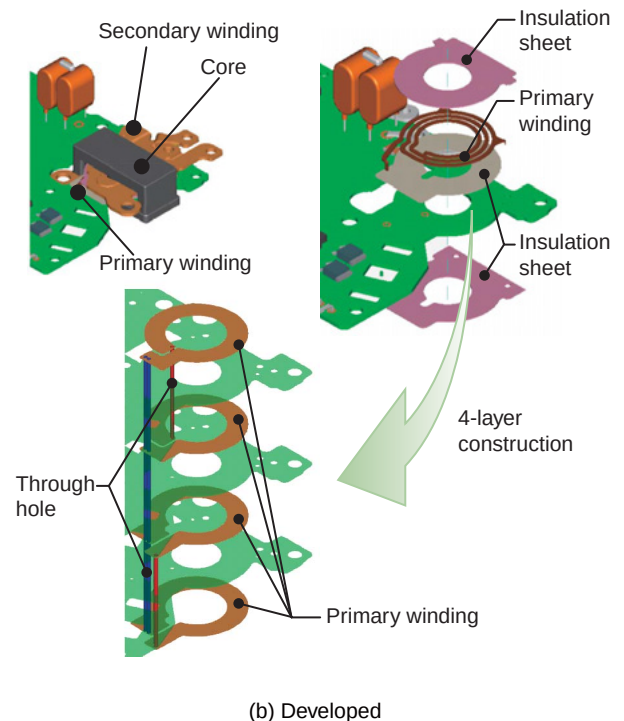
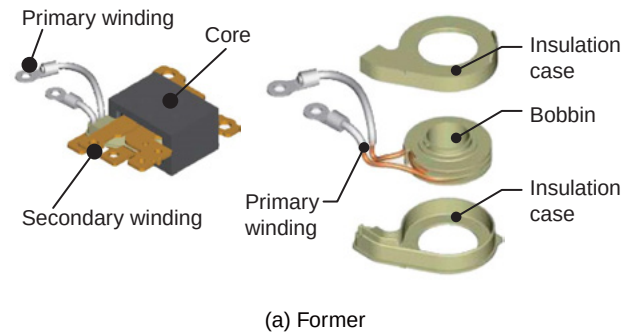


Fig. 8 Comparison of transformer

3.3.1. Transformer winding wire structure

Figure 8 shows the exterior appearance of the transformer. Figure 8(a) is the former structure while Fig. 8(b) is the new structure.

The primary winding wire in the former transformer was a single-core copper winding wire wound on a bobbin, as shown in Fig. 8(a). The electrical connection, therefore, was made with screws holding the wires to a terminal, thereby necessitating an additional connecting process other than the soldering process used with circuit components on printed circuit board, and complicating the manufacturing process. The primary winding wire in the new structure, as shown in Fig. 8(b), is made up of a printed circuit board with conductive patterns in coil form equivalent to a total of two turns in 4 layers, together with copper sheet equivalent to three turns. This enables the electrical connections to be made by the same soldering process used for circuit components on the printed circuit

board, so that the manufacturing process is simplified and reduced in cost⁽³⁾.

In the former structure, using the necessary number of turns in the primary winding wire resulted in a greater total height. With the new structure, however, the use of a printed circuit board also contributes to the reduction in the height of the DC-DC converter.

3.3.2. Reducing the high-voltage input filter circuit

The former DC-DC converter had a filter circuit of the kind shown in Fig. 9 in order to mitigate disturbance noise in the high-voltage input portion of the device. In the newly developed device, the location of the DC-DC converter inside the PCU case has been optimized so that the ferrite-core filter and Y-capacitor in the input portion could be eliminated, and the electrostatic capacity of the DC-DC converter input filter capacitor has been reduced 60% relative to the former device.

3.3.3. Enhanced efficiency

In order to achieve the desired fuel economy performance, it was necessary to help prevent the degradation of power conversion efficiency resulting from the PCU unit being made more compact. Another important factor is enhancement of device and circuitry efficiency in the power conversion portion.

The former device included a resonant snubber circuit on the secondary side in order to employ a rectifier diode with low withstand voltage and low loss. The newly developed device, however, uses a circuit to limit surge voltage due to switching on the primary side instead, enabling use of a still lower-loss rectifier diode. As a result, efficiency increases of 0.5% to 1% were achieved in the normal range of use. The reduction of losses also contributed to a 30% reduction in the weight of the heat sink. Figure 10 shows a comparison of efficiency.

The use of these technologies realized a 46% reduction in weight and 3% reduction in volume in the DC-DC converter, compared to the former device.

4. Conclusion

Development of the PCU for the new model INSIGHT achieved the following items relative to the former PCU:

- (1) In the inverter, the capacity of the smoothing capacitor was reduced by 50% by increasing the speed of over-voltage detection. The use of DLB for the wiring and a 17% increase in the allowable junction temperature realized a 40% reduction in the volume of the power semiconductor module without sacrificing efficiency. As a result, the weight was lowered 36% and the volume reduced by 35%.
- (2) The DC-DC converter weight was reduced by 46% and its volume reduced by 3% by converting part of the transformer winding wire to printed circuit form, adopting a low-loss rectifier diode, and optimizing the

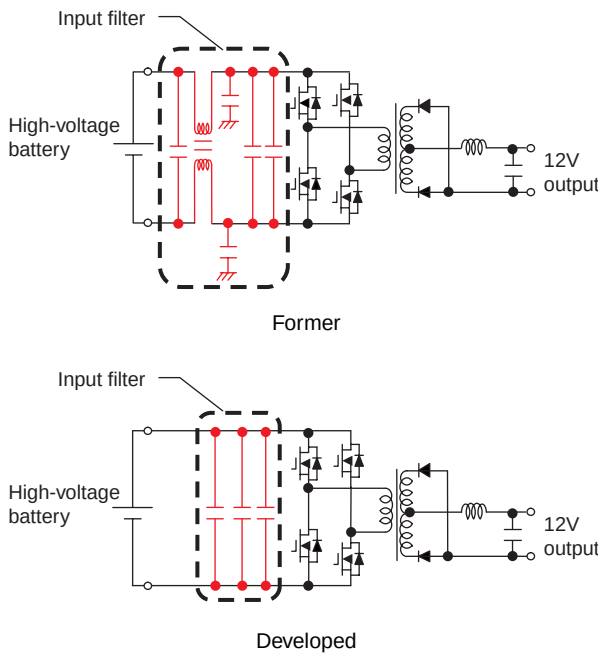


Fig. 9 Comparison of input filter

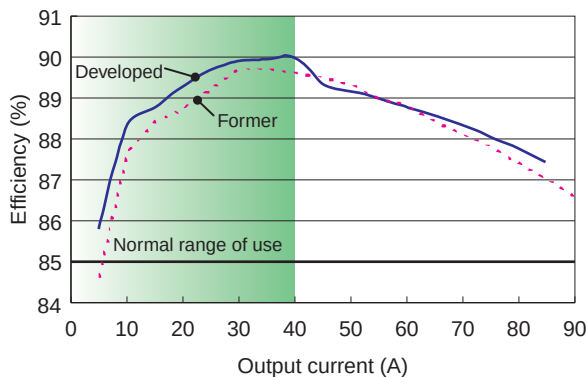


Fig. 10 Comparison of conversion efficiency

internal layout of the PCU to reduce the input filter.

- (3) In the PCU, the electromagnetically shielded structure and power distribution mechanism were optimized. In the inverter, the smoothing capacitor electrostatic capacity was reduced and a DLB structure was adopted. In the DC-DC converter, a new structure was adopted for the transformer winding wire and the input filter was reduced. These measures enabled reductions in cost.

As a result, the overall PCU height was reduced by 41%, its weight was reduced by 31%, and its volume was reduced by 10%.

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Development of 2009 Model Year U.S. FIT

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ABSTRACT

The first-generation U.S. Fit embodied a new appeal — a utility and sportiness not previously seen in the North American sub-compact car market — which gave the vehicle a unique position in the market.

The development of the second-generation U.S. FIT sought to polish and enhance the strengths of the original vehicle, while also attempting to both surprise and please North American customers.

The Fit is now a global vehicle platform, and the development of the new model focused on redesigning even small details in order to optimize the first generation Fit's "DNA," its stylish small body and spacious cabin, ease of use, safety and environmental performance, and fuel efficiency, as well as the riding and driving pleasure it offers, for the North American market.

This has advanced the appeal of the Fit beyond the previous model and resulted in the realization of a "next-generation sub-compact car" able to satisfy not only younger users, but also users seeking a smaller car as their next vehicle, a market segment expected to increase in the future.

1. Introduction

The first-generation U.S. Fit, featuring an automatic transmission and modified design, debuted in 2006, four years after the Japan domestic model.

Up to that point, the selling points of sub-compact cars in the U.S. had been their low price and fuel efficiency. The Fit added a high level of utility created from their center tank layout and a sporty feel designed exclusively for the North American market to forge a unique market position.

Market evaluation of the first generation U.S. Fit had not yet reached Japan, even when the project for the new FIT started, and the team was unsure as to whether the Fit's unique utility and sportiness had achieved significant market acceptance.

In order to obtain information in a short period, the team conducted interviews with user groups, conducted dealer surveys and design clinics, and offered test rides in sub-compact cars at various locations in the U.S. The information obtained and the impressions received at these events became the axis around which the concept for the new FIT was generated.

2. Concept

Interviews with users and surveys of dealers had indicated that the first generation U.S. Fit had been accepted in accordance with the initial goals for the design of the vehicle, and that the market was expecting a further evolution of the vehicle.

The first generation Fit's spacious cabin, roomier than it appears from the exterior, its ease of use, in particular the "Magic Seats," its nimble handling in the city, and its acceleration on freeway convergences were particularly well received.

The market in both North America and Japan also showed that there were signs of a shift away from larger vehicles, due in part to the spiraling price of gasoline.

Based on this information, the team was determined to refine the utility and sportiness that were important aspects of the appeal of the original Fit, and to increase the sense of quality of the vehicle to also satisfy customers who sought to move to a smaller car. These goals were summed up in the concept "Sport & Utility + Refinement" (Fig. 1).

* Automobile R&D Center



Fig. 1 Development concept

It was also decided that the design of the North American FIT would be considered during the development of the Japan domestic FIT, which would serve as its base vehicle, thus realizing an increase in the level of commonality between the vehicles. By this means, the developers sought to achieve a good balance between cost and the realization of increased commercial appeal.

3. Exterior

3.1. Design

The concept for the exterior design of the U.S. FIT was set as a “Super Forwarding Aero Form.” This design sought to incorporate an advanced cabin design in which the cabin reached far to the front of the vehicle and a dynamic forward inclination into a vehicle that gave the sense of being a single compact unit (Fig. 2).

In order to realize the Super Forwarding Aero Form and to increase up-down and left-right visibility:

- The A-pillars were moved forward (110 mm in comparison to the previous model).

- The A-pillars were reduced in width (by 10-40 mm against the previous model).
- The area of the windscreen and the front quarter windows was increased.

This design would reduce the cross-section of the framework. The response to this issue was to use stiffeners and to modify the structure of the A-pillars in order to reduce the amount of change to the framework cross-section (Fig. 3).

The front overhang was extended by 150 mm compared to the domestic FIT in order to emphasize the vehicle's sportiness, and designs developed exclusively for the North American model were used for the front bumper, front fender, and hood.

In addition, two front bumper designs were prepared for the FIT Sport and the base FIT in order to satisfy the desires of a wide range of users, from younger users to users seeking to move to a smaller vehicle.

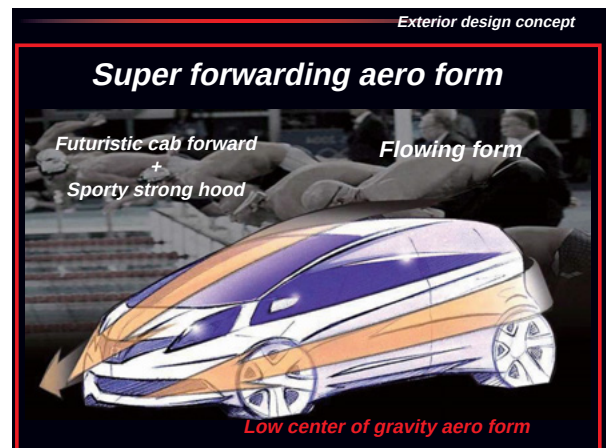


Fig. 2 Exterior design concept

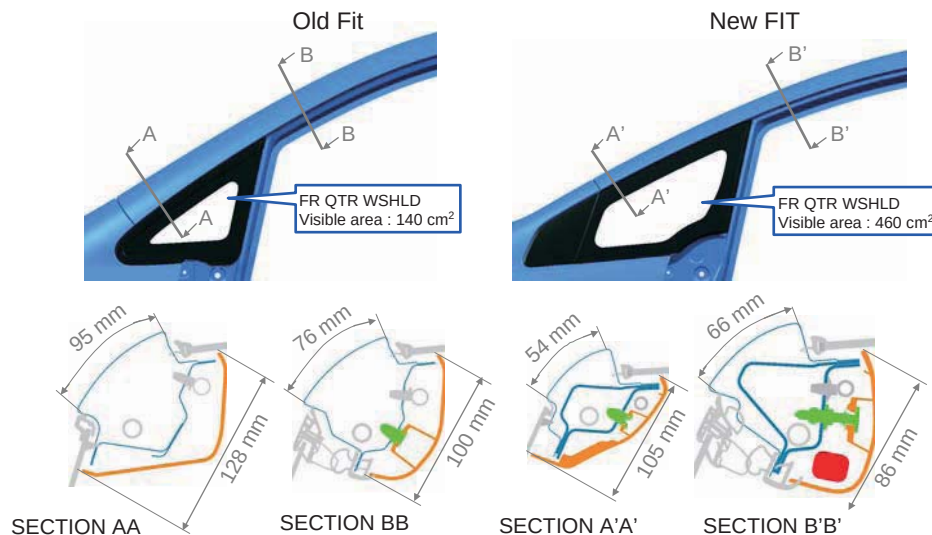


Fig. 3 Structure of front pillar

3.2. Dimensions

The wheel base of the U.S. Fit has been increased by 50 mm against the previous model to 2500 mm (the same length as the Japan domestic Fit). Ride comfort has been enhanced by extending the tandem distance and redesigning the rear suspension geometry.

In addition, structural modifications, including the expansion of the rear frame cross-section, use of a polygonal cross-section, and optimization of frame thickness, have enabled the achievement of higher

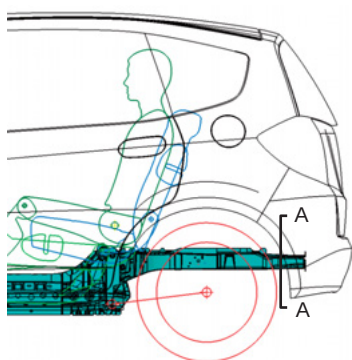


Fig. 4 Section of rear frame

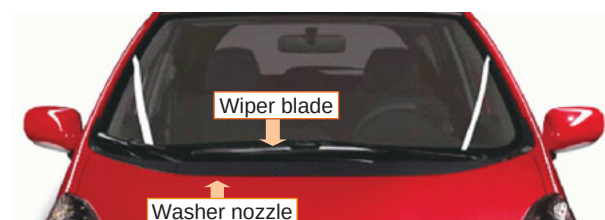


Fig. 5 Forward visibility

energy absorption performance within short strokes, which minimized the increase in total length and also enhanced the vehicle's safety performance in a rear collision (Fig. 4).

3.3. Increased Visibility

In order to increase up-down visibility and visible area on the left and right:

- An extra-large windscreen with a total length of 1076 mm (an increase of 166 mm against the previous model) has been employed.
- A new wiper blade design has been employed.
- The positioning of the wipers helps prevent them from obstructing forward visibility.
- The washer nozzle has been positioned inside the cowl top.
- Small mirror bases have been employed for a streamlined design.

These innovations have enabled the achievement of a panoramic field of vision (Fig. 5).

4. Interior

4.1. Design

In order to realize ease of use and comfort based on the concept "Advanced Personal Interior," the goal for the interior design was a sporty design that located the vehicle operating switches in the center of the driver's position and offered the pleasure of operation with instantaneous recognition and response (Fig. 6).

4.2. Package

The design of the vehicle package sought to increase cabin space and realize greater ease of use while maintaining the center tank layout.

Various modifications were made to all the fittings, such as changes to the shape of the door linings and the seats, in order to achieve increased cabin space.

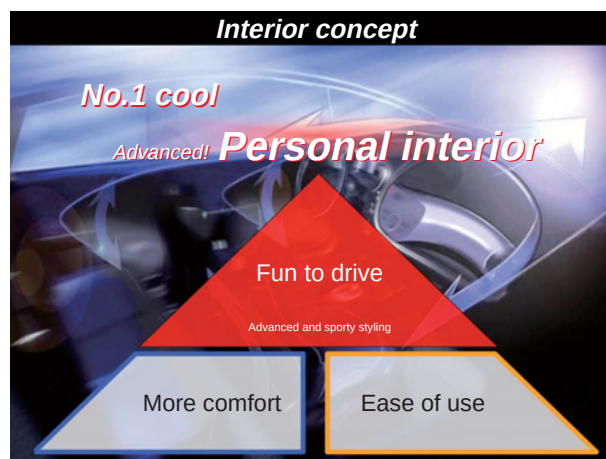


Fig. 6 Interior concept

In addition, employing a large footrest and telescopic steering mechanism and increasing the range of choice for seat adjustments has enabled each driver to find the optimum, most comfortable driving position (Fig. 7).

4.3. Instrument Panel

The heater control switches have been positioned to be within reach of the driver's hand, thus enabling them

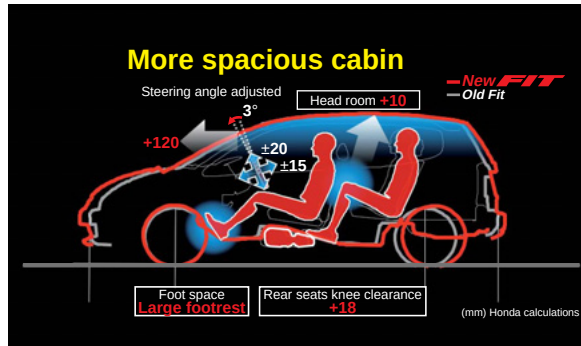


Fig. 7 Package layout

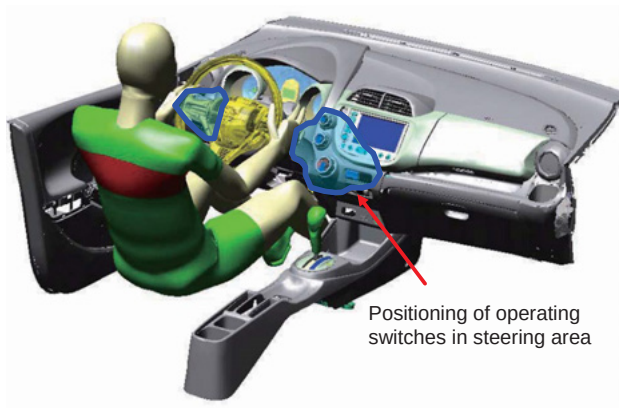


Fig. 8 Enhanced maneuverability

to be operated with the minimum change in the driver's line of sight. The use of thinner ducts in the instrument panel has enabled the top section of the instrument panel to be lowered in order to achieve increased visibility (Fig. 8).

4.4. Storage Space

Cup holders have been placed at the sides of the instrument panel in order to enhance their ease of use and to make them more accessible. Drink holders have also been fitted in 10 other locations, including the console box and door pockets (Fig. 9).

4.5. Seats

The use of a luxury vehicle seat frame in which the



Fig. 10 New seat frame



Fig. 11 One motion dive-down

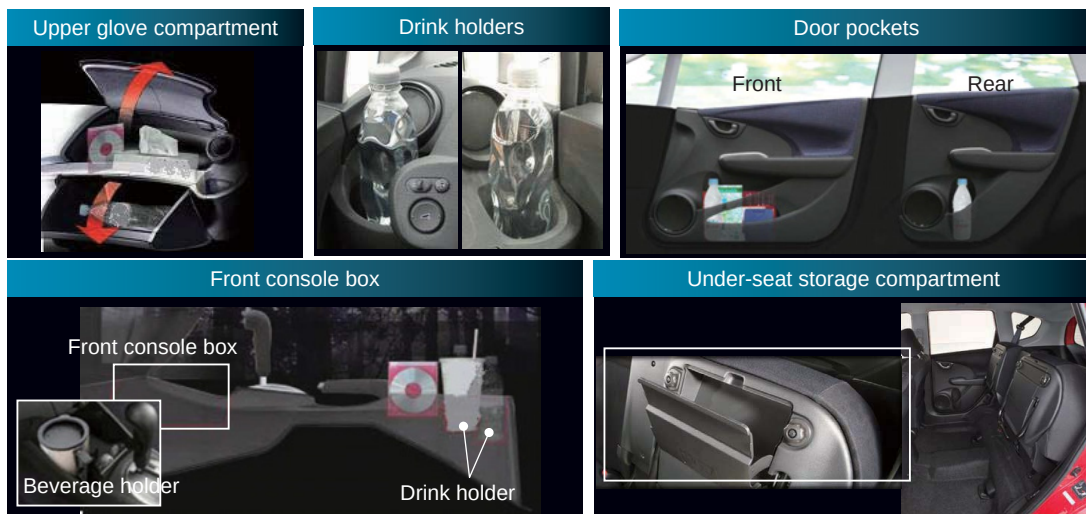


Fig. 9 Storage

cushion springs run horizontally was set as the basic specification for the seats. The seat and the side supports have been separated by deep grooves, realizing a balance between cushioning and holding performance and enabling the achievement of a “wrap-around” feel (Fig. 10).

Thicker cushions in the rear seats have enhanced ride comfort, and a lever-operated seat shift mechanism has been employed to enable one-touch operation (Fig. 11).

5. Powerplant

In order to create a powerplant suitable for the model change of the Fit as a global model, an i-VTEC engine⁽¹⁾ displaying increased power and reduced friction was developed to increase fuel efficiency and environmental performance (Fig. 12).

The replacement of the one-valve VTEC mechanism used in the previous model with a high-power VTEC mechanism that switches the two intake valves between “Lo” and “Hi” modes, has increased maximum power. Reducing knocking by means of redesigning the flow of cooling water in the engine and employing an intake manifold with a resonator enabled the achievement of increased power.

In terms of environmental performance, high-

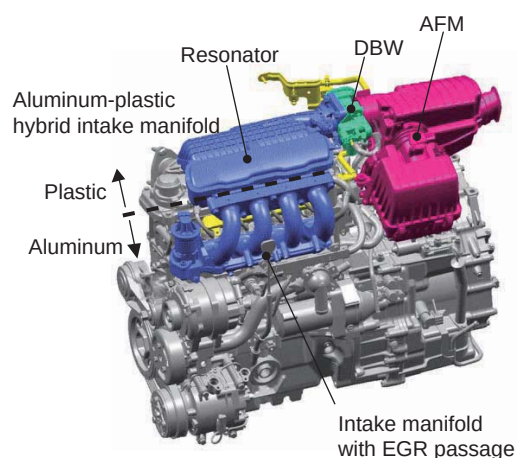


Fig. 12 2009 Model engine

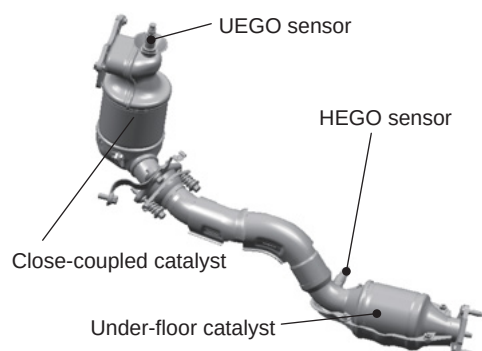


Fig. 13 Configuration of exhaust system

precision A/F control using an airflow meter and LAF sensors, integration of the cylinder heads with the exhaust manifold, and the use of a close-coupled catalyst for enhanced cleaning performance have enabled the new FIT to meet the California LEV II ULEV regulations (Fig. 13).

In order to increase commercial appeal, the characteristic of the DBW (electric throttle) control, which is a factor in the vehicle’s drive feel, has been set to suit modes of use in the North American market, in particular city driving, enabling the achievement of a rapid acceleration feeling.

6. Electrical Equipment

6.1. Meters

The use of dials with a three-dimensional hairline finish and acrylic calibrations on the speedometer, tachometer, and fuel gauge has increased the three-dimensionality of the meters. To increase visibility, the

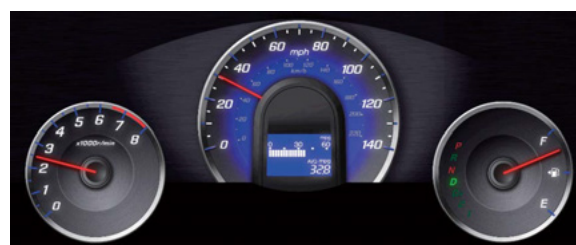


Fig. 14 Meters



Fig. 15 Audio



Fig. 16 Navigation system

meters are constantly lit, and the use of blue illumination imparts a sense of sophistication and refined quality. In addition, the information display located in the center of the panel matches the meters with a blue backlight, and includes constantly displayed instantaneous and average fuel consumption meters, providing support for eco-driving (Fig. 14).

6.2. Audio

Provided with its own face panel, the audio system is MP3/WMA-compliant, and a volume adjustment in relation to vehicle speed is standard. In order to respond to the need to connect portable audio players, and the like, an external input jack has been provided as standard in the front of the audio panel. The FIT Sport features a USB interface that enables an iPod to be operated through the audio as standard (Fig. 15).

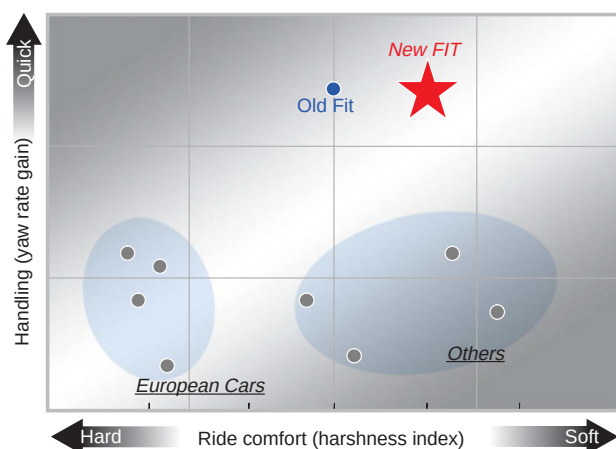


Fig. 17 Handling

6.3. Navigation System

The FIT Sport offers a DVD navigation system as an option for the first time in the U.S. Fit. The system features a 6.5-inch-wide touch screen display and voice recognition function (Fig. 16).

7. Handling / Ride Comfort / NV Performance

With regard to handling and ride comfort, the development of the new FIT maintained the “Fun to Drive” experience that was a strength of the previous model in the North American Market, while also working to realize the concept of solidifying the Fit’s position as the “Sporty No. 1.”

In order to bring this concept to fruition, the rigidity of the new vehicle’s body was increased against the first-generation U.S. Fit, and the steering system and front and rear suspension systems were redesigned.

In addition, the steering and suspension settings were established exclusively for North America. Testing was done continuously based on North American road conditions in an attempt to realize both nimble handling and a level of ride comfort in which there would be minimal shock even on rough roads (Fig. 17).

7.1. Body Rigidity

The body of the new FIT, the basis of the vehicle’s dynamic performance, was redesigned from the ground up (Fig. 18).

However, no increases in weight were permissible in achieving enhanced performance. To help prevent weight increases, CAE was employed at every step of the body design process. The shapes and thicknesses of plates were optimized and the efficiency of couplings was increased

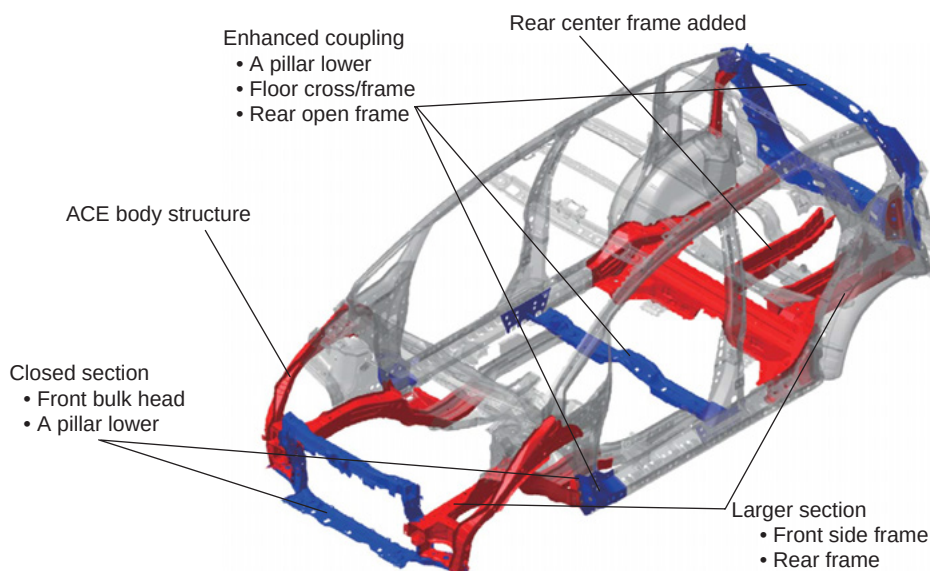


Fig. 18 High-rigidity body structure

in a continuing process aimed at achieving weight savings in units of grams.

The main points in this process were:

- 1) Realization of a larger cross-section for the body framework
- 2) Employment of an ACE (Advanced Compatibility Engineering) body framework
- 3) Addition of a rear center frame
- 4) Achievement of increased coupling efficiency
- 5) The use of a closed cross-section

In terms of dynamic performance, the use of an ACE framework and other measures employed have increased body rigidity by 164% against the previous model.

In addition, the rigidity of the rear suspension mounting has been increased by 27% (Fig. 19), resulting in the achievement of a level of body rigidity that rivals that of the CIVIC, a vehicle one class higher.

7.2. Steering System

In order to provide highly responsive steering, a high-capacity EPS system has been used exclusively for the North American model, and a quick ratio has been set. The rigidity of the gear box has also been increased, and a redesigned mounting method helps to ensure high support rigidity (Fig. 20).

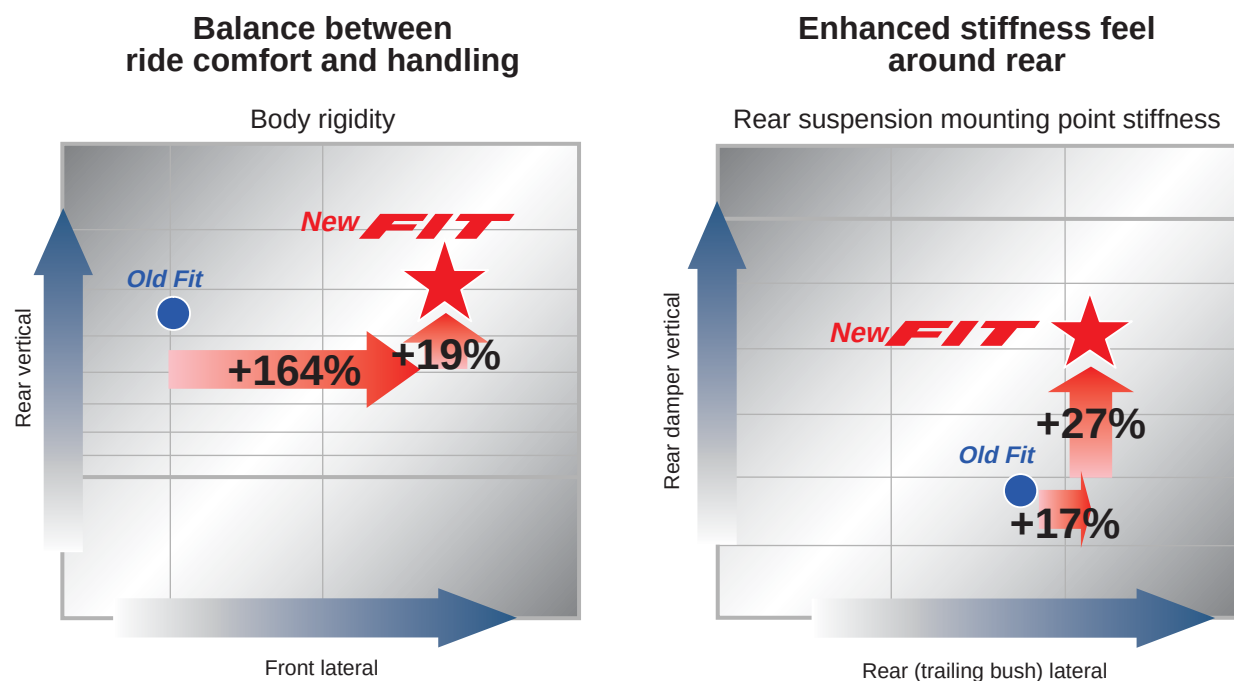


Fig. 19 Body rigidity

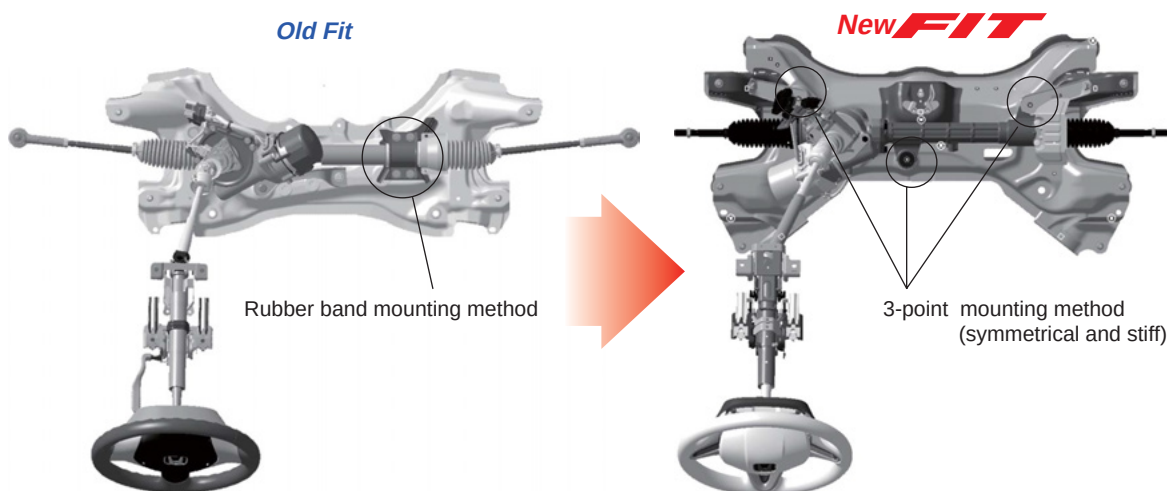


Fig. 20 Steering system

7.3. Suspension

The coaxial configuration of the compliance bushing axes with the supporting section of lower arm has reduced precession of the axes in the front suspension during the stroke, resulting in increased ride comfort (Fig. 21).

In the rear suspension, enhanced ride comfort has been achieved by increasing the length of the trailing arm by 40 mm and reducing the spring lever ratio by 10% (Fig. 22). The FIT Sport uses a $\phi 17$ rear stabilizer and sportier suspension settings. The capacity of the bushes has been increased in both the front and rear suspensions, and detailed attention has been paid to optimizing settings for the damping force of the dampers, and the like, resulting in the realization of sporty and responsive driving performance.

7.4. NV Performance

The targeted characteristic in the new model U.S. FIT was that of a “sporty feel in city driving,” and engine noise in various scenarios, for example when stopping and starting at traffic lights, would have an effect on the realization of this attribute.

In addition, in the North American market, the Fit, a vehicle falling into the low-displacement vehicle category, would often be called upon to operate at wide-open throttle, for example at highway convergences. Engine noise during acceleration at wide-open throttle was

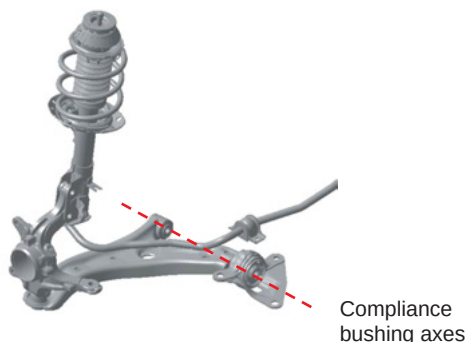


Fig. 21 Front suspension



Fig. 22 Rear suspension

therefore focused on in order to increase quietness.

Both methods of reducing input and of increasing vehicle body performance were employed in order to increase quietness. A torque rod engine mount that reduces radiated engine noise⁽¹⁾ was employed as a method of reducing input (Fig. 23).

As a method of increasing vehicle body performance, the previous sound-proofing materials were replaced by sound-absorbing materials, enabling quietness to be increased while achieving a 6.3 kg weight saving.

As a result, middle-frequency engine noise during acceleration in the frequently-used 3000-6000 rpm range has been reduced by approximately 5 dB (Fig. 24).

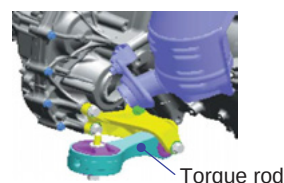


Fig. 23 Torque rod-type engine mount

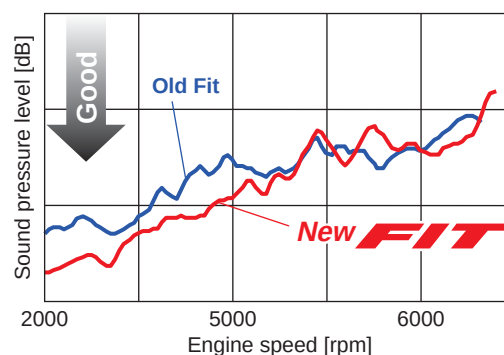


Fig. 24 Middle-frequency engine noise

8. Conclusion

During the period of development of the 2009 model year U.S. FIT, the level of changes in the global environment, users' awareness, and price of gasoline exceeded previous expectations, and the sub-compact car class came to assume a more important position, even in the North American market.

This is, of course, in line with the trend towards more economical and environmentally-friendlier vehicles. However, the development team also received the impression from numerous sources that these were not the only desires for the sub-compact class.

In order to respond to those desires, while maintaining the level of fuel efficiency and environmental performance that are basic factors in the appeal of the class, the development of the new FIT focused on fine details to

achieve a design and drive feel specifically tailored to the North American market, adding to basic performance parameters the achievement of a level of “Fun to Drive” and sense of quality that sets the FIT apart from competitors’ sub-compacts.

Honda is confident that the freshness of the new FIT will both surprise and please an increasing number of North American customers who have previously preferred larger, higher-displacement cars.

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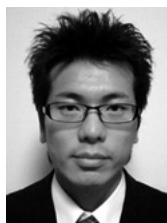
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Development of FREED Assisted Mobility Vehicle

Kimito NISHIYAMA*

ABSTRACT

This paper introduces two approaches that were applied when developing the FREED assisted mobility vehicle, and gives specific examples of each.

The first approach is to simultaneously develop an assisted mobility vehicle with various specifications from the base specification vehicle planning and layout stage. The optimum package layout for an assisted mobility vehicle was realized within that process. In addition, common parts were shared by establishing the streamlined parts design and investigating design feasibility.

The second approach consists of simultaneously producing assisted mobility vehicles within the same production line as mass production vehicles.

Applying the above development methods enhanced product appeal and productivity compared to conventional models.

1. Introduction

Research on small-scale production technology for special vehicles was previously reported in “Development of Small-scale Production Method for Large-size Exterior Plastic Parts”⁽¹⁾. Development of the FREED assisted mobility vehicle introduced in this paper aimed to enable various usage formats and enhance product appeal, while also reducing costs by minimizing investment amounts in consideration of small-scale production.

2. FREED Assisted Mobility Vehicle

The FREED package characteristics, which offer large cabin space in a compact body size, were used to provide sufficient functions as an assisted mobility vehicle with the minimum possible changes from the base vehicle. In addition, various specifications are provided to meet

diverse user needs.

Figure 1 shows the three FREED assisted mobility vehicle types.

Side lift-up seat (equipment) car is illustrated in Fig. 1(a).

This is a 4-passenger specification with two rows of captain seats for easy assistance when getting in and out of the vehicle. A sliding door with an opening width of 600 mm is included to provide sufficient space when raising and lowering the lift-up seat.

Next, assistant lift-up seat (equipment) car is shown in Fig. 1(b).

This is a 7-passenger specification with three rows of captain seats for easy communication. The door opening angle is 16.5 degrees wider than that of the base vehicle in order to provide sufficient space when getting in and out of the vehicle.

Finally, wheelchair vehicle is shown in Fig. 1(c).



Fig. 1 FREED with side and assistant lift-up seat and wheelchair vehicle (with third seat)

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Two types were set: a 5-passenger specification with two rows of captain seats, and a 7-passenger specification with three rows of captain seats. A 3-point seatbelt for the wheelchair passenger is provided, and various usage formats are supported.

3. Package Layout and Common Parts with the Base Vehicle

3.1. Simultaneous Development of Three Assisted Mobility Vehicle Types

Assisted mobility vehicles have conventionally been modified versions of mass production vehicles, so the following items are desired⁽²⁾.

- (1) Large-scale modifications should not be necessary.
- (2) It should be possible to flexibly add different equipment to suit the needs of the disabled.

In addition, development of assisted mobility vehicles easy to use for every user has also been desired from the viewpoint of universal design⁽³⁾. FREED development did not take the approach of developing separate general vehicle and modified vehicle types, but was instead conducted in a manner that includes the modified vehicle as a variation of the general vehicle. The assisted mobility vehicle with various specifications was simultaneously developed in order to effectively achieve the items specified below.

- (1) Establishment of the optimum FREED package layout that also includes assisted mobility specifications
- (2) Sharing of common parts between the FREED base vehicle and assisted mobility vehicles
- (3) Line production of assisted mobility vehicle with minimum changes to line facilities

Specific examples in the development of wheelchair vehicle are presented below.

3.2. Specific Examples of a Streamlined Package Layout

A package that can also be used as a wheelchair vehicle has been developed by making use of base vehicle package.

The layout of the exhaust pipe, fuel tank and fuel lines, rear suspension, harness, and other parts was investigated. This resulted in a better package layout with a wider slope, taller rear opening, and more expansive wheelchair passenger space compared to other compact class minivan wheelchair vehicles in the same class as the FREED. Note that values of these sections are almost the same as those for a middle class minivan wheelchair vehicle.

The main wheelchair vehicle dimensions and comparisons are set forth below and in Fig. 2.

- (1) Lateral width of the slope for getting in and out of the vehicle: 700 mm (+20 mm wider than other vehicles in the same class)
- (2) Height of wheelchair passenger space: 1 345 mm (+95 mm higher than other vehicles in the same class)
- (3) Opening height: 1 315 mm (+90 mm higher than other vehicles in the same class)

3.3. Specific Examples of Common Parts Shared with the Base Vehicle

After securing the main wheelchair vehicle dimensions described in section 3.2. above, common parts were shared with the base vehicle as shown in Fig. 3 below in order to reduce new investment.

The main common parts with the base vehicle are shown in Fig. 4.

- (1) Exhaust pipe in front of the low floor section
- (2) Fuel tank in front of the low floor section

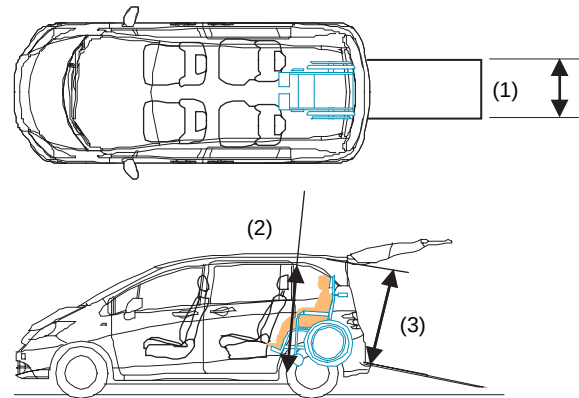


Fig. 2 Main dimensions of FREED wheelchair vehicle

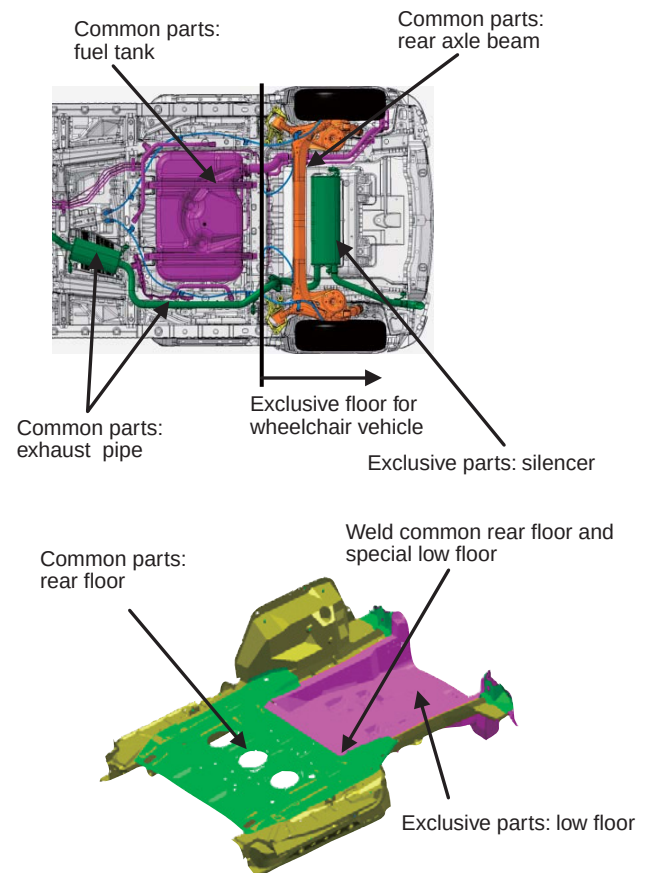


Fig. 3 Structure of FREED wheelchair vehicle

- (3) Rear axle beam
- (4) Floor (structural portion) in front of the low floor section

The above parts contribute greatly to basic vehicle performance, and the use of common parts enables the development process time to be streamlined.

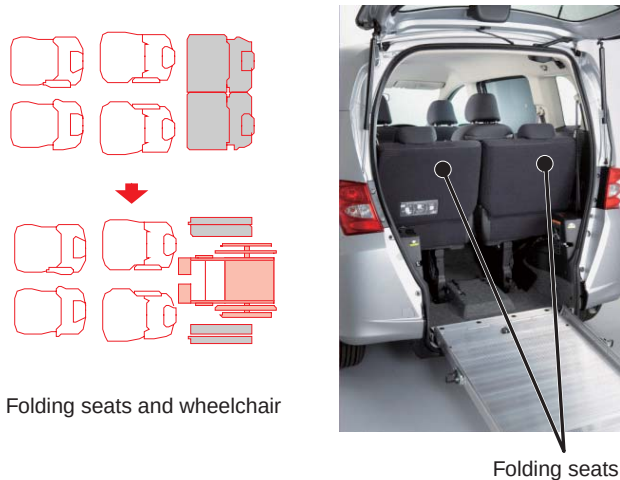


Fig. 4 FREED wheelchair vehicle (with third seat)

3.4. Compact Size Wheelchair Vehicle that also Enables Use of a Third Seating Row

Investigations were also conducted to enable the FREED wheelchair vehicle to make the use of the 7/5-passenger package layout of the base vehicle. This resulted in a compact size minivan vehicle with a 3-row seating specification that can accommodate seven adult passengers when not using a wheelchair, and a 2-row seating specification that can accommodate five adult passengers when using a wheelchair. This enabled the support of various usage formats.

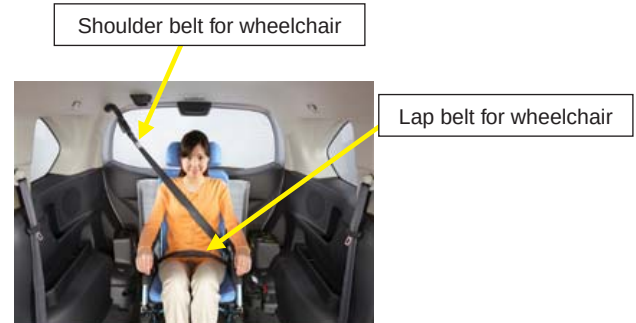


Fig. 5 Photograph of FREED wheelchair vehicle (with 3-point seatbelt for wheelchair)

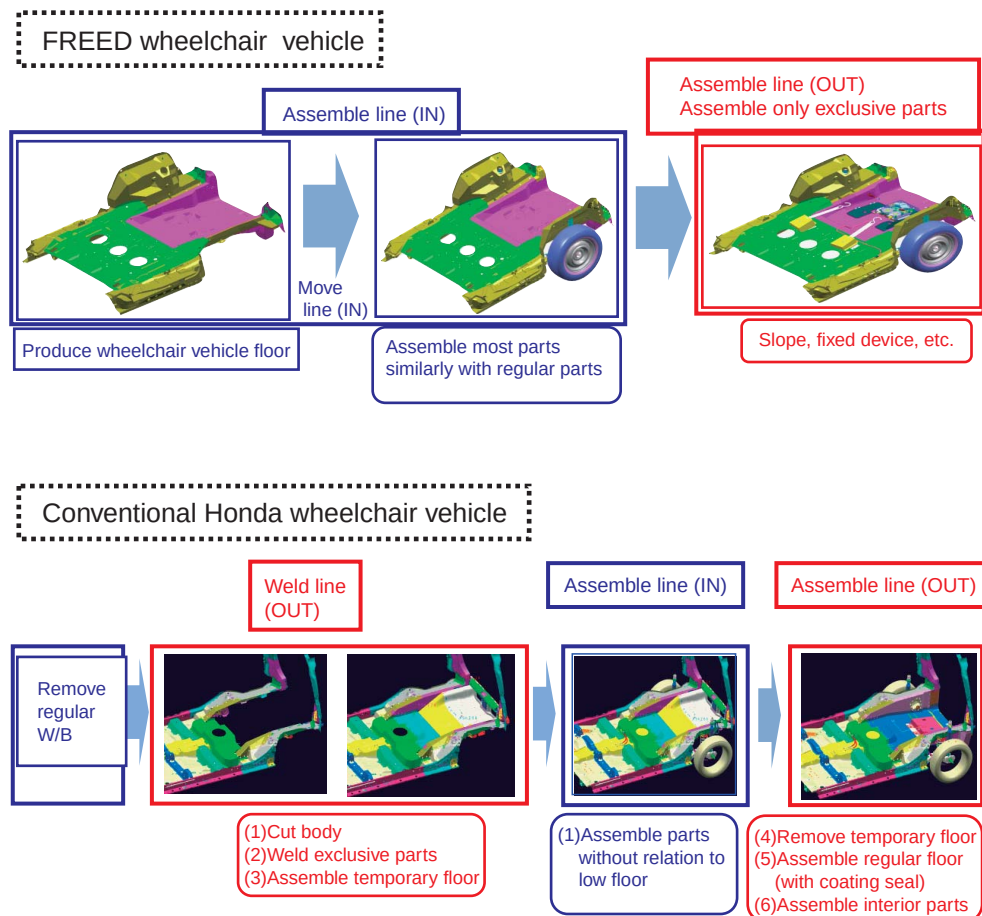


Fig. 6 Manufacturing outline of FREED and conventional wheelchair vehicles

3.5. Three-point Seatbelt and Other Features for Wheelchair Passenger

The FREED wheelchair vehicle has a 3-point ELR seatbelt for the wheelchair passenger that consists of a shoulder belt and a lap belt as shown in Fig. 5. The locations where this seatbelt is attached to the chassis share common parts with the base specifications. In addition, the two belts are separate in consideration of different usage formats. Assist grips are also provided on the backs of the second row seats, enabling the wheelchair passenger to more easily maintain position and posture.

4. Common Manufacturing Processes with the Base Vehicle

Figure 6 shows the process outline diagrams for the FREED wheelchair vehicle and the conventional Honda wheelchair vehicle. Processes enclosed by blue frames in the figure are processes in the mass production vehicle manufacturing line, and those enclosed by red frames are processes outside the mass production vehicle manufacturing line. In addition, the upper level shows the process outline for the FREED wheelchair vehicle and the lower level shows the process outline for the conventional vehicle. The conventional vehicle shown on the lower level has many processes that are performed outside the manufacturing line, such as cutting the body, welding exclusive parts, assembling and then removing a temporary floor, bolting to the floor of the base vehicle (including sealing), and assembling exclusive interior parts. In contrast, for the FREED wheelchair vehicle shown on the upper level, the only processes performed outside the manufacturing line are assembly of some parts such as the slope and wheelchair fixing devices.

5. Conclusion

This development applied the two development methods of:

- (1) Developing assisted mobility vehicles simultaneously with the base specification vehicle
- (2) Producing assisted mobility vehicles within the same production line as mass production vehicles

This enabled the use of the characteristics of the FREED package, which offers high utility in a compact size, and solidified the FREED assisted mobility vehicle concept, while also giving consideration to the viewpoints of conservation of resources and energy.

It is thought that the social need for assisted mobility vehicles and other assistance measures will increase further as Japan moves towards an aging society. However, production volumes will still be small, and it is important that assisted mobility vehicle development enables the support of additional usage formats and increases product appeal further, and those steps be taken to advance rational vehicle creation as small-scale

production vehicles.

Numerous opinions and advice were received during the course of this development from people on the front lines at care facilities, and the authors would like to take this opportunity to express their deepest thanks to everyone for their cooperation.

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■ Author ■



Kimito NISHIYAMA

Super Ultra Flat Torque Converter with Multi Plate Lock-up Clutch

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ABSTRACT

Three-dimensional CFD was used to optimize the torus section and stator blade shape, and newly developed stator blades were also employed to develop the fluid flow channels for a thin type torque converter with a flattening ratio of 50%, while maintaining torque converter performance. A multi-plate lock-up clutch with heat-resistant performance approximately twice that of a conventional single-plate lock-up clutch has been positioned in the space created by the use of these new fluid flow channels. The developed Super Ultra Flat torque converter with multi-plate lock-up clutch enables all-area LC control without increasing the overall width compared to the previous torque converter, and can be fitted in an FF vehicle. As a result, in real-world driving, the new system increased fuel efficiency by approximately 3% in US06 mode, and enabled a more direct driving feeling by shortening the time from 0 to 100 km/h at 100% throttle acceleration by 0.2 s.

1. Introduction

Heightened awareness towards global warming in recent years has led to increased demands on torque converters, which are the launch device for automatic transmissions (AT). In particular, demands are increasing to simultaneously achieve both smooth torque transfer characteristics, which are important when the vehicle launches, and direct torque transfer characteristics with little slip, which contribute to increasing fuel efficiency. At the same time, from the viewpoint of securing collision safety, design restrictions on the overall torque converter width are also important and should be observed to help prevent the AT axial dimension from increasing.

The flattening ratio of fluid flow channels is generally in the 80% range, but Honda led the industry in mass-producing an Ultra Flat torque converter with a flattening ratio in the 60% range from 2001 year models. Now, a Super Ultra Flat torque converter with multi-plate lock-up clutch has been developed with the goal of further increasing fuel economy (Fig. 1). This new torque converter consists of a lock-up clutch (LC) system that enables all-area LC control and new fluid flow channels

with a flattening ratio of 50%, and can be fitted in an FF vehicle.

This paper describes the development goals, technical issues, and countermeasures for this Super Ultra Flat torque converter with multi-plate lock-up clutch, and reports the resulting enhanced performance.



Fig. 1 Side view of developed torque converter

* Automobile R&D Center

2. Development Goals

In order to simultaneously achieve both smooth and direct torque transfer characteristics, the following goals were set for development.

- (1) Enhanced LC performance
Comparison with previous model:
LC heat-resistant performance: Double
LC lock-up performance
Operating range: $0.5 < \text{speed ratio} < 1.5$
Response time: 0.5 s or less
- (2) Torque converter performance equal to that of the previous model
- (3) Overall torque converter width equal to that of the previous model

All-area LC control is employed to achieve direct torque transfer characteristics that contribute to increasing fuel efficiency. As shown in Fig. 2, the LC operates over a wider range than the previous model, from the high throttle range to the low vehicle velocity range.

The development goal of LC heat-resistant performance is needed to enable unhindered LC lock-up and release in the high throttle range and while shifting gears (area A in Fig. 2). In addition, the development goal of LC lock-up performance is needed to reliably operate the LC when decelerating in order to extend the fuel cut time that contributes to increasing fuel efficiency (area B in Fig. 2).

Furthermore, in order to shift to LC lock-up driving without placing an excessive thermal load on the LC facing material immediately after a smooth launch, the torque capacity factor, torque ratio and other torque converter performances should be the same level as that of the previous model.

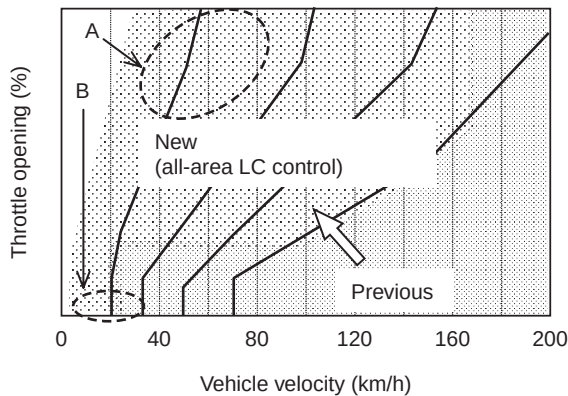


Fig. 2 Operating range of lock-up clutch

3. LC System

To achieve the LC performance development goals, LC systems were investigated from the viewpoints shown in Table 1. Single-plate LCs have a simple parts

Table 1 Comparison of LC systems

	Single-plate type (previous)	Multi-plate type
Typical layout		
LC operating pressure		
Comparison	○ : Good ⊙ : Very good	
(1) LC heat resistant-performance	○	⊙
(2) LC response time operating range	○	⊙
(3) Torque capacity	○	⊙
(4) Width	⊙	○

configuration, so they have good space efficiency and are employed in the torque converters of FF vehicles.

On the other hand, Table 1 shows that in contrast to single-plate LCs, the LC operating pressure in a multi-plate LC acts on the piston chamber, which is a closed space, so there is no need for the facing material to seal the working fluid pressure. This enables the establishment of lubrication grooves on the facing material surface, which increases the heat-resistant performance [Table 1 (1)]. In addition, the LC piston operation is not affected by the fluid flow inside the torque converter, which is advantageous in terms of LC operating range and operating time [Table 1 (2)]. Moreover, the multiple facing surfaces also provide a larger clutch capacity, which is advantageous for use in high displacement engines [Table 1 (3)].

Therefore, in consideration of the above performance comparisons and displacement of the target engines, a multi-plate LC system was selected. However, multi-plate LCs have many parts such as a piston chamber, multiple disc plates, and a clutch guide, so using the conventional fluid flow channel width would unavoidably increase the axial dimension of the torque converter. Therefore, the axial dimension of the fluid flow channels, which consist of the blades and the torus section, should be shortened to enable them to fit into an FF vehicle.

4. Development of Super Ultra Flat Fluid Flow Channels

Torque converter performance is represented by the torque capacity factor and torque ratio. The larger the torque capacity factor, the smaller the energy loss during

torque transmission, and the larger the torque ratio, the larger the driving force at launch. These torque converter performances are determined by the change in fluid momentum produced by turning the flow using the pump elements, turbine, and stator. To obtain high torque converter performance, the flow should be turned within a rotating three-dimensional space while minimizing both the secondary flow produced by imbalances between the centrifugal force and pressure gradient, and also separation flow and other flow loss factors.

4.1. Setting the Flattening Ratio and Aspect Ratio of the Torus Section

There are two approaches to shortening the axial dimension of the fluid flow channels as shown in Fig. 3. The torus section can be flattened without changing the inlet radius r_1 and outlet radius r_2 of each blade element, and the overall torus section can be reduced by keeping the flattening ratio the same and increasing the aspect ratio.

When the torus section is flattened, the curvature of the fluid flow channels becomes smaller, producing an increase in the flow loss inside the torque converter, and this tendency is particularly evident in the turbine, which generates torque by inverting the flow. On the other hand, when the aspect ratio of the torus section is increased, the radii ratio r_2/r_1 decreases as shown in Fig. 3. This makes it challenging to create a difference in flow momentum at the pump inlet and outlet, which tends to reduce the torque capacity factor.

The relationship between the axial dimension shortening effects and the ratio of change in torque converter performance of these two approaches was verified with the goal of reducing the axial dimension of

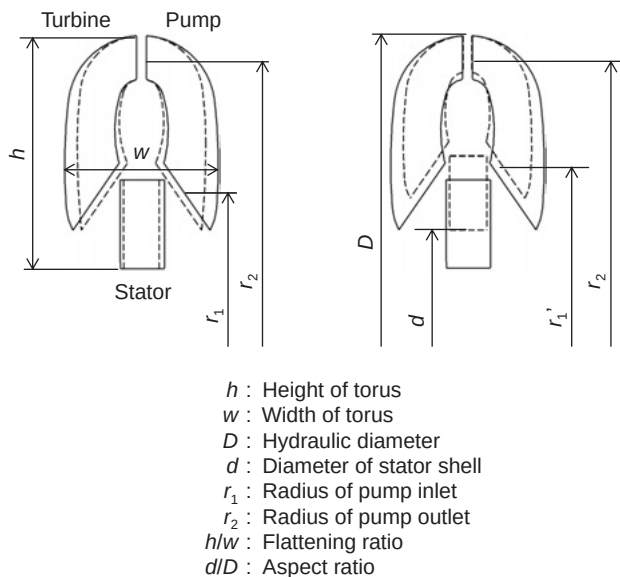


Fig. 3 Approaches to decrease axial length of torus

Table 2 Torque converter specifications and calculation conditions

Hydraulic diameter (mm)		252
Flattening ratio (%)		65
Number of blades	Pump	29
	Turbine	27
	Stator	15
Aspect ratio of torus		38
Rotation speed of input shaft (rpm)		2000
ATF temperature (°C)		85

the fluid flow channels. Table 2 shows the specifications used for this comparison. Figure 4 shows the results of calculating the torque capacity factor at a speed ratio of 0.8, which has a large influence on fuel efficiency, when the blade angle, wing form function, and number of blades are kept the same, and only the flattening ratio or the aspect ratio of the torus section is changed.

The torque converter performance was calculated from three-dimensional CFD by general-purpose software that uses the finite volume method for the separation flow. The turbulence kinematic model was a standard κ - ϵ model⁽¹⁾, and the SIMPLE method⁽²⁾ was used as the analysis algorithm.

The verification results confirmed that to obtain the same axial dimension shortening effects, reducing the flattening ratio produced a smaller ratio of decrease in torque converter performance than increasing the aspect ratio.

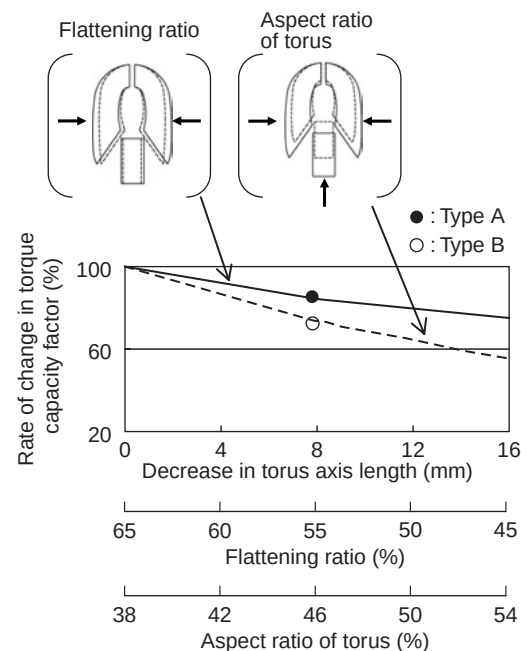


Fig. 4 Influence of flattening ratio and aspect ratio of torus

In addition, Fig. 5 shows the results of calculating the turbulence kinematic energy for Type A and Type B shown in Fig. 4. This turbulence kinematic energy κ is obtained by solving the κ - ε model, and shows that the flow loss increases as κ becomes larger.

Figure 5 shows that compared to Type A, the turbulence kinematic energy tends to increase together with the internal flow speed for Type B. Based on this and the aforementioned verification results, it was decided to keep the same aspect ratio and reduce the flattening ratio of the fluid flow channels compared to the previous torus section.

After the aspect ratio and flattening ratio of the torus section were set as described above, efforts were made to further reduce flow loss due to the core shape of the torus section, the blade shape, and the stator blades.

4.2. Optimization of the Core Shape of the Torus Section and Blade Shape

To optimize the core shape of the torus section and the blade shape, it is necessary to verify an immense number of combinations of design parameters, such as the torus section shape, the fluid flow channel area setting of each blade at the inlet, the blade angle, blade output and inlet radii, wing form function, bias angle, number of blades, and the like.

Therefore, a performance prediction method⁽³⁾, developed by Honda, based on one-dimensional angular momentum theory was used. The features of this calculation prediction method are as follows.

- (1) Loss within the fluid flow channels due to flattening is expressed using a virtual curved tube model.
- (2) The effects of stator loss on torque capacity characteristics are expressed using the outlet flow angle of the stator as the parameter.

This calculation prediction method was used to determine the base combination of design parameters, such as the fluid flow channel area at the pump outlet, the

flattening ratio, blade angle, blade outlet and inlet radii, and other reference parameters. Then, three-dimensional CFD was used to set the wing form function, bias angle, and number of blades from the viewpoint of optimizing the pressure gradient inside the turbine fluid flow channels.

Figure 6 compares the results for the fluid flow channel section, which has the largest calculated turbulence kinematic energy value, in order to illustrate the change before and after optimization in the turbulence kinematic energy inside the turbine fluid flow channels in the stall state. As shown in Fig. 6, the reduction effects on the flow loss are particularly large on the suction side near the turbine inlet, and the average reduction in flow loss within the fluid flow channel space was confirmed to be approximately 13%.

4.3. Development of New Stator Blades

The conventional blade shape that has been used for the stator of Ultra Flat torque converters has a bulbous protrusion in the part where the coring and the pressure surfaces of the stator blades contact. This bulbous blade shape was further developed, and the blade shape, fluid flow channel area, and number of blades were newly set from the viewpoint of reducing flow loss, produced by the separation flow phenomenon⁽⁴⁾, that occurs on the suction side (Fig. 7). This bulbous shape is able to reduce the flow loss that occurs around the stator coring without complicating the die-cast mold division plans, which is the method used to manufacture the stator. This effectively increases the torque ratio by approximately 5% in the stall state that has a fast internal flow speed.

Figure 8 compares the calculated turbulence kinematic energy at the stator blade center section in the stall state before and after optimization. These results show that the turbulence kinematic energy at the front and rear edges of the stator blade decreases [Fig. 8(1)].

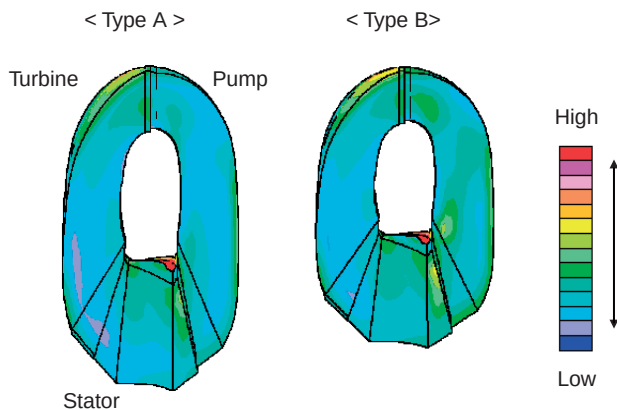


Fig. 5 Calculation results of turbulence kinematic energy

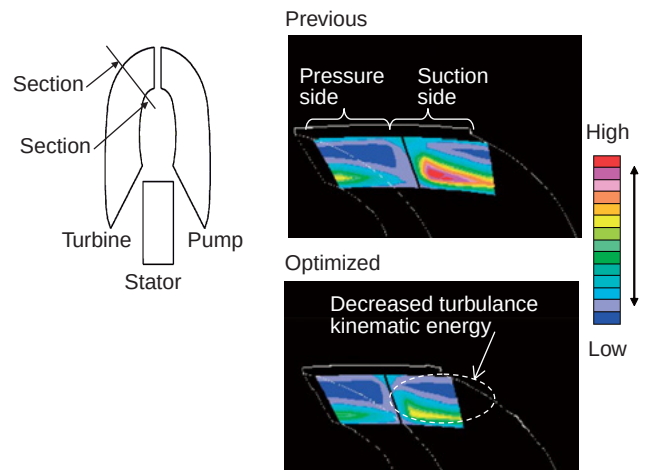
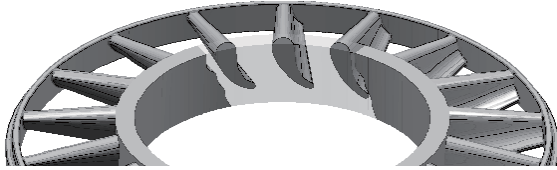


Fig. 6 Calculation results of turbulence kinematic energy at turbine inlet

Previous



Optimized

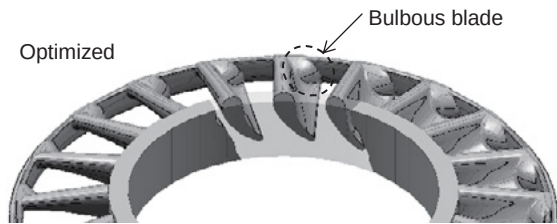


Fig. 7 Comparison of conventional and optimized stator blades

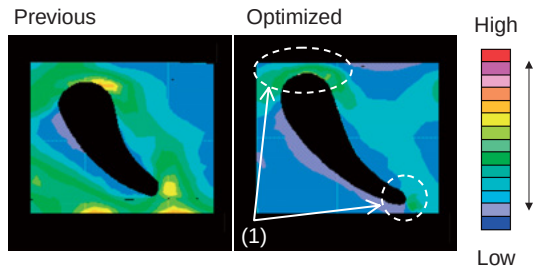


Fig. 8 Calculation results of turbulence kinematic energy at stator blades

5. Confirmation of Performance and Effects

5.1. LC Performance

A multi-plate LC with the structural merits shown in Table 1, aforementioned, was positioned in the space created by employing the Super Ultra Flat fluid flow channel developed by the above process. The LC size, facing surface inner and outer diameters, and number of clutch plates were set based on the allowable axial dimension and required clutch capacity, and the number and shape of the lubrication grooves on the facing material surface were set in consideration of the LC heat-resistant performance and clutch drag torque.

Figures 9 and 10 show the results of comparing LC lock-up performance and LC heat-resistant performance, respectively, for the previous and new torque converters. Figure 9 shows that the LC operating range is wider and the LC response time is also shorter and more stable compared to the previous torque converter.

Figure 10 shows the results obtained by measuring the facing surface temperature during LC slip control in the assembled torque converter state under the test conditions

Table 3 Test conditions

Measuring point: ●	
Single-plate type (previous)	Multi-plate type
ATF temperature (°C)	100
LC transmitted power (W)	3100
Cooling flow rate (L/min)	4

given in Table 3. This shows that during LC slip control, the facing surface temperature of the previous single plate LC exceeds 200°C at 3 100 W in the high load range. In contrast, the temperature for the new torque converter at this time is approximately 160°C, which is within the tolerance range.

In addition, the confirmation results showed that the

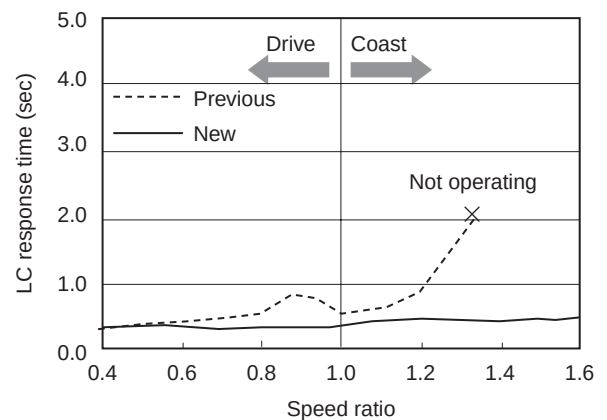


Fig. 9 Test results of LC response time

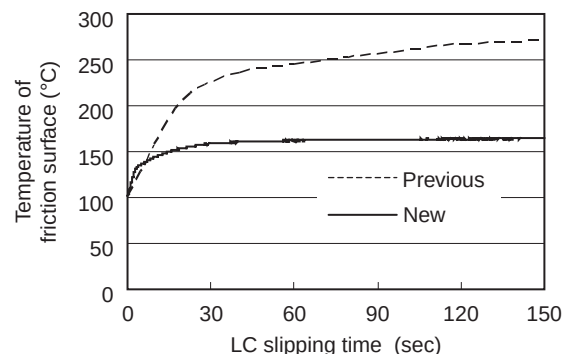


Fig. 10 Test results of facing temperature

Table 4 Comparison of torque converter specifications

		Previous	New
Hydraulic diameter (mm)		252	260
Flattening ratio (%)		65	50
Number of blades	Pump	29	31
	Turbine	27	29
	Stator	15	21

amount of LC work needed to reach a facing surface temperature of 160°C was approximately 1 330 kW for the single-plate LC. This confirmed that the new LC has LC heat-resistant performance almost double that of the previous LC.

These enhancements in both LC lock-up performance and LC heat-resistant performance enabled LC operation from the high throttle range to the low vehicle velocity range in which LC lock-up was not possible for the previous torque converter, and increased the LC lock-up driving frequency by approximately 1.5 times in real-world driving. When applied in an actual vehicle, the effects of this all-area LC control increased fuel efficiency in US06 mode by approximately 3%, and enabled the time from 0 to 100 km/h at 100% throttle acceleration to be shortened by 0.2 s.

5.2. Torque Converter Performance

Table 4 compares the main specifications of the new and previous torque converters. Figure 11 shows the torque converter performance for a 3.5-liter engine. This shows that even though the flattening ratio was changed from 65% to 50%, the new torque converter achieved the equivalent torque capacity factor and torque ratio to those of the previous torque converter.

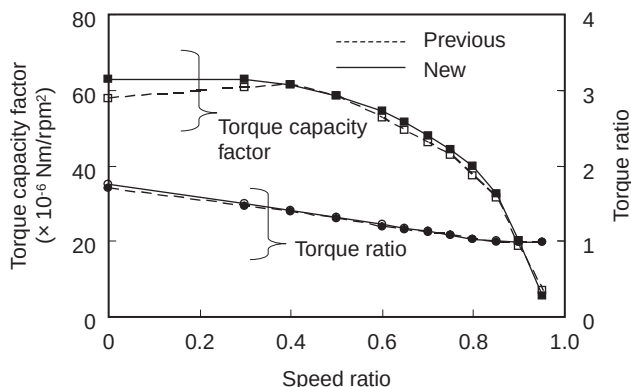


Fig. 11 Torque converter performance

6. Conclusion

This development simultaneously achieved both smooth torque transfer characteristics, which are important when the vehicle launches, and direct torque transfer characteristics with little slip, which contribute to increasing fuel efficiency, within the design restrictions on the overall torque converter width.

All-area lock-up control was enabled by employing thin-type fluid flow channels with a flattening ratio of 50% and a multi-plate lock-up clutch that has heat-resistant performance approximately double that of the previous single-plate lock-up clutch. As a result, the new system increased fuel efficiency by approximately 3% in US06 mode, and enabled a more direct driving feeling by shortening the time from 0 to 100 km/h at 100% throttle acceleration by 0.2 s, and the like.

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Research on FM Band Radio Noise Visualization System for Hybrid Vehicle

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ABSTRACT

Research has been conducted on a system enabling visualization of FM band radio noise produced by the high-voltage components in a hybrid vehicle.

The system is composed of an electromagnetic field detection device that detects electromagnetic radiation noise, a measuring device and interface that scan the output of the electromagnetic field detection device, a spectrum analyzer that analyzes the frequency of the electromagnetic radiation noise, and a PC that processes the data.

The system employs a camera mounted above the high-voltage components in the hybrid vehicle. The camera records images of the components. In combination with the measuring device, this enables noise to be detected in each operating state of the high-voltage components. Electric field strength data that has been analyzed for frequency by the spectrum analyzer and photographic data are then synthesized to produce contour diagrams, enabling noise sources to be identified.

This method has enabled simple identification of noise sources and noise frequency bands, thereby reducing the time needed to identify a noise source and apply a countermeasure to one-third against conventional methods.

1. Introduction

Today's spiraling crude oil prices and increasing concern towards the protection of the environment are generating greater demand for hybrid vehicles, given their high fuel efficiency and effectiveness in reducing CO₂ emissions.

In the development of hybrid vehicles, the FM band (76-108 MHz) radio noise produced by the high-voltage components in the vehicles represents an issue from the perspective of commercial appeal. The common mode noise and differential mode noise produced by the high-voltage components propagates through the vehicle body, cables, and harnesses, and is radiated as electromagnetic radiation noise. This electromagnetic radiation noise is received by the radio antenna, and introduces extraneous noise to the audio output of the radio, thus affecting the consumer appeal of the radios in the vehicles.

In order to contribute to a resolution of this issue, research was conducted on an FM band radio noise visualization system that would enable identification of

noise sources among the high-voltage components and rapid verification of the effectiveness of countermeasures. This paper will discuss the development concept and the developed visualization technology.

2. System Concept

Figure 1 shows a flow chart for conventional noise source identification. Figure 2 shows the conventional method of identification and different types of noise. In the conventional process, frequencies at which noise increases are identified through hearing tests, manual measurements of noise in the high-voltage components are taken using an electromagnetic field probe, and a spectrum analyzer is used to analyze the frequencies, after which countermeasures are implemented. This method uses hearing tests to determine noise increases, without identifying noise increases in other components or other frequency ranges. Due to the repetition of the process, a certain amount of time is needed for the identification of noise sources and the adoption of countermeasures.

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The developed system is the product of research on an evaluative system enabling rapid visual quantification of noise sources.

The system concept was established as follows:

- (1) Able to automatically measure the common mode noise produced between the high-voltage components and vehicle body ground and the differential mode noise produced between the circuits, and to identify noise sources via a visual display
- (2) Able to specify and analyze the frequency of the noise source

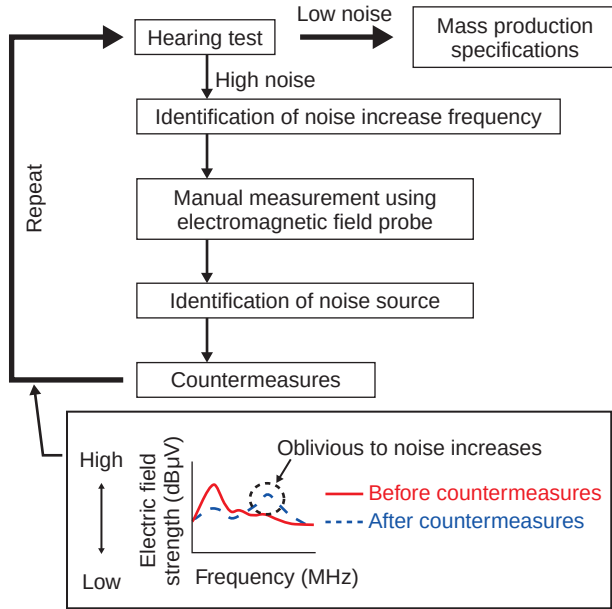


Fig. 1 Noise source specific flow chart

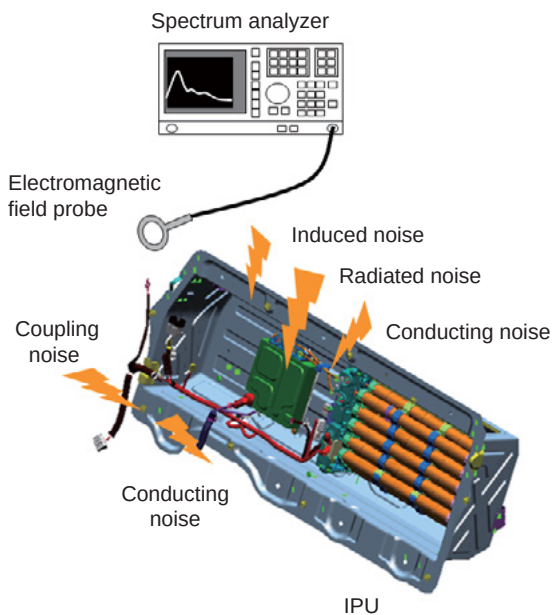


Fig. 2 Conventional method and kind of noise

3. System Overview

The Intelligent Power Unit (IPU), a high-voltage component fitted in a hybrid vehicle, is made up of a Power Control Unit (PCU), a high-voltage battery, and a junction board. The PCU is made up of a Power Drive Unit (PDU), a DC-DC converter, and an ECU. The configuration of the units is shown in Fig. 3.

The noise produced by these units propagates through the cases, covers, harnesses, cables, and the vehicle body, and is radiated.

3.1. System Configuration and Specifications

Figure 4 shows the configuration of the developed system. An electromagnetic field detection device is positioned above the high-voltage components in a hybrid vehicle and measures electromagnetic radiation noise by conducting scans in an X-Y plane. A spectrum analyzer is used to analyze the frequency of the noise. The analyzed noise electric field strength data is synthesized with a photograph of the high-voltage components that has been taken separately, to enable the distribution of the electric field strength of the noise to be visualized and displayed as a contour diagram, thereby enabling identification of the noise source.

Table 1 shows the specifications of the developed FM band radio noise visualization system.

3.2. Configuration of Measuring Device

The measuring device has both large and small configurations, which can be selected depending on the area in which the measurements will be conducted. In order to increase the measurement sensitivity of the probe, both the large and small devices are able to be moved up

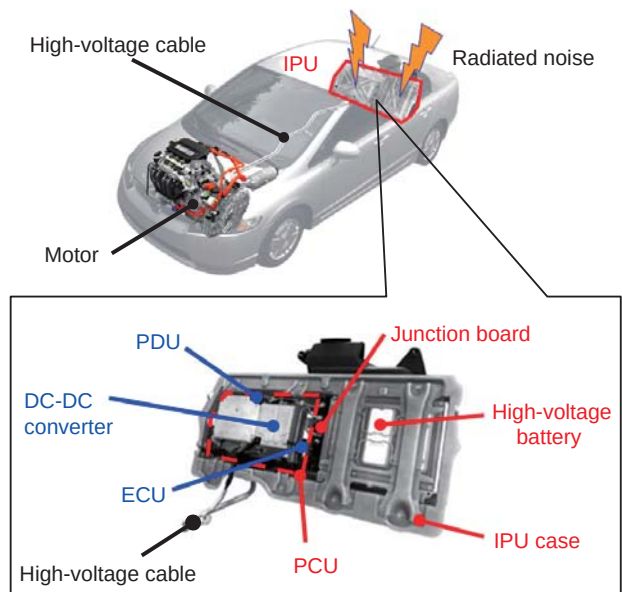


Fig. 3 IPU configuration

Table 1 Noise visualization system specifications

Item	Specification	
	Large measuring device	Small measuring device
Measurement object	High-voltage electronic components	Small electronic components and harness
Electromagnetic field detection device	4 probes in array configuration	3 probes in series configuration
Sweep method	X-Y plane	Y axis
Z axis movable range	50 (mm)	50 (mm)
Measurement area	500 × 600 (mm)	60 × 100 (mm)
Detailed measurement area	Arbitrarily set	Arbitrarily set
Measurement prohibition area	Arbitrarily set	—
Pitch measurement	1, 2, 4, 5, 10, 20, 50 (mm)	1, 2, 4, 5, 10 (mm)
Probe type	Vertical magnetic field probe Horizontal magnetic field probe Electric field probe	
Amplifier amplification rate	40 dB	
Measurement frequency bandwidth	76 - 108 MHz	
Frequency resolution	1001 point	
Display resolution	Electric field strength is displayed by dividing the range of the maximum value and the minimum value into 8 sections.	

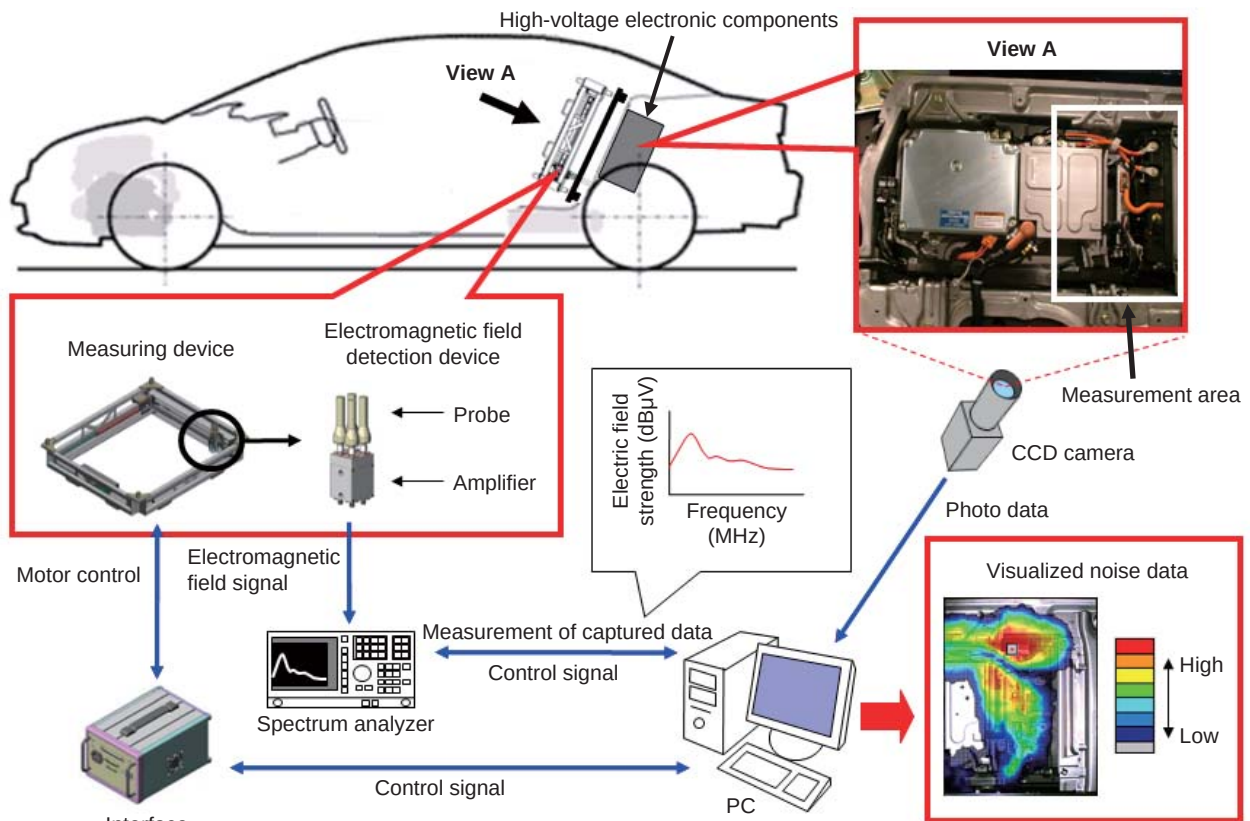


Fig. 4 Configuration of noise visualization system

and down along the Z axis in order to bring the probe closer to the object of measurement.

The mechanism enabling vertical motion along the Z axis in the large measuring device is a plunger, fitted to the amplifier of the electromagnetic field detection device, that enables the height to be adjusted in 5 mm increments from 0 to 50 mm, thus allowing adjustment of the distance between the high-voltage components and the probe. The device can therefore be adjusted for different vehicle layouts. The configuration of the large measuring device is shown in Fig. 5.

The small measuring device is provided with four feet, the height of which can be independently adjusted using feed screws, to enable adjustment of the distance of the probe and positioning of the device in relation to the raised and recessed sections of the structures being measured, thus enabling measurements of harnesses and small electronic components. The height can be adjusted from 0 to 50 mm, and each single rotation of a foot adjusts the height by 0.7 mm. The device is also provided with a quick release mechanism for rougher adjustments, and pressing buttons at the sides of the feet enables the height of the device to be changed rapidly. Figure 6 shows the configuration of the small measuring device.

The electromagnetic field detection device is made up of a probe and an amplifier. In order to increase the sensitivity of the probe and match its impedance to the impedance of the 50Ω spectrum analyzer coaxial cable,

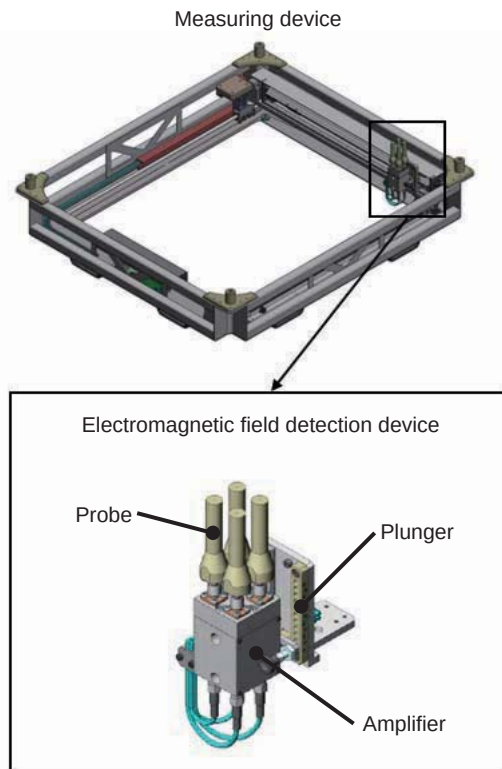


Fig. 5 Configuration of large measuring device

the amplifier is positioned directly below the probe. Figures 5 and 6 show the configurations of the electromagnetic field detection devices.

In order to increase measurement speed and reduce measurement time, the large measuring device employs four probes positioned in an array, enabling four sections of the object of measurement to be measured with each movement of the probes in the X-Y plane.

Because the purpose of the small measuring device is to conduct measurements of harnesses and small components, it employs three probes arranged in a line, enabling the measurement area to be set narrow, and helping to ensure that measurement accuracy does not decline. The specifications of the probes have been selected to enable measurement of vertical magnetic fields, horizontal magnetic fields, and electric fields. Figure 7 shows the internal configurations of the probes.

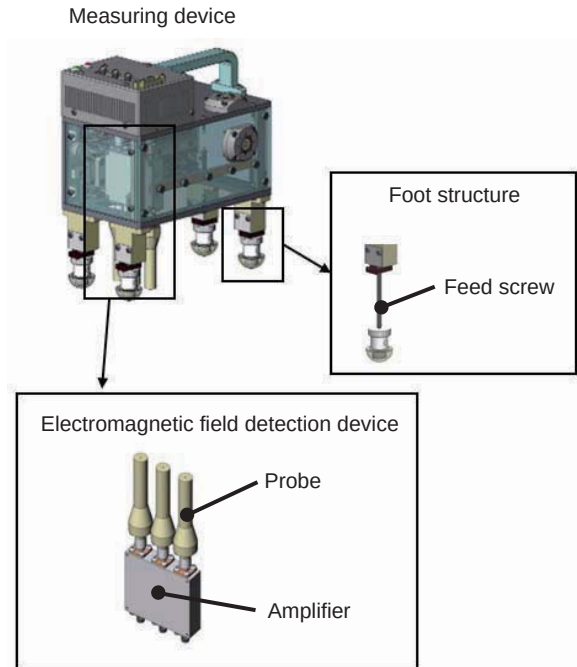


Fig. 6 Configuration of small measuring device

Vertical magnetic field probe	Horizontal magnetic field probe	Electric field probe
Ferrite cores Coil	Coil	Metal plate

Fig. 7 Internal configuration of probes

4. Details of Visualization Technology

4.1. Mechanism of Generation of Electromagnetic Radiation Noise

When high-frequency current passes through a conductor, a magnetic field is generated. Through the principle of electromagnetic induction, changes in the magnetic field produce electromotive force, and this

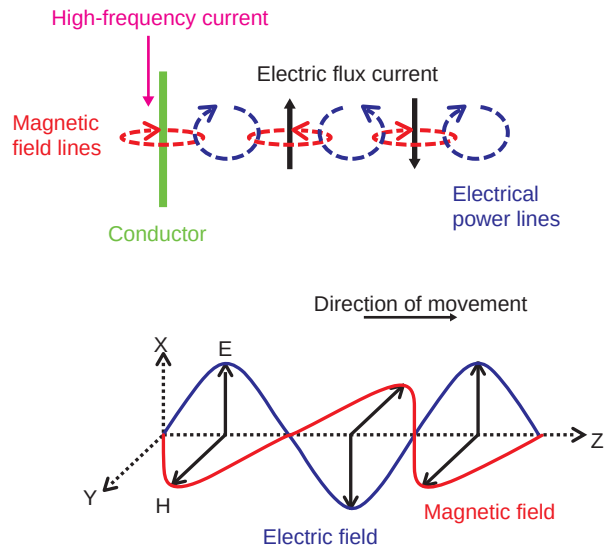


Fig. 8 Mechanism of generation of electromagnetic radiation noise

generates an electric field. Changes in the electric field produce an electric flux current, generating a magnetic field. In this way, electric fields and magnetic fields are formed successively in a chain reaction, and spread through space as electromagnetic radiation noise ^{(1),(2)}. The mechanism of generation of electromagnetic radiation noise is shown in Fig. 8.

4.2. Principle of Noise Visualization

Measurement areas are established in the high-voltage components. The electromagnetic field detection device scans the measurement areas in the X-Y direction, and the frequencies of the detected electromagnetic field noise are analyzed by a spectrum analyzer. Data for each area are input into a PC at a resolution of 1001 points on the frequency axis, and the data for all areas is saved. The peak levels are isolated for either the entire measured frequency range or a specified range, and the noise electric field strength data for each area are displayed as strength distribution diagrams.

Figure 9 shows the principle of visualization, assuming 9 (3 x 3) measurement areas. In terms of the resolution of the displayed data, the range between the maximum and minimum points for each area is divided into eight equally-spaced intervals, which are distinguished by color; the areas of highest electric field strength noise are shown as red, and the areas of lowest electric field strength noise are shown as blue or uncolored. Software is employed for interpolation

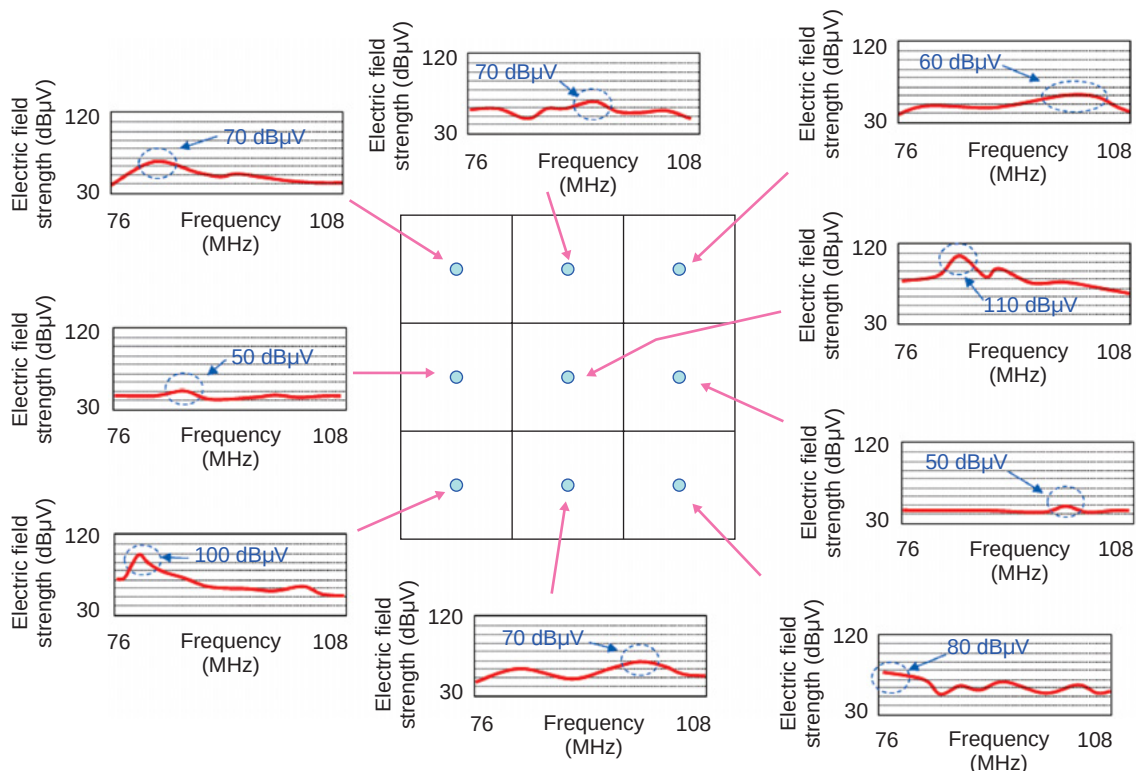


Fig. 9 Principle of visualization

processing of the obtained strength distribution diagrams, enabling a smooth contour diagram to be displayed. The interpolation program linearly interpolates strength levels with adjacent strength levels, and calculates intermediate strength levels and positions.

Figure 10 shows a conceptual diagram of interpolation. The level of strength of the electric fields at the centers of each measurement area (shown in bold) are connected to the strength levels of the adjacent areas in eight directions by straight lines, and the level of difference in strength is calculated at equal intervals. The density of change differs depending on the magnitude of the level of difference. The smaller font figures shown in Fig. 10 are intermediate levels obtained from calculations. Identical levels obtained using this processing method are connected using identical colors, enabling a diagram of the distribution of strength levels to be formulated. The processing is applied to all the measurement areas in order to calculate intermediate levels and formulate a strength distribution diagram.

4.3. Measurement Resolution and Specification of Areas

The developed system enables identification of noise sources by superimposition of the noise electric field

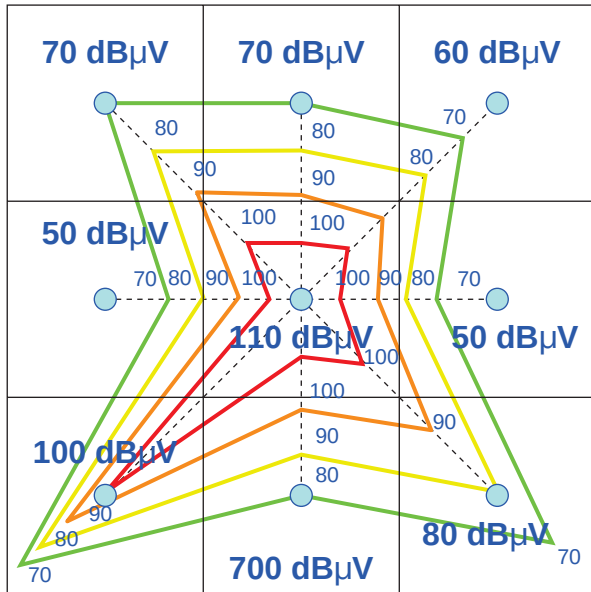


Fig. 10 Concept of interpolation

Parts	Size (mm)	Pitch measurement (mm)					
		10	20	30	40	50	60
PCU	600						
PDU	200						
DC-DC converter	150						
Connector	50						
Cable	10						
Harness	2						

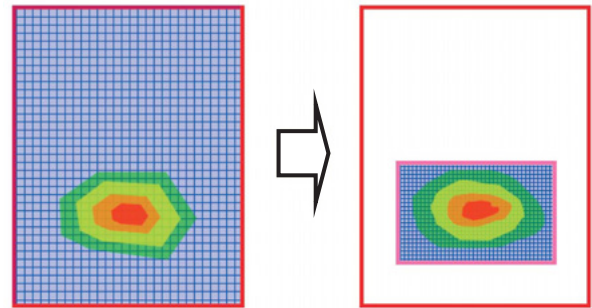
Fig. 11 Pitch measurement

strength data on a photograph of the object of measurement. This makes measurement resolution an important factor. As shown in Fig. 11, the resolution tolerances for the measuring device in relation to the size of the high-voltage components were studied, and resolution tolerances of 1, 2, 4, 5, 10, 20, and 50 (mm) were set for the device. Measurement of a measurement area of 500 x 600 (mm) at a minimum resolution pitch of 1 mm would take an enormous amount of time, and would therefore be impractical. Therefore, the area producing noise is first rapidly identified at a rough resolution, with a maximum pitch of 50 mm. Next, a measurement resolution is selected on the basis of the size of the components in the area producing noise, and the noise source is identified with more detailed measurements. This method has enabled measurement time to be reduced.

The developed system therefore incorporates, in addition to a function enabling measurement resolution to be set, a function enabling the measurement area to be freely set. Figure 12 shows an image of a detailed measurement.

4.4. Frequency Analysis

A pointer indicating the maximum area of noise electric field strength in the relevant bandwidth appears on the noise electric field strength distribution displayed as a contour diagram, and the spectrum of this area is determined. The system is also provided with a function enabling the pointer to be freely moved in order to enable the spectrum of each area to be determined. In addition, the system enables the range of frequencies for visualization and display as contour diagrams to be freely specified, and results can therefore be restricted to the frequencies at which noise increases in hearing tests and



	Measurement area: 500 × 600 mm
	Detailed measurement area: (Ex)
	Pitch measurement: 50 mm
	Pitch measurement: (Ex) 10 mm

Fig. 12 Image of detailed measurement

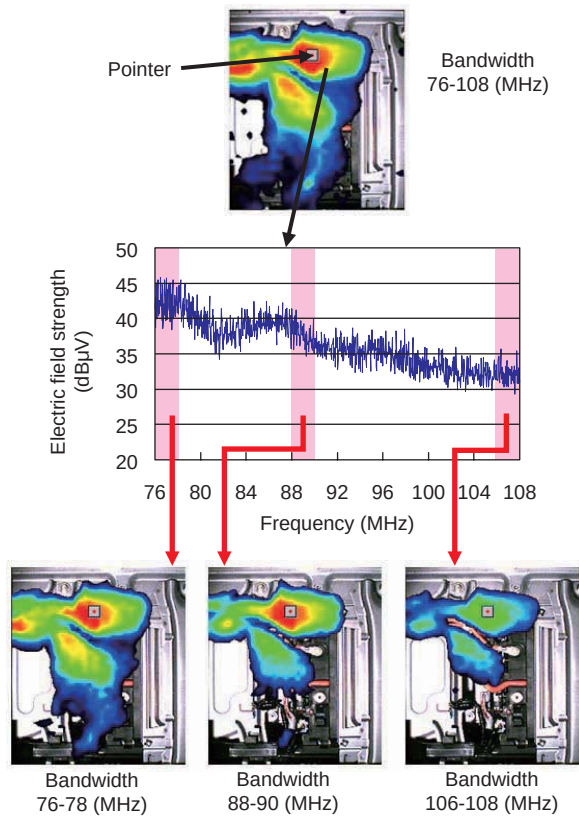


Fig. 13 Frequency analysis

the electric field strength distribution at these frequencies rendered as contour diagrams, thus enabling detailed noise source identification. Figure 13 shows a contour diagram and spectrum analysis data for the entire bandwidth, and contour diagrams for specific partial bandwidths.

5. Results of Verification

The effectiveness of the developed system was verified using a 2009 model year INSIGHT. The tests demonstrated the effectiveness of the system in identifying

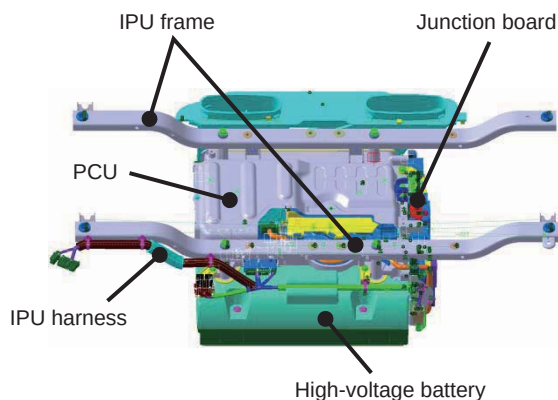


Fig. 14 IPU configuration

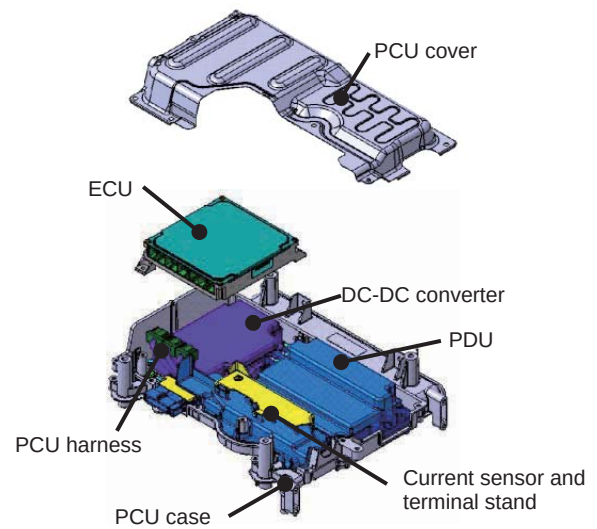


Fig. 15 PCU configuration

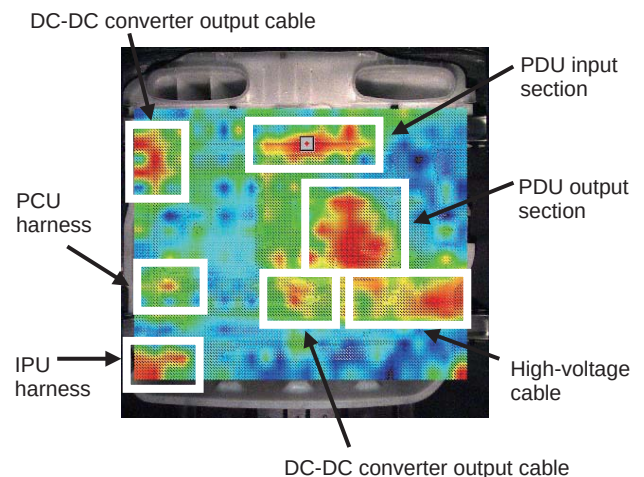
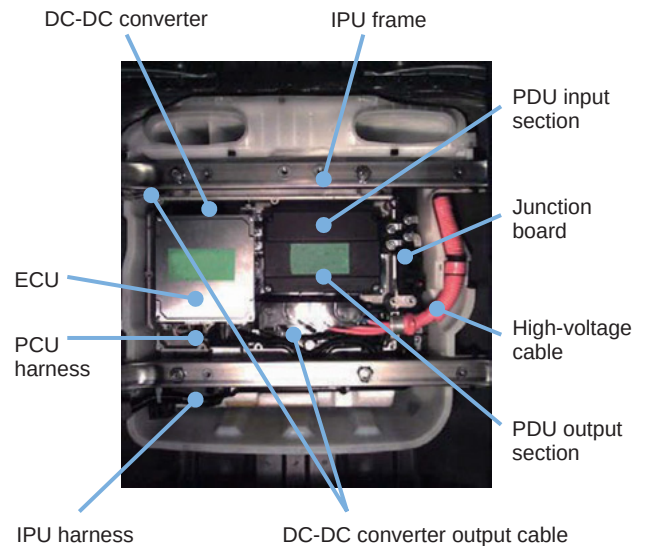


Fig. 16 Identification of noise sources

noise sources and propagation paths. In addition, the system enabled identification of noise sources in one-ninth or less than one-ninth of the time and implementation of countermeasures in one-third or less than one-third of the time required by the conventional method.

The high-voltage components of the 2009 model year INSIGHT used in the verification are shown in Figs. 14 and 15. The IPU is made up of a PCU, high-voltage battery, junction board, IPU harness, and IPU frame. The PCU in the IPU is made up of a PDU, DC-DC converter, ECU, PCU harness, PCU case, and PCU cover.

5.1. Identification of Noise Sources and Propagation Paths

A large measuring device was positioned above the IPU, and measurements were conducted of the interior of the PCU with the PCU cover removed, in order to identify which of the components were noise sources, and which harnesses and cables were propagation paths.

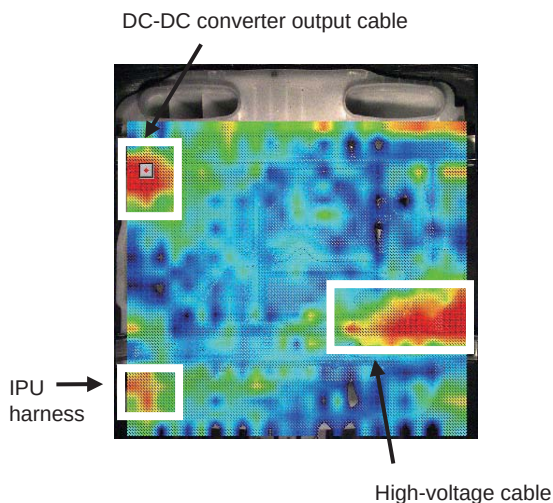
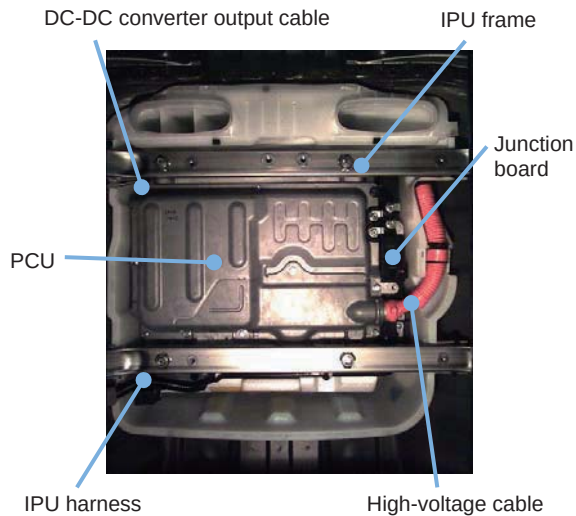


Fig. 17 Identification of propagation route

Figure 16 shows the results of measurement of the noise electric field strength distribution of the electromagnetic field noise radiating from each component in the PCU. The results show that the system was able to conduct a detailed identification of noise sources in the IPU, with noise being identified as being produced by the input and output sections of the PDU, the output section of the DC-DC converter, the PCU harness, IPU harness, DC-DC converter output cable, and high-voltage cable.

Next, the PCU cover was replaced, and the system was used to determine which paths the noise produced by each of the components inside the PCU was propagating along and radiating from. Figure 17 shows the noise electric field strength distribution. These results show that the

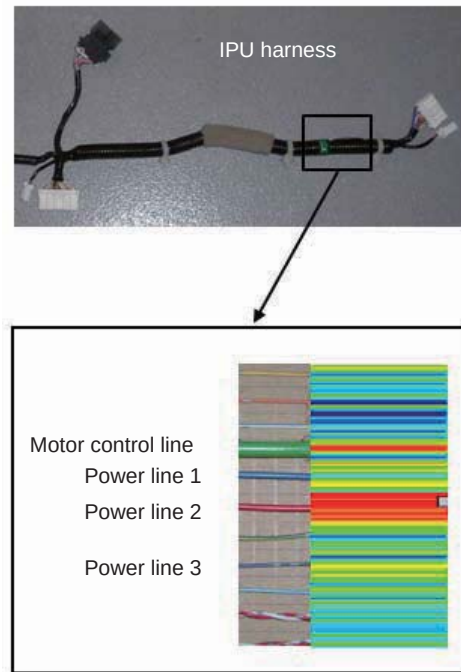


Fig. 18 Measurement results for harness

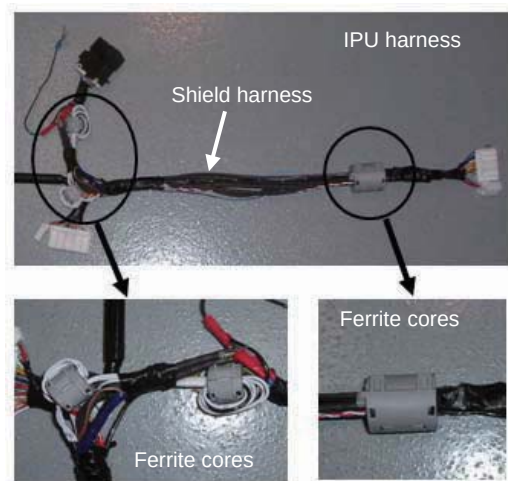


Fig. 19 Countermeasures

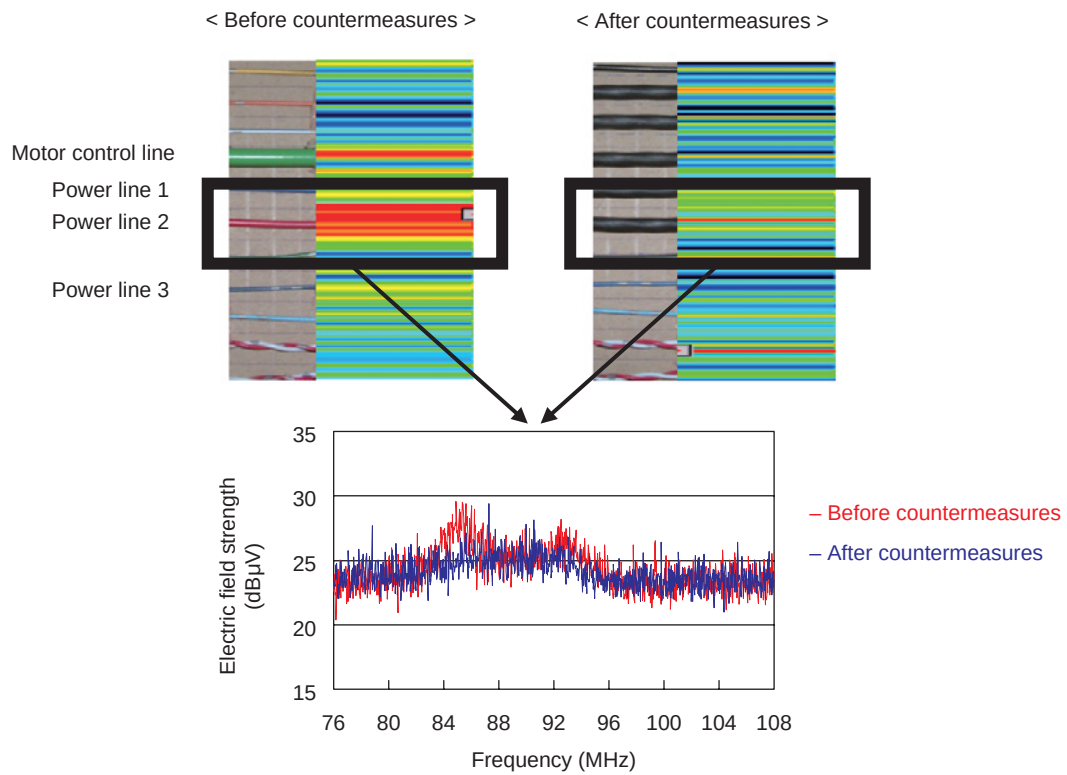


Fig. 20 Noise reduction effect of IPU harness

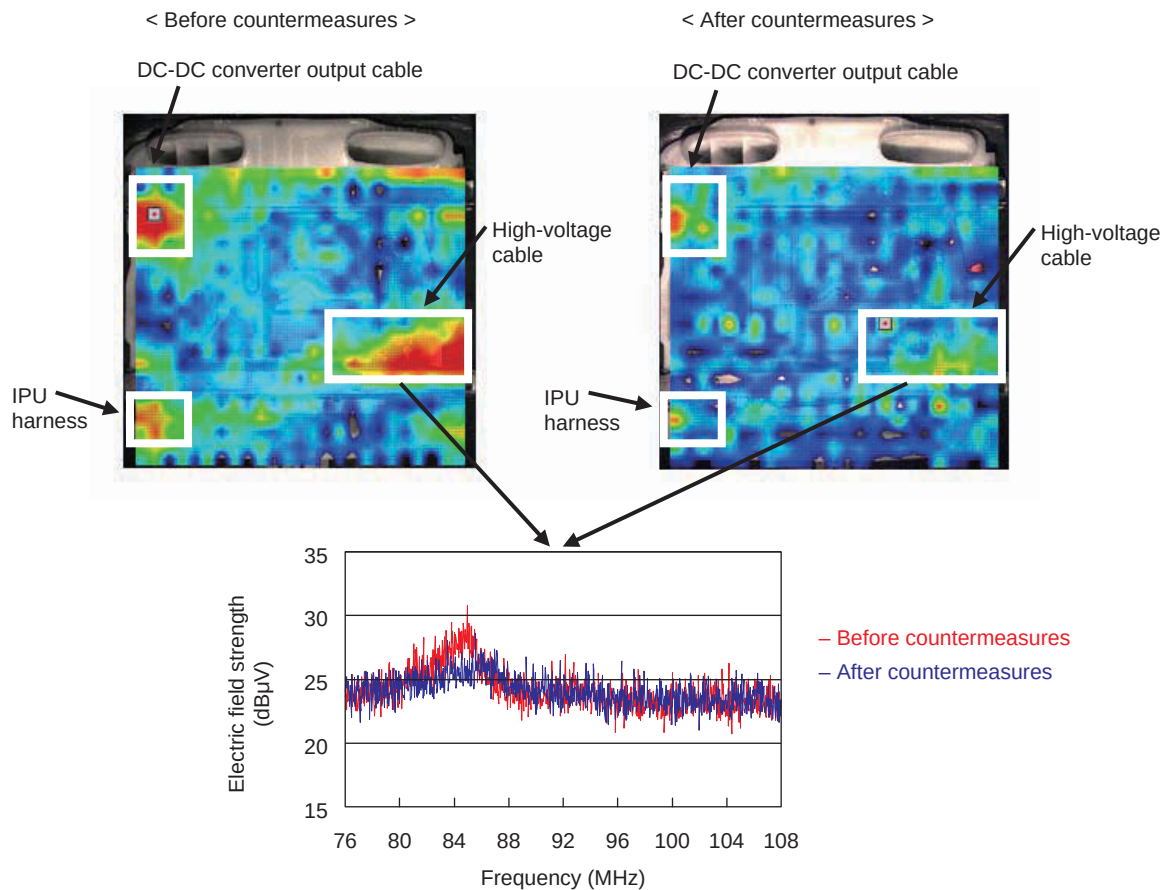


Fig. 21 Noise reduction effect of IPU unit

noise produced by the PDU and the DC-DC converter is shielded by the PCU cover, and propagates along and radiates from the IPU harness, the DC-DC converter output cable, and the high-voltage cable. The system was therefore able to verify the noise reduction effect of the PCU cover, and identify noise propagation routes.

The small measuring device was then used to identify noise propagation circuits in the IPU harness. Figure 18 shows the results of measurement of the noise electric field strength distribution in the IPU harness. As the results show, the system was able to identify noise propagation from the power lines between the PDU and ECU in the PCU and the motor control line.

These results verify that the use of the large and small measuring devices, depending on the area being measured, enables detailed identification of noise sources and propagation routes.

5.2. Frequency Analysis

The frequency analysis function of the system is used to enable optimum countermeasures to be applied for the frequencies at which noise was aurally perceived as increasing.

In the 2009 model year INSIGHT, the bandwidth in which noise increased in the IPU harness noise propagation route was identified, and the noise-reducing components that would have the greatest noise reduction effect in that bandwidth were selected and reflected in the vehicle specifications.

In this case, a shield harness and ferrite cores were selected for use, and the number of windings in the core was specified. These countermeasures against noise propagating through the IPU harness are shown in Fig. 19, and their noise reduction effect is shown in Fig. 20. Figure 21 shows the noise reduction effect in the IPU unit achieved by the employment of the IPU harness as a countermeasure. This noise reduction in the IPU as a whole occurs as a result of reducing the ground impedance of the PCU and controlling fluctuations in the electric potential of the ground by grounding the drains at both ends of the shield harness against the PCU case and vehicle body.

6. Conclusion

Research was conducted on an FM band radio noise visualization system, with the following results.

- (1) Fitting the measuring device in the vehicle enabled identification of noise sources in relation to the load on the high-voltage components.
- (2) The employment of automatic measurement, the use of probes positioned in an array, and the employment of detailed measurement functions enabling the measurement resolution to be set and the measurement area to be specified reduced measurement time.
- (3) The display of contour diagrams and pointers enabled

noise sources to be identified instantaneously, and enabled frequency components to also be identified.

- (4) The system was able to clearly identify noise sources, and enabled accurate selection of specifications and components to act as countermeasures to the specific frequency bandwidths of the noise sources.

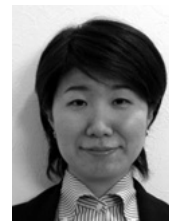
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Toshiyuki UEMORI



Masami SUNAGA

Development of Brake-by-Wire System for Super Sport Motorcycles

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ABSTRACT

Honda has developed a variety of innovative motorcycle brake technologies. Super sport motorcycles, CBR600/1000RR, have a unique combination of short wheel bases and center of gravity locations as vehicle characteristics, and the possibility towards pitching under hard braking has prevented conventional ABS from being used in these motorcycles. In addition, the characteristics of mechanical combined brake systems (CBS) make it challenging to adapt them for sport riding. Therefore, a new brake-by-wire system has been developed in which brake input is detected by a hydraulic sensor and the brake hydraulic pressure is varied by a power unit. A newly developed stroke simulator has enabled the achievement of an operation feeling similar to that of a conventional brake. In addition, the precise control of hydraulic pressure achieved by this system has enabled pitching to be controlled and made ABS/CBS compatible with sport riding.

The new system represents a first for super sport motorcycles, and has been employed in the 2009 model year CBR600/1000RR.

1. Introduction

To enhance motorcycle braking performance, Honda has offered anti-lock braking systems (ABS), combined braking systems (CBS), or combined braking systems with ABS (Combined ABS) on much of its on-road motorcycle lineup with engine sizes 250cm³ and greater^{(1), (2)}. However, the following issues have been encountered in attempting to use these systems in super sport motorcycles, CBR600/1000RR:

- (1) Super sport motorcycles have short wheel bases, increasing the possibility for some degree of pitching during high rates of deceleration, such as those that characterize ABS⁽³⁾.
- (2) Conventional CBS distributes braking force by varying the braking force on the front and rear wheels by means of mechanical valves. As a result, the front-rear braking force distribution pattern cannot be changed.
- (3) The necessity of having to make super sport motorcycles compact and lightweight and to concentrate their mass means that there is a minimal margin of space for additional equipments in the body.

- (4) The use of special CBS components such as dual channel calipers and secondary master cylinder increases the unsprung mass of the motorcycles.

In order to address these issues, a brake-by-wire system was developed that detects the brake input force by means of hydraulic sensors and applies completely independent control to the front and rear wheel braking force.

This paper will provide an overview of the new braking system, which is offered in the 2009 model year CBR600/1000RR.

2. Development Goals

The following targets were established for the development of a brake-by-wire system for super sport motorcycles:

- (1) Achievement of a balance between a high level of deceleration during ABS operation and motorcycle stability
- (2) CBS characteristics that did not impede sports riding
- (3) Appropriate size to enable fitting on a super sports motorcycle

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- (4) No significant increase in unsprung mass due to the employment of conventional brake calipers
- (5) No sense of discomfort in brake feeling in comparison to conventional brakes
- (6) Guaranteed brake operation in the event of system failure

3. Components

Figure 1 shows a schematic of the system when activated. Table 1 defines the codes used in the diagram in Fig. 1. This chapter will discuss the individual components of the system.

3.1. ECU

The electronic control unit (ECU) is the central element in the control system of the new brake system. The ECU processes signals from the wheel speed sensors (Section 3.6.) and pressure sensors (Section 3.3.) in its internal CPU, and controls the solenoid (electromagnetic) valves and motors, and conducts system diagnoses.

3.2. Power Units

The power units adjust brake pressures in response to control signals from the ECU. Figure 2 shows the basic configuration of the power unit and the initial set position of the power piston. When the power piston moves upward from this set position, the pressure on the brake caliper increases. The pressure decreases as the power piston returns to its set position. The motors and power pistons have been positioned in parallel configurations in

Table 1 Codes used in diagram of brake-by-wire system

Codes	Meanings
SNOM	Solenoid valve, normal open, master cylinder
SNCS	Solenoid valve, normal close, simulator
SNCP	Solenoid valve, normal close, power unit
PMC	Pressure sensor (master cylinder)
PFS	Pressure sensor (fail safe)
PCC	Pressure sensor (brake caliper cylinder)
VOW1	One-way valve
VOW2	One-way valve
SWSF	Front wheel speed sensor
SWSR	Rear wheel speed sensor
SymF	Front brake stroke simulator
SymR	Rear brake stroke simulator
MCF	Front brake master cylinder
MCR	Rear brake master cylinder
LWG	Warning lamp
Batt	Battery
ECU	Electronic control unit
M	Motor

order to reduce the size of the power units. Ball screws have been employed to efficiently convert the rotation of the motors into linear motion of the power pistons.

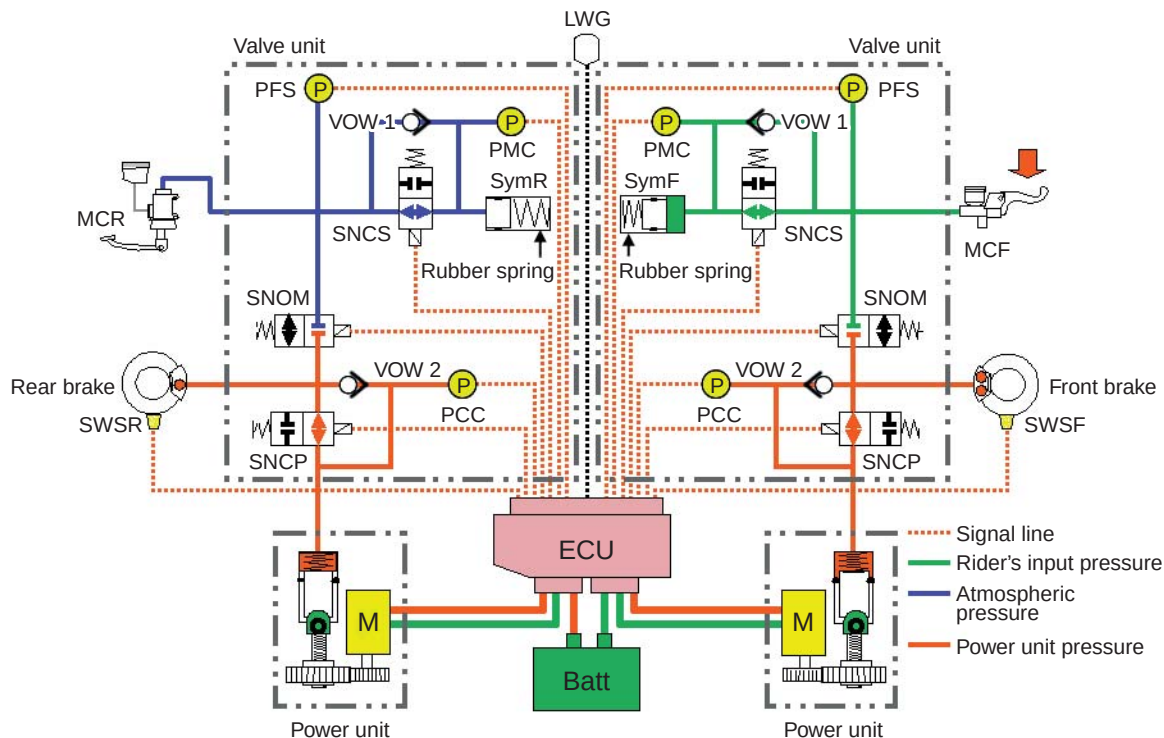


Fig. 1 Diagram of brake by wire system (Activated)

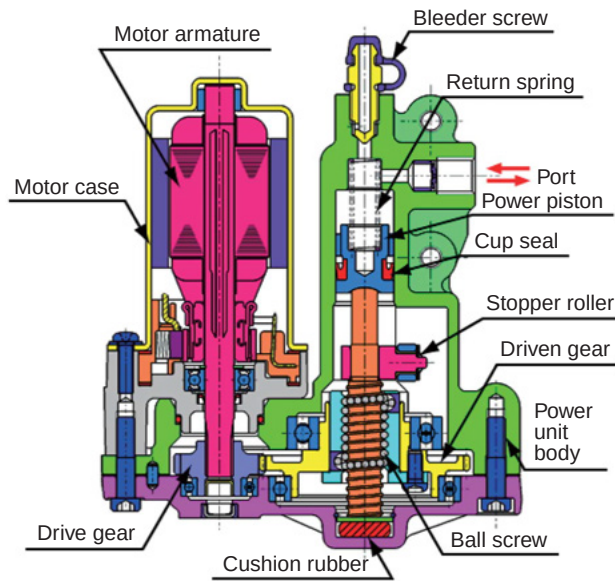


Fig. 2 Power unit

3.3. Pressure Sensors

Pressure sensors are employed to detect brake hydraulic pressure. Pressure is normally detected as the degree of strain on the pressure-detection section of the sensor. The resolution of the sensors is therefore higher the higher the degree of strain in relation to pressure, but a higher degree of strain reduces the pressure resistance of the detection section of the sensor. During motorcycle operation, there are occasions on which a high instantaneous hydraulic pressure is generated. However, pressure sensors have not previously balanced high resolution with high pressure resistance.

In the developed system, all forms of brake control are based on the input pressure, making it vital to attempt to prevent inaccuracies in pressure detection. High pressure resistance sensors (PFS) and high-resolution sensors (PMC) were therefore positioned upstream and downstream, respectively, of the SNCS solenoid valves, enabling the system to self-diagnose inaccuracies by comparing pressure measurements from the sensors. In addition, the SNCS valves close when a high input pressure exceeding a preset value is generated in order to protect the high-resolution sensors on the input side. High pressure resistance sensors, with a level of pressure resistance equivalent to that of the brake calipers, have been positioned upstream of the SNCS valves.

The high-resolution sensors that detect the brake caliper hydraulic pressure (PCC) have identical specifications to the input-side PMC sensors, and have been positioned downstream of the SNCP for the same reasons. The output pressure sensors (PCC) are able to self-diagnose inaccuracies by monitoring pressure in relation to the operating state of the motor.

3.4. Stroke Simulator

The stroke simulators reproduce the reaction force produced when the rider operates the brake lever or the brake pedal. The rider operates the brakes while feeling the reaction force generated by lever operation and the braking force. The difference of the slight reaction force transmitted to the rider's hand or foot is important in the braking control during a turning. In order to enable response to subtle operation of the brakes by the rider, it is therefore necessary for brake systems to generate a reaction force in response to input. Figure 3 shows a cross-sectional view of the stroke simulator section of a valve unit (Section 3.5.). The stroke simulators are positioned downstream of the SNCS solenoid valves, and are therefore connected to the master cylinders when the valves are open. When the rider operates the master cylinders, the pressure generated passes through the SNCS valves and acts on the simulator pistons. The load characteristic of the rubber springs that are fitted to the pistons is set to be identical to a normal brake operation feeling. In order to enable reproduction of an operation feeling even closer to that of an actual motorcycle, two rubber springs with two different load characteristics have been employed per piston in the developed system (Fig. 4).

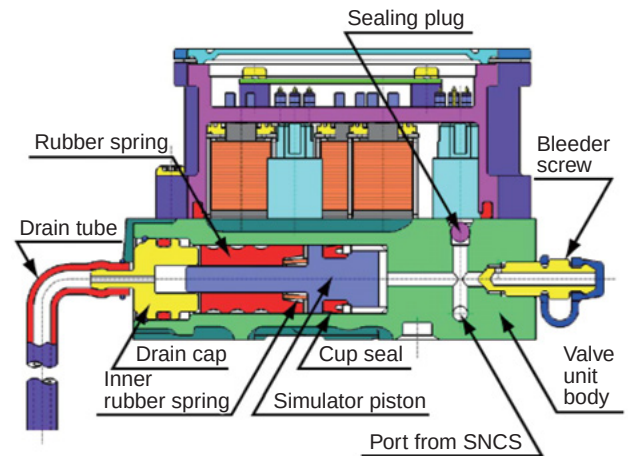


Fig. 3 Valve unit

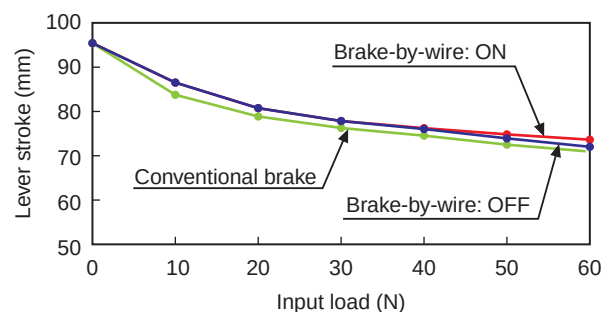


Fig. 4 Load deformation characteristics of stroke simulator

3.5. Valve Units

Solenoid valves, pressure sensors, and stroke simulators are built into the valve units. The solenoid valves are opened and closed by current from electromagnetic coils. Two types of valves are employed: Normal open valves, which close when current flows to them, and normal closed valves, which open when current flows. The solenoid valves operate in response to signals from the ECU, connecting and disconnecting the master cylinders, brake calipers, power units, and stroke simulators.

3.6. Wheel Speed Sensors

The wheel speed sensors (SWSF and SWSR) use hall elements (magneto-electric transducers) to detect the rotation of pulsar rings manufactured by punching slits in thin, circular steel sheets. This type of wheel speed sensor is widely used in the ABS systems of current production motorcycles.

4. Mounting in Actual Motorcycle

Hydraulic brake systems for motorcycles are made up of a master cylinder that generates hydraulic pressure when the rider operates the brake levers (or brake pedal), brake calipers that convert the hydraulic pressure into braking force, and brake hoses that connect these components. Figure 5 shows the layout of the developed system in an actual motorcycle.

The developed system adds an ECU, power units, valve units, and wheel speed sensors to a conventional hydraulic brake system. Due to the concentration of mass in a super sport motorcycle, there is a minimal margin of space in the body of the vehicle.

To enable the new system to be easily mounted in an actual motorcycle, it was divided into a number of smaller units.

The power units and the valve units are connected

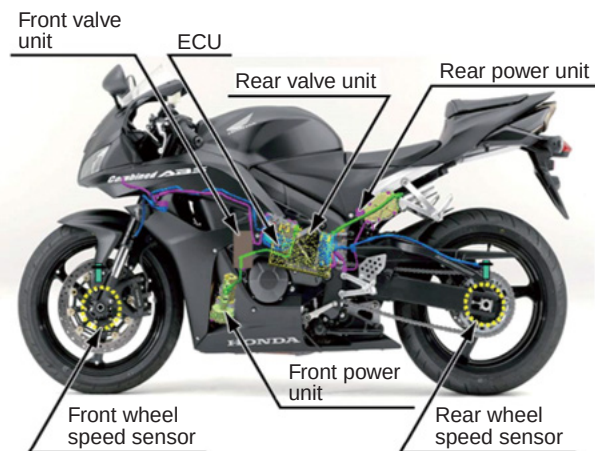


Fig. 5 Layout of brake-by-wire system

between the master cylinders and brake calipers, and the units are electrically connected to the ECU by signal lines. The system does not require pipes to connect the front and rear brakes, dual channel calipers or secondary master cylinder similar to a conventional mechanical CBS, and enables CBS to be realized using standard brake calipers. Other than the wheel speed sensors, the system does not increase the unsprung mass of the motorcycle, and thus enables the high level of dynamic performance required from a super sport motorcycle to be maintained.

5. Operation of Brake-by-Wire System

5.1. System in Deactivated State

The system diagram in Fig. 6 shows the system when it is deactivated. When the motorcycle is stopped, the master cylinders on the front wheel side (MCF) and the rear wheel side (MCR) are connected to their respective brake calipers via the normal open solenoid valves (SNOM). Due to their normal closed configuration, the solenoid valves (SNCS) that connect the master cylinders to the stroke simulators and the solenoid valves (SNCP) that connect the brake calipers to the power units are closed at this time. If the rider operates the brakes in this state, the hydraulic pressure acts directly on the brake calipers through the pathway shown in green in Fig. 6. The rider is therefore able to operate the brakes like a conventional system.

5.2. System in Activated State

The brake system goes into stand-by mode when the speed of the motorcycle exceeds 6 km/h. In this state, the SNCS solenoid valves open, and the PMC high-resolution pressure sensors are able to detect input pressure. When the input hydraulic pressure exceeds a preset level after the brake lever operation by the rider, the system enters by-wire mode (Fig. 1). The SNOM solenoid valves close the pathways from the master cylinders to the brake calipers, and at the same time the input pressure from the rider's brake operation is transmitted to the simulator piston of the SymF stroke simulator via the SNCS solenoid valve. The ECU calculates the required brake caliper pressure from the input pressure and the rate of pressure increase. As the motor rotates, the ball screws raise the power pistons, and brake hydraulic pressure is generated. The hydraulic pressure generated in the power units on the front wheel side is transmitted to the brake calipers via the SNCP solenoid valves, and braking force acts on the wheels. When the brake lever is operated, the ECU applies CBS control to the rear wheel brake power unit in order to produce the optimum braking force in relation to the brake caliper pressure on the front wheel. The system functions in the same manner when the brake pedal is operated. If the slip ratio of the wheels exceeds a preset value, the ECU puts the motor into reverse, the power pistons are lowered and brake pressure is reduced,

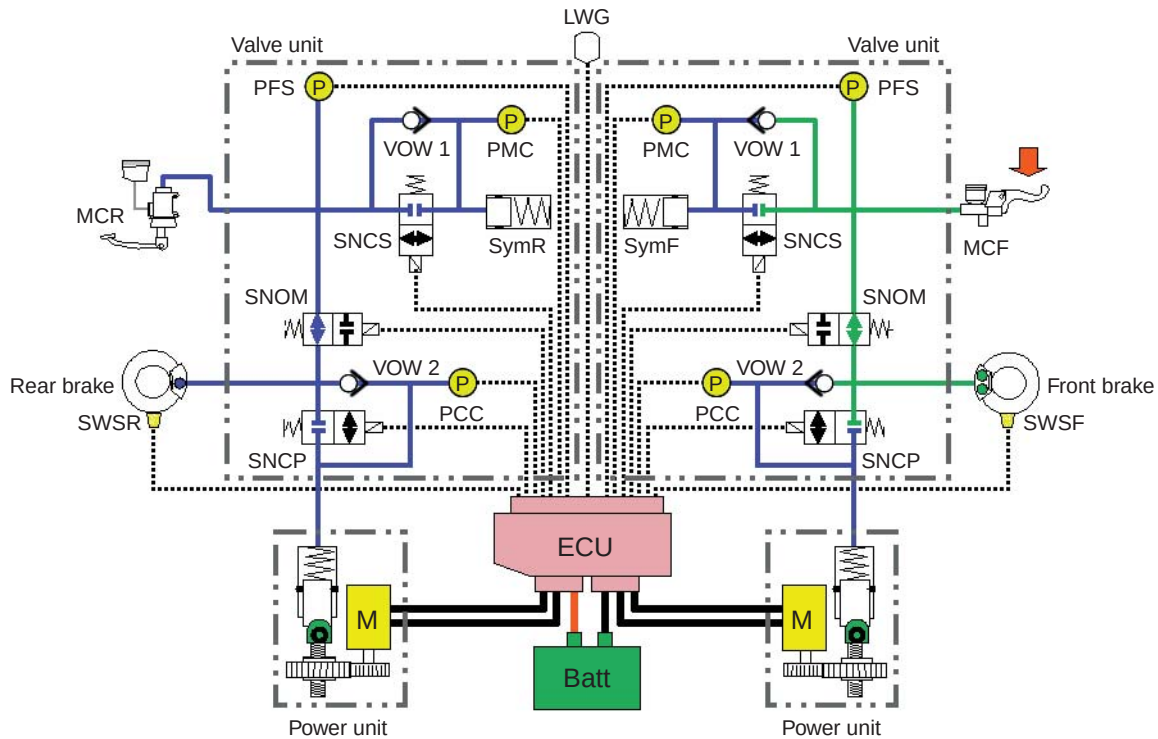


Fig. 6 Diagram of brake-by-wire system (Deactivated)

and the ECU controls brake pressures by ABS to keep an appropriate slip ratio.

5.3. System Failure

When the ignition switch is ON, the ECU constantly conducts self-diagnosis, irrespective of whether the system is activated or deactivated. In the event of a system failure during operation, the ECU will disconnect the power from the solenoid valves and the motors. The internal springs in the solenoid valves will induce them to return to their original positions, enabling normal braking pathways to be maintained.

6. Effect of Use of Brake-by-Wire

6.1. Setting Braking Force Distribution in CBS

Figure 7 shows a conceptual image of CBS control by hand operated combined brake. The figures on the left show front and rear wheel braking force distribution patterns, and the figures on the right show the changes over time in the braking force on the rear wheel from when the brakes are applied, to their release, resulting from these patterns. Figure 7(a) shows a conventional CBS, and Fig. 7(b) shows the brake-by-wire system.

Because in conventional CBS the front and rear wheel braking force distribution patterns are the same for application and release of the brakes, when the hand operated combined brake is released there is a temporary increase in the hydraulic pressure in the rear wheel brakes.

This could result in an instantaneous brake drag on the rear wheel. This peculiar character disturbed sport riding. By contrast, in the brake-by-wire system, the braking force distribution between the front and rear wheels is independent and can be freely set, and therefore there is no increase in rear wheel hydraulic pressure when the hand operated combined brake is released.

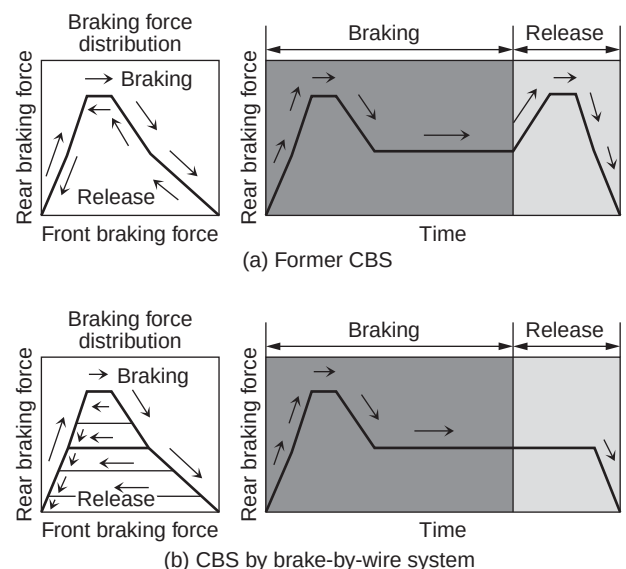


Fig. 7 Braking force distribution pattern of CBS

Table 2 Test conditions

	Brake-by-wire system	Brake operation	Brake control system		
			ABS	Combined brake	Pitching control
Test 1	Deactivated	Conventional brake (hand and foot)	Deactivated	Deactivated	Deactivated
Test 2	Activated	Hand-operated combined brake	Activated	Activated	Deactivated
Test 3	Activated	Hand-operated combined brake	Activated	Activated	Activated

6.2. Pitching Control

Super sport motorcycles can experience pitching which in some circumstances may result in the rear wheel lifting off the ground in the high deceleration range. The operation of ABS at an appropriate timing is able to help prevent this pitching. However, conventional ABS which is responded by deceleration has not been able to be used for this purpose because the slip ratio does not increase in response to the speed of the brake input or the change in the front and rear wheel loads.

The new system controls pitching by predicting the occurrence of pitching from the brake input pressure and the rate of pressure increase, and operating with optimum timing of the ABS⁽³⁾.

The effectiveness of this pitching control was verified in tests conducted using a super sport motorcycle fitted with the system under the test conditions shown in Table 2. Table 3 shows the main specifications of the motorcycle used in the tests. Figure 8 shows an example of results when pitching control was deactivated. There is a 0.8 G deceleration immediately following the application of the brakes, but this is directly followed by a high degree of pitching, and the rider releases the brakes.

Figure 9 shows an example of results when pitching

Table 3 Specifications of test motorcycle

Items		Test motorcycle
Brake system	—	Brake-by-wire
Wheelbase	L (m)	1.372
Curb mass	M (kg)	185.6
Height of C.G.	H (m)	0.681
Distance between C.G. and rear axle	X (m)	0.663
Distributed load of motorcycle with one rider	Front (N)	1 235
	Rear (N)	1 321
Pitching factor	(L-X)/H	1.04
Engine displacement	V (cm ³)	599

control was activated. The ECU predicts pitching from the pressure rise rate of the front wheel brake input, and rapidly increases the front wheel brake caliper pressure. This result in ABS operation of the front brake, and even at its maximum, the pitching is controlled to around 5 deg. Given that the pitching that occurs when conventional brakes are used is around 6 deg, the pitching control in the new system can be judged to be effective.

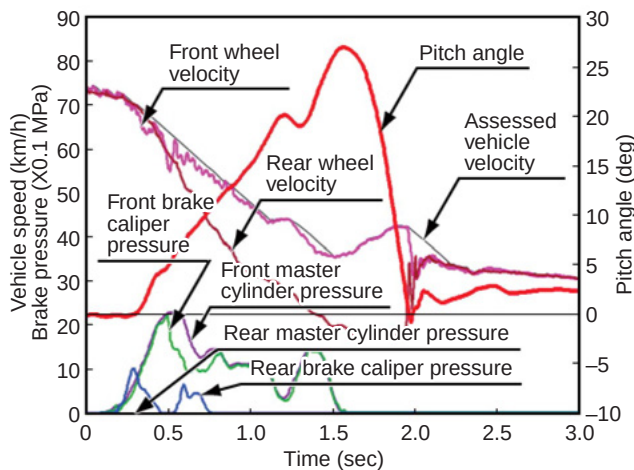


Fig. 8 Example of time history data (Pitching control deactivated)

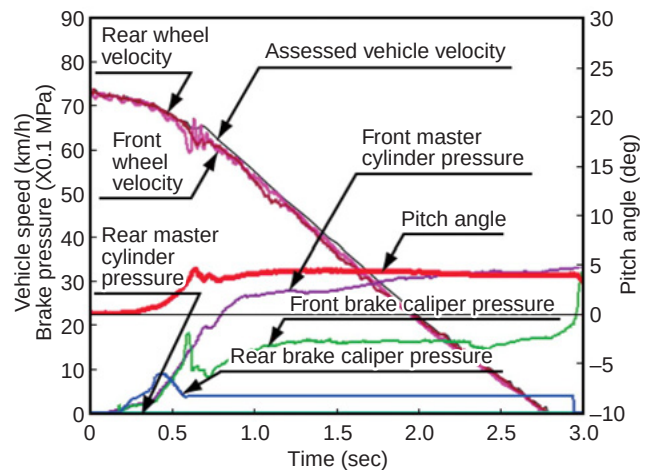


Fig. 9 Example of time history data (Pitching control activated)

7. Conclusion

A brake-by-wire system has been developed for super sport motorcycles. The following results were obtained:

- (1) Pitching control has reconciled a high rate of deceleration with rear wheel lift during ABS operation.
- (2) The new system reconciles sport riding and the use of CBS.
- (3) A brake-by-wire system able to be fitted in a super sport motorcycle has been developed.
- (4) Increase in unsprung mass has been limited to a level that does not affect operation or stability.
- (5) A newly developed stroke simulator has enabled the achievement of an operation feeling similar to that of a normal brake system.
- (6) The brake operation has been guaranteed in the event of power-off and system failure.

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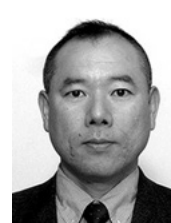
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Development of Small-sized Multilayer Fuel Tank for Off-road Motorcycles and ATVs

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ABSTRACT

A multilayer polyethylene resin fuel tank for off-road motorcycles and All-Terrain Vehicles (ATVs) has been developed to comply with U.S. EPA gasoline permeation control regulations. Motorcycles and ATVs employ complex saddle-shaped tanks with weld-molded insert parts and requirements for exterior design, and the following four issues were therefore identified and addressed for the multilayer blow molding process. To enable the saddle shape to be molded, pre-blowing and depressuring were employed in the blow molding process in order to achieve a uniform wall thickness. A temperature range that balanced resistance to drawdown in the parison, resistance to melting damage in the insert parts, and adhesion of the insert parts enabled the adhesive strength of the insert parts to be increased. Changes to the design of the modified polyethylene adhesive used to bond the EVOH, which forms a gasoline permeation barrier, have increased the resistance of the insert parts to delamination as a result of impact. The employment of a mold surface treatment producing a surface roughness able to conceal mold transfer defects has enabled the external design requirements for the tank to be satisfied.

1. Introduction

Efforts for the reduction of automotive environmental pollution are becoming increasingly important during recent years. Requirements for environmental pollution control in motorcycles, in particular with regard to exhaust gases, are also increasing year by year. Against the background of this trend, North American exhaust emission regulations have been more stringent than those of other countries; also, the U.S. Environmental Protection Agency's (EPA) new regulations concerning gasoline exhaust emissions for 2008 year model motorcycles have been established. To change to a metal fuel tank helps eliminate fuel permeation from the tank surface. However, we set out to meet the fuel permeation regulations using a resin fuel tank, which is superior from the perspectives of lightening, corrosion resistance, shape flexibility, cost, and recovery from impact. Conforming to the regulations using the single-layer polyethylene (PE) resin fuel tanks employed in conventional models represents a challenge. Therefore, 4-materials and 6-layers multilayer continuous blow molding with an ethylene-vinyl alcohol copolymer (EVOH) barrier was selected considering permeability,

durability, and tank function.

Blowouts occur during blow molding of the ductile parts of saddle-shaped, highly ductile exterior fuel tanks. There is a greater decline in the temperature of the resin following extrusion of the parison when using a combination of multilayer continuous blow molding and insert molding, a characteristic method used for motorcycles and ATV, than there is when using conventional PE single-layer accumulative blow molding. This results in a reduction of the welding performance of the insert parts, which can produce fuel leaks. By contrast with single-layer PE, when the joints of the insert-molded parts receive loads during impacts, the multilayered EVOH and high-density polyethylene (HDPE) undergo delamination, producing damage which can result in fuel leaks. In addition, orange peel and die lines on the surface of the tank detract from its external appearance.

In order to address the issues indicated above, a control of the pre-blow conditions and the depressuring and exhaust conditions during blow molding, the temperature behavior of the adhering sections, the design of the adhesive strength of the adhesion layers, and the mold surface treatment were focused on in this

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development. A multilayer continuous blow molding and insert molding manufacturing method, which complied with the gasoline permeation regulations, were sought to be established, and, to satisfy functions of multilayer resin external fuel tank, the development of a high-ductility saddle-shaped fuel tank was targeted⁽¹⁾.

2. Characteristics of Off-road Motorcycle Fuel Tanks

Off-road motorcycles, which have a steel main pipe positioned in the center of the body, need saddle-shaped fuel tanks that straddle the main pipe. The difference between concavities and convexities is greater than the box-shaped fuel tanks employed in automobiles. Figure 1 shows the external appearance of a saddle-shaped motorcycle fuel tank.

Molds are available for the insert parts – the fuel filler cap, fuel cock mount, and metal nuts for mounting the tank shroud and seat. The insert parts are connected to the parison, the matrix of the tank, by adhesion. For the fuel tanks of off-road motorcycles and ATVs, which have a small clearance with the external cover, insert molding is a suitable method from the perspective of space efficiency and to help ensure sufficient volume, and presents advantages over post-welding adhesion. Figure 2 shows the layout for the insert parts of an off-road motorcycle fuel tank.

An impact load on the adhesion surfaces of the insert parts has resulted in delamination of the multilayered materials, a phenomenon that does not occur in a single-layered PE tank. This phenomenon occurs because the strength of the adhesion layer that bonds the HDPE and EVOH layers is low. The stress on the extremities of weld-molded insert parts adhering to the parison is predicted to be higher than for insert parts that are attached following molding.

In an off-road motorcycle fuel tank, the surface of the blow-molded molding is used as the exterior surface. The exterior of the fuel tank is formed by transferring the form of the surface treatment of the mold to the surface of the fuel tank. Before discussing individual technologies below, this section will provide an outline of the flow of blow



Fig. 1 Appearance of saddle-type fuel tank for motorcycle

molding procedures.

- (1) The resin raw material is heated, melted, and kneaded in the screw sections of the extrusion cylinders.
- (2) The resin extruded from the screws flows in a ring shape from the tips of the cylinders to the die head.
- (3) Ring-shaped resin flows in the same way from the cylinders used for each layer towards the die head. The ring-shaped resin materials sink downwards in tubular shapes. The resin materials adhere closely, forming a laminated tube. This high-temperature resin tube is the parison.
- (4) The parison is gradually formed lower than the die head by extrusion of resin from each of the cylinders.
- (5) The lower section of the parison is pinched to form a bag shape.
- (6) The depressuring and exhaust of the bag-shaped parison are conducted by means of the air channels in the die head.
- (7) The parison is set in the mold.
- (8) The blow pin in the mold is inserted into the parison.
- (9) Air is blown in via the inserted blow pin, and the parison is formed by pressing to the mold.
- (10) The mold is opened and the molded product is ejected⁽²⁾.

This flow is illustrated in Fig. 3.

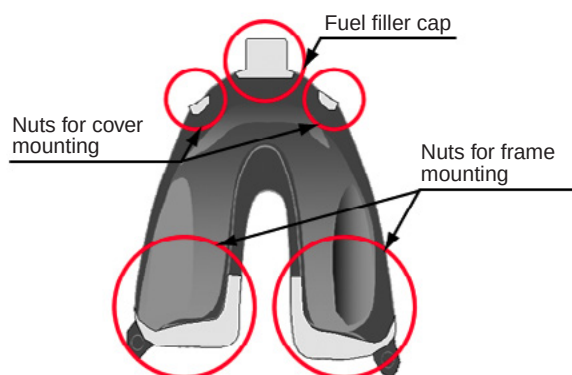


Fig. 2 Insert parts layout of fuel tank for off-road motorcycle

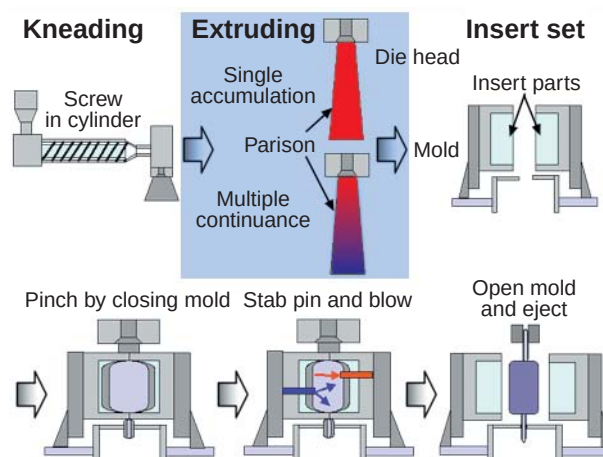


Fig. 3 Blowing process

3. Issues of Multilayer Continuous Blow-molded Motorcycle Tanks

- (1) Compared to a fuel tank for a four-wheel vehicle, highly ductile molding is needed for a saddle-shaped motorcycle fuel tank, and blowouts occur during blow molding of the ductile sections.
- (2) The decline in the temperature of the resin following extrusion of the parison is greater in the case of multilayer continuous blow molding than in conventional single-layer accumulative blow method using PE. Because of this, a decline in the adhesion performance of the insert parts may occur when using a combination of multilayer continuous blow molding and the insert molding that is used specifically for motorcycles and ATVs, and this may lead to fuel leaks.
- (3) By contrast with single-layer PE, when the joints of the insert-molded parts receive an impact load, delamination of the EVOH and HDPE that adhere in the multilayer structure occurs, producing damage, which may result in fuel leaks.
- (4) The fuel tank of a motorcycle is an external part, and it is therefore necessary to take measures to enhance the external appearance of the tank in response to orange peel and die lines.

It was necessary to develop technologies to address these characteristic motorcycle fuel tank issues.

4. Development of a High-ductility Saddle-shaped Fuel Tank

The fuel tank for off-road motorcycles should be formed into a saddle shape with large concavities and convexities. Figure 4 compares the forming ductility of a saddle-shape motorcycle tank with that of a four-wheel vehicle tank.

4.1. Pre-blow Molding

During blow molding, if the parison ejected from the

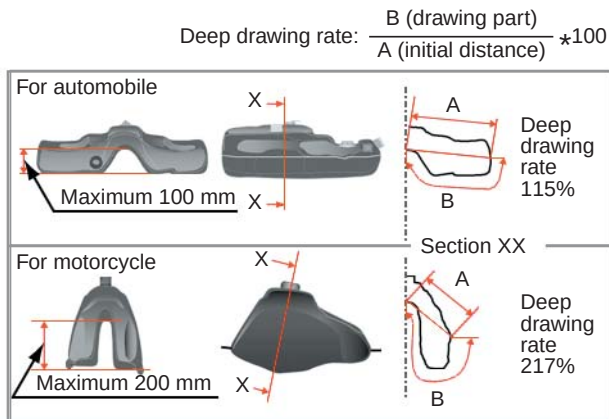


Fig. 4 Comparison of forming ductility with automobile fuel tank

molding machine die core is directly blown, the parison easily makes contact with the mold. The parison adheres to the mold, and the parison between the contact points are stretched and melted. The stretched sections between the contact points are reduced in thickness. Then, in the case of saddle-shaped tanks with large concavities and convexities, blowouts occur. Performing pre-blowing before blow molding to form the necessary tank perimeter in advance, to help ensure uniform thickness after blow molding, is an important molding method. If this method is not used, it is not possible to mold a high-ductility saddle-shape tank. Figure 5 shows the behavior of the parison during pre-blowing.

If the pre-blowing time is too short, blowouts will occur from the thin sections of the parison during blowing; if the pre-blowing time is too long, blowouts will occur during pre-blowing. The setting of pre-blowing conditions is important from the perspective of achieving a uniform wall thickness.

For the formation of the saddle-shape fuel tank discussed in this paper, setting a pre-blowing time of 0.4 sec resulted in a more uniform wall thickness. Figure 6 shows the relationship between the pre-blowing time and the minimum wall thickness of the product.

4.2. Depressuring and Exhaust Molding

The depressuring and exhaust process is even more important than pre-blowing in terms of achieving a uniform wall thickness. If the depressuring and exhaust process is not conducted, after mold closure, the pre-blown parison will contact the surface of the mold, in addition to which the mold will bite the parison in areas other than the predetermined areas, which can result in thin walls in the molding. Depressurizing the pre-blown parison produces sagging in the radial direction, enabling the outermost external radius of the parison to be compressed without change, relative to the actual perimeter of the parison, helping to prevent unwanted biting of the parison by the mold and enabling maximal

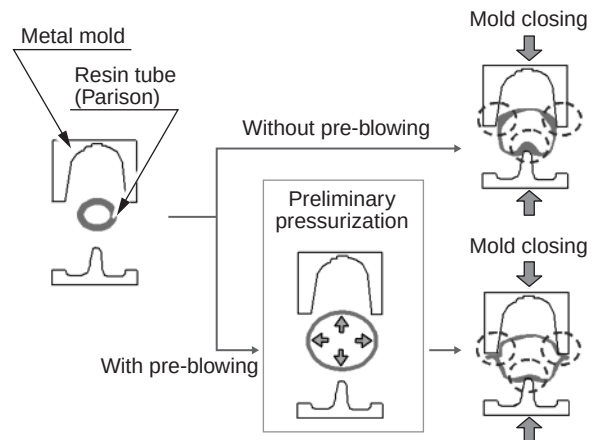


Fig. 5 Behavior of parison during pre-blowing

use of its perimeter. When depressuring and exhaust continues, adherence occurs in the interior of the parison, and the parison cannot be blow molded.

In the case of the saddle-shape fuel tank discussed in

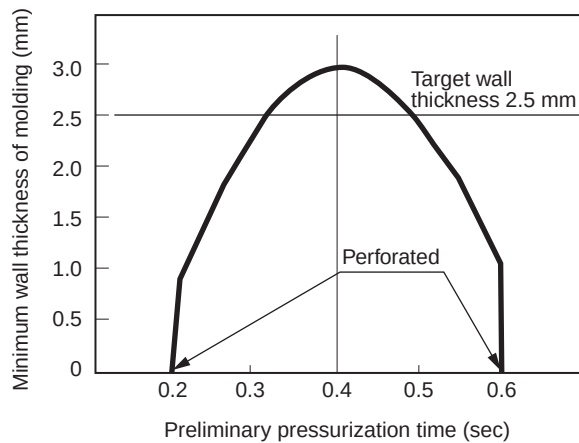


Fig. 6 Relationship between preliminary pressurization time and minimum wall thickness of molding

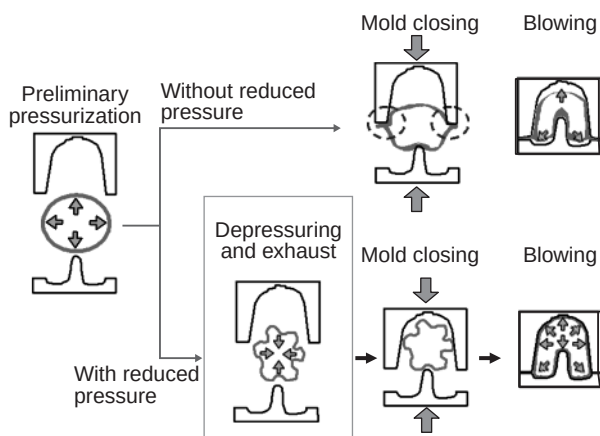


Fig. 7 Behavior of parison during depressuring and exhaust

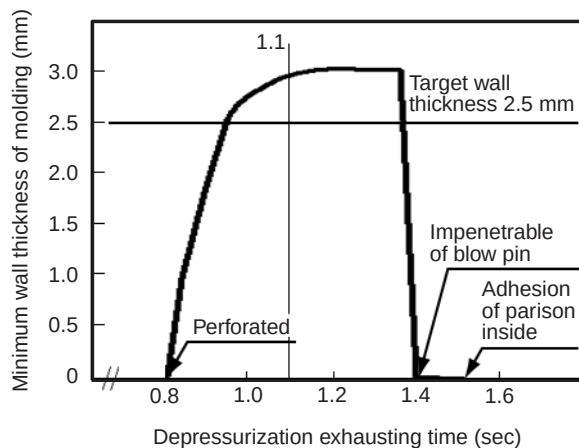


Fig. 8 Relation between depressurization exhaust and minimum thickness of molding

this paper, a depressuring and exhaust time of 1.1 sec was set, balancing contact with the mold, avoidance of unnecessary biting, and adhesion resistance in the interior of the parison, enabling the achievement of a more uniform wall thickness. Figure 7 shows the behavior of the parison during the depressuring and exhaust process. Figure 8 shows the relationship between depressuring and exhaust time and the minimum wall thickness of the product.

5. Achievement of Insert Parts Adhesion Performance

5.1. Parison Temperature

The insert parts adhere to the parison, which forms the body of the fuel tank. The adhesion performance of the insert parts is important, because fuel leaks can result in particular from adhesion imperfections in the fuel filler cap and fuel cock mount. Conventional single-layer PE tanks are molded by accumulative blowing. In this method, resin is accumulated in the molding machine. Immediately prior to molding, the parison is extruded and blow-molded in a mold in which the resin insert parts, having been preheated in an oven, have been set, enabling the insert parts and the parison to adhere. The time that elapses following extrusion of the parison is short, enabling the achievement of good adhesion with the insert parts.

In the multilayer continuous blow molding method, by contrast, no resin is accumulated in the molding machine and the parison is continuously extruded in order to avoid heat degradation of the EVOH. Because of this, the resin temperature is lower at the initially extruded parison tip. This is an unfavorable condition for adhesion molding of the insert parts. At the same temperature setting as used for conventional single-layer PE molding, as the temperature is insufficient when the parison and insert parts were connected, peeling would result in gasoline leaks. Increasing the parison temperature by 30 or 40 °C against the setting used in single-layer accumulative PE molding, in order to enhance adhesion performance, resulted in the phenomenon of drawdown, in which the parison breaks under its own weight and falls to the bottom. Simply setting a higher resin temperature would result in issues such as heat degradation of the parison resin and breaking of the parison due to drawdown. This meant that there was a limit to the ability to increase the resin temperature, and that a balance between temperature and adhesion would have to be sought. Increasing the temperature by 20 °C against the conventional temperature setting in an attempt to enhance adhesion performance enabled the achievement of a balance between drawdown resistance and adhesion. Figure 9 shows the temperature transition following extrusion of the parison.

5.2. Temperature of Insert Parts

Section 5.1. focused on the temperature of the parison, but it was determined that the temperature of the parison

was not the only factor in achieving adhesion of the insert parts. It was judged that it would also be necessary to consider the temperature of the insert parts. The temperature setting for the insert parts was therefore reexamined. In advance, the insert parts are preheated in a heating oven and set in the mold before blow molding, and then blow molding is conducted. The preheating temperature for the insert parts was set 30 or 40 °C higher than the temperature for conventional single-layer PE accumulative molding, and molding was attempted. This resulted in melting damage to the insert parts in the oven and deformation of the insert parts during handling following removal from the oven. The temperature of the insert parts decreases following removal from the heating oven. When the insert parts have been set in the mold, there is an even more noticeable decline in the temperature of the parts due to heat radiation into the mold. For the insert parts also, the preheating temperature setting was increased by 20 °C against the conventional temperature in order to enhance adhesion performance.

In addition, in order to enhance safety and productivity, a robot arm has been employed to set the insert parts in the mold rather than setting them manually. This increased the time required to remove the parts from

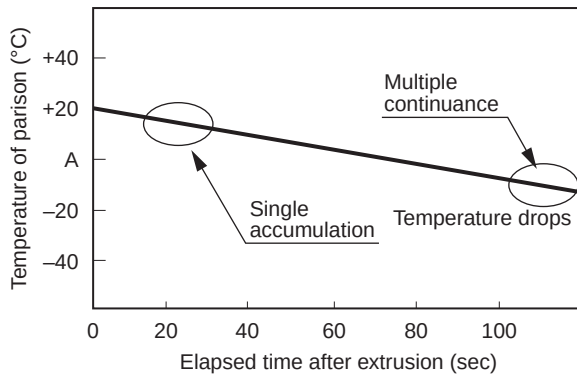


Fig. 9 Temperature transition after parison extrusion

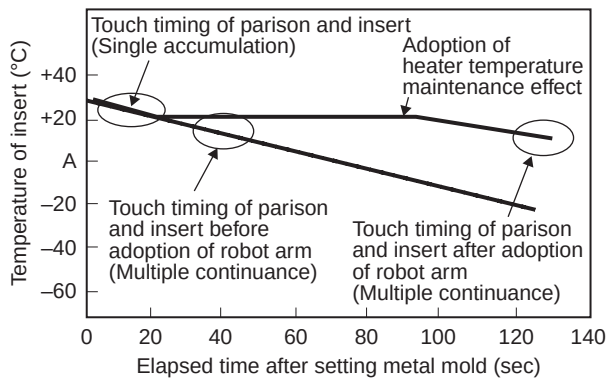


Fig. 10 Temperature transition of insert parts with thermal heater temperature maintenance effect after removal of insert parts from heating oven

the oven and set them in the mold, resulting in a decline in their temperature. In order to avoid this decline in temperature during the standby time before the robot arm grips the insert parts, a thermal heater has been added to provide concurrent heating, enabling the temperature of the adhesion surfaces to be maintained. Figure 10 shows the temperature transition of the insert parts as a result of employment of the thermal heater.

6. Design of Adhesive for Bonding Multiple Resin Layers

Achieving adhesion with PE represents a challenge, and a modified polyethylene adhesive is therefore used between EVOH and PE when bonding the two substances. The adhesive strength of the adhesive affects the strength of the fuel tank under impacts at low temperatures, and the form of damage it sustains under these impacts. A tank that contains insert parts sustains damage due to peeling of the adhesive layer originating at the lateral ends of the insert parts. This type of damage is specific to insert-molded tanks, and does not occur in the case of tanks in which the insert parts have been attached following molding. It is one aspect of insert-molded tanks that need to be considered in order to increase impact strength. Adhesives are subjected to maleic anhydride modification in order to enhance adhesion performance. Maleic

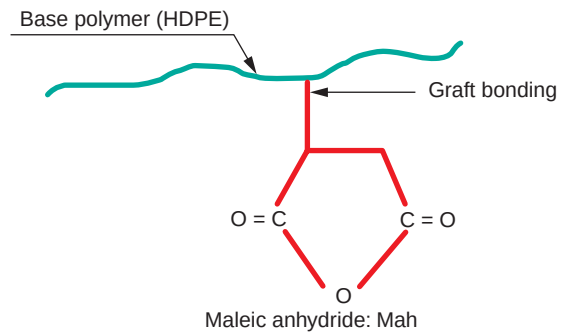


Fig. 11 Bonding structure of olefins and maleic anhydride

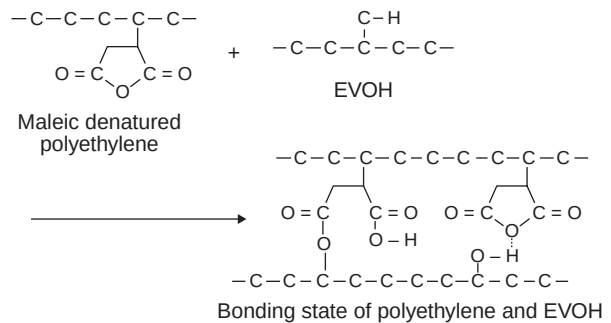


Fig. 12 Reaction mechanism of maleic anhydride and EVOH

anhydride contains a vinyl radical, and therefore reacts readily with polyolefins. Figure 11 shows the bonding structure of olefins and maleic anhydride, and Fig. 12 shows the reaction mechanism of maleic anhydride and EVOH.

The amount of maleic anhydride in the adhesive is the main factor affecting adhesive strength. Increasing the amount of maleic anhydride in the adhesive increases its adhesive strength. However, maleic anhydride is a costly substance, and it is therefore necessary to achieve a balance between adhesion performance and cost with regard to the amount of maleic anhydride used. Low-temperature impact tests were conducted on molded fuel tanks using adhesives containing 0.05, 0.1, and 0.15% maleic anhydride, as used to bond insert parts in post-molding welding. Following the tests, the tanks showed damage occurring as a result of peeling of the adhesion layer originating at the ends of the insert parts. No peeling of the adhesion layer was observed in low-temperature impact tests when adhesives containing more than 0.3% maleic anhydride were used. The percentage of maleic anhydride, an index of the amount of maleic anhydride in the adhesive, was set at 0.35, based on considerations of safety. This enabled stable adhesive strength to be

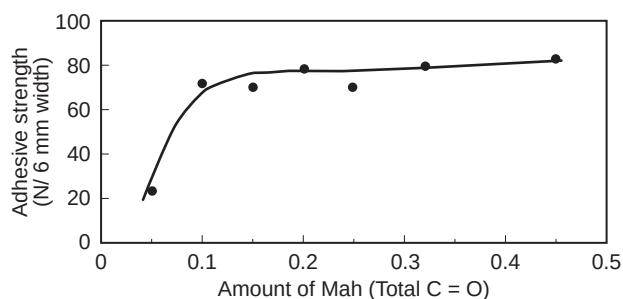


Fig. 13 Relationship between Adhesive strength and amount of maleic anhydride

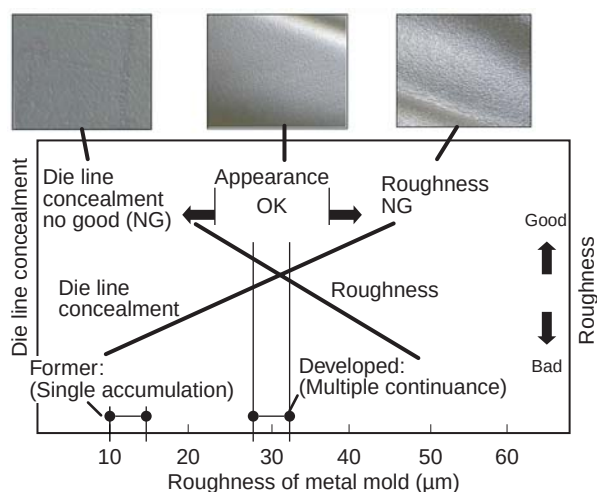


Fig. 14 Relationship between roughness and roughness of metal mold for finishing

achieved at minimum cost. Figure 13 shows the relationship between adhesive strength and the amount of maleic anhydride in the adhesive^{(3), (4)}.

7. Mold Surface Treatment for Exterior Design of Fuel Tank

Contrary to the internal fuel tanks used in four-wheel vehicles, resin fuel tanks for off-road motorcycles are colored and used as an aspect of the exterior design of the vehicle. The temperature of the resin forming the parison is lower in multilayer continuous blow molding than in PE single-layer accumulative blow molding, and mold transfer is therefore inferior when using the former method. This represents a disadvantage in terms of coloring the resin at the exterior of the tank. Shot blasting is conventionally used as a mold surface treatment, producing minute concavities and convexities of 10-20 μm . The lower parison resin temperature in multilayer continuous blow molding means that the undulations of the die lines on the surface of the parison cannot be filled by the shot-blasted concavities and convexities, and a style design therefore cannot be formed. In order to conceal the concavities and convexities on the surface of the parison, a chemical treatment was applied to produce a grained surface of a sufficient depth (30 μm) and form, enabling the exterior design requirements to be satisfied. Figure 14 shows the relationship between the external appearance requirements of the tank and the depth of the mold surface treatment.

8. Effect of Employment of Developed Fuel Tank

The developed fuel tank has been employed in the 2008 model year CRF-F and CRF-X Series off-road motorcycles and all Honda 2008 model year ATVs. This has contributed to the protection of the environment by reducing the emission of gasoline to the atmosphere substantially. In addition, the development of a manufacturing method combining multilayer continuous blow molding and insert molding has contributed to the achievement of a high level of space efficiency without modification of the layout of the covers and seat base plates employed around the fuel tanks of off-road motorcycles and ATVs. Honda intends to expand its use of the new tanks, fitting them to models falling under the purview of the U.S. EPA gasoline permeation control regulations from 2008 year models.

9. Conclusion

A fuel tank for off-road motorcycles and ATVs manufactured using a combination of multilayer continuous blow molding and insert molding has been developed. The salient points of this development project are as follows:

- (1) The setting of a pre-blow time of 0.4 sec and a depressuring and exhaust time of 1.1 sec for the molding of a high-ductility saddle-shape fuel tank of the shape discussed in this paper enabled molding with a highly uniform wall thickness.
- (2) With regard to the adhesion performance of the continuous blow-molded parts and weld-molded insert parts, raising the pre-heating temperature of the insert parts and the parison temperature by 20 °C each and employing a thermal heater for concurrent heating of the insert parts before they are set in the mold by a robot arm enabled the achievement of a balance between adhesion performance, resistance to melting damage in the insert parts, and resistance to drawdown in the parison.
- (3) The use of a maleic anhydride content of 0.35% in the adhesive employed to bond the EVOH and HDPE layers in the multilayer parison enabled the achievement of the necessary adhesive strength for impact resistance in the blow-molded and insert-molded fuel tank.
- (4) The application of a chemical surface graining treatment to the mold and a grain depth setting of 30 μm enabled the achievement of a good external appearance for an off-road motorcycle fuel tank.

10. Acknowledgments

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Rearward Visibility Simulation System of Motorcycle Rearview Mirrors

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ABSTRACT

Visibility provided by the rearview mirrors of motorcycles is regulated by law, and should be measured. Thus far, compliance with legal standards has been checked first by investigation using 2-dimensional CAD, and then manufacturing a prototype motorcycle. However, an issue unique to motorcycles is displacement of the rearview mirror when the rider rides the motorcycle, which affects rearward visibility. This factor hindered sufficient investigation in two dimensions, which delayed determination of the design specifications. Therefore, a simulation system has been developed that uses 3-dimensional CAD, which incorporates a function that automatically sets the legal prerequisites using a combination of the Newton-Raphson method and bisection method. This enabled assessment in the design layout stage, and contributed to early layout maturation.

1. Introduction

Countries around the world have prescribed standards governing the reflection of the rider and the rearward visibility of rearview mirrors in order to enhance motorcycle safety. Thus far, compliance with legal standards has been judged first by checking with 2-dimensional CAD, and then manufacturing a prototype motorcycle, which has delayed assessment of legal compliance. It is challenging to investigate factors such as displacement of the rearview mirror when the rider rides the motorcycle by using 2-dimensional CAD. Therefore, a system has been developed to conduct more realistic simulations by using 3-dimensional CAD data. This system enables accurate assessment of compliance with legal standards in a short time based on the specifications data of the rearview mirror and the rider.

This paper presents an overview of the rearward visibility simulation system for motorcycle rearview mirrors, and reports on rearview mirror surface control.

2. Laws and Regulations for Motorcycle Rearview Mirror

Table 1 lists typical examples of laws and regulations governing rearward visibility for motorcycles. To assess legal compliance, it is necessary to actualize the given setting conditions. Therefore, a function that sets these conditions in 3-dimensional CAD is important to realize a simulation system.

3. Features of Rearward Visibility of Motorcycle Rearview Mirrors

Unlike cars, displacement of the rearview mirrors resulting from tire deformation when the rider rides the motorcycle has a large effect on rearward visibility. In addition, more realistic simulations can be performed by adjusting the rearview mirror position. Therefore, when conducting simulations it is necessary to first obtain the relative positions of the rearview mirrors and the rider when the rider rides the motorcycle (Fig. 1).

4. System Overview

4.1. Flowchart

Figure 2 shows the simulation system flowchart. First, the rider position, rearview mirror position and other data are input. Next, the rider position when riding the motorcycle is obtained from the 3-dimensional layout data and specifications data, and after that the simulation starts. Note that the “JASO T009-94⁽¹⁾ Measurement Method of Field of Rear-view Mirror for Motorcycles” is used as the method for measuring the actual motorcycle. The simulation adjusts the rearview mirror position by positioning the mirror surface in accordance with this measurement method.

4.2. Geometrical Background

The system obtains the point (Reflection Point) on the mirror surface at which the object (Object Point) appears

* Motorcycle R&D Center

Table 1 Laws and regulations for motorcycle rearview mirror

Regulations	Setup	Points to be checked
Rearward visibility of rearview mirror in Japan	Align center point of mirror surface with point, reflecting on mirror surface, 50 m behind.	Non-visible range equal to or less than 25%
Verification of regulation conformity in Belgium / Netherlands	Align corner point of maximum rectangle which is inscribed around edge of mirror surface, with point on ground line, reflecting on mirror surface, 10 m behind reference eye point.	Reflection on 10 m line equal to or more than 2.5 m.
Verification of regulation conformity in Germany	Align lower corner point (vehicle side) of maximum rectangle which is inscribed around edge of mirror surface, with point on ground line, reflecting on mirror surface, 20 m behind reference rear view mirror point.	Reflection on 20 m line equal to or more than 2.5 m.
Verification of regulation conformity in EEC	Align lower corner point (vehicle side) of maximum rectangle which is inscribed around edge of mirror surface, with point on ground line, reflecting on mirror surface, 10 m behind reference rear view mirror point.	Reflection on 10 m line equal to or more than 2.5 m.

as viewed from the position of the motorcycle rider's eye (Eye Point). The reflection point to the mirror surface is calculated geometrically by varying the center point of the sphere (Center Point) while fixing the connection position (Joint Point) of the rearview mirror. Figure 3 shows an image of the mirror surface. The reflection point can be obtained using the following three conditions.

- (1) The reflection point is located on the plane that passes through the position of the eye (Eye Point), the position of the object (Object Point), and the center point of the mirror surface point.
- (2) The reflection point is located on the sphere with radius r , centering on the mirror surface central point.

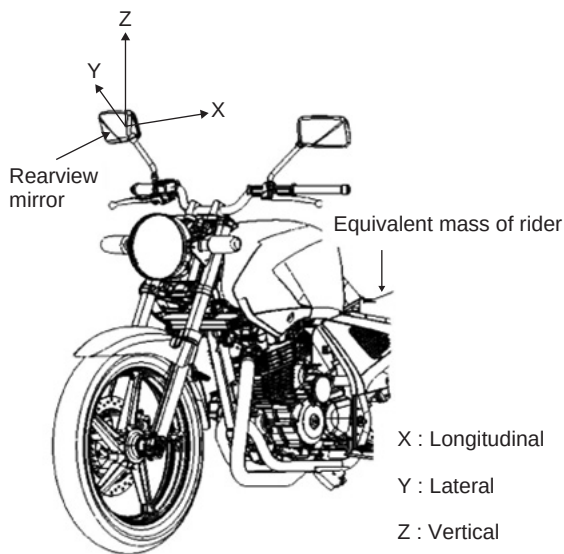


Fig. 1 Positional relationship of rearview mirror and rider when rider rides

- (3) The angle formed by the vector from the reflection point to the eye point and the vector from the reflection point to the object point is bisected equally by the vector from the mirror surface central point to the reflection point.

4.3. Default Value Settings

The parameters were set as follows.

$R(x, y, z)$: The reflection point (Reflection Point)

$E(E_x, E_y, E_z)$: The position of the eye point (Eye Point)

$O(O_x, O_y, O_z)$: The position of the object (Object Point)

$C(C_x, C_y, C_z)$: The center point of the sphere (Center Point)

$M(M_x, M_y, M_z)$: The mirror surface center point (Mirror Point)

$J(J_x, J_y, J_z)$: The connection position with the mirror (Joint Point)

$G(G_x, G_y, G_z)$: The default estimated value (Guess Point)

r : The mirror radius (Mirror Radius)

α, β : Reference points for obtaining the guess point

$f(x, y, z)$: Discriminant for judging presence on the plane CEO

a, b, c, d : Coefficients of $f(x, y, z)$

$g(x, y, z)$: Discriminant for judging presence on a plane

A : Amount of Guess point movement

Figure 4 shows the coordinate illustration of the default value. When the default value is inappropriate, for example when an inflection point is present in the interval to be searched for a solution, either the calculation does not converge on a nearby solution or the calculation

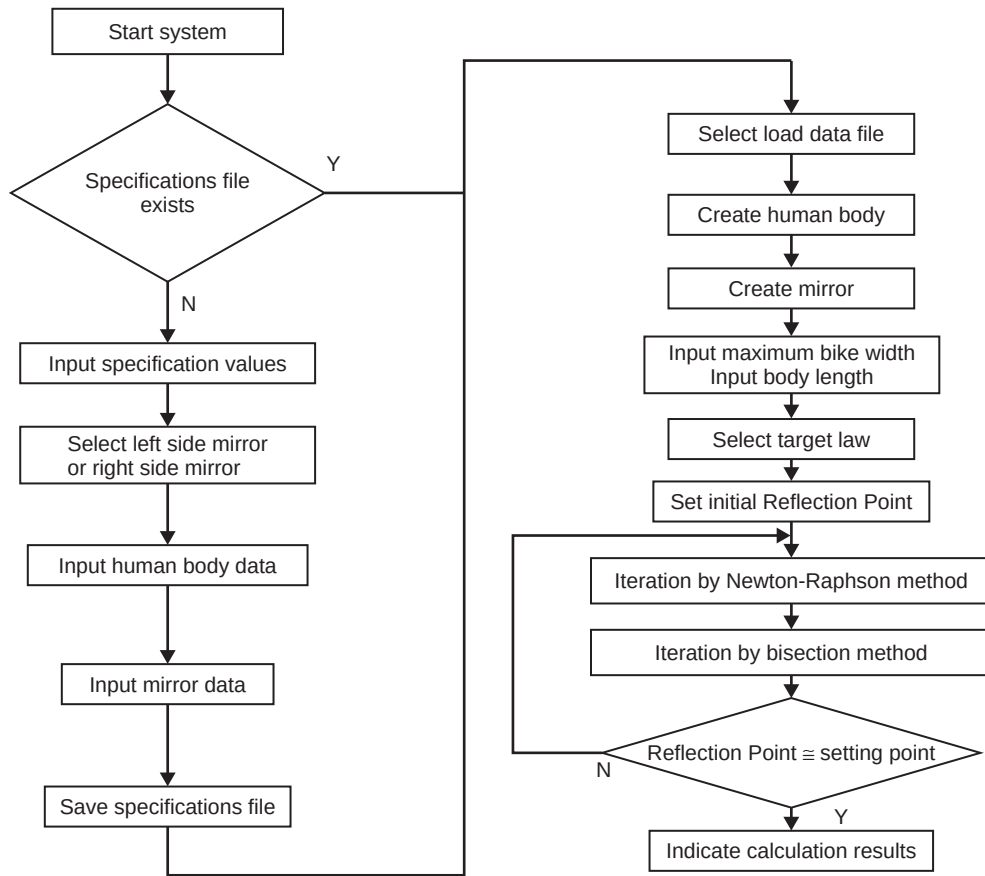


Fig. 2 Simulation flowchart

converges but the convergence speed is slow. Therefore, the default value should be a value thought to be close to the solution. Here, the midpoint between the two points α and β on the mirror surface shown in Fig. 4 is set as the default value.

Therefore, the vector of the default value from the mirror surface central point is as follows.

$$\vec{G} - \vec{C} = \frac{\vec{\alpha} + \vec{\beta}}{2} \quad (1)$$

The four vectors $\vec{G}$, $\vec{C}$, $\vec{\alpha}$ and $\vec{\beta}$ represent the vectors from the origin to the respective points.

That is to say, the guess point of the default value can be expressed as follows.

$$G(Gx, Gy, Gz) = \frac{1}{2} \left\{ \left(\frac{\vec{CE}}{|\vec{CE}|} + \frac{\vec{CO}}{|\vec{CO}|} \right) \cdot r + 2\vec{C} \right\} \quad (2)$$

4.4. Calculation Algorithms

The calculation is performed by the following procedure.

- (1) The reflection point on the mirror surface is obtained by iterative calculations starting from the default value.

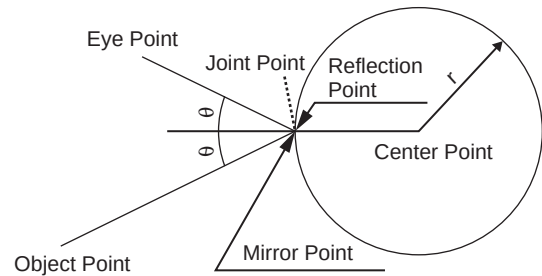


Fig. 3 Mirror surface image

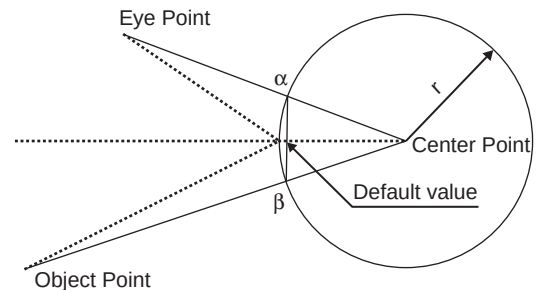


Fig. 4 Image of default coordinate value

- (2) The rearview mirror is positioned so that the obtained reflection point matches the legally stipulated position. At this time, as “Center Point” also moves, some deviations appear in the reflection point.
 - (3) A new default value is given.
- Steps (1) to (3) above are repeated until the values converge.

4.4.1. Obtaining the mirror surface reflection point

The Newton-Raphson method, which has a fast convergence speed, was expanded to three dimensions and used to obtain the reflection point⁽²⁾. Figure 5 shows the algorithm flow for obtaining the reflection point. The algorithm exits the loop when each component of A in this flow infinitely approaches zero, or after 100 loops in consideration of cases when the solution does not converge.

When the function Eq.(3) obtained from each component and intercept of each vector has zero, the “Reflection Point” is considered to be on the plane that passes through C, E and O.

$$f(x, y, z) = ax + by + cz + d \quad (3)$$

When the function Eq.(4) obtained from the equation of sphere has zero, the reflection point passes through C and is considered to be on the sphere with radius r .

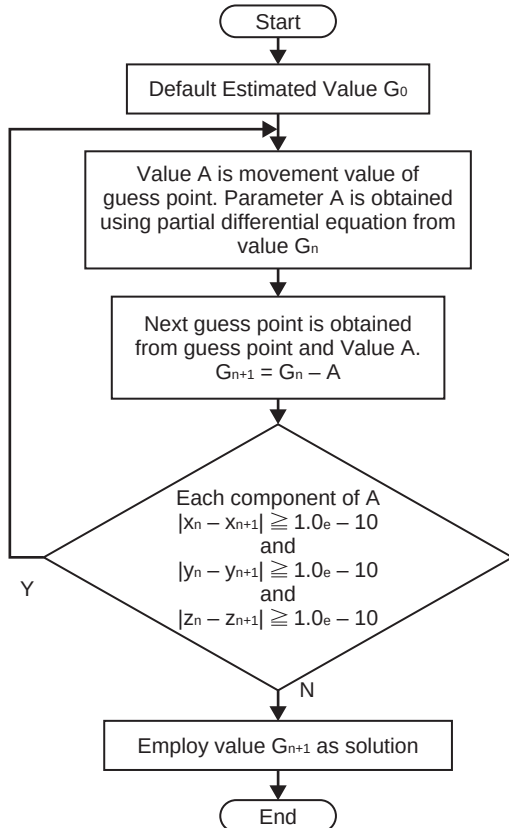


Fig. 5 Newton-Raphson method algorithm flow

$$g(x, y, z) = (x - Cx)^2 - (y - Cy)^2 - (z - Cz)^2 - r^2 \quad (4)$$

According to the conditions of angle bisection described in 4.2. (3) above, when the angle formed by $\overline{CR}$ and $\overline{RE}$ and the angle formed by $\overline{CR}$ and $\overline{RO}$ are equal, the light coming from O is reflected by R and reaches E.

The following equation uses Eq. (3) with not just one variable, but expanded into a three-dimensional simultaneous nonlinear equation.

$$\begin{bmatrix} x_n - x_{n+1} \\ y_n - y_{n+1} \\ z_n - z_{n+1} \end{bmatrix} = \begin{bmatrix} f(x_n, y_n, z_n) \\ g(x_n, y_n, z_n) \\ q(x_n, y_n, z_n) \end{bmatrix} \begin{bmatrix} \frac{\partial f}{\partial x_n} & \frac{\partial f}{\partial y_n} & \frac{\partial f}{\partial z_n} \\ \frac{\partial g}{\partial x_n} & \frac{\partial g}{\partial y_n} & \frac{\partial g}{\partial z_n} \\ \frac{\partial q}{\partial x_n} & \frac{\partial q}{\partial y_n} & \frac{\partial q}{\partial z_n} \end{bmatrix}^{-1} \quad (5)$$

Here, when the values of $|x_n - x_{n+1}|$, $|y_n - y_{n+1}|$ and $|z_n - z_{n+1}|$ each infinitely approach zero, the solution to the simultaneous equation is considered to have converged.

4.4.2. Positioning the rearview mirror

Once the mirror surface reflection point has been obtained, the rearview mirror is positioned so that the point is at the set position prescribed by law. When positioning the rearview mirror, the Newton-Raphson method obtains the solution at relatively high speed, but the solution may diverge. Therefore, the bisection method was used from the standpoint that the solution always converges as long as the calculation interval is determined. The outline of the bisection method and the application of this method to calculate the reflection on the mirror surface are described below.

(1) Bisection method

The bisection method theorem (calculation method)⁽²⁾ used to position the rearview mirror is described below.

The actual numerical calculation starts with the two points a and b ($a < b$) such that $f(a)f(b) < 0$ as default values, and the calculation $f(c)$ is performed on the point $c = (a + b)/2$ that bisects the interval $[a, b]$. When $f(c)f(a) < 0$, b is replaced with c . When $f(c)f(a) > 0$, a is replaced with c . This operation is repeated so that the solution is always within the interval $[a, b]$, and the calculation ends when the interval width $|b - a|$ becomes less than the determination value ε that controls the solution accuracy. Figure 6 shows the bisection method algorithm flow.

(2) Application of the bisection method to calculation of mirror surface reflection

Figure 7 shows an image of applying the bisection method to mirror surface reflection. The vector that joins “Center Point” and “Reflection Point 1” is labeled $\overline{CR_1}$, the vector that joins “Center Point” and “Reflection Point

2" is labeled $\overrightarrow{CR_2}$, and the vector that joins "Center Point" and "Target Point" is labeled $\overrightarrow{CT}$. The angle formed by $\overrightarrow{CT}$ and $\overrightarrow{CR_1}$ is labeled θ_1 (- direction), and the angle formed by $\overrightarrow{CT}$ and $\overrightarrow{CR_2}$ is labeled θ_2 (+ direction). These θ_1 and θ_2 are substituted to a and b in the bisection method described in 4.4.2. (1) above.

(3) Adjusting the reflection standard position on the mirror surface and the line of sight by rearview mirror automatic adjustment

Figure 8 shows the algorithm flow to actually obtain the line of sight and rearview mirror adjustment angle using the bisection method. As shown in Fig. 8, there are

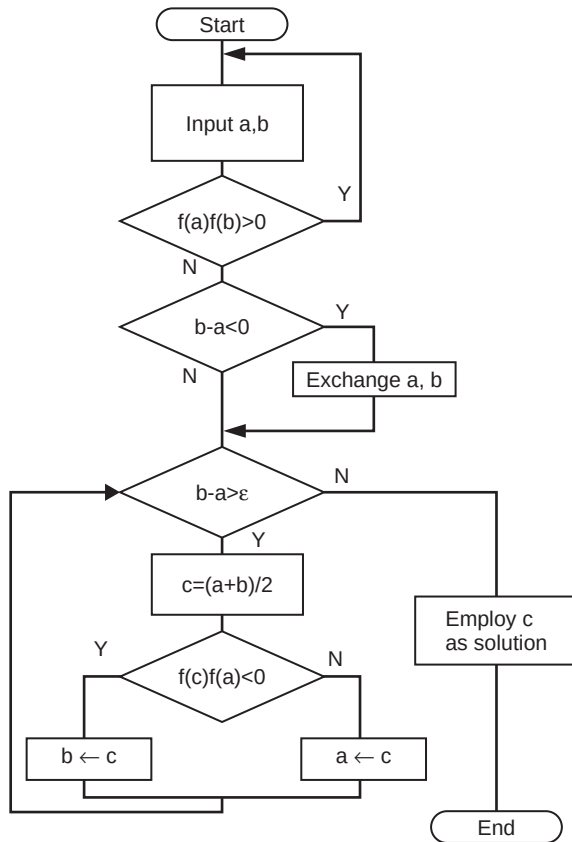


Fig. 6 Bisection method algorithm flow

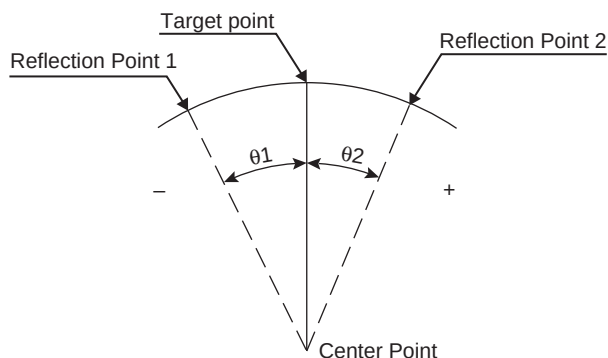


Fig. 7 Image of bisection method in mirror reflection

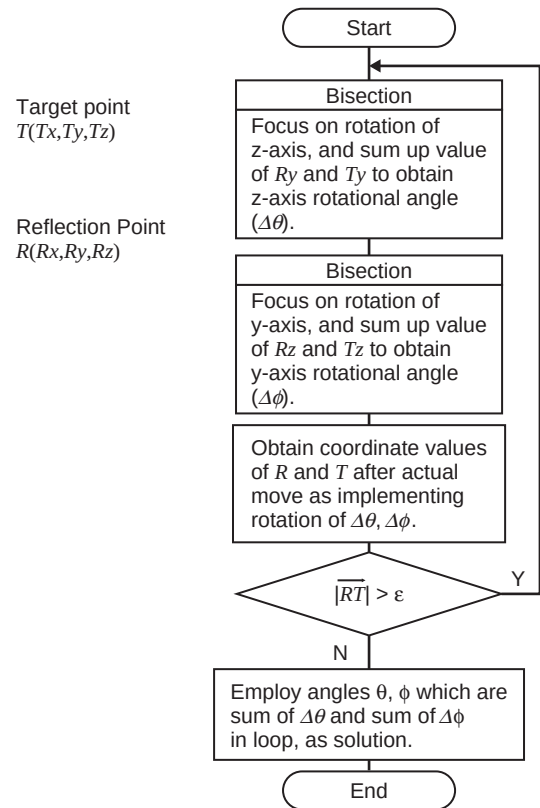


Fig. 8 Mirror automatic adjustment flow

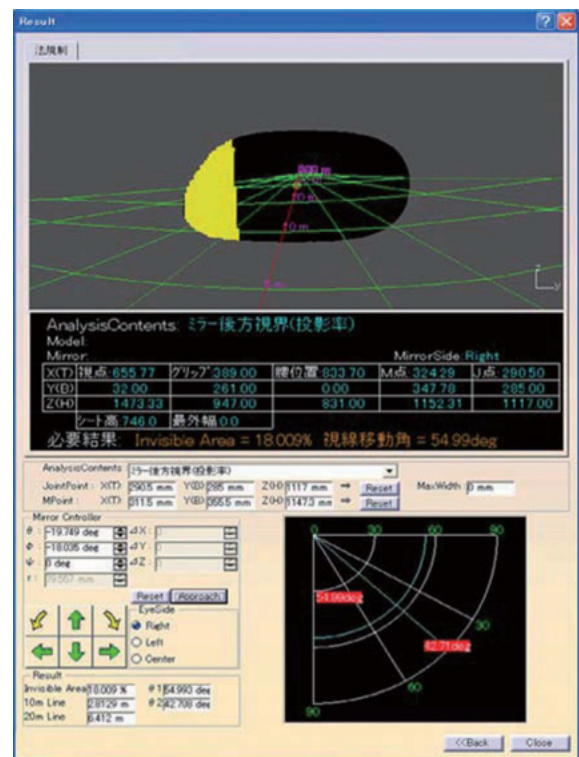


Fig. 9 Graphical user interface (GUI) of simulation system

an infinite number of patterns for aligning the standard position on the mirror surface with the line of sight, so calculation is performed without taking into account the X axis.

4.5. Graphical User Interface (GUI)

A GUI was created to simplify the applicable legal settings and the results display of object reflection to the rearview mirror (Fig. 9). This made it easy to reselect the applicable laws and regulations, and enabled both visual and data display of the results simply by pressing the automatic adjustment button.

5. Results of Positioning Rearview Mirror in the Simulation System

The results of positioning the rearview mirror in order to check the legal compliance of the rearward visibility provided by a rearview mirror are introduced here as an

example of application. Figure 10 shows the simulated conditions based only on specifications, before making adjustment in order to comply with the regulations. Figure 11 shows the results after automatically adjusting the position to check legal compliance. Note that the yellow portion on the mirror surface represents the reflection of the rider's arm.

The above-mentioned results illustrate that the simulation system enables the rearview mirror positioning to comply with the setting specified in the regulation to align the mirror center point with the point 50 m to the rear. This helps eliminate the need for manual positioning, and the results can be obtained in a short time, enabling simulations to be performed as many times as necessary in order to check legal compliance.

The rearview mirror position can also be adjusted automatically in the same manner to comply with other legal standards.

6. Conclusion

In order to check the legal compliance of motorcycle rearview mirrors, a simulation system that uses 3-dimensional CAD, which has a function that automatically sets the legal prerequisites using the Newton-Raphson method and the bisection method, has been developed.

- (1) The ability to perform simulations using 3-dimensional CAD enables checking of legal compliance without creating a prototype motorcycle.
- (2) Simulations can be performed for various legal standards, enabling designers to perform simulations using the latest information.
- (3) This contributed to early maturation of the 3-dimensional layout in the design stage.

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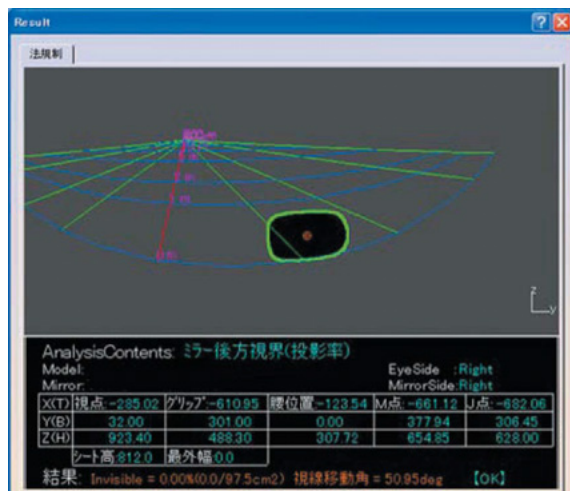


Fig. 10 Before mirror adjustment

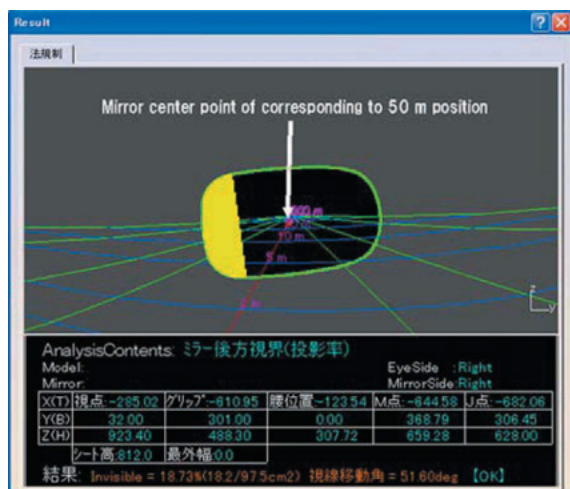


Fig. 11 After mirror automatic adjustment

Introduction of Kart Racing Engine Using 4-stroke General Purpose Engine

Yoshikazu SATO*

Masanori TSUBOUCHI*

ABSTRACT

As a part of Honda R&D's Special Self-study Activities, a kart racing engine was manufactured using the GX200 4-stroke general purpose engine with a displacement of 196.1 cm³ as the base engine. A cylinder head that drives four valves using an OHV configuration was newly designed to secure charging efficiency at higher engine speeds than the typical range of general purpose engines. In addition, the lubrication system was changed from that of a splash lubricant system to a pressure lubrication system to help stabilize lubrication at high engine speeds and during actual driving. As a result, the developed kart engine achieved a power output of 12.0 kW or more at 8 500 rpm, as compared to the maximum power output of 4.1 kW at 3 600 rpm of the base engine. A kart fitted with the developed engine recorded actual lap times equivalent to or better than that of a kart fitted with an existing 2-stroke kart engine. Using a general purpose engine as the base engine enabled an engine to be manufactured that can be mounted in a beginner class racing kart with equivalent ease to that of a 2-stroke kart engine.

1. Introduction

Honda R&D Center provides activity space and manufacturing support for company-sponsored Special Self-study Activities. These activities aim to help employees realize their dreams and acquire skills and knowledge through autonomously creating things and other activities. As a part of these activities, a general purpose engine that can be mounted in a racing kart was manufactured.

Karts are divided into a number of classes (Fig. 1), and among the most easily enjoyed of these are rental karts, which can be used for a fee by anyone that meets certain requirements such as height and age. Most engines mounted in rental karts are 4-stroke general purpose engines with a displacement between 160 and 200 cm³ and a maximum power output of around 4 kW. This is because these engines have good starting performance, do not require gasoline mixed with oil, and have better durability and actual driving fuel economy compared to 2-stroke engines, which gives them good running cost performance.

Next, the class called sports karts^{(1), (2)} uses a general purpose engine mounted on a racing kart chassis, and has easy-to-handle power characteristics that make this vehicle class ideal for drivers starting motor sports. Formerly, sports karts often used 2-stroke scooter engines, but

general purpose engines have come to be widely used in recent years. Sports kart series races are starting to be held in various locations around Japan, and 4-stroke general purpose engines are also gaining widespread acceptance in Europe and America, as typically seen on indoor kart courses, because they produce less white smoke and exhaust gas smell, and are quieter than 2-stroke engines.

The upper class of karts is racing karts, which generally use lightweight, high power output 2-stroke engines. The beginner class uses engines with a displacement of 100 cm³ that produce axial power output

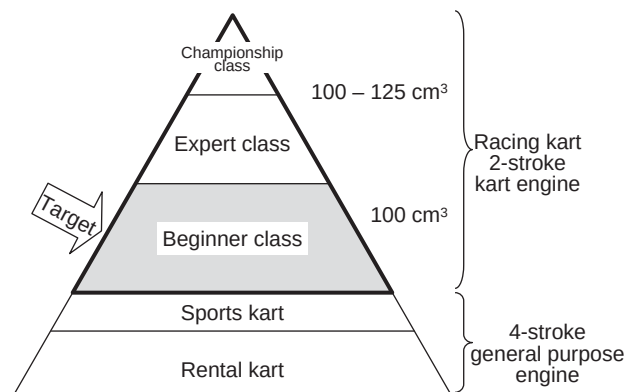


Fig. 1 Kart class conception diagram

* Power Products R&D Center

Table 1 Major specifications

		Kart engine	Base engine
Engine type		Air-cooled 4-stroke OHV single-cylinder	
Displacement (cm ³)		196.1	
Bore x Stroke (mm x mm)		68.0 x 54.0	
Compression ratio		10.0 : 1	8.5 : 1
Valve diameter	Inlet (mm)	25.5	25
	Exhaust (mm)	22	24
Number of valves		4	2
Max valve lift	Inlet (mm)	6.4	5.7
	Exhaust (mm)	5.9	5.7
Max power (kW/rpm)		12.0 / 8,500	4.1 / 3,600
Ignition system		Digital C.D.I.	Transistor magneto
Lubrication system		Pressure lubrication	Forced spray

of 10 kW or more, and the expert class uses engines with a displacement of 125 cm³ that produce axial power output of 20 kW or more and lateral acceleration of 3 G or more while running.

General purpose engines do not have a transmission like a motorcycle engine, so the engine itself is relatively lightweight and easily mounted. For this reason, mounting in racing karts has been desired, but general purpose engines produce lower axial output than that of 2-stroke engines, so they have been considered unsuitable for kart racing, which requires high output. General purpose engines with large displacement are sometimes mounted to obtain high output, but the larger engine size makes mounting on the chassis a challenge, and the increased engine weight hampers the quick driving that is characteristic of karts.

Therefore, the mid-sized GX200 general purpose engine, which is easily mounted, was used as the base engine. A cylinder head with revised inlet and exhaust systems, and the lubrication system were newly designed to achieve high engine speeds and high output, and a general purpose engine for kart racing that is suitable as a beginner class engine was manufactured. This paper reports this process.

2. Design Concept

The following two points are given as the basic performances required of kart racing engines.

- (1) Lightweight, unimpaired steering stability, compactness, and easy mountability.
- (2) High engine speed, high power output, providing good instantaneous power when accelerating out of corners.

Kart racing engines are mainly 2-stroke engines, so the following three items were set as design goals to manufacture a kart racing engine based on a 4-stroke general purpose engine.

- (1) The circuit lap time should be equal to or better than that of a kart fitted with a 2-stroke kart engine with a displacement of 100 cm³.

- (2) The engine should be lightweight, compact, and offer the same ease of mounting as a 2-stroke kart engine.
- (3) The structure should enable easy gear ratio setting at a race circuit.

3. Overview of Manufactured Engine

Figure 2 shows an external view of the newly manufactured engine. Table 1 lists the main specifications, and Fig. 3 compares the axial power output



Fig. 2 External view of engine

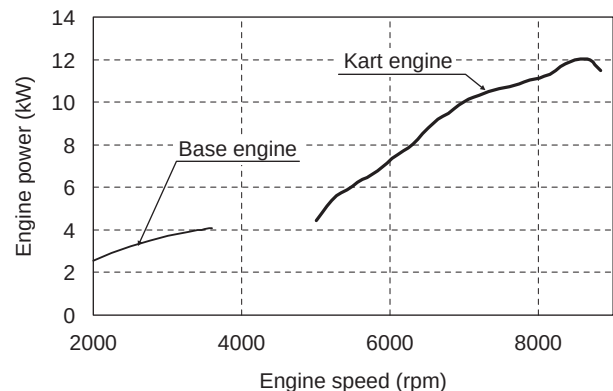


Fig. 3 Engine performance

with that of the base engine. The bore and stroke are the same as those of the base engine, but the cylinder head has been newly manufactured to support a 4-valve valve train. The new engine is designed for use with high-octane gasoline, and the compression ratio has been raised from 8.5 to 10.0 as a means of increasing power output. The lubrication system is a pressure lubrication system that uses an oil pump in order to support high engine speeds.

Figure 4 shows the engine layout drawing. When mounting a general purpose engine, the output is normally led from the crank case cover side, so when mounted on a kart, the power transmission chain is located between the engine and the driver, making it a challenge to change the sprocket. In karts that do not have transmissions, selecting the proper gear ratio to match each circuit is an important setting element, which means that the sprocket should be easy to change. Two-stroke kart engines have a structure that leads the output from the right side of the vehicle between the engine and the tires. Therefore, the output lead-out direction of the newly manufactured engine was made the same as that of a 2-stroke kart engine. This structure is described in detail in Section 4.5.

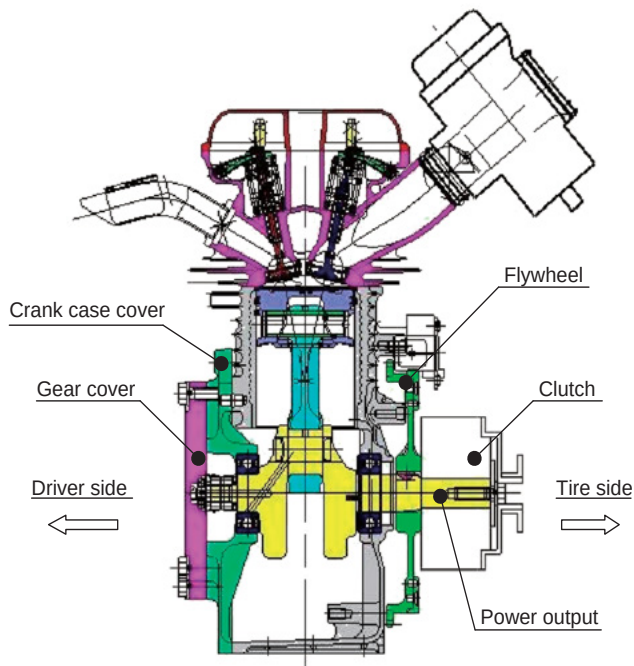


Fig. 4 Layout of kart engine

4. Technologies Enabling High Engine Speeds and High Power Output

The base engine has a maximum engine speed of 3 600 rpm, but the goal for the newly manufactured engine was set as 8 000 rpm or more, without changing the basic framework. The main technologies changed from the base engine are introduced below.

4.1. Increased Inlet Air Volume

The inlet port performance should be enhanced to secure charging efficiency at high engine speeds. The base engine is mounted in various power products such as generators and farm tractors, and is designed to achieve good charging efficiency at 2 500 to 3 000 rpm. However, when the engine speed is increased further, the inlet airflow speed also rises, which increases the airflow resistance and produces a drop in the charging efficiency at higher engine speeds.

In addition, the cylinder head of the base engine is manufactured by high pressure die casting, which accentuated productivity through mass production, so parts with the acute angles needed by the mold structure are formed within the inlet port. Figure 5 (a) shows the inlet and exhaust port shapes of the base engine, and Fig. 5 (b) shows the inlet and exhaust port shapes of the newly manufactured engine. The newly manufactured engine accentuated inlet performance, so the ports were changed to smooth ports manufactured by sand mold casting, and the number of inlet valves was increased from the one valve of the base engine to two valves.

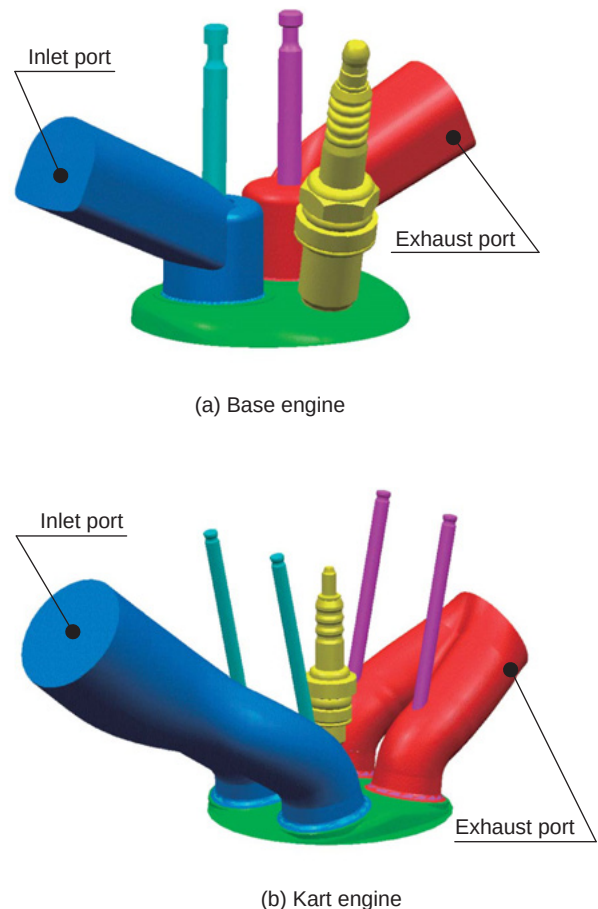


Fig. 5 Inlet and exhaust port 3D shape

Here, the effective opening area (Z) relative to the valve lift was calculated using originally developed Computational Fluid Dynamics (CFD) software⁽³⁾ as an index for evaluating inlet performance. Figure 6 shows the calculation results. The newly designed inlet port has a maximum effective opening area value 2.3 times larger than that of the base engine, which was predicted to reduce airflow resistance at high engine speeds.

The combustion chamber has a pent roof shape, and spark plugs with the small diameter of 8 mm were applied and positioned in the center of the combustion chamber. Note that the thermal load between the two exhaust valve seats increases, so cooling holes were provided to actively introduce cooling air.

4.2. High-speed Valve Train Operation

The basic valve train structure uses the same OHV mechanism as the base engine, but a new mechanism was devised to drive four valves using two push rods in order to increase the number of inlet and exhaust valves. Various mechanisms for driving four valves in an OHV configuration have been reported in the past^{(4), (5)}, but the new engine uses a layout with sub-rocker arms positioned orthogonally to the main rocker arm. This enables two inlet and two exhaust valves to be operated by the conventional rocker arm motion. Figure 7 shows the valve train configuration.

The valve size was changed from the IN: $\phi 25$ mm x 1, EXH: $\phi 24$ mm x 1 of the base engine to IN: $\phi 25.5$ mm x 2, EXH: $\phi 22$ mm x 2. In addition, the valve stem

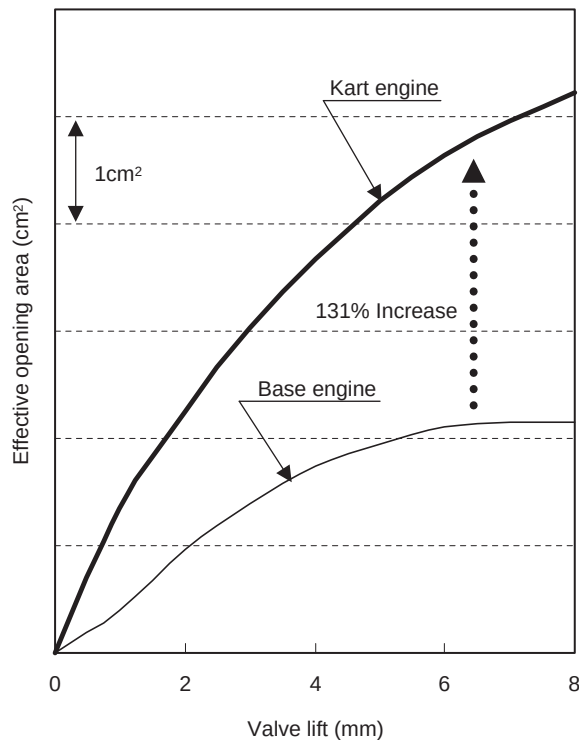


Fig. 6 Calculated intake effective opening area

diameter was reduced from the $\phi 5.5$ mm of the base engine to $\phi 4.0$ mm, which reduced the weight by 4 g per inlet valve and 7 g per outlet valve, and reduced the inlet resistance. The retainer was changed to a cotter type that can withstand high engine speeds, and special valve springs with an uneven pitch were newly designed. The new engine used $\phi 7.5$ mm solid aluminum shaft OHV push rods that have the same coefficient of thermal expansion as the cylinder block, while maintaining equal or greater rigidity than that of the $\phi 4.0$ mm steel push rods in the base engine. In addition, the valve lifter weight was reduced by 3 g by machining. Figure 8 shows the new valve lifter shape.

These measures resulted in a reciprocating part equivalent weight of 128 g compared to 70 g for the base engine, so the increase in weight was kept to 1.8 times despite the addition of two valves and two rocker arms to the base engine.

However, the new engine operates two valves with a single push rod, so the equivalent weight per cam

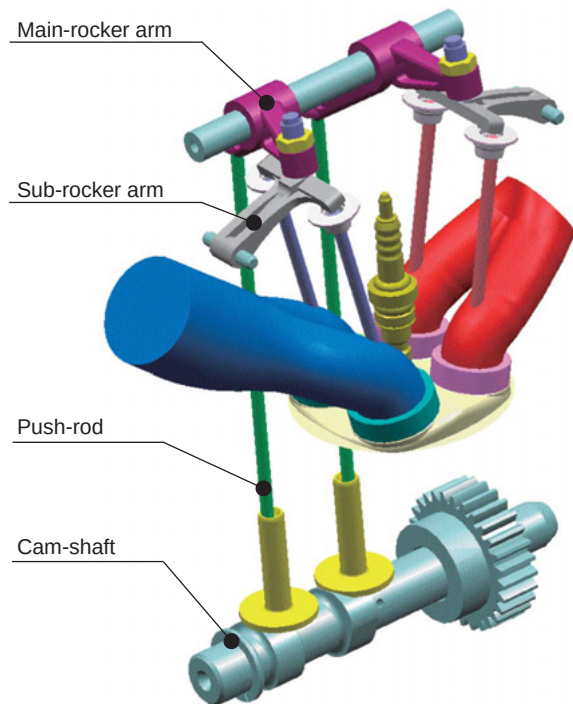


Fig. 7 Cylinder head for OHV 4-valve configuration

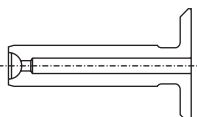
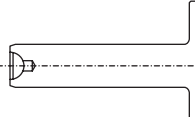
	Kart engine	Base engine
Shape		
Mass (g)	16	19

Fig. 8 Comparison of valve lifter shape

increases, and valve behavior may become unstable due to the effects of the dynamic rigidity of the valve train at high engine speeds. Therefore, the positive acceleration on the valve lift was reviewed to enhance charging efficiency and stabilize valve behavior at high engine speeds. Specifically, the maximum positive acceleration value was suppressed to 47% of that of the base engine by expanding the overlap and delaying the inlet valve close timing angle. The larger maximum valve lift amount increased the axial torque, but reduced power performance and drivability were confirmed, which were thought to be due to unstable valve behavior occurring at engine speeds lower than the maximum engine speed. Therefore, the valve lift amount was set at IN: 6.4 mm, EXH: 5.9 mm. This achieved a maximum engine speed of 8 000 rpm or more.

4.3. Pistons and Piston Rings

The piston skirt was shortened and the weight reduced by approximately 10 g to reduce the reciprocating part inertial force that is generated during high-speed operation and to minimize vibration during operation. In addition, the sliding surface with the cylinder bore was given a molybdenum coat to reduce friction. Figure 9 shows the differences in the piston shape.

The piston ring thickness was reduced to reduce the reciprocating part weight and friction. The top and second rings were made of steel, and the h1 dimension was reduced from 1.2 to 0.8 mm. The oil ring thickness was also reduced from the conventional h1 dimension of 2.5 to 2.0 mm. In addition, the total tension was reduced by 47% while still satisfying oil consumption performance during races and maintaining appropriate trackability. Table 2 compares the section shapes of the piston rings.

4.4. Lubrication System

The base engine uses a splash lubrication system that collects the lubricating oil on the inner bottom surface of the engine block, and splashes the lubricating oil using a dipper integrated with the connecting rod. The newly manufactured engine calls for a stable supply of lubricating oil to the large end of the connecting rod to support engine speeds in excess of 8 000 rpm and changes in the oil surface when subject to the lateral acceleration

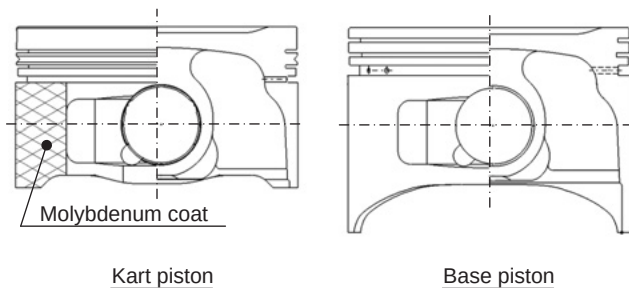
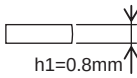
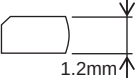
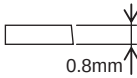
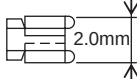
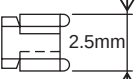


Fig. 9 Comparison of piston shape

Table 2 Comparison of piston ring

	Kart engine	Base engine
Top Ring	 h1=0.8mm	 1.2mm
2nd Ring	 0.8mm	 1.2mm
Oil Ring	 2.0mm	 2.5mm

described hereafter. Therefore, an oil pump was built into the newly designed crank case cover, and a pressure lubrication system was used to drive the oil pump gear from the crank shaft via the cam shaft. There are two sets of lubrication channels: one that passes from the oil pump through the cylinder head to the rocker arm parts, and another that passes from the oil pump through the crank shaft to the large end of the connecting rod.

The connecting rod of the base engine is made of aluminum, and the rod portion used in the new engine is the same as that of the base engine. Figure 10 shows the difference in the connecting rod shapes. The inertial load of the reciprocating part that is generated near the top dead center increases at high engine speeds, so the thickness of the cap side was increased by 1.7 times to increase the loop rigidity. In contrast to the mass production specifications that assume use in diverse environments from low- to high-temperature regions, races are not held at sub-zero temperatures, so the oil clearance of the newly manufactured engine was reduced, the rigidity of the large end increased, and the oil film holding toughness at high engine speeds also enhanced.

Next, the effects of changes in the oil surface on lubrication during actual driving were assessed. A big difference between bench tests and actual driving tests is the presence of lateral acceleration (hereafter, "lateral G")

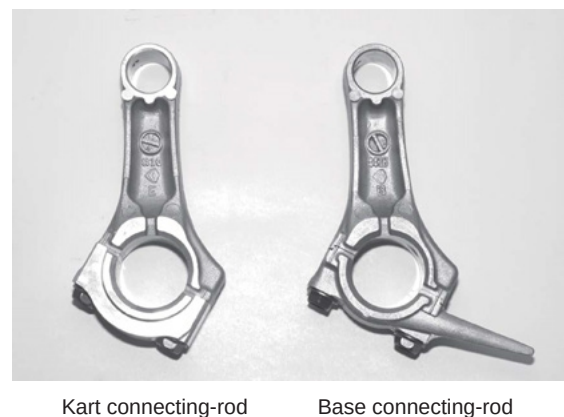


Fig. 10 Comparison of connecting-rod shape

generated during cornering. The general purpose engines mounted in generators and farm tractors are rarely subjected to sharp lateral G for long periods. However, when mounted in a kart, lateral G equivalent to 1.5 to 2.0 G is produced, and can continue for about 5 to 7 seconds. At this time, the oil surface is thought to become inclined as shown in Fig. 11, which reduces the amount of contact between the dipper and the oil surface, and the amount of splash produced by the cam gear in a splash lubrication system.

Figure 12 shows the lateral G and the change in oil temperature produced during actual driving. The horizontal axis shows the time, and the vertical axis shows the change in oil temperature and the lateral G produced. The red line in the upper level indicates the change in oil temperature for the pressure lubrication system, and the blue line in the middle level indicates the change in oil temperature for the conventional splash lubrication system. The orange line in the lower level indicates the lateral G,

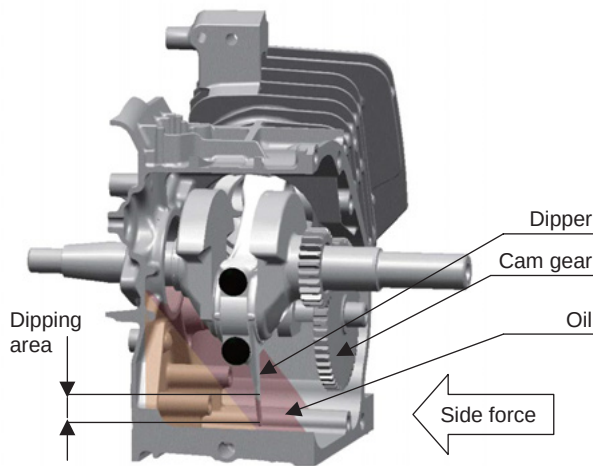


Fig. 11 Lubricant system of general purpose engine

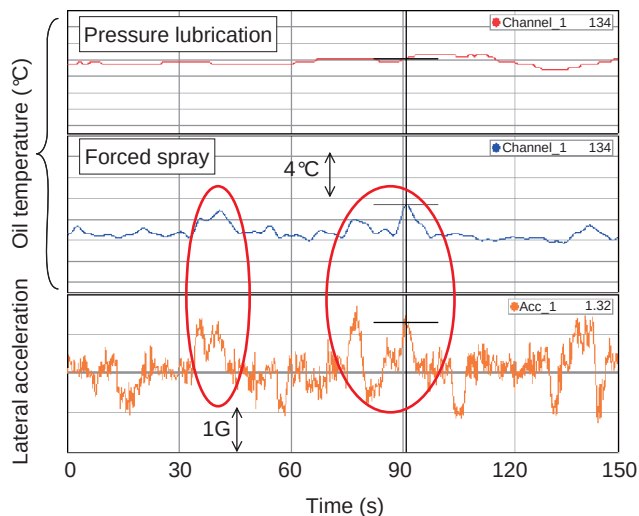


Fig. 12 Relation between lateral acceleration and oil temperature

and shows that the oil temperature rises at the timing when lateral G is generated. This is thought to be because lubrication becomes insufficient during cornering, which increases the friction heat on the large end of the connecting rod. Figure 13 shows a diagram of the new engine with the pressure lubrication system. The oil inlet is positioned near the center inside the case where there is little change in the oil surface due to lateral G, and a baffle plate has also been installed around the inlet to reduce the effects of changes in the oil surface. Changing to a pressure lubrication system and providing a stable supply of lubricating oil to the large end of the connecting rod was confirmed to effectively suppress fluctuations in the oil temperature due to changes in the oil surface under lateral G conditions.

4.5. Drive Train

In karts that do not have transmissions, the sprocket is replaced to change the gear ratio setting, depending on whether the maximum speed, cornering acceleration, or other factor is prioritized in the circuit course. Figure 14

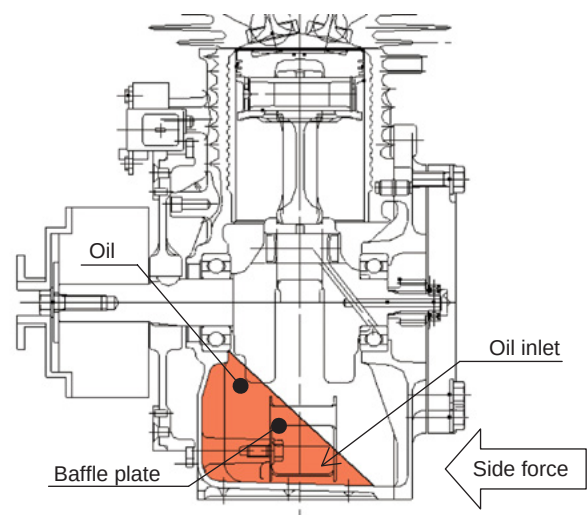


Fig. 13 Lubrication system of kart engine

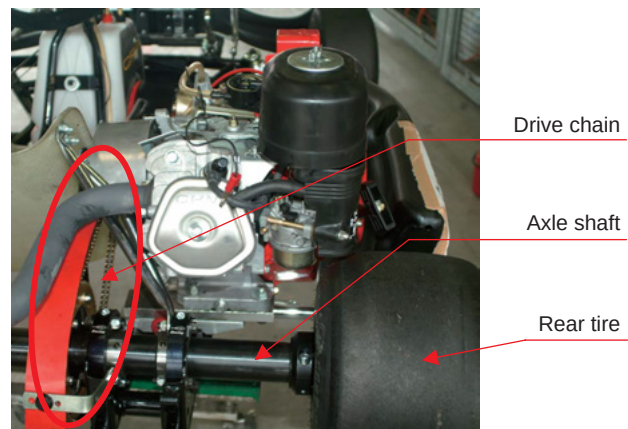


Fig. 14 Drive chain placement with general purpose engine

shows a typical kart fitted with a general purpose engine, and here the output is led from the crank case cover side, which is on the inside of the kart. This made it necessary to remove the axle shaft and replace it together with the chain when changing the rear sprocket, which entailed much time and effort. The rear sprocket can also be split in half and assembled, but level differences in the assembled condition may cause the chain to slip off. Furthermore, when it is necessary to also change the chain length when changing the sprocket, it is not enough to merely replace the rear sprocket.

Therefore, the output lead-out in the newly manufactured engine was changed to the flywheel side, which is located on the outside of the kart, and a centrifugal clutch was added. Figure 15 shows a photograph of the engine mounted on a kart. This enabled the sprocket and chain to be replaced in the engine-mounted condition without removing the axle shaft, which enabled easy setting of the gear ratio at the race circuit in the same manner as a conventional kart engine.

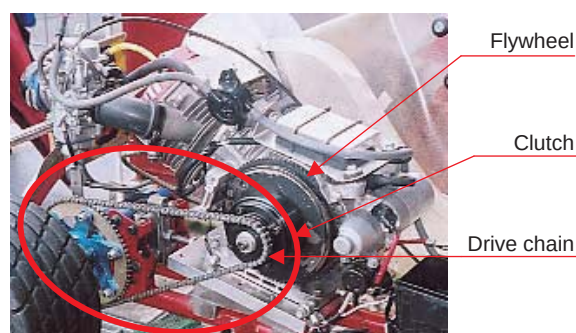


Fig. 15 Drive chain placement with kart engine

5. Actual Driving Results

The newly manufactured engine and the base engine were mounted in karts, and power performance was compared by the lap times on the Twin Ring Motegi road course. This road course is a circuit with a length of 4.8013 km. Figure 16 shows the lap time taken for the same test driver. A kart fitted with the newly manufactured engine achieved a lap time 52 s shorter than specifications with the speed governor removed from the GX200 base engine. This means that the kart fitted with the newly manufactured engine achieved an average lap speed of 108 km/h, in contrast to an average lap speed of 81 km/h for the kart fitted with the base engine. Furthermore, the new engine was also found to provide a running advantage of 5 s or more, compared to a kart fitted with a beginner class 2-stroke kart engine with a displacement of 100 cm³.

In addition, Fig. 17 compares the acceleration times from 85 to 95 km/h as a comparison of speeds when

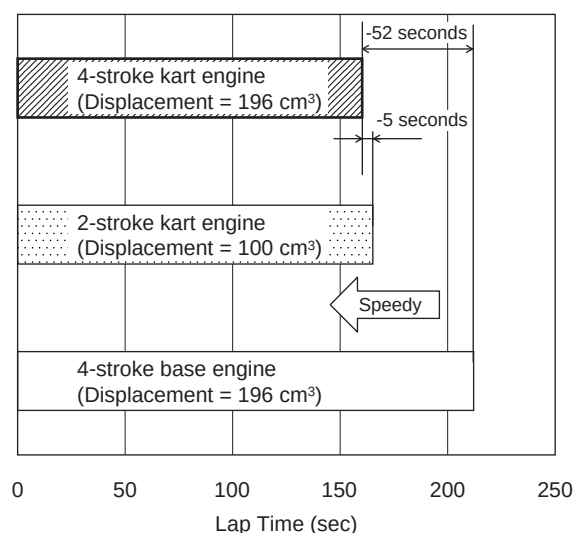


Fig. 16 Comparison of lap time

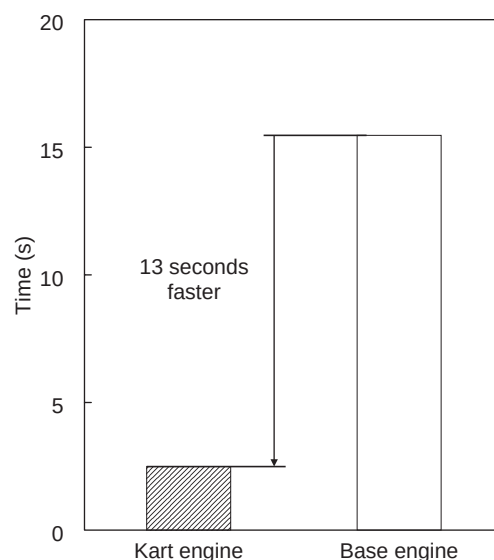


Fig. 17 Comparison of acceleration time

accelerating out of corners. Here as well, the new engine provides enhanced acceleration performance compared to the base engine.

6. Conclusion

A lightweight, compact, high power output kart racing engine was manufactured that makes the most of the advantages of a general purpose engine. Figure 18 shows a racing kart fitted with the newly manufactured engine.

- (1) A cylinder head that supports an OHV 4-valve valve train mechanism was newly manufactured, and achieved an engine power output of 12.0 kW or more at 8 500 rpm.
- (2) An engine with the same ease of mounting as that of a 2-stroke kart engine was manufactured using a 4-stroke



Fig. 18 Kart race at TWIN RING MOTEGI

general purpose engine as the base engine. In addition, the structure leads the engine power output from the flywheel side, which enables the gear ratio to be set as easily as that of a conventional kart engine.

- (3) Use of a pressure lubrication system stabilized oil temperature fluctuations under conditions with lateral G applied for long periods, and increased reliability at high engine speeds.
- (4) In actual driving tests on a circuit with a length of 4.8013 km, the developed kart engine was confirmed to shorten the lap time by approximately 5 s, compared to a kart fitted with a 2-stroke kart engine with a displacement of 100 cm³.

Afterward

Power products tend to be thought of as relatively estranged from motor sports. However, autonomously designing and manufacturing a race engine and entering a kart fitted with this engine in kart races enabled the authors to actualize their own ideas, and to autonomously put into practice Honda's Three Realities Principle of knowing the actual situation, going to the actual place, and being realistic. This greatly increases the motivation of young engineers, and it is also hoped that the skills and ideas acquired through this process can be fed back to other power products in the future.

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Development of Enhanced Honeycomb Barrier Finite Element Model for Rear Crash Simulations

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ABSTRACT

This paper explains the development of an enhanced finite element barrier model for use in simulating the FMVSS 301 high speed rear crash test.

To create the barrier model, material testing was conducted on the aluminum honeycomb, cladding sheets, and adhesive which comprise the barrier. Particular emphasis was placed on the modeling of the aluminum honeycomb, which has been traditionally very challenging to model due to its highly anisotropic behavior. Recently developed modeling features were used to capture the complex behavior of the honeycomb material and extensive testing was conducted to define its behavior.

To validate the barrier model, a series of specification impact tests were conducted. These tests were chosen to represent the various load conditions the barrier might experience during crash testing. Each impact test was simulated and the barrier model was correlated to the test results.

Finally the barrier model was used in a full-vehicle crash simulation for verification. In this full-vehicle crash simulation, a significant improvement in the correlation of the model to test results was achieved. The results clearly indicate that improved predictive accuracy in crash test simulations can be obtained with the use of this new barrier model.

1. Introduction

Many automotive crash tests involve impact with a honeycomb barrier as a surrogate for another vehicle. During these crash test events, the honeycomb barrier typically undergoes severe deformations and extreme non-linear behavior. The simplicity of the construction of these barriers can lead one to believe that modeling them would be rather straightforward, but this simplicity is

deceptive. To capture the behavior of these barriers with a finite element model has been, and continues to be, a major challenge for the automotive industry.

The new FMVSS 301 high speed rear crash test uses the same Moving Deformable Barrier (MDB) as the FMVSS 214 side impact test⁽¹⁾. The MDB has a honeycomb barrier face mounted to the front of the cart. In the new high speed rear crash test, the MDB impacts the vehicle at 50 mph with 70% overlap. As Fig. 1



Fig. 1 Typical damage to honeycomb barrier seen after FMVSS 301 high speed rear crash test

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shows, the severity of this collision causes extreme deformation of the honeycomb barrier face. The current barrier model is not able to capture this behavior accurately; so as a result, there is poor correlation of the simulation model.

To enhance the predictive accuracy of rear crash test simulations a new barrier model was created. This model utilizes recently developed features in LS-DYNA for modeling aluminum honeycomb to more accurately capture its off-axis strength and shear behavior. Also, a new approach for modeling the adhesive that bonds the honeycomb to the cladding was developed. This model was correlated to a number of different specification tests and has shown significant improvement to rear impact simulation correlation.

2. Barrier Modeling

2.1. Barrier Construction

The barrier used in the rear impact testing consists of a face assembly and a bumper assembly⁽²⁾. The face assembly is comprised of the main honeycomb block and cladding sheets. The crush strength of the face assembly honeycomb is 45 psi. This assembly has two aluminum cladding sheets with a thickness of 0.812 mm. The face sheet is bonded to the front side of the main honeycomb block and wraps around the top of the block (no adhesive is applied to the top portion). The main honeycomb block is also bonded to an aluminum backing sheet. The bumper assembly consists of the honeycomb bumper having a crush strength of 245 psi and two aluminum cladding sheets with a thickness of 3.175 mm. The front cladding sheet is bonded to the front face of the bumper honeycomb while the bumper backing sheet is bonded to the back face of the bumper honeycomb. Finally the bumper assembly is bonded to the face assembly. This barrier construction and adhesive layers are detailed in Fig. 2.

The barrier is mounted to a cart designed to meet specified values for center of gravity and inertia. This cart is typically a space frame style design made from

steel or aluminum tubes. The cart has four wheels with pneumatic tires mounted to it, but no suspension.

2.2. Barrier Model Construction

The aluminum honeycomb material is modeled using solid elements. In LS-DYNA, a solid element formulation is assigned that allows for the elements to undergo the extreme shear behavior possible in honeycomb material. The cladding is modeled with shell elements, and all adhesive layers are modeled with solid elements. The material modeling used for these components will be described in the next sections.

The cart for the barrier is modeled as a rigid body. Rigid solid elements are used to represent the basic space frame structure of the cart and the proper values for the inertia and center of gravity are applied using an inertial card in LS-DYNA. Each tire is modeled with shell elements and an air volume is defined between the tire and wheel. Each tire is attached to the cart using revolute joints to allow for rotation.

2.3. Aluminum Honeycomb Modeling

The deformation behavior of aluminum honeycomb is extremely difficult to capture accurately in a finite element model. This material is extremely anisotropic in its behavior: it can exhibit high crush strength in the strong direction (T direction), while showing extremely weak crush strength in an orthogonal direction (W and L directions). These directions are illustrated in Fig. 3.

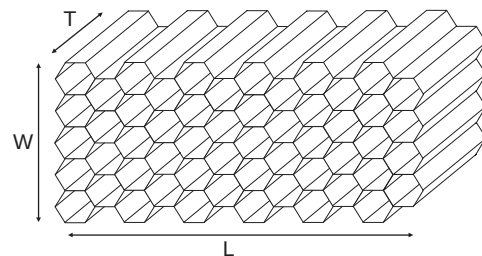


Fig. 3 Description of aluminum honeycomb

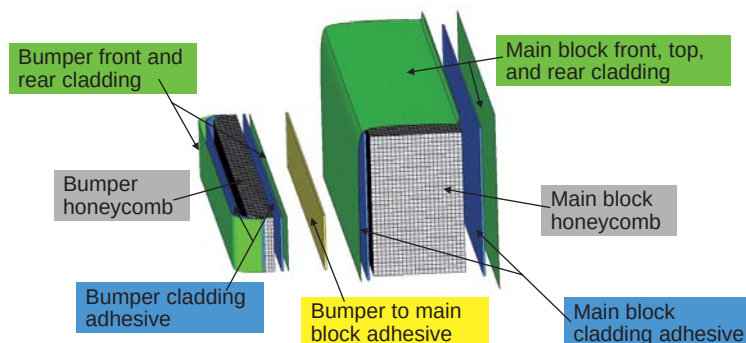


Fig. 2 Barrier construction

While many recent published attempts to model aluminum honeycomb have used shell elements to provide an explicit representation of the cell structure⁽³⁾⁻⁽⁵⁾, the model described here uses brick elements to limit the model's size and run time. Recently developed features in LS-DYNA are used to allow the brick elements to capture the honeycomb behavior.

With the traditional material model for aluminum honeycomb, a user could define the crush strength of the honeycomb in the strong and the weak directions. This material modeling method was found to over predict the material strength in an off-axis loading case⁽⁶⁾. To address this issue an improved honeycomb material model was developed by Kojima et. al. and implemented in LS-DYNA with the Mat_Modified_Honeycomb material model^{(6), (7)}. With this material model, a yield surface can be defined such that the limit stress, σ^y , is a function of the angle, φ , with the strong axis and the volumetric strain, ε^{vol} :

$$\sigma^y(\varphi, \varepsilon^{vol}) = \sigma^b(\varphi) + (\cos \varphi)^2 \sigma^s(\varepsilon^{vol}) + (\sin \varphi)^2 \sigma^w(\varepsilon^{vol}) \quad (1)$$

In LS-DYNA, load curves are used to define σ^b , σ^s , and σ^w in Eq. (1) with respect to volumetric strain. The crush strength of the honeycomb with respect to the angle of crush is defined by σ^b . Stiffening effects observed in

the honeycomb as it is crushed to its fully compacted state are defined by σ^s and σ^w in the strong axis or weak axes respectively.

Defining σ^b is critical to the accuracy of this material model and is not a trivial undertaking⁽⁶⁾. To produce this, load curve samples must be cut from the honeycomb block at a variety of angles. Figure 4 shows the samples removed from the honeycomb block. Each sample was crushed, and a load versus displacement graph was generated. A number of the resulting curves are shown in Fig. 5. From these curves, the plateau strength of the honeycomb can be determined for each sample. Finally, a curve can be generated which shows the strength versus angle of crush. A representation of this curve is shown in Fig. 6. By using this curve, a steady decrease in honeycomb strength can be obtained as the orientation of the crush direction transitions from the strong direction to a weak direction. Thus, the off-axis loading can be accurately defined.

2.4. Cladding Modeling

The aluminum cladding sheets were tested using uniaxial samples to provide a stress-strain curve. This curve was adjusted to a true stress versus effective plastic strain curve and smoothed using a power law fit. Finally,



Fig. 4 Samples cut from honeycomb block at different angles

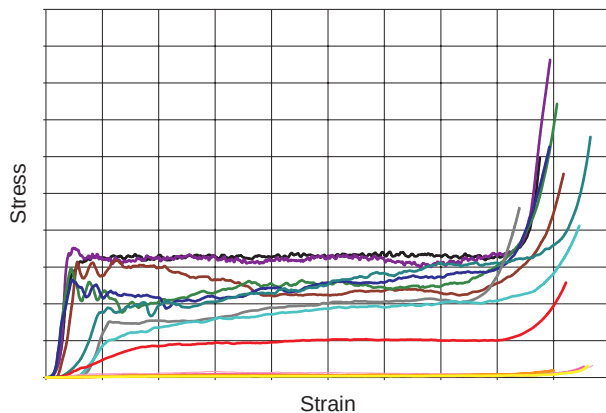


Fig. 5 Load versus displacement curves for honeycomb samples

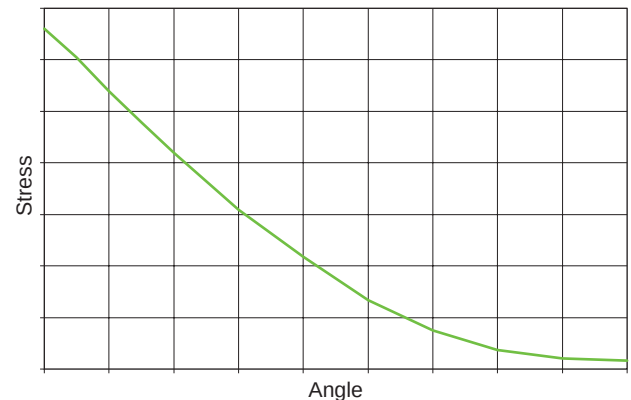


Fig. 6 Representation of curve defining crush strength versus angle of crush

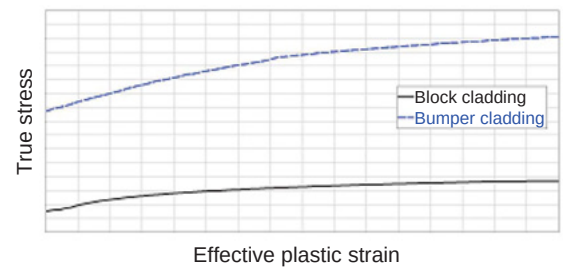


Fig. 7 True stress versus effective plastic strain for aluminum cladding material

it was sampled at 100 equally spaced intervals and used with a piecewise-linear plasticity material model in LS-DYNA. This curve is shown in Fig. 7. Failure was allowed using an effective plastic strain value beyond which the element would be eliminated. This failure strain was adjusted from the tested value to provide reasonable correlation of the cladding failure in each of the specification tests.

2.5. Adhesive Modeling

2.5.1. Honeycomb to cladding adhesive

The honeycomb to cladding adhesive, indicated in Fig. 2, is modeled as a single layer of solid elements that is attached to the honeycomb and cladding using a tied contact. The adhesive layer material definition is identical to the main honeycomb block with the exception that failure parameters are defined using tensile and shear strain limits. As Fig. 8 shows, the specification tests demonstrate that the failure mode is the tearing of the honeycomb material rather than the separation of the adhesive layer. Therefore, the model represents the failure in the honeycomb, and the adhesive is represented as the tied contact.



Fig. 8 Honeycomb layer torn away with cladding (note undisturbed layer of adhesive present under torn honeycomb layer)

2.5.2. Cladding to cladding adhesive

The bumper rear cladding and block front cladding are attached using an adhesive. This is represented in the model as a thin layer of solid elements which are attached to the cladding sheets through a tied contact definition. The solid adhesive layer material model includes the following failure function to represent the adhesive failure:

$$\left[\frac{\max(N_{rr}, 0)}{N_{rrf}} \right]^2 + \left[\frac{N_{rs}}{N_{rsf}} \right]^2 + \left[\frac{N_{rt}}{N_{rtf}} \right]^2 + \left[\frac{M_{rr}}{M_{rrf}} \right]^2 + \left[\frac{M_{rs}}{M_{rsf}} \right]^2 + \left[\frac{M_{rt}}{M_{rtf}} \right]^2 - 1 = 0 \quad (2)$$

The numerators in the failure definition are the resultants calculated in the model, and the denominators are the failure values specified in the material model⁽⁷⁾.

The failure parameters for all adhesive layers in the barrier model were determined using barrier specification tests that were conducted in order to validate the material models. Each barrier specification test produced unique failures of the adhesive layers. By adjusting the failure parameters in Eq. (2), the simulation could match the adhesive failure seen in the specification test. It was found to be impossible to exactly match the adhesive failure seen in all specification tests with the same failure parameters. This is likely due to the variability seen in the barriers where the adhesive bond between the honeycomb and cladding is not consistent. Ultimately, a single set of failure parameters was selected where the adhesive layer showed reasonable failure correlation for all specification tests.

3. Barrier Specification Testing and Modeling

3.1. Flat Wall Specification Test

The flat wall specification test validates the material properties for the main block honeycomb material. There is no failure in the honeycomb block since the test applies

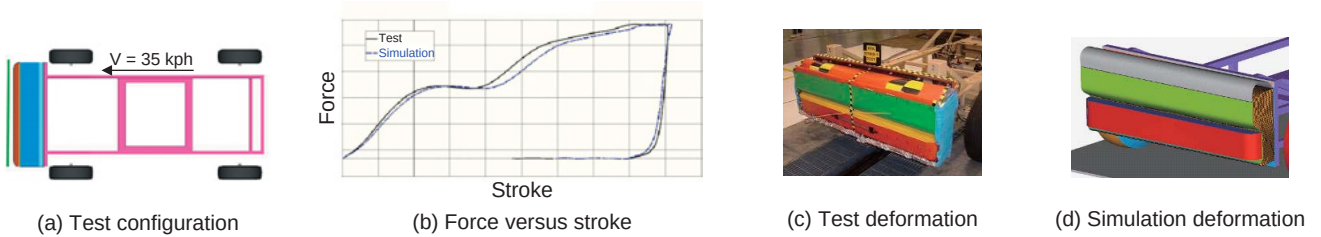


Fig. 9 Flat wall specification test

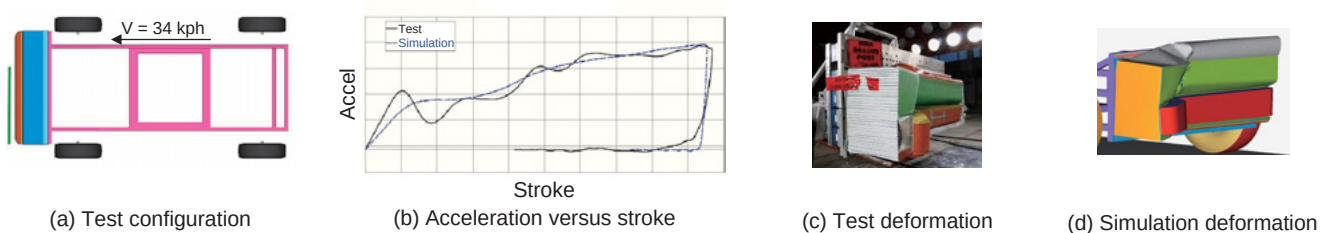


Fig. 10 Offset wall specification test

a uniform load on the face of the barrier. Since there is no block failure in the test, the simulation model and the test show a high level of correlation as shown in Fig. 9.

3.2. Offset Flat Wall Specification Test

The 70% offset wall specification test showed no cladding failure. The plastic failure strain for the cladding material was adjusted so that little or no cladding failure was present in the model. The failure parameters from Eq. (2) for the adhesive layer between the block front cladding and bumper rear cladding were also set in order to correlate to this test. Although separation of the bumper assembly from the block does not occur in the simulation model as is does in the test, the correlation level shown in Fig. 10 is judged to be well within the variability for the barrier's performance.

3.3. Pole Specification Test

In the initial stages of the pole impact, the test and the simulation model correlate very well. However, towards the end of the event, the simulation model diverges from the test results by behaving stiffer than the physical barrier. Although the honeycomb material definition allows the user to define various failure functions, the main honeycomb block does not have a defined failure

behavior. Failure was not added to the honeycomb material at this point to try and ensure that the barrier model would behave very robustly in full vehicle simulations (this is discussed further in Section 4.1). As can be noted in Fig. 11 (c), the main honeycomb block clearly has some failure that allows the pole fixture to punch into the honeycomb block. Since the simulation model correlates very well until late into the event, the discrepancy is judged to be reasonable, and the overall correlation level is judged to be acceptable.

3.4. Offset Bumper Test

The overall deformed shape of the offset bumper simulation model compared to the test result shows an acceptable level of agreement as shown in Fig. 12. The bumper component remains attached to the block face as the block cladding tears at the impact point. However, there is a great deal of failure of the honeycomb block in this test. If failure in the honeycomb block were to be modeled, the correlation to the test would become very close. However, in order to maintain contact stability and overall robustness of the simulation model, failure is not applied to the main block honeycomb. This results in a stiffer response to this load case. This type of isolated failure in the barrier block is not usually noted in a full

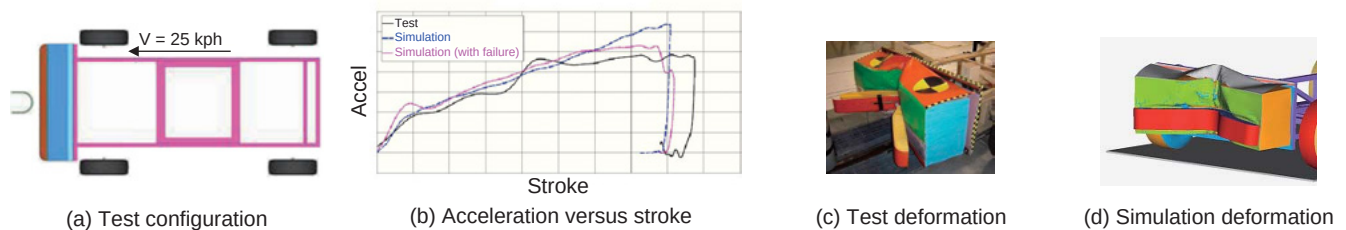


Fig. 11 Pole specification test

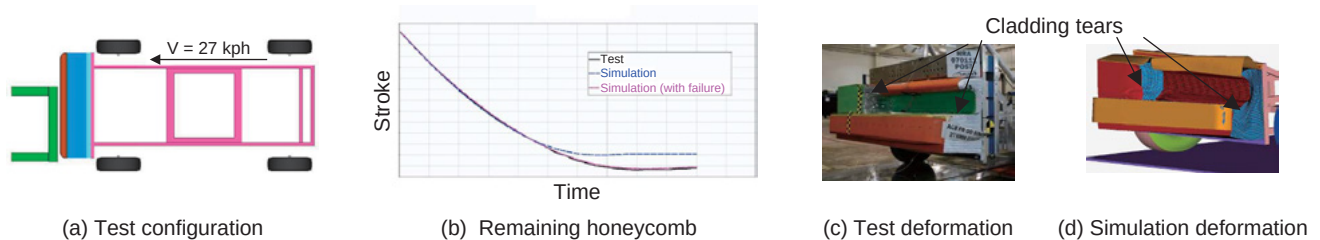


Fig. 12 Offset bumper test

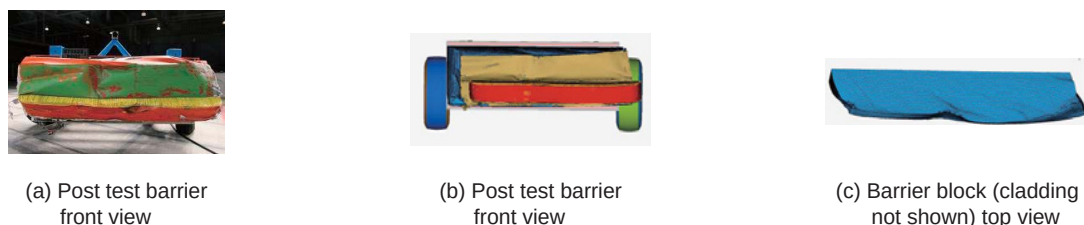


Fig. 13 Top view of initial model's barrier block illustrating non-physical shearing behavior

vehicle testing, and the overall correlation level is judged to be acceptable.

3.5. Angle Wall Specification Test

With the selected material model used for the honeycomb material, shear stability can at times be poor. This is noted in Fig. 13 where the barrier model was used in a FMVSS 214 side impact full vehicle simulation. With this test mode, the barrier impacts the vehicle in a “crabbed” manner which introduces a shear load into the honeycomb block. In the simulation model, the barrier block was sheared in a manner that was not physically representative of the actual behavior.

To prevent this type of unstable shear behavior, LS-DYNA has the option to apply limits to the shear and hydrostatic stresses⁽⁷⁾. These limits allow for the material to develop resistance to shear and hydrostatic stress while the uniaxial response remains unaffected and is still governed by Eq. (1). The hydrostatic and shear limit stresses are given as functions of volumetric strain by Eqs. (3) and (4) respectively:

$$\sigma_p^Y(\epsilon^{vol}) = \sigma_p^Y + \sigma^s(\epsilon^{vol}) \quad (3)$$

$$\sigma_d^Y(\epsilon^{vol}) = \sigma_d^Y + \sigma^s(\epsilon^{vol}) \quad (4)$$

The same load curve defined for Eq. (1) is represented by σ^s , which provides for stiffening of the material as it is crushed to its fully compacted state. The parameters σ_p^Y and σ_d^Y were determined using the offset bumper specification test from Fig. 12 and the angled wall specification test shown in Fig. 14.

For the angle wall impact test, the barrier model without any limits to the shear or hydrostatic stresses

shows the unstable shear behavior noted in the side impact test. This can be seen in Fig. 15 (b) where the simulation model shows excessive shearing of the honeycomb elements to the side versus the actual behavior shown in the test barrier of Fig. 15 (a). Applying the limit stresses can cause excessive stiffening in the shear behavior of the honeycomb so a compromise must be implemented. By adjusting the limit stresses for the angle wall specification test while monitoring their effect on the offset bumper impact, values were obtained that provided the needed shear stability, and also kept the correlation of the offset bumper impact at an acceptable level. The result for the angle wall simulation is shown in Fig. 15 (c).

4. Model Robustness and Efficiency

4.1. Model Robustness

A major goal for updating the barrier models, other than enhanced accuracy, is improved barrier robustness. This is a critical issue when running large models such as full vehicle crash models. With the material models and element formulations chosen for this model, the new barrier has proven to be very robust in full vehicle simulations. It can undergo the severe deformations seen in the actual barriers and still remain stable. Thus, the new barrier model can be applied in production use.

It has been shown that adding failure to the honeycomb block can improve the correlation to the specification tests. Also, honeycomb failure in the main block can be present in rear offset tests as is evidenced by Fig. 1. This may lead to the desire to incorporate honeycomb material failure into the material model, but adding this failure to the honeycomb block can lead to stability issues and unexpected results if the barrier model

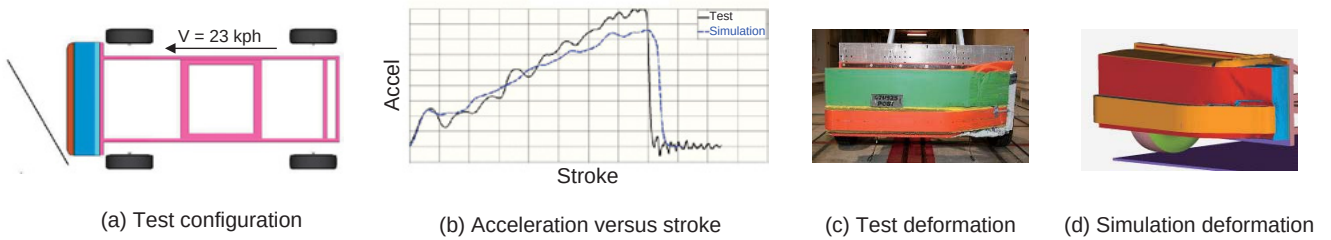


Fig. 14 Angle wall test

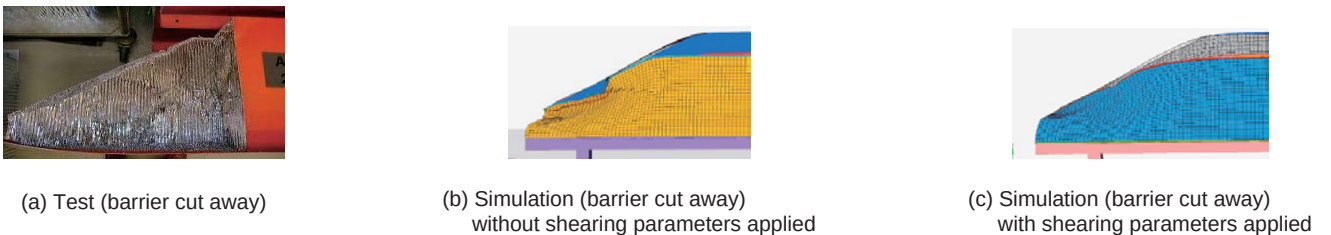


Fig. 15 Top view of section cut of test barrier compared to simulation

were to experience a deformation mode significantly different from those encompassed by the specification tests. To avoid these troubles, an attempt was made to avoid applying failure to the honeycomb block material model. This has proven to be successful as the barrier shows good correlation for all specification tests and for full vehicle simulations (detailed in Section 5). Thus, the goal of achieving a barrier model with excellent robustness and accuracy has been achieved.

4.2. Model Efficiency

Another goal for the barrier model is to maintain computational efficiency while enhancing accuracy. Ultimately, barrier models should be a shell mesh of the aluminum foil that makes up the actual honeycomb structure; however with current techniques this would be far too computationally expensive (the run time penalty factor would be approximately 15), so compromises must be made. One possibility is to use a shell mesh of the of a honeycomb structure where the cells are scaled up from their actual size. This leads to a lower cell density, fewer elements, and thus, improved computational efficiency over a barrier with a shell mesh at the actual cell density⁽³⁾⁻⁽⁵⁾. But, for the rear impact barrier this would still lead to a barrier with a large computational expense penalty over the current barrier model. Therefore, an attempt was made to develop a barrier model utilizing solid elements to keep the computational cost penalty to a minimum. With the barrier model developed here, the computation

time is increased by a factor of 1.25 compared to the old barrier model. This is an acceptable increase, given the improvements in accuracy, because this increase is well within the range that can be overcome by adding additional CPU's for each simulation.

5. Full Vehicle Verification

With the barrier properly correlated to spec tests, its performance in full-vehicle rear crash models was evaluated. The old barrier showed significant weakness in rear crash simulations - often producing results with excessive deformation to the vehicle and even incorrect deformation modes. It was thought that an improved barrier model would resolve these issues.

The new barrier model has shown a significant improvement to full-vehicle simulation accuracy. The difference in intrusion between the simulation prediction and the test result has been greatly reduced with the use of the new barrier model. Also, the deformation modes appear much closer to that of the test. These results are shown in Fig. 16.

6. Conclusion

A new barrier model for rear crash simulation has been developed. This model takes advantage of new modeling features in LS-DYNA to more accurately capture the behavior of the aluminum honeycomb. The model was

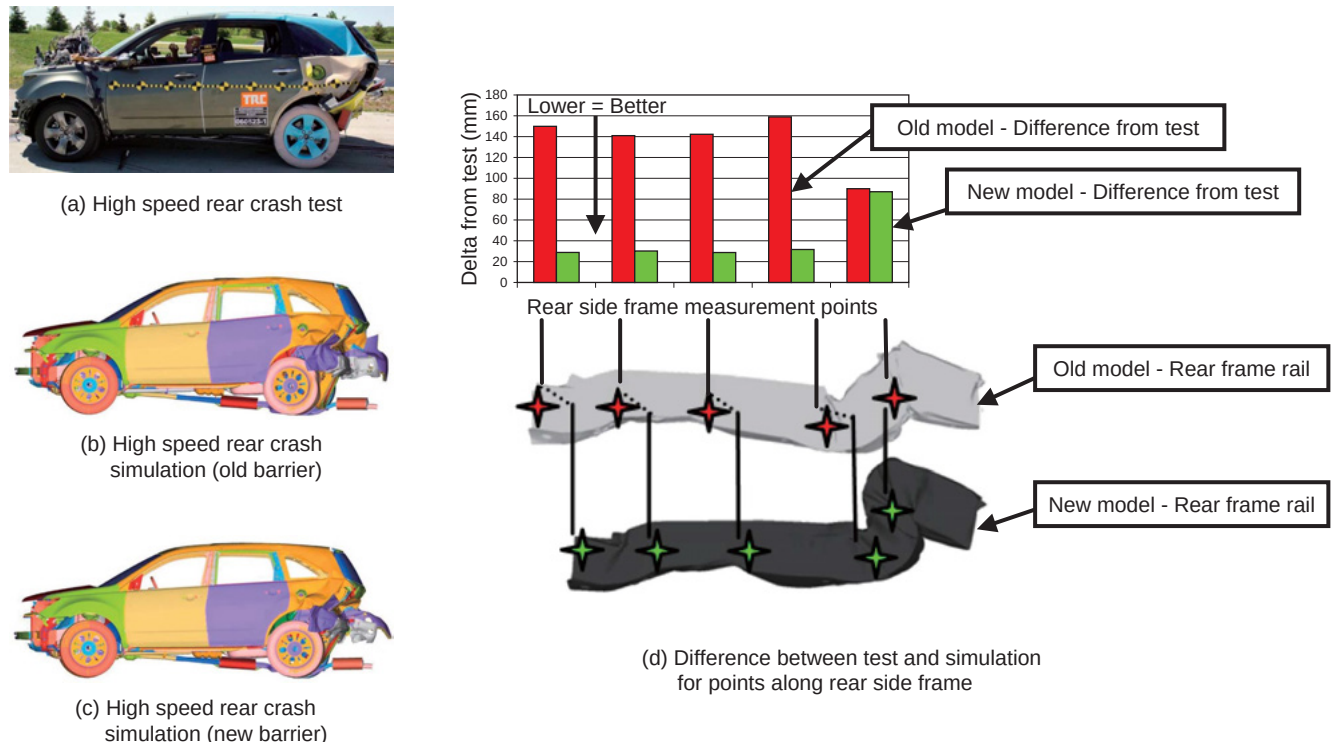


Fig. 16 Correlation of rear frame rail with new barrier model

correlated to a number of specification tests. While perfect correlation to every specification test was found to be impossible, a reasonable compromise in correlation to each specification test was achieved. This new barrier model has been used in full-vehicle, rear crash simulations where it has shown significant improvements in its correlation to test results. The barrier model has proven to be very robust in actual usage by avoiding any instabilities that could lead to incomplete or pre-maturely terminated simulation runs. The barrier does increase the computational time needed to run a model. However, in comparison to an approach that would introduce computationally intensive shell elements, the increase in time is deemed reasonable. These improvements to the barrier model provide for an accurate, predictive, rear crash simulation tool that can be used in vehicle development.

There are of course still areas where further enhancements could be made to the barrier model. Future work should address items such as 1) incorporating failure into the aluminum honeycomb block in a robust accurate manner, 2) moving toward a shell element model when computational speed increases, and 3) accounting for any dynamic effects present in the barrier at high speeds.

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■ Author ■



Allen Sheldon



Emily Nutwell

Improved Method to Assess Spot Weld Rupture in LS-DYNA using Solid Element Assemblies

Skye Malcolm* Emily Nutwell*

ABSTRACT

Various methods of modeling spot welds with failure criteria are considered for the solver LS-DYNA, a general purpose transient dynamic finite element program. These methods include the failure methodology of the spot weld, the meshing technique of the weld itself, and the resulting dependence on the mesh of the sheet metal attached to the weld. A force-based failure criterion as measured through an assembly of 8 hexahedrons is used to best judge spot weld rupture. This judgment method is calibrated in weld coupon simulations. A script is written in the simulation preprocessor to apply the calibrated failure parameters efficiently in simulations with thousands of spot welds. Finally the method is applied to full vehicle safety simulations where the correlation to actual test results is verified.

1. Introduction

Our overall goal is to create a reliable and efficient method to evaluate the strength of spot welds in automotive structures in computer simulations of static and dynamic safety tests. We set out to reliably predict the rupture of spot welds during the simulation so as to be able to visualize the effects of that rupture on the overall structure and improve the robustness of the design.

Our work was driven by the more specific goal of using a spot weld failure methodology based on peak force through the weld. This matches our current evaluation criteria for spot welds in actual vehicles. In fact, we could draw upon an existing library of spot weld strength data from single spot weld coupon tests. This library is based primarily on material strength and thickness of the sheet metal being welded. Therefore, our strategy is to use a force-based failure criterion in the simulations.

There are several existing weld modeling methods which can be used to assess the strength of those welds. The first existing method is the mesh dependent rigid body weld. This method has a proper failure criterion, but due to efficiency considerations and the inability to model 3 sheet welds easily, its disadvantages outweigh its

benefits. Another method is the beam weld which is not directly attached to the nodes of the shell mesh but is instead tied to an arbitrary point on the shell's surface with a so-called tied contact. This method is very efficient when constructing models and can model 3 sheet welds easily. Unfortunately, due to the single point contact with the shells, the force measured by this type of weld varies enough to add an unacceptable level of uncertainty into the measured force. The final existing method is a single hexahedron element, again tied to the shell surface through a tied contact and not through coincident mesh methods. However, even this type of element displays distortion of the force due to penetrations of the shells into this solid element in certain cases. Since all these existing methods can not be improved upon, we worked in conjunction with a software maker to create a new weld method made from an assembly of hexahedron elements through which the sum of all forces was combined into a single resultant force representing the load measured by the whole weld.

Having made a truly mesh independent representation of the force in the spot weld, we then show the simulations representing the library of spot weld coupon tests to determine the failure parameters. These failure parameters are set using the failure forces from the test library and confirmed by a DOE study of all the material

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strength and gage combination in a parameterized model run using optimization software to automatically prepare, run and analyze several dozen simulations efficiently.

Then we describe an efficient method to assign the proper failure parameters to each weld stack up in vehicle model with thousands of spot welds. An algorithm designed and programmed as a custom macro in the preprocessor allows the engineers working on a large model to select some or all of the spot welds and quickly assign or reassign weld strengths to those welds. Finally, the accuracy of this method is shown in a full vehicle side impact model and compared to an actual crash test.

2. Failure Method Based on Peak Force

In order to select which failure methodology to focus on, the available methods in LS-DYNA are evaluated and relative merits are considered.

The oldest failure method, so-called the “*CONSTRAINED_SPOTWELD” method, is also closely tied to the meshing technique, because it is a mesh dependent method. It requires a node of the shell elements representing the sheet metal to be coincident with the node of the rigid body representing the spot weld. The failure criterion is shown in Eq. 1⁽¹⁾ for this method and is valid once FC becomes greater than 1 as the normal force f_n and the shear force f_s increase to their respective strengths, S_n and S_s .

$$FC = \left(\frac{|f_n|}{S_n} \right)^p + \left(\frac{|f_s|}{S_s} \right)^q \quad (1)$$

Although the failure method available for *CONSTRAINED_SPOTWELD takes into account shear and tensile loads, the resulting inefficiency imposed on the model build process by having to mesh the shells with the weld points predefined is simply too great an expense. In addition, this method is difficult to apply to 3 or 4 sheet welding because there will be nodes that are constrained

by more than one rigid body. For these practical considerations this method is rejected.

A second method considered called *MAT_100_DA is still in development during this period of research. Because it is undergoing changes, there are practical issues of limited access to versions of both the solver, LS-DYNA, and of a preprocessor which supports all the features necessary to implement that failure method. Because of these practical considerations this method is also rejected.

The third and final failure method we ultimately chose to use is based on force and initially available for beams or a single hexahedron. This method is called *MAT_100 with the option flag set to zero. Ultimately, the LS-DYNA code is modified to also allow this failure criterion to work with the resultant force-based failure criteria as measured through an assembly of 8 hexahedrons. This method takes into account forces and moments in three axes for a total of six failure criteria as shown in Fig. 1 and Eq. 2⁽¹⁾.

$$FC = \sqrt{\left(\frac{\max(N_r, 0)}{NRR} \right)^2 + \left(\frac{N_s}{NRS} \right)^2 + \left(\frac{N_t}{NRT} \right)^2 + \left(\frac{M_r}{MRR} \right)^2 + \left(\frac{M_s}{MSS} \right)^2 + \left(\frac{M_t}{MTT} \right)^2} \quad (2)$$

NRR (tension) is a function of thickness and nugget diameter. NRS equals NRT (shear) and they are a function of thickness, nugget diameter and material grade. MRR (torsion) is a function of thickness, and nugget diameter. MSS equals MTT (peel) and they are a function of thickness, nugget diameter, and material grade.

It should be noted that there is no strain rate component to the weld evaluation data or criteria. The basis for this omission is the research from Madasamy⁽²⁾. In his study, spot weld coupon tests using mild and high strength steels in hundreds of thickness and material combinations have showed insignificant strain rate dependence.

3. Achieving True Independence from Artifacts of the Mesh

First, although it may be assumed that the mesh dependence of beam welds with single point constraints to the shell mesh representing the sheet metal is well known, a small experiment is done to demonstrate this fact. A coupon welded by a single beam is put in shear and the force through the beam is measured. This beam is placed at the center of the quad shell element, then at a node and finally at an edge and the force of the beam plotted is easily shown to be dependent on its position on the shell mesh. This violates the assumption of mesh independence our failure criterion requires and thus invalidates the use of beams of this type. Figure 2 below shows this variability in the force.

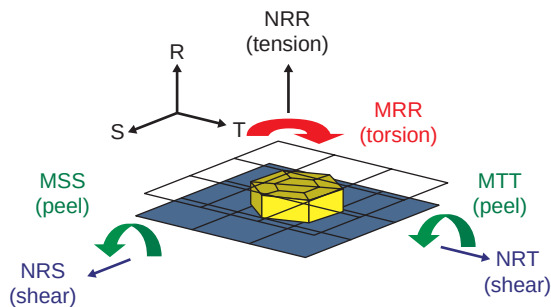


Fig. 1 Single weld assembly illustrating directions of failure criteria

Next, the more subtle distinction can be shown between hex representations of the spot weld. The internal weld forces used as the weld failure criteria should remain consistent regardless of relative placement of the weld to the master segments of the sheet metal in order to successfully apply predictive weld failure. In this study, the mesh of the weld is refined from 1-hex representation and then further refinement of 4-hex, 8-hex, and 16-hex representation is performed in order to determine the best weld configuration for the application of weld failure. The internal weld forces are then compared to the external forces of the model to confirm that they are in good agreement, and in order to evaluate the mesh independence of the weld⁽³⁾. The mesh size of

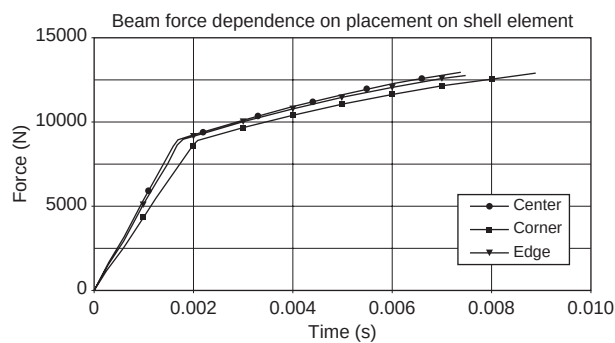


Fig. 2 Illustration of mesh dependence of beam welds

the master segment is set to 5 mm, which is an appropriate mesh size for full vehicle crash models, while the diameter of the weld is 8 mm, a large but plausible size of a spot weld applied to the vehicles. Four cases are considered as shown in Fig. 3 and the experimental set up is shown in Fig. 4.

Comparing the internal weld forces in Fig. 5, the single hex weld is subject to spurious weld forces due to the master segment passing through the weld. The 4-hex assembly compares reasonably well to the more detailed weld representations. There is a minimal difference between the 8-hex and 16-hex configurations indicating that further mesh refinement beyond eight elements is unnecessary. By taking a cross section cut view of the model through the weld in the deformed shape, the importance of the mesh refinement becomes clear, as shown in Fig. 6. As the number of nodes is increased on the slave side of the weld contact, the contact becomes more robust; therefore, it becomes a more stable representation of the weld connection.

The underlying cause of this phenomenon is illustrated in Fig. 7(a) through Fig. 7(c) where we can see almost by inspection that the root of this problem is the simple mismatch between the master shells of the contact and the slave nodes in the weld. Fig. 7(a) shows the tied contact for a single hex weld. The center element of the master

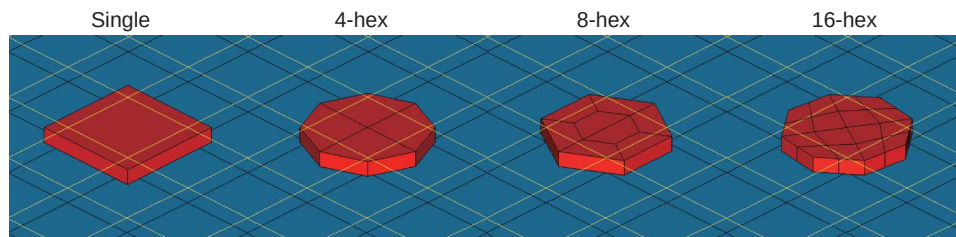


Fig. 3 Hex weld mesh refinement ranging from 1-hex to 16-hex configurations

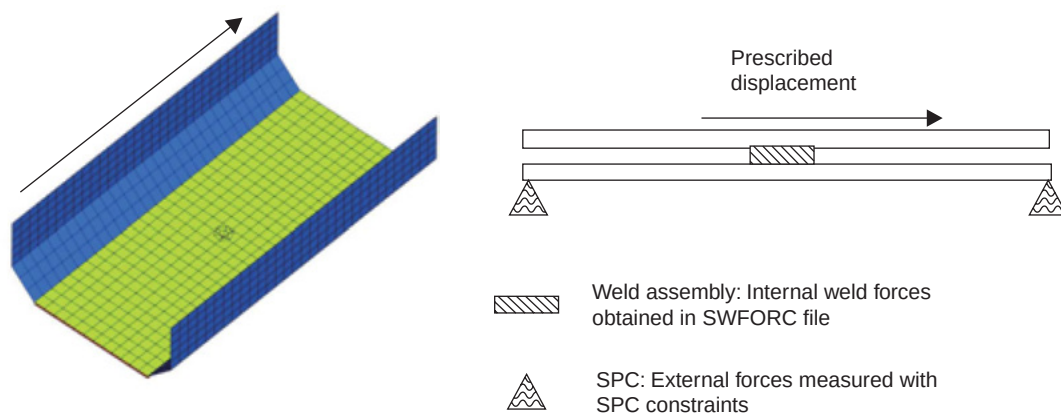


Fig. 4 Mesh and schematic diagram of simple test model to evaluate mesh independence within LS-DYNA

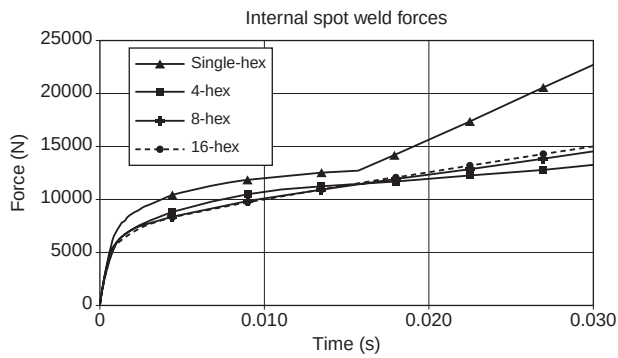


Fig. 5 Internal weld forces comparing effects of mesh refinement of weld

segment is not included in the tied contact because the weld element is larger than the master segment mesh. This allows the master segment to pass through the weld. This results in a non physical representation of the weld. Fig. 7(b) shows the 4-hex configuration. As the master segment deforms, there remains some pass through with the tied contact because the mesh of the weld continues to be coarse. This is accounted for to a point by using the SPOTHIN parameter. This results in a softening of the weld internal forces because it allows the master sheets to move closer together inside the weld which is not physical. Finally, Fig. 7(c) shows the contact patch for the 8-hex configuration which is similar to that of the 4-hex weld. The tied contact is more robust due to the

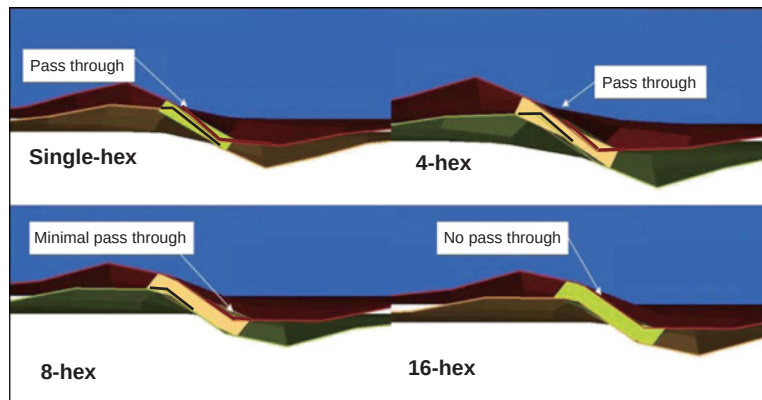


Fig. 6 Cross section view of deformed shape for various weld mesh densities

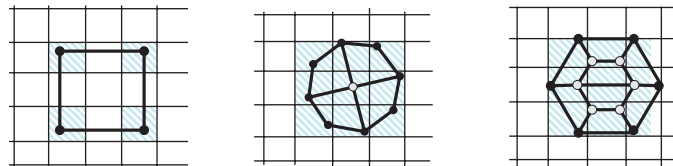


Fig. 7 Schematic of tied contact patch for 1-hex (a), 4-hex (b), and 8-hex (c)

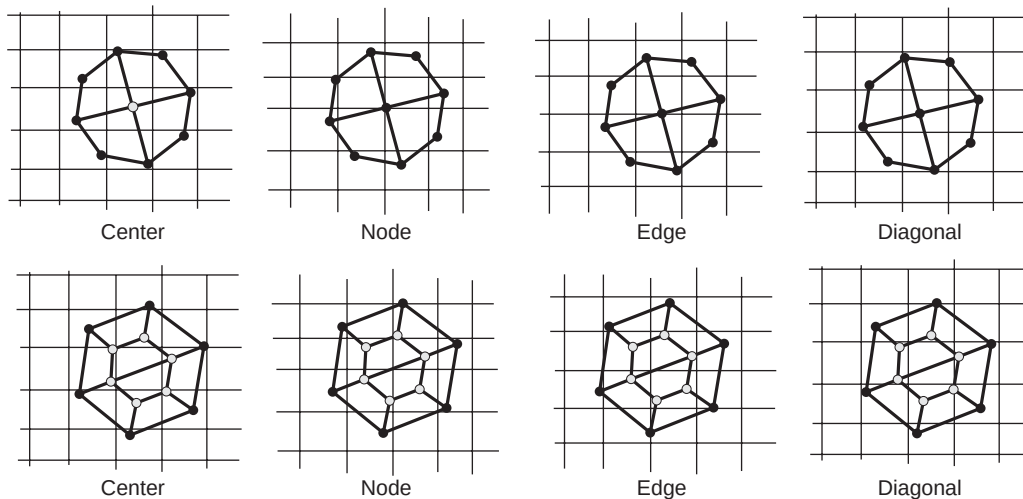


Fig. 8 Positional cases evaluating mesh dependence for 4-hex and 8-hex weld assemblies

weld consisting of an inner ring of nodes, rather than a single node for the 4-hex assembly. The internal weld force is within 1% of the 16-hex configuration indicating that the refinement of eight elements for the weld is sufficient.

In order to properly connect the two under all circumstances it is necessary to have at least one node from the weld on each master segment of all the shells in direct contact with the weld. For the range of mesh sizes studied here, it was determined that the 8-hex

assembly shown in Fig. 7(c) was independent enough of the adjoining shell mesh as to render it reliable.

Mesh dependence is considered for the case of changing the relative position of the weld to the master segment as shown in Fig. 8. The mesh density of the master segment is slightly altered in order to avoid moving the weld which would change the physicality of the model.

The internal and external forces shown in Figs. 9 and

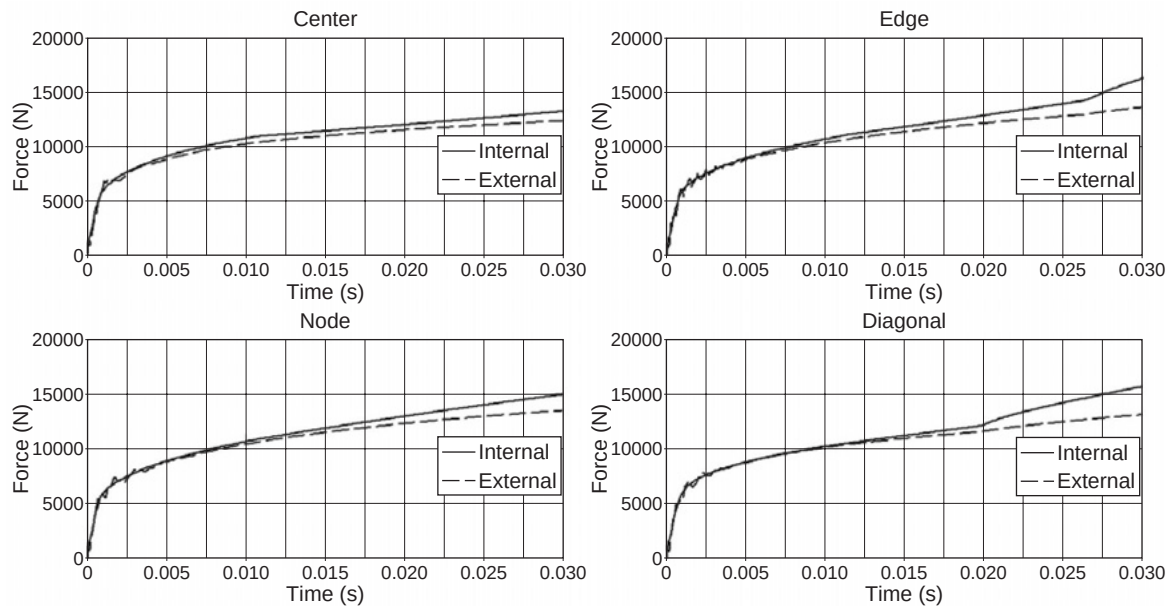


Fig. 9 Comparison of internal weld forces to external forces of 4-hex welds

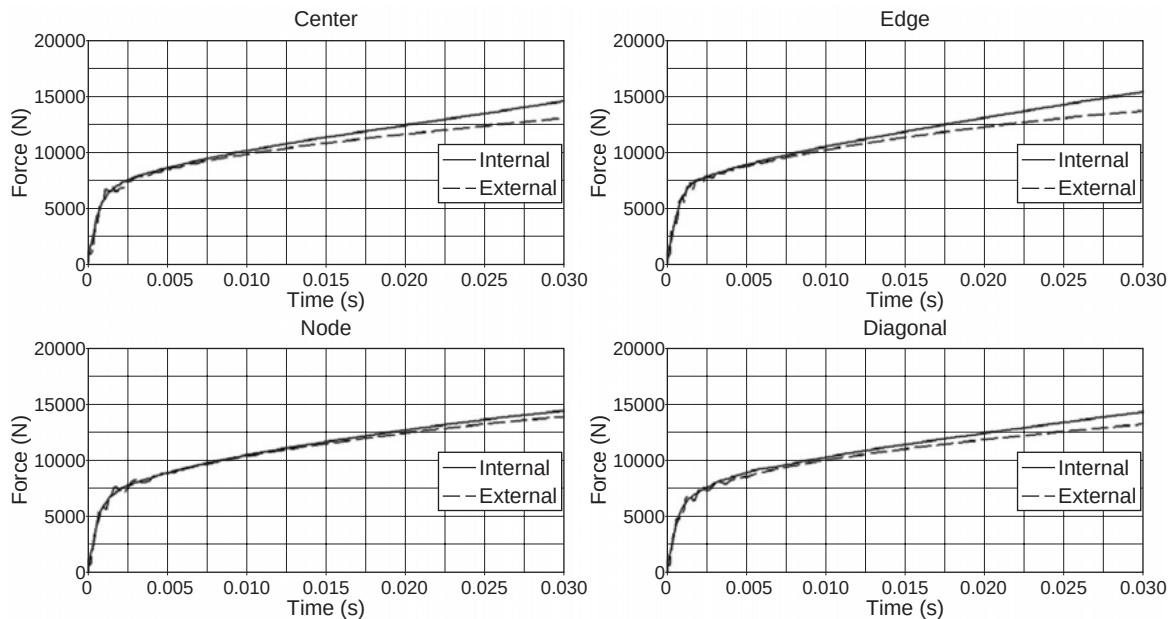


Fig. 10 Comparison of internal weld forces to external forces of 8-hex welds

10 compare well for both the 4-hex and 8-hex welds. However, there are spurious weld forces in the 4-hex case due to the contact not being as robust and allowing the master segment to pass through the weld. The severity of this pass through is dependent on the location of the weld relative to the master segment. This issue does not present itself in the case of the 8-hex weld assembly.

Mesh dependence is then considered for the case of rotating the weld relative to the master segment as shown in Fig. 11.

Mesh dependence of the 4-hex assembly is noted when comparing the internal and external forces [Fig. 12(a)]. The 8-hex weld assembly has a much greater internal

energy since the tied contact is much more robust and the mesh is fine enough to capture the deformed shape of the weld [Fig. 12(b)]. This results in the internal forces of the weld to be very stable and in good agreement with the external forces. However, the 4-hex weld assembly exhibits severe hour glassing, indicating a lack of robustness with the tied contact, and that the mesh is too coarse to capture physical behavior of the weld. This results in measurable mesh dependence, and a lack of agreement between the internal weld forces and the external forces.

Although not shown in detail here, the 16-hex weld assembly does not exhibit any further benefits from the 8-hex version. Therefore to avoid unnecessary complexity

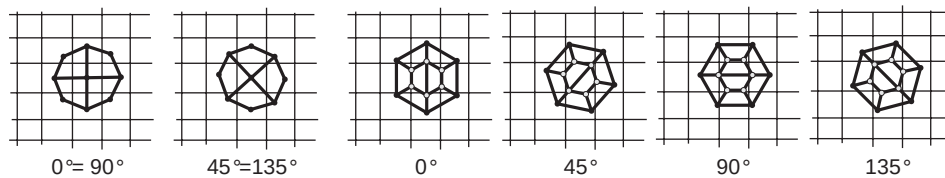


Fig. 11 Rotational cases evaluating mesh dependence for 4-hex and 8-hex weld assemblies

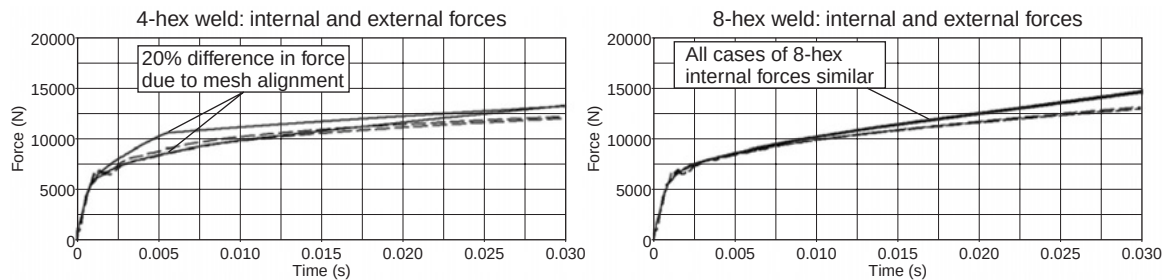


Fig. 12(a) Comparison of internal weld forces to external forces of 4-hex and 8-hex welds for each rotational case

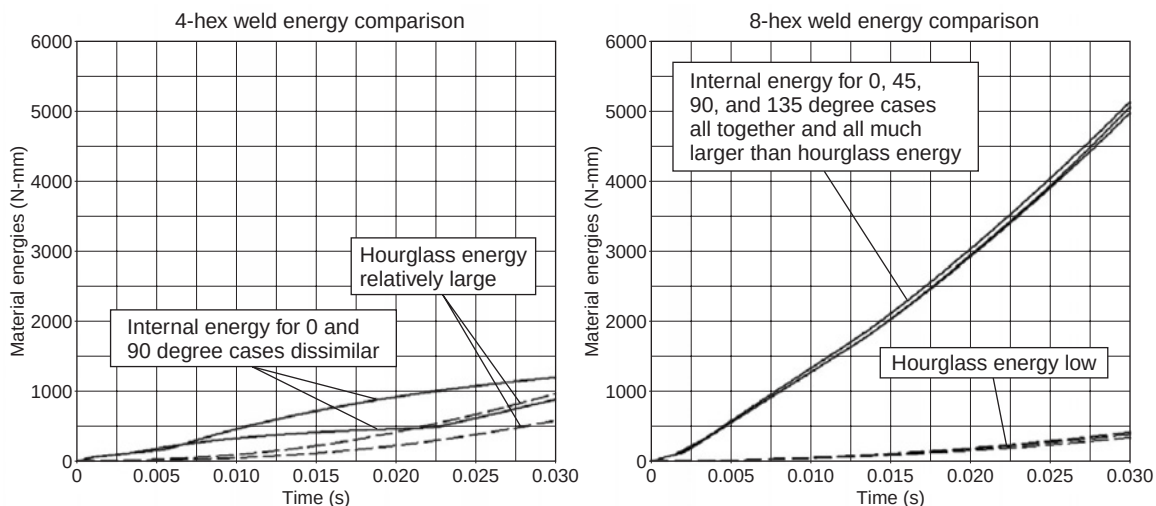


Fig. 12(b) Internal and hourglass energy for 4-hex and 8-hex weld assemblies for each rotational case

and based on these results, the best weld configuration is an 8-hex weld assembly in order to successfully apply weld failure in a predictive manner. Internal weld forces, which are used as the criteria for the failure model, are stable. No error will be introduced as a result of any mesh dependence of the weld.

4. Calibration of the Failure Parameters to Coupon Tests

One assumption of this work is that there exists a database of spot weld strengths for the entire matrix of spot welds. These strengths depend on the thickness and material of each sheet in the weld, the number of sheets and the nugget diameter (which itself depends on the welding schedule). From an iterative routine using commercially available optimization software the correct

values for the failure parameters NRR, NRS/NRT, MRR, and MSS/MTT are determined stochastically. The starting values for the failure parameters are the external peak loads achieved by actual tests. However, due to the slight differences seen in the previous section between the internal weld forces and the externally measured load, these values do undergo adjustment on the order of 10% until the measured load achieved by simulation matches the target values. This method is sufficient to determine NRR, NRS and NRT. The weld coupon simulation models used for this correlation are shown in Fig. 13.

These two experiments are sufficient to determine the shear and tensile failure parameters. Furthermore, these are the only existing experiments where complete data for all material grades and sheet thicknesses are available. However, the torsion and moment failure parameters of MRR and MSS/MTT are still unknown. If the values are left as 0 in the material definition, then the failure criteria may not be reached in certain limited load cases, which are not common as the initial rupture, but can become the failure mode after several welds have ruptured. The following process is used here. MRR is assumed to be proportional to NRS/NRT and MSS/MTT are assumed to be proportional to NRR. Keep in mind that MRR, MSS, and MTT are not forces but moments. Once this proportionality relationship is assumed a very limited subset of simulation experiments is run and checked against a single case in the published literature and a single internal experiment⁽⁴⁾. These simulations are shown in Fig. 14.

5. Automatic Assigning of Failure Parameters in Large Models

A proprietary script is developed for multiple preprocessors which interrogates each weld and determines its connections. Thickness of the sheet metal, material grade of the sheet metal and the total number of sheets in the weld are all taken into consideration for the script. Once the script determines this data for a given spot weld, a look up table of weld failure parameters is consulted by the script and a MAT_100 material is created with those failure parameters and that weld element is moved into that part. If a part with those same properties already exists, then the weld is simply moved into that existing part with other welds of the same configuration.

This script works in conjunction with the entity known as a connector which already contains the connection information and as is the case can create the mesh element from the connector format efficiently. As a result of careful programming of the script, we are able to assign or reassign all the welds in a full vehicle model on the order of 10 minutes using commonly available engineering computer workstations.

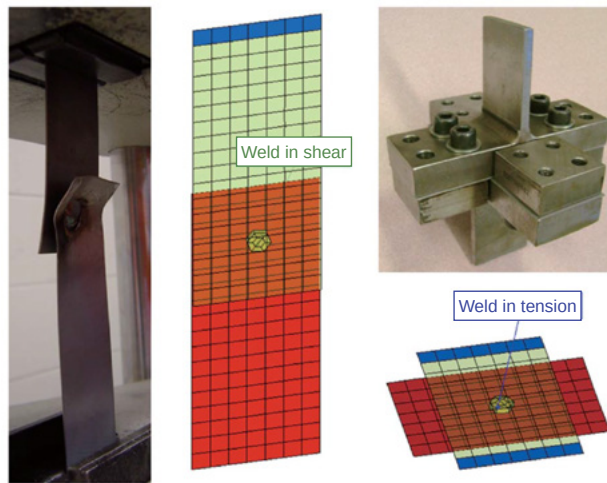


Fig. 13 Test and simulation spot weld coupons showing shear and tensile load cases

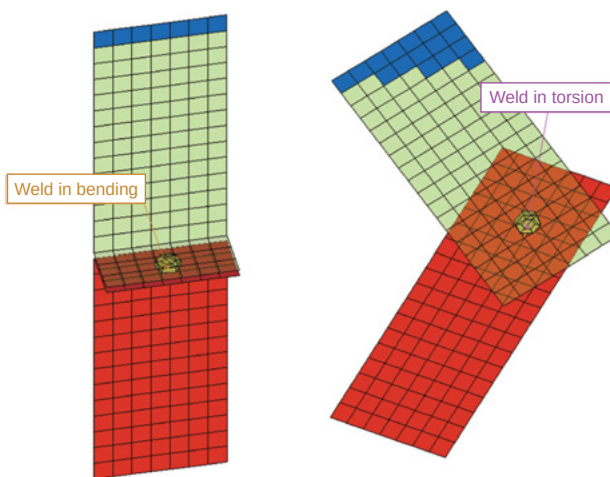


Fig. 14 Bending and torsion parameter check simulations

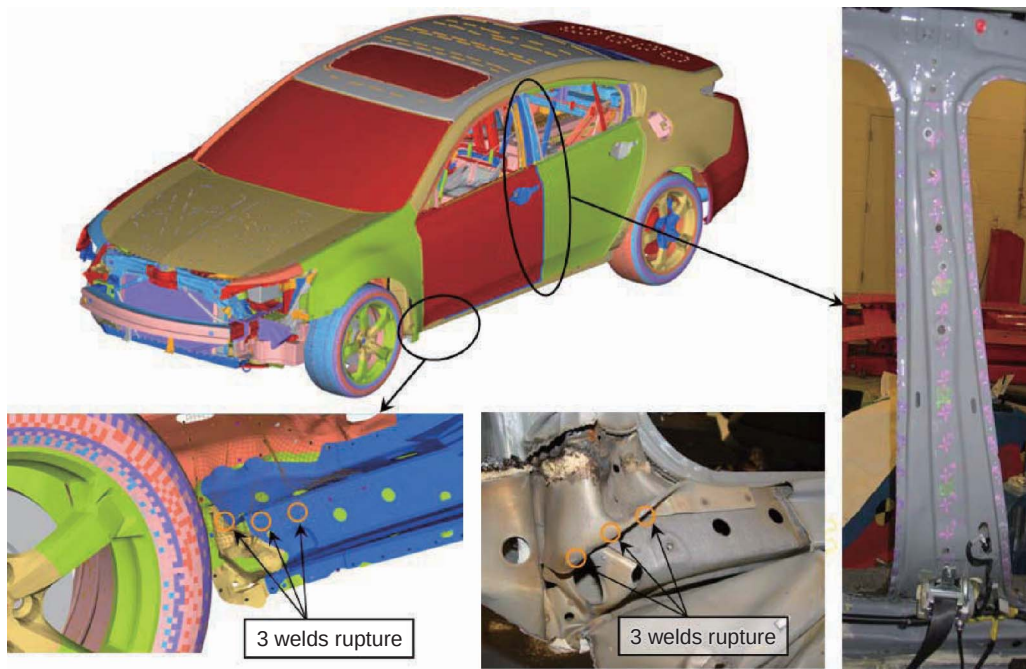


Fig. 15 Comparison of side impact simulation and test results

6. Results in Full Vehicle Crash Simulations

The 8-hex welds and the automatic assigning of the material failure parameters through the preprocessor scripting have shown to be usable in a development engineering setting. An example illustrated here is a side impact crash simulation of a vehicle with several million elements. The automatic assigning script process has been used to set the material failure parameters for all the spot welds in the vehicle, so each has the potential for failure criteria to be achieved depending on the loads seen during the simulation. Furthermore, the impact on analysis run time between beam welds and 8-hex assembly welds with failure criteria was approximately 5% of the overall run time of approximately 24 hours in this case. This lack of time penalty caused by the decrease in mesh size is due to the artificial scaling of the modulus of elasticity in the weld material definition. Although this decrease is significant and smaller than what is being applied to the beams, it does not seem to cause a significant change in the intrusion measurements compared to the beams or compared to the actual test result. The correlation between the welds which ruptured in the actual test compared to the welds which failed in the simulation was good. The simulation showed only 3 welds in the front of the side sill area near the front wheel which ruptured. This comparison is shown in Fig 15.

7. Conclusion

By determining a method to create truly mesh independent weld elements with an appropriate failure criteria method, a practical method to help predict spot weld rupture behavior has been achieved.

As a practical matter, since the precision of the prediction methods is within the variability of the test results, the methods developed here have been proven useful for development work. In addition, other practical considerations such as model fidelity, computational run time, and model set up have only been impacted to a small degree. Therefore these methods should be viewed as viable tools during the development of spot welded structures.

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Skye Malcolm



Emily Nutwell

Development for New Bumper Face Material in Asia-Pacific Region

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Monchai CHAICHANA*

ABSTRACT

This paper describes the development of a low-cost polypropylene (PP) bumper face material at Honda R&D Asia Pacific. This material realizes high impact performance at low temperature, and low specific gravity, which are Honda's worldwide quality requirements.

In order to simultaneously realize good low-temperature properties, high rigidity, and low specific gravity as the conventional bumper characteristics, it was necessary to polymerize block co-polymer polypropylene (BCPP), the main component, and then add expensive rubber material and high crystallization PP during the subsequent compounding process.

In this development, new catalyst properties and profile rationalization were employed, permitting manufacture of a more rigid BCPP that contains a large amount of rubber, which is advantageous for impact performance. By using this BCPP as a base resin, a less expensive bumper face material that had good productivity was developed.

1. Introduction

In recent years, the rising price of crude oil has brought about an increase in the price of naphtha, and because naphtha is a raw material of polypropylene (PP), the price of PP has also risen, as shown in Fig. 1^{(1), (2)}. This situation shows the same tendency as that of steel, copper, and other metallic materials. In order to cope with this increase in the price of materials, it is important for suppliers who handle industrial products to search for, and implement, ways of reducing the cost of materials and the products made from them, from a variety of fronts.

Regarding the recent situation of automobiles in the Asia-Pacific region, sales of small automobiles are tending to spread centered on the Indian market, while in Thailand the eco-car concept has been realized, and it is considered a matter of urgency to concentrate automobile development on these vehicles in order to strengthen the cost competitiveness of small models.

Considering measures for reducing costs, firstly, it is necessary to develop inexpensive and high-grade materials. In other words, it is necessary to search for high performance, highly functional materials that can be substituted for expensive materials or contribute to a reduction in their use. Secondly, it is important to reduce waste as far as possible and strive to reduce investment costs and raise productivity, through means such as

reducing the number of processes, rationalization, integration, increased accuracy and yield, and the like. Thirdly, it is necessary to reduce the cost of raw materials, particularly at Honda's overseas centers, and use specialty products of each region concerned as well as materials assumed to be locally procured, in order to help eliminate transportation costs, customs duty, administration costs, and the like. As a start in this direction, development aimed at realizing an inexpensive cost bumper face material was carried out. A description of this development is set out below.

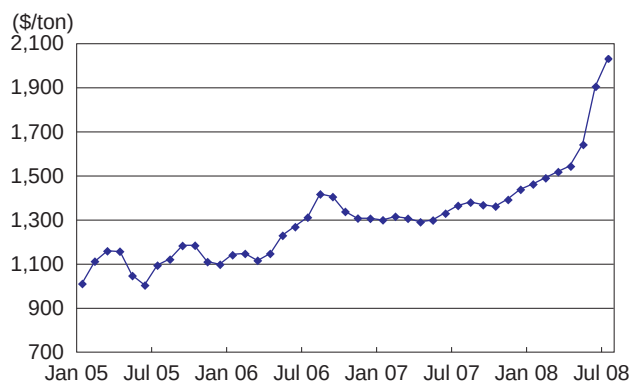


Fig. 1 Polypropylene price (platts)

* HONDA R&D ASIA PACIFIC CO.,LTD.

2. Material Composition, and Proposals Aimed at Cost Reduction

The role of the bumper face material is to retain the shape of the bumper, even when it is thin; to help ensure that the bumper is rigid so that it does not deform when hot; and to help prevent deformation or cracking of the bumper in the event of a minor collision. Needless to say, it is desirable that a bumper face material be light and inexpensive. In order to satisfy these requirements, the bumper face material consists of block co-polymer polypropylene (BCPP) as a base, and high crystallization PP, rubber, and talc, as shown in Table 1. The manufacturing process consists of a process in which PP is polymerized by naphtha, followed by a compounding process in which the physical properties and color are adjusted, as shown in Fig. 2. The resulting material is formed into resin pellets that are finally sent to the injection molding stage⁽³⁾.

In this development, a new material design was considered from the viewpoint of cost reduction. Of the constituents of the bumper face material, the structure of BCPP consists of elastomer “islands” in a “sea” of high crystallization PP, as shown in Fig. 3. Conventionally, the characteristics of the bumper face material are determined with the goal of acquiring rigidity and impact resistance. This BCPP on its own does not have all of the physical properties necessary for achieving this goal, so in the subsequent compounding process, BCPP is blended with rubber, high crystallization PP, and talc. However, because of its low production efficiency, this rubber is a high-cost material compared to BCPP, so rightfully the amount added should be reduced in quantity or its use should be

Table 1 Role of material

Process	Composition	Role
Polymerization process	BCPP	Rigidity, paintability and impact
Compound process	High crystallization PP	Rigidity
	Post-addition elastomer	Impact and paintability
	Talc	Rigidity

eliminated altogether. Likewise, regarding high crystallization PP as well, promoting crystallization to reduce the required amount of high crystallization PP as far as possible should also help reduce the cost of the material. Figure 4 shows a diagrammatic representation of the material constituents based on the above conjecture.

In other words, material design for cost reduction involves approaching the final composition as much as possible during the BCPP polymerization process so as to help eliminate subsequent addition or adjustment.

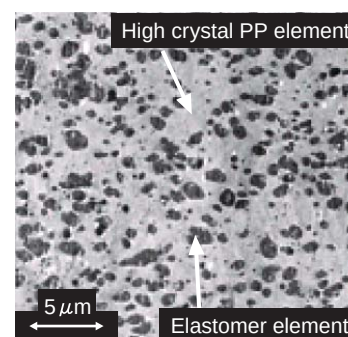


Fig. 3 Structure of BCPP

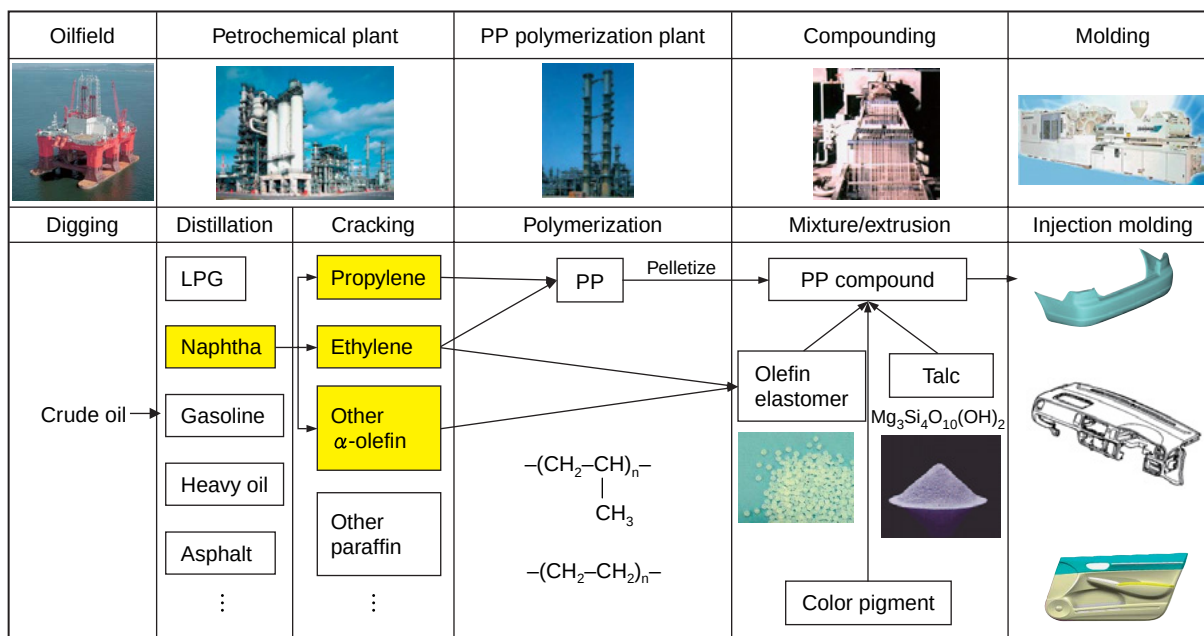


Fig. 2 Manufacturing process

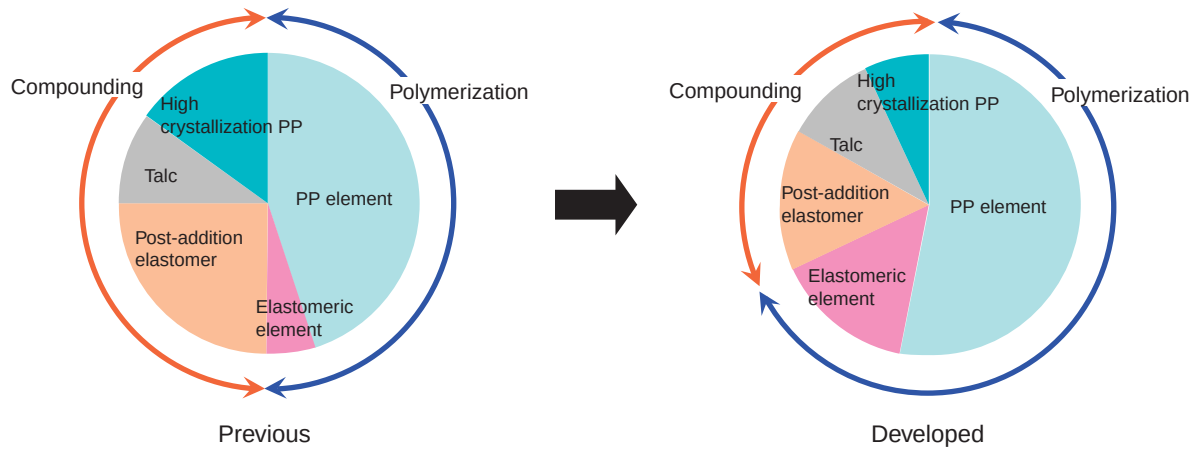


Fig. 4 Idea of bumper material composition design

Concretely, this means promoting high rubber content⁽⁴⁾ and high crystallization of PP. Each of these is described in the following section.

3. Promoting High Rubber Content in BCPP

The key to the promotion of high rubber content is the use of a PP polymerization catalyst. Conventionally, as shown in

Figure 5, BCPP that is extracted from the PP polymerization plant after the reaction that takes place there consists of a dry powder having a grain diameter of about 0.5 mm, which does not produce blocking. If the rubber content is increased in this condition, the amount of rubber existing on the powder grain surface will

increase as shown in Fig. 6, and the powder will become more tacky. As a result, sticking and reduction of fluidity will occur, leading to reduced productivity. In order to overcome this, first of all a study of the use of large grains in order to reduce the relative surface area of the BCPP powder and thus reduce the opportunities for the rubber to come into contact with it was carried out. Because the grain diameter of the powder depends upon the grain diameter of the polymerization catalyst, a large grain diameter catalyst was used, resulting in a powder that had a grain diameter of about 1.5 mm, as shown in Fig. 7⁽³⁾. In addition, a study of the shape of the catalyst was carried out with the goal of reducing the rubber content at the surface of the powder, and it was decided to use a porous catalyst. As a result, a large amount of rubber was

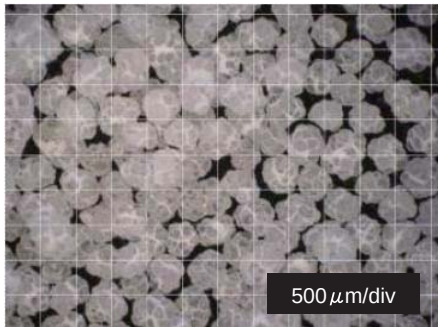


Fig. 5 BCPP powder

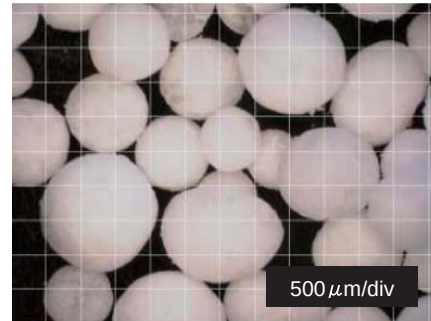


Fig. 7 BCPP powder

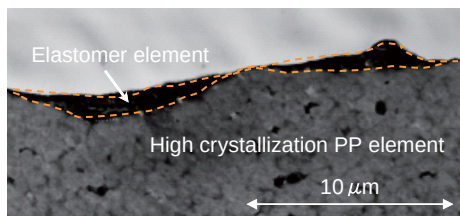


Fig. 6 Surface of BCPP powder

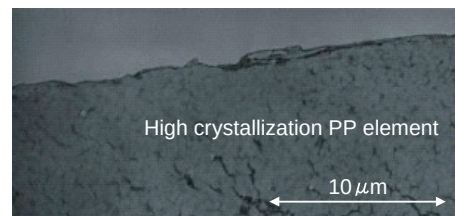


Fig. 8 Surface of BCPP powder (porous catalyst)

taken into the powder, and as shown in Fig. 8, the rubber concentration at the surface could no longer be seen.

Figure 9 shows the relationship between the rubber content and fluidity of the powder for different grain diameters. Fluidity is an index that expresses the degree of tackiness of the rubber component from the velocity at which the PP powder inside the polymerization reaction unit drops. In this development, the target fluidity was 10 or less. When using the conventional powder grain diameter, the rubber content should be no more than about 10% to help ensure adequate productivity. It was found that if the powder grain diameter is increased, the rubber content will rise, and if this measure is combined with the use of a porous catalyst, a rubber content of at least 23% or so can be achieved. However, because there are limits to the porosity and increase in the grain diameter of the catalyst itself, the catalyst was optimized in consideration of realistic production, and the rubber content was set.

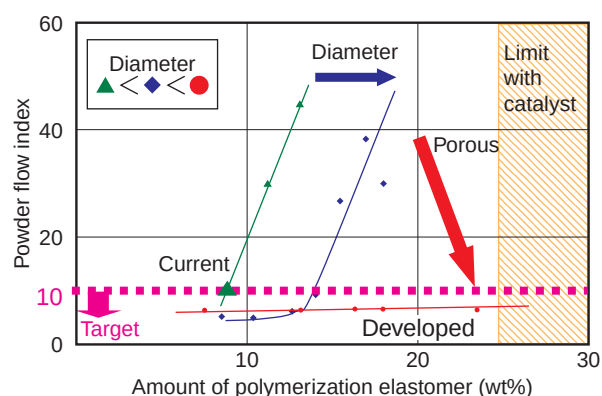


Fig. 9 Effect of BCPP elastomer content

4. Developing a Highly Rigid Bumper Face Material

In the conventional bumper face material, high crystallization PP was post-added during the compounding process in order to enhance the rigidity of BCPP. In the case of the newly developed bumper face material, however, the addition of a nucleator during the compounding process raised the crystallinity and increased rigidity, enabling the amount of post-added high crystallization PP to be reduced, thus contributing to cost reduction. Table 2 shows an example in which particular attention was placed on rigidity and the effectiveness of adding a nucleator to increase rigidity was verified using Homo PP.

5. Performance of the Developed Material

Table 3 shows the physical properties of the developed BCPP. It can be seen that the impact

Table 2 Effect of nucleator

Properties	Unit	Homo PP	Homo PP + nucleator
Specific gravity	—	0.9	0.9
Flexural strength	MPa	54	65
Flexural modulus	MPa	1850	2250
Izod impact (23 °C)	J/m	22	20

Table 3 Physical properties of BCPP

Properties	Unit	Previous	Developed
Specific gravity	—	0.9	0.9
Flexural strength	MPa	35	27
Flexural modulus	MPa	1200	1000
Izod impact (23 °C)	J/m	50	90

performance has been increased by 80% as a result of using high rubber content in a BCPP condition. By using this BCPP, a new bumper face material for use in the Asia-Pacific region having the same performance as that of the existing material and satisfying Honda's global standards was able to be developed.

6. Effectiveness of Using the New Material

As a result of the promotion of high rubber content during polymerization and the use of a nucleator for high rigidity, a cost reduction of 6% compared to that of the previous material was attained.

Previously, in the Asia-Pacific region, dedicated materials limited to that region were used in parallel with material that conformed to Honda's global standards. As a result of the development of the new inexpensive material, however, the previous materials can be integrated into a single material, thus reducing loss due to changeover between materials.

7. Conclusion

A new bumper face material employing a combination of high rubber content BCPP that is based on the use of a new catalyst technology and crystallization technology was developed, and is being used for the first time on the 2008 model year ACCORD produced in Thailand.

- (1) By using a large grain diameter porous catalyst as the polymerization catalyst for BCPP, which is the base polymer, high rubber content (approximately 20%) BCPP was developed, enabling the amount of expensive post-added rubber to be reduced (by approximately 5%).
- (2) At the same time, the degree of crystallization of PP was raised, enabling the amount of post-added high

- crystallization PP to be reduced (by approximately 5%).
- (3) As a result, the newly developed bumper face material had good productivity and was inexpensive to produce (approximately 6% lower than that of the previous material), and moreover, the material has retained Honda's worldwide required quality, low-temperature impact performance, high rigidity, and low specific gravity.
 - (4) Subsequent to its adoption on the 2008 model year ACCORD, the new material has come to be used on all models being produced in Thailand, and in addition is being used throughout the Asia-Pacific region (Taiwan, Malaysia, India, and Pakistan).

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Vaporization Characteristic of High-ethanol Gasoline (E85) and Cold Startability

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ABSTRACT

The research discussed in this paper focused on the vaporization characteristics of high-ethanol gasoline (E85) and cold startability. The cold startability of an E85 (ASTM Class 3) and a gasoline with identical Reid Vapor Pressures (RVP: 71 kPa at 37.8 °C) was tested at -30 °C. When the E85 was injected in the same quantity as that which enabled engine start using gasoline, the engine failed to start. The results of a simulation of the fuel concentration in the air-fuel mixture showed that the mixture concentration formed by the E85 was lower than the mixture concentration formed by the gasoline, and that the E85 mixture did not reach the concentration required for engine start. Meanwhile, the formation of a mixture with a fuel concentration of approximately 2 vol% during cranking was necessary for engine start at low temperatures. Both simulation and test results showed that increasing the fuel injection quantity was an effective method of increasing the mixture concentration. Increasing the RVP of the fuel is another effective method of increasing the mixture concentration. The results of a simulation of the conditions required for low-temperature engine start using E85 at the same injection quantity as gasoline indicated that it would be necessary to increase the RVP of the E85 by approximately 30-35 kPa against that of gasoline.

1. Introduction

Today, concern with energy issues and the reduction of CO₂ emissions are seeing increasing attention directed towards ethanol as an alternative fuel. Ethanol is chiefly produced from plants such as corn, sugar cane, and sugar beets, and can be considered to be carbon neutral, given that the amount of CO₂ released by combustion is equivalent to the amount of CO₂ absorbed by the plants. Ethanol is presently being used as a fuel around the world in a variety of forms. The most common method is to use a mixture of 5-10% ethanol (E5-E10) with gasoline. In Brazil, the use of 100% ethanol (E100) is widespread, and in the EU and U.S., the use of high-alcohol gasoline such as E85, a mixture of 15% gasoline with ethanol, is spreading. When high-ethanol fuels are employed, the reduced vaporization of the ethanol at low temperatures makes cold startability an issue⁽¹⁾⁻⁽⁷⁾. In Brazil, where E100 is in use, a sub-fuel tank containing gasohol, which has good vaporization characteristics, is used at low temperatures in order to increase startability. In Europe

and the U.S., the Reid Vapor Pressure (RVP) of E85 in use is changed seasonally to help ensure cold startability.

It has been reported that there is a close relationship between the RVP of gasoline and cold engine startability⁽⁸⁾. The appropriate mixture concentration is necessary for ignition of the mixture, and the higher the RVP, the easier it becomes to vaporize the fuel and control the air-fuel ratio to the range enabling combustion at low temperatures. E85 displays a different characteristic toward gasoline, and even if its RVP was equivalent to the level enabling engine start using gasoline, the engine would not start. As this indicates, the study of vaporization characteristics is of considerable interest from the perspective of cold startability. However, to date, the majority of studies have focused on modifying engine hardware as a method of increasing cold startability⁽¹⁾⁻⁽⁷⁾, and very few have considered vaporization characteristics⁽⁹⁾. The research discussed in this paper set out to study the vaporization characteristics of E85, and to consider the necessary conditions for cold startability based on the results.

* Automobile R&D Center

Table 1 Specifications for ethanol fuel

Property	US E85			EU E85	
	ASTM D5898			CWA 15293	
Volatility class	1	2	3	A	B
Alcohol [vol%]	79 min	74 min	70 min		
Ethanol [vol%]	–			75 min	70 min
Methanol [vol%]	0.5 max			1 max	
High alcohol (C3-C8) [vol%]	2 max			2 max	
Hydrocarbons [vol%]	17-21	17-26	17-30		
Water [vol%]	1 max			1 max	
RVP [kPa]	38-59	48-65	66-83	35-60	50-100

Table 2 Test fuel characteristics

Property	No. 1	No. 2	No. 3	No. 4	No. 5
Standard	ASTM D5898		CWA 15293	ASTM D4814	
Volatility class	1	3	B	C	D
Ethanol [vol%]	81	71	75	–	
Water [vol%]	1	1	1	1	1
RVP [kPa]	40	71	55	71	83

2. Test Conditions

Table 1 shows the current specifications for E85 fuels in the U.S. and EU. The ethanol concentration and RVP of E85 differ seasonally. The minimum ethanol concentration is 70%; the maximum concentration differs depending on the country, but is approximately 85%. RVP becomes higher as seasonal temperatures become colder, to aid cold startability. As a test of high-ethanol gasoline, the research discussed in this paper used E85-Class 1 (Fuel No. 1) and Class 3 (Fuel No. 2), as specified by the U.S. ASTM's D5788-06⁽¹⁰⁾, and E85-Class B (Fuel No. 3), as specified by the EU's CWA 15293⁽¹¹⁾. For comparison, the project used an ASTM D4814-06⁽¹²⁾ Class C gasoline (Fuel No. 4), modified to give it an RVP value equivalent to U.S. Fuel No. 4, and an ASTM D4814-06 Class D gasoline specified for fall and winter use (Fuel No. 5). Table 2 shows the representative characteristics of these fuels. An 1800 cm³ natural-aspiration 4-cylinder engine was employed to evaluate cold startability. Table 3 shows the specifications of the engine. The temperature for evaluation of cold startability was set at -30 °C.

3. Characteristics of High-ethanol Gasoline

3.1. Calculation of Mixture Concentration and Vapor Pressure

The calculation of the mixture concentration and the vapor pressure of multi-component fuels, such as gasoline and E85, is complex because of the presence of interaction of the chemical species, including the azeotropic phenomenon. Calculations in the project discussed in this paper were conducted using the chemical engineering process simulator CHEMDAD, produced by Chemstations, Inc. The components of each fuel for use in the

Table 3 Test engine specifications

Cylinder configuration	In-line 4-cylinder
Bore × Stroke [mm]	81.0 × 87.3
Displacement [cm ³]	1799
Compression ratio	11.5
Valve train	SOHC
Number of valves	4 valves / cylinder
Fuel injection system	MPI

calculations were qualitatively identified using Gas Chromatography-Mass Spectrometry (GC-MS), and were quantitatively determined using a flame ionization detector (FID).

3.2. Relationship between Saturated Vapor Pressure and Cold Startability

Figure 1 shows the temperature characteristics of the saturated vapor pressures of the test fuels. Actually measured values are plotted as points, while the calculated values produced by the simulation are shown as lines. RVP were measured using a SETAVAP TESTER manufactured by Stanhope-Seta, Ltd. Vapor pressure at low temperature was measured by supplying fuel to the stainless steel vessel shown in Fig. 2, and measuring its internal pressure. E85 (Fuel No. 2) and the gasoline modified to make its RVP identical to that of E85 (Fuel No. 4) display largely identical saturated vapor pressures, even at low temperatures. Figure 3 shows the saturated vapor pressure of hydrocarbons (HCs) by carbon number. The results indicate that it is mainly lighter HCs such as butane (C4) and pentane (C5) that form vapor at low temperatures. If it is the light HC components of the fuels that vaporize at -30 °C, then it can be predicted that fuels would display identical cold startability as long as their saturated vapor pressures were the same. Figure 4 shows

the results of tests of engine start at -30°C using Fuel Nos. 2 and 4. In order to help ensure that only the effects of the fuels themselves were reflected in the results, the fuel injection quantity, ignition timing, and other conditions employed in the tests were set at levels that would enable engine start using Fuel No. 4. Contrary to expectations, there was a difference in startability between the fuels, with the engine failing to start when E85 was used. This result suggests that even when their saturated

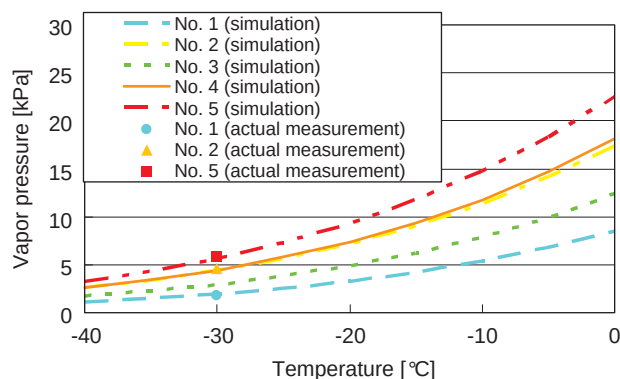


Fig. 1 Vapor pressure of test fuels

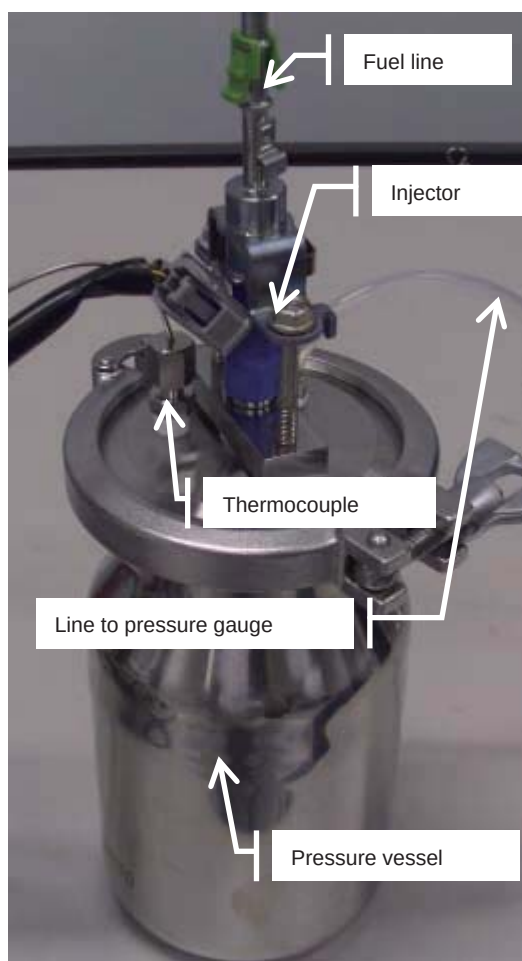


Fig. 2 Vapor pressure measurement device

vapor pressures are identical, the vaporization of E85 differs from that of gasoline.

3.3. Mixture Concentration in Unsaturated Vapor Range

Figure 5 shows the results of simulations of the relationship between fuel injection quantity (amount of fuel injected per 1000 cm^3 of air) and the mixture concentration for Fuel Nos. 2 and 4 at -30°C . When the fuel injection quantity is sufficient, that is, when a saturated vapor exists, both fuels form largely identical mixture concentrations. However, differences occur in the concentrations in ranges in which the fuel injection quantity is insufficient, that is, when the vapor is unsaturated. This is thought to be because the components making up the E85 fuel differ from the components of gasoline. Figure 6 shows the constituent ratio of Fuel Nos. 2 and 4. The ratio of HCs with a carbon number of 5 is higher in Fuel No. 4 (gasoline) than in Fuel No. 2 (E85). It is believed that when the fuel injection quantity is in the unsaturated vapor range, Fuel No. 4, which

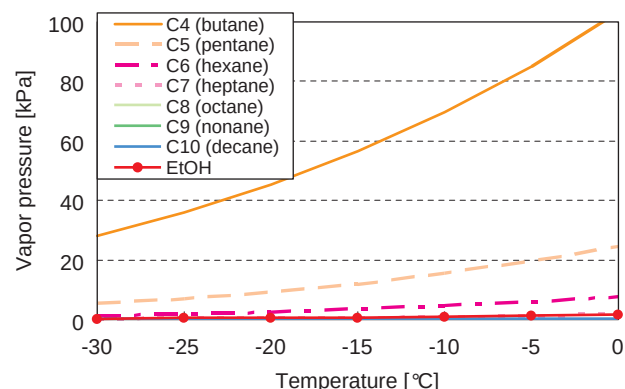
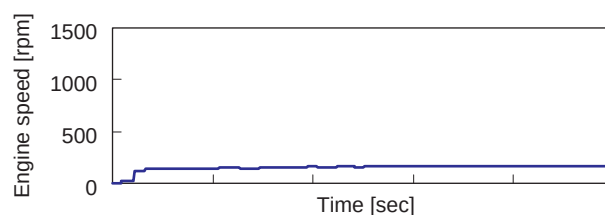
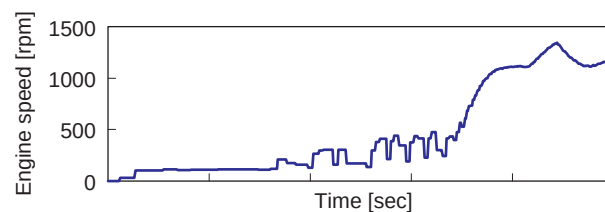


Fig. 3 Vapor pressures of ethanol and hydrocarbons [C4-C10]



(a) Fuel No. 2



(b) Fuel No. 4

Fig. 4 Cold startability with Fuel No. 2 and No. 4 at -30°C

contains a greater quantity of C5 HCs able to vaporize at low temperatures, forms a higher concentration mixture. The fuel injection quantity employed in the engine start tests shown in Fig. 4 was approximately 1 g per 1000 cm³ of air, and the mixture was unsaturated. Under these conditions, the mixture concentration formed by Fuel No. 2 was only approximately 60% that of the Fuel No. 4 mixture. It is believed that this difference in the mixture concentration is related to the difference in cold startability between the fuels.

The results discussed above indicate that even at an identical RVP, E85 vaporizes less than gasoline, and its cold startability is inferior to that of gasoline. The achievement of equivalent cold startability to gasoline would be necessary for the practical use of E85. If the difference in startability was a result of the difference in the mixture concentration formed by the fuels, it was considered that it would be possible to increase the cold startability of E85 by increasing its level of vaporization to equal that of gasoline. Imparting heat energy to the fuel is an effective means of increasing mixture concentration. Increasing the compression ratio or increasing the cranking speed at engine start in order to approach adiabatic compression as closely as possible can be considered methods of imparting heat energy to the fuel. As Fig. 5 shows, increasing the fuel injection quantity is also an effective method of increasing the vaporization of the fuel.

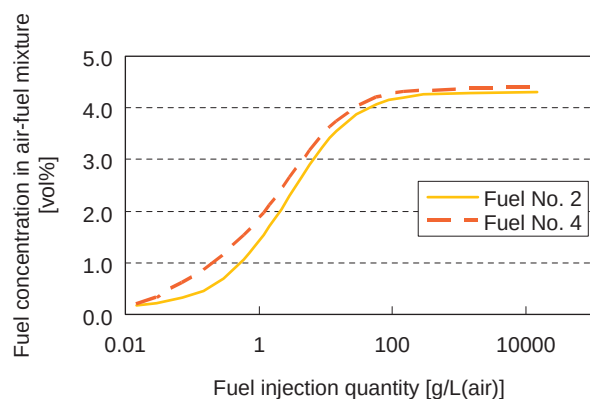


Fig. 5 Relationship between fuel injection quantity and fuel concentration in mixture at -30 °C

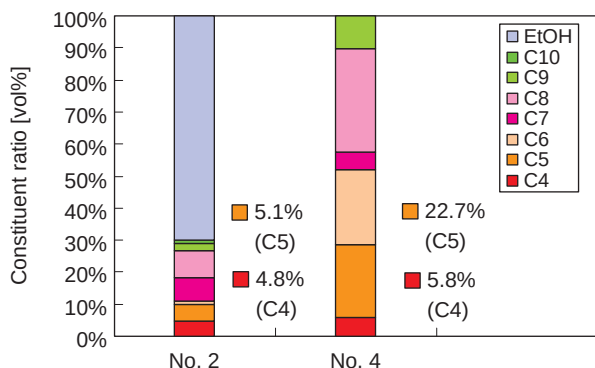


Fig. 6 Constituent ratio of fuel No. 2 and No. 4

Table 4 Combustible concentration of hydrocarbons

	Butane	Pentane
Combustible concentration [vol%]	1.6 – 8.5	1.5 – 7.8

4. Mixture Concentration Required for Engine Start

4.1. Combustible Mixture Concentration

A specific mixture concentration is needed to ignite the air-fuel mixture. For the engine to start, it is necessary for the mixture concentration to enter the range for ignition during the initial engine cranking. The mixture concentration is determined by how much air is mixed with the injected fuel. For this reason, even when there is a small amount of fuel vapor, the amount of air can be adjusted to increase the mixture concentration. However, the pressure of the intake manifold during the initial cranking in actual engine is almost equivalent to the atmospheric pressure, making adjustment of the volume of air a challenge. It was therefore conjectured that the potential for ignition during the initial cranking could be estimated from the mixture concentration formed under atmospheric pressure. Table 4 shows the combustible concentrations⁽¹³⁾ for butane (C4) and pentane (C5), the main fuel components of the mixture at low temperatures. While there will be slight changes in the fuel constituents of the mixture due to the constituents of the fuel and the temperature, because butane and pentane are the main components, estimation of the minimum combustible concentration at below freezing point based on Le Chatelier's principle⁽¹³⁾ gives a value of approximately 1.5 vol% for both gasoline and E85. This means that at low temperatures, unless the mixture concentration is at least 1.5 vol% during initial cranking for both gasoline and E85, the engine will fail to start.

4.2. Mixture Concentration Enabling Engine Start at Low Temperatures

The minimum fuel concentration for ignition of the air-fuel mixture was estimated in Section 4.1, but because in

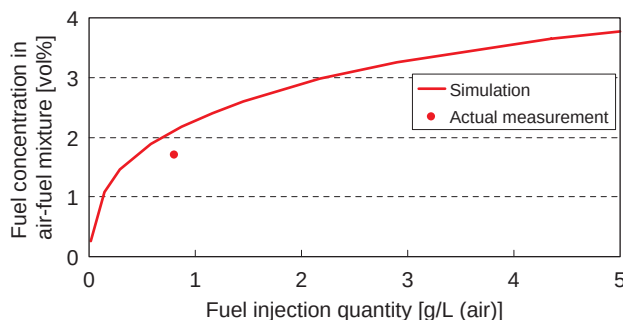


Fig. 7 Relationship between quantity of fuel injection and fuel concentration in combustion chamber for fuel No. 5 at -30 °C

an actual engine only sufficient energy is needed in order to push down the pistons and turn the crankshaft, there are cases in which the formation of a combustible mixture does not provide the condition for engine start, for example when ignition occurs but the engine still fails to start. The mixture concentration that would be necessary during cranking for engine start was therefore studied in the test engine. The fuel used for evaluation was the gasoline with the highest RVP (Fuel No. 5), because it would be possible to control the mixture concentration. A study of the minimum fuel injection quantity required for engine start at -30°C was conducted. The results showed that the minimum required fuel injection quantity was approximately 0.8 g for 1000 cm^3 of air. Figure 7 shows the relationship between the fuel injection quantity for Fuel No. 5 and the mixture concentration at -30°C , as determined by a simulation. The total pressure of the mixture was 1 atm (101.325 kPa). The results show that the mixture concentration reaches approximately 2 vol% at an injection quantity of 0.8 g of gasoline for 1000 cm^3 of air. Measurement of the concentration of a mixture sampled under identical conditions at compression top dead center produced a value of 1.7 vol%. Considering error in the analysis of the mixture, this is almost identical to the simulation result, and it was therefore conjectured that a mixture concentration of at least 2 vol% would be needed to enable the test engine to be started at low temperatures.

5. Effects of Compression Ratio and Cranking Speed

This section will consider the effectiveness of increasing the compression ratio and increasing the cranking speed as methods of increasing vaporization. The temperature changes occurring as a result of compression occur during such short periods that issues of responsiveness make their direct measurement via a thermocouple or other such device challenging. Fuel is made up of multiple components, and this project therefore focused on the changes occurring in the components of the vapor formed during heating. Figure 8 shows simulation results for the constituent ratio of fuel

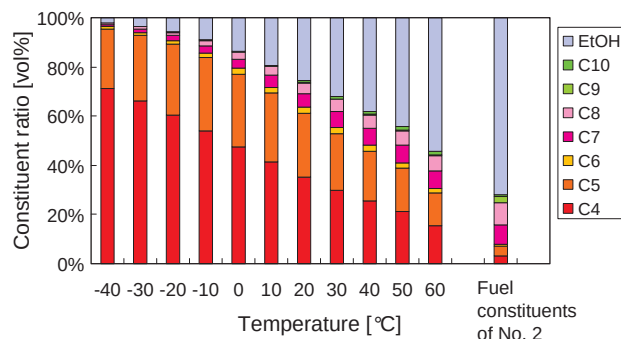


Fig. 8 Temperature dependence of mixture constituents for fuel No. 2

in a mixture of 2.4 g U.S. E85 Class 3 fuel (Fuel No. 2) and 1000 cm^3 of air at a total pressure of 101.325 kPa. The fraction of high-carbon-number HCs and ethanol increases as temperature increases. It was considered that this characteristic could be used to enable estimation of temperature by analyzing the composition of samples of the air-fuel mixture taken directly from the combustion chamber. Figure 9 shows a schematic diagram of the gas

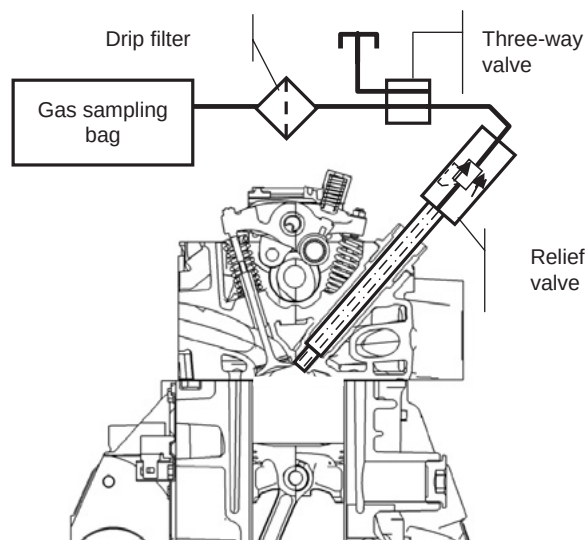


Fig. 9 Gas sampling device

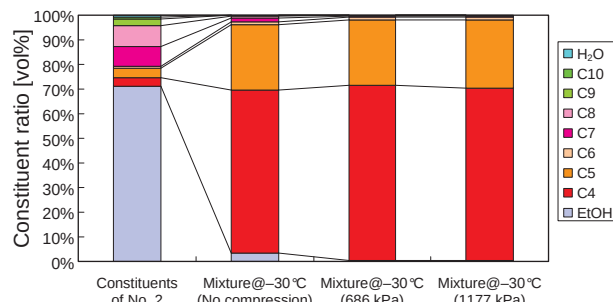


Fig. 10 Comparison of mixture constituents at 668 kPa and 1177 kPa at -30°C

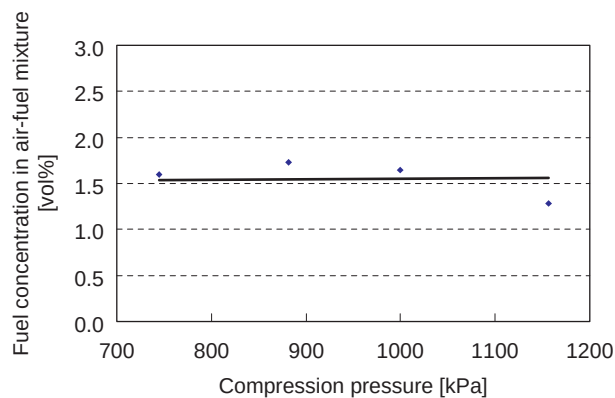


Fig. 11 Relationship between compression pressure and fuel concentration in combustion chamber at -30°C

sampling method. A pipe inserted in the spark plug hole was connected to the pressure relief valve, enabling gas in the combustion to be sampled at any pressure. The sampled gas was collected in an air bag, and its component ratios were analyzed using GC-MS and FID.

5.1. Effect of Compression Ratio

Because gas is sampled from the combustion chamber via the pressure relief valve, it can be sampled at any pressure. Changing the pressure at which the gas is sampled could be considered to be equivalent to changing the compression ratio. The effect of the compression ratio was therefore determined by changing the pressure of the gas sample. The fuel injection quantity was 2.4 g per 450 cm³ cylinder. Figure 10 shows the constituent ratio of mixtures sampled with relief pressures set at 686 kPa and 1177 kPa. The fractions for both gases are almost identical, and are similar to the fractions obtained from the simulation at -30 °C. Figure 11 shows the concentrations of mixtures sampled at various relief pressures. There is almost no change in mixture concentration in relation to relief pressure (that is, compression pressure). These results suggest that almost no increase in vaporization results from a change in the compression pressure. If the pressure prior to compression is 1 atm, and the polytropic exponent is 1.3, then a difference of around 2.3 will be produced in the

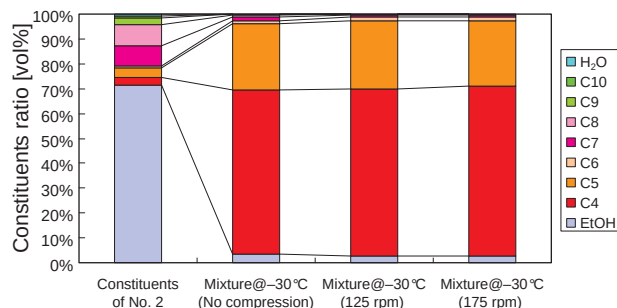


Fig. 12 Comparison of mixture constituents at 125 rpm and 175 rpm at -30 °C

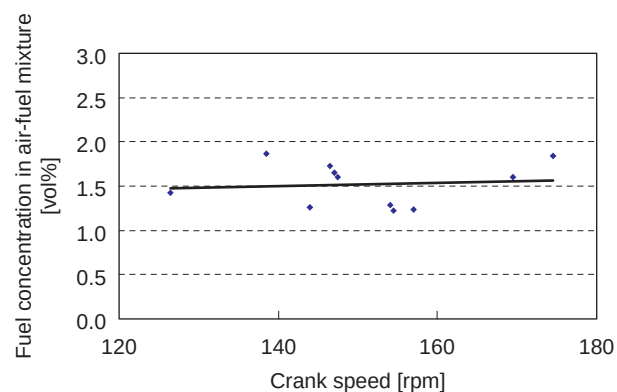


Fig. 13 Relationship between crank speed and fuel concentration in combustion chamber at -30 °C

compression ratio, which should result in a difference of 50 °C in the temperature at compression top dead center. The reason for the minimal effect of increasing the compression pressure can be considered as follows. Under low temperature conditions, the engine itself is at a low temperature, and heat generated by compression flows away to the engine without generating a substantial increase in temperature; or, even if there is an increase in temperature, because the thermal conductivity of the fuel droplets is low and only an instant is available for the droplets to absorb compression heat, they do not increase in temperature and fail to vaporize.

5.2. Effect of Cranking Speed

Figure 12 shows the constituent ratio of mixtures sampled at cranking speeds of 125 rpm and 175 rpm. Fuel and fuel injection quantity were identical to those employed in the study of the effect of the compression ratio. As in the case of the previous study, when cranking speed is varied, fractions for both mixtures are almost identical, and are similar to the fractions obtained from the simulation at -30 °C. Figure 13 shows the mixture concentration sampled at various cranking speeds, as in the previous tests. No increase in mixture concentration is observed as the cranking speed is varied. These results indicate that at low temperatures, increasing the compression ratio or cranking speed would have little effect in terms of increasing cold startability.

6. Effect of Increasing Fuel Injection Quantity

The results discussed in Chapter 5 showed that the fuel compositions of air-fuel mixtures sampled following compression were largely identical to those of mixtures sampled prior to compression. This suggests that the mixture concentration before the mixture enters the cylinder is important for cold startability. As Fig. 5 shows, increasing the fuel injection quantity is an effective method of increasing the mixture concentration before the mixture enters the cylinder. Figure 14 shows simulation

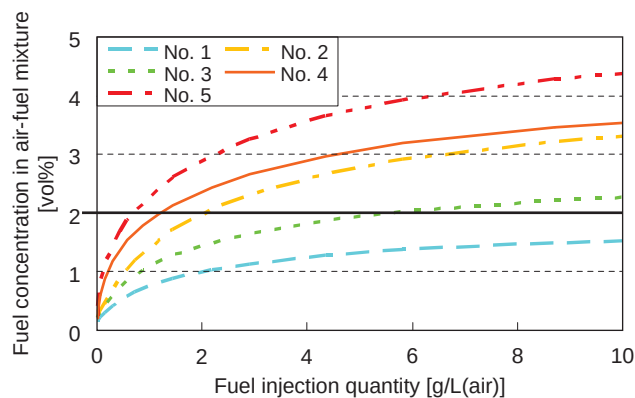


Fig. 14 Relationship between fuel injection quantity and simulated fuel concentration in mixture for test fuels at -30 °C

results for the relationship between fuel injection quantity and mixture concentration at -30°C . The results show that increasing the fuel injection quantity enables cold season E85 (Fuel Nos. 2 and 3) to reach a mixture concentration of 2 vol%. As indicated in Section 3.2, when the fuel injection quantity was not adjusted, the test engine failed to start at -30°C when Fuel No. 2 was used. It was concluded in Section 4.2 that a mixture concentration of approximately 2.0 vol% would be necessary for engine start, but as Fig. 14 shows, increasing the injection amount of Fuel No. 2 would potentially enable engine start at -30°C . By contrast, even if the injection quantity of Fuel No. 1 was increased, the mixture would not reach a concentration of 2 vol%, and there would be little possibility of engine start at -30°C . This is because the mixture concentration as determined from the saturated vapor pressure at -30°C , 1.85 vol%, is the highest concentration that can be formed by this fuel. A simulation based on the maximum injection quantity possible in the test engine indicated that under these conditions Fuel No. 1 would reach a mixture concentration of approximately 1.2 vol%. Based on simulation results, therefore, increasing the injection amount of Fuel No. 1 would not enable engine start at -30°C using this fuel.

7. Verification in Actual Vehicle

Figure 15 shows the results of tests of cold startability

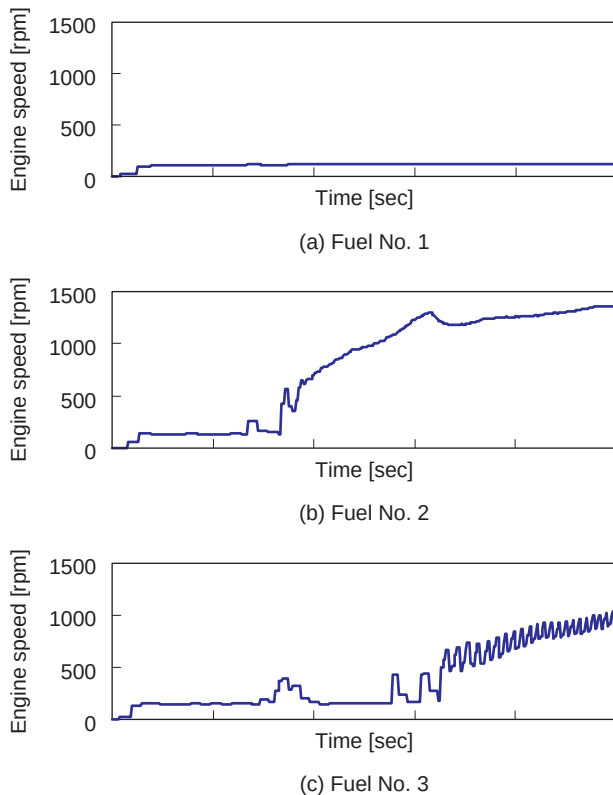


Fig. 15 Cold startability with optimized fuel injection quantity at -30°C

at -30°C using E85 fuels (Fuel Nos. 1, 2, and 3) at the fuel injection quantities that the simulation had indicated would produce a mixture concentration of 2 vol%. The simulation had indicated that Fuel No. 1 would not reach a concentration of 2 vol%, and therefore in this case the fuel was supplied in the highest quantity possible to the test engine. The engine took longer to start the lower the RVP of the fuel, but adjustment of the injection quantity of Fuel Nos. 2 and 3 enabled engine start at -30°C . The engine failed to start when Fuel No. 1, which the simulation had indicated could not achieve a combustible mixture concentration, was employed. The results indicate that increasing the fuel injection quantity is effective in increasing the mixture concentration, but that it is only effective for fuels that are able to reach the necessary engine start concentration when the injection quantity is increased.

8. Limit Temperatures for Engine Start using E85 Fuels

Figure 16 shows simulation results for mixture concentrations at -30°C for E85 fuels with various RVPs. The fuel injection quantity was the maximum amount that could be supplied to the test engine, 4.8 g

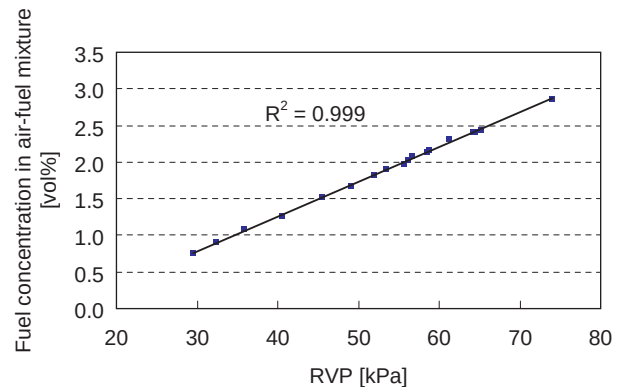


Fig. 16 Relationship between RVP and fuel concentration in mixture at maximum fuel injection quantity in test engine at -30°C

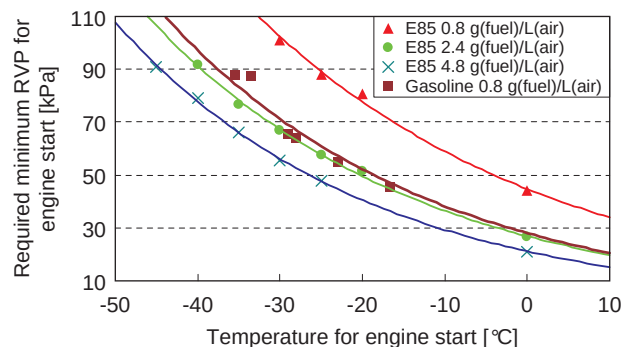


Fig. 17 Estimation of required RVP for cold engine start

per 1000 cm³ of air. While there is some variation due to the fuel composition, there is an almost exact correlation between RVP and the mixture concentration. At an RVP of 55 kPa, E85 reaches a mixture concentration of 2 vol%. It was therefore predicted that the test engine would start at -30 °C using E85 with an RVP of 55 kPa or more at the maximum injection quantity. The RVPs at which the engine would start at various temperatures and injection quantities were also estimated. The results are shown in Fig. 17. Figures for gasoline are also included for purposes of comparison. The results indicate that the minimum temperature for engine start using E85 is dependent on RVP and fuel injection quantity, as it is in the case of gasoline. Comparing the fuels at the same injection quantity (0.8 g per 1000 cm³ of air), it would be necessary to set the RVP of the E85 30-35 kPa higher than the gasoline in order to achieve engine start. If the injection quantity was to be increased, it would be necessary to increase the quantity of the E85 by three times when ASTM Class 3 fuel is used, and by six times when CWA Class B fuel is used, against the gasoline.

9. Discussion

The results of the project discussed here show that E85 displays lower vaporization characteristics, and thus inferior cold startability when compared to the gasoline with an identical RVP. This is because the fuel concentration in the mixture formed by the E85 is lower than that of the mixture formed by the gasoline, even when the RVP of the fuels is identical. At low temperatures, light HCs with carbon numbers of C4 and C5 are the main HC components of the air-fuel mixture, and it is therefore important to increase the concentration of these HCs. The results of this project have shown that increasing the fuel injection quantity is an effective means of doing so. This is because increasing the supply of fuel increases the absolute volume of C4 and C5 HCs. However, there are drawbacks to increasing the fuel injection quantity, such as the fact that fuel efficiency declines when the fuel injection quantity is increased⁽¹⁴⁾. In addition, unless the fuel has a specific vapor pressure, increasing the fuel injection quantity will not have any effect. An alternative method is to increase the RVP of the fuel. This method is able to increase the mixture concentration, and is therefore a certain method of increasing cold startability. However, it would be necessary to increase the RVP of the E85 by 30-35 kPa against that of gasoline. If the RVP is increased to some extent, it may result in vapor lock. Given this, the achievement of a balance between the fuel injection quantity and fuel properties is important from the perspective of cold startability, and it will be necessary to sufficiently research these parameters in order to enable practical use of high-ethanol gasoline.

10. Conclusion

The research discussed in this paper studied the relationship between the vaporization characteristics and the cold startability of high-ethanol gasoline. The following results were obtained:

- (1) When E85 and gasoline with identical RVP values were compared, the E85 displayed less vaporization than the gasoline, and its cold startability was inferior to that of the gasoline.
- (2) When the fuel concentration in the E85 air-fuel mixture was made equivalent to that of the gasoline mixture, the E85 displayed equivalent startability to gasoline. A mixture concentration of approximately 2.0 vol% would be necessary for engine start at low temperatures.
- (3) The main fuel components of the mixture at low temperatures are HCs with carbon numbers of C4 and C5.
- (4) Increasing the fuel injection quantity and increasing the RVP of the fuel are both effective methods of increasing the mixture concentration.
- (5) In order to achieve equivalent startability to gasoline, it would be necessary to increase the quantity of the E85 by three times when ASTM Class 3 fuel is used, and by six times when CWA Class B fuel is used, against the gasoline.

11. Afterword

The current U.S. ASTM D5788-06 and EU E85 specification CWA-15293 stipulate minimum RVPs for cold season E85 of 66 kPa and 50 kPa, respectively. Based on these values, for cold startability at -30 °C and below, the respective injection quantities required for the fuels would be, respectively, three times and six times that of gasoline (ASTM D4814-06 Class D, RVP83), as discussed in Chapter 10. Revision of the RVP specifications would therefore be desirable from the perspective of realizing more efficient cold startability.

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Effect of Bio Diesel Fuel on Sludge Prevention Performance of Engine Oil

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ABSTRACT

The research discussed in this paper clarified the effect of biodiesel fuel (BDF) on two effective sludge generation indexes of engine oil, the base number (BN) and the oxidation induction time (OIT). The results of laboratory tests of oil deterioration using an engine oil mixed with BDF showed that BDF, which contains a large number of unsaturated bonds, accelerates sludge generation of engine oil.

1. Introduction

The use of bio diesel fuel (BDF) as an alternative fuel that effectively addresses crude oil depletion and CO₂ reduction is expanding around the world. BDF is a mixture of diesel oil with fatty acid methyl esters (FAME) obtained by transesterification of animal and vegetable oils with methanol. The use of BDF has already advanced in Europe and the United States. The main FAME used in Europe is rapeseed methyl ester (RME), and in the U.S. it is soybean methyl ester (SME).

Although BDF is useful, there are many concerns about its fuel stability. The acids generated by BDF and the oxidation products from BDF deposits have been reported to affect fuel systems by clogging fuel filters, forming injector deposits, causing corrosion of materials, and the like^{(1), (2)}.

As controls on exhaust gases have become more stringent, it has become common to install diesel particulate filters (DPF) in diesel vehicles. DPF-equipped vehicles perform a main fuel injection, followed by a post-injection, to regenerate the DPF. As a result of this multi-stage injection, fuel tends to become intermixed with the engine oil. Consequently, the engine oil is also affected, and it is important to ascertain that effect in order to determine when the engine oil should be changed.

The present research investigated the influence of BDF on engine oil properties mainly in terms of the effect on sludge prevention performance.

2. Engine Oil Sludge Generation Indexes

It was previously reported that the remaining amounts of anti-oxidation performance and cleaning performance in engine oil provide valid sludge generation indexes for engine oil^{(3), (4)}.

It is known that the amount of remaining cleaning performance can be evaluated by the engine oil base number (BN)⁽⁵⁾. Figure 1 illustrates the relationship between BN and sludge with photographs of piston interiors showing the results from tests of engine oil deterioration in actual engines.

The fall of the BN results when the detergent used as an engine oil additive is neutralized and consumed by heat and the nitric acid that is generated by the NO_x gas in the blow-by gas. It was confirmed that, as shown in Fig. 1, a fall in the BN is accompanied by a reduction in cleaning performance and the formation and progressive accumulation of sludge. Maintaining the BN is therefore crucial in limiting the formation of sludge.

Meanwhile, it is known that the remaining amount of anti-oxidation performance can be evaluated from the oxidation induction time (OIT) using a high pressure differential scanning calorimeter (HPDSC)⁽⁶⁾. Figure 2 shows the relationship between the OIT and the generated content of nPB insolubles⁽⁷⁾ in sludge-producing and non-producing vehicles.

Reduction in the OIT results when the anti-oxidant used as an additive in engine oil undergoes a radical

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reaction with heat, NO_x gas, and unburned fuel. As shown in Fig. 2, sludge is not generated when there is remaining OIT. Maintaining the OIT is therefore important in preventing the generation of sludge.

It is apparent from the above that BN and OIT are valid indexes of sludge generation in engine oil.

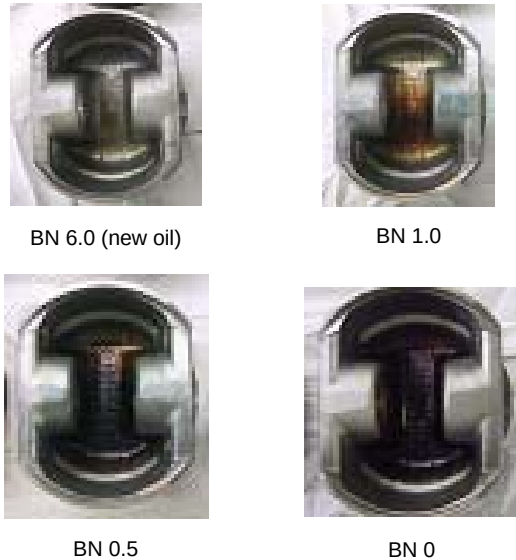


Fig. 1 Relationship between BN and deposits

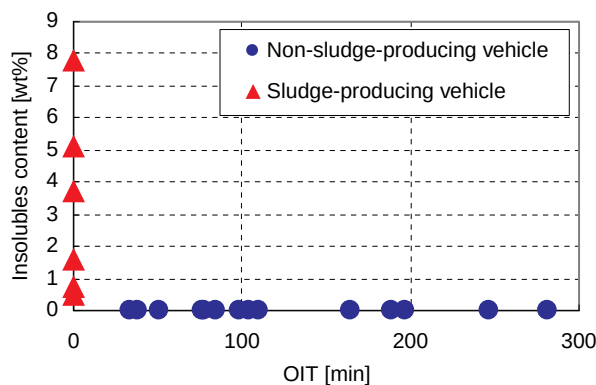


Fig. 2 Relationship between OIT and nPB insolubles

3. Effect on Engine Oil Sludge Prevention Performance

A NO_x bubbling test was conducted in order to study the effects on engine oil sludge prevention in BDF in a diluted state. Figure 3 shows a schematic diagram of the device⁽³⁾. The NO_x bubbling test is a laboratory test that simulates the deterioration of engine oil in the crankcase as a result of heat and NO_x being blown through it. Honda genuine oil combined with the types of BDF

Table 1 Properties of fuel for NO_x bubbling test

Items	Diesel	RME10	RME100	SME20	SME100
FAME in diesel (%)	0	10	100	20	100
Kinematic viscosity (40°C, mm ² /s)	2.721	2.792	4.452	2.833	4.137
Sulfur (ppm)	3	3	<3	3	<3
Density (15°C, mgKOH/g)	0.8356	0.8322	0.8831	0.8377	0.8831
Acid number (mgKOH/g)	0.01	0.03	0.1	0.06	0.17
Oxidation stability (hr) *	24.0<	18.6	7.8	2.9	1.1

* Rancimat

shown in Table 1 in a 10 wt% mixture of each was used for the test oil. The test measured the rate of reduction in the BN and OIT over time. The rate of reduction in the BN and OIT are shown in Figs. 4 and 5, respectively.

The BN and OIT reduction rates with the RME10 mix showed no difference from the mixture containing diesel oil. With the BDF mix, however, which has greater FAME content, the reduction tended to accelerate, indicating a marked effect. When BDF of equal concentrations was used, mixed with RME100 and with SME100, the rate of reduction was greater when mixed with SME100. This indicates that differences in the rate of reduction can occur with different BDF base materials.

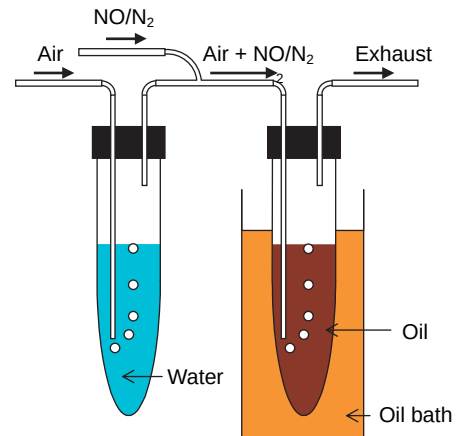


Fig. 3 NO_x bubbling device

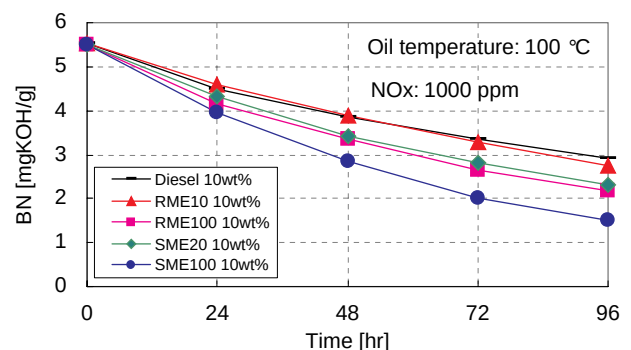


Fig. 4 BN reduction rate of BDF/oil mixture

determine whether the line of superposition of the reduction effects of the various FAMES coincided with the mode of reduction in actual mixtures with BDF. Table 2 shows the ratios of the individual FAMES that are mixed in the oil when the BDF in Table 1 is a 10 wt% mixture with engine oil. As one example, the BN reduction effect in a 10 wt% mixture with methyl linolenate is shown in Fig. 7, and the reduction effect of a 5 wt% mixture can be approximated by the roughly intermediate curve shown in Fig. 9. A similar method was used to create the estimated reduction curves for the corresponding amounts of each FAME present in the oil for 10 wt% mixtures of RME100 and SME100. The results for BN and OIT when those curves are superposed are shown in Figs. 10 and 11. There was a reliable correspondence between the superposition lines and the modes of RME100 and SME100 reduction. Therefore, the difference in the rate of reduction attributable to the difference in base materials is due to the percentage composition of FAMES with different numbers of unsaturated bonds.

4.3. Detailed Analysis of FAME Oxidation Products

Conceivable reduction factors that accelerate the lowering of BN and OIT include the involvement of acids and substances with radical properties. With the NO_x bubbling test, however, it is challenging to analyze and identify the reduction factors in the deteriorated oil, which is a complex chemical mixture. An attempt was therefore

Table 2 FAME rate at BDF 10 wt% mixture in engine oil

Items	Diesel	RME10	RME100	SME20	SME100
Methyl linolenate (wt%)	0.00	0.09	6.16	0.15	2.37
Methyl linoleate (wt%)	0.00	0.20	1.98	1.07	5.37
Methyl oleate (wt%)	0.00	0.62	0.85	0.47	0.75

made to estimate the BN and OIT reduction factors by forcing oxidation of the individual FAME constituents at a constant temperature under blown air, and observing the resulting products after the test.

Figures 12 and 13 show the results of analysis of low molecular weight products by means of ion chromatography (IC) and GC-MS. The IC analysis results indicate the production of lower fatty acids. However, acids were formed in the largest quantities with methyl oleate, which is counter to the effect on BN reduction shown in Fig. 7. This is probably because the rate of acid formation in engine oil mixtures is different from that in FAMES alone. The results of GC-MS analysis confirm the production of several aldehydes.

Figure 14 shows the results of analysis of high molecular weight products by means of gel permeation chromatography (GPC). The molecular weight distribution results after testing confirm the production of higher molecular weight substance than FAME. High molecular weight products occurred in the largest quantities with

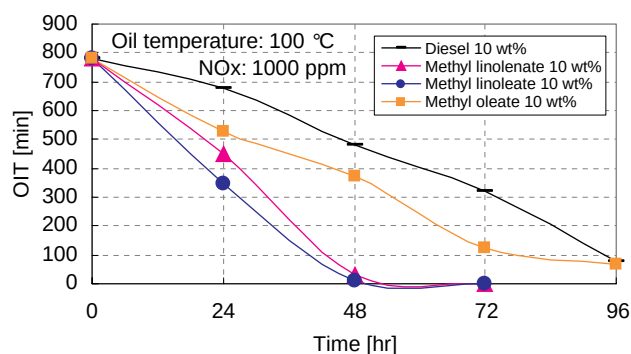


Fig. 8 OIT reduction rate of FAME/oil mixture

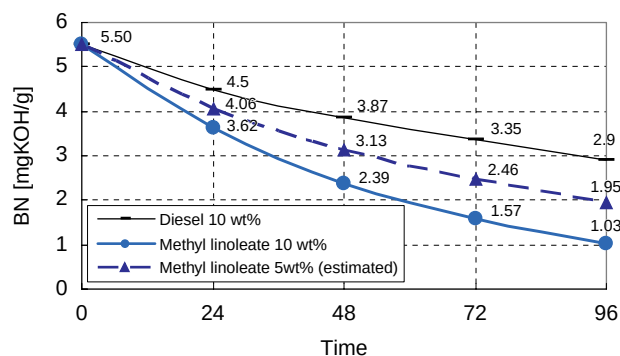


Fig. 9 Example of estimated reduction curve

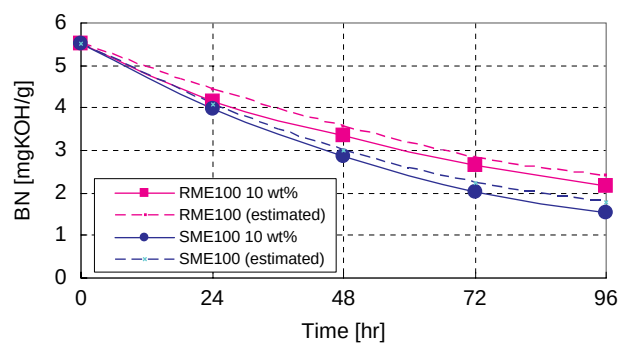


Fig. 10 Estimated BN reduction curve

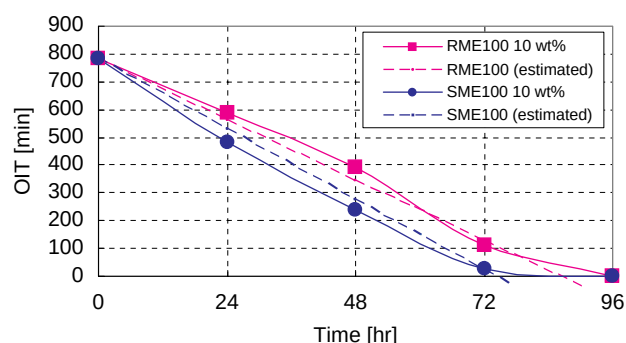


Fig. 11 Estimated OIT reduction curve

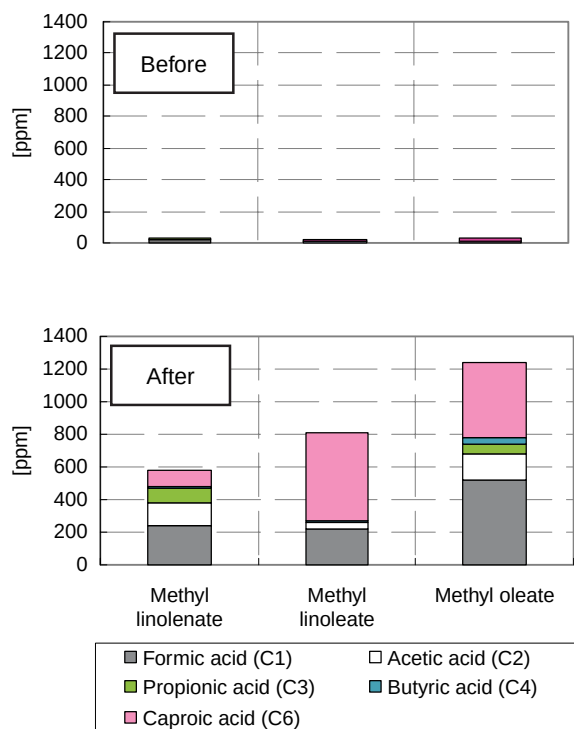


Fig. 12 Results of IC analysis after FAME oxidation test

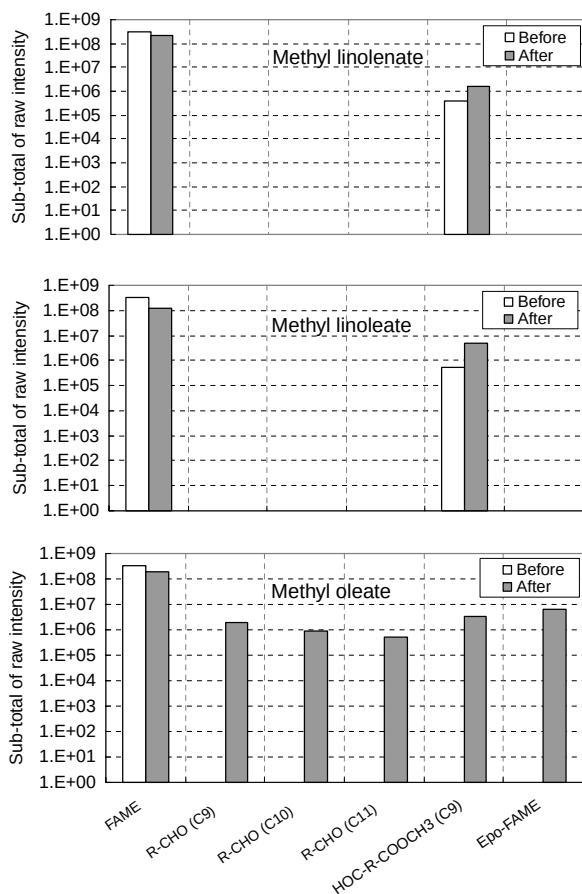


Fig. 13 Results of GC-MS analysis after FAME oxidation test

methyl linoleate. These high molecular weight products were then further separated and refined using high performance liquid chromatography (HPLC) and analyzed by means of atmospheric pressure chemical ionization mass spectrometry (APCI-MS). The results confirmed their mass to be equivalent to a dyad in each FAME. It is conjectured that these high molecular weight products are dimers resulting from polymerization reactions.

4.4. FAME Oxide Production Mechanism

It is known that FAME chemical reactions are generally started by methylene hydrogen elimination reactions near unsaturated bonds. Methylene hydrogens

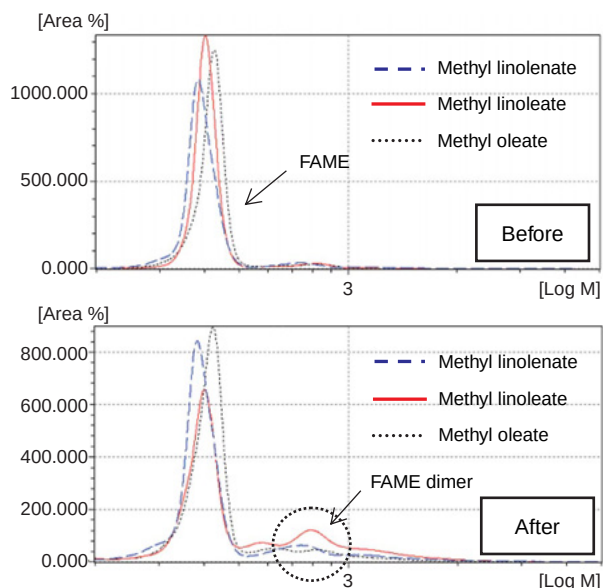
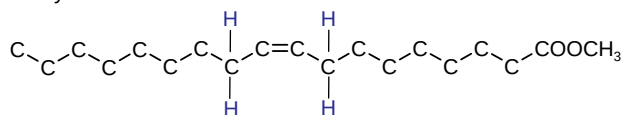
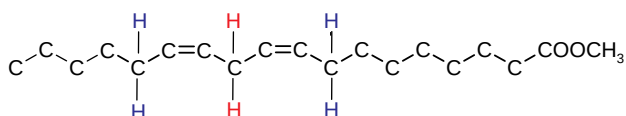


Fig. 14 Results of GPC analysis after FAME oxidation test

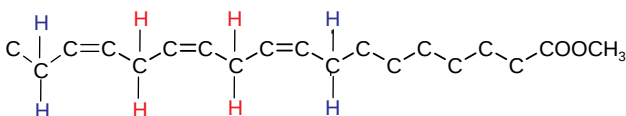
Methyl oleate



Methyl linoleate



Methyl linolenate



H: Allyl H: Bis-allyl

Fig. 15 Model of chemical structure of FAME

that bond to carbon atoms near unsaturated bonds, as shown in Fig. 15, are termed allyl hydrogens, while methylene hydrogens that bond to carbon atoms between unsaturated bonds are termed bis-allyl hydrogens. These hydrogens produce highly chemically reactive free radicals on carbon by elimination reactions. It has also been reported that the bis-allyl hydrogens undergo elimination reactions at a rate 100 times greater than the allyl hydrogens⁽⁸⁾. Methyl linolenate and methyl linoleate with numerous unsaturated bonds that can have bis-allyl hydrogens are highly chemically reactive.

Figure 16 shows the oxide production mechanism in FAMES that are the main constituent of RME and SME. FAMES produce free radical intermediates through hydrogen elimination reactions. Free radical intermediates proceed selectively to oxidation reactions or polymerization reactions. The oxidation reactions result in the carbon chains splitting and producing lower fatty acids. The polymerization reactions also result in the dyads acting as cross-links to produce dimers.

In the FAME oxidation test in section 4.3, methyl oleate generated the greatest quantities of acids, a result that is counter to the influence on BN reduction. The highly chemically reactive methyl linolenate and methyl

linoleate should produce greater quantities of acids. They formed acids in small quantities, however, and formed dimers in large quantities. It is conjectured that FAME in mixture with engine oil is less likely to collide with the

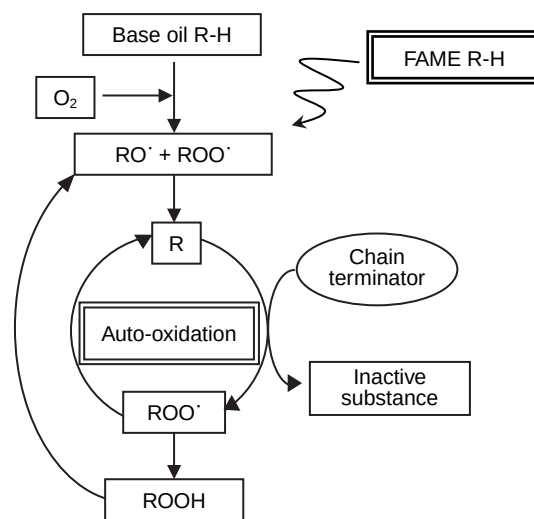


Fig. 17 Mechanism of engine oil auto-oxidation and terminal reaction with FAME mixture

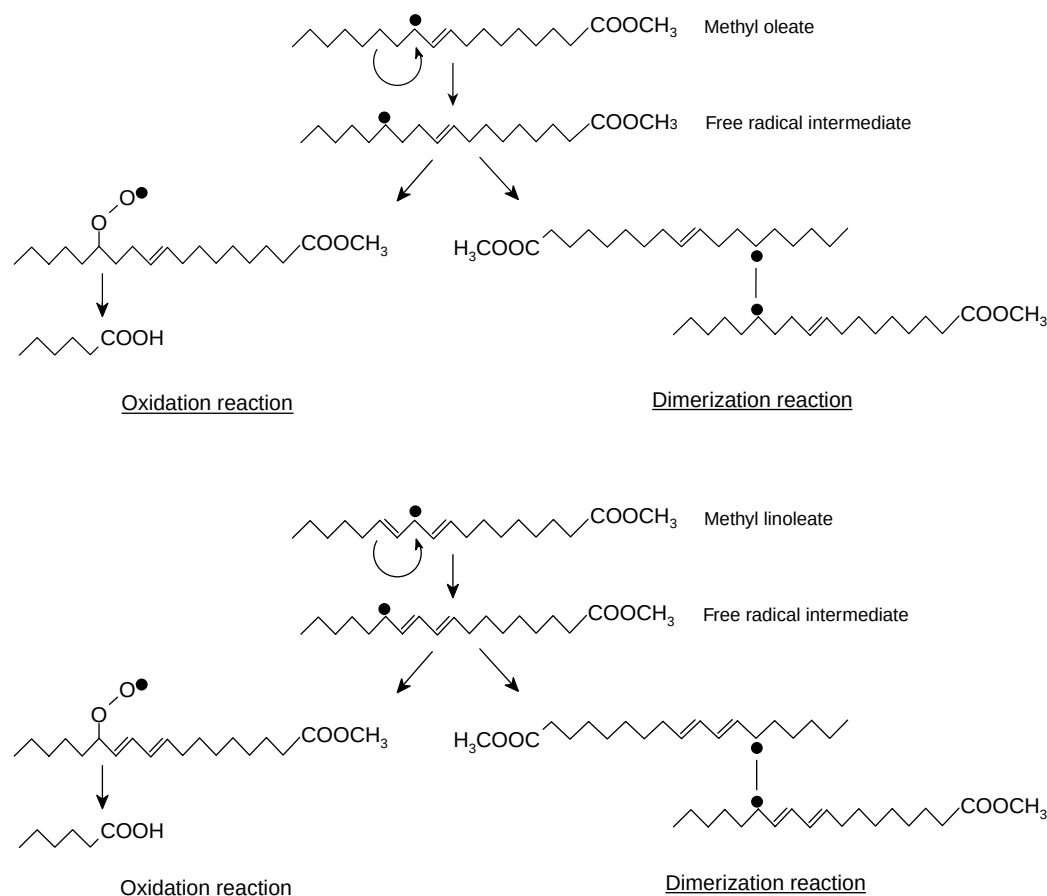


Fig. 16 Mechanism of generation of FAME oxidation compound

counterpart molecules required for a polymerization reaction since the engine oil would intervene, so that the oxidation reaction, rather than the polymerization reaction, is more likely to be selected.

4.5. BN and OIT Reduction Factors due to FAME

The above investigation suggests that the BN reduction factors are the lower fatty acids that are oxidation products of the FAME radical intermediates. It also appears that the OIT reduction factors are ROO radicals (ROO $\cdot$) that are generated in the course of producing the lower fatty acids. As shown in Fig. 17, once RO radicals (RO $\cdot$) and ROO radicals (ROO $\cdot$) are generated, the base oil of engine oil undergoes a chain of oxidation reactions. Antioxidants are known to be additives that function to stop these oxidation reactions and render the RO $\cdot$ and ROO $\cdot$ inactive⁽⁹⁾. Antioxidants function by means of a mechanism called chain termination to render the RO $\cdot$ and ROO $\cdot$ inactive. In a BDF mixture, however, ROO $\cdot$ is produced by the FAME as well as by the base oil for engine oil, and this promotes the consumption of antioxidant. It is conjectured that, as a result, OIT reduction tends to accelerate when in a BDF mixture.

5. Conclusion

Investigation of the effects that bio diesel fuel has on BN and OIT, which are engine oil sludge generation indexes, yielded the following findings:

- (1) Mixture with BDF results in the BN and OIT falling, so that sludge prevention performance is reduced.
- (2) The rate of reduction in sludge prevention performance is greater when BDFs include more unsaturated bonds.
- (3) It is thought that BN reduction factors are lower fatty acids, and OIT reduction factors are ROO radicals (ROO $\cdot$) that are formed during the production of lower fatty acids. These are factors accelerating the generation of engine oil sludge from BDF.

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Model Based Air-path Management Control for Clean Diesel Engine

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ABSTRACT

Research has been conducted on a variety of combustion technologies in order to reduce diesel engine emissions. These technologies should precisely control the state of in-cylinder gas (EGR mass flow, air mass flow, and the like). However, because the controlled object is a multi-input, multi-output (MIMO) system and a coupled system, the use of control systems based on conventional methods that employ PID controllers represents a challenge. Model predictive control (MPC) is well known as a MIMO algorithm. An intake control system that could be applied to the intake system of a diesel engine was constructed by supplementing MPC with a feedback function using a disturbance observer and compensator for the nonlinear characteristic of the actuators. Performance tests using an actual vehicle verified that, when applied to a two-input (throttle valve, EGR valve), two-output (air mass flow, intake chamber pressure) system, the proposed MPC was able to rapidly control each output independently to the target value.

1. Introduction

Diesel engines have higher thermal efficiency and produce a lower amount of CO₂ exhaust than gasoline engines, and are thus gathering attention as an effective internal-combustion engine in helping avert global warming. However, the simple aftertreatment system used in gasoline engines that employ a three-way catalyst with a reduction efficiency close to 100% cannot be applied to lean combustion diesel engines. This makes it necessary to sufficiently reduce NO_x, HC, soot and other exhaust gas components during combustion within the cylinder.

The Exhaust Gas Recirculation (hereafter, "EGR") system is known to be an effective means of reducing NO_x emissions, which is particularly important for diesel engines. This system suppresses the generation of NO_x, which increases rapidly at high combustion temperatures, by mixing the inert gas included in the exhaust gas into the intake air to lower the oxygen concentration, thus lowering the maximum temperature during combustion. However, as the EGR ratio is increased, the amount of soot produced also tends to increase after a certain excess air factor is reached. Therefore, to effectively reduce NO_x emission while also suppressing soot generation, it is necessary to use EGR control and fuel injection control

to accurately control the combustion state, represented by the excess air factor and other parameters, to within the desired range.

Recent studies have focused on technology that can simultaneously reduce both NO_x and soot by using PCCI combustion, which introduces even larger amounts of EGR to aggressively extend the ignition delay period, and incorporates premixed combustion⁽¹⁾. However, the PCCI combustion test results in Figs. 1 and 2 show that the exhaust gas characteristics fluctuate greatly according to the EGR ratio, and practical application needs enhancement of fuel injection control and EGR control performance to enable construction of a high response control system that is capable of tracking transient operation.

On the other hand, research aimed at practical application is also actively underway to develop technology for further reducing tailpipe emissions using a NO_x catalyst and other exhaust aftertreatment systems in order to further reduce exhaust gas in the future. NO_x catalyst systems need regenerating combustion control, called rich control, to reduce the NO_x adsorbed during normal lean combustion⁽²⁾. Rich control needs high-level combustion state control to match the air-fuel ratio to the target value so as not to adversely affect the combustion

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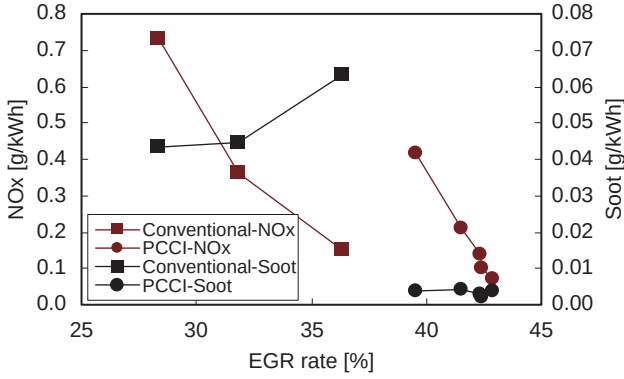


Fig. 1 Emission characteristics of conventional and PCCI combustion

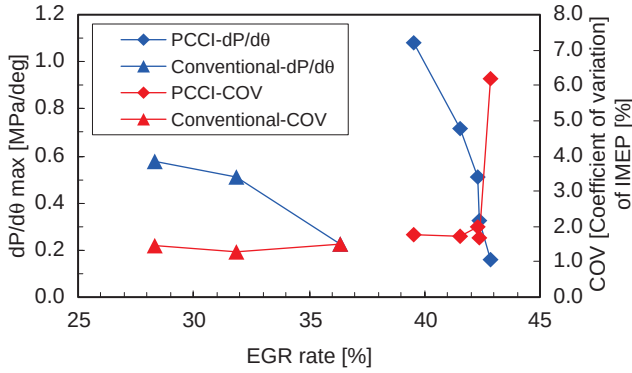


Fig. 2 Noise and vibration characteristics of conventional and PCCI combustion

noise and drivability. Therefore, in addition to fuel injection control, it is also important to increase the accuracy of the intake gas control that controls the distribution of the amount of air and the amount of EGR taken into the cylinder (hereafter “intake gas state”).

The above demands are increasing the need for more precise control of the intake gas state compared to previous systems.

2. Research Goals

This study designed and verified the effects of a control system that applies model predictive control (hereafter, “MPC”) to construct an intake gas state feedback control system in a diesel engine.

The physical amounts to be controlled by intake gas control are the amounts of oxygen and inert gas within the cylinder, that is to say the air mass and the amount of EGR gas. Here, among all the gas components taken into the cylinder, the air mass is the intake air component that excludes the gas returned by EGR. The directly operable intake actuators that control these physical amounts include the EGR valve and the throttle valve.

However, generally when any one of the intake actuators is changed, both physical amounts of the intake

gas state vary. This is the controlled object, called a coupled system, and construction of a feedback control system by conventional methods such as using a PID controller is known to be a challenge. Figure 3 shows the control effects when a general PID controller is applied to an engine simulator that simulates the above controlled object, and shows that coupling produces oscillating behavior.

On the other hand, MPC is known as a control algorithm that can be applied to a multi-input, multi-output system capable of feeding back multiple outputs to the target values. However, optimization calculations should be performed in real time in each sampling period, so MPC is mainly applied as in petrochemical plant systems and other systems that have long sampling periods.

However, in addition to increased ECU calculation processing speeds, remarkable progress is also being made on technology to further increase speeds by methods such as revising calculation algorithms⁽³⁾, and prospects look promising for MPC’s application to relatively fast systems⁽⁴⁾. Therefore, MPC was applied to design a feedback control system for the air mass flow and EGR mass flow, and the effects were confirmed.

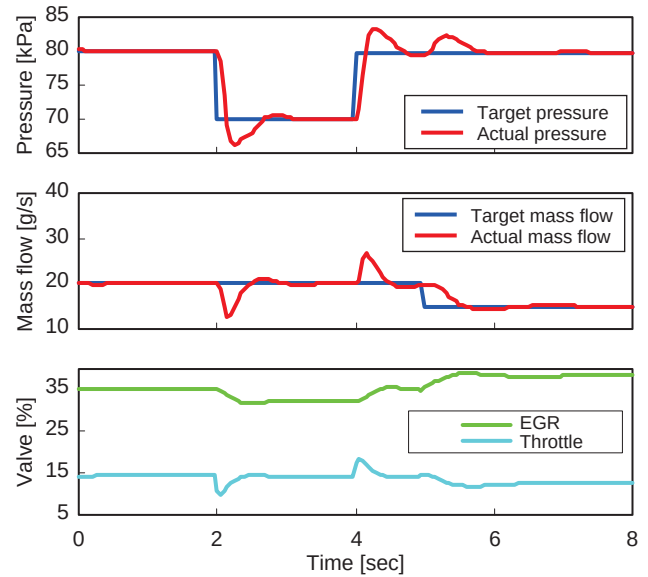


Fig. 3 PI control simulation results

3. Modeling

Figure 4 shows the intake and exhaust system configuration used in general diesel engines. This study considered the controlled object of a 2-input, 2-output system with the input and output variables defined as follows.

Inputs: EGR valve opening, throttle valve opening

Outputs: Intake chamber pressure, air mass flow

Normally, the air mass and the EGR amount taken into the cylinder should be selected as the output parameters, but it is a challenge to directly measure both of these, so the air mass flow and intake chamber pressure were selected as the respective substitute output parameters. Here, the air mass flow is the mass flow of the air measured at the position indicated in Fig. 4. In addition, the EGR amount can be estimated from the intake chamber pressure and the air mass flow, so the intake chamber pressure was selected as the other output parameter.

When applying MPC, optimal control, adaptive control, or other model-based control algorithms, the dynamic characteristics of the controlled object should be expressed by a state-space equation, that is to say a simultaneous equation using differential equations or difference equations.

First, modeling based on physical equations is performed to understand the structure of the controlled object, and the degree of the state-space equation is selected. Next, the physical parameters that prescribe the state-space equation are determined by system identification to obtain the state-space equation used by the control algorithm built into an actual diesel engine.

3.1. Physical Modeling

As shown in Fig. 4, the intake and exhaust system of a diesel engine is composed of a combination of valve elements and chamber elements. Therefore, modeling was performed based on physical principles, focusing on the throttle valve and its downstream chamber as shown by the example in Fig. 5.

Labeling the throttle valve opening as α_{th} , the upstream pressure and downstream pressure as p_0 and p_i , respectively, the upstream density as ρ_0 , and the specific heat ratio as κ , the air mass flow passing through the throttle valve is expressed by the equation of convergent nozzle^{(5), (6)}.

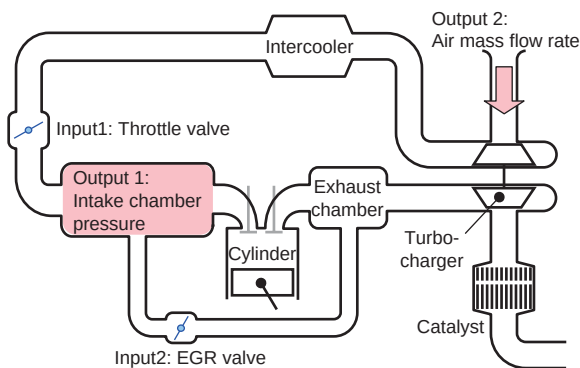


Fig. 4 Air-path system of diesel engine and definition of the controlled object

$$\dot{m}_{th} = A_{th}(\alpha_{th}) \cdot \sqrt{2p_0\rho_0} \cdot \Phi$$

$$\Phi = \begin{cases} \sqrt{\frac{\kappa}{\kappa-1} \left\{ \left(\frac{p_i}{p_0} \right)^{\frac{2}{\kappa}} - \left(\frac{p_i}{p_0} \right)^{\frac{\kappa+1}{\kappa}} \right\}}, & \left(\frac{p_i}{p_0} \right) > \left(\frac{2}{\kappa+1} \right)^{\frac{\kappa}{\kappa-1}} \\ \left(\frac{2}{\kappa+1} \right)^{\frac{\kappa}{\kappa-1}} \sqrt{\frac{\kappa}{\kappa+1}}, & \left(\frac{p_i}{p_0} \right) \leq \left(\frac{2}{\kappa+1} \right)^{\frac{\kappa}{\kappa-1}} \end{cases} \quad (1)$$

Here, $A_{th}(\alpha_{th})$ is the function that expresses the effective aperture area A_{th} relative to the opening α_{th} .

On the other hand, the change in the intake chamber pressure is expressed by the equation of ideal gas state within the chamber and the Law of Conservation of Mass⁽⁷⁾ as follows.

$$p_i V_i = M_i R_i T_i \quad (2)$$

$$\frac{dM_i}{dt} = \dot{m}_{th} - \dot{m}_{cyl} \quad (3)$$

At this time, the change in the intake chamber pressure is expressed as follows.

$$\frac{dp_i}{dt} = n \frac{RT_i}{V_i} (\dot{m}_{th} - \dot{m}_{cyl}) \quad (4)$$

Here, n is the polytropic index.

In addition, the cylinder intake air mass flow can be expressed as the 1st order function of the chamber internal pressure p_i as shown by the following equation.

$$\dot{m}_{cyl} = k_{\eta v} n_e p_i \quad (5)$$

Here, n_e is the engine speed, and $k_{\eta v}$ is a coefficient that includes the volumetric efficiency.

From the above, the following equation that is expressed by a 1st order lag system can be obtained from Eqs. (1), (4) and (5).

$$\frac{dp_i}{dt} = a \cdot p_i + b \cdot A_{th}(\alpha_{th})$$

$$a = -n \frac{RT_i}{V_i} k_{\eta v} n_e, \quad b = n \frac{RT_i}{V_i} \sqrt{2p_0\rho_0} \cdot \Phi \quad (6)$$

However, delay parameter a and gain parameter b are both expressed by non-linear functions, so this is a so-called non-linear system in which the dynamic behavior varies according to the operation point. Therefore, linear

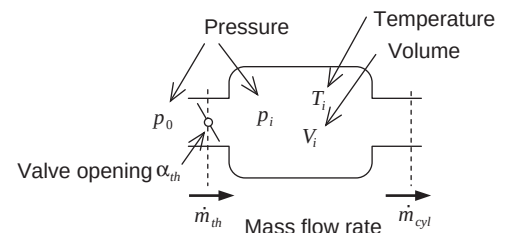


Fig. 5 Element model of valve and chamber system

control theory that includes MPC is unable to be applied in this format.

For this reason, an attempt was made to create a linear system using methods that establish input-state linearization⁽⁸⁾. Figure 6 shows the control system configuration achieved by input-state linearization. When input linearity transformation using the equation of convergent nozzle [Eq. (1)] is positioned at the controller output, the controlled object as viewed from the controller can be considered linear. This enables configuration of a linear controller that uses Eqs. (4) and (5) as the state equations.

The controlled object also has element systems other than those shown in the example above. The dynamic characteristics of each actuator are the dead time element and the 1st-order lag element, and the pressure lag from the air flow meter position to the throttle valve due to gas charging of the chamber can be approximated using the 1st-order lag element. Therefore, the controlled object is considered to be a system that combines dead time elements and lag elements.

3.2. System Identification

The characteristics of the controlled object were examined in the previous section, but the parameters included in the controlled object model are composed of a combination of numerous physical constants, and it is a challenge to physically determine these values. Therefore, system identification⁽⁹⁾ was performed to obtain the state-space equation used by the control theory applied to an actual engine.

Inputs: Command value for mass flow in throttle valve section m_{th_cmd} [g/s]

Command value for mass flow in EGR section m_{egr_cmd} [g/s]

Outputs: Intake chamber pressure p_i [kPa]

Air mass flow m_a [g/s]

The input and output data used for system identification were measured by applying a binary pseudo random signal called a signal of maximal-length sequences⁽¹⁰⁾ to the two input values noted above, and measuring the two output values at that time. Figure 7 shows the input and output time series data at that time.

The dead time T of the controlled object was determined from the measured input and output time series data using a method that estimates the impulse response⁽⁹⁾.

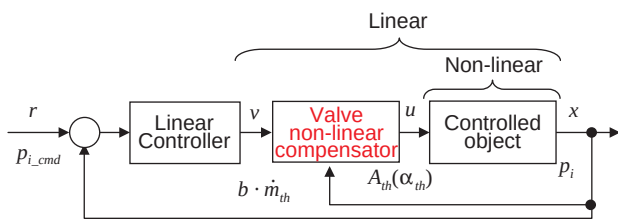


Fig. 6 Control structure with non-linear compensator

Next, the model structure used the 2-input, 2-output multivariable Auto Regressive with eXtra input model (hereafter “ARX model”) expressed by Eq. (7), and system identification was performed using the measured input and output time series data.

$$y(k+1) + A_1 y(k) + A_2 y(k-1) = B_1 u(k-T) + B_2 u(k-T-1) \quad (7)$$

$$y(k) = \begin{bmatrix} y_1(k) \\ y_2(k) \end{bmatrix}, u(k) = \begin{bmatrix} u_1(k) \\ u_2(k) \end{bmatrix} \quad (8)$$

$$A_1 = \begin{bmatrix} a_{111} & a_{112} \\ a_{121} & a_{122} \end{bmatrix}, A_2 = \begin{bmatrix} a_{211} & a_{212} \\ a_{221} & a_{222} \end{bmatrix}, B_1 = \begin{bmatrix} b_{111} & b_{112} \\ b_{121} & b_{122} \end{bmatrix}, B_2 = \begin{bmatrix} b_{211} & b_{212} \\ b_{221} & b_{222} \end{bmatrix} \quad (9)$$

Figure 8 shows the time series data of the actual output and the ARX model output to verify the accuracy of the identified 2-input, 2-output ARX model. The ARX model output accurately represents the actual output behavior, indicating that the obtained ARX model has sufficient accuracy. This is thought to be because inserting a non-linear compensator between the system identification input values and the input values applied to

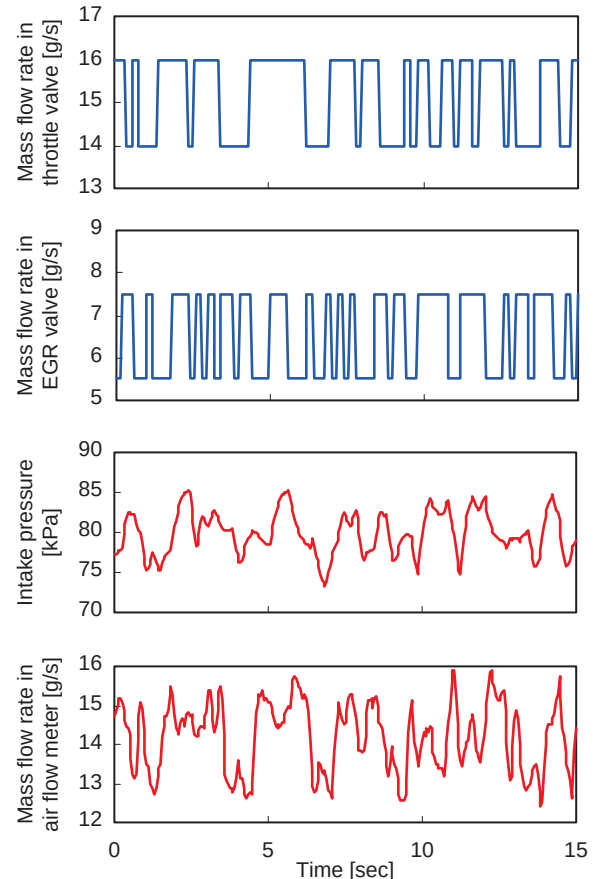


Fig. 7 Input and output measurement data

the actuators enabled the non-linear characteristics of the controlled object to be sufficiently canceled, and thus provided input and output data with high signal to noise ratios.

In addition, verification using the spectral analysis method⁽⁹⁾ also confirmed that the spectrum of the measured values and that of the obtained ARX model matched closely, which indicated that the ARX model is also accurate in terms of frequency range.

3.3. Augmented State-space Modeling

The ARX model obtained by system identification was transformed into a state-space equation for use as an MPC internal model.

Eqs. (7), (8), and (9) can be expressed in matrix form as follows.

$$\begin{cases} X(k+1) = AX(k) + Bu(k-T) \\ y(k) = CX(k) \end{cases}$$

$$X(k) = \begin{bmatrix} y(k) \\ y(k-1) \\ u(k-1) \end{bmatrix}, A = \begin{bmatrix} A_1 & A_2 & B_2 \\ I & O & O \\ O & O & O \end{bmatrix}, B = \begin{bmatrix} B_1 \\ O \\ I \end{bmatrix}, \quad (10)$$

$$C = [I \quad O \quad O]$$

In addition, the following augmented model is defined by adding a constant output disturbance to the states.

$$\begin{cases} \begin{bmatrix} X(k+1) \\ d(k+1) \end{bmatrix} = \begin{bmatrix} A & 0 \\ 0 & I \end{bmatrix} \begin{bmatrix} X(k) \\ d(k) \end{bmatrix} + \begin{bmatrix} B \\ 0 \end{bmatrix} u(k-T) \\ y(k) = [C \quad I] \begin{bmatrix} X(k) \\ d(k) \end{bmatrix} \end{cases} \quad (11)$$

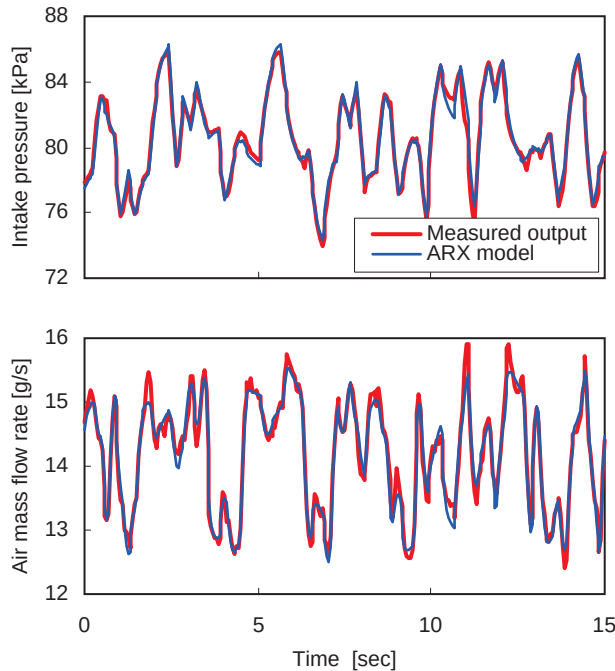


Fig. 8 Time series result

The constant disturbance item $d(k)$ of this augmented model is successively corrected using a disturbance observer.

Here, the input and output vectors are defined as shown in the following equation.

$$u(k) = \begin{bmatrix} m_{th_cmd}(k) \\ m_{egr_cmd}(k) \end{bmatrix}, y(k) = \begin{bmatrix} p_i(k) \\ m_a(k) \end{bmatrix} \quad (12)$$

At this time, the state-space equation is expressed by Eqs. (13) and (14), so this study used these equations to design the MPC controller.

$$\begin{cases} \begin{bmatrix} p_i(k+1) \\ m_a(k+1) \\ p_i(k) \\ m_a(k) \\ m_{th_cmd}(k) \\ m_{egr_cmd}(k) \\ d_{pi}(k+1) \\ d_{ma}(k+1) \end{bmatrix} = \begin{bmatrix} A_1 & A_2 & B_2 & O \\ I & O & O & O \\ O & O & O & O \\ O & O & O & I \end{bmatrix} \begin{bmatrix} p_i(k-T) \\ m_a(k-T) \\ p_i(k-T-1) \\ m_a(k-T-1) \\ m_{th_cmd}(k-T-1) \\ m_{egr_cmd}(k-T-1) \\ d_{pi}(k-T) \\ d_{ma}(k-T) \end{bmatrix} \\ + \begin{bmatrix} B_1 \\ O \\ I \\ O \end{bmatrix} \begin{bmatrix} m_{th_cmd}(k-T) \\ m_{egr_cmd}(k-T) \end{bmatrix} \\ \begin{bmatrix} p_i(k) \\ m_a(k) \end{bmatrix} = C \begin{bmatrix} p_i(k) \\ m_a(k) \\ p_i(k-1) \\ m_a(k-1) \\ m_{th_cmd}(k-1) \\ m_{egr_cmd}(k-1) \\ d_{pi}(k) \\ d_{ma}(k) \end{bmatrix} \end{cases} \quad (13)$$

$$C = [I \quad O \quad O \quad I], I = \begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix}, O = \begin{bmatrix} 0 & 0 \\ 0 & 0 \end{bmatrix} \quad (14)$$

4. Control System Design

4.1. Control Goals

The following requirements were given for the design of a control system for the controlled object in this study.

- (1) Increasing the output tracking speed with respect to changes in target values
- (2) Suppressing the effects of coupling in a multi-input, multi-output system
- (3) Removing the non-linearity of the valve mass flow characteristics
- (4) Compensating the pressure and mass flow lag characteristics produced by the actuator and chamber capacities
- (5) Achieving control performance and stability with respect to manufacturing variance and changes in the mass flow characteristics of actuators and other parts

- (6) Achieving stability with respect to disturbances produced by environmental changes, and other factors

4.2. Control System Configuration

This study addressed the requirements listed in the previous section using the following elements. Figure 9 shows the control system configuration.

Elements	Requirements
(a) Model predictive controller (MPC)	(1), (2), (4)
(b) Reference trajectory	(1), (6)
(c) Disturbance observer	(5), (6)
(d) Valve block static non-linear compensator (3)	

4.2.1. Model predictive control

One of the practical advantages of MPC is the ability to naturally handle multi-input, multi-output control. Additional practical advantages are that the basic operation concept is easy to understand, and adjustment in an actual engine is easy and intuitive. Therefore, focusing on these advantages, MPC was applied as the controller.

Figure 10 shows a concept drawing of basic operation, and the operation procedure is described below.

- (1) Generation of reference trajectory: Designate the speed

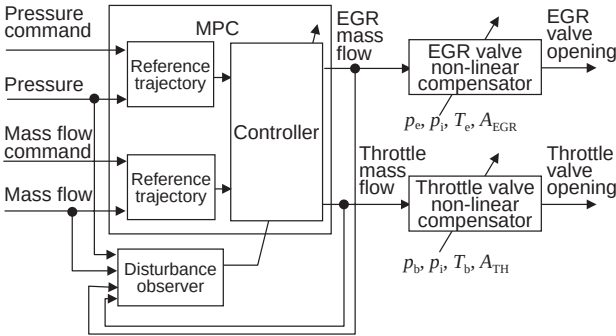


Fig. 9 Block diagram of controller

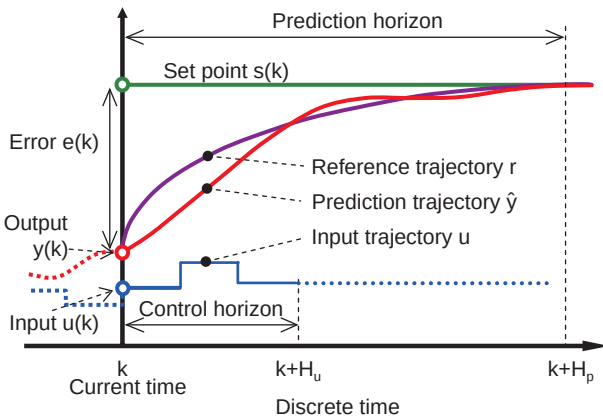


Fig. 10 Basic concept of MPC

at which the plant moves towards the setting value using the current time error $e(k)$.

- (2) Calculation of prediction trajectory: Predict the behavior of the output during the prediction horizon period from the internal model and the input during the control horizon period.
- (3) Determination of input trajectory: Perform optimization calculation to determine the ideal combination of inputs (input trajectory) so that the reference trajectory (1) and prediction trajectory (2) match as closely as possible.
- (4) Application of input values: Apply the first element $u(k)$ of the input trajectory (3) to the actual controlled object.

The entire cycle from step (1) through step (4) is repeated after a single sampling period.

4.2.2. Disturbance observer

MPC is a control theory based on a state equation (hereafter, “model”) that expresses the dynamic characteristics of the controlled object, so if that model contains errors, it will produce error relative to the intended behavior. In particular, if the model steady state gain is inaccurate, steady state error will occur in the controlled object output. As a result, the device characteristics may change or have variance, or the control performance may drop due to unpredictable changes in the environment, and other factors. Therefore, a disturbance observer was introduced to suppress the effects of disturbance, which are a challenge to include in the model.

Dynamic Matrix Control (DMC)⁽¹¹⁾ is known as a method that applies a disturbance observer to MPC by the procedure shown below. This study used DMC to apply a disturbance observer that helps eliminate the effects of constant disturbance.

The disturbance observer in MPC performs the following standard procedure at time k .

- (1) Measure the actual output $y(k)$.
- (2) Estimate the disturbance as the difference between the actual output and the estimated output.
- (3) Use that estimated disturbance value to predict the output during the prediction horizon period.

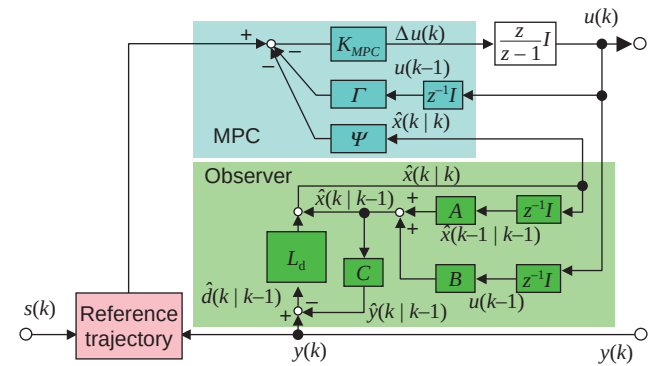


Fig. 11 Block diagram of MPC with disturbance observer

This method should have information on all state variables of the controlled object, so a so-called state observer is needed to estimate state variables that cannot be directly measured.

The state vector that includes the constant output disturbance can be estimated using an augmented state-space model that includes the output disturbance model derived by Eq. (13). Labeling the state estimation gain matrix as L , the standard observer equation is expressed by the following equation.

$$\begin{bmatrix} \dot{\hat{x}}(k+1|k) \\ \hat{d}(k+1|k) \end{bmatrix} = \begin{bmatrix} A & 0 \\ 0 & I \end{bmatrix} \begin{bmatrix} L_x \\ L_d \end{bmatrix} [C \quad I] \begin{bmatrix} \hat{x}(k|k-1) \\ \hat{d}(k|k-1) \end{bmatrix} + \begin{bmatrix} B \\ 0 \end{bmatrix} u(k) + \begin{bmatrix} L_x \\ L_d \end{bmatrix} y(k) \quad (15)$$

From the above, the overall configuration that combines MPC with the observer is as shown in Fig. 11.

5. Verification of Operation by Simulations

5.1. Conditions

The MPC logic is large-scale and complex, so operation was verified using an engine simulator to debug the algorithm and perform pre-calibration before application to an actual engine. The conditions of the engine simulator used are as follows.

- One-dimensional fluid simulation software GT-Power Ver. 6.1 for analyzing engine performance
- An average value model using a neural network (MeanV Cyl)

Figure 12 shows the simulation environment configuration. In the environment shown in the figure, the control algorithm used in the actual engine is described by MATLAB/Simulink, and is linked with GT-Power. This enables verification of control algorithm operation using a controlled object having similar dynamic characteristics to those of an actual engine.

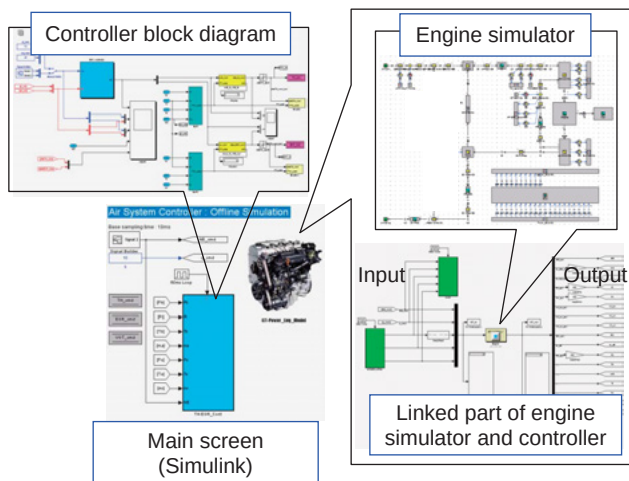


Fig. 12 Simulation environment

5.2. Results

Target value response simulations were conducted under fixed engine speed and fuel injection amount conditions.

MPC includes multiple calibration parameters. These calibration parameters are selected by the generally known MPC tuning index⁽¹¹⁾, and ultimately determined through simulations so that the output characteristics have fast

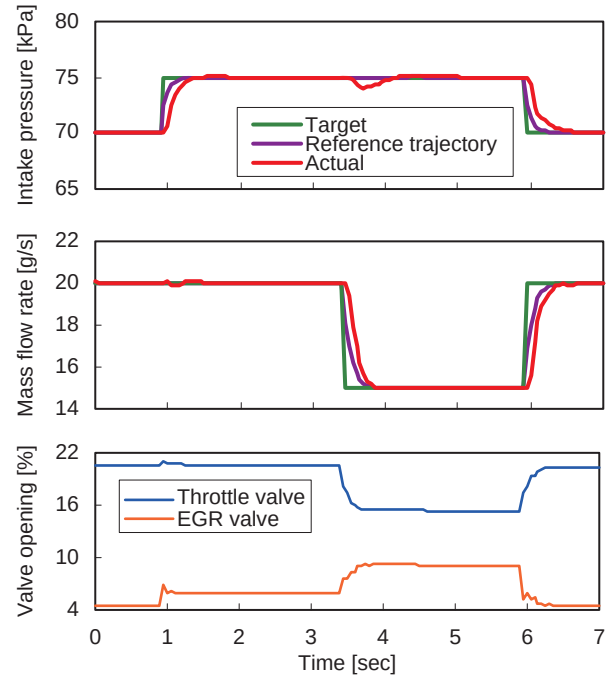


Fig. 13 Simulation results for variable coincidence point

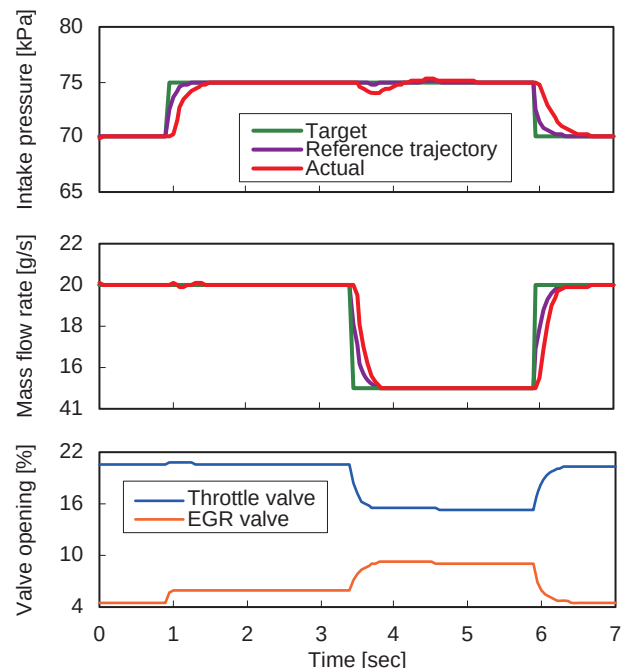


Fig. 14 Simulation results with variation of target value

responsiveness with little overshoot. Here, the effects of the coincidence point parameter are described as an example. The coincidence point parameter is a calibration parameter that stipulates in which steps of the prediction horizon period to match the prediction value to the reference trajectory obtained from the output target value, within the optimization calculations used to determine the control input.

Figures 13 and 14 show the target value response simulation results. Figure 13 shows the results when coincidence points are set over the entire prediction horizon period. This obtained a sufficiently fast output response, but unstable, oscillating behavior tended to appear in the input variables of the throttle valve opening and EGR valve opening, particularly during the transient period in which the output changes. In contrast, Fig. 14 shows the results when coincidence points are set at only the two points of the start and end of the prediction horizon period. Here, relatively well-balanced results were obtained with good response and stability for both inputs and outputs. In addition, setting fewer coincidence points also enables the calculation load on the ECU to be reduced.

The simulation results show that the proposed MPC

operates as intended, and that the two output values converge rapidly, without oscillation due to coupling, relative to the changes in two independent target values.

6. Verification of Operation Using Actual Vehicles

Next, target value response tests were conducted using MPC and the conventional control method mounted in actual vehicles. The test conditions were steady state operation with a constant engine speed and fuel injection amount.

Figure 15 shows the test results for the intake chamber pressure target value step response using a constant air mass flow target value. Here, comparison was performed using the time to reach 90% of the target value (90% response time, hereafter, “T90”) as the index of the response speed to the target value. For the conventional control, T90 needed approximately 500 ms in the decrease direction (a in the figure) and approximately 2 s or more in the increase direction (b in the figure). In contrast, while MPC exhibited some overshoot, it had a faster response speed than the conventional control, with a T90 of only approximately

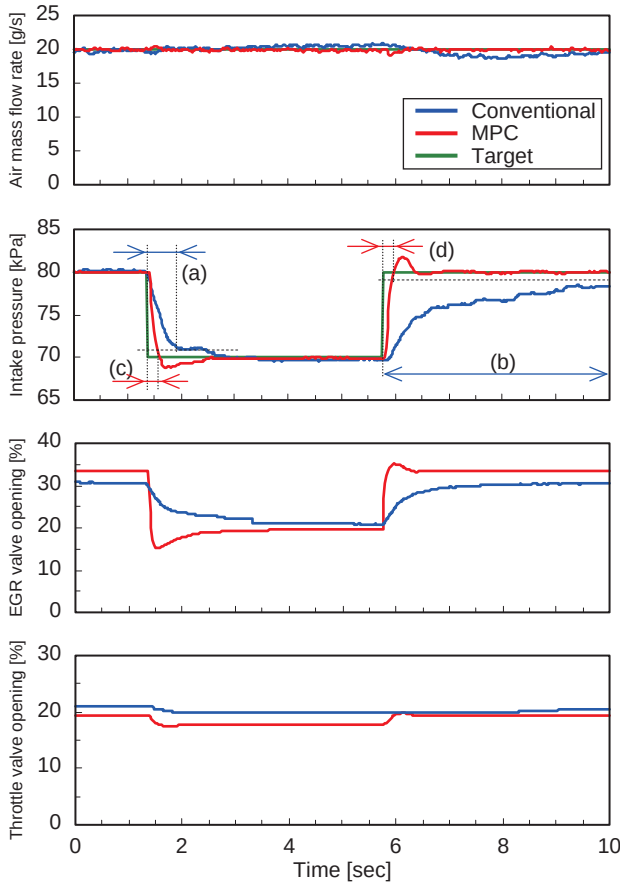


Fig. 15 Test results with variation of target pressure controlled by MPC

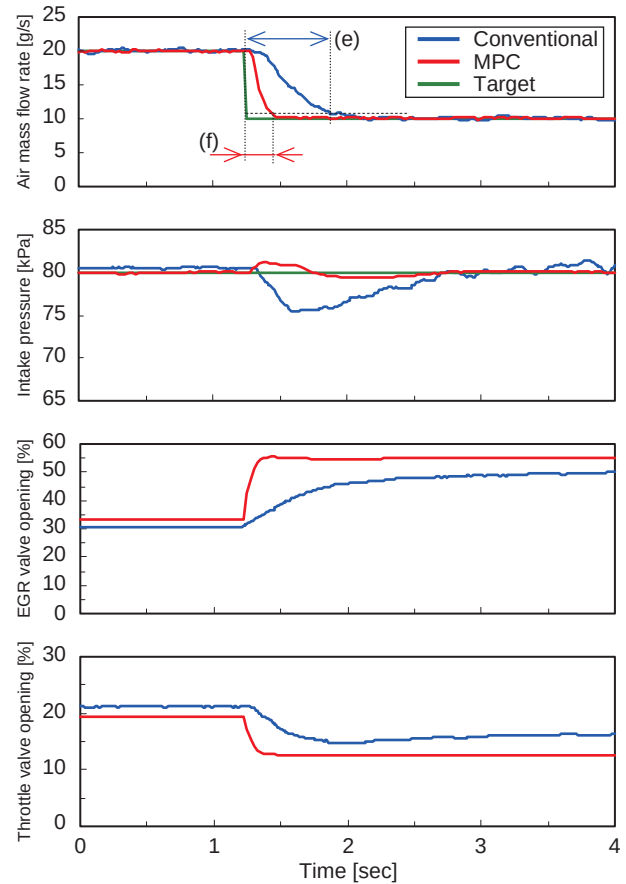


Fig. 16 Test results with variation of target mass flow controlled by MPC

200 ms in both the decrease and increase directions (c and d in the figure). In addition, the conventional control exhibited some fluctuation in the actual output, and did not converge completely with the air mass flow target value.

Figure 16 shows the test results for the air mass flow target value step response using a constant intake chamber pressure target value. T90 for the conventional control was approximately 800 ms (e in the figure). In addition, fluctuating behavior was seen in the actual pressure with respect to a constant intake chamber pressure target value due to the effects of coupling. In contrast, MPC had a T90 of approximately 200 ms (f in the figure), exhibited no pressure fluctuation due to coupling, and achieved a faster response speed than the conventional control.

The above results showed that applying MPC enabled two outputs to rapidly converge, independently, to two target values in the same manner as the engine simulator results.

Under these test conditions, MPC control converged to the target values at least 2.5 times faster than the conventional control. In addition, these enhancement effects further increased when the intake chamber pressure target value was increased and the air mass flow target value was varied.

7. Conclusion

Intake control that uses MPC was applied to a diesel engine intake system, which is a multi-input, multi-output coupled system, and the following items were verified.

- (1) An MPC internal model was constructed using a multivariable ARX model obtained through system identification, and non-linear compensation of the actuator mass flow characteristics based on the equation of convergent nozzle. This enabled an accurate internal model to be obtained.
- (2) MPC was applied to the intake control of a diesel engine. The results showed that two outputs could be rapidly converged, independently, to the respective target values, and a convergence speed of at least 2.5 times faster than that of conventional control was achieved.

As mentioned in the Introduction, intake state deviation is a factor that can adversely affect engine performance such as emissions and combustion noise. Therefore, application of intake control that uses MPC, which increases the intake state responsiveness, can be expected to enhance emissions, combustion noise, and other engine performance features in transient states.

In the future, the authors intend to further refine algorithms to enable operation over the entire engine range, including transient states during which the engine speed and fuel injection amount vary, and to practically apply highly accurate intake control systems to realize various advanced combustion technologies.

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Measurement of Local Air/Fuel Ratio and Evaluation of Accuracy of Engine Simulation Using Micro-Cassegrain Sensor

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ABSTRACT

The fuel consumption rate of gasoline engines can be decreased by reducing cyclic variation and variations in the combustion state between cylinders, but this necessitates high-precision combustion control. Control of the air-fuel ratio (A/F) in the cylinders, in particular the local A/F near the spark plugs, is an important factor in combustion control. A three-dimensional engine simulation model enabling prediction of the local A/F near the spark plugs was therefore constructed. A micro-Cassegrain sensor able to measure the local A/F near the spark plugs in a real engine was used to evaluate the accuracy of the prediction. The results indicated that the local A/F can be predicted within the range of cyclic variation.

1. Introduction

Today's increased focus on environmental issues has generated demands for the achievement of cleaner exhaust emissions and decreased fuel consumption rate in gasoline engines. Conventional gasoline engines precisely control the air-fuel ratio of the exhaust gas in order to maximally exploit the performance of the catalyst. In the future, it will be necessary to increase combustion efficiency and maintain optimal combustion at all times, even in transient states, in order to achieve decreased fuel consumption rate.

Achieving normal combustion in all cylinders during each cycle is desirable from the perspective of increasing combustion efficiency. If it is hypothesized that the in-cylinder pressure in each cycle in which the pressure drops below the average figure could be increased to the average pressure, the engine shown in Fig. 1 would achieve a mean fuel efficiency per cylinder of 3.3%, and a fuel efficiency of 13.3% overall. However, in real engines, variations in the fuel concentration⁽¹⁾ and the distribution of internal EGR gas⁽²⁾ produces cyclic variations⁽³⁾ and variation between cylinders. These are considered to be factors contributing to a decline in the

average performance of the engine.

It has been determined experimentally that the ignition delay period in a spark ignition engine varies depending on whether or not the air-fuel mixture around the spark plugs at ignition is in an ideal state, and this results in fluctuations in main combustion in each cycle. Given this, the research discussed in this paper focused on the air-fuel mixture, in particular the air-fuel mixture around the

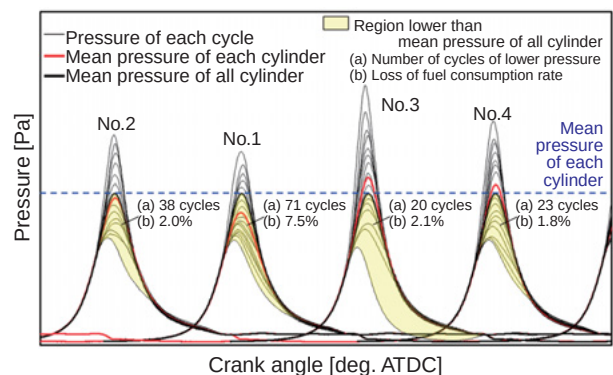


Fig. 1 Pressure history of each cylinder and cycle

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** Imagineering, Inc.

spark plugs which affects ignitability, with the goal of developing a method to enable its control.

In order to study the behavior of the air-fuel mixture, it was decided to apply three-dimensional simulations, which are today extensively employed in automotive applications. However, in the case of research reported by Ishikawa, et al⁽⁴⁾, despite the fact that the results obtained for the distribution of the air-fuel mixture in the combustion chamber in a simulation of the in-cylinder A/F distribution in a port-injected visualized engine corresponded well with actual measurements, the study was not able to quantitatively evaluate the A/F around the spark plugs. In order to enable quantitative evaluation of the accuracy of prediction of the A/F around the spark plugs in the present study, a micro-Cassegrain sensor able to quantitatively measure the A/F around the spark plugs in a vehicle-mounted engine without engine modification was employed in evaluating the accuracy of A/F predictions in a three-dimensional simulation.

2. Principle of Measurement of Local In-cylinder A/F

In-cylinder A/F measurements have normally been conducted using gas sampling or visualization methods such as laser-induced fluorescence (LIF) in a one-cylinder engine. However, these methods involve a number of issues. Numerous engine modifications are necessary; changing the shape of the pistons and changing the in-cylinder flow, for example by reducing the compression ratio, results in changes in charging efficiency and A/F, among other factors. As a result, setting conditions for the simulations represents a challenge. In addition to these issues, given the focus in the present study on variations between cycles and variations between cylinders, it would be desirable to conduct measurements in a multi-cylinder engine fitted in a vehicle rather than in a single cylinder engine. The micro-Cassegrain sensor used in the present study provided the advantage of enabling the analysis to be conducted without modifying the engine.

2.1. Principle of Measurement Using Micro-Cassegrain Sensor

Figure 2 shows the structure of the micro-Cassegrain sensor. A micro-Cassegrain sensor permits the radical chemiluminescence produced by combustion to pass through concave and convex mirrors that face each other. The light passes through an optic fiber and, with the involvement of a dichroic mirror and narrow band pass filter, the emission intensity of the characteristic wavelengths of OH radicals, and the like is detected by a photomultiplier tube.

The core diameter of the fiber is 230 μm , and it can be fitted inside a spark plug. The measurement domain is 0.1 mm and below, and the distance from the sensor tip to the measurement position is 3 mm.

2.2. Method of Measurement of Local A/F

As Fig. 3 shows, the micro-Cassegrain sensor detects the emission intensity of a variety of radicals, including OH*, CH*, and C₂*. A/F can be identified by means of the ratio of the peak values of these emission intensities of these radicals⁽⁵⁾⁻⁽¹⁰⁾, and if the A/F is changed, the ratio also changes linearly. Figure 4 shows a conceptual diagram of the relationship between flame propagation around the spark plug and the measurement position in the cylinder. A flame kernel is formed from the air-fuel mixture ignited by the spark plug; the flame propagates and passes the measurement position, and the luminescence of the radicals is detected. The ignition delay period is calculated from the time of detection of the emission intensity peaks by the sensor and the timing of the ignition signal. Using this data, the flame propagation speed can be calculated from the distance

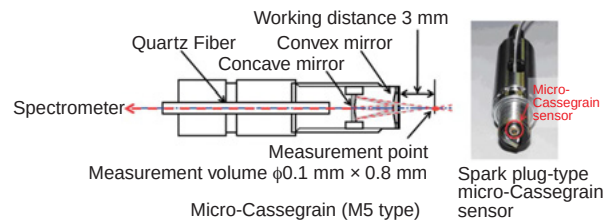


Fig. 2 Schematic of micro-Cassegrain sensor and shape of spark plug-type micro-Cassegrain sensor

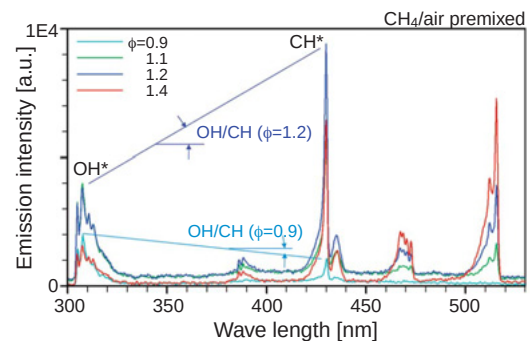


Fig. 3 Emission intensity of fuel wave length

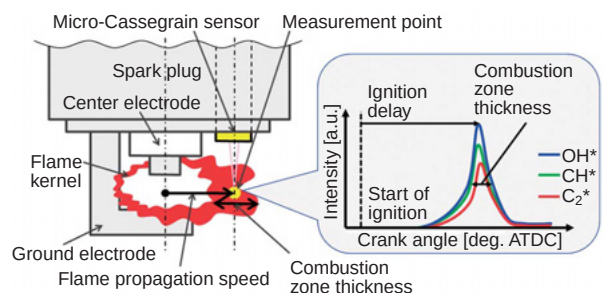


Fig. 4 Schematic of flame propagation and measurement point in cylinder

Table 1 Engine specifications

Engine	2.2L 4-cylinder gasoline engine
Bore × Stroke	87 × 90.7 [mm]
Compression ratio	11.1
Engine speed	1500 [rpm]
Tumble control valve opening	15 [deg.]
Injection	Twin, Primary, Secondary
Injection timing	44.5 [deg. ATDC] (Twin) 25.5 [deg. ATDC] (Primary, Secondary) =>60 [deg. ATDC] (Injection end)
Spark timing	35 [deg. BTDC]

from the plug to the measurement position. In addition, the width of the combustion domain can be determined from the width of the time series data for emission intensity.

3. Test Method and Measurement System

3.1. Characteristics of Test Engine

Figure 5 shows the layout of the test engine intake manifold. To enable the tumble ratio to be varied, the engine was fitted with a tumble control valve and a tumble plate which enables adjustment of the gas flow in the intake port. In addition, an inlet for external EGR gas was positioned below the tumble plate to enable the EGR flow to be controlled, and one injector was positioned in each port, facing the respective intake valves, to enable control of air-fuel mixture formation, even during fuel supply.

3.2. Operating Conditions and Measurement Position

Table 1 shows the measurement conditions employed in this study of A/F prediction accuracy. The tests were conducted using the same fuel injection amount and injection end timing under different injection conditions: with both of the injectors discussed above in operation, only the primary-side injector in operation, and only the secondary-side injector in operation.

Figure 6 shows the positioning of the micro-Cassegrain sensor and pressure sensors employed to

enable evaluation of calculation accuracy. Pressure was measured at seven points from upstream to downstream to enable evaluation of the gas flow and charging efficiency, parameters which are important in evaluating A/F. A sensor manufactured by Kistler (type 6052B) was employed inside the cylinder, for a total of eight pressure sensors recording simultaneous measurements. The pressure sensor in the cylinder was replaced by the micro-Cassegrain sensor when it was used to measure A/F.

3.3. Measurement System

Figure 7 shows the measurement system employing a micro-Cassegrain sensor used in this study to measure local in-cylinder A/F in a vehicle-mounted engine. The measurement system, which enables bench-top measurement, is made up of a sensor that is integrated with the spark plug which detects in-cylinder emissions, a light divider unit that detects the emissions from the sensor, and analytic software. The sensor was created to match the specifications of the test engine. The system synchronizes the angle signal output by an engine crank angle detector and the fuel injection signals used to discriminate cylinders with the detected emission signal, and determines the flame propagation speed, and the like, based on emission intensity, detection timing, and the like.

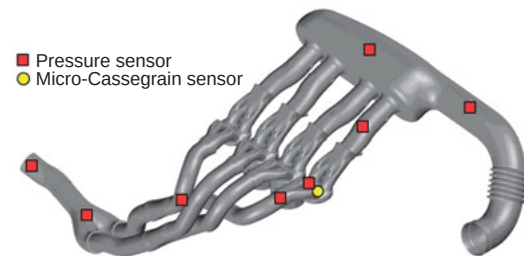


Fig. 6 Measurement point

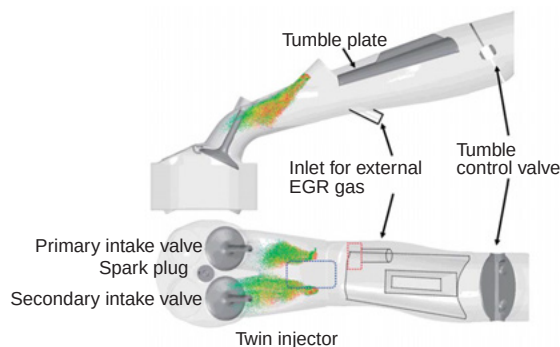


Fig. 5 Test engine intake manifold layout

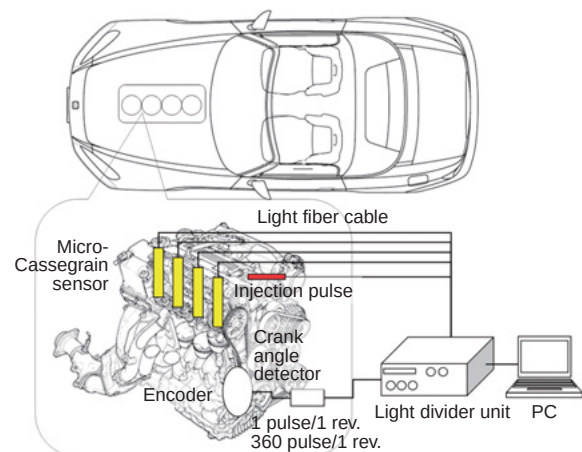


Fig. 7 Micro-Cassegrain measurement system

4. Measurement Results

4.1. Results of Measurement of Local A/F Using Micro-Cassegrain Sensor

Figure 8 shows the results of measurement of local A/F in each cycle when the pre-set engine A/F was varied. When the pre-set A/F was varied from 12 to 20, the average value of the local A/F also increased, with 3.5% or less deviation between pre-set A/F and average local A/F.

Figure 9 shows the results of A/F measurements using a UEGO sensor manufactured by NTK and local A/F measurements using the micro-Cassegrain sensor when the intake pressure was varied. The results for the UEGO

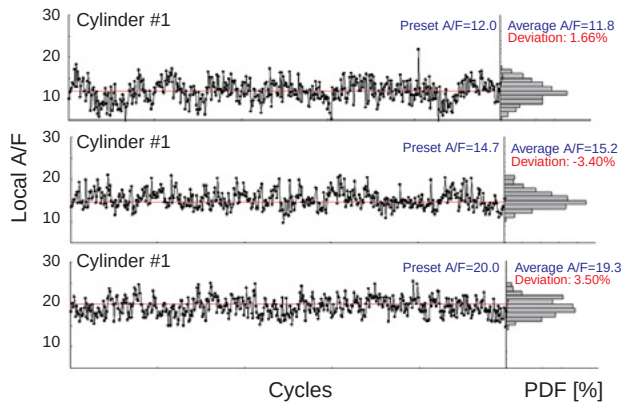


Fig. 8 Results for local A/F using micro-Cassegrain sensor

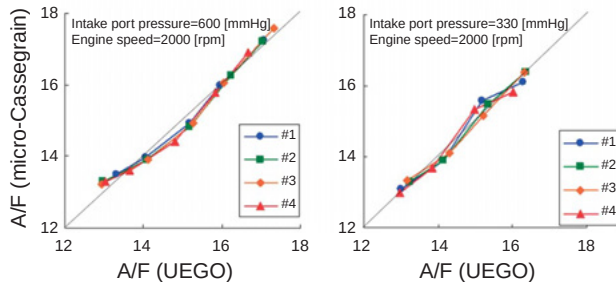


Fig. 9 Comparison of micro-Cassegrain and UEGO sensors

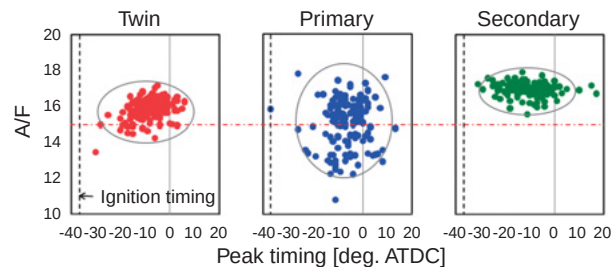


Fig. 10 Results for peak timing and A/F for different injection conditions

sensor and the micro-Cassegrain sensor are virtually identical for both intake port pressures, indicating that the output of the micro-Cassegrain sensor demonstrates sufficient accuracy.

4.2. Results of Local A/F Measurements when Fuel Injection Conditions Were Varied

The differences in A/F for each cycle under different injection conditions [injection using both injectors (“Twin”), injection only in the port fitted with an inlet for external EGR (“Primary”), and injection only in the opposite port (“Secondary”)] were measured and compared. The results are shown in Fig. 10. The areas bounded by circles in the graphs show the operating conditions produced by each form of injection. An ideal combustion state would be shown by a circle with a low radius, centered on a stoichiometric A/F. Compared to Twin injection, injection on the Primary side only produced greater variation in A/F. Variation was lower in the case of injection only in the Secondary side, but while the chain of points in the graph indicates a gasoline stoichiometric A/F, overall there is a shift to the lean side. These results show that when injection conditions are varied, there is a considerable variation in the local air-fuel mixture concentration around the spark plugs in each cycle.

The horizontal axes of the graphs show the timing of the radical emission intensity peaks, more specifically, the ignition delay period. Variations in the ignition delay period are displayed for all the forms of injection, but there is no significant difference in the radii of the circles or the position of their centers in the direction of the time axis, indicating that there is little correlation between A/F fluctuations and the ignition delay period. It was therefore considered that some factor other than A/F fluctuations, such as EGR gas concentration, is involved in the phenomenon of ignition delay.

5. Results of Verification of A/F Prediction Accuracy Using Three-dimensional Simulation

Calculations were conducted following the flow shown in Fig. 11 in order to evaluate the accuracy of predicting local A/F using a three-dimensional simulation. Using measured intake and exhaust pressure and temperature as a reference, a one-dimensional gas exchange simulation was conducted to determine boundary and initial conditions for the three-dimensional simulation. Fuel spray parameters were created for the three-dimensional simulation based on the fuel spray conditions used in unit tests and the results obtained in the tests. An air-fuel mixture distribution was calculated taking into consideration the mixture of the intake air and fuel, and the dispersion and wall-adhesion of the fuel as the input data for the three-dimensional gas flow simulation. To

evaluate the accuracy of the simulation results, A/F values were determined at the same position as the measurement position used for the micro-Cassegrain sensor, and compared with measurement results for the three injection conditions.

5.1. Three-dimensional Engine Simulation

VECTIS, manufactured by Ricardo, was employed for the three-dimensional engine simulation. Figure 12 shows the simulation model. The boundary and initial conditions formulated using the one-dimensional simulation and the fuel spray parameters were input into this model and calculations were conducted. The mesh used for the calculations is shown in Fig. 13. A mesh size of 1 mm was used for the combustion chamber and the areas around the intake and exhaust valves, 0.1 mm for the tumble control valve flow channel, and 2 mm for all other areas.

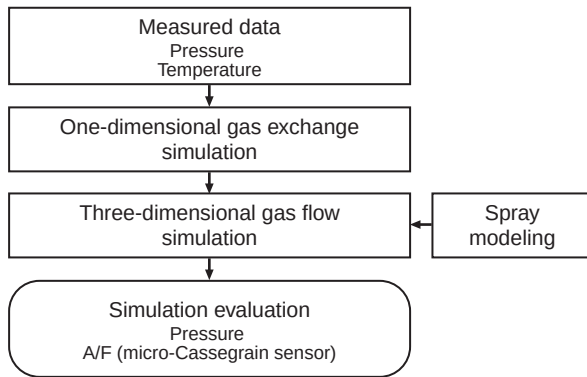


Fig. 11 Engine simulation flow

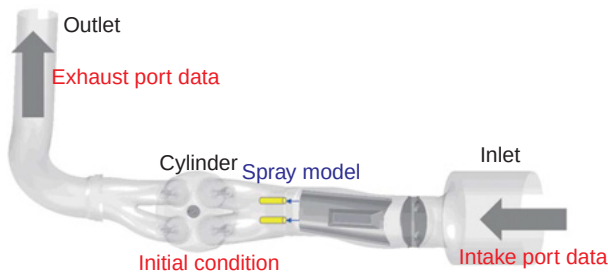


Fig. 12 Three-dimensional simulation model

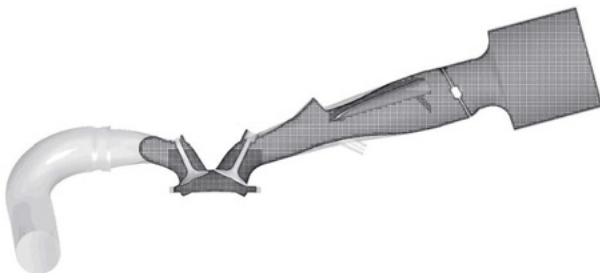


Fig. 13 Mesh used for calculation

5.2. One-dimensional Gas Exchange Simulation

WAVE, again manufactured by Ricardo, was employed for the one-dimensional gas exchange simulation. The simulation model is shown in Fig. 14. The tumble control valve and external EGR pipe used in the test engine were modeled. Figures 15(a) and (b) show measurement and simulation results for in-cylinder pressure and intake port pressure. The measurement results for in-cylinder pressure in Fig. 15(a) are an average for 100 cycles, but the simulation reproduces them well. The measured intake port pressure in Fig. 15(b) is always 60 kPa or below as a result of restriction due to the tumble control valve fitted in the test engine, but optimal setting of the restriction parameters has enabled good reproduction of the pressure history in the simulation.

5.3. Determination of Fuel Spray Calculation Parameters

A fuel spray injected into a constant volume chamber was visualized in order to determine the fuel spray calculation parameters. Scattered light was employed to

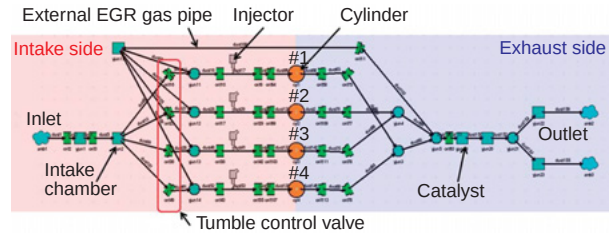
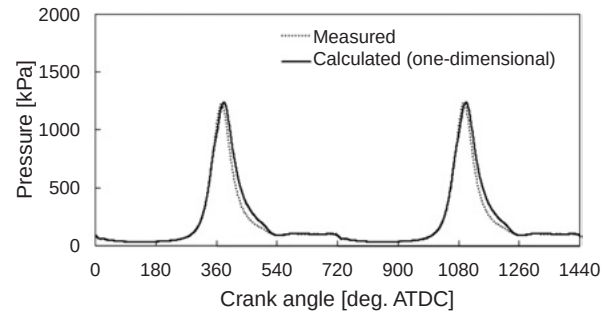
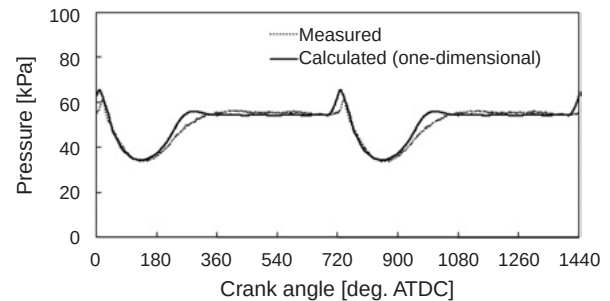


Fig. 14 One-dimensional simulation model



(a) In-cylinder pressure



(b) Intake port pressure

Fig. 15 Comparison between measured pressure and one-dimensional gas exchange simulation

enable imaging of the fuel injected into the bottom of the chamber. The atmospheric conditions for the measurements were atmospheric pressure and ambient temperature.

A discrete droplet model (DDM) was used in the fuel spray calculations. The measured droplet size distribution was employed as the boundary condition, and a Reitz-Diwakar model⁽¹¹⁾ was used for droplet break-up (“secondary break-up”). The atmospheric conditions at the time of the measurements were used as the initial conditions for the simulation. The mesh size was set at 2 mm, and the time step at 1.0×10^{-5} s. Figure 16 shows the visualized spray and the calculated spray. The time is 0.4 ms following injection. The spray angle for the measurements was set at 18°; while the angle is slightly wider in the calculation results, they show an oval-shaped spray overall, and can thus be judged as having reproduced the measurement results well. Figure 17 shows a comparison of measured spray tip penetration and calculation results. There is good correlation between the results for all points in time, indicating that the simulation demonstrates sufficient accuracy.

5.4. Evaluation of Accuracy

Figure 18 shows measurement results and three-dimensional simulation results for in-cylinder pressure, and the level of deviation between the results. The

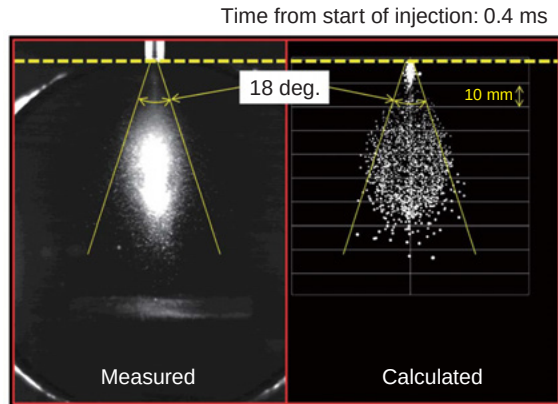


Fig. 16 Comparison of form of fuel spray

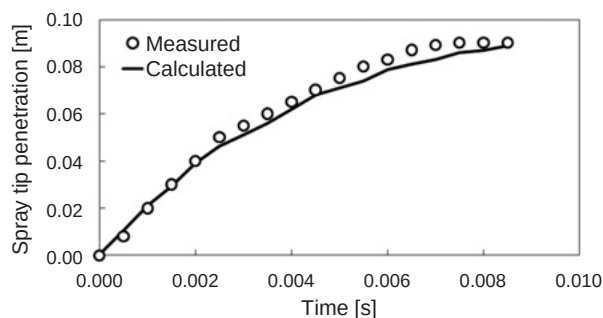


Fig. 17 Comparison between measured spray tip penetration and calculation result

calculations were conducted up to the point of gas flow and mixing of the fuel and air, and are therefore complete at the ignition timing of 35 deg. BTDC. The deviation between the results is high at 4.8% at the ignition timing, but the pressure history in both sets of results corresponds well, indicating that the simulation reproduces the in-cylinder pressure field with good accuracy. Given this, the simulation can be considered to accurately reproduce the flow rate and behavior per unit time of the air and fuel flowing into the cylinder through the intake port and the valve. The 4.8% deviation in results for pressure increases by 0.8% if expressed as charging efficiency and decreases by 0.66% if converted to A/F. This is quite low when compared to the amplitude of cyclic variation in measured local A/F, and sufficient accuracy to enable evaluation can therefore be considered to have been achieved.

Figure 19 shows simulation results for the A/F around the spark plug and a cyclic average of measurement results. The error bars in the figure show the cyclic variation in local A/F shown in Fig. 10. The calculated local A/F is within the range of cyclic variation, and the trend for mixture concentration also corresponds well, indicating that the A/F around the spark plug has been calculated with good accuracy.

Figure 20 shows the calculated in-cylinder A/F distribution. There is considerable spatial variation in A/F in the case of injection on the Primary side only. The

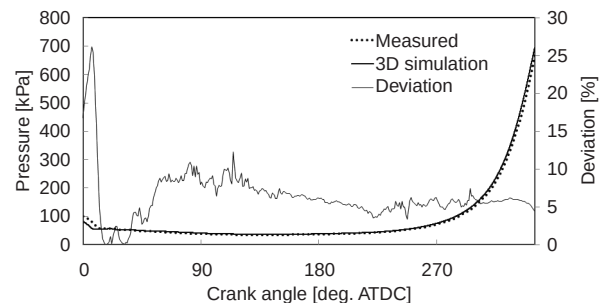


Fig. 18 Comparison between measured in-cylinder pressure and in-cylinder pressure calculation result

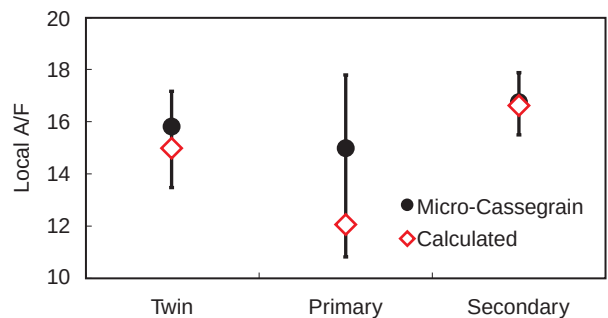


Fig. 19 Comparison between local A/F using micro-Casagrain sensor and calculation result

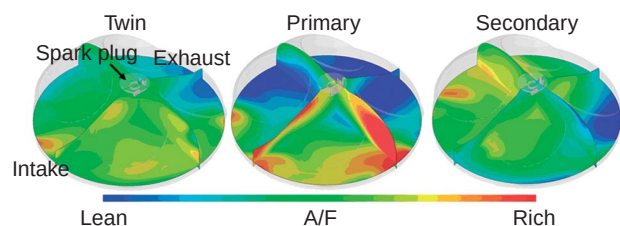


Fig. 20 Three-dimensional A/F distribution

reason that the error bar - the variation in each cycle - is larger for injection on the Primary side only than for the other injection conditions in Fig. 19 can be conjectured to be the fact that the spatial distribution of the air-fuel mixture concentration is greater than it is for the other injection conditions, and there is a minute change in the flow pattern in each cycle, resulting in a greater amplitude of variation in local A/F. Because of this, it is considered that the range and distribution of the air-fuel mixture flow around the spark plugs at the ignition timing would be effective as control parameters in reducing cyclic variation.

6. Conclusion

The A/F around the spark plugs was focused on as a factor in cyclic variation in gasoline engines, and the accuracy of a three-dimensional simulation was evaluated, with the following results:

- (1) A micro-Cassegrain sensor was used to measure the A/F around the spark plugs in a vehicle-mounted engine, previously regarded as a challenge, and the results were virtually identical to results obtained using conventional measurement methods such as UEGO sensors.
- (2) A comparison of the results for the A/F around the spark plug obtained from the micro-Cassegrain sensor and the three-dimensional simulation showed that the calculation results were within the range of cyclic variation, and there was a good correlation between the results for different injection conditions.
- (3) The three-dimensional simulation is judged to be an effective tool for the analysis of the behavior of the air-fuel mixture around the spark plugs and in the surrounding area based on cyclic variation.

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Mechanism of Torque Capacity Change during Operation of AT with Focus on Friction Properties

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ABSTRACT

The project discussed in this paper focused on the frictional characteristics of clutch facing materials in order to elucidate the mechanism by which the torque capacity of the multiple-plate clutches used in automatic transmissions (ATs) varies with their mode of use. The frictional characteristics of the facing materials are affected by the temperature, contact pressure, and sliding velocity of the friction surfaces. The temperature of the clutch plate, sliding velocity of the friction surfaces, and degree of misalignment of the gears transmitting torque were therefore measured in an AT during torque transmission. Next, a dynamic behavior analysis model, able to simulate the flow of torque transmission in the AT, was constructed on the basis of these measurement results. The results showed that the torque capacity of an AT operating at a constant temperature is affected by radial components of the friction force vectors on the friction surfaces.

1. Introduction

Contemporary increases in engine power have led to a demand for automatic transmissions that display high torque capacity while remaining lightweight and compact⁽¹⁾. Honda ATs feature a compact design that positions a multiple plate clutch for each speed on three or four parallel shafts⁽²⁾. A smooth shift response with minimal shift shock is realized by direct control of the piston pressure of the clutches that are engaged when the gears are shifted. However, the parallel shaft configuration limits the diameter of the clutch; heat production per unit area of the disks is controlled in order to increase durability by maximizing the difference between the internal and external diameters of the disks⁽³⁾. Because of this, it is important to control disk contact pressure distribution and plate temperature distribution when the clutch is engaged. As has previously been reported⁽⁴⁾, a simulation technology has been developed that is able to quantify the disk contact pressure distribution and plate temperature distribution, enabling the wear history of the clutch to be predicted based on the wear properties of the facing material.

Torque capacity is increased by increasing the number of disks in the clutch. The clutch unit is generally

designed using a simple torque capacity equation. However, this equation was formulated for conditions of uniform contact pressure distribution on each disk and an identical coefficient of friction at all points on the surfaces of the disks during clutch engagement. As previously reported⁽⁴⁾, the disk contact pressure distribution in a clutch unit is not uniform. The friction properties are known from the results of SAE#2 performance tests and other results to be dependent upon the temperature, the contact pressure, and sliding velocity of the disk surfaces⁽⁵⁾. For these reasons, average values for the disk contact pressure and coefficient of friction are employed in the equation used to calculate torque capacity.

In addition, it is known from experience that a variety of factors, including the inclination of the drive gears and the flexibility of the shafts, affect the torque capacity of a clutch fitted in an AT. However, these effects vary depending on the layout, friction properties and operating conditions of the AT, among other factors, and it is therefore challenging to consider them using a clutch unit equation. For this reason, the torque capacity of ATs is evaluated exclusively by manufacturing test transmissions and conducting torque capacity tests.

The research discussed in this paper focused on the friction properties of the clutch facing materials. A

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dynamic behavior analysis technology that is able to simulate the torque flow to each clutch during AT operation using these friction properties was developed. This enabled the mechanism by which the direction of friction force on the friction surfaces of the clutches results in changes in torque capacity during AT operation to be clarified.

2. Development Goals

Figure 1 shows the main section of the AT with a 5th gear used in the analysis. As the figure shows, the clutches and gears for each speed are positioned on parallel shafts, and shifting from clutch to clutch in the shifting process is enabled by controlling the oil pressure in oil channels located inside the rotating shafts.

Figure 2 shows the results of measurement of the torque capacity of the clutch. During the measurements, a constant temperature, T_0 , was maintained in the interior of the transmission, including the lubricant oil and clutch plates, and the piston thrust was also constant. The horizontal axis of the graph shows the torque input to the clutch, and the vertical axis shows the rotational speed differential between the disks and the plates. Torque capacity was defined as the torque input when the rotational speed differential reached a specified value. It was necessary for this specified value to be sufficiently small for the cooling effect of the lubricant oil to help prevent the temperature of the plates from changing. As the graph indicates, torque transmission increases as the rotational speed differential increases. This phenomenon is conjectured to be influenced by the friction properties of the clutch facing materials themselves.

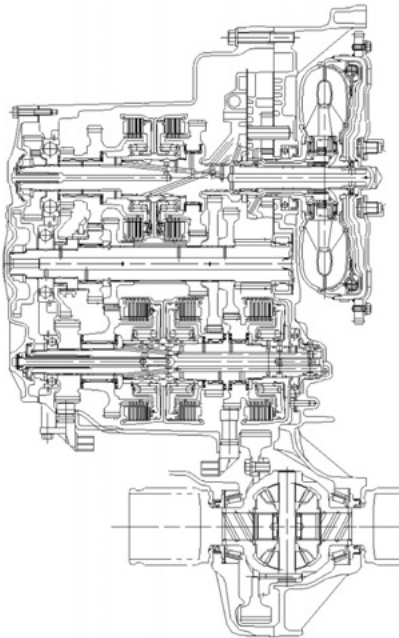


Fig. 1 Main section

Figure 3 shows the clutch configuration. The 4th and 5th clutches were identical clutches made up of four disks. The back surfaces of the guides of each clutch were welded together. The same facing material was used in both clutches. However, when the torque capacity of the clutches was measured, it was found that the torque capacity of the 4th clutch was higher. Given that there were no differences in the configurations of the individual clutch units, it was conjectured that this phenomenon was due to a difference in the torque flow in the transmission. If a clutch displays insufficient torque capacity, slipping occurs between the disks and the plates, and the rotational speed differential increases. If this rotational speed differential becomes too high, the configuration or layout of the clutch may need modification, and this can delay development.

Based on these considerations, the goal of the research project discussed in this paper was established as clarification of the mechanism by which torque capacity changes during AT operation.

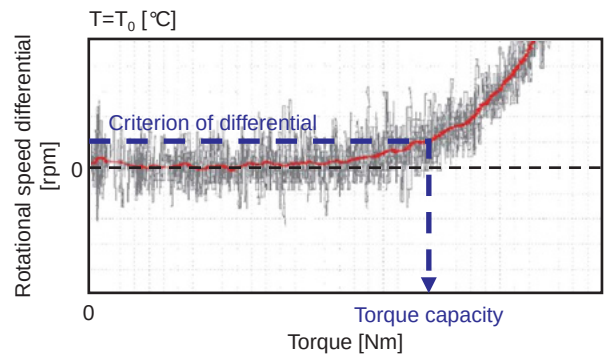


Fig. 2 Torque capacity measurement results

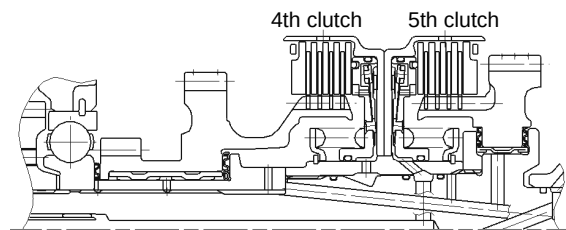


Fig. 3 Detail of clutch

3. Mechanism of Change in Torque Capacity

3.1. Differences in Torque Flow

This section will consider the mechanism causing torque capacity to change in the 4th and 5th clutches that formed the focus of this study during actual AT operation. The sectional shape of the gears and their method of mounting can be indicated as factors producing differences

in torque flow. The 4th gears are supported by long needle bearings, and this configuration makes them less prone to misalignment than the 5th gears when loads are being transmitted by the helical gear sections. The degree of misalignment of the 4th and 5th gears was therefore measured using a gap sensor positioned in an AT [Fig. 4(a)]. The degree of misalignment is clearly lower in the 4th clutch, in which torque capacity is higher [Fig. 4(b)].

3.2. Hypothesized Mechanism of Change in Torque Capacity

This section will consider the mechanism by which gear misalignment results in a decline in torque capacity. Plate temperature, disk contact pressure, and sliding velocity are assumed to be constant.

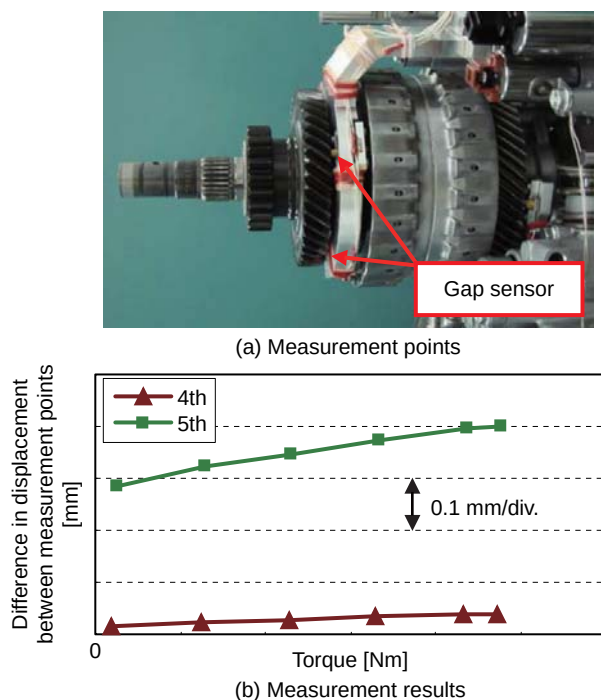


Fig. 4 Comparison of gear displacement

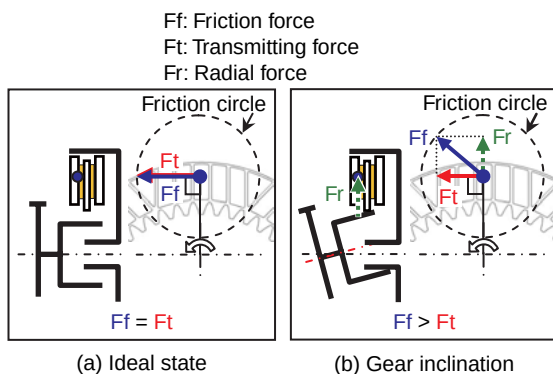


Fig. 5 Hypothesized mechanism of change in torque capacity

A friction circle drawn around one point on the friction surface of the clutch (Fig. 5) will be considered first. If the radius of this friction circle is considered as friction force, F_f , assuming that the coefficient of friction is constant, the size of the friction circle will not change irrespective of the direction of F_f . When the system is in an ideal state with no gear misalignment [Fig. 5(a)], the disks and the plates rotate on the same axis, and F_f acts in a tangential direction. The entire value of F_f can therefore be considered to contribute to torque transmission. Considering next the case of gear misalignment [Fig. 5(b)], when the gears are misaligned, a radial load, F_r , is generated where the gears and the disks mesh. This is transmitted to the friction surface, and depending on the magnitude of F_r , F_f can incline in a radial direction. When this happens, because the size of the friction circle does not change, the torque force, F_t , which is the component of F_f acting in the tangential direction, becomes lower than F_f . This is believed to be the mechanism by which the radial load on the disk increases and torque capacity declines as a result of misalignment of the gears.

4. Torque Capacity Simulations

The measurement of the magnitude and direction of the friction forces on the clutch friction surfaces during AT operation in order to prove the foregoing hypothesis would represent a challenge. Dynamic behavior analysis models, which modeled the flow of torque transmission in a clutch in an ideal state with no gear misalignment and the 5th clutch were therefore constructed, and results were compared.

4.1. Construction of AT Clutch Torque Capacity Analysis Models

Figure 6 shows the dynamic behavior analysis model constructed to simulate the torque flow in the 5th clutch of the AT. Care was taken to reproduce the actual clearance of the needle bearings that support the gears. With regard to load conditions, because the clutch is positioned on the M shaft, the right extremity of the M

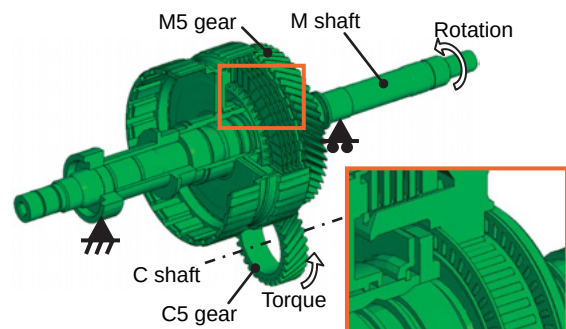


Fig. 6 Torque capacity prediction model for 5th clutch

shaft was forced to rotate, gradually increasing the torque load on the 5th driven gear, C5, positioned on the C shaft. During this process, the piston thrust was held constant. As the torque load increased, a sliding speed differential began to develop between the disks and the plates.

4.2. Friction Properties of Clutch Facing Materials

SAE#2 test equipment and LVFA test equipment can be used to measure the friction properties of the clutch facing material. However, the facing material is immersed in lubricant oil for measurement by these test equipments, preventing measurement of the effect of the amount of lubricant oil; it is also challenging to heat the entire facing material to a high temperature and to maintain a constant temperature. The torque transmission characteristic test apparatus, shown in Fig. 7, was therefore newly developed. The design of the apparatus incorporates uniform disk contact pressure, a lubricating system that supplies oil via the shaft, as used in an AT, and a configuration that enables simple control of temperature. This enables the torque transmission characteristics to be measured while holding the temperature and contact pressure of the disk surfaces and the rotational speed differential constant.

Figure 8(a) shows the clutch facing material friction properties obtained using this test apparatus with the

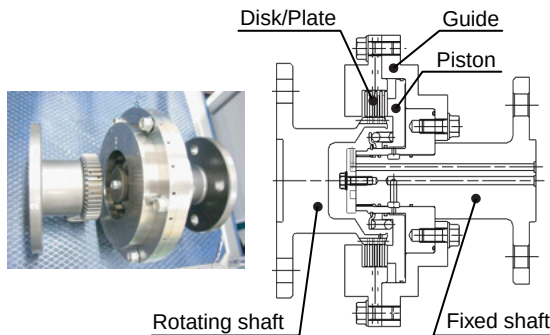


Fig. 7 Test apparatus for torque transmission characteristics

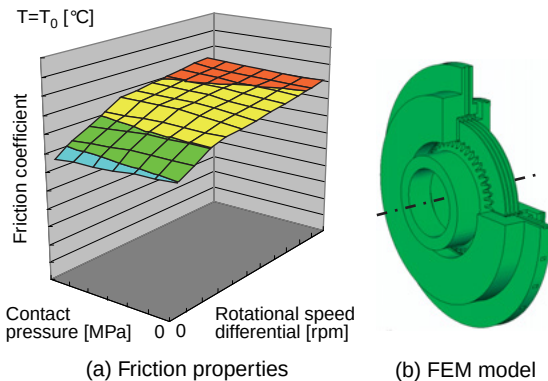


Fig. 8 Identification of friction properties

temperature held constant at T_0 . The vertical axis shows the coefficient of friction, and the horizontal axis shows the disk contact pressure and sliding velocity. The coefficient of friction during the test was identified using a structural analysis model [Fig. 8(b)] which modeled the test apparatus. The friction properties of the facing material discussed here were not significantly affected by changes in contact pressure, but a positive correlation with sliding velocity was obtained.

Figure 9 shows the results of simulation and measurement of the torque capacity of the 5th clutch, using these friction characteristics. The results show that the simulation was able to reproduce the qualitative tendency, in which rotational speed differential increased as the torque load increased.

The plate temperature for these results is the constant temperature T_0 .

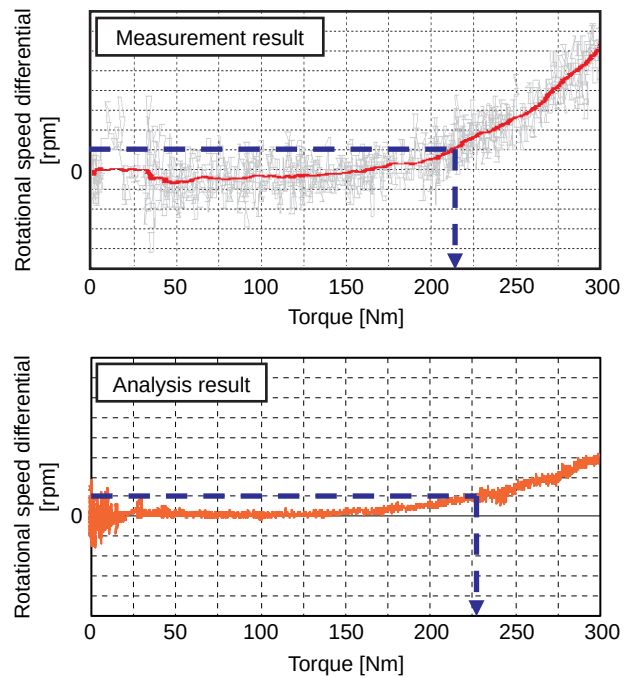


Fig. 9 Comparison of torque transmission by 5th clutch

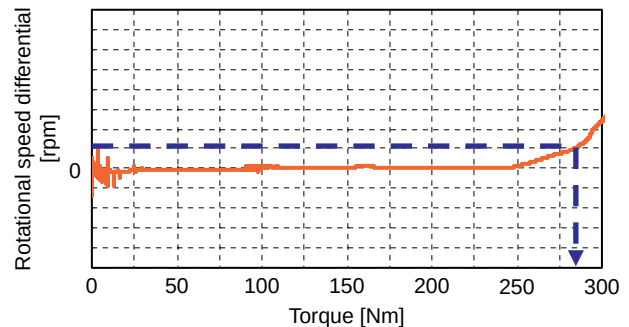


Fig. 10 Torque capacity of clutch without gear misalignment

4.3. Torque Capacity of Clutch with No Gear Misalignment

Next, the clutch in an ideal state with no gear misalignment was analyzed. In this analysis, the 5th driven gear, C5, was removed from the 5th clutch dynamic behavior analysis model, with the torque load applied directly to the 5th drive gear, M5. Figure 10 shows the results for the rotational speed differential. The results clearly show that torque capacity was higher in the clutch without gear misalignment than in the 5th clutch.

4.4. Friction Force Vectors on Surface of Disk

Figure 11 shows the friction force vectors acting on the surfaces of the disks of the 5th clutch and the clutch without gear misalignment when transmitting identical torque. In the clutch without gear misalignment, the friction force vectors all act in the direction of rotation, while in the 5th clutch there are also friction force components acting simultaneously in the radial direction. Because both clutches are transmitting the same amount of torque, the friction force components acting in the direction of rotation are identical, but the total friction force is higher in the 5th clutch because of the radial components. As a result, the increase in the rotational speed differential is higher in the 5th clutch than in the clutch without gear misalignment, and thus transmits torque at a higher coefficient of friction.

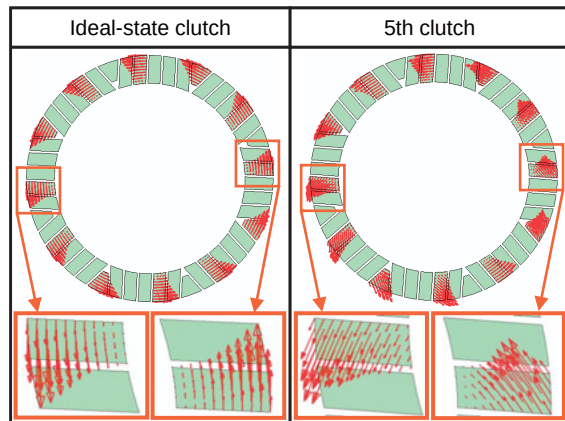


Fig. 11 Direction of friction force on disk surface

5. Torque Capacity during AT Operation

This section will consider the degree of gear misalignment and the friction force vectors for the 4th and 5th clutches. As Fig. 12 shows, the degree of misalignment of the drive gears is lower in the 4th than in the 5th clutch. This means that the radial load transmitted by the drive gears via contact with the disk teeth is also lower in the 4th clutch (Fig. 13). As a result, the radial components of the friction force vectors of the

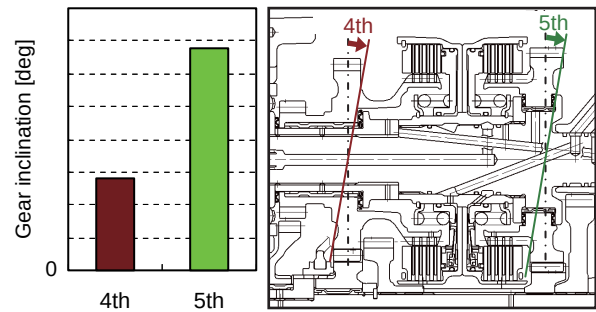


Fig. 12 Comparison of degree of gear misalignment in 4th and 5th clutches

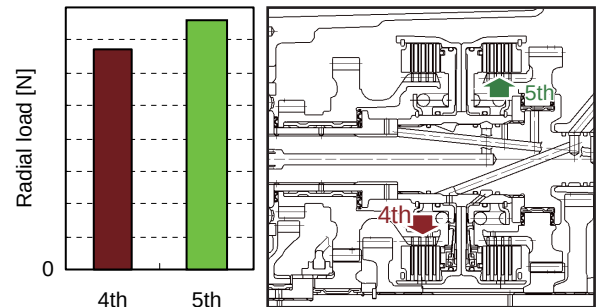


Fig. 13 Comparison of radial load in 4th and 5th clutches

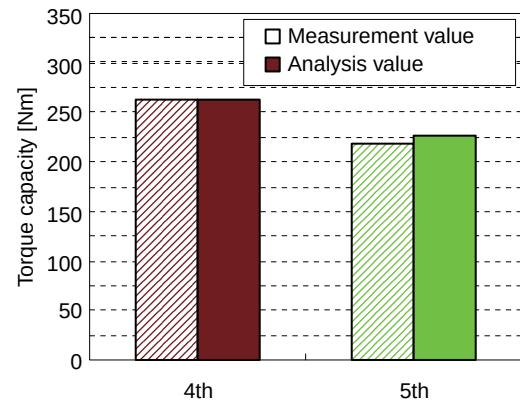


Fig. 14 Torque capacity of 4th and 5th clutches

4th clutch are smaller, and its torque capacity is therefore higher than the 5th clutch. Figure 14 shows simulation results and measurement results for torque capacity in the 4th and 5th clutches during AT operation.

6. Conclusion

Measurements of gear misalignment and dynamic behavior analysis models that reproduce the flow of torque transmission have been used to enable clarification of the mechanism of changes in torque capacity in an operating AT. The following results were obtained:

- (1) Study of clutch friction vectors showed that the torque capacity of an AT at a constant temperature is not only affected by friction force components acting in the direction of rotation, but by simultaneously occurring friction force components acting in the radial direction, and that as these friction force components become higher, torque capacity declines.
- (2) Reproduction of misalignment in the clutch drive gears, treating the friction properties of the clutch facing material as dependent on the temperature, contact pressure, and sliding velocity of the friction surfaces, enables prediction of torque capacity during AT operation.

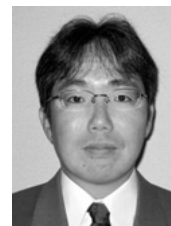
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Research on Longitudinal Vibration of MT Vehicle at Clutch Engagement

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ABSTRACT

Formulas have been developed to describe the longitudinal vibration that occurs when the clutch engages in an MT vehicle, and the vibration waveforms have been used to formulate evaluation indices that possess a high degree of correlation with sensory evaluations.

The parts that contribute to longitudinal vibration of the vehicle when the clutch engages were determined, and a simulation of the first-order vibration of the torsion system of the body and drivetrain was constructed, enabling specifications to be evaluated at the initial stage of vehicle development, when no prototype vehicle has yet been constructed.

The use of the new method in the development of a production vehicle enabled early determination of specifications, such as driveshaft stiffness, and the application of control to rapidly reduce engine speed, enabling effective responses to be proposed.

1. Introduction

Smooth running when shifting is an important aspect of performance for an MT vehicle.

However, when the vehicle gives rise to transient input to the drivetrain, it excites resonance and subsequent longitudinal vibration of the vehicle occurs. In this case, a driver would feel uncomfortable and find it challenging to drive smoothly.

Pressing down on the accelerator pedal is one of the operations that introduce transient input to the drivetrain. Past research⁽¹⁾⁻⁽³⁾ has enabled quantitative evaluation of the longitudinal vibration of the vehicle when the accelerator pedal is pressed down. In addition, simulations enable specifications to be examined from the initial stages of vehicle development in relation to this form of vibration.

However, the study of specifications in relation to the longitudinal vibration of the vehicle when the clutch is engaged relies on sensory evaluation and experience in most cases. In addition, because simulation techniques in relation to this form of vibration are as yet insufficiently developed, vehicle designers are unable to decide on specifications in the initial stages of development.

This research project therefore set out to develop

formulas to enable numerical evaluation of the longitudinal vibration generated by clutch engagement, and to construct a simulation model of the phenomenon. This paper will report the details of this project.

2. Development Goals

Refining vehicle specifications as much as possible in the initial stages of the development process, thus reducing the degree of trial and error during vehicle tests, is an important factor in enabling more effective countermeasures to be adopted during vehicle development.

Changes to specifications following completion of the prototype have a large effect on other aspects of performance, and as a result in many cases insufficient changes are made. This makes it challenging to efficiently develop countermeasures relative to the person-hours spent on the project.

The research discussed in this paper therefore established the following main objectives:

- (1) To develop a method of numerical evaluation that is able to replace sensory evaluation
- (2) To formulate a simulation that is able to be employed from the initial stages of development

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3. Formularization of Evaluation Indexes

In order to formulate evaluation indexes, the longitudinal vibration of the vehicle at clutch engagement was first formularized using equations of motion.

To enhance ride comfort, it is necessary to establish evaluation values for longitudinal vibration based on sensory evaluations of the driver's feeling of comfort. ISO 2631⁽⁴⁾ has been established as an international standard for exposure to whole-body vibration, providing an international standard for automotive ride comfort. It is known that humans are sensitive to 4-8 Hz vertical vibration and 1-2 Hz horizontal vibration.

A variety of methods of evaluation in relation to vibration and ride comfort have been proposed⁽⁵⁾⁻⁽⁹⁾, but the majority use a restricted set of conditions, such as relying on laboratory test equipment, and do not achieve a sufficient correlation with objective evaluations of ride comfort in actual vehicles. These methods have therefore not been applied to vehicle development. The project discussed here isolated the physical factors that represented the main components of sensory evaluations in actual vehicle driving tests and elucidated the correlation between these physical factors and physical equations describing vehicle behavior, in order to develop a numerical evaluation method that is able to be applied in the vehicle development process.

Here, the physical equations describing vehicle behavior will be considered first. The equations will be considered using a model for the evaluation of the magnitude of impact, of the type shown in Fig. 1. A sphere impacts with the vehicle, and stops inside the vehicle. The momentum, which indicates the magnitude of the impact, is

$$p = m \cdot v \quad (1)$$

If the sum of the forces imparted to the vehicle at the instant of impact is termed P , and the impact is considered to occur instantaneously at time $t=0$, then the change in momentum after the impact can be expressed as

$$\int_{0^-}^{0^+} dp = \int_{0^-}^{0^+} P(t) dt \quad (2)$$

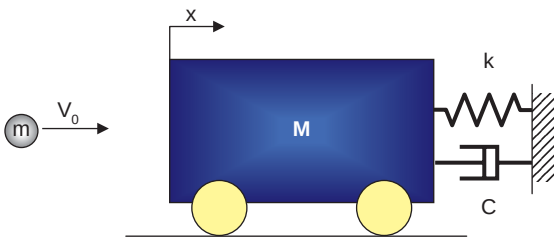


Fig. 1 Impact vibration model

If the initial momentum of the sphere is termed p_0 , the impact received by the vehicle is

$$-P = p_0 \quad (3)$$

The vehicle undergoes displacement x_0 as a result of the impact. The equation of motion for damped free vibration is

$$\frac{d^2x}{dt^2} + 2h\omega \cdot \frac{dx}{dt} + \omega^2 x = 0 \quad (4)$$

If the solution is assumed to be $x = \exp^{\lambda t}$ from the theory of differential equations, then the envelope curve for damped free vibration is

$$x(t) = \left(\frac{1}{\omega \sqrt{h^2 - 1}} \cdot \frac{dx_0}{dt} \right) \cdot \exp^{-h \cdot \omega \cdot t} \quad (5)$$

Nomenclature

p : Impact power [N]

m : Mass [kg]

v : Velocity [km/s]

P : Reaction force [N]

M : Mass [kg]

c : Damping property [Nms/rad]

k : Torsional stiffness [Nm/rad]

ω : Angular velocity [rad/s]

x : Displacement [m]

By this means, an evaluation method was able to be formularized based on a simple model.

4. Test Method

Next, a test method was developed and indexes were formulated based on the concepts that were formularized in the preceding section.

4.1. Test Method

The frequency of the longitudinal vibration generated by shifting differs depending on the gear ratio, and the frequency becomes higher the further the gear position moves towards the high-speed range. The magnitude of vibration is higher and the decline in ride comfort greater for shifting immediately following acceleration than for shifting during deceleration.

The 3 Hz longitudinal vibration generated when shifting up from 1st to 2nd gear is a particular issue. Figure 2 shows measurements of this vibration.

In order to determine the mechanism of production of longitudinal vibration during shift, torque and vibration were measured during shifting in test vehicles with sudden clutch engagement under conditions of a fixed predetermined engine speed and fully-closed accelerator. Measurements were conducted in more than ten types of test vehicles, changing in turn shift position, engine speed, and the degree of accelerator opening.

4.2. Formulation of Indexes

In order to formulate evaluation indexes, subjective evaluations were conducted in which a large number of drivers compared test drives in commercial vehicles. Indexes enabling numerical evaluation were formulated by means of comparative analysis of the results of these subjective evaluations and measurement results.

The drivers felt the sudden vibration peak when the clutch was engaged as a kick, and the subsequent counter-reaction as jerks. Figure 3 shows two evaluation indexes

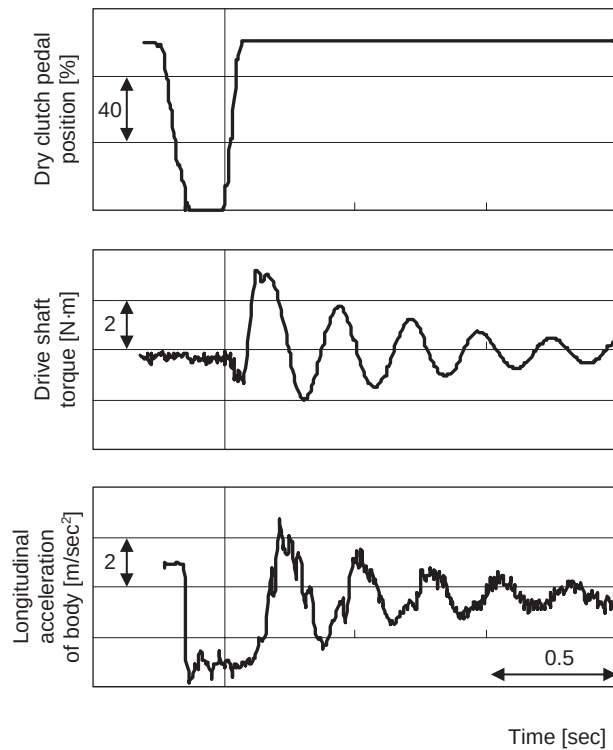


Fig. 2 Time history of body vibration by clutch engagement shock Closed accelerator, 1st-2nd gear, 3000 rpm

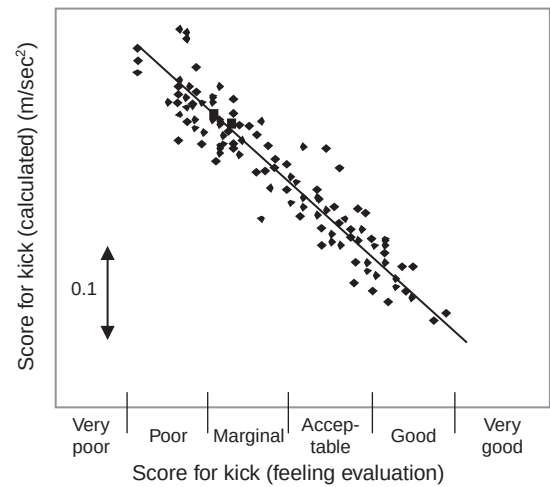


Fig. 4 Comparison of measured and calculated kick score

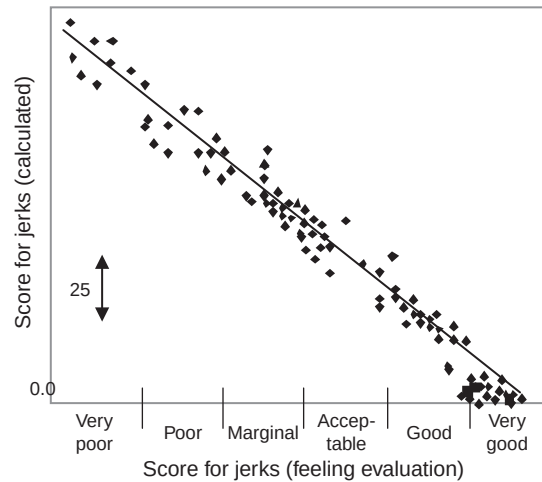


Fig. 5 Comparison of measured and calculated jerks score

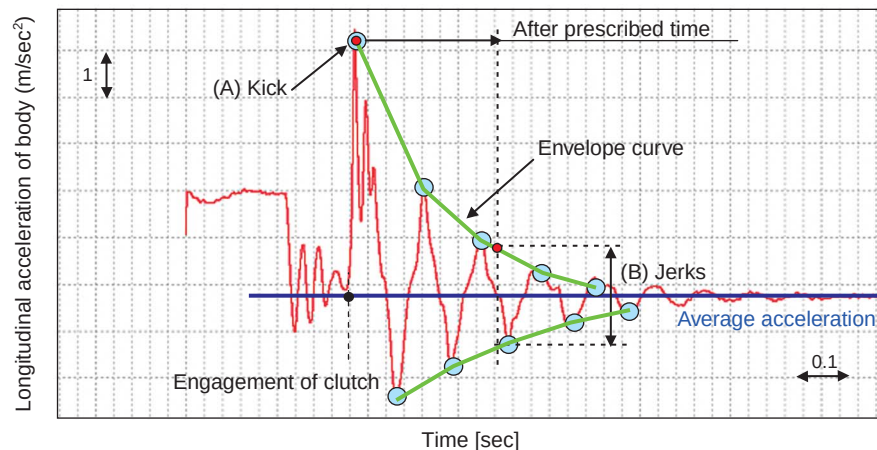


Fig. 3 Definition of index

formulated by quantifying these subjective drive feelings based on the results of a questionnaire survey. The equations used to calculate the respective numerical indexes were formulated from Eqs. (3) and (5) as follows:

(a) Kick: Correlated with sum of impact forces

The higher the mass, m , and the degree of change in speed, $dv(t)/dt$, the greater the impact forces. Because m is a value unique to the specific vehicle, the index has been formulated as the change in speed per unit weight to enable general application.

$$Kick = \frac{1}{m} \cdot \frac{dv(t)}{dt} \quad (6)$$

(b) Jerks: Correlated with the degree of change in the envelope curve of the damped free vibration

The higher the index, $h \cdot \omega$, the better the damping performance of the vehicle. To enable general application, an index formulated by making the initial amplitude of the envelope curve constant was selected for use as the index.

$$Jerks = h \cdot \omega \quad (7)$$

Figures 4 and 5 show the results of verification of the correlation between the two numerical evaluation indexes, formulated on the basis of sensory evaluations in test vehicles, and from measurement data based on the equations discussed above. The good correlation indicates the validity of the indexes.

5. Formulation of Simulation Model

The transient torque generated by the engine speed differential while shifting is an important factor in reproducing the longitudinal vibration of the vehicle resulting from clutch engagement. This torque produced by the engine speed differential during shifting was reproduced by adding a clutch operation element to a conventional body-drivetrain first-order torsional vibration mechanism analysis simulation model.

Figure 6 shows an outline of the entire model. Figure

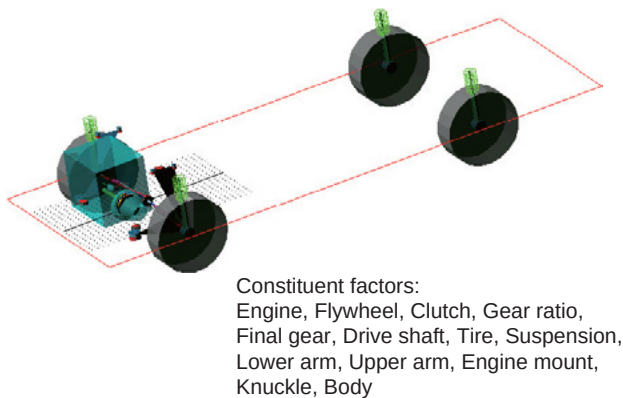


Fig. 6 Analysis Model

7 shows an outline of the clutch model. Clutch disks are positioned between the crankshaft and the clutch hub, and the torque capacity and clutch stroke are defined at the plates. The torque capacity is linked to the clutch stroke, and is transmitted to the transmission side as the amount of transmitted torque changes. In addition, the inclusion of the amount of play in the clutch pedal to the clutch stroke elements has enabled reproduction of the torque fluctuation resulting from the engine speed differential generated by operation of the clutch in a real vehicle. The formulation of a model in which clutch capacity can be varied in accordance with clutch operation has also

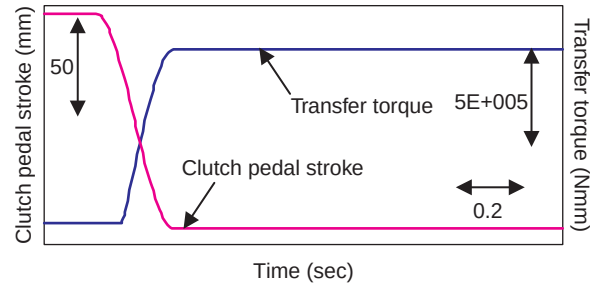


Fig. 7 Clutch model characteristic

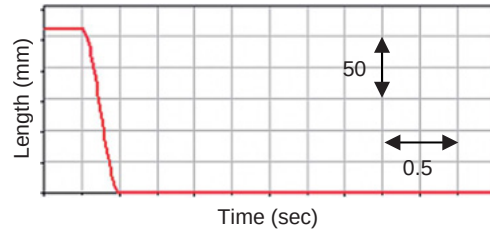


Fig. 8 Clutch operation

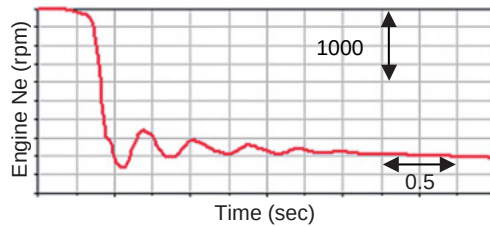


Fig. 9 Engine rotation

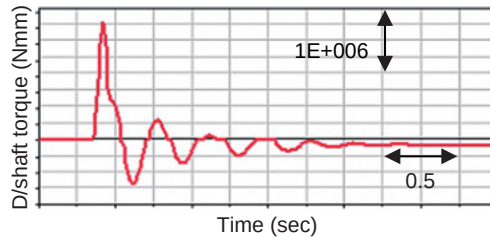


Fig. 10 Drive shaft torque

enabled reproduction of the effect on the kick feeling of differences in the time taken for operation of the clutch.

Figures 8 through 10 show simulation results for the drivetrain when shifting up from 1st to 2nd gear. As Fig. 8 shows, engine speed is absorbed by time series changes in the clutch stroke (clutch engagement), and as Fig. 9 shows, an engine speed differential is generated in response to the geometric ratio of 1st and 2nd gear. Torque is generated by the engine speed differential that is absorbed at this time, and is transmitted to the driveshaft, as Fig. 10 shows.

Design values were employed as the characteristics data for each of the simulation elements discussed above.

With regard to the vehicle body system model, consideration was given to nonlinear characteristics, and table functions were employed to provide static load-displacement characteristics for the engine mounts, suspension, longitudinal compliance, clutch, and driveshaft.

Figure 11 shows a comparison of simulation results for longitudinal vibration when shifting up from 1st to 2nd gear with measurement data from an actual vehicle.

The simulation has reproduced the first vibration peak and the following jerk vibrations.

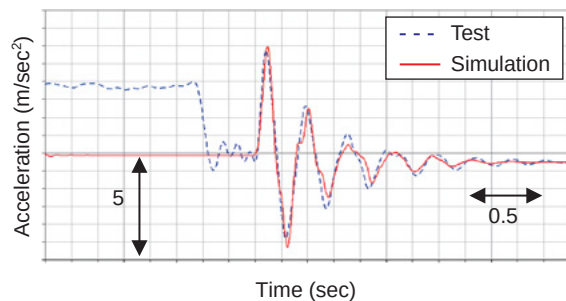


Fig. 11 Results of verification of simulation

6. Example of Application in Vehicle Development

6.1. Degree of Effect of Vehicle Body System

The developed simulation incorporating elements for clutch operation was employed in the initial stages of a vehicle development, and a parameter study of each component affecting the longitudinal vibration of the vehicle at clutch engagement was conducted. The results are shown in Figs. 12 and 13.

The effect of the geometric ratio between 1st and 2nd gear, the compliance bush characteristics, and the engine mount characteristics on the kick feeling was highest. The effect of the torsional stiffness of the driveshaft on the jerk feeling was highest.

Focusing on these areas that had a strong effect on kicks and jerks, a simulation was conducted in order to

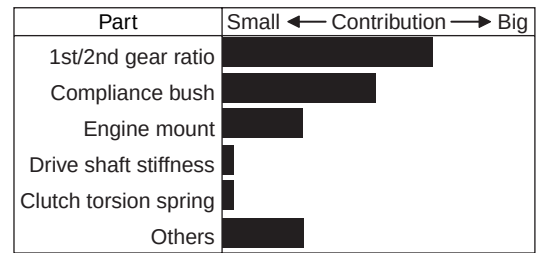


Fig. 12 Contribution to kick of body and drivetrain parts except tires by ADAMS simulation

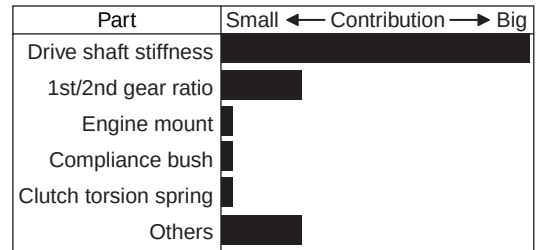


Fig. 13 Contribution to jerks of body and drivetrain parts except tires by ADAMS simulation

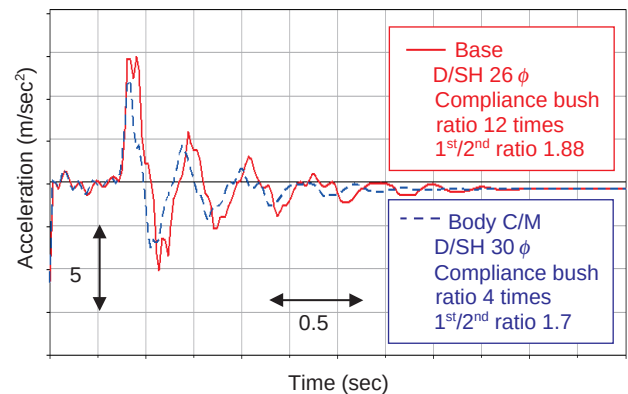


Fig. 14 Effect of body and drivetrain countermeasures by ADAMS simulation

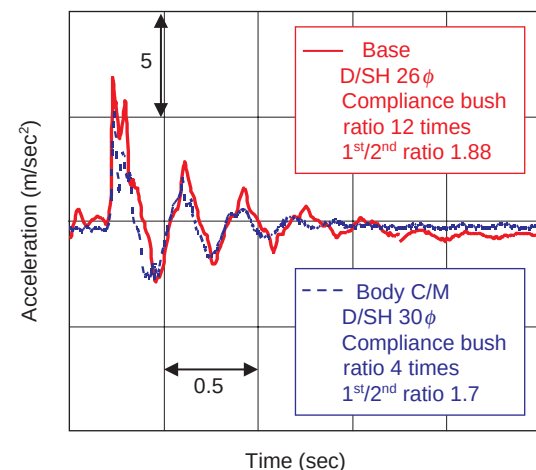


Fig. 15 Effect of body and drivetrain countermeasures in actual car

determine specifications that would reduce both kicks and jerks to an acceptable level. The results are shown in Fig. 14. The results of applying countermeasures in an actual vehicle are shown in Fig. 15. The kick feeling and the jerk feeling when shifting from 1st to 2nd gear have been reduced, as predicted by the simulation.

6.2. Engine Synchronizer Control

Figures 16 and 17 show the results of a simulation in which the engine speed differential was reduced by approximately 50% at clutch engagement as a measure to reduce the kick feeling when shifting from 1st to 2nd gear. The results show a reduction of approximately two-thirds in the first vibration peak occurring in the vehicle.

Figures 18 and 19 show the results of a simulation in which engine torque is imparted in the opposite direction to the direction of the torque at the driveshaft⁽¹⁰⁾,⁽¹¹⁾ as a measure to reduce the jerk feeling when shifting from 1st to 2nd gear. In these results, there is a reduction in the second and following vibration peaks produced in the vehicle body.

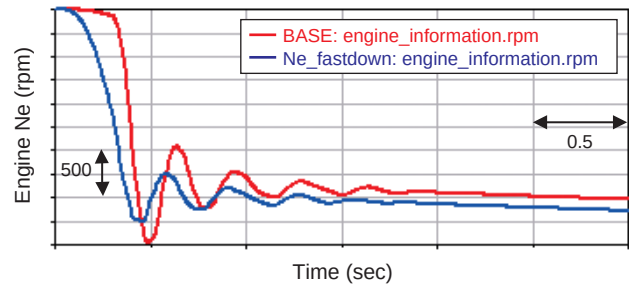


Fig. 16 Engine Ne control – Engine Ne

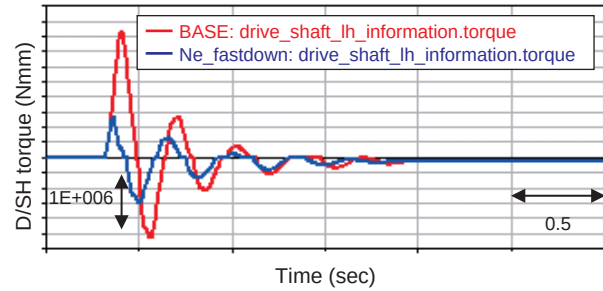


Fig. 17 Engine Ne control – D/SH torque

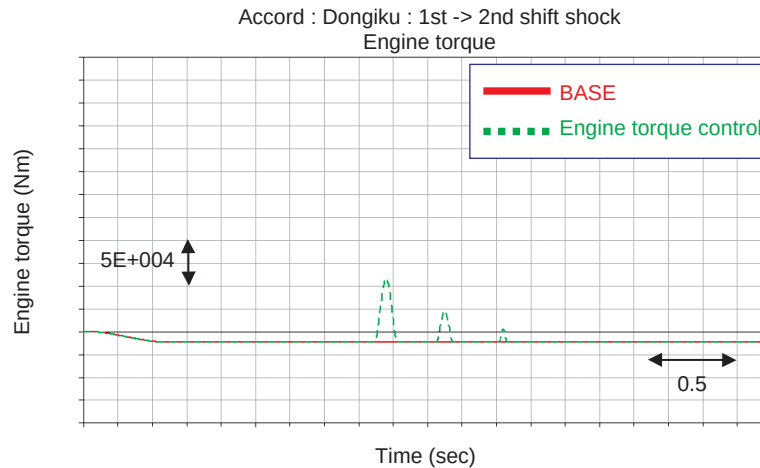


Fig. 18 Engine torque control – Engine torque

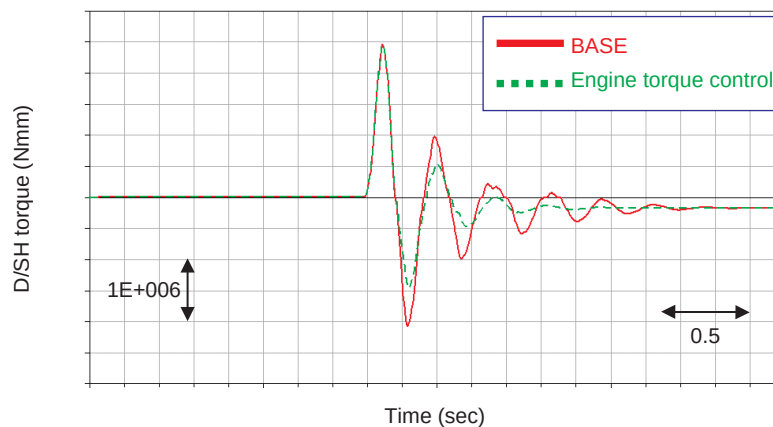


Fig. 19 Engine torque control – D/SH torque

Figure 20 shows the engine control method, termed synchronizer control, employed in the actual vehicle on the basis of the simulation predictions discussed above.

Deceleration kick is an issue when the accelerator pedal is suddenly released after it has been pressed down to accelerate the vehicle. In this case, it is necessary to reduce engine speed as slowly as possible.

However, it is necessary to reduce engine speed as quickly as possible in order to reduce the longitudinal vibration produced by engagement of the clutch. In order to reconcile these contradictory demands, as a measure

against kicks, a clutch sensor was included in the vehicle specifications, and control is applied to reduce engine speed rapidly only during shifting. Giving consideration to the robustness of vehicle characteristics, ignition timing control is applied for one cycle only as a measure against jerks.

Figure 21 shows the results of employing synchronizer control in a test vehicle. As predicted by the simulation, the control has reduced the kick feeling and jerk feeling when shifting from 1st to 2nd gear.

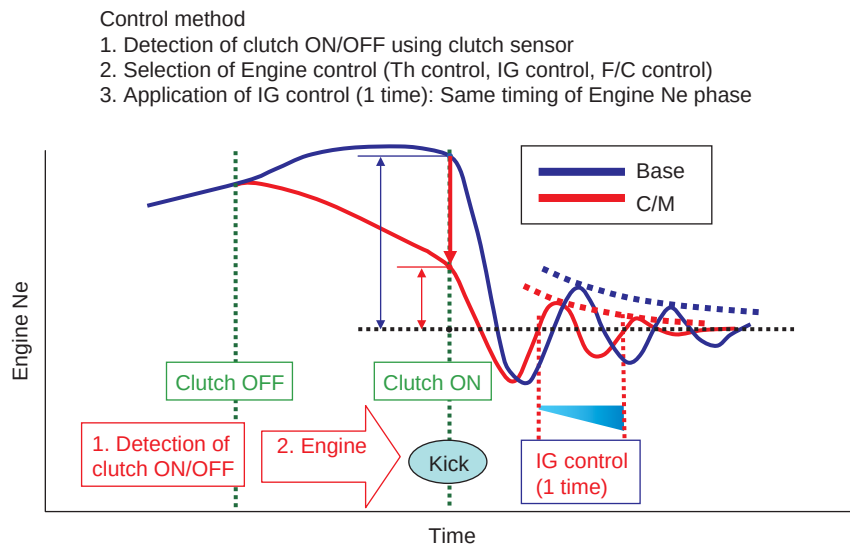


Fig. 20 Synchronizer control concept

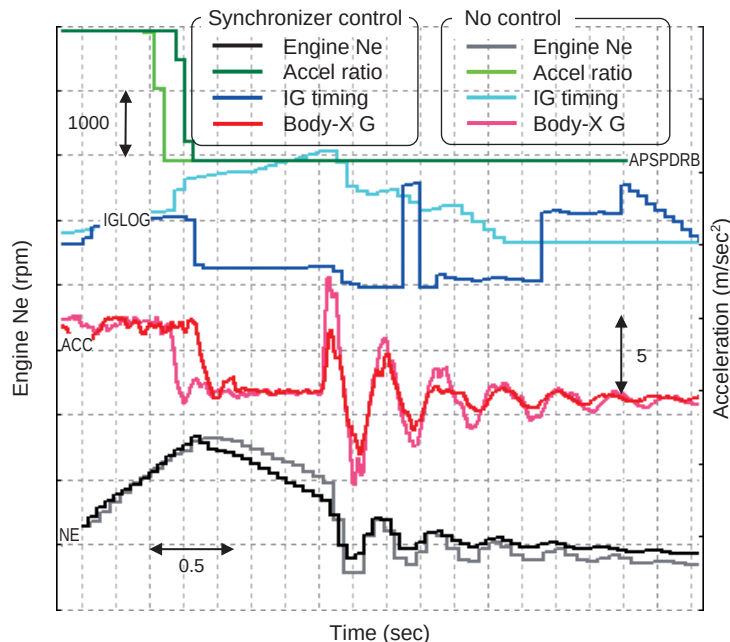


Fig. 21 Effect of synchronizer control

7. Conclusion

Research was conducted on the longitudinal vibration in an MT vehicle at clutch engagement. This research produced the following results:

- (1) A method enabling numerical evaluation of kick feeling and jerk feeling and able to replace sensory evaluation of vehicle body vibration at clutch engagement was developed.
- (2) A simulation model was formulated that displayed good correlation with results obtained from actual vehicles.
- (3) These developments have enabled study of specifications at the initial stages of vehicle development, when no prototype vehicle exists.

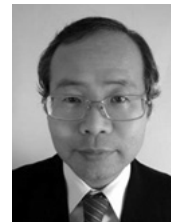
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Method for Three-dimensional Evaluation of Surface Form of Metal Rings for CVT Metal Pushing V-belts

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ABSTRACT

In order to respond to the increased amounts of torque being transmitted by the metal pushing V-belts in CVTs, it was necessary to increase precision in the formation of the metal rings employed in the belts. This necessitated the achievement of a higher level of precision in the surface forms of the rings, a requirement that demanded the development of a three-dimensional surface form measurement technology enabling quantitative evaluation of the manufacturing processes and manufacturing history of the rings. While there is a standard method of evaluating the precision of the surface form of the rings in the direction of width, no integrated method of evaluating the continuous sectional form in the direction of the circumference has been available. The developed method uses general purpose laser sensors to gather continuous data on the sectional form of the rings, enabling reproduction of the three-dimensional surface form of the rings around the entire circumference. This has enabled quantification of the cyclic deviations in surface form that occur during the manufacturing process.

1. Introduction

A change in specifications from a start clutch to a torque converter-type CVT amplifies torque at take-off and when the vehicle stalls, thus increasing the torque transmitted by the metal pushing V-belt. It is necessary to increase the accuracy of the shape of the rings used in the belts in order to stabilize their behavior when high tensile force is acting on them during torque amplification. This is because amplifying the dynamic behavior of the rings results in interference between the rings and the elements (Fig. 1), which is a factor producing wear of the ring edges. This makes it necessary to control the profile deviation in the rings, i.e., the difference between the design dimensions and actual shape of the rings in the direction of the circumference.

Conventionally, the profile deviation in the direction of width of the plate, i.e., across the entire circumference of the ring, has been controlled using a device that employs a measurement method that does not involve cutting the ring⁽¹⁾. Using this device, cyclic high-frequency form components originating in the accuracy of the roller employed in the ring rolling process can be evaluated, and a high level of accuracy achieved in the form of the ring in the direction of the width of the steel plate⁽²⁾. However,

the accuracy of the roller also contributes to the accuracy of the surface form of the rings, demanding a method of evaluating the three-dimensional surface form. In addition, interference of the jig during the application of the heat treatment used in the manufacture of the rings can prevent the treatment from being properly applied in some areas, changing the surface form of the rings. However, no measurement method has been available, and it has been challenging to measure and evaluate the surface form of the thin rings. The developed method of evaluating the form of the rings discussed in this paper uses a device

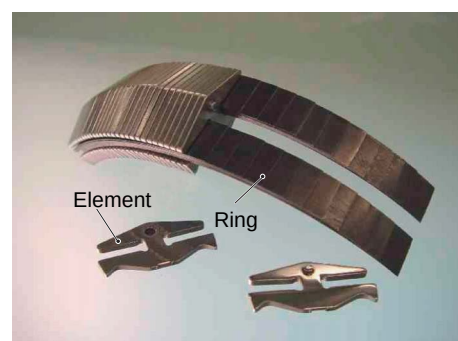


Fig. 1 Components of metal pushing belt

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made up of a general-purpose line laser sensor and simple positioning mechanism to measure the three-dimensional form of the ring surface. The application of a frequency analysis to the results enables the production history of the rings, over multiple processes, to be quantified as the degree of profile deviation of the ring.

With regard to the goal of achieving a high level of accuracy in the form of the rings, this method offers the advantage of enabling the efficient identification of the processes that produce profile deviations in the rings.

2. Mechanism of Occurrence of Profile Deviation in Rings

The processes that generate conspicuous profile deviations during ring manufacture are the rolling and heat treatment processes. Figure 2 shows a two-dimensional schematic diagram of the rolling process. The rings are formed by rolling at clearance f between a rolling roller and receiver roller. On a system of X-Y coordinates, with the axis of rotation of the rolling roller as o , the eccentricity of the axis, og , in relation to the center of the roller, g , is termed e , the initial angle of eccentricity is termed ϕ_0 , and the angle of rotation of the rolling roller is termed θ . With the X axis as the benchmark, the angle towards the Y axis is positive. If the ideal radius of the rolling roller is termed R , the profile deviation in the

direction of the radius can be termed ΔR , and the change in the clearance f , Δf , can be expressed by Eq. (1).

$$\begin{aligned}\Delta f(\theta) &= \sqrt{(R + \Delta R)^2 - e^2 \cos^2(\theta + \phi_0)} + e \sin(\theta + \phi_0) - R \\ &\approx \Delta R + e \sin(\theta + \phi_0)\end{aligned}\quad (1)$$

As Fig. 3(a) shows, actual measurements of the surface form of the rolling roller indicate oval-shaped second-order profile deviations. ΔR , the surface profile deviation of the rolling roller in Eq. (1), can be modeled by Eq. (2), where a is the amplitude and k is the order. ε_0 is the initial angle in relation to θ . Figure 3(b) shows a model of DR for $k=3$.

$$\Delta R(\theta) = a \sin(k\theta + \varepsilon_0) \quad (2)$$

If 360 deg is termed one cycle for angle θ in Eqs. (1) and (2), Δf displays a 1st-order and k^{th} -order cyclicity for one revolution of the rolling roller. Given this, the pressure applied to the ring is cyclically non-uniform, and it can be predicted that when it is necessary to use m revolutions of the rolling roller in rolling the entire length of the circumference of the ring, undulations displaying m^{th} -order and $m \times k^{\text{th}}$ -order cyclicity will be formed on the surface of the ring.

In the heat treatment process, local deformation in the surface of the ring is generated by physical interference when the ring is placed in the jig.

3. Principle of Measurement

The surface form of the rings across their entire circumference is measured using the instrument shown in Fig. 4. The ring revolves around two connected pulleys. A weight of approximately 5 kg is applied to pulley B, which sits on top of the guide mechanism, via a roller. This exerts tensile force on the ring. A line laser sensor (an LJ-G030, manufactured by Keyence), is positioned at the median point between pulleys A and B, and acquires instantaneous data on the surface form of the ring across its entire width, as shown in Fig. 5. The measurement time per cross-section is approximately 4 ms, and approximately 300 measurement points are employed for the 10 mm width under evaluation. The acquired data is rotationally transformed in order to render it parallel with

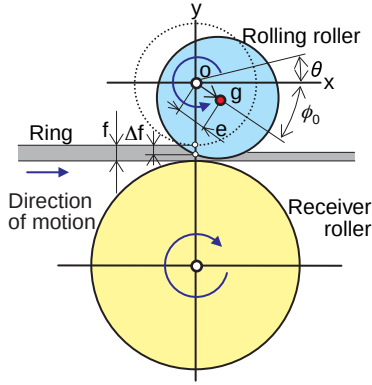
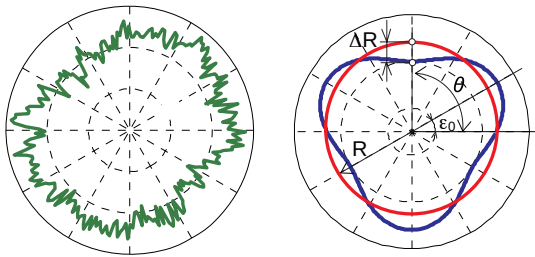


Fig. 2 Schematic diagram of rolling of steel ring



(a) Deviation of real form (b) Deviation of modeled form

Fig. 3 Deviation of real and modeled forms on rolling roller

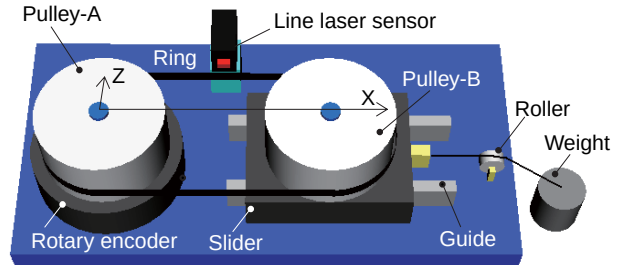


Fig. 4 Ring surface 3D measurement instrument

the y axis with the two edge points of the ring as the benchmarks, and the maximum height of the arc, h , is calculated.

When data has been acquired for one cross-section, the pulleys are successively rotated in identical angular increments, enabling continuous ring cross-section data to be acquired. This equally spaced feed is converted into length, and the cross-sectional form is arrayed along this length to create a three-dimensional surface form. In order to do so, it is necessary to reduce the noise that is generated by the optical characteristics of the laser sensor to help ensure accurate measurement of the individual cross-sectional surface forms.

3.1. Method of Evaluation of Ring Cross-sectional Form

The reduction of ring surface roughness and noise in the data acquired by the laser sensor was studied.

Figure 6 shows an example of surface form data acquired by the laser sensor. Oscillation of short-wavelength components can be seen on the arc-shaped surface. Because ring surface roughness components and noise exist in the short-wavelength components, a low-order approximate curve, P , is determined for the cross-sectional form data, as shown in Fig. 6, and the residual error is separated out as surface form Q , as shown in Fig. 7. The power spectrum of Q and its integrated frequency is then calculated, as shown in Fig. 8(a).

The same cross-section on the same ring is then measured using a contact-type form measurement machine (a CV-500 manufactured by Mitutoyo) with a stylus of 25 μm in radius, and the power spectrum of the surface form and its integrated frequency is calculated, as shown in Fig. 8(b). The integrated frequency distribution in Fig. 8(b)

increases logarithmically up to approximately the 10th-order, and includes approximately 80% of the form components. In addition, the form components up to the 10th-order are in the range of undulations with wavelengths of 0.9 mm and higher, and do not correspond to the wavelength range of roughness.

The integrated frequencies of the power spectrum calculated from the surface form measurements obtained using the contact-type form measurement machine and the laser sensor exceed 80% at the 12th and 32nd orders respectively. Given this, it was judged that the short wavelength components of the 12th-order and above in the surface form data acquired using the laser sensor represented noise occurring due to the optical characteristics of the laser, and it was therefore decided to employ a Gaussian filter⁽³⁾, which would shield the short-wavelength components. The cutoff frequency, λ_c , was set at 0.8 mm. As a result of using this filter, amplitude transmission characteristics of approximately

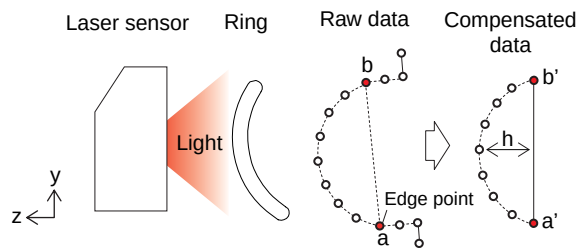


Fig. 5 Principle of data acquisition and compensation

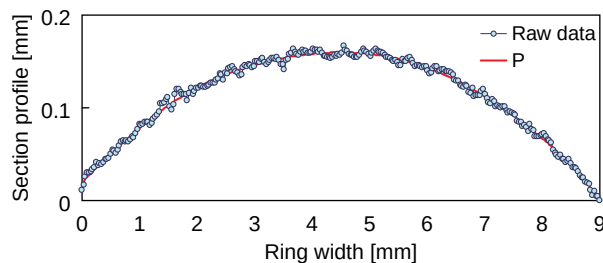


Fig. 6 Example of data acquisition by line laser sensor

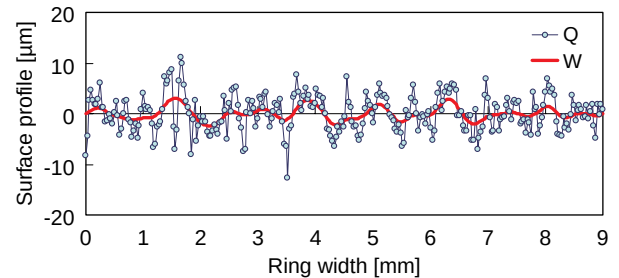
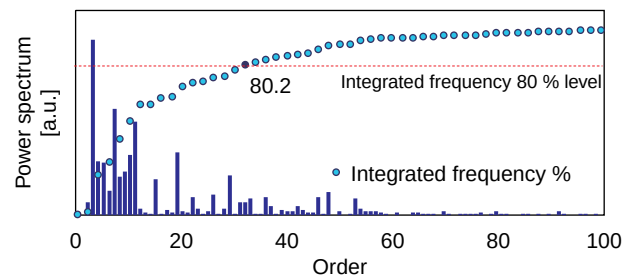
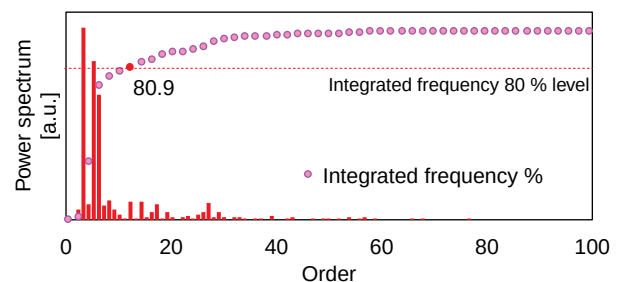


Fig. 7 Ring surface measured by line laser sensor and filtering



(a) Line laser measurement



(b) Contact-type form measurement machine

Fig. 8 Comparison of power spectrum on ring surface

50% and 10% were obtained for the 10th-order and 20th-order form components respectively.

The effect of the filter on the surface form Q is shown as undulation W in Fig. 7. The surface form of the ring is defined as the form obtained from the addition of P to the extracted undulation W. By separating the noise that is prone to occur at the ring edges from the surface form, this method stabilizes the accuracy of the measurement of the height of the arc, h, which is the object of measurement.

3.2. Evaluation of Accuracy of Non-contact-type Measurement

The developed measurement instrument was used to measure the arc heights, h, of ten cross-sections of a randomly-selected ring, and the results were compared with results obtained using the contact-type form measurement machine. As Fig. 9 shows, the average deviation for the ten cross-sections was 4.3 μm , and the coefficient of correlation was 0.99. The perpendicularity of the angle of incidence of the laser light on the ring surfaces was the main factor producing the deviation between measurements of h at the ten cross-sections made using the contact-type measurement machine and the laser sensor, and this was amenable to geometric correction.

The ten cross-sections of the randomly-selected ring were each measured ten times and an analysis of variance was conducted on the results in order to verify the repeatability of results obtained using the developed measurement instrument. The factors listed below were evaluated by resetting the roller in the measurement instrument for each measurement. σ represents standard deviation.

- (a) σ_R : Variation in the height, h, of the arc shape obtained from measurement points across the entire ring
- (b) σ_E : Variation in the height, h, of the arc shape obtained from repeated measurements in the same position
- (c) σ_M : Repeatability of the measurements including positioning of the pulleys and setting of the ring in the instrument

Using Eq. (3), X_{ij} was calculated for the data format shown in Table 1 as the measurement result for the ith

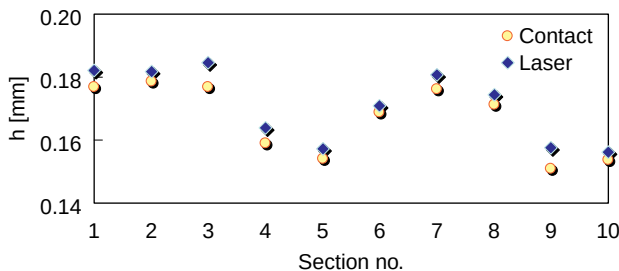


Fig. 9 Measurement accuracy based on laser sensor

Table 1 Raw data for evaluation of measurement repeatability

h mm	No. 1	...	No. 10	Average
Section 1	0.1796	...	0.1855	0.1820
...				
Section 10	0.1525	...	0.1543	0.1562
Average	0.1659	...	0.1720	0.1709

Table 2 Result of ANOV for measurement repeatability

$S_i \mu\text{m}^2$		n_i	$M_i=S_i/n_i \mu\text{m}^2$		μm^2
S_1	2.9204	1	M_1	2.92038	—
S_2	0.0117	9	M_2	0.001303	$\sigma_M^2 + 10 \sigma_R^2$
S_3	0.0004	9	M_3	4.05×10^{-5}	$\sigma_M^2 + 10 \sigma_E^2$
S_4	0.0017	81	M_4	2.06×10^{-5}	σ_M^2

measurement of the jth cross-section. σ_R , σ_E , and σ_M were calculated from an analysis of variance table (Table 2) and Eq. (4). n in Table 2 is the degree of freedom.

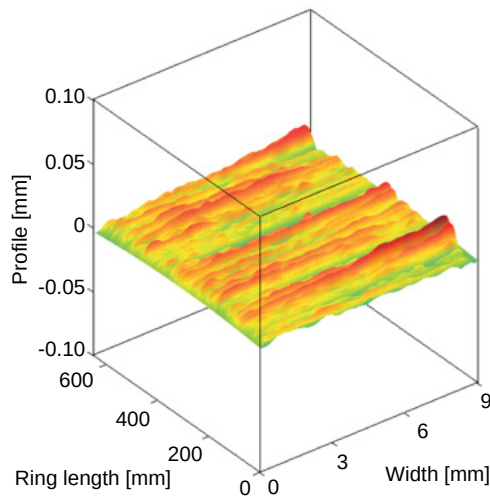
$$\begin{aligned} \bar{X}_i &= \frac{1}{10} \sum_{j=1}^{10} X_{ij}, \bar{X}_j = \frac{1}{10} \sum_{i=1}^{10} X_{ij}, \bar{X} = \frac{1}{100} \sum_{i=1}^{10} \sum_{j=1}^{10} X_{ij} \\ S_1 &= 100\bar{X}^2, S_2 = 10 \sum_{j=1}^{10} (\bar{X}_j - \bar{X})^2 \\ S_3 &= 10 \sum_{j=1}^{10} (\bar{X}_j - \bar{X})^2, S_4 = \sum_{j=1}^{10} \sum_{i=1}^{10} (X_{ij} - \bar{X}_i - \bar{X}_j + \bar{X})^2 \end{aligned} \quad (3)$$

$$\begin{aligned} \sigma_R &= \sqrt{(M_2 - M_4)/10} = 11.3 \mu\text{m} \\ \sigma_E &= \sqrt{(M_3 - M_4)/10} = 1.4 \mu\text{m} \\ \sigma_M &= \sqrt{M_4} = 4.5 \mu\text{m} \end{aligned} \quad (4)$$

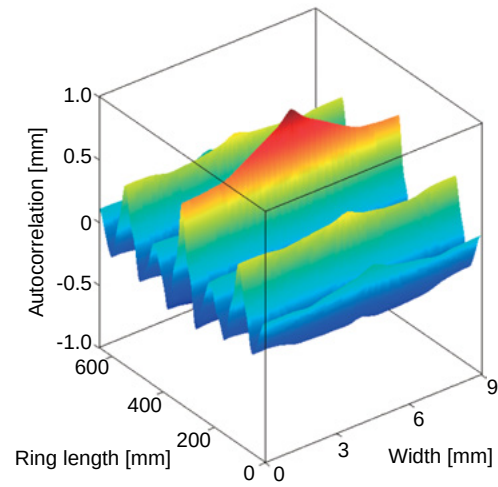
The measurement accuracy for the arc shape per cross-section for the measurement instrument, σ_E , was 1.4 μm , and the repeatability, σ_M , was 4.5 μm . This cannot be regarded as a high level of accuracy when compared to the accuracy of the contact-type measurement machine. However, the measurement principle employs a low-cost, general-purpose laser sensor and simple positioning mechanism, and can therefore be expected to find application at sites of manufacture of rings.

4. Measurement and Analysis Results

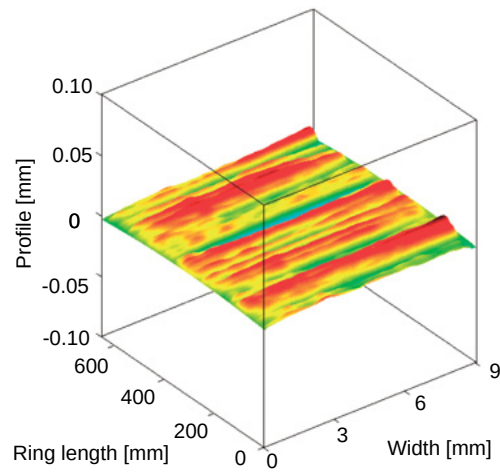
In order to help enable quantitative understanding of the processing history of the rings through the different manufacturing processes, the newly developed measurement instrument was used to evaluate the surface forms of (1) rings following rolling, (2) manufactured



(a) Raw data of ring surface after rolling



(c) Autocorrelation of the surface in Fig. 10 (b)



(b) After low pass filtering of the surface in Fig. 10 (a)

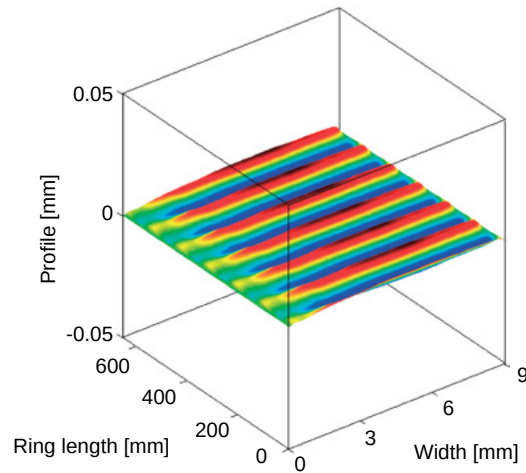
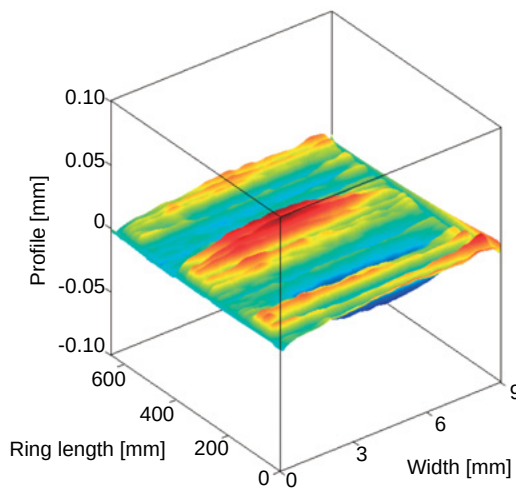
(d) Extraction of 6th-order form from Fig. 10 (b)

Fig. 10 Measurement results of ring surface after rolling



(a) After low pass filtering of raw surface data

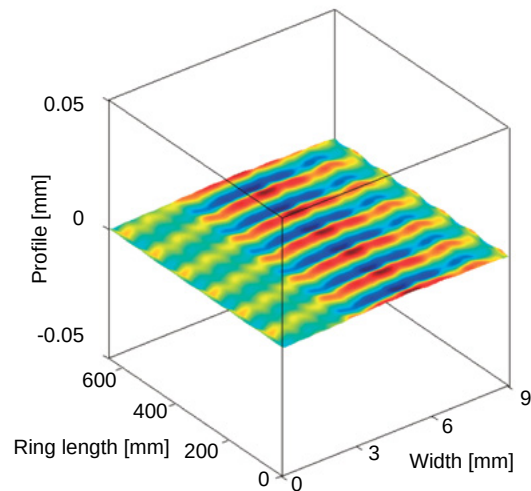
(b) Extraction of 6th-order form from Fig. 11 (a)

Fig. 11 Measurement results of ring surface of product

rings, and (3) areas of the rings that undergo distortion during heat treatment. The results of evaluation of (1), (2) and (3) are shown in Figs. 10, 11, and 12, respectively.

For the evaluations, the minimum angle of rotation of the pulley in the instrument was set at 3 deg, and approximately 250 cross-sections were used across the entire length of the rings. The number of cross-sections was arbitrarily set.

In order to separate the cylindrical and the surface form of the rings, least squares approximation was applied to the obtained three-dimensional surface form data, z , using Eq. (5). The surface form, dz , was defined as the deviation between z and z_p , the approximated cylindrical curved surface.

$$z_p(x, y) = ax^2 + by^2 + 2cxy + 2dx + 2ey + f \quad (5)$$

The surface form, dz , of the ring following rolling is shown in Fig. 10(a). Using Eq. (6), a filter that did not affect the form of the ring in the direction of width was applied in order to isolate the low-order form components of dz . λ_x and λ_y represent, respectively, the wavelength in the direction of the ring circumference, x , and in the direction of the ring width, y . The cutoff wavelength, λ_c , was set at 30 mm. This filter resulted in an amplitude transmission characteristic of approximately 50% for the 20th-order form components, and approximately 24% for the 30th-order form components.

$$H(\lambda_x, \lambda_y) = e^{-\pi \left(\frac{\alpha \lambda_c}{\lambda_x} \right)^2} \quad (6)$$

The effect of the filter applied using Eq. (6) is shown in Fig. 10(b). A series of six evenly-spaced asperities can be observed in the direction of the ring circumference. In addition, the amplitude of the form in the positive direction of ring width displays a tendency to increase.

Terming the surface area of the ring across its entire

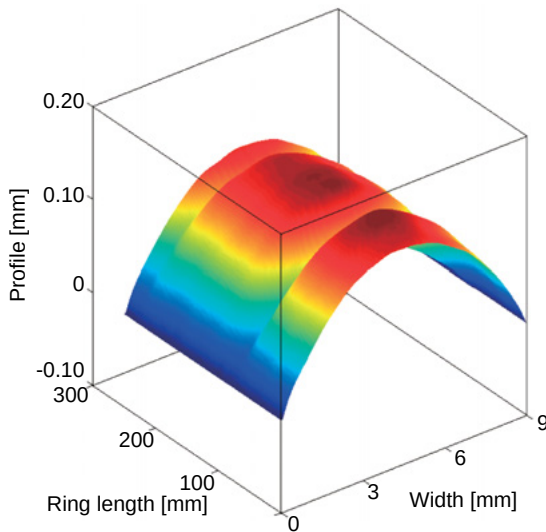


Fig. 12 Form deformation generated by heat treatment

length A , and the arithmetic mean of the surface height as found using Eq. (7) SW_a , an evaluation of the surface form shown in Fig. 10(b) resulted in a figure of 2.4 μm for SW_a .

$$SW_a = \frac{1}{A} \iint_A |dz(x, y)| dx dy \quad (7)$$

Normalization of C , calculated from a reverse Fourier transform, IFT [Eq. (9)], of S , the two-dimensional power spectrum found using Eq. (8), enables the auto-correlation function, A_c , to be found using Eq. (10).

$$S(f_x, f_y) = |\text{FT}(dz(x, y))|^2 \quad (8)$$

$$C(x, y) = \text{IFT}(S(f_x, f_y)) \quad (9)$$

$$A_c(x, y) = \frac{C(x, y)}{C(x_0, y_0)} \quad (10)$$

The auto-correlation function is employed in the study of processed surfaces in order to help enable discrimination of the direction of the main components, and as an index of directional cyclicity⁽⁴⁾. Figure 10(c) shows the results of calculation of the auto-correlation function with (x_0, y_0) , the centers of the length of the circumference and the plate width, as the benchmarks.

A 6th-order correlation that decreases in proportion to the distance from x_0 in the direction of the circumference can be observed in the results. The increases in correlation that are observable at 110 mm and 540 mm in the direction of the circumference result from the effect of the 3rd-order components. Figure 10(d) shows only the 6th-order components in the direction of the circumference extracted from the surface form shown in Fig. 10(b). The cyclical form continues across the entire width of the plate, with a maximum amplitude, S_6 , of 3.5 μm .

This results, as indicated by Eq. (1), from the eccentricity of the roller in relation to its axis of rotation, and shows that six rotations of the roller were required to roll the entire length of the circumference of the ring.

Figure 11 shows the results of measurement of the surface form of the manufactured ring. Figure 11(a) shows the results of application of the filter in Eq. (6) to the surface data for the manufactured ring. SW_a in Fig. 11(a) is 4.3 μm . Figure 11(b) shows only the 6th-order components extracted from Fig. 11(a). The maximum amplitude, S_6 , in these results is 1.5 μm , a reduction of 60% against the results for the ring following rolling. This is conjectured to be a result of the correction of the length of the circumference of the ring by the application of a roller of a different diameter to the rolling roller in order to increase the length of the ring following rolling.

Figure 12 shows the surface form of the ring following the application of the heat treatment. Two areas of deformation of approximately 40 μm can be observed within the 300 mm evaluation range. The positions of these areas of deformation correspond to the positions of

the white areas shown in Fig. 13. Their positions also match the positions of physical interference when the ring is placed in the jig.

Figure 14 shows the change in S_6 and SW_a between the ring following rolling and the completed ring. While S_6 decreases in the course of the processes following rolling up to the completion of the ring, SW_a increases. This is because the deformations induced by the application of the heat treatment, shown in Fig. 12, are included in the surface form. Evaluation of SW_a in isolation makes it challenging to achieve an understanding of the profile deviation resulting from the processes used in the manufacture of the rings, but simultaneous evaluation of S_6 , involving the evaluation of specific cyclical components, helps to explain the relationship between manufacturing processes and profile deviation.

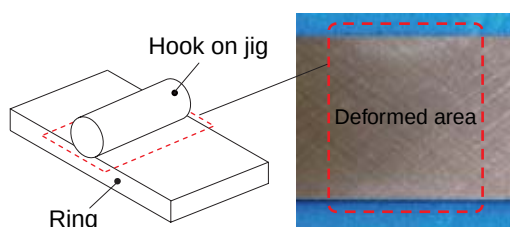


Fig. 13 Surface deformed area generated by heat treatment

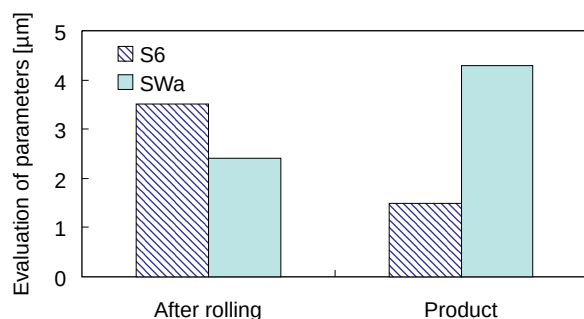


Fig. 14 Evaluation of surface height parameters

5. Conclusion

A technology enabling evaluation of the three-dimensional form of the metal rings employed in CVTs has been developed with the goal of achieving a high level of accuracy in the form of the rings. The results of this project and the benefits offered by the technology are as follows.

- (1) Acquisition of data on the cross-sectional form of the rings in the direction of the circumference using a general-purpose line laser sensor and continuous array of the data enabled the realization of three-dimensional form measurements. The developed measurement instrument has a simple configuration, is inexpensive,

and can be easily used at sites of manufacture of the rings.

- (2) The application of a frequency analysis to the three-dimensional surface form data enabled the profile deviations produced by processes employed in the manufacture of the rings to be revealed and quantified. In addition, expression of the surface form as an auto-correlation function enabled visualization of the characteristics produced by processing and the history of processing.
- (3) The eccentricity of the roller used in the rolling process produced a 6th-order cycle with a maximum amplitude, S_6 , of 3.5 μm , but this figure was reduced to 1.5 μm in the manufactured ring, following the process of correction of the length of the circumference. In addition, a case was observed in which the arithmetic mean of the height of the ring surface, SW_a , increased from 2.4 μm in the ring following rolling to 4.3 μm in the manufactured ring following heat treatment application.

This measurement and analysis technology can be expected to enable rapid detection of issues in manufacturing processes, and to contribute to enhancing the efficiency of prototype production.

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Psychological Effects of Use of Electric Scooter

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ABSTRACT

A questionnaire survey and interviews were conducted with users of electric scooter in order to determine the psychological effect of using these vehicles. It was found that there was a significant increase in the level of satisfaction reported by respondents in going shopping, running household errands, and traveling between their house and fields after they commenced using an electric scooter. Forty-five percent (45%) of respondents indicated that they felt less fatigued when leaving the house since they began using a scooter. In addition, more than 40% of respondents answered that the use of an electric scooter had increased their confidence when they left the house, even if they went out alone. Prior to purchasing an electric scooter, most respondents had lost the means of mobility that they had used previously, and were experiencing trouble walking. This can be considered to have resulted in a decline in their feelings of self-efficacy and self-esteem. After beginning to use an electric scooter, survey subjects recovered feelings of self-efficacy and self-esteem, and regained self-confidence. It can therefore be concluded that electric scooters are able to contribute to the maintenance or enhancement of the quality of life (QOL) of the elderly.

1. Introduction

For the elderly, leaving the house is of great significance not merely in terms of performing activities necessary to daily life, but also in terms of leading a physically and psychologically healthy and independent life. Many experts warn that becoming housebound is the preliminary stage of becoming bedridden^{(1), (2)}. When an elderly person becomes housebound, their level of physical activity is reduced, they have less contact with sources of visual stimulation, and their exchanges with other people are limited. This results in a decline in performance of activities of daily living (ADL) and in intellectual activity, and the occurrence of symptoms associated with disuse syndrome (muscular atrophy, contracture of joints, and reduction of cardiopulmonary function due to a lack of physical activity). This situation can easily lead to a transition to a bedridden state. A 2005 survey by the Cabinet Office showed that the number of times that elderly citizens left the house per day declined with advancing age, with the average number of times for the latter-stage elderly (75 and older) approximately 60%

of the figure for the early-stage elderly (aged 65-74)⁽³⁾. Behind these figures lie reductions in reasons and opportunities to leave the house as the elderly individual's role in society and the family changes, the loss of former means of mobility with declines in physical ability, and, as a result of these factors, a decline in desire to leave the house^{(4), (5)}. What is required, therefore, is to find a means of increasing the desire of elderly citizens to leave the house, to provide them with support to do so, and to provide an environment that enables unhindered mobility when they do so.

Taking Japan's aging society into consideration, in 1999 Honda commenced marketing the Monpal ML100, a four-wheeled electric scooter. The year 2006 saw the second-generation of the vehicle, the Monpal ML200, based on a concept of compact packaging and stylish design. As an individual increases in age, the choices of means of mobility available become narrower, and an electric scooter can be viewed as a highly serviceable mode of transport that elderly people can use with confidence^{(6), (7)}.

Electric scooters (JIS specifications: Total length: 120

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cm; Total width: 70 cm; Total height: 109 cm; Turning radius: approximately 110-160 cm) are classified with pedestrians in the road trucking vehicle law, and are classified as wheelchairs by the nursing care insurance system. Their maximum speed is limited to no more than 6 km/hr by the road trucking vehicle law. The JIS standards established for the vehicles require that they are appropriate for outdoor use, able to climb a 10 degree slope, ride over a stepped rise of 4 cm (with a running start), and cross a gap of 10 cm in width.

User surveys, by questionnaire and interview, have been conducted as part of the normal development loop, to enable the product to be refined and further evolved. However, while these questionnaires and interviews have elucidated the type of users purchasing electric scooters and potential points for enhancement in the products, they have not made clear how the use of a scooter has enhanced users' quality of life (QOL).

2. Purpose of Research

For a manufacturer of machines that increase human mobility, it is important to understand the ways in which the products that are offered affect the mobility and lifestyles of elderly users. Achieving such an understanding can enable the manufacturer to present enhanced products. However, to date there have not been any reports in the literature that clarified the psychological effects of using an electric scooter on elderly users. The research discussed in this paper therefore set out to study the psychological effects on users of using an electric scooter, in order to clarify the contribution made by the vehicles to QOL.

3. Questionnaire Survey

3.1. Purpose of Survey

The purpose of the questionnaire survey conducted in this project was to determine what types of individuals made use of electric scooters, how they came to use the vehicles, what they used them for, and whether the use of the vehicles had changed their experience of activities outside the house and their feelings regarding those activities.

3.2. Method

In June 2007, a questionnaire was distributed to 1000 users of the Monpal ML200 electric scooter; 458 of these surveys were completed and returned (Response rate: 45.8%). The questionnaires were distributed by staff members of regional vendors of the Monpal ML200. The questionnaires were distributed and collected either directly or by mail, as decided by the individual vendors. The ratio between male and female survey subjects was 50-50, with 88% of subjects aged 70 and above (Fig. 1). In terms of means of mobility prior to using the Monpal,

Table 1 How subject learned about electric scooter

I saw a user	50% (229)
I saw an electric scooter in a shop	21% (94)
I saw a leaflet	12% (53)
I saw an advertisement in the newspaper	12% (53)
I saw an advertisement in a magazine	5% (25)
I saw a TV commercial	4% (19)
I saw one on the internet	4% (19)
Other	21% (94)
No response	3% (15)
N = 458	

51% of survey subjects had driven cars or ridden motorbikes, 32% had ridden bicycles, and 16% had walked. Seventy-three percent (73%) of subjects were first-time purchasers of an electric scooter, while 20% had owned an electric scooter previously.

3.3. Results and Considerations

When asked by the questionnaire how they had come to learn about electric scooters, half (50%) of respondents indicated that they had seen the scooters being used (Table 1). It may be conjectured that many elderly individuals would view operating an electric scooter as too difficult for them, but that seeing other elderly people using the vehicles can prompt them to feel that they could operate the vehicles without being anxious because their speed is low, or to feel, for example, "If that person can do it, then so can I."

When asked what prompted them to purchase an electric scooter, many subjects responded in terms of

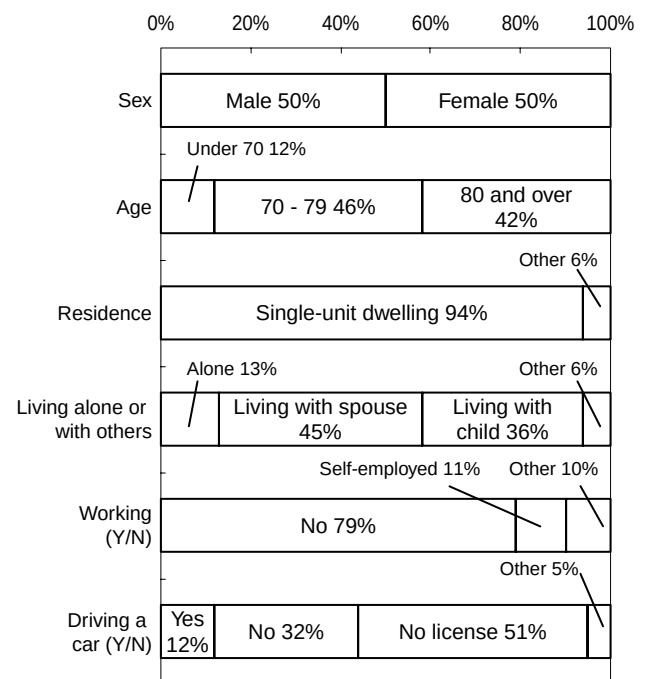


Fig. 1 Subject attributes

increasing their mobility, with 271 respondents (59%) answering “It had become difficult to walk outside,” 122 (27%) answering “It had become difficult to use a car or motorcycle to get around,” 98 (21%) answering “I wanted to go out independently, rather than being helped by others,” and 96 (21%) answering “It had become difficult to use a bicycle to get around.” Other respondents indicated that they had been encouraged to purchase an electric scooter, for example by family or friends (111 people; 24%) or retail store staff (56 people; 12%). Other responses included “A person I know started using one, and seemed to enjoy it” (46 people; 10%), “I saw one in a magazine or newspaper, and it looked appealing” (31 people; 7%), and “I saw one at an exhibition or test ride meeting, and it seemed appealing” (25 people; 5%).

With regard to the actual purchase, 204 respondents (38%) indicated that they decided to purchase a scooter by themselves, 176 (45%) indicated that they decided on purchasing a scooter in consultation with their family, and 50 (11%) indicated that the scooter was a gift from their family. When asked whether they had compared the Monpal to competitors’ models, 89 respondents (19%) answered that they had, and 296 respondents (65%) answered that they had not. This suggests that the majority of users did not gather information about the scooters from the Internet, pamphlets, and the like, but were introduced to the vehicle by more personal sources of information (staff members of stores with which they had dealt previously, personal acquaintances, and the like) prior to their purchase.

3.3.1. Mode of use of the vehicle

The literature suggests that an elderly individual’s desire to leave the house, and the necessity for that individual to leave the house, is affected by whether or not they live with another person or persons⁽⁵⁾. It can therefore be reasonably assumed that the type of use to

which an elderly individual will put an electric scooter in order to leave the house will also be affected by the same factor. Specifically, an elderly person who lives alone will have to run all their errands – shopping, going to the post office, visiting the hospital – by themselves, and the necessity for them to leave the house will be greater than that of a person who does not live alone. With regard to the status of respondents’ outdoor activities and changes in that status, in addition to reviewing overall results, the researchers divided the results into three groups, “Living alone” (60 people), “Living with spouse” (204 people), and “Living with children” (163 people), and applied an χ^2 test to determine whether responses differed between the three groups.

When asked their reason for using the electric scooter, the majority of respondents answered “Going shopping” (273 of 458 people, or 59% of the total), followed by “Going to the bank/post office” (188 people; 41%), “Visiting the hospital” (165 people; 36%), and “Visiting a friend’s home” (159 people; 35%) (Fig. 2). The application of an χ^2 test to determine a difference between respondents living alone and respondents living with others showed no significant difference. This demonstrates that it is not merely elderly people living alone who have to purchase food and daily necessities by themselves, but also elderly people living with their families whose main purpose in using the electric scooter is to go shopping. Sixty-three percent (63%) of respondents who live alone indicated that they use their electric scooter for shopping.

With regard to the frequency with which they use their scooter, 31% of all respondents (143 of 458 people) answered “Every day,” 13% (61 people) answered “Once every two days,” 27% (123 people) answered “Two or three times a week,” and 15% (69 people) answered “Once a week.” No difference was observed between respondents who live alone and respondents who live with others, with 10 respondents in the former group (17% of

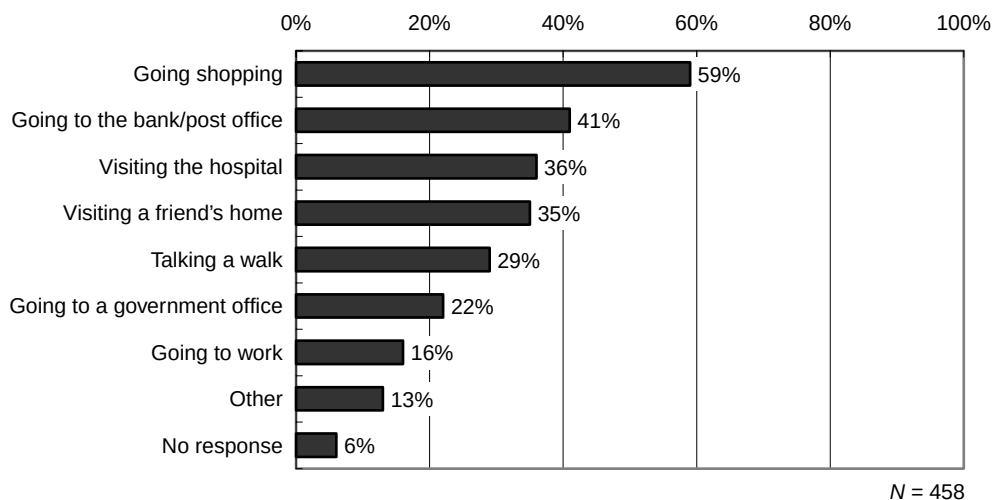


Fig. 2 Reason for using the electric scooter

Table 2 Time spent using the electric scooter on each trip outside the house

	Overall N = 458	Living alone n = 60	Living with spouse n = 204	Living with a child n = 163	χ^2 value
For less than 30 min	31% (142)	22% (13)	33% (67)	31% (51)	2.77
For 30 min to 1 h	37% (167)	23% (14)	40% (82)	38% (62)	5.78 [†]
For 1 to 2 h	20% (93)	35% (21)	17% (35)	19% (31)	9.40**
For 2 h or longer	4% (20)	7% (4)	4% (8)	3% (5)	1.49
No response	8% (36)	13% (8)	6% (12)	9% (14)	–

** : $p < 0.01$, [†] : $p < 0.1$

Table 3 Changes in status of trips outside the house

	Overall N = 458	Living alone n = 60	Living with spouse n = 204	Living with a child n = 163	χ^2 value
I no longer become very tired when leave the house	45% (208)	45% (27)	45% (92)	45% (74)	0.00
I am now able to go places that I had been visiting less frequently	39% (183)	37% (22)	38% (77)	41% (67)	0.57
I leave the house more often	37% (174)	33% (20)	38% (77)	38% (62)	0.46
I take more walks and strolls	21% (97)	8% (5)	22% (45)	25% (40)	7.15*
I can go to places that I've never been to before	16% (77)	10% (6)	18% (37)	17% (28)	2.27
Other	9% (43)	8% (5)	11% (22)	9% (15)	–
No noticeable change	14% (66)	20% (12)	16% (33)	13% (21)	1.85
No response	4% (20)	11% (7)	29% (7)	7% (12)	–

* : $p < 0.05$

60 people) answering that they only used their electric scooters once a week.

When asked the average length of time for which they used their electric scooter each time, the largest number of respondents answered “30 minutes to one hour” (37% of the total), followed by “Less than 30 minutes” (31%), and “One to two hours” (20%) (Table 2). Approximately 70% of respondents indicated that each trip would take them no longer than one hour, allowing for the conjecture that the majority were using their scooters to shuttle between their houses and nearby shops, post offices, banks, hospitals, and friends’ houses. Checking for differences between respondents living alone and respondent living with others showed a significant trend for the response “30 minutes to one hour” ($\chi^2(2) = 5.78$, $p < 0.1$) and a significant difference for the response “One to two hours” ($\chi^2(2) = 9.40$, $p < 0.01$) (Table 2). This demonstrates that the periods spent using their electric scooters tend to be longer among respondents who live alone. Elderly people living with their families are able to request their family members to run errands for them outside the house, or can be driven around outside the house in the family car. By contrast, elderly people living alone must perform all their necessary daily activities – getting cash from the bank, making payments, buying food – by themselves, and many have no means of getting around other than their electric scooter. This means that on each trip that they take, they will tend to visit multiple destinations, for example banks, post offices, shops, and

hospitals. It can therefore be considered only natural that the periods that they spend using their scooters will be longer.

3.3.2. Changes in status of outdoor activities and feelings regarding outdoor activities with use of electric scooter

Respondents were asked how the status of their outdoor activities had changed since they began using an electric scooter (Table 3). “I no longer become very tired when I leave the house” was the most popular response (45% of the total), followed by “I am now able to go places that I had been visiting less frequently” (39%), “I leave the house more often” (37%), “I take more walks and strolls” (21%), and “I can go to places that I’ve never been to before” (16%). Muronaga and Morozumi (2004) indicate that the desire of elderly individuals to leave the house is affected by changes in their physical capacity and state of health⁽⁵⁾. The fact that a scooter can help prevent the elderly from becoming fatigued can therefore be pointed to as an important element in encouraging them to leave the house. In considering why the use of an electric scooter reduces fatigue, it can be assumed that it is related not merely to the fact that the vehicle makes the user mobile without having to walk, but also that when using one, they do not have to walk with heavy parcels.

Testing for differences between respondents living alone and respondents living with others in terms of changes in the status of outdoor activities following the

Table 4 Changes in feeling towards trips outside the house

	Overall N = 458	Living alone n = 60	Living with spouse n = 204	Living with a child n = 163	χ^2 value
I am more confident when leaving the house, even if I go alone	44% (206)	42% (25)	47% (95)	42% (69)	0.85
I no longer feel that leaving the house is a hassle	43% (201)	42% (25)	42% (86)	47% (77)	1.11
I am now able to enjoy a variety of sights and scenery	34% (157)	32% (19)	35% (71)	33% (54)	0.25
I enjoy meeting people	25% (116)	23% (14)	23% (47)	26% (42)	0.39
I am more comfortable since I have to ask for help less and don't need someone to go with me	19% (90)	8% (5)	20% (41)	24% (39)	6.70*
Other	5% (24)	8% (5)	6% (12)	4% (7)	—
No noticeable change	14% (65)	12% (7)	15% (30)	15% (25)	0.49
No response	5% (23)	6% (4)	4% (9)	2% (4)	—

*: $p < 0.05$

purchase of an electric scooter showed a significant difference ($\chi^2(2) = 7.15$, $p < 0.05$) for the response “I take more walks and strolls” (Table 3). This indicates that the number of users who go for more walks and strolls following their scooter purchases tends to be higher among individuals who live with others. One reason that might be considered for this is that elderly individuals who live with others have more free time. Given that they must perform all their necessary activities for themselves, leaving the house will require time and energy for elderly people who live alone, and many will not have more opportunity to walk or stroll. Elderly individuals living with others, by contrast, will often have household errands performed by family members, and will thus have free time available for walking or strolling. In addition, many elderly people who are being looked after try to be considerate of the family members they live with, and find it difficult to be with them constantly. It may be considered that some individuals go for walks or strolls to escape these types of psychological pressures.

The survey questioned respondents on changes in the way they feel about leaving the house since beginning to use an electric scooter (Table 4). “I am more confident when leaving the house, even if I go alone” (44% of the total) and “I no longer feel that leaving the house is a hassle” (43%) were the top responses, followed by “I am now able to enjoy a variety of sights and scenery” (34%), and “I enjoy meeting people” (25%). Testing for differences between respondents living alone and respondents living with their spouses or children showed a significant difference for the response “I am more comfortable since I have to ask for help less and don't need someone to go with me” ($\chi^2(2) = 6.70$, $p < 0.05$). This indicates that this feeling is most common among respondents living with others, but this result can be considered natural because respondents living alone have no one around them who they can turn to for assistance or go somewhere with.

Helping the elderly to maintain feelings of self-efficacy (the feeling that one can successfully do things for

oneself) and self-esteem (a feeling of positive self-acceptance) is important in helping to prevent them from becoming housebound or bedridden. A decline in self-efficacy and self-esteem can lead to a vicious circle in which the individual tends to stay indoors, leading to a decline in physical ability, which in turn leads to a loss of self-confidence and further declines in self-efficacy and self-esteem. The majority of users of electric scooters who were surveyed can be characterized as having felt anxious about walking and having lost a previous mode of mobility before they began using the vehicle, resulting in a situation in which their self-efficacy and self-esteem could decline or had already declined. Almost half of respondents indicated that the use of the electric scooter had made them “...more confident when leaving the house, even if I go alone,” and almost 20% answered that it had made them “...more comfortable since I have to ask for help less and don't need someone to go with me” (Table 4). These results justify the conjecture that there are cases in which the use of an electric scooter can assist an elderly person in maintaining or recovering their sense of self-efficacy and self-esteem.

4. Interview Survey

4.1. Purpose of Interview Survey

To attempt to determine the degree to which using an electric scooter has increased the user's level of satisfaction with their outdoor activities.

4.2. Method

Nine subjects were selected from respondents to the questionnaire survey who indicated that they would be able to participate in interviews, and these interviews were conducted either at the subject's house or at the store from which they purchased their electric scooter. Table 5 shows the attributes of the subjects (ability to walk, and the like). Three male and six female subjects were interviewed. One subject was in her 60s, three were in their 70s, and five were in their 80s. All subjects resided

Table 5 Subject attributes

Subject	Sex	Age	Living alone or with others	Residence	Ability to walk
A	F	80 - 89	Living with son	Single-unit dwelling	Difficulty walking (use of cane required)
B	M	80 - 89	Living with spouse (C)	Single-unit dwelling	Ambulatory but only for short distances
C	F	70 - 79	Living with spouse (B)	Single-unit dwelling	Ambulatory
D	M	80 - 89	Living with spouse	Single-unit dwelling	Difficulty walking
E	F	70 - 79	Living with child	Single-unit dwelling	Ambulatory but only for short distances
F	F	60 - 69	Living with spouse and son	Single-unit dwelling	Ambulatory
G	F	70 - 79	Living with spouse	Single-unit dwelling	Ambulatory but only for short distances
H	F	80 - 89	Living with spouse	Single-unit dwelling	Ambulatory
I	M	80 - 89	Living alone	Single-unit dwelling	Ambulatory but only for short distances

1) Sex: F = Female, M = Male

in single-unit dwellings.

Eight of the interview subjects lived with others; only one subject lived alone. Interviews were conducted with three subjects (A, B, and C) on July 12, 2007, with one subject (D) on July 13, 2007, with three subjects (E, F, and G) on August 6, 2007, and with two subjects (H and I) on October 17, 2007.

4.3. Results and Considerations

Everyday visits outside the house were divided into eight categories according to purpose: 1) Enjoying a hobby; 2) Going shopping; 3) Running household errands such as sending mail; 4) Interacting with friends; 5) Visiting the hospital; 6) Traveling between home and fields; 7) Participating in neighborhood associations; and 8) Taking a walk. Interview subjects were asked to compare their feelings of satisfaction in performing these activities before and after they commenced using their electric scooter, and to rank their degree of satisfaction on a scale from 1 to 5, with 1 representing "Not satisfied at all," and 5 representing "Extremely satisfied." In

addition to determining average figures for each category, the research team conducted *t*-tests in order to determine whether there were any statistically significant differences before and after the subjects commenced using their scooters (Fig. 3). Significant differences were determined for the categories "Going shopping" ($t(8) = 2.12$, $p < 0.05$), "Running household errands" ($t(8) = 2.87$, $p < 0.05$), "Interacting with friends" ($t(8) = 2.24$, $p = 0.05$), and "Traveling between home and fields" ($t(8) = 1.90$, $p < 0.05$).

Significant increases in the subjects' level of satisfaction in leaving the house for shopping, running errands, and traveling between the house and fields were observed after they commenced using their electric scooters, indicating that they had experienced considerable loss of mobility prior to using the vehicles. Five of the nine subjects (A, B, D, E, and G) had in fact found walking difficult due to pain in their legs and other factors. Subject A, for example, had not been able to walk without a cane, and had to be driven in a car by her husband, but lost this means of mobility following her husband's death. Subject D's legs gave him trouble, and

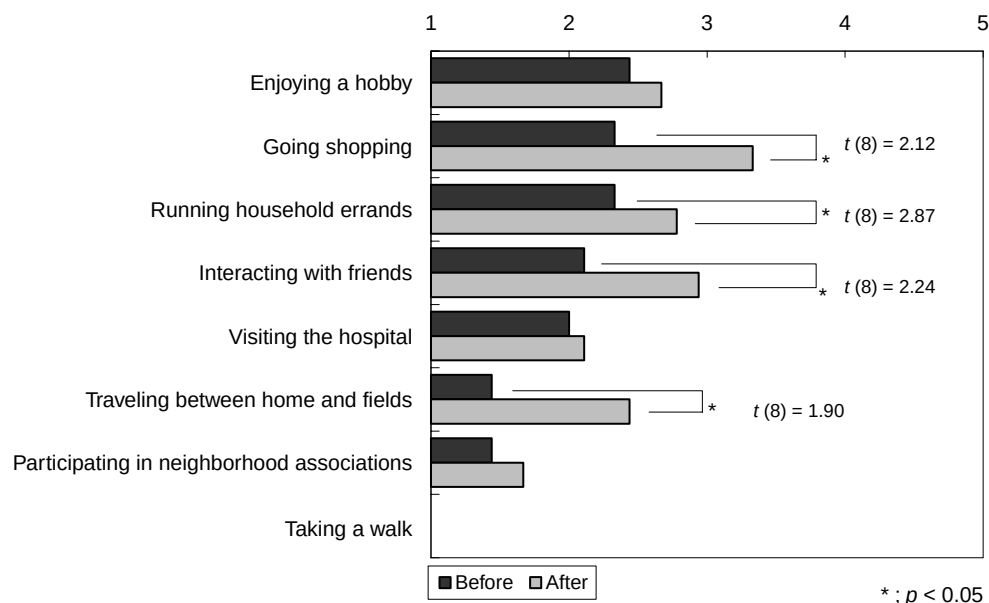


Fig. 3 Comparison of level of satisfaction when leaving the house before and after beginning to use an electric scooter

he could not ride a bicycle, as a result of which he tended to be confined to his house. In addition, three of the nine subjects (F, H, and I) had given up driving or riding motorcycles due to their advanced age. The majority of subjects were obliged to rely on the assistance of family members to perform their daily errands. They would, for example, request family members to do their shopping for them, or ask to be driven to their fields. It is presumably because of this that the majority of subjects experienced a strong feeling of satisfaction when they became able to perform their household errands and daily business by themselves. At present, subject A uses her electric scooter to pay her respects at her husband's grave in addition to going shopping and visiting the hospital (Fig. 4). Subject D indicated that the electric scooter had enabled him to meet his wife on her way home when she goes out to do the shopping, and to carry her purchases home for her. Subjects E and F are able to make the trip between their fields and their homes when they feel like it without bothering members of their family. Subject B has found pleasure in being able to go shopping at the local mall with his wife, subject C.

The results of the interviews also suggested that the use of electric scooters had expanded and enhanced contacts with friends. It may be conjectured that this is due to the fact that the vehicle enables its user to comfortably visit their friends' homes by themselves, that being able to progress at a walking pace in the vehicle enables the user to easily converse with acquaintances that they meet, and that they are also able to stop the vehicle and remain seated, enjoying a conversation without becoming fatigued. Subject F pointed out that "When I'm out on my Monpal and I meet a friend, I can stay seated while we stop and talk. When I want to rest, the Monpal is my very own seat." Having a place to sit and rest when they go out of the house is important for the elderly; it has been reported that elderly people will limit the amount

of times that they leave the house if they cannot be sure that they will find a place to rest⁽⁵⁾. Elderly people can feel relaxed when going out using an electric scooter, because it functions as a seat when they need to rest – when the vehicle is stopped, the user does not have to support the vehicle in an upright position. With regard to the ability to interact with friends when they went out, subjects indicated that "I am now able to visit friends who live a long way away," and "Before, I had to have my family drive me to the field, but now I can go there and come back on the sidewalk, and that gives me a lot more opportunities to chat with friends and acquaintances." Subject G said that her scooter had enabled her once again to visit a friend who lives 6 km away.

The use of an electric scooter also reduced the burden of carrying parcels when shopping. Subject B pointed out that "With my scooter, I can go out and buy a case of beer. I can buy rice!" As this indicates, in addition to offering increased mobility, the electric scooter is a vehicle that provides its users with various other functions, including the ability to carry parcels, and functioning as a place to rest.

5. Conclusion

Reasons for the elderly becoming housebound include a lack of inclination on the part of the individual to go out, a lack of desire on the part of family members to let the individual go out, an inability to go out due to social barriers, and a lack of places to go⁽⁶⁾. The latter stage elderly in particular are obliged, unwillingly, to rely on their families and others for even their minimum requirements for leaving the house, such as running necessary errands or receiving medical care. This can be considered to result in some level of decline in their QOL. The survey discussed here has indicated that an electric scooter can offer assistance to elderly people who find it difficult to walk but cannot use other means of mobility, and can help people who tend not to leave the house because of a lack of confidence. The results have shown that, by means of changing elderly peoples' experience of leaving the house and their feelings about leaving the house, the use of an electric scooter can enable an elderly person to continue to perform their basic daily requirements independently, can increase their freedom and their enjoyment of life, and can enable them to recover their sense of self-efficacy and self-esteem. It can therefore be concluded that electric scooters are able to contribute to the maintenance or enhancement of the QOL of the elderly. The survey produced the following results leading to this conclusion:

- 59% of respondents had found it troublesome to walk outside the house before beginning to use an electric scooter.
- Almost half of respondents (45%) indicated that the use of an electric scooter enabled them to leave the house without feeling tired.



Fig. 4 Subject A using an electric scooter to pay respects at a gravesite

- A high percentage of respondents answered that with the use of an electric scooter “I am more confident when leaving the house, even if I go alone” (44%) and “I no longer feel that leaving the house is a hassle” (43%).
- In addition to reducing the troublesomeness of leaving the house, 34% of respondents indicated that their electric scooter had enabled them to enjoy sights and scenery at a variety of different destinations, and 25% indicated that it had enabled them to enjoy spending time with other people.
- The level of satisfaction of the user increased with the use of the electric scooter for the responses “Going shopping” ($t(8)=2.12$, $p<0.05$), “Running household errands” ($t(8)=2.87$, $p<0.05$), “Interacting with friends” ($t(8)=2.24$, $p<0.05$), and “Traveling between house and fields” ($t(8)=1.90$, $p<0.05$).

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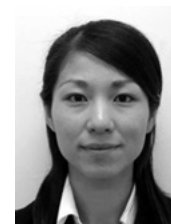
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Editorial Afterword

Everyone recognizes the importance of compiling the results and the intellectual processes of research and development in reports. That is why we try to leave a record of these activities, even if they might just be in the form of notes and drawings, because these become a “storehouse of technology,” with value both for the authors and for the organization. When it comes to putting this type of information in organized form as papers for publication, however, I think there are many people who think it is just too difficult or troublesome. One reason for this is that, unlike a PowerPoint presentation, which can be put together from paragraph points and graphics, a paper places one’s writing ability on the line.

A certain novelist used to say that we should write “difficult things so they are simple, simple things so they are profound, and profound things so they are interesting.” This certainly makes sense, and I wish that I could write that way. No doubt the ability to do so is one of the secret arts of a master writer who has to describe people and things related to them.

In the case of a technical paper, it is more likely to be mathematical formulas that are essential, and specialized terminology that must be used in order to explain the subject. When the readers are experts, the writing may be made more understandable not by making it simple but rather by stating the matter clearly and concisely and using the appropriate technical terminology. A paper for the Technical Review, however, should be written so that it can be understood even by people from other specialized fields, just as long as they have the necessary technical foundation. That is what we consider important.

Engineers are not professional writers, so it’s no simple matter for them to write like a master, even if they wanted to. The first step is to create writing that is logical and unambiguous, and does not cause stress for the reader. I would like to see our engineers keep the idea of writing “difficult things so they are simple” somewhere in their minds, and go ahead and write and write. Then they can publish the results. They should read many, many papers and write a great deal, and by doing so, they may find that they, too, are approaching that realm of the profound and the interesting.

(Toru FUJIHIRA)

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