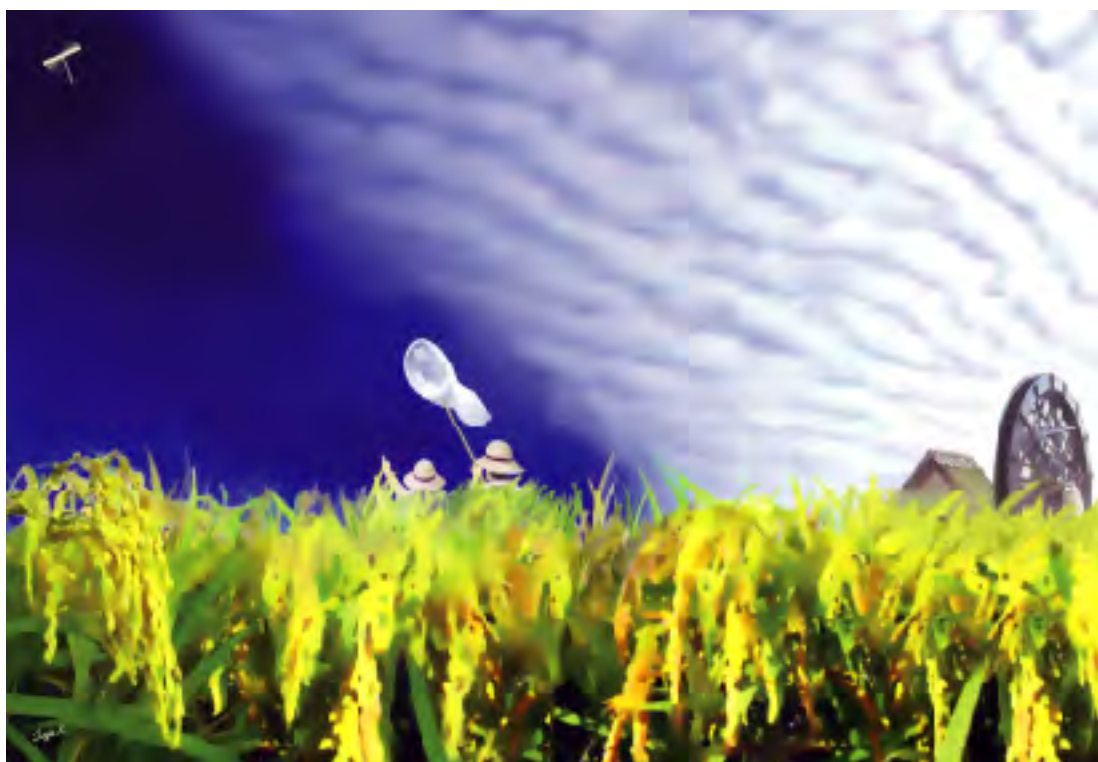


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Comment on the Cover

Focusing all of our intelligence and ingenuity
on the pursuit of our dreams; the spirit of craftsmanship
transcends time.



Taiji Koyama
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Styling Design of New STREAM

Hideaki UCHINO*

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ABSTRACT

The concept of Motion Designing was used in the development of the new model STREAM to produce a vehicle that offers a sense of integration at every point of contact between vehicle and human user and that urges the owner to drive it.

The aim of the design of the STREAM was to reproduce the dynamism of a high-performance sports coupe. A “Crouching Coupe Form” was the basis of the exterior design. In addition to this, the image of a “Silver Wolf” was adopted as a design motif, and the form was designed to express the sense of aggressiveness and sharpness suggested by that image.

The design theme for the interior was “Modern Sporty Interior,” representing a fusion between “Emotional Sports Cockpit” and “Modern Comfort.” This has resulted in an advanced cockpit with comfortable, high-quality seats producing the feel of a cozy living space.

1. Introduction

It is now more than 10 years since the minivan category became established in the Japanese market. When the first-generation STREAM made its appearance on October 26, 2000, the new concepts it embodied attracted numerous customers.

In the almost six years that have passed since then, the market, customers, and vehicles themselves have further diversified. Against this background, the new second-generation STREAM was developed to embody new values appropriate to the Honda name.

The development of the new STREAM commenced with the concept of Motion Designing, a design that offered a sense of integration at every point of contact between vehicle and human user and that actively urges the owner to drive the vehicle. The design team kept certain questions in mind throughout the design process: How could they create a vehicle that would become a

new standard in the category, that would make customers proud to own it, and that would provide an entirely stress-free driving experience? The motivating force behind the design was the team’s desire to create a minivan that would do away with any lingering resistance to the purchase of a vehicle in this category, and that would grow to be loved by customers owning it; a sporty vehicle that incorporated new values that would, for example, make owners want to look at it, feel pleasure in washing it and be proud of driving it. This can be summed up in the phrase “Fun to Have STREAM.”

To actualize the design concept for the new STREAM, designers focused on users who took seriously the challenge of balancing their lifestyles and their social responsibilities. Aiming towards a stylish design, the team realized a passenger vehicle that is both sporty and comfortable to drive around town. This paper presents an overview of the design of the new STREAM (Fig. 1).

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Fig. 1 Fun To Have STREAM

2. Design Aims

To enable the creation of a more distinctive sporty passenger vehicle, “Crouching Coupe STREAM” was established as the design concept. Based on this concept, the design team lowered the front slope of the vehicle to give it a sense of leaning forward, ready to spring into action. They also aimed for the high quality and individualistic style of a coupe.

The issues in realizing this concept were to give the vehicle a low profile and a sporty and individualistic appearance when viewed from the outside, while bringing these qualities into harmony with the seemingly contradictory requirements for the interior space: a spacious and comfortable cabin with a level of quality that provides a sense of custom design throughout (Fig. 2).

3. Package Layout

Numerous studies were conducted to enable the exterior to be lowered to give the vehicle the profile of a sports coupe, while making the interior more spacious than the first-generation STREAM, while at the same time ensuring high quality. The basic requirement in terms of the package layout was to ensure a comfortable ride for occupants despite the cabin height, which is low for a minivan, and the space limitations of a C segment vehicle.



Fig. 2 CROUCHING COUPE STREAM

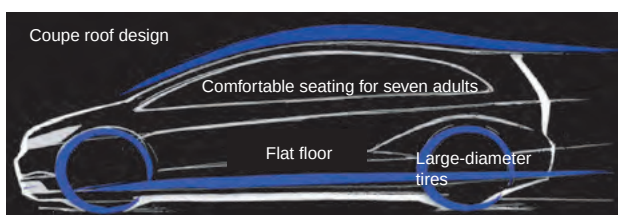


Fig. 3 Package layout

Discussion of the numbers was of course important in achieving these goals, but first maximum use was made of package models, with the entire design team contributing ideas. Development proceeded with the pursuit of increased comfort in the limited space from the floor to the seat as a specific goal.

To achieve this goal, the design team focused on the design of the floor. The use of a floor design that is flat throughout the vehicle (like a bathtub) and that has been lowered significantly at the position of the third row of seats, has ensured that users feel no sense of restriction on any part of their lower bodies no matter what seat they sit in. This has made the interior seem larger than its actual dimensions and offers a sense of ease of use, fulfilling the design goal of providing a comfortable living space (Fig. 3).

4. Exterior Design

The design theme for the exterior of the vehicle is the “Crouching Coupe Form,” a low-riding form that emphasizes the large-diameter tires, with a front inclination making the vehicle appear ready for take-off; the first impression created by the compact, sophisticated exterior is that the vehicle could not possibly accommodate seven occupants. The compactness and refined dynamism of the new STREAM makes it a vehicle that is at home anywhere in the city.

- (1) The powerfully dynamic exterior form is the result of the integration of a number of design elements: A roof and side window graphics that slope to the rear as they do in a coupe, a solid front nose, a large, rounded and sloping tail end, and the use of large, overhanging fenders despite the compactness of the body (Fig. 4).
- (2) The motif of a “Silver Wolf” was established to further heighten the sense of dynamism produced by the vehicle. In an attempt to project an image of the aggressiveness and sharpness of these animals, the front mask was lowered to represent the face of a wolf, the sculpting of the body was deepened, and a tense, firm front structure was employed, while maintaining a total width of 1695 mm (Fig. 5).



Fig. 4 Side view

Image not displayed for copyright reasons.

Fig. 5 SILVER WOLF

- (3) The rear windshield has been angled as much as possible while ensuring space in the third row of seats, and the rear pillar graphics gradual slope even more strongly from the tail lights. This produces a sense of lightness in the form, making it hard to judge from the exterior that the vehicle is spacious enough for three rows of seats (Fig. 6).

The design elements discussed in (1) to (3) above respond to the needs of users to whom family life is important, but who also value their individual style, and integrate to form an overall design that makes the new STREAM a stylish driving experience whether the driver is alone or accompanied by someone else.

5. Interior Design

The design theme for the interior of the vehicle was “Modern Sporty Interior,” representing a fusion between an advanced “Emotional Sports Cockpit” and the “Modern Comfort” offered by the high-quality, comfortable and spacious seats.

5.1. Emotional Sports Cockpit

On the driver’s side, vehicle controls such as the shift lever and the air conditioner controls have been concentrated in one area, which provides metallic accents; the navigator’s side is simpler but textured and soft to the touch (Fig. 7). In keeping with the



Fig. 6 Rear view



Fig. 7 Emotional sports cockpit



Fig. 8 DUALFACE METER

concept of a “Dual Face,” controls and accessories look different during the day and at night. During the day they have the compactness and texture of metal, while at night the figures glow white and an indirect blue light illuminates the meters, the switches and the cup holders. This means that the cabin has a different “emotional expression” during the day and at night. The Dual Face Meters provide an example of this (Fig. 8).

5.2. Modern Comfort

Under the category of “Modern Comfort,” designers aimed to create an interior that offers quality and comfort whether passengers are sitting in the first, second, or third row of seats. Meticulous attention to detail was lavished on the interior to achieve this goal, extending to the flatness of the floor, the comfort of the seats, and the shape, volume and positioning of storage spaces that can be used everyday (Fig. 9).

6. Color Design

6.1. Exterior

The optimum color design to express the dynamism of the vehicle and the “Silver Wolf” motif was developed under the theme “Sharp & Dynamic.” This was achieved with the development of “Cool Umber M.” This color design involves the use of a sparkling metallic tone integrated with a dark gray base suggesting the aggressiveness of a wolf, but also providing an adult flavor. This color becomes a subtle umber when the light strikes it, and gently



Fig. 9 Modern comfort



Fig. 10 Color design

highlights the design details of the exterior. In addition, a further six colors from the existing color lineup have been matched to the new STREAM and are offered as choices for customers (Fig. 10).

6.2. Interior

“Sport Modern” was selected as the theme for the color design of the vehicle interior to express the style and sophistication of the STREAM’s users. Functional materials offering hard-wearing performance while also appealing to modern tastes were used for the seat covers. The elegant beiges and browns offer a refined color scheme that coordinates with the rest of the cabin (Fig. 11).

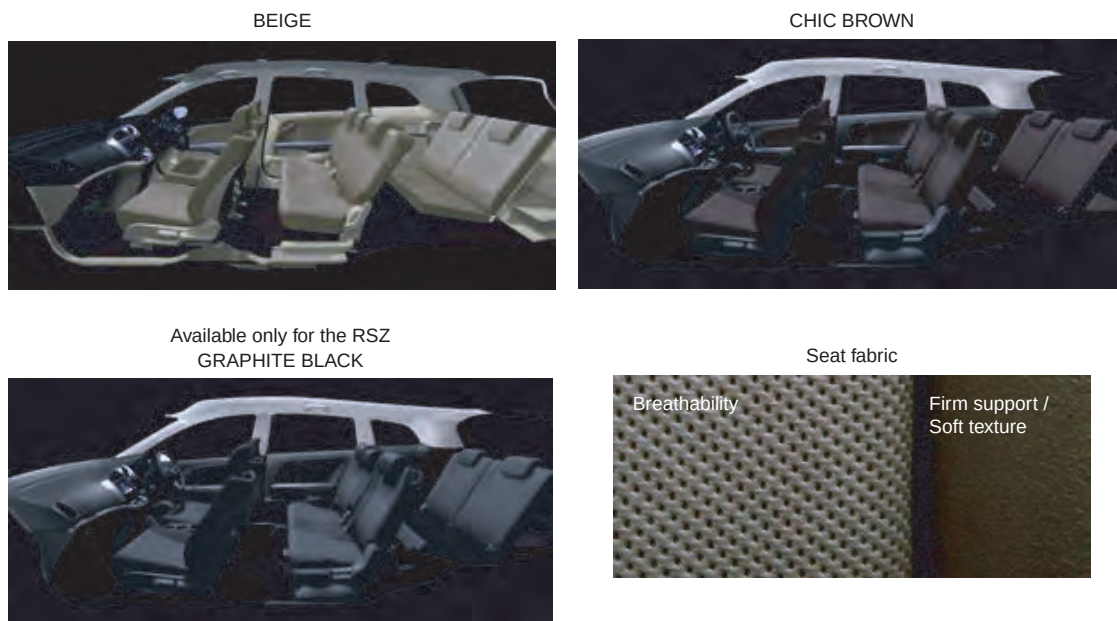


Fig. 11 Interior design

7. RSZ Design

Users seeking sportier performance can select the RSZ model, which offers tuned suspension and a design based on the theme “More Sporty” (“more aggressive”) (Fig. 12).

7.1. Exterior

Centering on a blacked-out grille, the front mask has been designed for a feeling of aggression, with dark-toned headlights and the fog lamps positioned low. That same sense is emphasized even more when viewed from the side, with an aluminum wheel



Fig. 12 RSZ design

design that makes the 17-inch large-diameter tires look even bigger, and a low side sill garnish that makes the body of the vehicle seem firmly in touch with the ground.

7.2. Interior

All the interior fittings of the RSZ are black-toned; the functional components are outlined in a silver metallic color to bring out their forms sharply, creating an adult, sporty feeling. The use of the silver metallic color for the knob of the shift lever and the inner surface of the steering wheel, which are touched by the driver's hands, and the use of silver stitching on the seats, create a sporty feel while coordinating with the rest of the interior.

In "RSZ," "RS" represents the youthful sportiness of the CIVIC RS, while "Z" indicates an ultimate rendition. The sharp and modern design of the RSZ emblem used inside and outside the vehicle appeals to the user seeking a distinctively sporty vehicle.

8. Conclusion

The development of the new STREAM was based on a design concept incorporating the idea of a "Crouching Coupe STREAM" and the aggressiveness and sharpness suggested by the motif of a "Silver Wolf." The result is an exterior possessing a dynamism suggestive of a high-performance sports coupe, and an interior design offering a luxurious and comfortable living space.

The new STREAM embodies a new concept: It is at once a driver's car that symbolizes a stylish lifestyle and offers pride of ownership, and a practical car that accommodates any number of occupants in comfort and increases the pleasure of motoring the more it is used.

Living up to its name, Honda's new minivan gives birth to a new STREAM of automotive excellence.

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Introduction of Honda ASV-3 (Passenger Car)

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ABSTRACT

Honda participated in the Advanced Safety Vehicle Project Phase 3 (ASV-3), which was a national project managed by the Ministry of Land, Infrastructure and Transport (MLIT). In addition to Phase 3, Honda participated in Phases 1 and 2.

Honda's work for the ASV-3 project was based on accident analysis results and focused on the types of accidents caused by phenomena that are outside or barely within a driver's field of view. The three technical approaches of "Collision avoidance," "Collision mitigation," and "Lifesaver" were selected as the three principles supporting the basic Honda ASV-3 concept. The relevant clusters of technology were organized under these headings. The new systems applied the latest sensor and vehicle dynamics control technologies for the driving support systems developed during Phases 1 and 2. The systems use inter-vehicle communication (IVC) technology and a system to assist in emergency rescue activities after accidents occur. These systems were developed with the aim of expanding the range of accident types that ASV systems are capable of handling. An experimental vehicle with these systems installed was built, and future safety technology suggestions were made.

1. Introduction

Following participation in Phases 1 and 2, Honda participated⁽¹⁾ in the Advanced Safety Vehicle Project Phase 3 (ASV-3) managed by the Road Transport Bureau of the Ministry of Land, Infrastructure and Transport (MLIT).

The goal of the Honda ASV-3 project was to build the technologies that had the capability to deal with the types of accidents caused by phenomena that are outside or barely within the driver's field of view. This involved enhancing on-board sensors

that provide external sensing and vehicle dynamics control technology. Both systems were developed for ASV-1 and 2. An additional application of inter-vehicle communication (IVC) technology was also studied. Furthermore, the project involved the study of technology which assists rescue activities after accidents. Additionally, the concepts for these applications were suggested.

This article will provide an overview and introduction of the suggested technical concepts of the Honda ASV-3 on-board systems.

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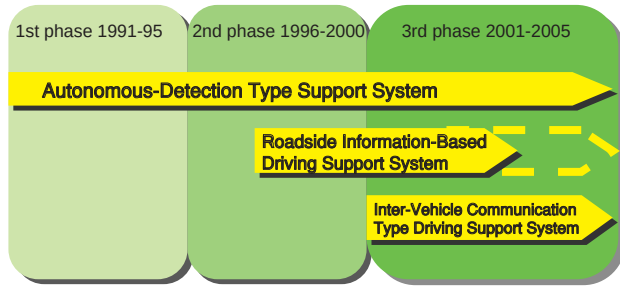


Fig. 1 Technical developments pursued in ASV project

2. The Development Process of ASV

MLIT advocated the ASV project and inaugurated the ASV Promotion Study Group, with the joint participation of representatives from industries, government, and academia. The ASV Promotion Study Group formulated the concept plan for advanced safety technology (hereafter ASV technology). Through the open trials of ASV, it intended to appeal and penetrate the availability of ASV technology to society at large⁽²⁾. MLIT initiated the ASV project in 1991 as a national government project. It finished at the end of fiscal 2005 (Fig. 1).

The objective of Phase 1 (ASV-1, FY91–95) was to verify technical feasibility. ASV technology that could be realized using current advanced technology was developed on that basis, and the effectiveness of the resulting ASV technology in reducing accidents was then verified.

During Phase 1, Honda developed the adaptive front lighting system (AFS) and adaptive cruise control (ACC)⁽³⁾. These technologies were then put to practical use.

The objective of Phase 2 (ASV-2, FY1996–2000) was to perform research and development to find practical applications for ASV technologies. Basic principles and development guidelines were therefore formulated so that the development and dissemination of ASV technology could proceed properly. The development guidelines organized 58 technical items, and projected their effect. New technological developments included a driving support system that uses roadside infrastructure and the roadside information-based driving system. The system was built and proving tests^{(4),(5)} were conducted.

During Phase 2, Honda carried out technical developments for a collision mitigation brake system (CMBS), a night vision system (NVS) and a lane-keep assist system (LKAS)⁽⁶⁾, and implemented practical applications of these technologies. Motorcycles were also included, beginning with Phase 2.

The objective of Phase 3 (ASV-3, FY01–05) was to perform studies for the purpose of penetration of ASV technologies. The technologies which evolved in Phases 1 and 2 were applied, along with the IVC technology.

As described above, the objectives of the ASV project were defined for each phase, with the development and penetration of ASV technology carried out accordingly.

3. Development Goals

An investigation of current traffic accident situations in Japan shows there are a total of over 930 000 accidents and over 1 160 000 casualties (with 9 000 fatalities) in 2000, that is previous FY of the

beginning of ASV-3. This continues to be a major social issue.

According to the percentage of accidents resulting in deaths and severe injuries by accident type (Fig. 2), collisions with pedestrians make up the greatest proportion of fatal accidents (30.9%). Intersection collisions make up the greatest proportion of accidents resulting in severe injury (27.9%).

Figure 3 shows the causes of accidents related to human factors that result in deaths and severe injuries. Accidents caused by recognition errors (failure to confirm safety and failure to watch the road ahead) account for 71% of the total. The next most frequent causes are errors in judgment (20%) and errors in operation (9%).

Given the traffic accident situation described above, Honda worked to develop ASV-3 technology that would reduce collisions with pedestrian and intersection collisions. Three technological approaches were derived from the perspective of mitigating the severity of these kinds of accidents, and were defined as the guiding principles behind the basic Honda ASV-3 concept. The associated clusters of technology were organized as described below. Building on the foundation of these clusters, the on-board systems for the experimental vehicle of Honda ASV-3 were created.

(1) Collision avoidance

This is a technology to prevent and avoid collision, for example, to provide information on the presence of other vehicles or pedestrian for recognition error, and to support the driver's operation for operation error.

(2) Collision mitigation

This is a technology to mitigate collision damage under the circumstances an accident is unavoidable. For example, it provides operational support for misjudgment or operation error.

(3) Lifesaver

This is a technology to assist rescue activities, for example, to automatically notify the status of accident and passengers.

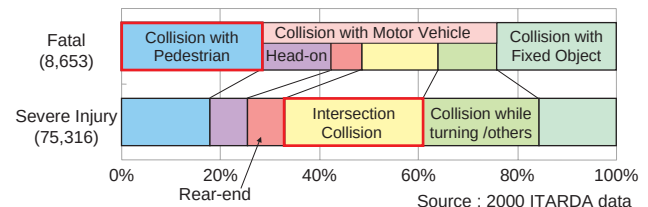


Fig. 2 Rate of the fatalities and severe injuries

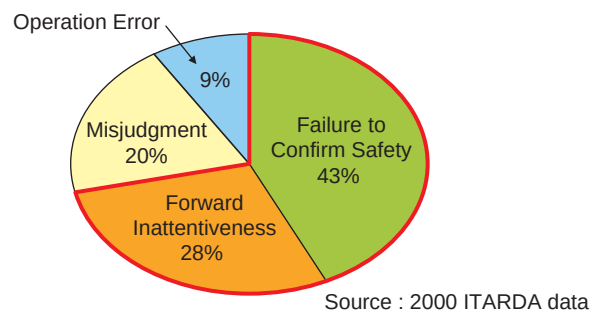


Fig. 3 Rate of Accident Causes

4. Construction of On-board Systems

The three guiding principles of the basic concept, the purposes of the various on-board systems, and the roles played by the systems are depicted in relation to each other in Fig. 4.

4.1. Systems for “Collision Avoidance”

The systems described below were developed to address collisions with pedestrians and intersection collisions.

- Pedestrian detection system with IVC
- Vision-based pedestrian detection system
- Intersection stop assistance system
- Intersection go assistance system
- Forward obstacle collision avoidance assistance system

These systems inform the driver of the presence of vehicles and pedestrians that are outside or barely within the driver’s field of view, which helps prevent collisions. With using on-board sensors, they provide driving assistance, which aids collision avoidance. The systems are intended to help reduce the occurrence of errors in recognition and operation.

4.2. Systems for “Collision Mitigation”

Technology development was conducted for an intersection collision accident mitigation brake system that was intended to address intersection collisions.

This system is intended to mitigate collision damage with using on-board sensors.

4.3. Systems for “Lifesaver”

Technology was developed for the Advanced Mayday System to help assist rescue activities after accidents occur.

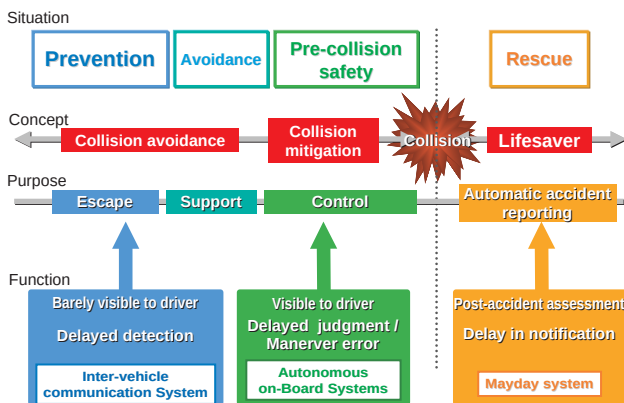


Fig. 4 System concept of Honda ASV-3

5. Technology Introduction of On-Board Systems

Honda ASV-3 experimental vehicle was fabricated as a way of moving this project forward. An appearance of the vehicle is shown in Fig. 5.

5.1. System Configuration

The configuration of the on-board systems is shown in Fig. 6.

5.2. Systems for “Collision Avoidance”

5.2.1. Pedestrian detection system with IVC

This system coordinates information from the on-board sensors with IVC. The system helps detect the presence of pedestrians who are outside or barely within the driver’s field of view.

(1) Functions

The system communicates with mobile communication terminal equipment for pedestrian use and thus can acquire positioning information on pedestrians carrying the equipment, even when they are hidden in the shadow of obstacles ahead of the vehicle. This information is provided to drivers using the on-board screen and voice guidance (Fig. 7, A).

(2) Basic technologies in this system

- Pedestrian detection technology with IVC

The IVC experimental equipment for pedestrians was developed so it can directly acquire the location data of pedestrians and determine the distance and direction of pedestrians with respect to the vehicle. The basic specifications are as shown below. The appearance of the equipment and its circuit diagram are shown in Fig. 8.



Fig. 5 Experimental vehicle of Honda ASV-3

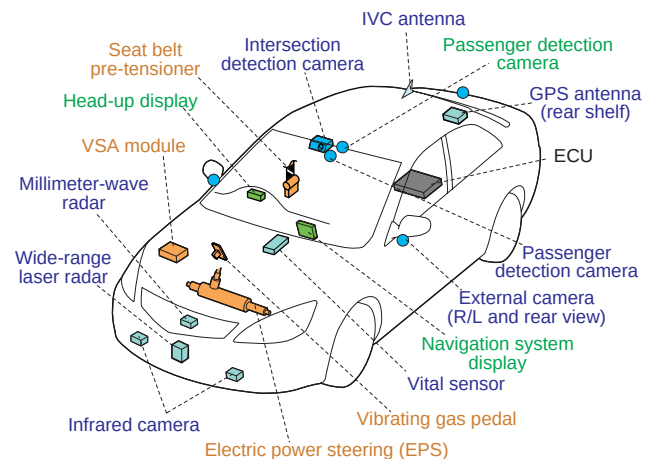


Fig. 6 Configuration of Honda ASV-3

- Basic specifications of IVC experimental equipment for pedestrian
 - Transmission frequency: 5.8 GHz band
 - Transmission power: 10 dBm
 - Positioning system: GPS
 - Size: 150 mm × 110 mm × 39 mm

5.2.2. Vision-based pedestrian detection system

A pedestrian detection system was developed to help detect pedestrians in the daytime, which is out of function of NVS.

(1) Function

Images acquired by the on-board camera are subjected to recognition processing for pedestrians who are in front of the vehicle. The positions of pedestrians are detected and the pedestrian information is provided with the on-board screen and voice guidance (Fig. 7, B).

(2) Basic technology in this system

- Pedestrian detection technology using image processing

This technology is applicable to pedestrians within the driver's field of view. It detects pedestrians by means of pattern matching to recognize and isolate pedestrian silhouettes (Fig. 9).

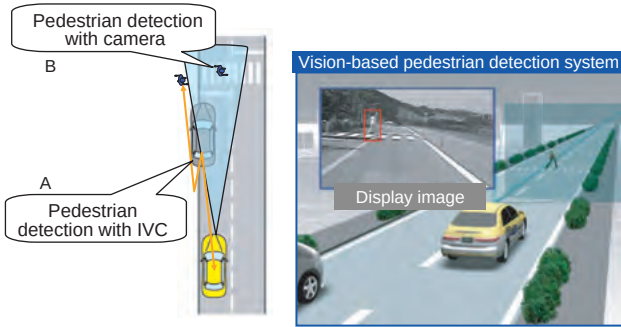


Fig. 7 Function of pedestrian detection system with IVC and vision-based pedestrian detection system

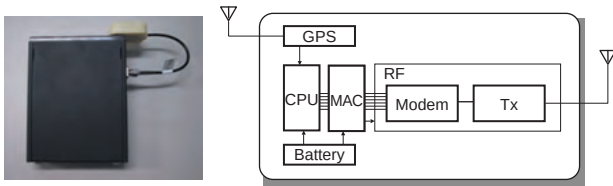


Fig. 8 Appearance and block diagram of IVC experimental equipment



Fig. 9 Vision-based pedestrian detection

5.2.3. Intersection stop assistance system

The purpose of this system is to help reduce intersection collisions due to failure to recognize a stop sign and/or stop line at an intersection (recognition error), resulting in a vehicle suddenly or carelessly entering the priority road.

This system decides the area to provide information with using digital data of intersections and roads in the GPS navigation system, along with Vision-based recognition of stop signs and stop lines.

(1) Function

- This system acquires data of frontal intersections without traffic signals using GPS navigation system. The system provides information on such intersections using the on-board screen to remind drivers stop before they enter the intersections (Fig. 10, A and B).
- The system detects stop signs and stop lines on the road with using Vision-based recognition technology (Fig. 10, C).
- The stopping distance L that allows stopping before the stop line is calculated by the following formula (1). If the vehicle approaches the line closer than the calculated distance without braking, the system provides voice guidance and haptic warning by brake actuation⁽⁷⁾.

$$L = \frac{V^2}{2a} + V \cdot T_d \quad (1)$$

where, V : Vehicle velocity

a : Deceleration in the emergency stop

T_d : Free running time

(2) Basic technologies in this system

- Intersections detection technology using GPS navigation system

The system acquires digital data of intersections by the GPS navigation system. The determination of whether the driver is on a priority road or not, is decided referring to the data of road width, category and structure.

- Vision-based recognition of stop sign

This technology detects stop signs to judge whether the intersections acquired by (a) requires to stop once or not. The system recognizes the placement of a stop sign near an intersection by means of the raw imagery captured with using an on-board camera (Fig. 11, A). It isolates the sign shape based on differences in luminance relative to the surroundings (Fig. 11, B). Next, it ascertains the

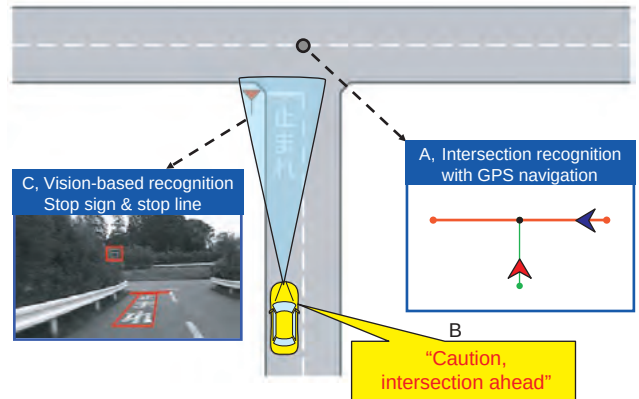


Fig. 10 Function of intersection stop assistance system

characteristic inverted triangle shape to determine that it has identified a stop sign (Fig. 11, C).

(c) Vision-based recognition of stop line

This technology detects stop line to correctly calculate the distance to an intersection. Stop lines are whiter than the road surface and are at right angles to the direction of movement. The line is detected using these characteristics (Fig. 12).

Next, the system calculates the distance to the detected stop line. The distance L to the stop line is derived from the height where on-board camera is mounted, and the angle between the line from the camera to the stop line and road surface, using the following formula (2).

$$L = \frac{H}{\tan \theta} + l' \quad (2)$$

where, H : Height of the camera

θ : Angle between the line from the camera to the stop line and road surface

l' : Correction value with pitching

5.2.4. Intersection go assistance system

The purpose of this system is to help reduce intersection collisions that occur at an intersection when a vehicle emerges suddenly and carelessly into a priority road. This could be caused by a failure of judgment or a failure to recognize an oncoming vehicle. This system detects other vehicles using IVC and provides information on the presence of oncoming vehicles.

(1) Function

- (a) This system acquires position and velocity data of other vehicles on priority roads with using IVC. Also, it provides information on the presence of such vehicles using the on-board screen (Fig. 13, A).

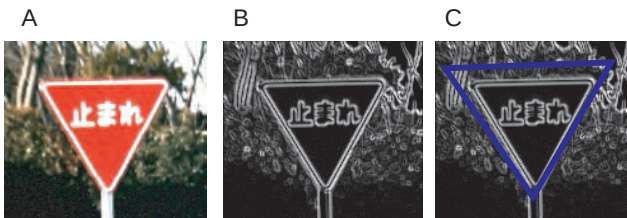


Fig. 11 Vision-based recognition of stop sign

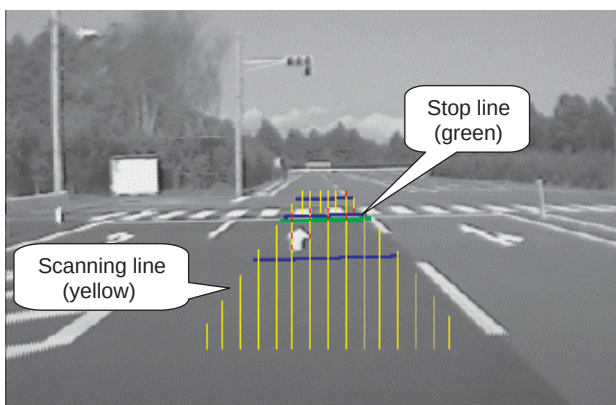


Fig. 12 Vision-Based Recognition of Stop Line

- (b) Based on the acquired data by means of (a), the system calculates a time to arrive at the intersection for other oncoming vehicles. Then it provides information on the presence of other vehicles and informs the driver which will arrive first at the intersection. To enhance the driver's awareness a high-intensity-display on the on-board screen is used (Fig. 13, B).

5.2.5. Forward obstacle collision avoidance assistance system

The purpose of this system is to help reduce intersection collisions with another vehicle that suddenly emerges from a branch road. It assists the driver's operation to avoid collisions by using new systems which were applied during Phase 1 and 2⁽⁸⁾. These systems employ the latest sensor and vehicle dynamics control technologies to support the driver.

(1) Function

- (a) It detects other vehicles that suddenly emerge from branch roads with using on-board radar, and provides braking with using CMBS to decelerate (Fig. 14, A).
- (b) It controls the delay of steering and understeer at the initial stage of driver's operation to avoid collision with using cooperative control between EPS and VSA (Fig. 14, B).
- (c) It controls overshooting the steering at the stage of driver's operation is to avoid a collision. It uses cooperative control between EPS and VSA (Fig. 14, C).
- (d) And it controls the delay of the steering wheel return and oversteer after the driver's operation to avoid a collision. It also uses cooperative control between EPS and VSA (Fig. 14, D).

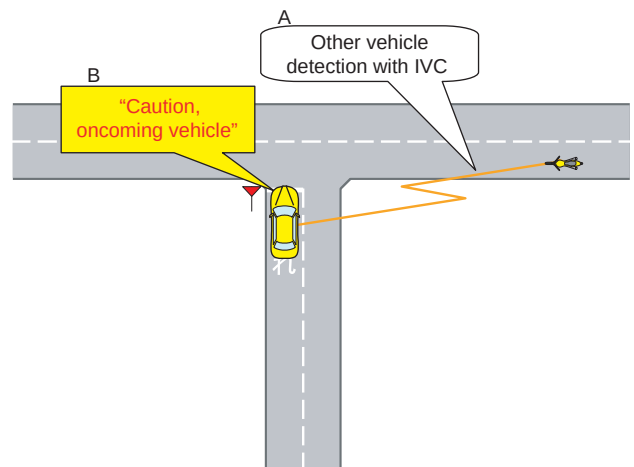


Fig. 13 Function of intersection go assistance system

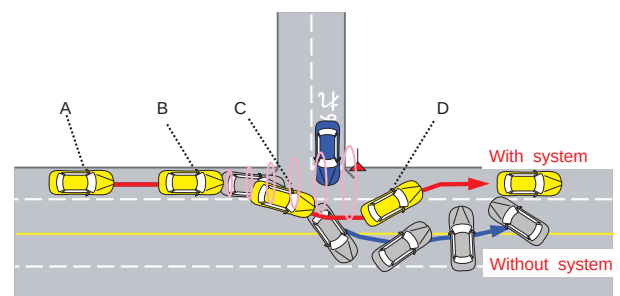


Fig. 14 Function of forward obstacle collision avoidance assistance system

5.3. Systems for “Collision Mitigation”

5.3.1. Intersection collision mitigation brake system

The purpose of this system is to help mitigate collision damage in intersection collisions with the other vehicle that suddenly emerges from a branch road. It helps assist the driver’s recognition for the other vehicle that suddenly emerges from a branch road and helps reduce collision damage by using a CMBS, which has had its sensor function enhanced.

(1) Function

- (a) This system detects the other vehicle that suddenly emerges from a branch road by using an on-board wide-range laser radar. And when there is a possibility of collision with another vehicle, the system applies a voice warning and haptic warning in where the E-Pretensioner gently retracts the seatbelt (Fig. 15, A).
- (b) If the system determines that a collision is unavoidable, the E-Pretensioner strongly retracts the seatbelt and brakes

forcefully. Braking forces from both the system and driver’s operation will help reduce vehicle velocity at the time of impact (Fig. 15, B).

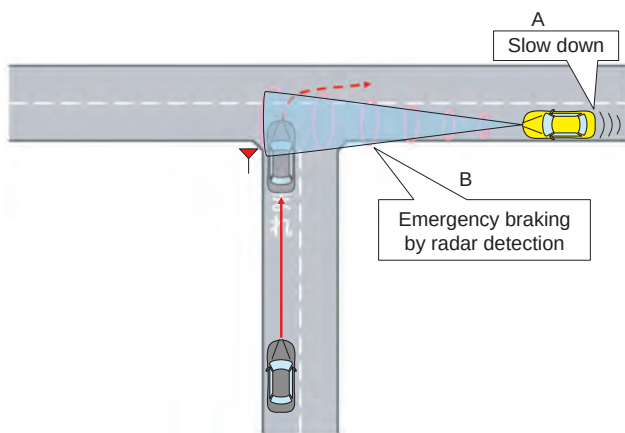


Fig. 15 Function of intersection collision mitigation brake system

6. Conclusion

- (1) Honda participated in the ASV-3 project managed by the MLIT and developed its technologies.
- (2) Based on the results of accident analysis, Collision avoidance, Collision mitigation, and Lifesaver were established as the three principles in support of the basic Honda ASV-3 concept.
- (3) The experimental vehicle of Honda ASV-3 was produced and the future safety technology suggestions were made based on the basic Honda ASV-3 concept.
- (4) The latest sensor and vehicle dynamics control technologies were applied to the ASV-3, therefore the technologies that had the capability to deal with more types of accidents were constructed.
- (5) IVC was applied to the ASV-3, therefore the foundation of technology which has the capability of dealing with the types of accidents which occur due to phenomena that are outside or barely within the driver’s field of view was constructed.

References

- (1) ASV Promotion Study Group (The Ministry of Land, Infrastructure and Transport): ASV: Inter-Vehicle Communication (IVC) based Driving Support System (Pamphlet) (2005)
- (2) ASV Promotion Study Group: Advanced Safety Vehicle Phase 3 (Pamphlet) (2002)
- (3) Tsuruga, T., Nakamura, Y., Matsuda, S., Enomoto, K.: Introduction of HONDA ASV, HONDA R&D Technical Review, Vol. 8, p. 1-11
- (4) Advanced Cruise-Assist Highway System Research Association (AHSRA): AHSRA REPORT, Vol. 6, p. 6-15 (2000)
- (5) Isogai, T.: Overview of Fiscal Year 2001 Research Activities, AHSRA Report, Vol. 10, p. 14-18 (2002)
- (6) Asanuma, N., Kawai, M., Takahashi, A., Tsuji, T.: Introduction of HONDA ASV-2 (Passenger Cars), HONDA R&D Technical Review, Vol. 12, No. 2, p. 1-8
- (7) Mashimo, H., Furukawa, Y., Fukumaru, T., Kaseyama, H.: Experimental Evaluation of Driver’s Braking Operation Assistant System Using Brake Motion Warning, Transactions of Society of Automotive Engineers of Japan, Vol. 33, No. 3, p. 139-144 (2002)

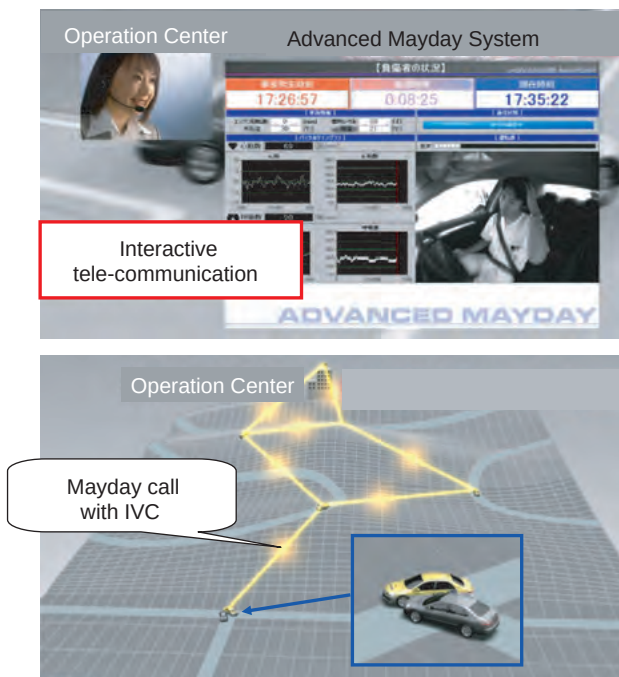


Fig. 16 Advanced Mayday System

- (8) Fujiwara Y., Ohshima, K., Shoda, Y., Kanaboshi, M.: Assistance Method of Obstacle Avoidance Maneuver by Steering Operation, Honda R&D Technical Review, Vol. 17, No. 2, p. 107-114
- (9) Home Page of Honda: Honda Completes Development of ASV-3 Advanced Safety Vehicles
<http://www.world.honda.com/news/2005/c050902.html>

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Introduction of Honda ASV-3 (Motorcycles)

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ABSTRACT

As part of the Advanced Safety Vehicle Phase 3 (ASV-3) project proposed by the Road Transport Bureau of the Ministry of Land, Infrastructure and Transport, Honda has conducted research on predictive active safety technologies. Honda developed the ASV-3 motorcycle based on the concepts of assistance of “visibility” and “conspicuity.” The predictive safety technologies used in the Honda ASV-3 motorcycle consist of inter-vehicle communications technologies, observing sensor technologies and conspicuity enhancement technologies. The inter-vehicle communications system is useful in enhancing both visibility and conspicuity. It is based on research conducted following the development of the inter-vehicle communications system used in the Honda ASV-2. Observing sensor technologies help enhance visibility by using a front-monitoring system and a rear-view system. This enables warning signals to be provided that help prevent rider oversight and reduce blind spots. Conspicuity enhancement technologies help make it easier for the motorcycle to be seen. Specifically, the conspicuity of the motorcycle has been increased using the “FACE design,” which exploits the drivers’ innate ability to recognize the pattern of a human face thus increasing the rate of recognition of the vehicle, and the “LONG design,” which increases awareness of the vehicle’s speed and distance.

1. Introduction

The Advanced Safety Vehicle Phase 3 (ASV-3) project being conducted by the Road Transport Bureau of the Ministry of Land, Infrastructure and Transport was concluded in FY2005, in accordance with its five-year plan. The focus of Phase 3 was inter-vehicle communications technology. This was the core technology for research by the 14 Japanese automotive manufacturers involved in the project. Proving tests using real vehicles, followed by public demonstrations, were conducted at the Tomakomai winter test track of the Civil Engineering Research Institute of Hokkaido between July and October 2005.

As part of this project, Honda conducted further R&D on the

inter-vehicle communications technology developed for the Honda ASV-2. This not only involved the development of inter-vehicle communications technologies, but also focused on research related to driving support systems and the design of motorcycles based on accident analyses. This increases their conspicuity and enables drivers of other vehicles to see them more easily, thus preventing errors in judgment. The results of these studies were incorporated into the Honda ASV-3 motorcycle and passenger car. This was made public in September 2005 (Fig. 1).

This paper provides an overview of the inter-vehicle communications technologies, observing sensor technologies and conspicuity enhancement technologies that were used in the Honda ASV-3 motorcycle.

* Motorcycle R&D Center



Fig. 1 Honda ASV-3

2. Accident Analyses

Figure 2 shows a breakdown of ratios of fatal collisions involving motorcycles by type of collision⁽¹⁾.

Collisions involving a right-turning automobile with straight-going motorcycle at intersection, and collisions at crossing paths account for more than 60% of the accidents.

Figure 3 shows a breakdown of the collisions caused by automobiles⁽¹⁾. Accidents caused by oversights and misjudgments of approach speed account for 70% of the collisions between automobiles and motorcycles. These are lapses in judgment and awareness on the part of the automobile drivers. It can also be assumed that lack of prediction and anticipation skills and lapses in judgment by motorcycle riders also played a part.

It is therefore highly possible that information related to the position of other vehicles will help reduce misjudgments and lapses in awareness by both motorcycle riders and automobile drivers. This will enable some accidents to be prevented or the severity mitigated.

The following causes of lapses on the part of automobile drivers can be suggested:

(1) Drivers misjudge the approach speed and the distance between

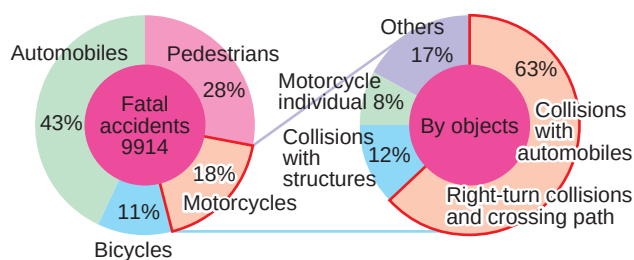


Fig. 2 Breakdown of motorcycle fatal accidents in Japan

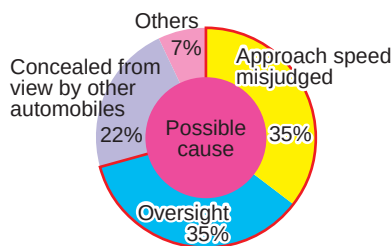


Fig. 3 Breakdown of accidents caused by automobiles

their vehicles and motorcycles because of the smaller motorcycle profile.

- (2) Drivers overlook the presence of motorcycles because their smaller profiles cause them to be hidden by other vehicles and surrounding objects.

Assisting automobile driver awareness by increasing the conspicuity of motorcycles could therefore be expected to help enable automobile drivers to more quickly identify a motorcycle in traffic and more appropriately judge distance and approach speed.

3. Features of the Honda ASV-3 Motorcycle

3.1. Concept

As accident analyses have indicated, assistance of “visibility” and “conspicuity” are essential in helping reduce the human error that causes accidents (Fig. 4). To enhance visibility, the Honda ASV-3 motorcycle is provided with a front-monitoring system and a rear-view system as well as an inter-vehicle communications system.

Crossing path collisions typically occur at intersections without traffic signals. When the motorcycle fails to come to a complete stop, driver oversight can easily occur. The Honda ASV-3 motorcycle uses a front-monitoring system with image recognition technology to assist the motorcycle rider when stopping at an intersection. This was developed with consideration to cost and ease of mounting on a motorcycle. Additionally, human error is further reduced by using an inter-vehicle communications system to reduce misjudgments of distance and speed of other vehicles approaching the intersection when the motorcycle starts again after stopping. Since there is a blind spot in a motorcycle's rear-view mirror, an increased awareness of the traffic situation behind the vehicle is also necessary during lane changes, etc. The Honda ASV-3 motorcycle is also provided with a rear-view system.

The inter-vehicle communications system is capable of sensing vehicles at long distances (around 50-200 m), but its accuracy in detecting position is low around 30 m. This system can be produced at a lower cost compared to observing sensor technology. The observing sensor technology displays a higher level of accuracy in detecting vehicle position. However, it can only detect vehicles at a range of around 100 m. The inter-vehicle communications system and observing sensor technology can therefore be considered highly compatible and can be used to supplement each other. In addition, sensor fusion is expected to provide a higher level of driver support.

To assist conspicuity, the Honda ASV-3 motorcycle is provided with an inter-vehicle communications system and technology for increased conspicuity. Analysis of accident statistics indicates that motorcycles can be easily overlooked. Their distance and speed

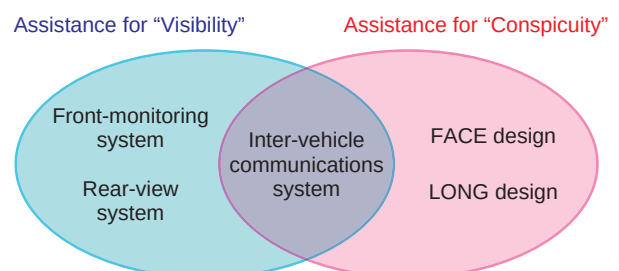


Fig. 4 Honda ASV-3 motorcycle concepts

can be easily misjudged by other vehicle drivers. This makes enhancements for increased conspicuity beneficial. The inter-vehicle communications system is also able to provide enhancement for conspicuity by transmitting its own information (position, speed, etc.) to other vehicles nearby. Motorcycles are also overlooked or have their distance and speed misjudged because they are smaller than automobiles. The Honda ASV-3 motorcycle further attempts to reduce human error by using the “FACE design” and the “LONG design.” These enhance conspicuity using the capacity for pattern recognition of the human face and characteristics of distance and speed recognition.

3.2. Technological Overview

Figure 5 shows the system layout when installed, and Fig. 6 shows the structure of the system.

3.2.1. Inter-vehicle communications system

(1) Communications system The inter-vehicle communications system provides an enhancement for both visibility and conspicuity.

Honda began research on safer vehicle operation when they applied the wireless communication technology to transfer information between vehicles in 1990s. The results of this research were used in the inter-vehicle communications system fitted into the Honda ASV-2. This was made public in 2000⁽²⁾.

The study of system requirements progressed further for the Honda ASV-3, and the system configuration was established as follows:

(A) Communications requirements:

- Maximum distance: 500 m
- Maximum number of vehicles for inter-vehicle communications: 120
- Communication interval: 100 ms

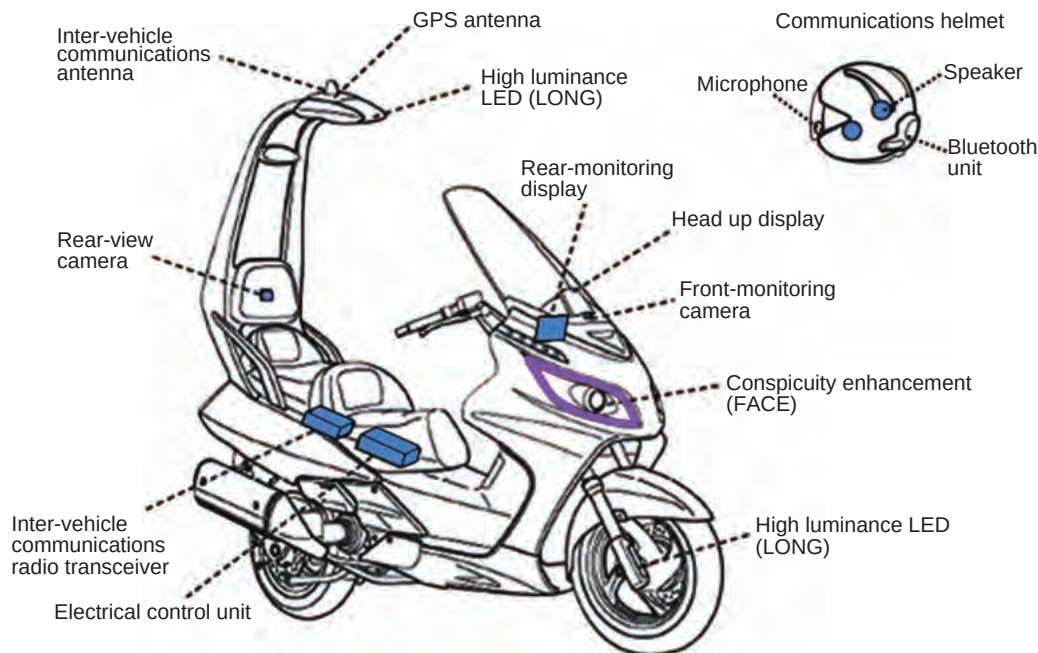


Fig. 5 System layout

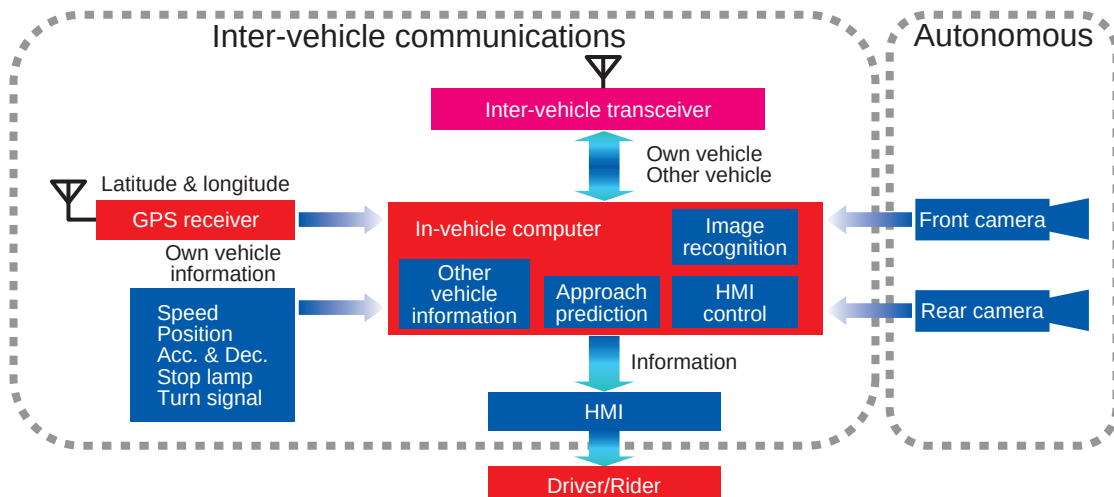


Fig. 6 Structure of inter-vehicle communications system and autonomous system

- (B) Wireless communications equipment:
 - Frequency: 5.8 GHz
 - Transmitting output: 10 dBm
- (C) System control
 - Control interval: 10 ms
- (D) GPS receiver
 - GPS data interval: 1 s
 - Estimation interval of subject vehicle's position: 100 ms

The information from the inter-vehicle communications radio transceiver, GPS receiver, front-monitoring camera and subject vehicle are inputted into the onboard control computer every 100 ms. The computer processes control timing and sequence of providing information to the rider, according to the synthetic situations of inputted information.

Because the communication interval is 100 ms while the GPS data interval is 1 s, the real position of the vehicle will have moved forward in the interval time under which the GPS data is updated. Assuming a vehicle speed of 90 km/h, the distance traveled in this time would be 25 m. Therefore, errors could occur in vehicle position of transmitted information. To correct these errors, the vehicle's longitude and latitude data are predicted every 100 ms from its wheel pulse and yaw rate.

(2) Head-Up Display (HUD) and Human-Machine Interface (HMI) providing acoustic information A method of providing information to the rider via an HUD using a system of zones to indicate the direction of danger has been developed. The merit of this method is that the rider can awake to indication of the HUD via peripheral vision, without having to focus on the image. The HUD also includes a speedometer area, enabling the rider to access the area with a comparatively small change in line of vision (Fig. 7).

The communications helmet used in the system enables acoustic information to be provided to the rider at the same time. The communications helmet is fitted with a speaker and a microphone. This also enables the rider and a passenger to talk. The onboard computer is preprogrammed with a variety of acoustic data that can be presented to the rider as required, depending on the situation. Bluetooth wireless technology is used to transmit data between the onboard computer and the communications helmet (Fig. 8).

3.2.2. Observing sensor technology

The external monitoring system is made up of a front-monitoring system and a rear-view system to enhance visibility.

(1) Front observation assistance Analysis of images from the front-monitoring camera shown in Fig. 9 enables the system to recognize an intersection ahead; information of the presence of an intersection can therefore be provided as a rider approaches the intersection.



Fig. 7 Head up display



Fig. 8 Communications helmet

As Fig. 10 shows, image analysis isolates the stop sign, stop line and "stop" painted on the road from the image data provided by the front-monitoring camera. If the rider does not apply the brakes, or slow down to less than a predetermined speed, information that the vehicle is approaching an intersection will be indicated on the HUD with acoustic information to encourage the rider to stop.

(2) Rear-view assistance As shown in Fig. 11, the ASV-3 motorcycle is equipped with a rear-view camera. Figures 12 and 13 show the type of images from the rear of the motorcycle that are displayed on the LCD at the bottom of the HUD.

Conventional motorcycle rear-view mirrors are sometimes partially obscured by the rider's body. This makes it difficult for the rider to check directly behind the vehicle during lane changes, etc. Positioning the rear-view mirrors even further out enables a wider range of monitoring around the rear of the vehicle than prior art. This helps reduce the blind spot (Fig. 14).

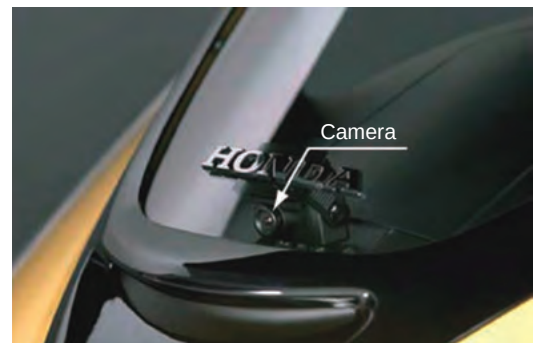


Fig. 9 Front-monitoring camera

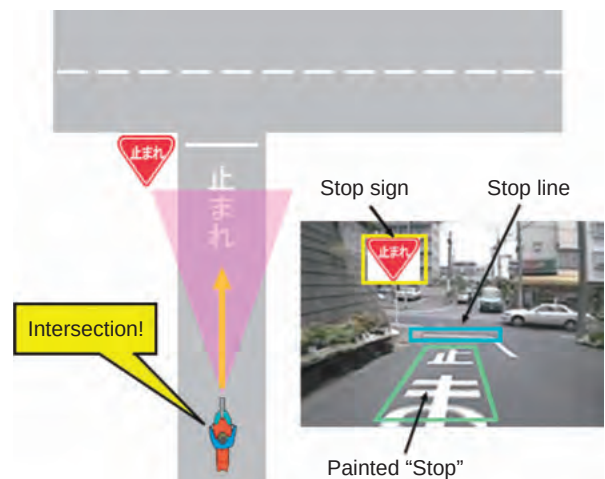


Fig. 10 Intersection approach support



Fig. 11 Rear view camera



Fig. 12 Example of HUD image: 1



Fig. 13 Example of HUD image: 2

3.2.3. Conspicuity enhancement technology

This technology is used to increase the ease with which other drivers are able to recognize motorcycles.

(1) FACE (Facial Attention to help Conspicuity Enhancement) design Increasing the conspicuity of motorcycles is effective in helping prevent them from being overlooked by the drivers of other vehicles.

Neurological research has indicated the existence of neurons that respond more quickly to specific patterns; in 1981, the discovery of “face neurons” was reported⁽³⁾.

The fact is well known that awareness of motorcycles is delayed, because motorcycles are small and can be hidden by trucks and/or passenger cars in a mixed traffic environment. Research has therefore been conducted on a design of the front of motorcycles (FACE design) that exploits the ability to recognize faces that is innate to humans. This further enhances the conspicuity of motorcycles in mixed traffic environments. Specifically, this

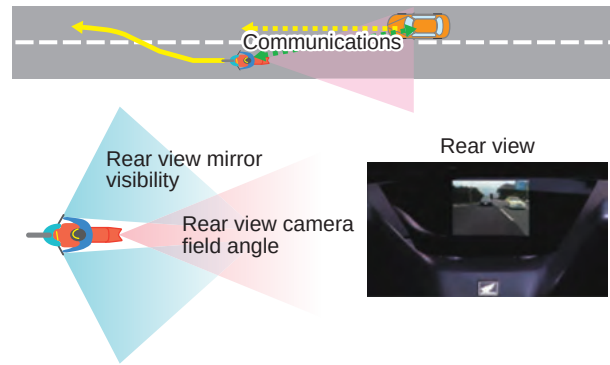


Fig. 14 Rear view support

involves a lighting design on the front of a motorcycle that resembles a human face. The design genuinely suggests a face during the day and at night⁽⁴⁾.

Images of mixed traffic of the type shown in Fig. 15 were formulated using an image simulator. These images were used in tests to determine whether test subjects could identify the motorcycle when shown the images for a short period. The results of these tests showed that subjects were approximately twice as likely to recognize a motorcycle with front design elements resembling a face than a conventional motorcycle (Fig. 16).

Functional Magnetic Resonance Imaging (fMRI) measurements of the brain function (Fig. 17) when shown an image of a real human face and a motorcycle with design elements suggesting a face are very similar. This indicates that the design causes activity in part of the brain that is extremely sensitive to the “face” pattern (Fig. 18).

To ensure that the front of the Honda ASV-3 motorcycle suggests a face at a glance, the projector headlamps are shaped like pupils;

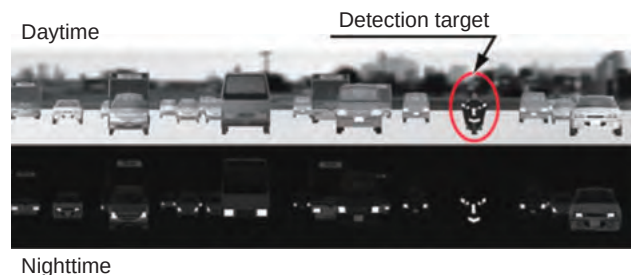


Fig. 15 Mixed traffic by image simulator

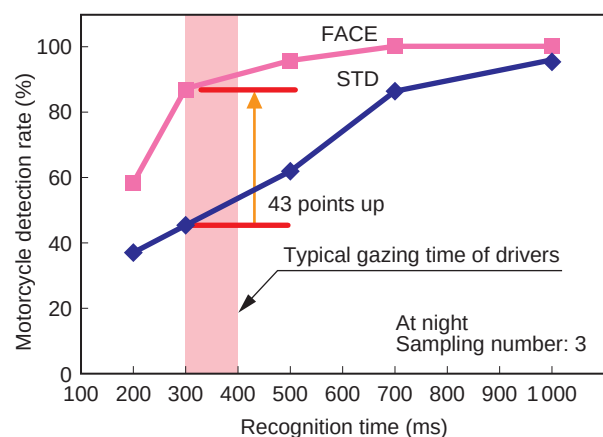


Fig. 16 Results of probability-of-detection experiment at night

the outline of the “eyes” and their “whites” also glow when the lights are on. This further reinforces the image of eyes, especially at night (Figs. 19 and 20).

(2) Longitudinal Oriented Normative time-Gap compensate (LONG) design Driver misjudgment of the distance and speed of oncoming motorcycles has been shown to be one of the main causes of accidents involving a right-turn collision or a crossing paths collision (Fig. 3).

To enhance distance awareness, lights have been positioned low on the body of the motorcycle to emphasize the point at which the vehicle contacts the road. At the same time, to increase awareness of the motorcycle’s speed, lights have been positioned high on the body to emphasize the size of the body, making it easier for drivers to judge exact speed (Fig. 21)⁽⁵⁾.

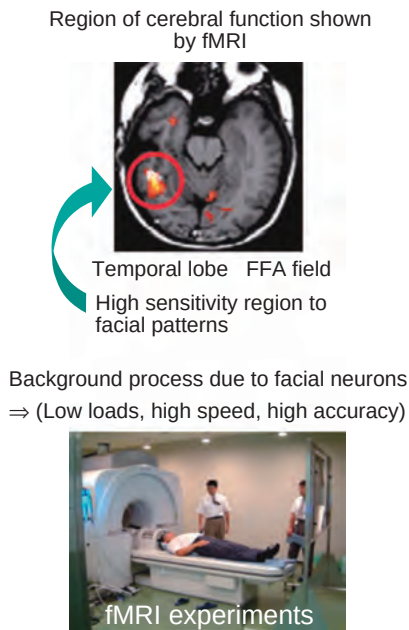


Fig. 17 Brain function analysis using fMRI

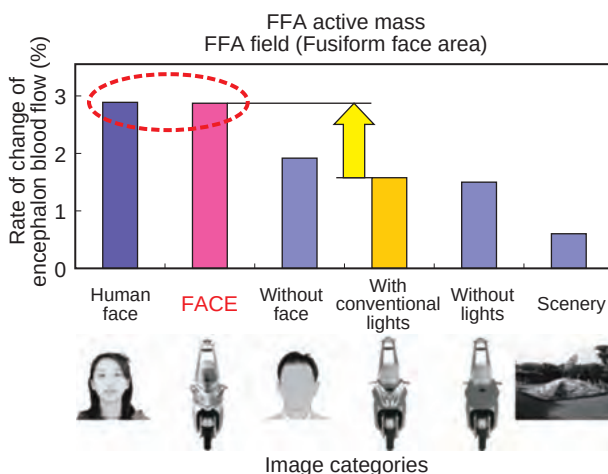


Fig. 18 Assessment of face-recognition using fMRI

A motorcycle equipped with lights following the LONG design (LONG) produces an almost identical awareness of the distance as for a normal passenger vehicle (CAR). This represents an enhancement of approximately 10% compared with a standard motorcycle (STD) (Fig. 22). In addition, LONG displays an enhancement of 20% over STD in speed awareness. The results of measurements of distances at which speed differentials could be discriminated for a standard stimulus (60 km/h) showed that a more accurate discrimination was possible for a greater distance for LONG than for STD (Fig. 23).



Fig. 19 FACE design for Honda ASV-3



Fig. 20 Comparison of FACE design (daytime and nighttime comparison)

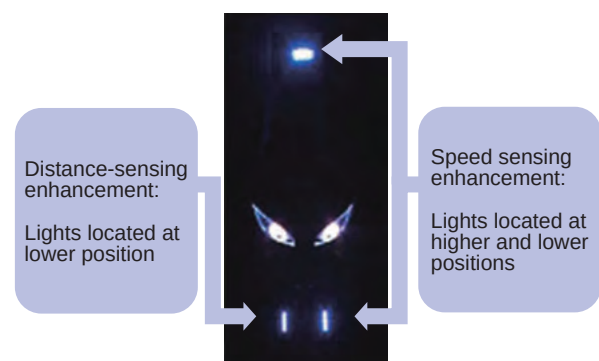


Fig. 21 LONG design lamp arrangement

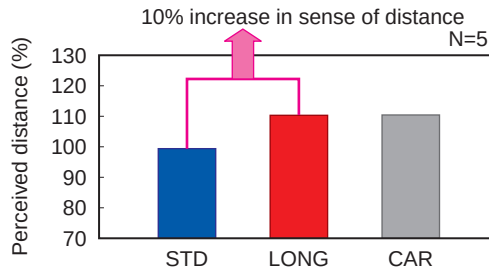


Fig. 22 LONG effect (sense of distance)

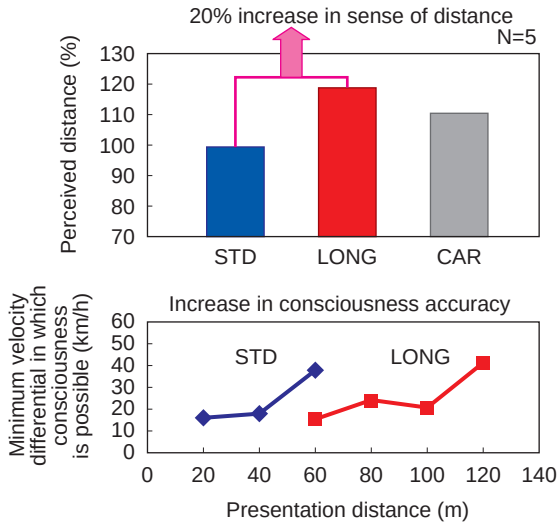


Fig. 23 LONG effect (sense of speed)

4. Proving Tests of Inter-Vehicle Communications System

4.1. Overview

In phase 3 of ASV by MLIT, proving tests of driving support systems using inter-vehicle communications were conducted jointly by 14 domestic automotive manufacturers at the Tomakomai winter test track of the Civil Engineering Research Institute of Hokkaido.

These tests focused on seven total accident types: four types of accidents for which oversight was considered the major cause (crossing paths collisions, right-turn collisions, head-on collisions and pedestrian collisions); and three types of accidents that cause serious damage, generating a significant social need for mitigation (rear-end collisions, left-turn collisions and lane change collisions).

Table 1 shows the categories in which the respective manufacturers were involved. Honda's tests, for the Honda ASV-3 motorcycle and passenger car, covered all seven accident types.

4.2. Results of Proving Tests and Considerations

The driving support system used in the Honda ASV-3 motorcycle does not use digital maps for car navigation systems; the application has been designed to encourage direct visual perception and thus increase awareness.

Tests related to motorcycle accident scenarios involving right-turn collisions and crossing path collisions indicate the inter-vehicle communications system was effective in supporting the ability to judge the possibility of a collision. This stimulates visual awareness in riders.

In tests of accident scenarios involving head-on, rear-end and left-turn collisions, insufficient accuracy in lane information caused the effectiveness of the system to be limited to the level of support of providing information on the presence of other vehicles. However, a driving support system using inter-vehicle communications can be expected to be effective in helping reduce the number of left-turn collisions by notifying vehicle drivers to the presence of motorcycles.

5. Conclusion

This paper has mentioned the inter-vehicle communications system, observing sensor technology and conspicuity enhancement technology used in the Honda ASV-3 motorcycle. The basic concept behind these safety technologies is assistance for enhanced visibility and conspicuity. The aim is to help reduce and possibly prevent accidents by reducing human errors that can cause them. The effectiveness of the inter-vehicle communications system and observing sensor technology was verified in proving tests conducted

Table 1 Domestic car maker participation in operation support system verification experiment

○: Participated

Accident type		Right turn	Crossing path	Pedestrian	Head-on	Rear end	Left turn	Lane change
Honda	(Motorcycles)	○	○		○	○	○	
Honda	(Passenger cars)		○	○		○		○
Isuzu	(Trucks)		○			○	○	
Suzuki and Kawasaki	(Motorcycles)	○	○				○	
Suzuki	(Passenger cars)	○	○				○	
Daihatsu	(Passenger cars)		○			○		
Toyota	(Passenger cars)	○	○		○	○		
Nissan	(Passenger cars)	○	○			○	○	○
Nissan Diesel	(Trucks)					○	○	
Hino	(Trucks)	○				○		
Fuji Heavy Industries	(Passenger cars)	○	○		○	○		○
Mazda	(Passenger cars)	○	○	○	○	○	○	○
Mitsubishi	(Passenger cars)		○			○		
Mitsubishi Fuso	(Trucks)					○	○	
Yamaha	(Motorcycles)	○	○	○		○		○

jointly by 14 domestic automotive manufacturers. Conspicuity enhancement using the “FACE design” and the “LONG design” were shown in simulator tests to increase the ratio of recognition and awareness of speed and distance. Research is ongoing on the practical application of these designs.

6. Afterword

The IT New Reform Strategy⁽⁶⁾ announced by the Cabinet Office on January 18, 2006, specified a government target for the reduction of traffic accident fatalities. The target is below 5 000 persons annually by the end of 2012. From 2006, the public and private sectors will begin R&D collaboration that includes large-scale tests on systems to link infrastructure (inter-vehicle communications, vehicle-to-road communications, road-to-vehicle communications) with national diffusion of the systems. The target for practical use is early 2010.

References

- (1) Institute for Traffic Accident Research and Data Analysis, Japan: Koutsujiko Toukei Nenpou 1998 (in Japanese)
- (2) Yamamoto, K., Kushida, K., Iijima, S., Yamagawa, T., Kodaira, S.: Introduction of HONDA ASV-2 (Motorcycle), HONDA R&D Technical Review, Vol. 12, No. 2, p. 9-16
- (3) Bruce, C., Desimone, R., Gross, C. G.: Visual properties of neurons in a polysensory area in superior temporal sulcus of the macaque, *Journal of Neurophysiology*, 46, p. 369-384 (1981)
- (4) Sugawara, T., Matsumoto, Y., Kimura, M., Kushida, K., Murata, Y.: Enhancement of Motorcycle Conspicuity Using Brain Activities Induced by the Facial Image Stimuli, 2005 JSAE Annual Congress Proceedings, No. 50-5 (2005)
- (5) Maruyama, K., Tsutsumi, Y., Kushida, K.: A Study of a New Lighting System for Better Conspicuity of a Motorcycle, 2005 JSAE Annual Congress Proceedings, No. 94-05 (2005)
- (6) IT Strategic Headquarters: IT Shin Kaikaku Senryaku, p. 19 (2006) (in Japanese), http://www.kantei.go.jp/foreign/koizumipphoto/2006/01/19it_e.html

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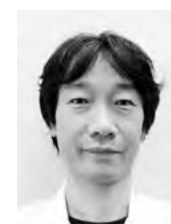
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Research on Electrically Powered Small Commuter Scooter

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ABSTRACT

For the study of electrically powered small commuter scooter, we focused on short distance urban travel to set a practical cruising distance. High energy density Ni-MH battery, power unit module with inverter, and a motor with single gear reduction were selected to ensure a compact lightweight design with necessary drive power. Part layout was optimized with consideration for front axle weight to provide good handling and stability while reducing vehicle size. While keeping the riding posture the same as a basic scooter, the lower seat height and narrower floor enhanced handling and the user's ability to touch the ground. A lever type accelerator was applied to enhance the operation of this scooter. As a result, an electrically powered small scooter with flat floor which is clean, quiet, and smooth, was realized with practical handling and stability. The scooter with its 1 285 mm in length, 44 kg mass, and 15 km cruising distance can be easily handled even by novice riders.

1. Introduction

During the 1980s the number of scooter users increased rapidly as a result of the commercialization of the lightweight low-priced class 1 motorized bicycle which could be easily used by people switching from a bicycle to a scooter for the first time. With the subsequent evolution of scooters however, their size and mass have increased which has made it more difficult for first-time scooter users to adequately handle a scooter.

In 1994, Honda commenced mass production of its first electrically powered scooter, the CUV-ES. This scooter was intended as a replacement model for some engine-driven small scooters and was aimed at keeping the same driving performance as an engine-driven scooter. However, to attain the target cruising distance it was necessary to install a large capacity battery which

posed the issue of increased mass of the scooter. Also, because this scooter was sold on a lease basis it did not become common on the market. Subsequently, Electric Vehicle (EV) technology for the battery, power devices for the inverter, and control of the motor current have evolved, making it possible to achieve higher performance with a greater degree of compactness.

In light of this situation research started on an electrically powered small scooter suitable for first-time scooter users which was intended to be quiet, smooth running, and easy handling. From the viewpoint of popularizing electrically powered scooters the concept of the scooter was developed to that of a commuter for city driving. Under this concept, easy handling and the ability to park it in small spaces were featured. The mass of the expensive battery was reduced with limitation to driving performance and cruising distance and research was carried out accordingly.

* Motorcycle R&D Center

This paper describes the vehicle concept of a prototype electrically powered small scooter and the main technologies used in it (Fig. 1).

2. Vehicle Concept

2.1. Size and Mass of the Body

As a result of analyzing the body size and mass of a class 1 motorized bicycle (gasoline engine) it was considered desirable to use a small lightweight EV scooter as the target of research.

In the 1980s, small, lightweight scooters were commercialized, and because of their ease of handling the number of users increased, particularly first-time users switching from bicycles. These scooters were further evolved to meet the needs of the market by enhancing performance and providing storage space for a helmet among other features. Subsequently, scooters became progressively larger and heavier (Fig. 2). At present, there are no scooters with an overall length of 1 500 mm or less and a mass of 50 kg or less. These are the scooters that are gladly accepted by first-time users. It is thought that people still need to switch from using a bicycle to using a scooter so the target of the research was set as a compact and lightweight EV scooter that first-time users could handle easily.

2.2. Driving Performance and Cruising Distance

Increasing the mass of the battery to obtain the required driving

performance and cruising distance causes the mass and the cost of the EV scooter to increase. In this research, it was considered important to restrict the scope of application of the electrically powered small scooter to a certain extent and set the driving performance, cruising distance, and battery weight to match this scope of application.

In 1994, development of the CUV-ES (Fig. 3) for mass production started. This scooter was intended as a substitute for conventional scooters powered by engines and was designed to equal the driving performance of a conventional scooter as well as a relatively long cruising distance. Although this scooter had a maximum speed of 60 km/h and a cruising distance of 35 km (ECE40 mode), the amount of batteries required for this performance level were largely responsible for the scooter's overall mass of 130 kg, which was roughly twice that of a conventional scooter powered by an engine. Also, since the retail price was 850 000 yen (3-year lease), it could not satisfy users of conventional scooters. The EV scooter was commercially practical at that time, but it was unable to penetrate the market.

In the next approach, the CAIXA, MOBIMOB, and other concept models were announced at the 2001 Tokyo Motor Show (Fig. 4). Although these models were manufactured for separate applications, all of them were EV that used a motor as the power source. While complementing the applications of automobiles, these scooters were good for city driving or limited-distance urban travel and all featured reduced body size, limited driving performance, and cruising distance due to a smaller battery.



Fig. 1 Appearance of prototype small EV scooter

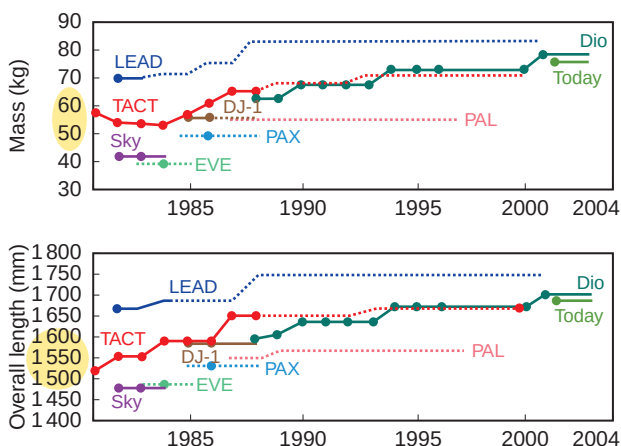


Fig. 2 Changes of size and mass of scooters



Fig. 3 Appearance of CUV-ES



Fig. 4 Appearance of concept models

From the above, it was judged best from the viewpoint of lowering the overall mass and price to design all EV small scooters specifically for city driving and to set the size of the battery, driving performance, and the cruising distance with consideration of practical applications.

2.3. Concrete Concept

A concrete concept was determined taking into account the fact that the scooter was intended for users who were switching from bicycles to scooters for the first time and the scooter would have various features as an EV. Based on that concept, we studied the direction to be emphasized to design an electrically powered small scooter that would be more readily accepted by users (Fig. 5).

For short distance urban travel, bicycles are used as well as scooters. It is anticipated that bicycle users who do not demand the cruising distance or driving performance of a full-fledged scooter will become users of electrically powered small scooters. We decided that it would be desirable to appeal to these users using the fact that this scooter uses a motor instead of an engine as the power source and is easier than a bicycle to use. Thus, the main concept was condensed into the catchphrase “Comfortable mobility appliance.” Based on this catchphrase, it was necessary to provide functions and performance that emphasized user-friendliness, convenience, and a compact lightweight body which represent the image of an electrical appliance.

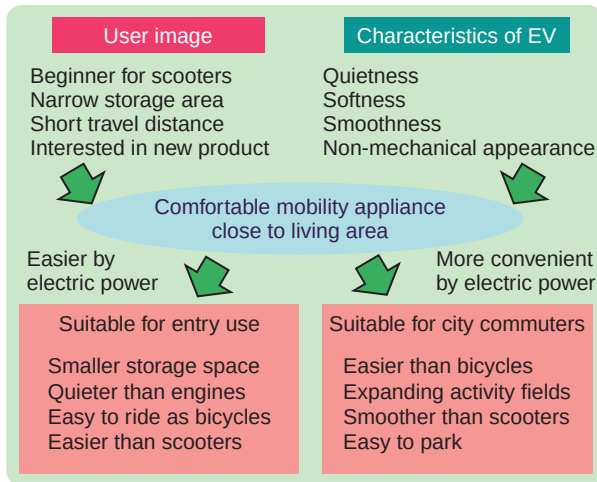


Fig. 5 Concept map of small EV scooter

Also, it was thought the ability to use such a scooter over a wide area, its small size, easy handling, and parking would contribute to easy use and convenience. Regarding driving performance, it was thought that potential users would make a comparison with a bicycle and would therefore not make strong demands for high performance.

3. Outline of an Electrically Powered Small Scooter

3.1. Specifications of an Electrically Powered Small Scooter

Table 1 shows the major specifications of an electrically powered small scooter. Emphasis was placed on reducing overall length. At the same time, the type and size of the battery was set by balancing mass, price, and performance.

The overall length was made 1 285 mm to allow the scooter to be parked inside the front entrance of a home or taken into the elevator of a condominium and smaller than the width of the rear hatchback door of a minivan. To enable the overall length to be reduced small diameter 8-inch tires were used instead of 10-inch tires which is the normal size of tires used on a scooter or motorized bicycle. The maximum speed was set to 40 km/h and the practical cruising distance was set to 15 km as the minimum performance required for a scooter used for city driving. The maximum speed setting was determined as the speed that enabled the difference in the time required to travel a short distance to be ignored and that was fast enough that the driver felt no psychological pressure on a road with other vehicles. The practical cruising distance was determined from the data in Fig. 6, which indicates that 92% of bicycle users travel no more than 9 km in a day⁽¹⁾ and also that 77% of users of engine-driven scooters rode an average distance of no more than 10 km per day⁽²⁾. The practical cruising distance of 15 km that was set included an allowance factor of 1.5. To reduce the body size, mass, and price while achieving the determined target driving performance and cruising distance, a high energy density Ni-MH battery was used. A specification comparison with the CUV-ES which has the same performance as a general engine-driven scooter shows that the vehicle curb mass is about 1/3 and the capacity of the battery is about 1/5. These specifications coupled with the easy handling and reduction of price of the battery resulted in specifications that are near the realistic zone in which popularization of the scooter is feasible.

Table 1 Specifications of EV scooter and CUV-ES

Items	Specifications	
	Small EV scooter	CUV-ES
Length × Width × Height (mm)	1 285 × 600 × 930	1 785 × 615 × 1 035
Vehicle curb mass (kg)	44	130
Motor type	Permanent magnet synchronous motor	Permanent magnet synchronous motor
Rated power (kW)	0.58	0.58
Maximum power (kW)	1.15	3.25
Battery type (V-Ah)	Ni-MH 48 V-7.5 Ah	Ni-Cd 86.4 V-20 Ah
Maximum speed (km/h)	40	60
Cruising distance (km/charge)	15 (300 m go-stop)	35 (ECE40 mode)
Charging hours (hr)	3	8

3.2. Electrically Powered System Configuration

Figure 7 shows the configuration of the electrically power system. A rated power supply voltage of 48 V was applied with consideration of capacity of cell, discharge performance of the battery, breakdown voltage of the power devices in the inverter, and the allowed voltage and current capacity of the wiring components. The rated voltage of the lights and other components was 12 V, so the 48 V power was converted by a DC-DC converter to 12 V and supplied to the lighting components. The 48 V drive system circuit was isolated from the 12 V lighting circuit.

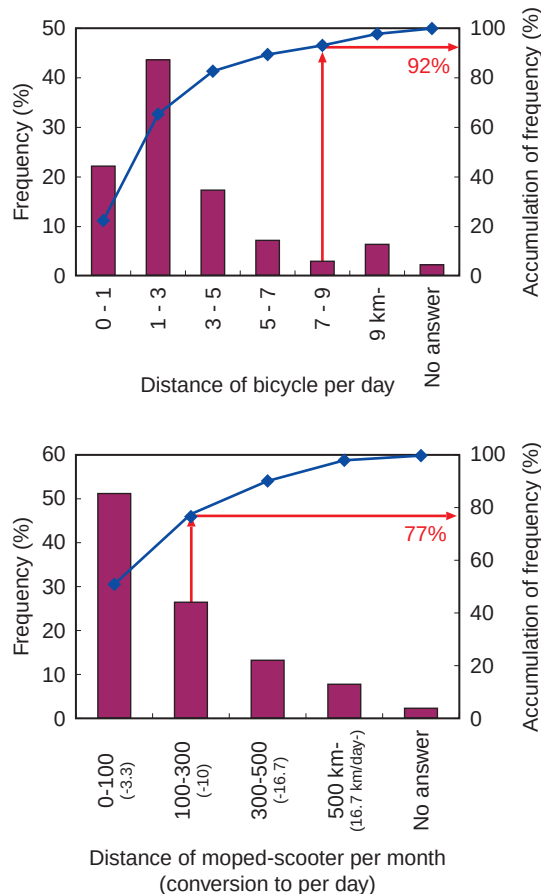


Fig. 6 Travel distance of bicycle and scooter users

The battery is a permanent type on the vehicle body. It is charged from a 100 V domestic power outlet. The fixed type battery has fewer electrical and mechanical components than those of a replaceable battery, thus contributing to the overall lightness and low price of the scooter.

3.3. Energy Consumed while Running

By making the body compact and light, running resistance is reduced and it is projected that the energy consumption of the scooter will be reduced. Energy consumption of the small EV scooter in an urban area running mode was compared with that of the CUV-ES and the Smart Dio Z4 (engine-driven scooter). The urban area running mode consisted of repeatedly starting and stopping every 300 m with the throttle fully open, and the maximum speed of 40 km/h regardless of the model.

Figure 8 shows energy consumption in terms of the energy extracted from the battery (or gasoline tank) per 1 km and also the energy consumption in terms of the energy drawn from the energy source which takes into account the energy generation process. The energy consumption of an electrically powered small scooter is 83 kJ (23 Wh)/km in terms of energy extracted from the battery and 172 kJ (48 Wh)/km as the energy generation converted value. From these results it was confirmed that if the scooter was restricted to short distance urban travel the energy efficiency in terms of energy drawn from the source was 1.6 times that of the CUV-ES, and approximately 4 times that of the Smart Dio Z4.

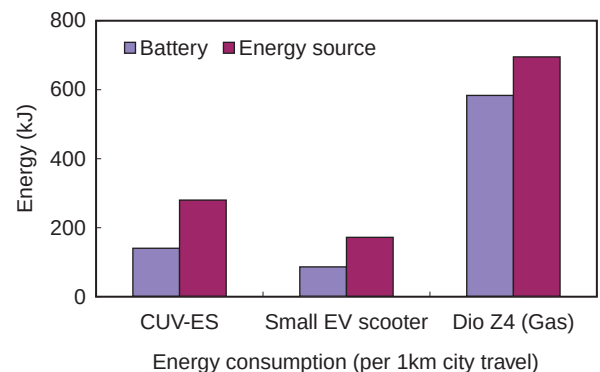


Fig. 8 Comparison of energy consumption of scooters

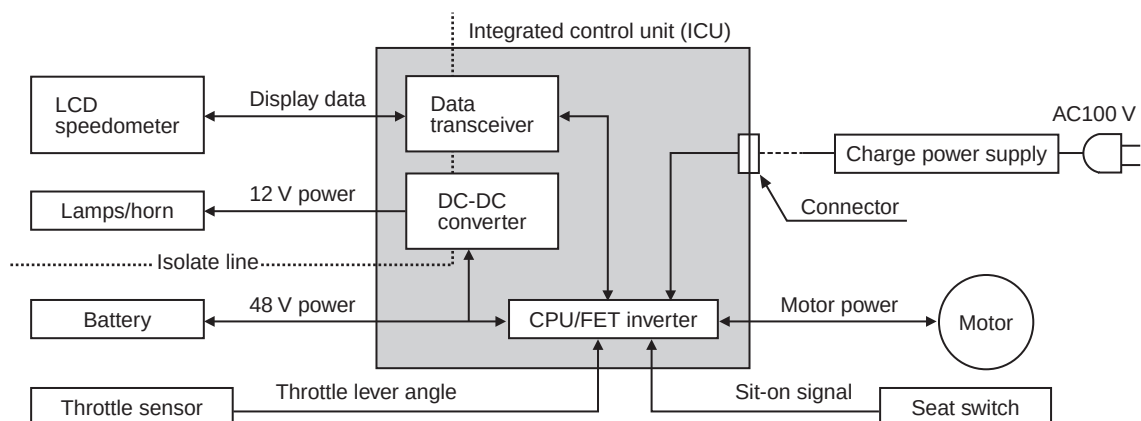


Fig. 7 Diagram of EV system

4. Major Technology

4.1. Electrically Powered Components

4.1.1. High energy density battery

To obtain the necessary driving performance and cruising distance while keeping the body compact, a high energy density Ni-MH battery was used. This battery is based on the one developed and mass-produced for the Racoon STEP COMPO (1999) power-assisted bicycle. The distribution and overall quantity of the active material was re-examined and both the mass- (weight-) energy density and volume-energy density were increased by 40% (Fig. 9). Generally, if a high energy density type battery is used the cyclic life tends to fall. By using an appropriate depth of charge for the full charge to minimize deterioration of the hydrogen storage alloy a cyclic life the same as that of a conventional battery could be achieved.

The battery used consisted of two packs each consisting of 20 cells of 1.2 V-7.5 Ah connected in series. The dimensions of each pack were $279 \times 80 \times 48$ mm with a mass of 2.9 kg. The total capacity of the two packs was 360 Wh (Fig. 10).

To reduce the time required to cool a battery heated as a result of use to a temperature low enough to charge it an intermediate material consisting of a silicone filler and rubber was inserted between the aluminum frame and the battery pack to enhance the thermal conductivity. As a result the time it takes for the battery temperature to fall from an initial value of 60 °C to 40 °C, a

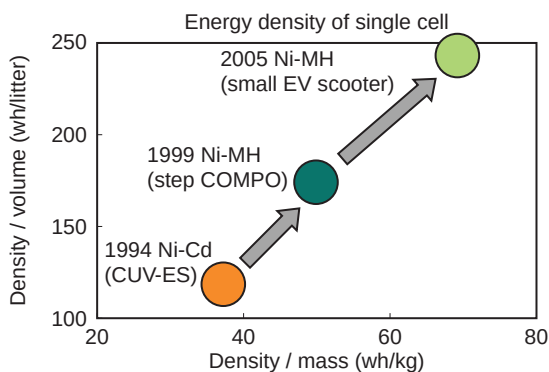


Fig. 9 Enhancement of battery performance

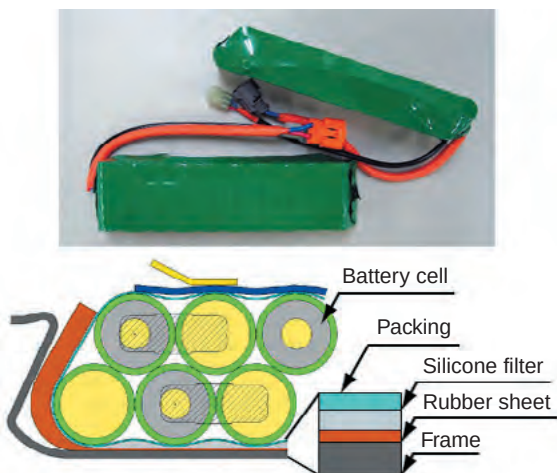


Fig. 10 Battery packs and cooling structure

temperature cool enough to allow the battery to be charged, was reduced from 95 minutes to 80 minutes.

The use of the high energy density battery as well as the use of the frame as a heat release structure contributed to body compactness.

4.1.2. Modular power unit

To reduce the size of the body the electrical and mechanical components of the driving system were modularized (Fig. 11). The engine, transmission, centrifugal clutch, and reduction gears of the scooter were integrated as a power unit inside the swing arm of the rear wheel which allowed the power train and the drive system to be laid out neatly. In this research the following configuration was used for increased compactness, lightness, and simplicity.

- (1) A high-torque low-speed outer rotor type motor was combined with a small 8-inch tire and the drive system between the motor and rear wheel was simplified consisting of a single gear reduction.
- (2) Current vector control was used to expand the range of rotational speed of the motor to the flux-weakening area while ensuring high driving efficiency thus eliminating the transmission and a clutch for starting.
- (3) The rear wheel, motor, and inverter were arranged as close as possible to each other and incorporated inside the swing arm enabling the overall length of the swing arm to be shortened. Also, the swing arm was made of cast aluminum allowing it to be used as a heat sink for the inverter.

As a result of the above a small power unit (overall length 356 × width 199 × height 175 mm) that was capable of efficiently converting the energy of the battery into drive power was developed for a small electrically powered scooter.

4.2. Frame and Parts Layout

To reduce the size of the body small diameter tires were selected and the wheelbase was shortened. Also, to obtain good handling and stability the layout of the parts was optimized.

With an EV it is possible to separate each component and also to select the parts layout relatively easily by adjusting the length of the wiring. This allows the benefit of adjustment of the weight distribution in the longitudinal direction.

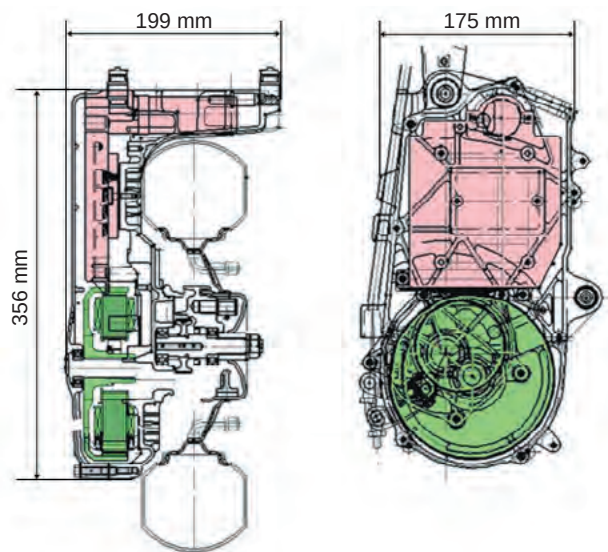


Fig. 11 Cross section of power unit

The weight of the overall vehicle was reduced with one of the two battery packs mounted on the down tube of the frame while the other one mounted on the step (Fig. 12). As a result the ratio of the front and rear weight distribution with a rider sitting on the seat was 34:66. Compared to an engine-driven scooter of the same class with a short wheelbase using 8-inch tires the weight was about the same and the distribution ratio was 3 points higher. As a result it was possible to obtain the same level of handling and stability as that of a scooter fitted with 10-inch tires.

4.3. User Convenience

4.3.1. Usability and riding feel

To make it easier for the rider to move the scooter or change direction while in a riding posture with both feet on the ground and to facilitate starting from a stable standing posture the seat height was made 700 mm with a floor width of 260 mm which is the minimum width over which both legs can be placed. As a result not only does the scooter handle well, but it also provides the sense of security for a first-time scooter user who has changed over from a bicycle. The body was designed in such a way that the rider can maintain the same riding posture as a conventional scooter with acceptable riding quality though overall size of the body was reduced (Fig. 13).

4.3.2. Lever type accelerator

Some first-time users of scooters complain it is difficult to use the twist grip type accelerator because the speed adjusting function of this type of accelerator is not separate from the steering function. A separate lever type accelerator was used in the new scooter enabling even first-time users to use it easily. A step was added to the operating load before and after a set operating angle of the lever and immediately before this step the motor output was set so that the scooter travels at a constant speed of about 8 km/h. The aim of this was to enable users who are not familiar with the speed adjusting function of a scooter to understand the relationship between the operating load, lever angle, speed of the scooter, and to become skillful at starting off at a moderate lower limit speed (Fig. 14).

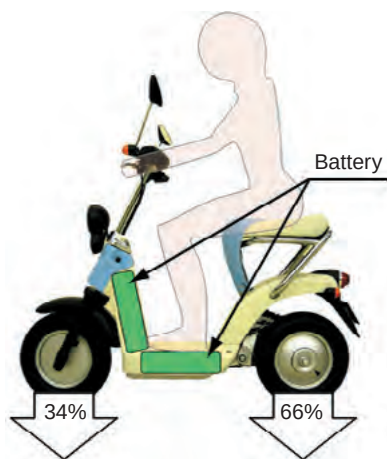


Fig. 12 Battery position and mass distribution

As a result of conducting test-rides on the in-house test course using people who possessed a license but mainly used a bicycle, it was found that 88.3% of the subjects gave positive opinions concerning the operability of the lever type accelerator. 6.5% of the subjects said that operation was easy because they could adjust the speed using one finger, thus indicating that the lever type accelerator was preferred over the twist grip type.

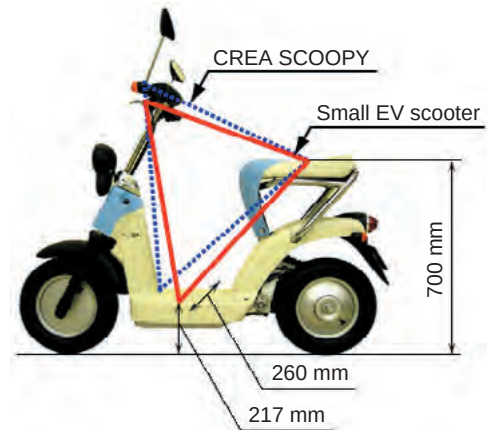


Fig. 13 Floor dimension and riding position

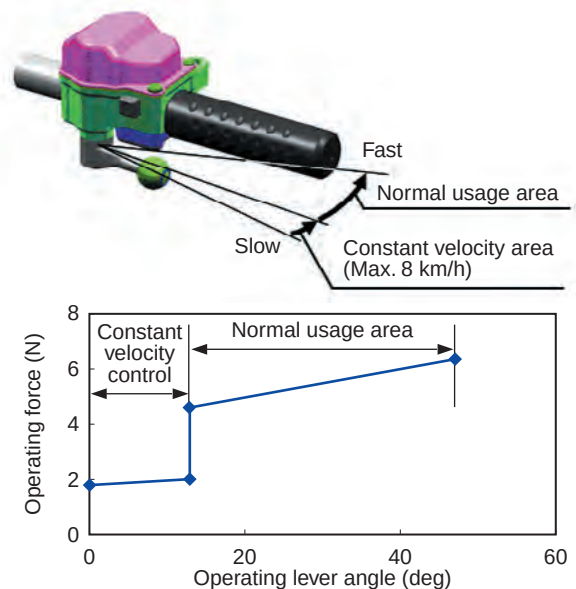


Fig. 14 Structure and characteristics of lever accelerator

5. Conclusion

The following results were obtained from the research into the electrically powered small scooters.

- (1) A new scooter was tailored to meet the needs of city commutings. A small lightweight body was combined with a small as practically possible battery resulting in higher energy efficiency compared to a conventional engine-driven scooter.
- (2) By optimizing the parts layout to maintain weight distribution on the front axle it was possible to obtain adequate handling stability despite the use of small diameter tires and a small lightweight body with a short wheelbase.
- (3) A lever type accelerator with a stepped operating load was used with a motor output characteristic that produced a constant speed of 8 km/h in the vicinity of the step, allowing first-time users of scooters to quickly acquire skill in using the accelerator.

References

- (1) Sendai Urban Research Forum: Jitensha Riyou ni Kannsuru Shimin Anketo Chousa Heisei 14 nendo ban, p. 20-21 (2003) (in Japanese)
- (2) Japan Automobile Manufacturers Association, Inc.: Nirinsha Shijou Doukou Chousa Heisei 15 nendo, p. 28 (2004) (in Japanese)

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Development of Monpal ML200 4-Wheel Scooter

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ABSTRACT

The Monpal ML200 4-wheel scooter has been developed based on the “Smart Package” concept.

Increased static and dynamic stability and greater user comfort were realized by adopting A-arm independent front and floating axle type rear suspensions and by reducing the center of gravity by locating the main parts lower down.

A metal base plate was adopted for the motor controller, while the use of a 3D map system identification speed control and a Honda-manufactured brushless motor with an integrated transmission improved efficiency and reduced drive resistance.

The scooter’s overall electrical consumption was improved by 9% over that of the current Monpal ML100 model, and a running distance of 34.3 km on a single battery charge was achieved.

Moreover, the arrangement of the lights, based on Honda ASV-3 LONG technology, has enhanced visibility.

1. Introduction

The Monpal ML100⁽¹⁾ was a pioneering 4-wheel scooter in 1999, when the market was dominated by 3-wheel models. The Monpal ML100 displayed excellent climbing performance, and was particularly well received by users in hilly and mountainous regions.

Honda’s new 4-wheel scooter, the Monpal ML200, has been developed to provide a more stable and comfortable ride, not just in hilly areas, but also in urban areas (the populations of which are expected to increase), in order to offer the joy of independent movement to even more customers.

Observation and customer surveys indicate that users of scooters in urban areas require the vehicles to be small enough not to intimidate pedestrians. They also need the vehicles to be easy to handle, so that they can, for example, be easily stored and maneuvered in and out of elevators.

Therefore, the product concept “Smart Package” was established, and the associated keywords “relaxation”, “comfort”, and “stylishness” were chosen.

2. Product Features

Figure 1 shows an external view and Fig. 2 shows a design sketch of the Monpal ML200; the scooter’s major specifications are shown in Table 1.

• Design

The design concept for the Monpal ML200 was “Quality Life – A relaxing, comfortable and stylish vehicle that increases the richness of daily life.” The design used has a low center of gravity and offers a sense of stability even with a slim and compact body.

* Power Products R&D Center

Table 1 Major specifications

Items	Specifications	
Model name	New Monpal ML200 (DX)	Monpal ML100 (DX)
Overall length (mm)	1,190	1,190
Overall width (mm)	595	680
Overall height (mm)	1,045	1,080
Tread (mm)	500	Front: 590/Rear: 530
Vehicle weight (kg)	112	94
Tire size	80/80-6	Front: 3.00-4 Rear: 70/100-8
Batteries	Valve Regulated Lead Acid 12V-35AH $\times$ 2	
Motor (30 min. rated output)	DC brushless motor (24V 430W) $\times$ 1	DC brush motor (24V 360W) $\times$ 1
Drive system	Direct 2-rear-wheel drive system (with differential)	
Brakes	Electromagnetic brake, regenerative brake, manual brake	
Max. speed (km/h)	6	6
Min. turning radius (mm)	1,430	1,470
Max. continuous driving range (Until stopping) (km)	34.3 (6km/h on level surface)	31.6 (6km/h on level surface)
Max. driver weight (kg)	100 (including baggage)	100 (including baggage)



Fig. 1 External view of Monpal ML200



Fig. 2 Design sketch

- (1) A front face with a welcoming look
 - (2) A front fender offering maneuverability
 - (3) A front panel with a U-shaped pillar design offering robustness, i.e. protection to the user, and a rear panel extending from the floor
- Convenient package
The width of the Monpal ML200 has been reduced by 85 mm to 595 mm, equivalent to the width of a bicycle. This new profile is highly convenient when driving the scooter, storing it when not in use, or maneuvering it in and out of an elevator. A front fender has also been added to make it easy for users to see which direction the wheels are pointing, resulting in excellent handling.
 - Comfortable ride
The use of front and rear suspensions has enabled the achievement of a balance between user comfort and stable handling.
 - Confidence-inspiring running performance
The use of a Honda brushless motor with integrated transmission and a motor controller featuring 3D map system identification speed control has enabled the achievement of a traveling distance of 34.3 km on a single battery charge and minimal reduction in vehicle speed when climbing hills.
 - Advanced drive system
Right and left driving levers, forward/reverse switching with automatic reset connected to the power source, large, clearly visible indicators, and a blinker switch with a canceller provide ease of use and a sense of security.
 - High visibility
Honda ASV-3 LONG technology is used on the body's side surface, and lights are arranged lower on the body, enhancing the user's sense of distance. The lights have also been increased in number and placed in a higher position, increasing visibility to vehicular traffic.
- This paper will discuss the body, power train and control technologies that have been used to develop the main features of the Monpal ML200: A smart package, user comfort and security, and driving performance offering peace of mind.

3. Body Technologies

3.1. Ensured Stability

The following points describe the differences between a scooter and a normal vehicle.

- The driver's weight is high relative to vehicle weight (approximately 100 kg), but ensured stability through shift of the driver's weight cannot be expected.
- There is a wide projected range of driver weight (40-100 kg).
- Vehicle dimensions are specified by the model standard, as follows: (Total length × total width × total height): 1200 mm × 700 mm × 1090 mm or shorter.
- Load shifting on inclines is high due to the comparatively high center of gravity.
- Considerations of ease of entry and exit, field of vision, and comfort place limits on the degree to which the sitting position can be lowered.
- Ease of use requires a turning radius of less than 1.5 m, and to achieve this, a turning angle of more than 55 deg. is required for the inner wheel.
- Drive and braking are dependent on the rear wheels, and it was therefore necessary to use a 30/70% front to rear weight ratio to ensure stable traction and braking force.

Given these restrictions, the development issues were to ensure stability while reducing the width of the Monpal ML200 against the Monpal ML100 and using a suspension system to increase user comfort.

3.1.1. Static stability

The reduction in static stability caused by the reduction in the width of the wheels was addressed by lowering the center of gravity of the vehicle.

Specifically, a low position layout of heavy accessories such as batteries, an increase in weight from using urethane-filled puncture-less tires, and addition of a suspension system and a swiveling seat mechanism lowered the vehicle's center of gravity by 22 mm against the Monpal ML100, resulting in an increase of approximately 5% in static turnover angle on inclines.

3.1.2. Dynamic stability

An important dynamic stability issue for the scooter was to help prevent lateral rollover when the user operates the steering handle while the vehicle is descending an incline.

The following three scenarios can be suggested as causing instability on an incline (Fig. 3):

- (1) The angle of the incline exceeds the scooter's lateral stability
- (2) The sum of the angle of the incline and the lateral G force (incline direction component) during a steady turn exceeds the scooter's lateral stability limit
- (3) The roll induced by the lateral G force during a turn causes the back inner wheel to leave the ground; the differential operation makes the sum of the lateral G force (incline direction component), increased by the acceleration due to the incline, and the angle of the incline exceed the scooter's lateral stability limit

It is difficult to imagine scenarios (1) and (2) occurring on a normal paved incline. However, scenario (3) is dependent on the characteristics of the vehicle, and in some cases scooters can become unstable under limited conditions in this scenario.

The following methods were used to control the scenarios described above:

- (1) Ground contact during a turn has been enhanced by imparting a degree of freedom to the rear suspension in the direction of roll (discussed below).

- (2) The timing of deceleration against the steering wheel angle was changed, and the ability of the deceleration control to follow rapid turning of the steering handle was increased.

As a result of these modifications and of lowering the scooter's center of gravity, as mentioned previously, the vehicle's dynamic stability has been enhanced despite the reduction in the width of the wheels.

3.2. Increased Comfort

There are numerous indented and raised sections such as stepped areas in sidewalks and curbs, joints and repairs in paving, manhole covers, pedestrian crosswalk paint, and steel plate surfaces used to prevent slipping. These cause impacts and vibrations to be transmitted to the seat and the steering handle. The Monpal ML200 has been fitted with front and rear suspensions to absorb these impacts and vibrations and increase user comfort.

3.2.1. Front suspension

The configuration of the front suspension is simple, with an A-arm and shock absorber used on each side. The left and right sides are independent. This configuration has significantly increased the camber, but changes in grip and potential uneven wear were judged, based on operating conditions (vehicle speed, lateral acceleration, etc.), to be within acceptable limits.

A multi-stage wheel rate was used to maximize sinking when the driver sits on the seat. This produces a more comfortable ride and enhances stability by ensuring ground contact when the load is reduced on steep hills despite the fact that a low load is distributed to the front of the vehicle.

The configuration helps control instability, such as when the vehicle is on an incline or affected by the push-under that occurs when the steering angle is altered to start the vehicle on a low μ surface, resulting in a level of handling that enables even the elderly to operate the vehicle with confidence.

In addition, the caster angle and kingpin angle were set to 2° and 7° respectively to reduce steering force (Monpal ML100 - caster angle: 5°, kingpin angle 10°). This minimizes the restorative force to allow for extended use by elderly users with limited muscular power.

Because the scooter basically stops when the user takes his or her hands off the handle, excessively strong neutral restorative force is not necessary. Therefore, the setting prioritizes comfort of the steering handle operation.

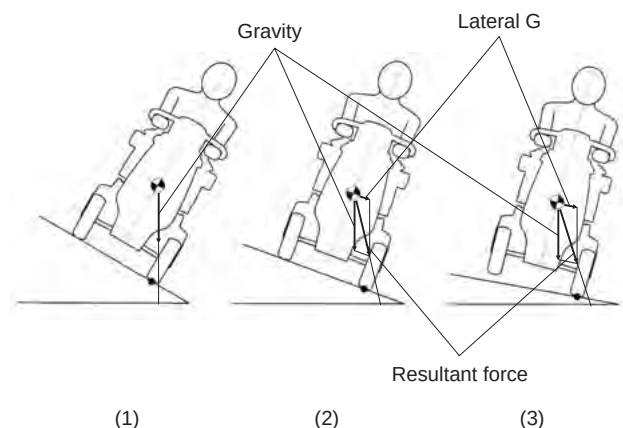


Fig. 3 Unstable vehicle model

Figure 4 shows the configuration of the front suspension, and Fig. 5 shows the relationship between front tire load and movement.

3.2.2. Rear suspension

The rear suspension generally used in scooters possesses a one degree of freedom link; this allows only an up-down motion supported by the left and right trailing arms of the motor/transaxle unit.

The rear suspension used in the Monpal ML200 is a floating axle type that possesses a two degrees of freedom link, allowing motion in the direction of roll using a pivot at one point on the frame of the trailing arm and rubber bushings at two points on the transaxle coupling. Studies of stability, comfort, and strength indicate that a lateral rod is not required.

Roll stiffness was controlled by adjusting the hardness of the three bushings used on each side of the vehicle and the lateral mounting span of the shock absorber. The bushing on the frame

has been recessed to ensure that it flexes easily in the direction of roll, which increases road holding when one wheel comes off the pavement and runs on a raised section, or when roll occurs. It also increases comfort.

Figure 6 shows the configuration of the rear suspension, and Fig. 7 shows the frame bushing cross section.

3.2.3. Tires

Urethane foam-filled puncture-less tires have been specially designed in place of the previous pneumatic tires to prevent anxiety over getting a puncture while out on a journey. Factors such as the shape of the tread and the hardness of the urethane (necessary for balancing comfort and running resistance), and the patterns required to achieve stable traction and braking force with changes in the pavement surface were decided on the basis of a variety of tests.

Figure 8 shows the tire structure and pattern, and Fig. 9 shows the contribution made to riding comfort by tire and suspension.

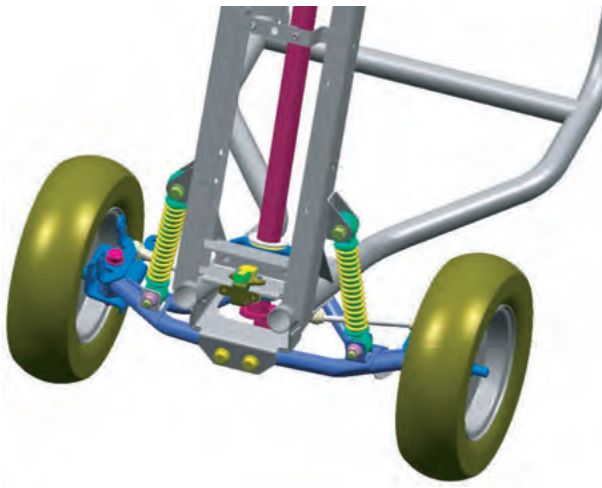


Fig. 4 Front suspension

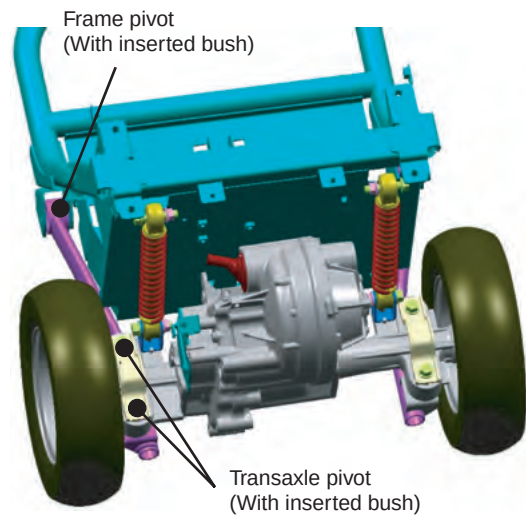


Fig. 6 Rear suspension

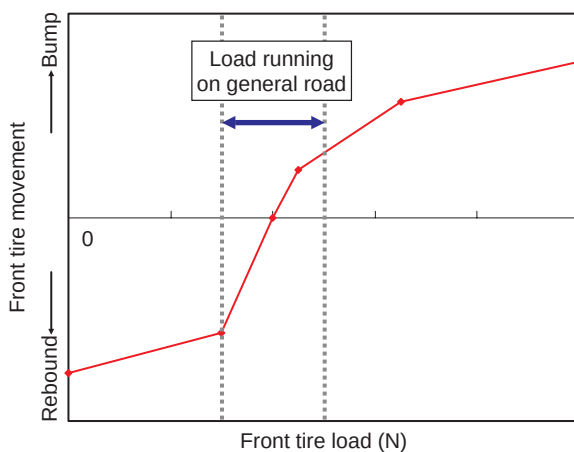


Fig. 5 Relationship between load and movement

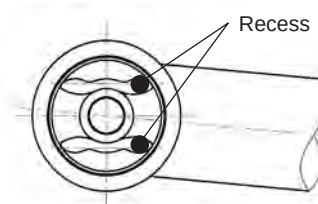


Fig. 7 Frame bush cross section

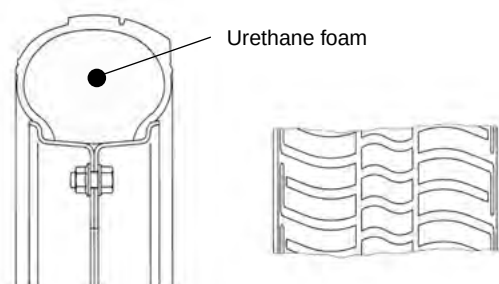


Fig. 8 Tire structure and pattern

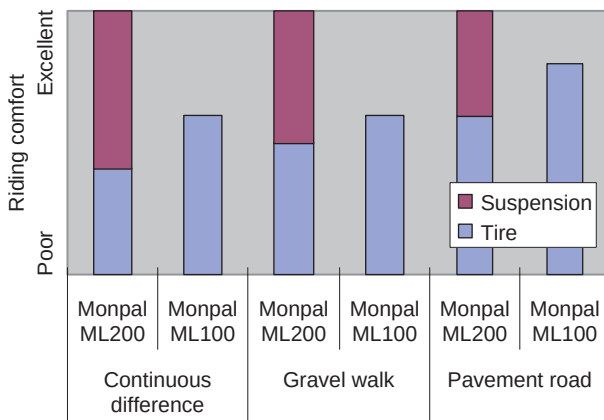


Fig. 9 Contribution to riding comfort

4. Power Train Technology

4.1. Power Train Configuration

In the power train, an original brushless motor is used for drive, and integrated with the gear case. This configuration reduces the number of parts and prevents gear noise from being generated by motor/gear case mounting tolerance.

Electromagnetic brakes are mounted on the motor shaft. The system detects the ON/OFF state of the power switch and driving lever and the speed of the shaft, and controls operation of the electromagnetic brakes, enabling the brakes to be applied while the vehicle is stationary. This allows the brakes to operate automatically, even when the scooter is stationary on an incline, ensuring that it remains immobile.

The gear case has three shafts; the clutch and the manual brake are mounted on the second counter-shaft.

A shifter enables the spline that runs between the counter-shaft and counter-driven gear to be disconnected to allow the shaft to move freely when the scooter is pushed manually, when the battery loses charge or during a malfunction.

The manual brake is operated by a brake lever provided with a lock mechanism. The brake is a standard internal expansion drum brake, and is used when the scooter is being pushed manually or when it is parked.

Figure 10 shows the structure of the power train.

4.2.2. Semi-closed slot stator

As methods to achieve smooth output RPM in a drive motor, a skewed pole configuration or a closed-slot stator are commonly used. However, both reduce magnetic flux, leading to a reduced power density for the same size of motor.

The new motor uses a stator with a semi-closed slot configuration. In addition, by optimizing the width of the magnet and the depth to which it is inserted in the IPM rotor, smooth revolution and high output are both achieved.

Figure 11 shows the configuration of the semi-closed slot stator.

4.2.3. Thicker wiring and internal bus ring configuration enabled by use of split-core stator

The use of a split-core stator enables bobbin winding of the coils. In addition, the use of self-fusing wires for windings enables the usual varnishing process to be eliminated.

When using a split-core stator, it is necessary to connect the concentrated windings of the coils at the core. In the new motor, the ends of the coils are positioned at the inner circumference of the stator. For this reason, the connections between the coils and the bus ring protrude into the inner circumference of the stator. This enables the electrode structures to be simplified and reduced in size, resulting in a reduction in the length of the wires and reduced electrical resistance.

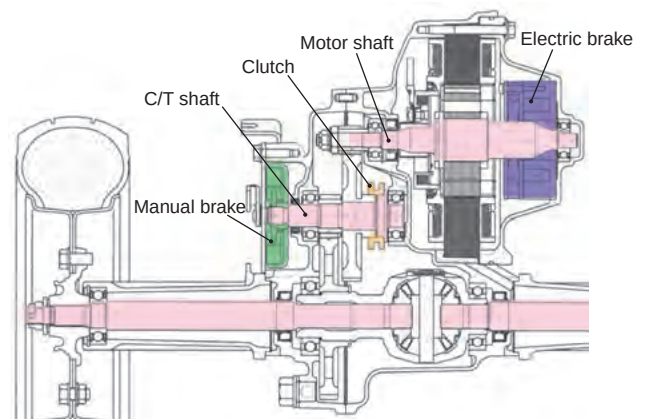


Fig. 10 Structure of power train

4.2. Integrated Motor/Transmission

4.2.1. IPM rotor

The Monpal ML100 used a Surface Permanent Magnet (SPM) rotor with a magnet attached to the core. Armature reaction in these rotors caused demagnetization at high temperatures and high loads.

The new scooter uses an Interior Permanent Magnet (IPM) type rotor that uses a flat plate magnet. This allows reluctance torque to be used at low speeds and field weakening control to be applied at high speeds. As a result, the weight of the magnet has been reduced, while the range of operation has been increased.

This IPM rotor was first used by Honda in the hybrid snow thrower HSS1170i, and is manufactured by the company's Kumamoto Plant⁽²⁾.

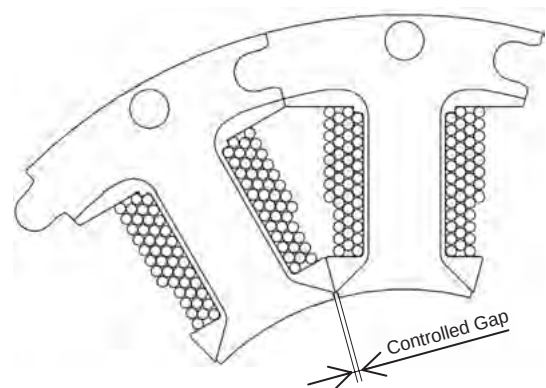


Fig. 11 Semi closed stator

4.3. Increased Drive System Efficiency

The traveling distance of a scooter on a single battery charge is an important performance indicator. The efficiency of the drive system used in the Monpal ML200 has been enhanced to achieve the development target of an increase of at least 5% in traveling distance against the Monpal ML100 (running on the flat road at 6 km/h).

In addition to using a high-efficiency drive system (to be discussed below), other measures used to increase efficiency include the optimization of motor speed and the use of low-friction sealed bearings in the motor to reduce electricity consumption, resulting in the achievement of a 0.2 A reduction in consumption against the Monpal ML100.

0W-20 oil, which is now being used in fuel-efficient engines, was adopted as the transmission oil, and agitation resistance equivalent to 0.1 A of current consumption was reduced.

Application of voltage duty control (75%, 500 Hz) to the electromagnetic brakes after they are released has reduced the effective voltage, resulting in a 0.1 A reduction in expended current.

As a result of these measures, the electrical consumption of the vehicle has been improved by 9% over that of the current Monpal ML100 model, and the running distance on a single battery charge of 34.3 km (DX type actual value) was achieved.

Figure 12 shows a comparison of current consumption in the Monpal ML100 and the Monpal ML200.

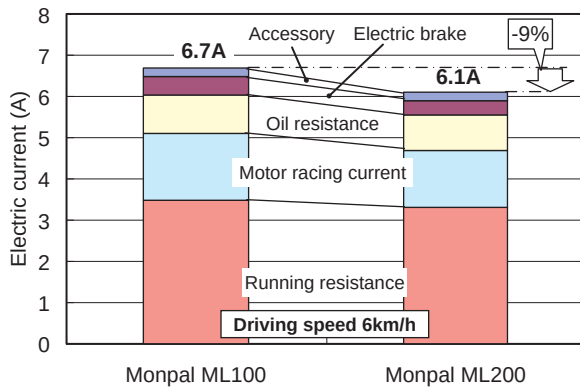


Fig. 12 Comparison of current

5. Control Technologies

5.1. High-efficiency Drive System

The use of a metal base plate for better heat radiation and 3D map system identification speed control technology for an enhanced power characteristic has resulted in the achievement of increased efficiency in the motor controller and has also enabled the unit to be reduced in size.

Figure 13 shows a comparison between the new brushless motor used in the Monpal ML200 and the brush motor used in the Monpal ML100.

5.1.1. Motor controller with metal base board

Reducing the size and weight of the motor controller while increasing power necessitated increased heat radiation performance for the drive section which produces the most heat in the unit. A

5 mm thick aluminum base board was therefore used in the motor controller.

In addition, the motor controller was coupled by bolts to a 1.2 mm thick steel frame located underneath the seat, with a heat radiation sheet sandwiched between.

Figure 14 shows the layout of the motor controller in the vehicle.

This configuration has enabled the heat generated by the motor drive section to be dispersed by the aluminum plate and radiated directly from the steel frame, enabling the previously used heat sink to be eliminated.

As a result, it was possible to increase the power of the motor controller while reducing its size and weight, and the efficiency of the unit was increased by 5% against the Monpal ML100.

Figure 15 shows a comparison of the motor drive systems used in the Monpal ML100 and the Monpal ML200 at a vehicle speed of 6 km/h.

5.1.2. Three-dimensional map system identification speed control

The optimization of power distribution patterns in response to motor speed and torque was essential to maximizing the potential of the IPM rotor/DC brushless motor.

The new system estimates output torque based on the difference between the RPM that are demanded by external input and actual RPM. It then calculates the duty values, lead angle and turn-on angle required to achieve the estimated torque using a 3D ROM map. These calculations are conducted in real time to enable stable speed control to be applied.

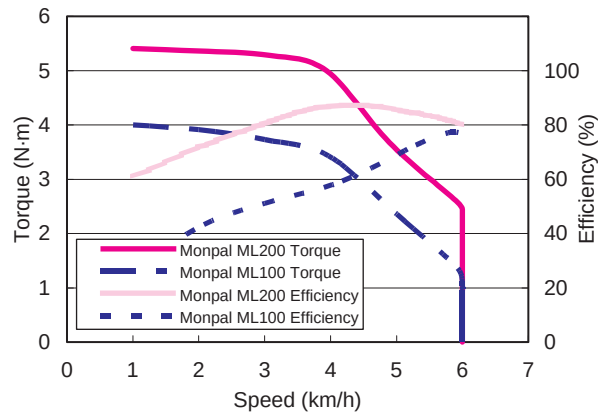


Fig. 13 Comparison of power & efficiency

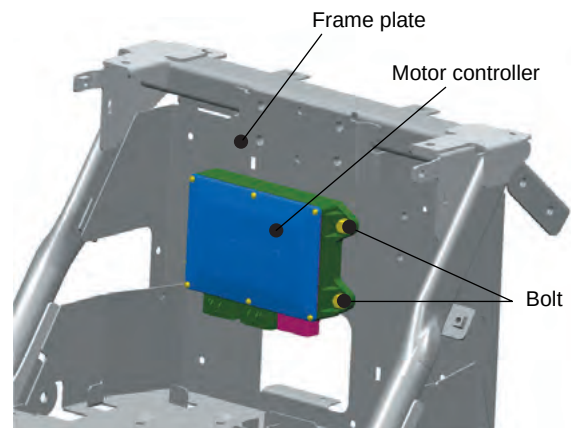


Fig. 14 Layout of motor controller

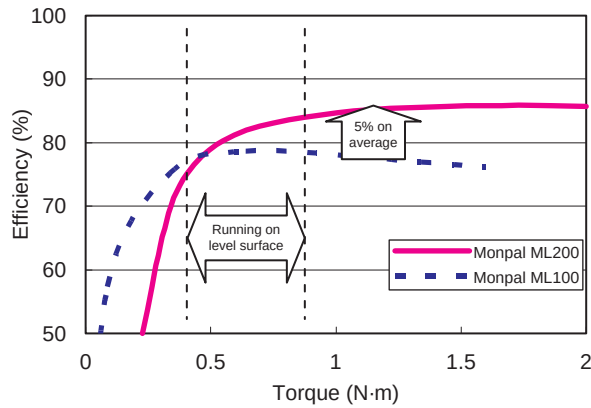


Fig. 15 Comparison of motor efficiency

Because the motor in the new system uses an IPM rotor and concentrated windings in the stator, an electromagnetic field model of optimal speed control would be extremely complex. For this reason, the 3D ROM map of duty value, lead angle and turn-on angle mentioned above are determined on the basis of calculations that are conducted in advance as well as from actual measurements.

Figure 16 shows representative characteristics of a 3D map of DUTY.

5.2. Other Control Technologies

5.2.1. Turning deceleration control

Generally, the scooter is provided with turning deceleration control that limits the maximum speed of the vehicle in response to the driver's steering handle input. This enhances drive feel and vehicle body stability during turns.

In the Monpal ML100, the handle input was detected according to ON/OFF states of a turning switch positioned radially on the steering shaft. To enhance detection of the degree of input to the steering shaft, steering input is detected in the Monpal ML200 using a potentiometer driven by the steering shaft via gears. Because

there is a linear relationship between the handle input from the driver and the potentiometer output value, a linear deceleration control according to the value enables smoother turning deceleration.

Figure 17 shows the relationship between setup speed and steering rate inside the motor controller. If the vehicle speed is higher than 4 km/h, it will be reduced to 4 km/h when the handles are operated. Equation (1) shows the relationship between these parameters.

$$Ncd = Ncm \times Rst$$

$$Nset = Niset - Ncd \quad (1)$$

Ncd: Degree of reduction of maximum speed

Ncm: Vehicle speed prior to turning deceleration

Rst: Turning ratio based on steering potentiometer reading

Niset: Setup speed based on operation of the accelerator by the user

Nset: Actual setup speed

5.2.2. CAN communication

The Monpal ML200 uses two ECUs with different functions, in the control panel and the motor drive section. Controller Area Network (CAN) communication is used to transmit information between these control units.

The vehicle uses a drive-by-wire (DBW) system in which the control panel ECU receives signals from the driving lever, the front-rear switch, the volume for maximum speed settings, and the steering sensor, and transmits this information via CAN communication to the motor drive ECU, which uses them to apply optimum motor drive control.

The use of CAN communication provides the following benefits:

- (1) The drive section, which radiates a significant amount of heat, can be concentrated in one section.
- (2) The harnesses used for operating controls can be significantly shortened.
- (3) The harnesses used for running control can be reduced by about 50% in number compared to the current Monpal ML100.
- (4) Advanced malfunction diagnostics can be applied because the ECUs are separated by function.

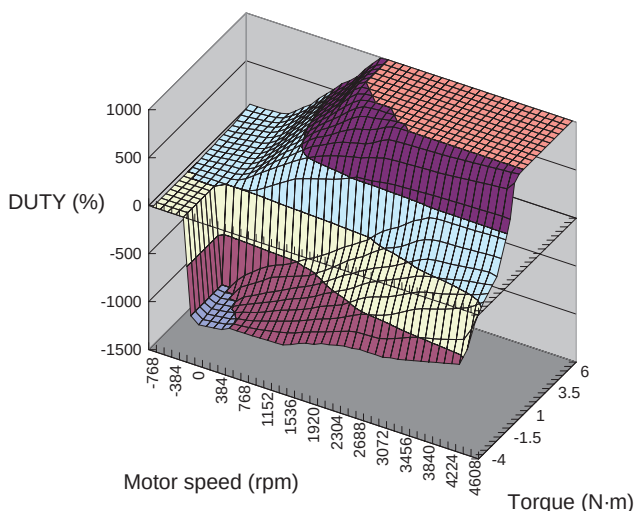


Fig. 16 Three-dimensional map of DUTY

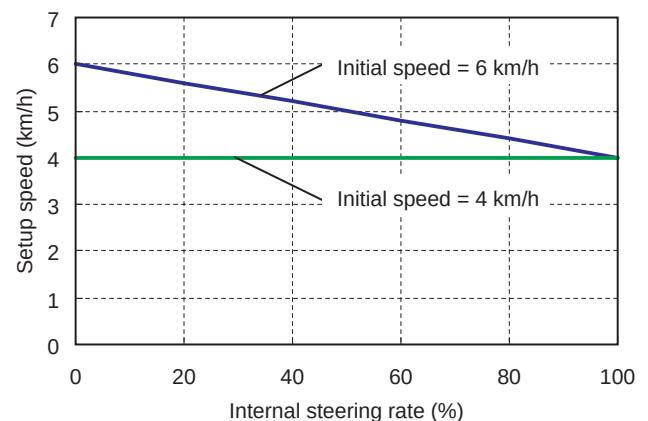


Fig. 17 Relationship between setup speed and internal steering rate

6. Conclusion

Using new suspension configurations and lowering the vehicle's center of gravity have increased steering stability and user comfort. The achievement of increased efficiency through the use of 3D map system identification speed control and a Honda brushless motor with an integrated transmission has increased the traveling distance on a single battery charge.

These body, power train, and control technologies have allowed realization of the "smart package" development concept, with the keywords "relaxation," "comfort" and "stylishness."

As a result, the Monpal ML200 4-wheel scooter is easier to use than the current Monpal ML100, and will undoubtedly be used with satisfaction by numerous customers.

References

- (1) Hosaka, J., Inui, T., Maesaka, K.: Development of a 4-wheel Electric Wheelchair, Monpal ML100, HONDA R&D Technical Review, No. 12, No. 1, p. 21-26
- (2) Wakitani, T., Yamagishi, Y., Kawakami, T., Mizoroke, T.: Development of Electronic Control System for Small Hybrid Snow Thrower, Honda R&D Technical Review, No. 16, No. 1, p. 63-70

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Development of Gasoline Turbo Engine Conforming to U.S. ULEV2 Regulations

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ABSTRACT

Honda has developed a 2.3L 4-cylinder gasoline turbo engine that balances high power output with high fuel economy and low emissions. The new engine adds a newly developed variable-capacity turbocharger to Honda's proprietary i-VTEC engine technology. This addition resulted in a peak power output of 179 kW with the low-speed torque characteristics of a 3.5L normal aspirated engine due in part to a high boost-pressure response. Rapid activation of the exhaust catalyst is achieved by positioning it directly underneath the turbocharger and by reducing the heat capacity of the exhaust system via an exhaust manifold constructed out of aluminum alloy. This, coupled with a highly optimized catalyst metal loading has enabled this powertrain to meet the requirements of U.S. ULEV 2 regulations. This engine will be used in a new model SUV for the U.S. market. The vehicle will display a dynamic performance equal to or better than that of SUVs fitted with 3.0-3.5L normal aspirated engines, while at the same time achieving top-in-class fuel economy.

1. Introduction

At the same time that environmental issues are being addressed on a global scale, the price of gasoline has risen markedly. Given these circumstances, the need for enhanced automotive fuel economy has grown, from the perspective of not only reducing tailpipe emissions, but also of CO₂ reduction. Several different approaches exist when considering automotive fuel economy enhancement technologies, but one approach is to reduce engine displacement without sacrificing power by means of forced induction. Forced induction via the use of a turbocharger leads to lower friction loss than with mechanical supercharging, and is appropriate for enhancing fuel economy through this method of reduced displacement. However, response and torque at low engine speeds, when there is little exhaust energy, are not as good as in a

large displacement, normal aspirated engine, and it was thought that applying supercharging to heavy vehicles would lead to reduced drivability. To address this, Honda's new engine, uses a Variable Flow Turbocharger (VFT), which enhances low-speed torque and response. Combining this VFT with an automatic transmission optimized for use with a turbo engine provides performance and drivability similar to that of other vehicles with large engine displacements. A turbocharger, however increases the heat capacity of the exhaust system, which inhibits exhaust catalyst warm-up, and thus greatly affects exhaust emission reduction performance at start-up. This engine conforms to U.S. ULEV2 standards through the use of Honda's original i-VTEC mechanism, reducing the heat capacity of the exhaust system, and optimizing exhaust catalyst performance. This paper describes an outline of the newly developed gasoline turbo engine, along with its major technologies.

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Table 1 Engine specifications

Engine code		K23A	K24A
Cylinder configuration		In-line 4-cylinder	In-line 4-cylinder
Bore × stroke (mm)		86 × 99	87 × 99
Displacement (cm ³)		2300	2354
Compression ratio		8.8 : 1	9.7 : 1
Valve train		DOHC i-VTEC	DOHC i-VTEC
Number of valves		4 per cylinder	4 per cylinder
Valve diameter (mm)	Inlet	35	35
	Exhaust	30	30
Turbo charger		Variable flow turbocharger	–
Impeller external diameter (mm)	Compressor	56	–
	Turbine	32	–
Exhaust manifold		Water-cooled aluminum alloy	Stainless steel pipe
Catalyst [vol.(L)]		Close coupled [1.4] + underfloor [1.0]	Underfloor [1.7]
Recommended gas (octane rating)		Unleaded premium (RON95)	Unleaded regular (RON91)
Max. power (kW/rpm)		179 / 6000	125 / 5800
Max. torque (Nm/rpm)		353 / 4500	220 / 4000

2. Development Goals

The following targets were set to develop an engine for use in a compact SUV for the U.S. market.

- (1) Achieve best-in-class fuel economy with acceleration performance equivalent to SUVs equipped with normal aspirated 3.0–3.5 L engines.
- (2) Achieve a smooth and responsive starting acceleration performance with no feeling of stagnation.
- (3) Achieve the best peak power among vehicles with turbo engines that conform to U.S. ULEV2 standards, and peak torque on par with normal aspirated 3.5 L engines.

3. Overview of the Engine and Major Specifications

3.1. Overview of Engine

Table 1 compares the major specifications of this engine with those of the base 2.4 L normal aspirated ACCORD (K24A) engine, and Fig. 1 shows the external view of the newly developed engine.

Based on the framework of the K24A 4-cylinder i-VTEC engine⁽²⁾ used in the ACCORD, various measures were taken, including strengthening the reciprocity system that accompanies the use of a turbocharger. Engine displacement was changed to 2.3 L by reducing the K24A cylinder bore by 1 mm to 86 mm. The valve-train system is the i-VTEC used in the K24A. The i-VTEC mechanism used in this engine combines a VTEC (variable valve timing & lift) mechanism with VTC (camshaft phase continuation variable control) on the intake side camshaft. A VTEC mechanism is used for the intake valves, switching between a combustion-stabilizing 1-valve pause and a high charge-efficiency 2-valve drive, according to the driving range. The combination of 1-valve pause and high swirl-intake ports is effective in stabilizing combustion during ignition timing delay immediately after start-up. This allows for ignition retard immediately after starting, which in turn leads to rapid catalyst warm-up. The VTC incorporated in the intake camshaft contributes to a reduction in feed gas NO_x through the

introduction of an internal EGR, while also contributing to enhanced fuel economy by reducing pumping losses. Moreover, a water-cooled aluminum alloy exhaust manifold, developed for this engine, minimizes any increase in the heat capacity of the exhaust system due to the installation of the turbocharger, and achieves rapid activation of the catalyst during start-up. The conventional cylinder head with integrated exhaust manifold⁽⁴⁾ that is used in many Honda engines has an integrated structure. In this set-up, all of the exhaust ports, from each cylinder to the collector, are completely housed inside the cylinder head. With the turbo engine, the exhaust manifold portion is separate, which enlarges the water jacket surface area. This enhances water cooling performance around the exhaust ports, which exhibit high exhaust pressure and high temperatures due to the installation of the turbocharger. It also enhances the cooling performance of the surfaces between the turbocharger and exhaust manifold. Figure 2 shows the structure of the exhaust manifold water jacket.

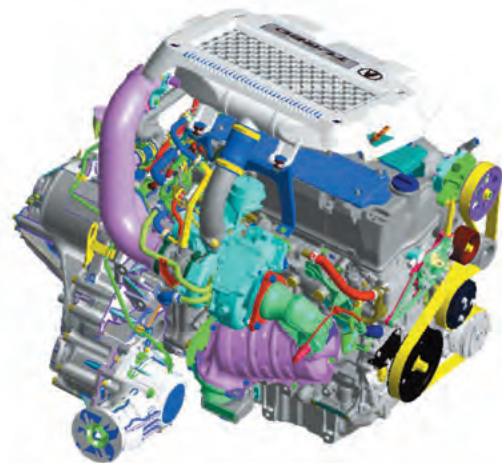


Fig. 1 K23A DOHC i-VTEC turbo engine

3.2. Overview of the Turbocharger

Figure 3 is an overview of the VFT used by this engine. The VFT is a redesigned variable capacity turbocharger⁽¹⁾. The turbine has two scrolls – inner and outer circumferences – separated by the fixed vane. A flow control valve, located at the turbine inlet port, directs exhaust gas to the inner or outer scrolls. This allows the turbine capacity to be changed arbitrarily, providing both the characteristics of a small-capacity turbine, emphasizing low-speed torque, and of a large-capacity turbine, emphasizing high power. The variable capacity turbines used in diesel engines are generally Variable Nozzle Turbochargers (VNT), but these turbochargers have a complex mechanism in the variable nozzle part, and have many issues concerning heat resistance. It is difficult to adopt this technology to high-power gasoline engines because of their high temperature exhaust gases. Compared to VNT, the VFT has fewer moving parts. The sliding part of the flow control valve, which is the VFT's only moving part, is located where it is not in direct contact with high-temperature exhaust gases, so the VFT can be used in high-power gasoline engines. The flow control valve and waste gate are driven by boost pressure, via their own separate positive pressure actuators. In addition, the actuators' applied pressure is electronically controlled by solenoid valves.

4. Supercharger System and Power Performance

4.1. Benefits of VFT Use

Turbo engines can recover energy in exhaust gasses, making them more efficient than engines with mechanical superchargers. However, some of the issues affecting turbo engines include insufficient torque at low engine speeds, when there is little exhaust energy, and significant delayed response times when shifting from low-load to high-load ranges. The newly developed engine addresses these issues by using the VFT. Figure 4 compares the W.O.T. torque characteristics of conventional turbochargers to those of the VFT. The VFT provides a wider range of torque characteristics, including both the low-speed range characteristics

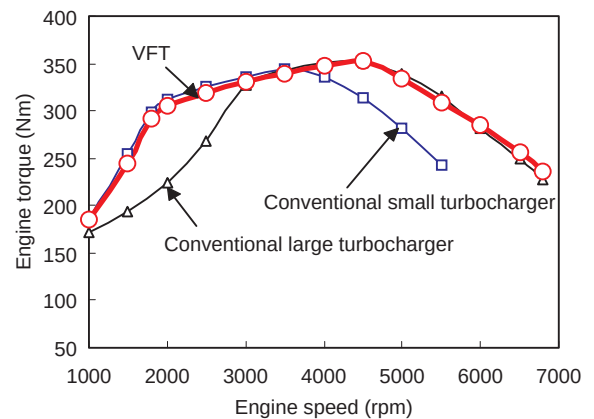


Fig. 4 Effect of VFT on engine torque

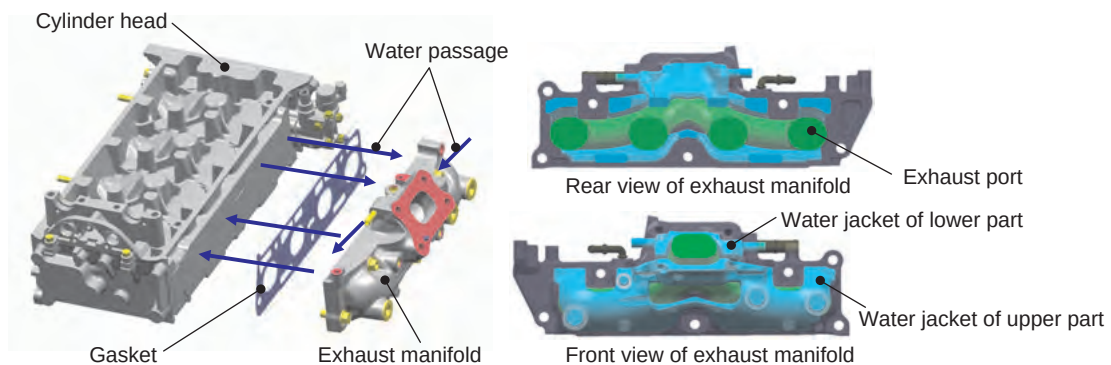


Fig. 2 View of water-cooled aluminum alloy exhaust manifold

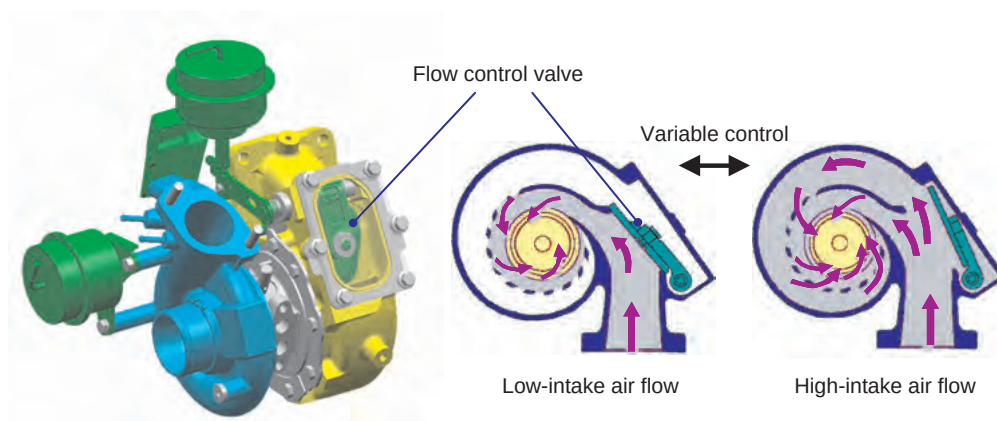


Fig. 3 View of VFT

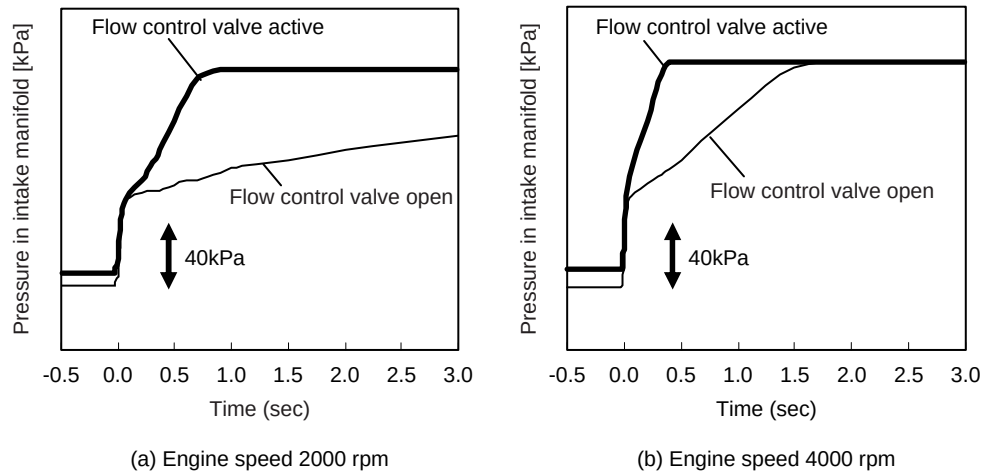


Fig. 5 Effect of flow control valve on boost pressure response

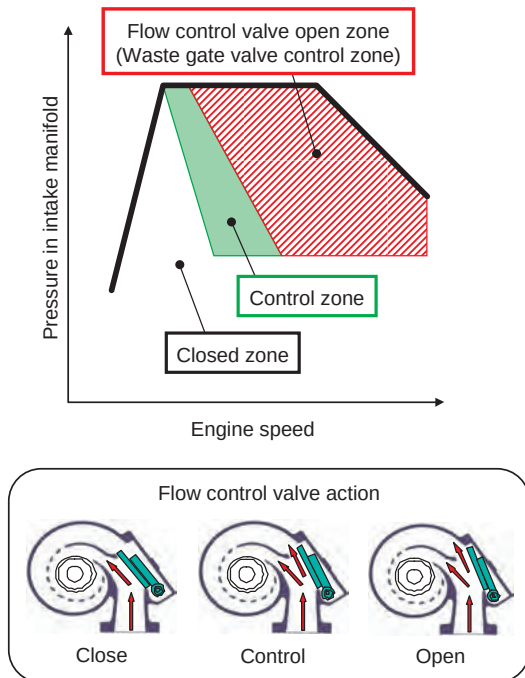


Fig. 6 VFT control zone

of a small turbocharger and the high-speed range characteristics of a large turbocharger.

As an evaluation of boost pressure responsiveness, Fig. 5 shows boost pressure characteristics when the throttle changes from closed to wide open at fixed engine speeds.

With the VFT flow control valve wide open (no flow control), turbo characteristics are equivalent to those of a conventional large turbocharger. At 2000 rpm (low speed), the target boost pressure cannot be achieved without implementing flow control. However, when flow control is applied, it is possible to achieve the target boost pressure within 1.0 sec. [Fig. 5(a)]. At 4000 rpm, the time it takes to reach the target boost pressure drops from 1.7 sec. to 0.4 sec. with flow control applied. This is less than 1/4th the time required when the flow control valve is not used [Fig. 5(b)].

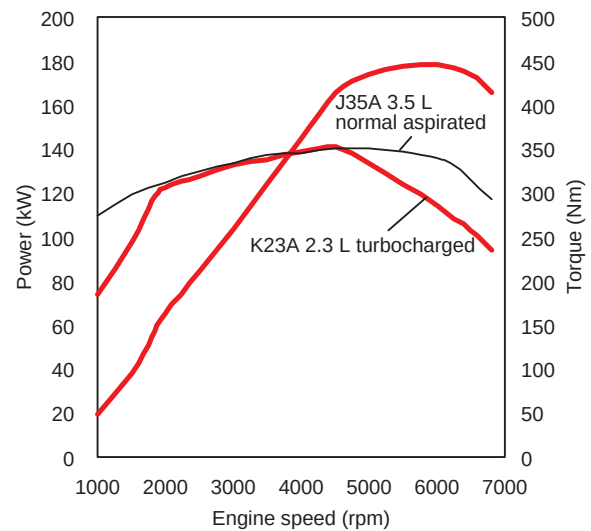


Fig. 7 Engine performance

4.2. Boost Pressure Control

Boost pressure is controlled, according to driving conditions, by the flow control valve and the waste gate valve. These valves are driven by two independent actuators. The flow control valve opening is basically set according to intake airflow. The boost pressure is controlled by the flow control valve in the low intake airflow range, and by the waste gate valve in the high intake airflow range. Figure 6 is a schematic of the boost pressure control ranges.

4.3. Power Performance

While reducing intake and exhaust resistance, this engine incorporates a charge air cooling effect thanks to a large capacity low pressure loss intercooler. All its technologies combined achieves a maximum power of 179 kW and a maximum torque of 353 Nm. Figure 7 shows this engine's W.O.T. power characteristics, and for comparison, the W.O.T. power characteristics of the J35A (3.5 L V6) used in the ACURA RL. In the normal operating range of 2000–4500 rpm, this engine achieves torque on a level equal to that of a normal aspirated 3.5 L engine.

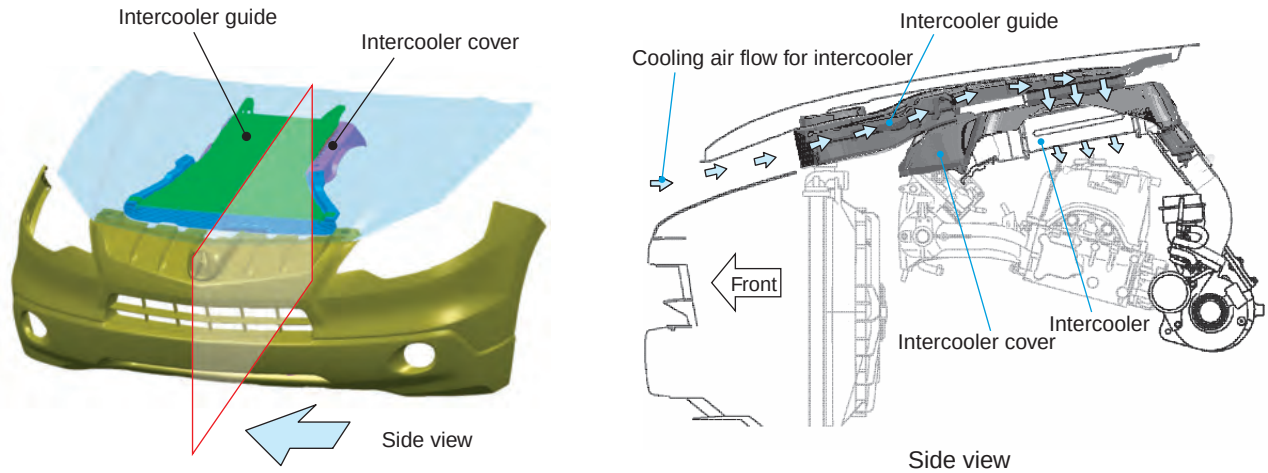


Fig. 8 View of intercooler guide

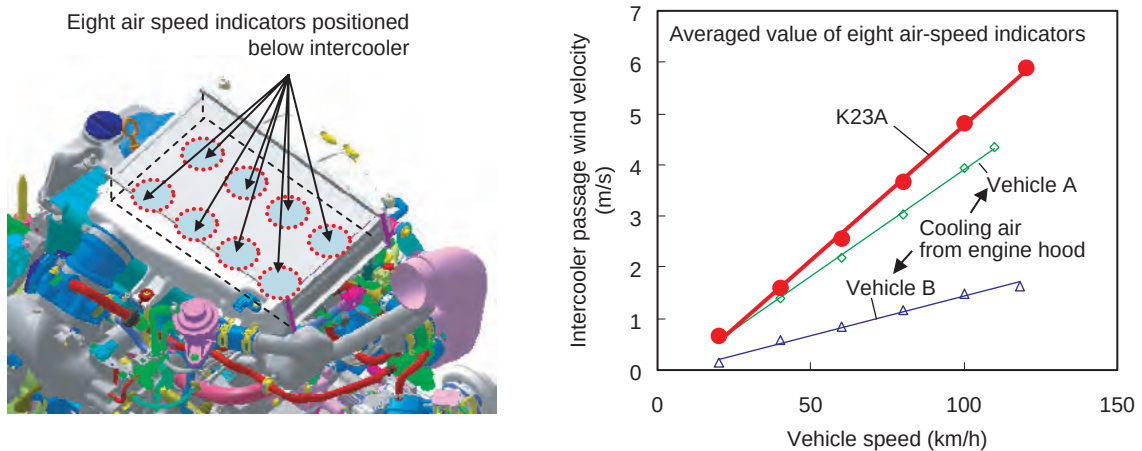


Fig. 9 Intercooler passage wind velocity

The intercooler is positioned at the top of the engine, and cooling air taken in from the front of the vehicle is directed by the intercooler guide and cover, as shown in Fig. 8. An outline of the intercooler guide is also shown in the figure.

Eight air speed indicators positioned directly below the intercooler constantly evaluate intercooler efficiency with respect to vehicle speed. Since cooling air is taken in at the front of the vehicle, an intake airspeed of 4.8 m/s is ensured at a vehicle speed of 100 km/h. This results in better cooling performance compared with conventional set-ups where the cooling air is taken in from the engine hood, as shown in Fig. 9.

5. Engine Unit Fuel Economy

Typically, forced induction engines must have a low compression ratio to avoid knocking, which in turn can cause lower fuel economy than in normal aspirated engines. To minimize the drop in fuel economy and make downsizing through forced induction effective, it is very important to address the issue of knock. In this engine, knocking was avoided by modifying the water jacket shape around the combustion chamber. The use of small diameter long-reach spark plugs also helped prevent a drop in fuel economy. With any forced induction engine, torque is directly proportional to boost pressure. In addition, high boost pressure calls for a lower

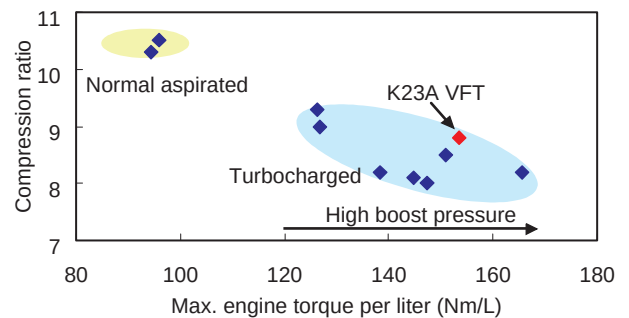


Fig. 10 Positioning of compression ratio

compression ratio to help reduce knocking. However, as shown in Fig. 10, torque per unit of displacement of this engine is enhanced by high boost pressure, while at the same time employing a higher compression ratio than that of many other charged engines.

As shown in Fig. 11, in addition to the higher compression ratio and reduction in pumping loss achieved with the internal EGR using the VTC, BSFC was reduced roughly 12% compared to that of a normal aspirated 3.5 L engine with an equivalent maximum torque.

Additionally, when looking at the relationship of BSFC to normal aspirated engine displacement, as shown in Fig. 12, this engine is positioned as equivalent to a normal aspirated 2.5 L engine.

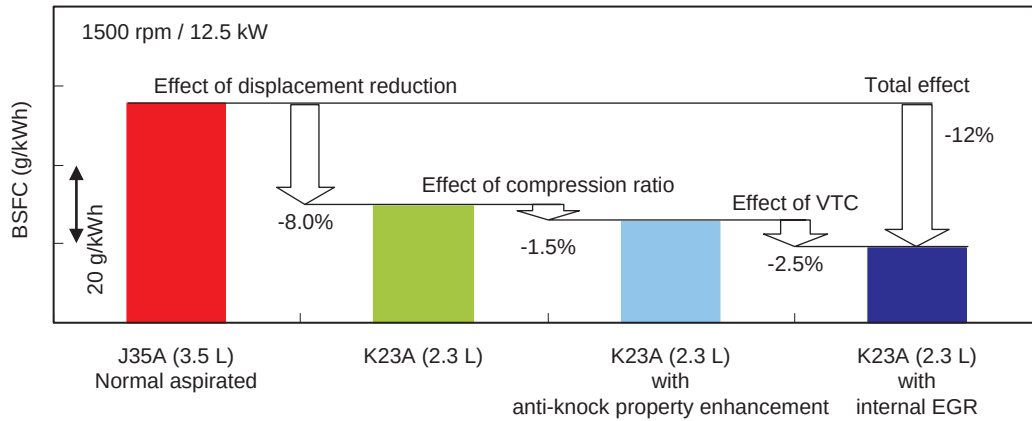


Fig. 11 Effect of displacement reduction on BSFC

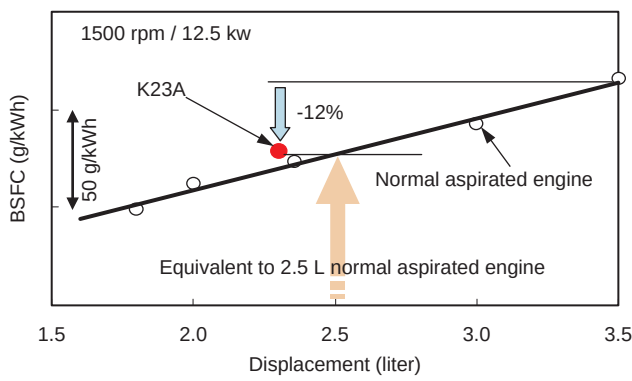


Fig. 12 Positioning of BSFC

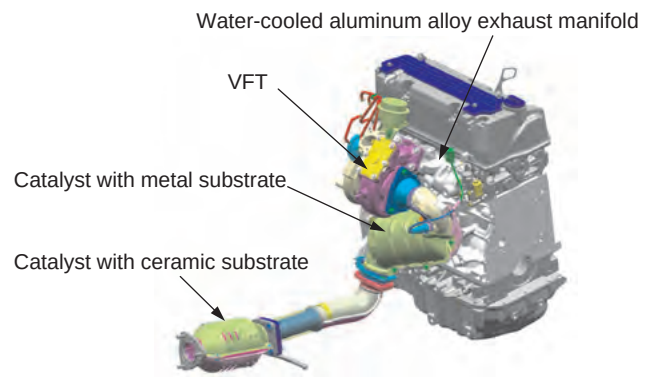


Fig. 13 View of exhaust system

6. Low Emission Technologies

The exhaust emission reduction system incorporates a metal substrate catalyst located directly beneath the VFT, as well as a ceramic substrate under-floor catalyst. Figure 13 shows this engine's exhaust system.

Because a water-cooled aluminum alloy exhaust manifold is used in the engine, the catalyst temperature 20 sec. after start-up was increased by approximately 120 °C over that of a conventional stainless steel exhaust manifold, contributing to rapid activation of the exhaust catalyst. Figure 14 shows the exhaust catalyst temperature change over time immediately after start-up.

A metal substrate with a lower cell density was used for the catalyst positioned immediately beneath the turbocharger to lower the exhaust back-pressure. Despite the fact that the metal substrate has higher thermal transfer characteristics than those of a ceramic substrate, which is a disadvantage in terms of warm-up time, this change was necessary to meet power targets. With this engine, the metal substrate was divided into two parts and the effect of the catalyst segmentation ratio on warm-up characteristics was studied using a 2D simulation of the catalyst reaction. This used chemical reaction and heat transfer models to calculate the catalyst temperature from the data on exhaust gas introduced into the catalyst. Dividing the catalyst enhanced the warm up characteristics

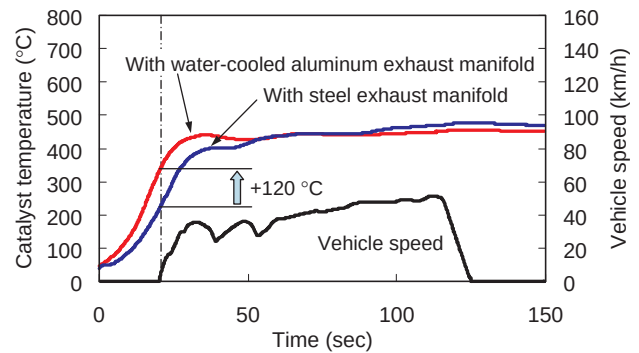


Fig. 14 Effect of water-cooled exhaust manifold on catalytic converter warm-up characteristics

of the upstream catalytic converter due to the insulating effect of the gap created between the bricks. It was shown that reducing the capacity of the upstream catalyst was more effective in enhancing warm-up characteristics. A comparison of warm-up characteristics in relation to the catalyst segmentation ratio is shown in Fig. 15.

From the results of the simulation, setting the catalyst segmentation ratio to 0.45 L upstream and 0.95 L downstream enabled catalyst temperature to rise by an additional 20 °C compared to when the catalyst was divided equally. As a result of

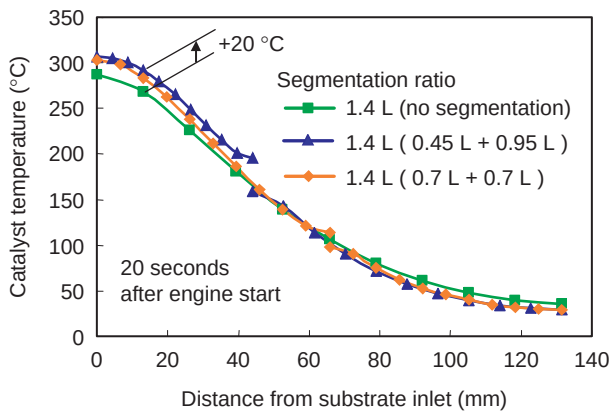


Fig. 15 Simulation results for catalyst temperature

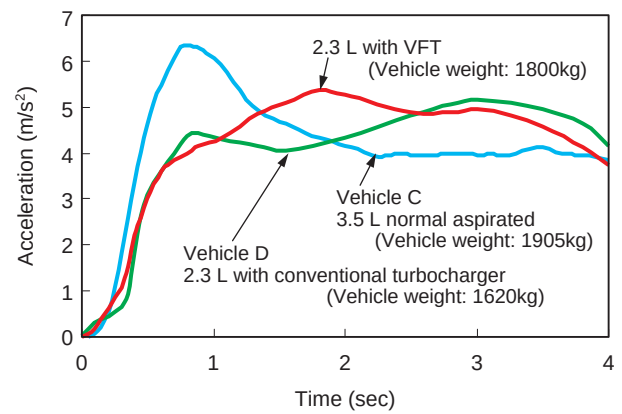


Fig. 17 Comparison of take-off acceleration performance

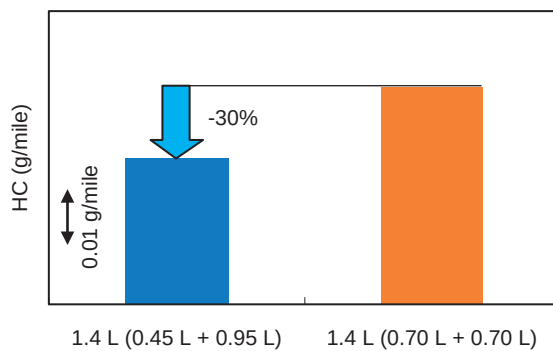


Fig. 16 Tail-pipe emissions

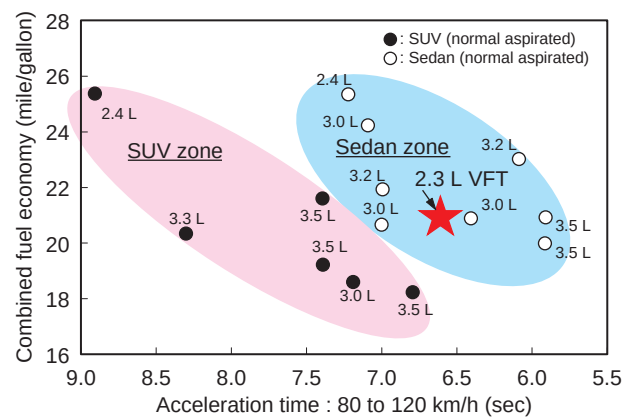


Fig. 18 Positioning of acceleration time and fuel economy

applying this catalyst segmentation ratio to the engine, it was possible to reduce the HC concentration in exhaust gases by 30% in U.S. emissions FTP mode (cold start), as shown in Fig. 16.

In addition to low emissions technologies⁽³⁾, such as the i-VTEC carried over from the K24A, enhancing the catalyst's warm-up characteristics using the aforementioned exhaust system allowed U.S. ULEV2 emission standards to be met.

7. Vehicle Performance and Fuel Economy

7.1. Standing Start Acceleration Performance

Torque converter characteristics of the automatic transmission were optimized with response characteristics of the VFT engine in mind to achieve the targets of smooth yet firm starting acceleration with no feeling of stagnation or lag. Figure 17 compares the starting acceleration performances of an SUV with the VFT engine installed, a vehicle with a normal aspirated 3.5 L engine (Vehicle C), and a vehicle with a 2.3 L engine with a conventional large turbocharger (Vehicle D).

Vehicle C, with the large displacement normal aspirated engine, generates maximum acceleration immediately after the throttle is wide open using the torque converter amplification effect. Then acceleration gently drops off. Vehicle D, using an engine with a conventional large turbocharger, reaches an acceleration peak somewhat later due to the effects of turbo lag, followed by a gentle drop in acceleration. After that, acceleration once again rises as boost pressure rises, resulting in a feeling of acceleration stagnation.

With the newly developed engine, the VFT makes the boost pressure rise earlier, and prevents a drop in acceleration, thus eliminating the feeling of acceleration stagnation produced by a conventional high-powered turbo engine.

7.2. Vehicle Fuel Economy

The fuel economy achieved with the VFT engine was a U.S. label combined 21 mpg. Compared to SUVs in the same class, it was best in both acceleration performance and fuel economy. Figure 18 shows the vehicle's relative acceleration performance and U.S. combined fuel economy.

8. Conclusion

One of the major differences in the new engine is the use of the newly developed VFT. This ensures driving performance that compares favorably to that of a large displacement normal aspirated engine. The fuel economy is also benefited by reducing engine displacement. In addition to good acceleration performance and fuel economy the forced induction VFT engine achieved low emissions by enhancing catalytic converter warm-up characteristics through technologies such as the aluminum alloy exhaust manifold and i-VTEC. The following results were obtained:

- (1) A peak power of 179 kW was achieved with maximum boost pressure being available at engine speeds of 2000 rpm or less. This made forced induction possible at low engine speeds with little response delay providing a smooth and responsive starting acceleration with no feeling of stagnation.
- (2) The development of an aluminum alloy exhaust manifold and optimization of the segmentation ratio of the metal substrate of the catalytic converter located immediately beneath the turbocharger enabled catalyst warm-up characteristics to be enhanced and U.S. ULEV2 emissions regulations to be met.
- (3) The effect of displacement reduction enabled a vehicle with this engine to achieve 21 mpg U.S. combined label fuel economy, which is the best for that class of SUVs.

References

- (1) Ishii, M., Adachi, K., Kono, S.: Variable turbocharger, Journal of Society of Automotive Engineers of Japan, Inc., Vol. 58, No. 4, p. 42-47 (2004)
- (2) Niizato, T., Hayashi, A.: Development of New 2.0L Lean-burn Engine, HONDA R&D Technical Review, Vol. 12, No. 2, p. 45-54
- (3) Kitagawa, H., Minobe, T., Okamatsu, K., Yasui, H.: Ultra-Low Emission System for the “Accord SULEV”, HONDA R&D Technical Review, Vol. 12, No. 2, p. 37-44
- (4) Wada, Y., Ohtsu, K., Narita, K., Shinohara, T.: Development of i-DSI 2-plug Engine for 2004 Model Year Honda LIFE, Honda R&D Technical Review, Vol. 16, No. 1, p. 93-102

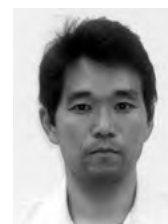
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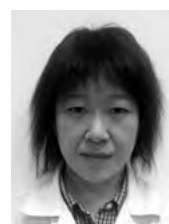
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Development of 3-stage i-VTEC VCM Engine for CIVIC Hybrid

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ABSTRACT

A 1.3 L 3-stage i-VTEC VCM engine, which satisfies both low fuel consumption and low emission requirements, was developed for the 2006 CIVIC Hybrid. In the development of this engine, friction was greatly reduced and volumetric efficiency was increased. In addition, VTEC technology was evolved to realize a 3-stage i-VTEC VCM that enables 3-stage valve motion switchover at low and high engine speeds and during cylinder idling. With the increased motor output, it is possible to run the automobile on the motor alone over a wider range of driving conditions, and to realize a further reduction in fuel consumption. The newly developed engine, higher power motor and more efficient CVT transmission enable the 2006 CIVIC Hybrid to achieve better vehicle dynamic performance and a fuel consumption of 31.0 km/L in the 10-15 mode. In addition, the performance of the high density 2-bed, 3-way catalyst has been increased, realizing low emissions equal to 75% of the low emissions standard for fiscal 2005.

1. Introduction

The necessity of protecting the global environment and conserving resources is increasing demands for greater automotive fuel economy and reduced exhaust emissions. Responding to those demands, Honda has continued to research and develop Variable Valve Timing and Lift Electronic Control (VTEC)-equipped next-generation engines^{(1),(2)}. Honda has also developed a proprietary Integrated Motor Assist (IMA) system, a hybrid powertrain combining an electric motor and an engine that was first used in the ultra-high fuel economy 1999 Insight⁽³⁾. The IMA system was also used to enhance fuel economy in the CIVIC Hybrid⁽⁴⁾, a hybrid small sedan marketed in 2001.

A new powertrain has been developed for the 2006 CIVIC Hybrid, using the LDA-MF3 engine⁽⁴⁾ from the 2001 model as the base engine. Friction has been reduced and volumetric efficiency enhanced in the new LDA-MF5 engine. In addition, the engine has been provided with a 3-stage i-VTEC Variable Cylinder Management (VCM) mechanism, enabling switching between three stages of valve operation: Low engine speeds, high engine speeds, and cylinder idling (valve deactivation). The power of the motor and the torque capacity and transmission efficiency of the CVT have also been enhanced, increasing the efficiency of the powertrain as a whole.

This paper will discuss the new technologies used in the LDA-MF5 engine, with a focus on the 3-stage i-VTEC VCM mechanism.

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2. Development Goals

The IMA system used in the 2001 CIVIC Hybrid enabled the vehicle to achieve the world's top level of fuel economy for a small 5-passenger sedan.

In the new development, all aspects of the hybrid powertrain were enhanced, and development goals were set for achieving enhanced driving performance and fuel economy as well as greater passenger comfort, all in a more compact package.

The specific targets were as follows:

- (1) Fuel economy of 31.0 km/L in domestic 10-15 mode
- (2) Acceleration performance of 0-100 km/h in 12 seconds
- (3) An emissions level 75% lower than the FY2005 low emissions standards
- (4) High responsiveness and a natural drivability equivalent to that of a gasoline vehicle, offering exhilarating driving in all ranges

3. Hybrid System

The new system uses a thin DC brushless motor, which is positioned between the engine and the CVT and is directly connected to the engine crankshaft. An integrated power unit (IPU), containing the high-voltage components (the inverter unit that drives the motor, a high-voltage Ni-MH battery pack, etc.) and a cooling system, is located in the trunk behind the rear seat. The power cable connecting the motor and the IPU runs underneath the floor

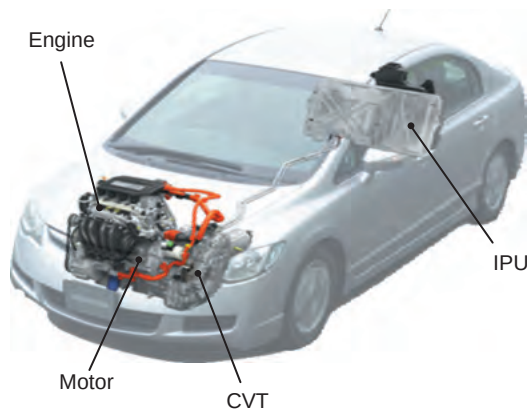


Fig. 1 IMA system construction

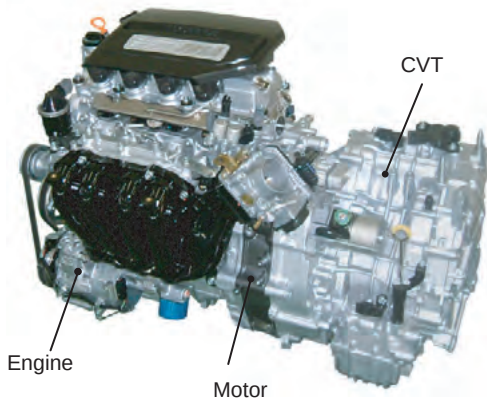


Fig. 2 View of newly developed hybrid powertrain

Table 1 Engine specifications

Engine code	LDA-MF5	LDA-MF3
Cylinder configuration	In-line 4-cylinder	In-line 4-cylinder
Bore × stroke (mm)	73 × 80	73 × 80
Displacement (cm ³)	1339	1339
Compression ratio	10.8 : 1	10.8 : 1
Valve train	SOHC 3-stage VTEC	SOHC 2-stage VTEC
Number of valves	2 per cylinder	2 per cylinder
Valve diameter (mm)		
Inlet	35	32
Exhaust	30	28
Fuel supply system	Programmed fuel injection	Programmed fuel injection
Ignition system	Dual point sequential ignition	Dual point sequential ignition
Intake manifold Diameter × length (mm)	34 × 340	30 × 200
Exhaust manifold	Integrated cylinder head	Conventional
Gasoline	Regular (RON91)	Regular (RON91)
Max. power (kW/rpm)	70/6000	63/5700
Max. torque (Nm/rpm)	123/4600	119/3300

of the cabin. Figure 1 shows the configuration of the components in the vehicle. Figure 2 shows an external view of the hybrid powertrain.

With the engine as the main power source and the motor functioning as a supplementary power source, it was possible to make the overall system simple and lightweight.

4. Overview of Engine and Main Specifications

Table 1 shows a comparison of the major specifications of the LDA-MF5 engine and the LDA-MF3 engine used in the 2001 CIVIC Hybrid. In the newly developed engine, the diameter of the inlet and exhaust valves has been increased, the shape of the inlet and exhaust ports has been optimized and a long, large-diameter intake manifold with a plastic resonator has been used. This resulted in increased volumetric efficiency and an 11% increase in power. A variety of measures were taken to reduce friction, including integrating the exhaust manifold and cylinder head to enable rapid heating of the cooling water. In addition, the new 3-stage i-VTEC VCM mechanism enables the system to switch between low and high engine speed valve timing and deactivate the valves to balance fuel economy and power.

Figure 3 shows a cross section of the intake manifold with attached plastic resonator. The optimum pipe length and diameter ($\phi 34 \times 340$ mm) for power at high engine speeds were used. This resulted in an increase of approximately 5% in volumetric efficiency at 4500 rpm and above compared to the intake manifold used in the LDA-MF3 engine. Positioning the resonator over the chamber synchronized resonating frequencies, resulting in increased volumetric efficiency in the medium engine speed range. As Fig. 4 shows, when the resonator is in operation, the intake port pressure directly before closure of the intake valve is higher, resulting in increased air intake. Figure 5 shows the effect of the resonator on the power characteristic. The resonator increases power between 3000 and 4000 rpm, resulting in the achievement of an excellent torque characteristic.

Figure 6 shows an external view of the exhaust manifold integrated cylinder head, and the configuration of the water jacket. This configuration ensures that exhaust gas concentrates in the cylinder head. Heat from the exhaust gas heats the cooling water, reducing the time required to warm up the engine. Also, eliminating the steel exhaust manifold decreased the weight of the engine.

Figure 7 shows a diagram of the cooling water circulation channels. The cooling water flows first to the cylinder head, enabling rapid warming using the heat of the exhaust gas. A bypass channel located in the cylinder block channel rising from the water pump to the cylinder head increases the coolant flow over the upper section of the cylinder block, suppressing engine knocking.

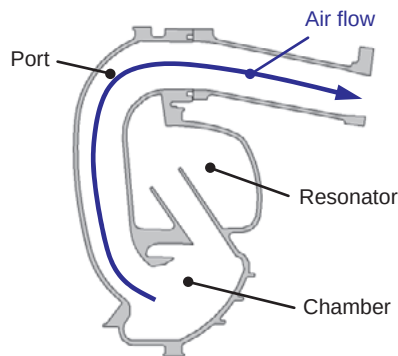


Fig. 3 Structure of intake manifold

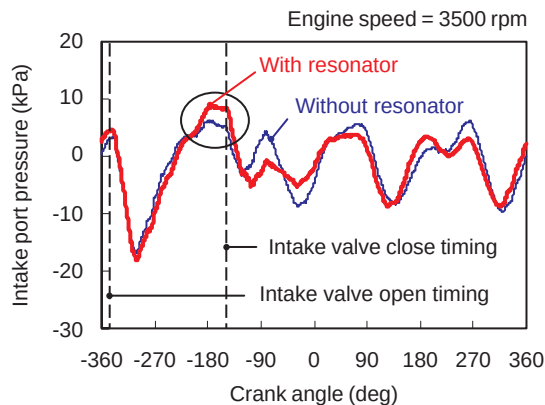


Fig. 4 Characteristics of intake manifold pressure

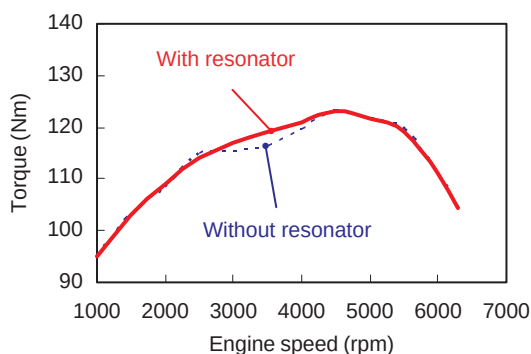


Fig. 5 Effect of intake manifold resonator

Figure 8 shows the effect of the bypass channel on ignition timing for knocking. The use of the bypass channel reduced knocking at 2000 rpm and above by approximately 2 deg.

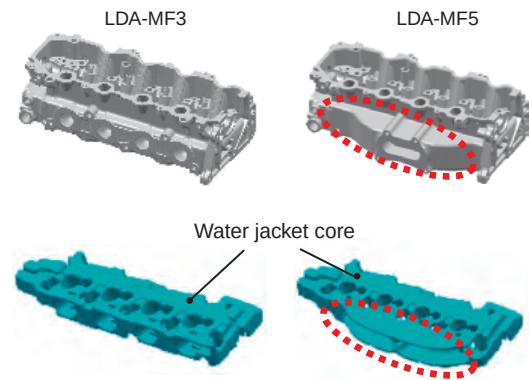


Fig. 6 View of exhaust manifold integrated cylinder head

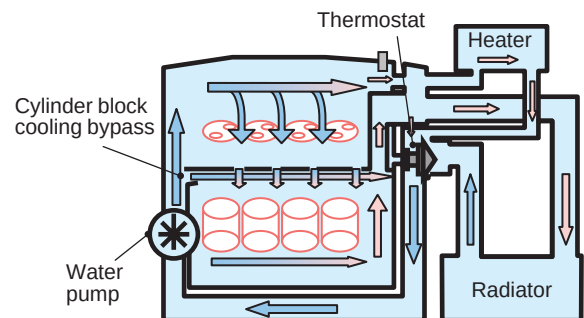


Fig. 7 View of cooling system

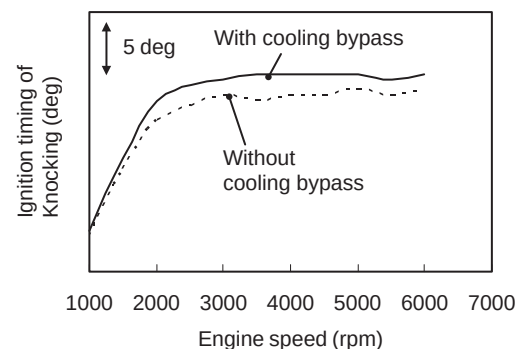


Fig. 8 Effect of newly developed cooling system

5. Power Performance

Figure 9 shows the output performance of the IMA system. The LDA-MF5 system displays superior power to the LDA-MF3 system across the entire driving range, reflecting the enhanced performance of both the engine and the motor. The high torque generated by the motor assist system at low engine speeds gives the system superior performance at 3000 rpm and below compared to a 1.8L i-VTEC (R18A) engine⁽⁵⁾ fitted in an identical vehicle.

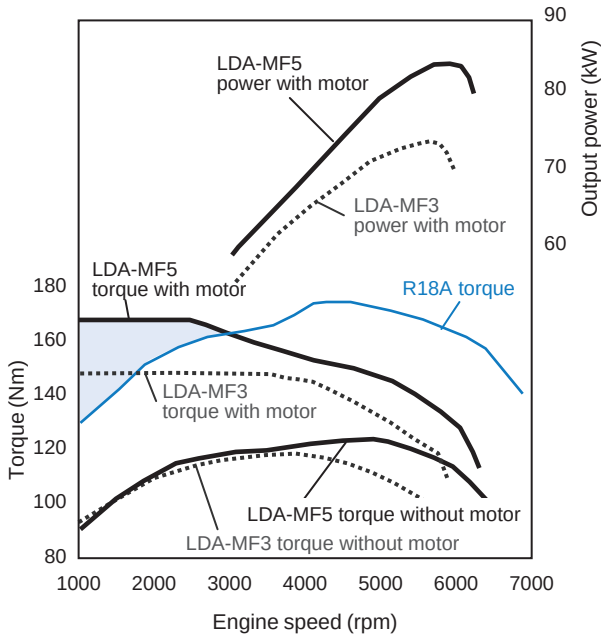


Fig. 9 Output performance of IMA system

6. 3-Stage i-VTEC VCM

6.1. Structure and Principle

The cylinder idling VTEC system used in the LDA-MF3 system provided 2-stage inlet and exhaust valve control: valve operation and cylinder idling. The LDA-MF5 system offers 3-stage control, enabling switching between different valve timing at low engine speeds, high engine speeds and during cylinder deactivation. This provides excellent fuel economy and torque at low speeds and high power at high speeds.

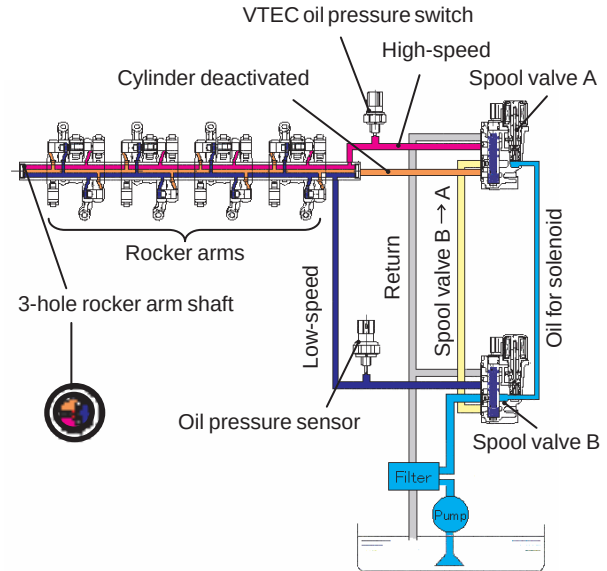
Figure 10 shows the component configuration of the 3-stage i-VTEC VCM mechanism. As in previous VTEC mechanisms, hydraulic pressure controls spool valves and synchro pistons built into the rocker arms, allowing the amount of lift of the intake and exhaust valves to be varied. The new system adds an extra spool valve to the single valve used in the LDA-MF3, and the hydraulic channel for use in cylinder idling has been made independent. These modifications have increased the reliability of start-up from cylinder idling, enabling all four cylinders to be idled, as compared to the three-cylinder idling used in the LDA-MF3.

Figure 11 shows the configuration of the rocker arms. Five rocker arms, which are composed of synchro pistons and return springs, are provided for each cylinder: three on the inlet side and two on the exhaust side. As in the LDA-MF3, a hydraulic/hydraulic switching mechanism that uses hydraulic pressure for connection and disconnection of the rocker arms was used for switching between low-speed valve timing and lift and cylinder idling. This ensures that it is possible to switch at low speeds of around 1000 rpm, when hydraulic pressure is low. A simple hydraulic/spring mechanism, in which the rocker arms are connected by hydraulic pressure and disconnected by the pressure of a spring, was used for switching between low-speed valve timing and lift and high-speed valve timing and lift.

Figure 11(a) shows valve operation at low speeds. Hydraulic pressure acts on the deactivation rocker arm, connecting it to the

low-speed rocker arm. The connected rocker arms drive the valves according to the low-speed cam profile.

Figure 11(b) shows valve operation at high speeds. The connection between the low-speed rocker arm and the deactivation rocker arm is maintained by the return spring. At the same time, hydraulic pressure acts on the hydraulic channel of the deactivation rocker arm on the high-speed rocker arm side. The operation of the



	Spool valve A	Spool valve B
Low-speed	OFF	OFF
High-speed	OFF	ON
Cylinder deactivated	ON	ON

Fig. 10 3-stage i-VTEC VCM valve train and hydraulic passage

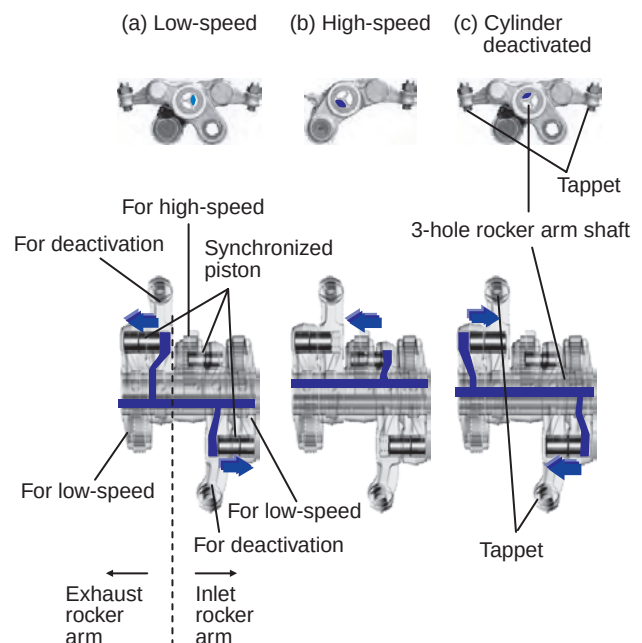


Fig. 11 3-stage i-VTEC VCM rocker arm and rocker arm shaft construction

synchro piston connects the deactivation rocker arm and the high-speed rocker arm. The valves are driven by the connected rocker arms according to the high-speed cam profile. On the exhaust side, only one type of cam is used, and therefore there is no cam switching.

Cylinder idling is shown in Figure 11(c). Hydraulic pressure acts on the low-speed rocker arm, and the operation of the synchro piston disconnects the deactivation and low-speed rocker arms. The valves are deactivated by the deactivation rocker arm (zero cam lift).

6.2. Effects

Figure 12 shows the output characteristic at low-speed and high-speed valve timing and lift. Switching between the low-speed and high-speed rocker arms at around 4800 rpm ensures an excellent power characteristic from low speeds to high speeds. In addition, the use of low-speed valve timing and lift provides excellent combustion at low speeds and low loads, resulting in superior fuel economy and emissions performance.

When the vehicle is decelerating and fuel is not being injected, the idling of the intake and exhaust valves reduces pumping loss. This friction reduction is captured by the motor as regenerated energy. Figure 13 shows engine friction during deceleration (fully closed throttle). Compared to deactivation of three cylinders, which reduces engine friction against full valve operation by a maximum of approximately 50%, deactivation of all four cylinders reduces pumping loss to virtually zero and reduces engine friction by a maximum of approximately 66% against full valve operation. This has increased the ratio of regenerated energy to braking energy from 57% to 64%, providing increased motor assist during

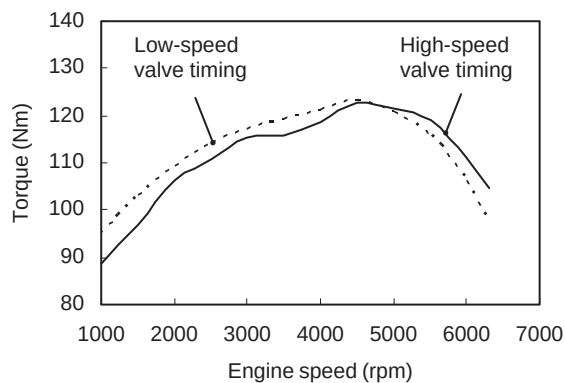


Fig. 12 Effect of valve timing and lift on torque

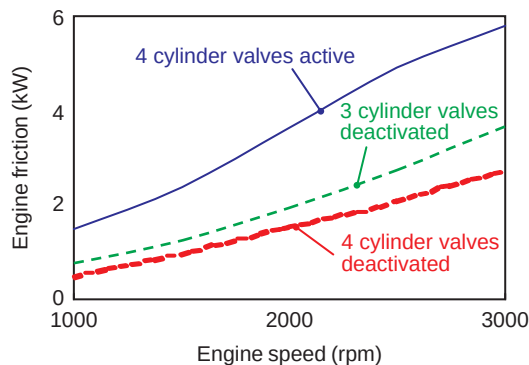


Fig. 13 Effect of VCM on reduction of engine friction

acceleration. In combination with increased motor power, this allows vehicle operation by the motor alone during low-speed cruising, resulting in increased fuel economy.

6.3. System Control

Figure 14 shows a diagram of the operation of the 3-stage i-VTEC VCM and the motor. When the vehicle becomes stationary, the engine automatically stops; the engine is restarted by the motor when the brakes are released. Low-speed valve timing and lift at take-off and at low vehicle speeds provides excellent fuel economy and torque. The motor generates drive power in response to the magnitude of torque required. The vehicle operates on motor power alone at low speeds and low loads. At this time, all the cylinders are deactivated and no fuel is injected. When the vehicle rapidly accelerates, the system switches to high-speed valve timing and lift at around 4800 rpm and above in order to generate high power. During deceleration, fuel injection is suspended and all the cylinders are deactivated, reducing pumping loss and increasing regenerated energy.

Switching between stages in the 3-stage i-VTEC VCM system is conducted based on a number of vehicle parameters, including engine speed, hydraulic pressure, water temperature, vehicle speed and remaining charge in the high-voltage batteries. This provides natural drivability and an acceleration and deceleration feel equivalent to those offered by a standard gasoline vehicle.

The system uses torque management control, a cooperative control involving drive by wire (DBW) and cylinder idling, to generate drive force. The target torque for the end of the crankshaft is decided from the degree of opening of the accelerator pedal and engine speed. The necessary allocation of engine and motor torque to realize this target torque is determined with consideration of energy management (remaining battery charge, etc.).

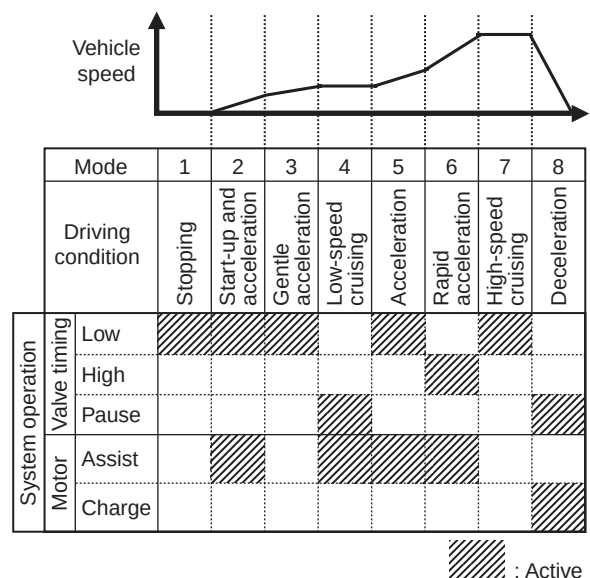


Fig. 14 3-stage i-VTEC+IMA operation

Table 2 Engine friction reduction technology

Engine code	LDA-MF5	LDA-MF3
Roller rocker arm	○	○
Off-set cylinder arrangement	○	○
Low-tension piston ring	○	○
Streak crank bearing	○	○
MoS ₂ processing of piston skirt	○	○
Cam shaft chain drive system	○	○
Low-viscosity engine oil	○	○
Cylinder sleeve plateau honing	○	○
Mirror surface treatment cam journal	○	—
Aluminum diecast piston	○	—
Ion-plate piston ring	○	—
Manifold-integrated cylinder head	○	—
Air conditioner belt-auto tensioner	○	—

7. Reduction of Engine Friction

Table 2 shows the main technologies used in the reduction of engine friction. In addition to the numerous friction reduction technologies previously used in the LDA-MF3, the new engine also features high-strength die-cast pistons⁽⁶⁾, an exhaust manifold integrated cylinder head, mirror surface treatment of the cam journals and an auto-tensioner for the accessories belt, among other new technologies that reduce friction.

Figure 15 shows the characteristics of the materials used in the high-strength die-cast pistons. The material used in the new pistons has a lower rate of thermal expansion than the previous material, controlling increases in friction generated by temperature changes. CAE was used in three-dimensional behavioral calculations to achieve uniform stiffness and optimize the shape of the pistons, in order to maximize friction reduction while ensuring durability and reliability.

Figure 16 shows a graph of engine friction against total displacement for existing Honda engines at an engine speed of 1500 rpm and an intake pressure of -53 kPa. In addition to new friction reduction technologies, the achievement of weight savings in the valve springs and piston rings and the reduction of contact area in the LDA-MF5 engine reduced friction by 8% against the

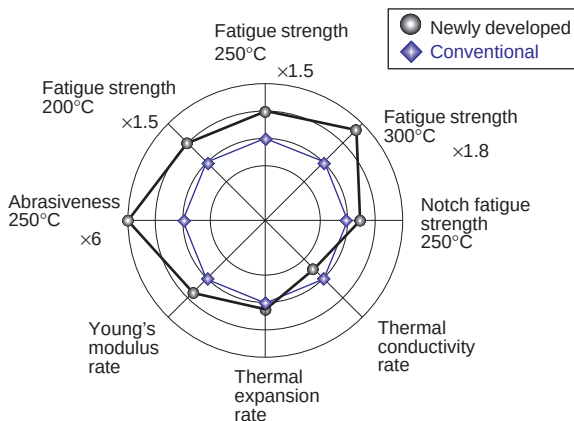


Fig. 15 Characteristics of piston materials

Engine speed = 1500 rpm, Intake pressure = -53 kPa

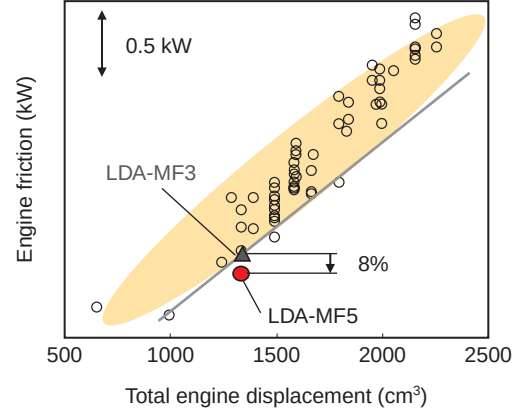


Fig. 16 Enhancement of engine friction

LDA-MF3 engine. This is the lowest figure for engine friction yet achieved by Honda in an engine of this displacement.

8. Reduction of Fuel Consumption through Enhanced Combustion

A new intake port that enhances turbulence at the end of the compression stroke (directly before ignition) by means of increased air intake and the generation of increased swirl (horizontal vortices) and tumble (vertical vortices) was developed. This new port reduces fuel consumption by enhancing combustion while at the same time increasing volumetric efficiency. Figure 17 shows the shape of the intake port, and Figs. 18 and 19 show CAE analyses of in-cylinder turbulence. The use of a restriction inside the port close to the valves in addition to a jumping spot increased the directionality of swirl and tumble during the intake stroke, resulting in the achievement of strong turbulence at 10 deg BTDC, directly before the completion of the compression stroke.

Figure 20 shows the relationship between the intake speed coefficient, M_i , defined by Eq. (1), and vortex strength, V_i , defined by Eq. (2). Vortex strength has been increased by approximately 25% against the previous model.

$$\text{Intake speed coefficient } M_i = (U_p/a_i) \cdot (A_p/Z_{MEI}) \quad (1)$$

U_p : Average piston speed (m/s)

a_i : Speed of sound passing the intake valves (m/s)

A_p : Cross-sectional area of piston (m²)

Z_{MEI} : Average effective open area of intake valve (m²)

According to C. F. Taylor⁽⁷⁾, the major determinant of volumetric efficiency during the intake stroke is the pressure drop of the intake valves. The pressure drop can be considered as a Mach number function of the intake valves, and defined as the intake speed coefficient, M_i .

$$\text{Vortex speed } V_i = (S_r^2 + T_r^2)^{0.5} \quad (2)$$

S_r : Swirl ratio

T_r : Tumble ratio

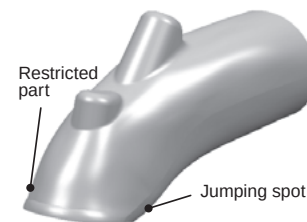


Fig. 17 View of intake head port

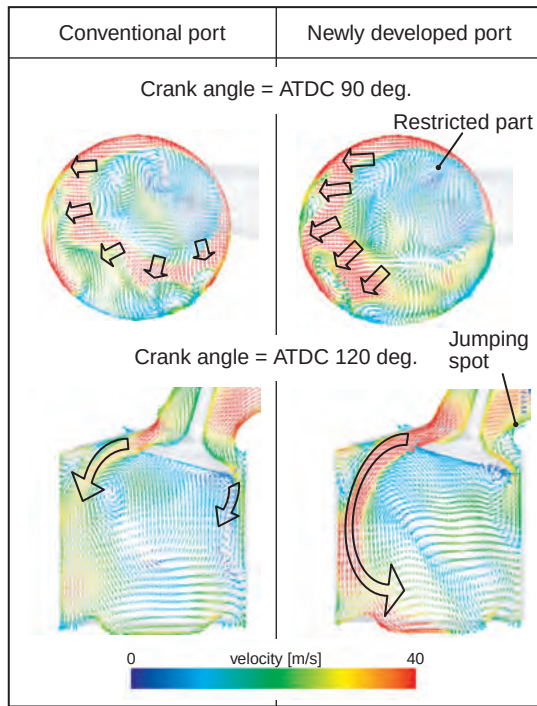


Fig. 18 Comparison of swirl and tumble flow

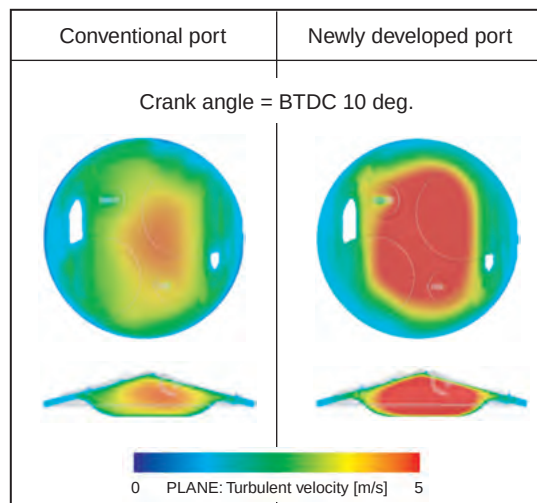


Fig. 19 Comparison of turbulence intensity

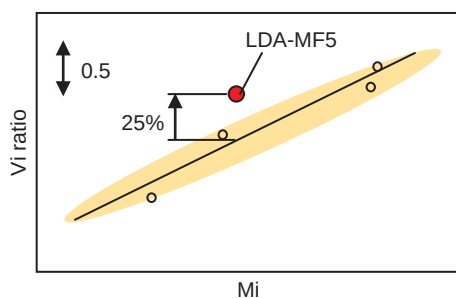


Fig. 20 Enhancement of Vi ratio

Figure 21 shows BSFC, Nox emissions and the co-variation of IMEP against EGR rate. Combustion is stable even at EGR rates of over 25%, enabling mass EGR to be used. As a result, NOx emissions have been reduced and a reduction of 5% in BSFC has been achieved.

Lean combustion was used in the LDA-MF3 to reduce fuel consumption, but not in the LDA-MF5. This is because motor operation takes place in the range in which lean combustion was previously used.

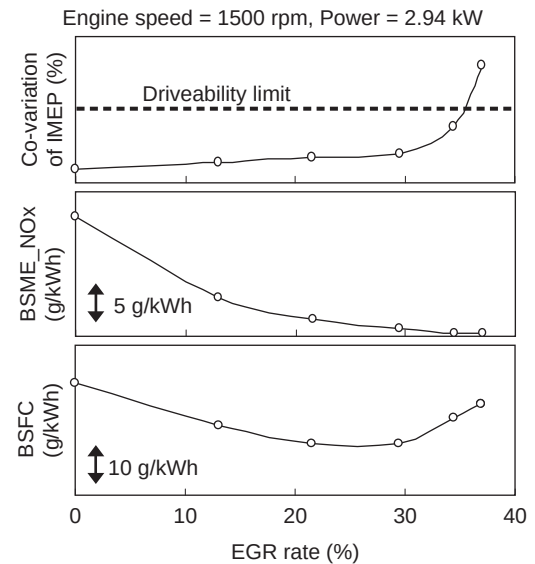


Fig. 21 Characteristics of EGR

9. Vehicle Fuel Economy and Dynamic Performance

The use of the 3-stage i-VTEC VCM mechanism, increased motor power, increased transmission efficiency in the CVT, and vehicle technologies including cooperative control of braking regeneration, have resulted in a fuel consumption of 31.0 km/L in 10-15 mode, a 5% increase against the previous vehicle. In terms of dynamic performance, increased engine and motor power have enabled 0-100 km/h acceleration in 12.1 sec. Figure 22 shows the positioning of the vehicle in terms of 0-100 km/h acceleration and 10-15 mode fuel economy.

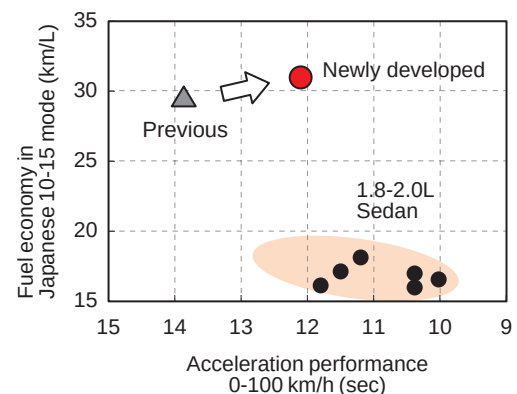


Fig. 22 0-100 km/h acceleration performance and fuel economy

10. Emissions Reduction Technologies

The catalyst used in the vehicle is a high-density, two-bed, three-way, close-coupled model. The enhancement of the catalytic converter⁽⁸⁾ has resulted in excellent purification performance, enabling the precious metals loading to be reduced. The exhaust manifold integrated cylinder head has eliminated the need for the exhaust manifold, thereby reducing the area radiating exhaust heat, thus enabling more rapid activation of the catalyst. Other emissions reduction technologies include a 6-hole injector and electronic EGR valves. The use of these technologies in addition to high-precision engine control has enabled the achievement of the emissions target, an emissions level 75% lower than the FY2005 low emissions standards.

11. Conclusion

A new engine, the LDA-MF5, has been developed for the 2006 CIVIC Hybrid. The following are the salient points of this development:

- (1) A 3-stage i-VTEC VCM mechanism has been developed that enables switching between low and high speed engine valve timing and cylinder idling, thus achieving a balance between excellent fuel economy and torque at low and medium engine speeds and high power at high engine speeds.
- (2) Deactivation of all four cylinders has reduced pumping loss to almost zero, resulting in increased regenerated energy and enabling operation by the motor alone.
- (3) In combination with vehicle technologies, the newly developed hybrid powertrain has enabled the achievement of a fuel economy of 31.0 km/L in 10-15 mode and 0-100 km/h acceleration in 12.1 sec.
- (4) The use of a high-density, 2-bed, 3-way catalyst and an exhaust manifold integrated cylinder head, in addition to high-precision engine control, has resulted in the achievement of an emissions level 75% lower than the FY2005 low emissions standards.

References

- (1) Oura, S., Miura, K.: Making Gasoline Engine More Efficient using Variable Structures, Journal of Society of Automotive Engineers of Japan, Vol. 45, No. 8, p. 93-98 (1991)
- (2) Niizato, T., Hayashi, A.: Development of New Lean-burn Engine, HONDA R&D Technical Review, Vol. 12, No. 2, p. 45-54
- (3) Aoki, K., Kajiwar, S., Sato, H., Yamamoto, Y.: Development of Motor Type Hybrid Power Train, 1999 JSAE Annual Congress Proceedings, No. 98-99, p. 9-12 (1999)
- (4) Matsuki, M., Wakashiro, T., Kamiyama, T., Sato, T., Kaku, T., Kanda, M.: Development of a New Power Train for the CIVIC Hybrid, Honda R&D Technical Review, Vol. 14, No. 1, p. 39-48
- (5) Seko, K., Taga, W., Torii, K., Nakamura, S., Akima, K.: Development of New 1.8L 4-Cylinder Gasoline Engine, 2005 JSAE Annual Congress Proceedings, No. 106-05, p. 1-4 (2005)
- (6) Tanihata, A., Sato, K., Furukawa, K.: Development of High-strength Piston Material with Die Casting, Honda R&D Technical Review, Vol. 17, No. 2, p. 122-128
- (7) Taylor, C. F.: The Internal Combustion Engine in Theory and Practice, MIT Press

- (8) Inose, S., Okayama, T.: Research for High Concentration of Precious Metal and Thin Layer of Catalyst Top Layer, Honda R&D Technical Review, Vol. 17, No. 2, p. 129-135

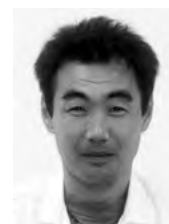
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Construction of Power Product Data Management System

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ABSTRACT

Manufacture of the iGX440, the first in the iGX series of next-generation general-purpose engines, began in September 2005. In place of a conventional governor mechanism providing mechanical control, the iGX440 uses an electronic governor that applies control via an ECU. Because the unit is fitted with an ECU, data can be altered, enabling the programming of optimal settings for engine speed and governor characteristics, allowing a diverse range of uses.

The new system makes it possible to ensure that all data is saved when the ECU requires replacement by storing data in one location, and has also made it possible to eliminate unnecessary model types by enabling data to be overwritten during the distribution process. In addition, the data accumulated in the data base enables the use patterns of the engines to be analyzed, enabling future development that will expand operation in entirely new directions.

1. Introduction

General purpose engines are used as power sources for various work equipment, including construction, agricultural, and industrial machinery. These machines generally use a mechanical governor mechanism, but it is necessary to change the springs that comprise the governor, adjust the idle speed, or make other adjustments in accordance with the demands for each work machine. However, there are numerous combinations related to the engine speed setting alone, so this makes it necessary to perform inventory management.

In response to these needs, the iGX440 electronically controlled general purpose engine⁽¹⁾ uses an electronic governor mechanism⁽²⁾, whereby the target engine speed and governor characteristics can

be changed by altering the ECU data. This allows manufacturers using this engine in their products to set target engine speeds suited to their work equipment, and dealers wholesaling the engine to also change the governor characteristics in accordance with customer requests. The governor characteristics can be changed after shipping, so inventory can be reduced and the lead time to the customer can also be shortened by reducing the number of model types. However, even though the same ECU is used, the ability to set different characteristics has created the issue that when an ECU fails and is brought to a service center, the original governor characteristics cannot be restored simply by replacing the ECU. In addition, it has also become necessary to create terms of agreement and other rules concerning characteristic changes after shipping,

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such as change privileges, change methods, division of responsibility, intellectual property rights, legal responses, and so on.

This paper presents technical methods and rules when changing the characteristics of power products equipped with ECU after shipping. In addition, technical aspects are also introduced as an example of database application using the Internet for centralized management of iGX440 change data on a global scale.

2. Development Goals

When an ECU fails and is brought to a service center, the original governor characteristics cannot be restored simply by replacing the ECU. This makes it essential to construct before-sale (B/S) and after-sale (A/S) systems. The following goals were set to realize an ECU management system, starting with an information system.

- (1) Centralized management of data change history from the time of shipping from the factory to the latest overwrite.
- (2) Standardization of equipment and tools used to overwrite data.

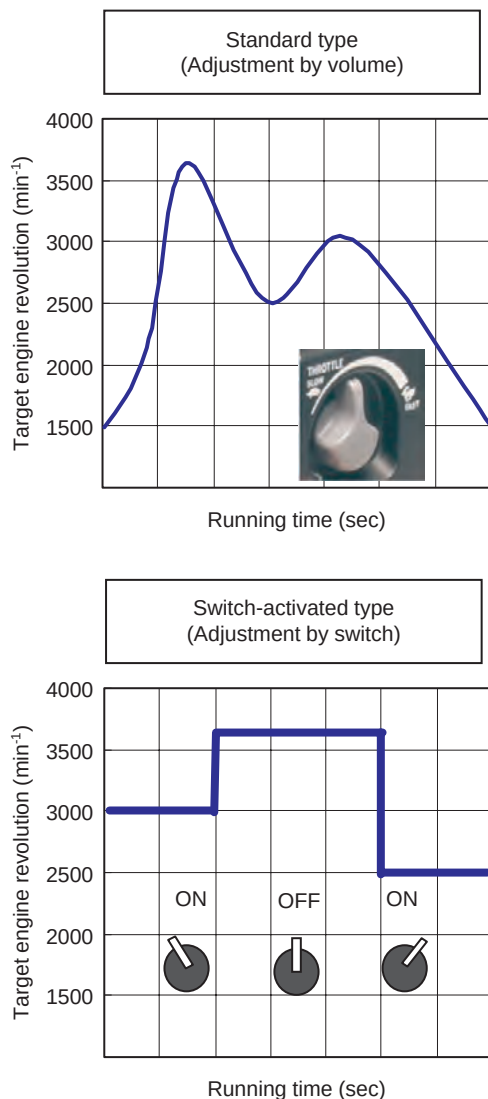


Fig. 1 Operation mode

- (3) Establishment of basic rules for writing and overwriting ECU data.

In addition, assuring data when ECUs are replaced and eliminating unnecessary model types were also set as development goals.

3. iGX Data Management System

3.1. iGX Engines

Conventional mechanical governor mechanisms set the target engine speed mechanically using a throttle lever. In contrast, iGX engines use an electronic governor mechanism, and can select the means by which the target engine speed is changed. Figure 1 shows an example of modes that can be selected. For example, when standard mode is selected, the target engine speed can be changed by an electrical potentiometer, and when switch-activated mode is selected, the target engine speed can be changed by a switch. In addition, the rate of change when the target engine speed is changed and other characteristics can also be optionally set. In this manner, various demands for each work machine can be supported by changing the governor characteristics.

3.2. ECU Data Management Method

The governor characteristics of iGX engines can be overwritten during the distribution process, so a data management system for centralized management of the ECU overwrite data is essential. The engine ECU data is stored in the database when the engine is shipped from the factory, and it is also mandatory that the changed data be updated to the database when the data is overwritten during the distribution process. This makes it possible to restore the latest data even when an ECU is replaced due to failure or other circumstances. Figure 2 shows the overall system configuration.

3.2.1. Initial registration

The governor characteristics, type code, completed engine inspection date, characteristics change date, person changing the characteristics, failure history, total running time and other data for each engine number are transferred to the database and registered as the initial data when the engine is inspected at the factory.

3.2.2. Uploading

Manufacturers who use iGX engines in their products must overwrite the ECU data when changing the settings to suit their work equipment. This changed data is uploaded to the database via the Internet. When uploading, only the changed portions are updated and stored in the database, so the history can be traced. In addition, when changes are not uploaded within 48 hours after overwriting the data, further overwriting is prevented to ensure that the database contents are constantly updated to the latest data.

3.2.3. Downloading

When an ECU fails and is brought to a service center, the service center downloads the ECU data from the database based on the engine number. The governor characteristics can then be restored to the original state by writing the downloaded data to the new ECU.

3.3. Web Features

The Internet is used when uploading and downloading data, so the database can be accessed from anywhere in the world regardless of time differences. In addition, a dedicated Web site is provided so that user information can also be checked in addition to uploading and downloading data.

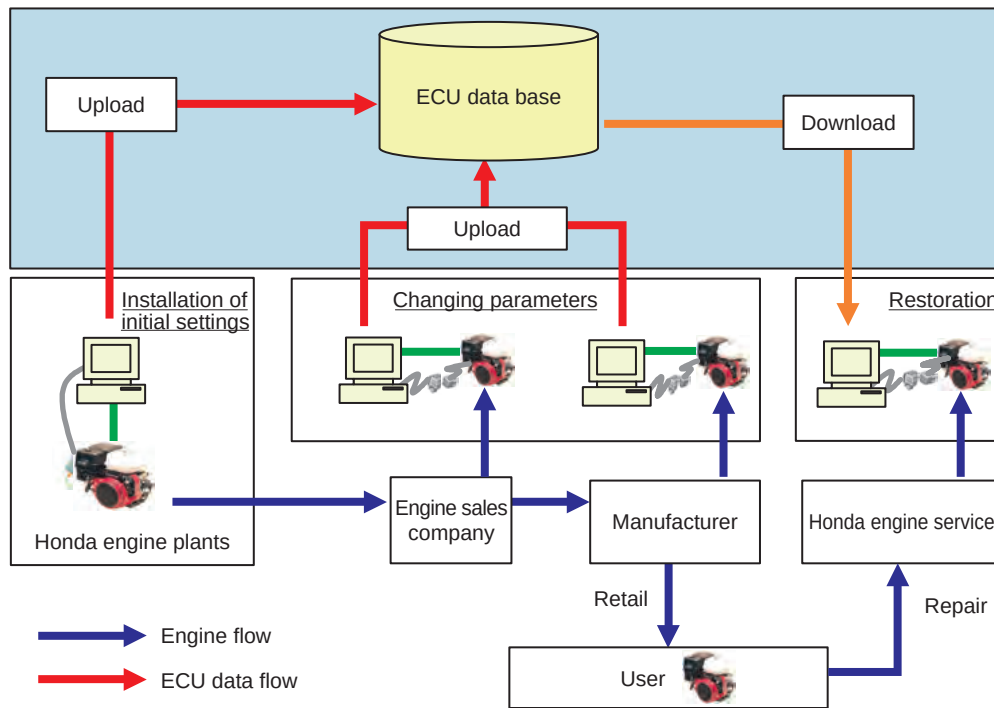


Fig. 2 System diagram

3.4. Database Features

The database server is located in Japan, where the data is centrally managed. Only the changed data is backed up on a daily basis, and all of the data is backed up once per week to prevent data loss. The data storage period is 20 years in consideration of general purpose engine life cycles. Basic security is protected by ID and password entry when accessing the database, and to further ensure safety, the Web operation history of all users also remains in the database so that administrators can understand who performed what operations when.

In addition, the ECU data accumulated in the database can be used to analyze the failure histories of general purpose engines and trends in the usage of work equipment with these engines. This makes it possible to obtain extremely useful data for the development of future general purpose engines.

3.5. ECU Character Management Tool

3.5.1. Interface unit

Personal computers (PC) are used to overwrite the target engine speed and other ECU data. The protocol between the PC and the ECU must be converted to write or overwrite data. Therefore, power products equipped with ECU must use a standardized protocol. This study designed the new interface unit shown in Fig. 3 for communication with external equipment. Design requirements for interface units are as follows.

- (1) The interface unit should be compact
- (2) The interface unit should have shock resistance capable of withstanding falls if dropped
- (3) The interface unit should have a general compatibility that allows connection with an optional PC

Compact size and shock resistance were realized by hot melt molding⁽³⁾, and general compatibility was ensured by using a USB connector to allow connection with almost all PCs currently



Fig. 3 Interface unit

available. In addition, DC 12 V can be supplied to the ECU by connecting a DC jack to the interface unit so that data communication is possible even when a battery is not connected.

3.5.2. Overwriting software

Conventional general purpose engine settings cannot be overwritten during the distribution process, and ECU character management tools also did not exist. Therefore, the ECU overwriting software (hereafter “iGX Creator”) shown in Fig. 4 was created to allow governor characteristic changes and access to the database server. Other functions include real-time monitoring, a failure diagnosis function, and a logging function.

3.6. Overwriting Rules

Illegal access and overwriting are prevented by requiring an encrypted ID and password that are issued according to the respective user privileges. IDs are ranked from higher to lower privileges, and lower privilege IDs are unable to perform higher privilege operations or view higher privilege ECU information. In addition, the IDs of people performing operations are written in the ECU to allow traceability of who performed what processes.

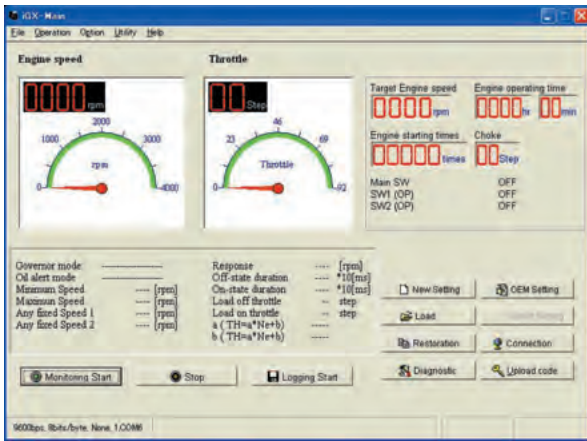


Fig. 4 iGX Creator (Main window)

3.6.1. Five basic principles

Five basic principles for writing and overwriting data to the ECU were established as follows:

- (1) ECU operations must be performed using the distributed tools and software.
- (2) ECU operations are restricted to the range that can be changed by the respective privileges.
- (3) Overwrite operations must be reported immediately.
- (4) ECU operations should be performed only by persons who have received the specified training, and operations should follow rules that are prescribed as necessary.
- (5) Confidentiality contracts or equivalent agreements must be concluded when non-employees are to perform ECU operations.

4. Conclusion

Data management methods and rules were constructed to allow ECU data to be changed and restored by the Sales Department, local subsidiaries, completed unit manufacturers, distributors and others during the distribution process.

- (1) The general purpose engine industry's first system for storing, viewing and updating engine change data in a database was realized, and operation rules were established.
- (2) The ability to change governor characteristics after shipment from the factory made it possible to eliminate unnecessary model types and reduce the lead time to the customer.
- (3) A data assurance system for ECU replacement was established by providing the ability to search for the governor characteristic information needed for restoration from the engine number and restore the data to the original condition.
- (4) Formerly unavailable data resources for future development were realized, such as the ability to analyze trends in general purpose engine usage from data that is accumulated on a daily basis.

References

- (1) Ito, K., Honda, S., Saito, T., Kurata, M., Naoe, G.: Development of Electronically Controlled Multi-purpose Engine iGX440, Honda R&D Technical Review, Vol. 17,

No. 2, p. 44-51

- (2) Fukushima, T., Matsuda, H., Shinogi, Y., Taniguchi, Y.: Governor System for General Purpose Engine Using Adaptive Control Theory, Honda R&D Technical Review, Vol. 17, No. 2, p. 58-64
- (3) Kimata, R., Maekawa, Y., Kuwasako, S., Tamechika, T.: Application of Thermoplastic Resin Molding to Control Unit of General Purpose Engine, Honda R&D Technical Review, Vol. 17, No. 2, p. 52-57

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Development of Automated Manual Transmission System for New CIVIC

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ABSTRACT

An automated manual transmission for the new CIVIC was developed to realize lower fuel consumption, easy driving and sportiness. The light and compact 6-speed transmission was developed by adding a gear change and clutch actuator to a conventional manual transmission, thus making it easier to drive and providing a sporty feel. The system achieved a 3% fuel economy improvement over the base transmission through system optimization.

1. Introduction

Recently in Europe, automated manual transmission vehicles aimed at reducing driving load and enhancing fuel economy have gradually increased their market presence. An automated manual transmission called “i-SHIFT” has been developed by Honda to provide improved fuel economy, easy driving and sportiness for the new model CIVIC⁽¹⁾. The aims of development and technical features of i-SHIFT are described below.

2. Outline of i-SHIFT

The i-SHIFT is an automated transmission system that uses electronic control to perform the starting and gear shift operations of a conventional manual transmission⁽²⁾. A system diagram of the i-SHIFT is shown in Fig. 1.

3. Development Goals

The i-SHIFT is intended to enhance fuel economy and minimize the inconvenience of manual gear changes, while retaining the sporty feeling of a conventional manual transmission. Specifically, the development goals are as follows:

- (1) To enhance the fuel economy to a level at least that of a manual transmission vehicle in EU mode
- (2) To realize a simple, lightweight and compact system configuration

- (3) To realize easy driving through complete automation of starting and gear shifting
- (4) To realize a direct & sporty driving feel by manual gear shifting

4. System Configuration

4.1. Actuators

Electro-hydraulic and electric motor actuator systems are typically used for small vehicles. An electro-hydraulic system can operate under large loads, and with the gear selection actuators hydraulically operated, such a system has the advantage of only needing one electric motor. However, a complicated hydraulic circuit is required, so there are weight and cost issues that must be considered. The electric motor system requires a high operating force due to clutch wear, so a large actuator is typically needed.

Based on the existing manual transmission, this 6-speed automated manual transmission uses three independently operated actuators to electronically control the shifts, gear selection and clutch operations. The clutch actuator performs clutch control by operating the release fork under the hydraulic pressure generated from the electric motor. The motor drive section contains a speed reduction mechanism and uses a self-adjusting clutch (hereafter called SAC), enabling the actuator to be lighter, more compact and less expensive. The actuator has a simple configuration and good power transmission efficiency. Also, it can be easily installed onto the base manual transmission.

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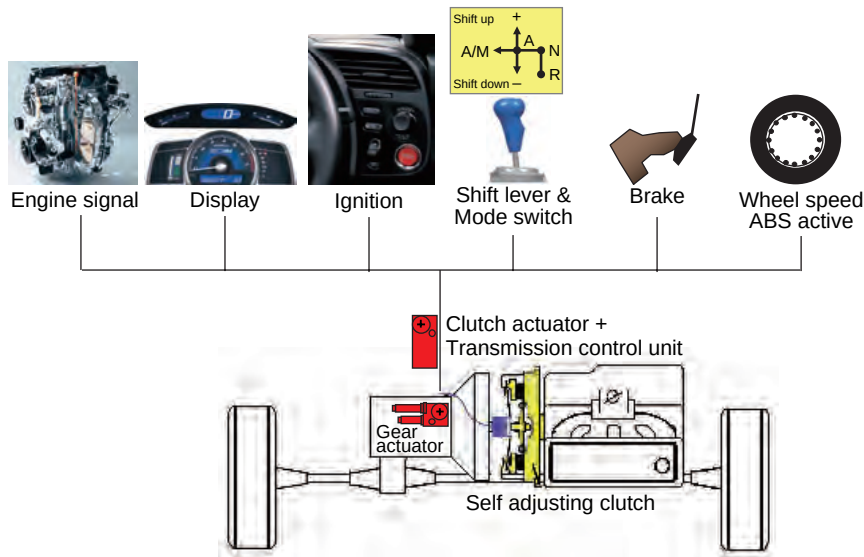


Fig. 1 Configuration of "i-SHIFT"

The gear-change actuator, which performs the shift and selection operations, performs gear changes using the drive force from an electric motor. The actuator is of an integral construction and has mounting compatibility with the chassis of conventional manual transmissions.

4.2. Clutch

In a conventional clutch, the diaphragm spring setting angle changes due to friction material wear. This causes an increase in

the release load which requires a larger actuator. In a SAC, the pivot of the diaphragm spring moves along with the wear surface, and the diaphragm spring setting angle is maintained at a nearly constant angle throughout the life of the clutch. Consequently, the release load does not increase with wear and the clutch operating force remains stable⁽³⁾. By using the SAC to stabilize the clutch operating force, it was possible to use a small electric motor. Figure 2 shows a typical SAC assembly and Fig. 3 shows operating characteristics.

5. Control

When the driver presses the accelerator or brake pedal, the i-SHIFT recognizes the driver's intent from the change in the rotational speed of the four wheels and performs the optimum starting or gear change operation. Figure 4 shows an outline of the system. The gear shift schedule is optimized according to the road gradient, loading situation and driver intent, based on the shift map which takes into account high fuel economy. In addition, a

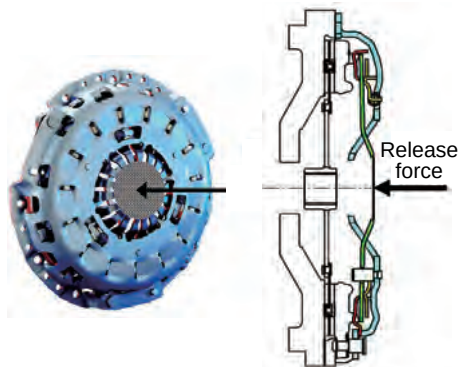


Fig. 2 Construction of Self-adjusting-clutch

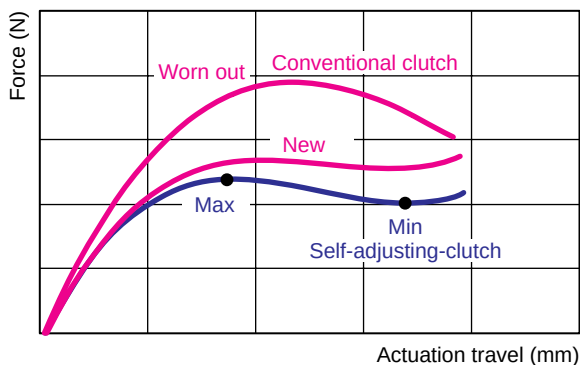


Fig. 3 Component characteristics of Self-adjusting-clutch

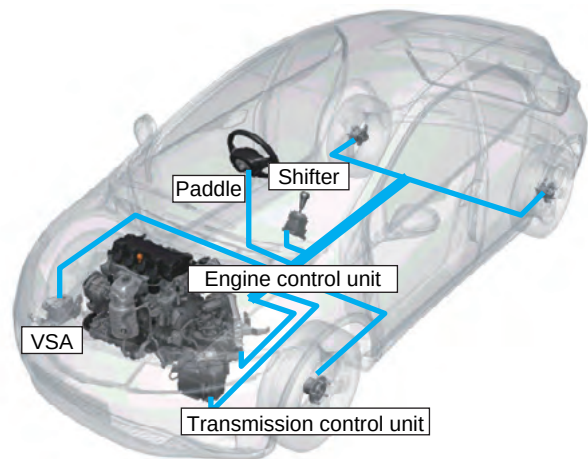


Fig. 4 System overview

control mechanism is implemented that eliminates the discomfort experienced by the driver when cornering at high speeds, abruptly decelerating, or braking (Fig. 5). For example, the driver of a vehicle traveling at high speed slows to enter a corner and then reapplies the accelerator pedal in order to make a fast exit. If normal control is used in this situation, the driver may feel discomfort due to the delay in drive force caused by excessive shifting. As another example, after overtaking another vehicle, the driver slightly releases the accelerator pedal. Under normal control logic, the transmission may up-shift excessively and would require subsequent down-shifts to maintain cruising speed. This can cause a reduction in the distance between the driver's vehicle and the overtaken vehicle. To avoid these kinds of situations and eliminate discomfort, up-shift is limited. Conversely, when braking, control is performed so as to promote an early down-shift, and thereby enhance the response during re-acceleration.

5.1. Starting Control

Regarding the starting control mechanism, when the shift lever is in the neutral position, the change mechanism and the clutch are in neutral. When the shift lever is put in the auto-mode position,

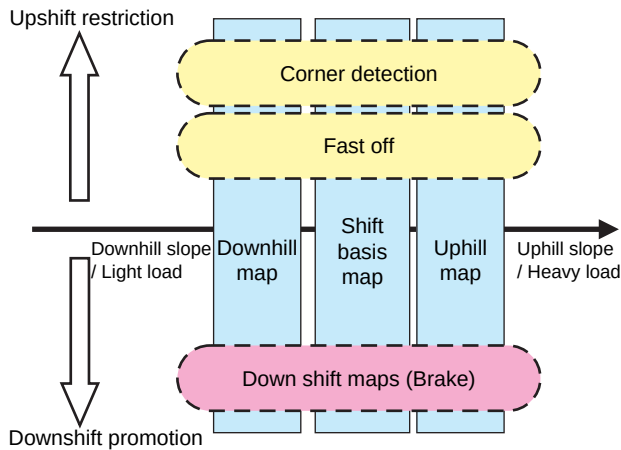


Fig. 5 Shift restriction

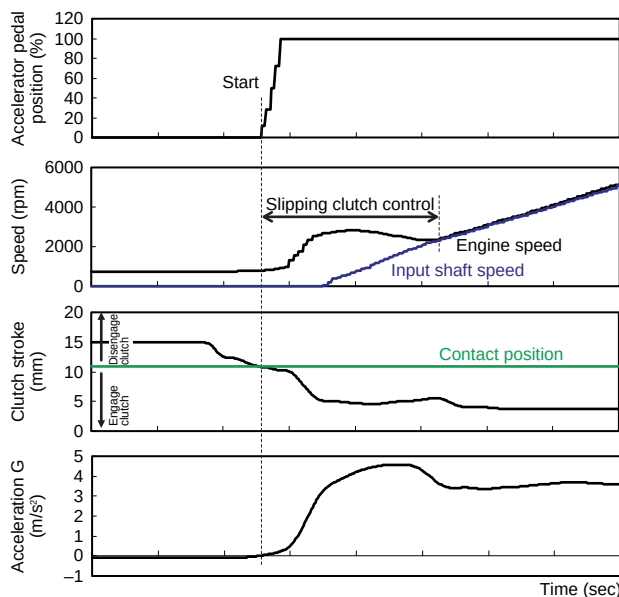


Fig. 6 Example of starting control

the gear enters the start stage, and the clutch stroke moves to its standby position to prepare for starting. When the driver depresses the accelerator pedal, the clutch strokes to the contact position, after which the vehicle starts smoothly under clutch position control. As a result, complicated manipulations when starting the vehicle have been eliminated, and clutch wear has been reduced and evened out. An example of this is shown in Fig. 6.

5.2. Gear Shifting Control

For engine control during shifting, the transmission controller commands the necessary torque to the Electronic Throttle Control System (ETCS). Torque reduction, gear change, speed synchronization and torque recovery are implemented cooperatively with the ETCS. By disengaging the clutch when the engine torque becomes zero, the clutch disengagement shock is reduced. Directly

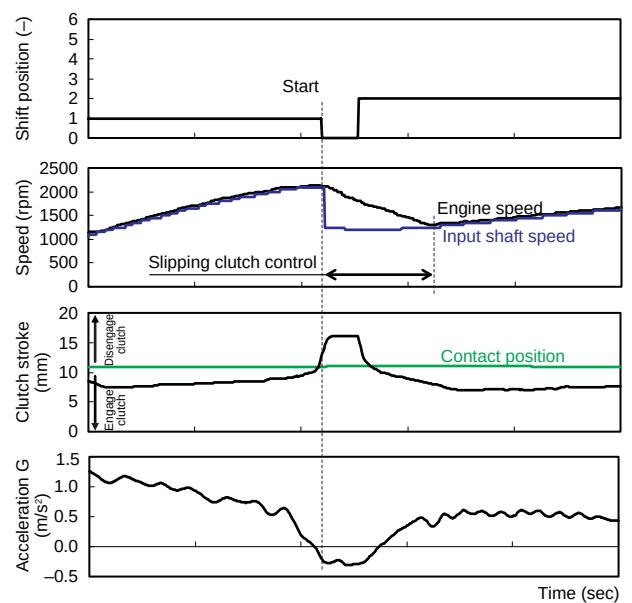


Fig. 7 Example of gear shifting control (upshift)

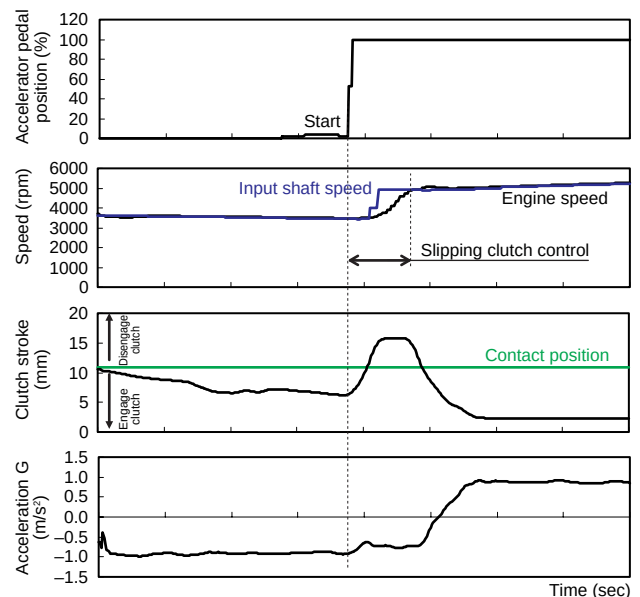


Fig. 8 Example of gear shifting control (downshift)

after clutch disengagement, the gear select, shift operation and speed synchronization take place. Subsequently, the clutch gradually moves in concert with the increasing engine torque - from zero to demand torque - and the gear shift operation is complete. During shifting, the optimum clutch position for good feeling is obtained from the accelerator pedal position, the rotational speed of the engine, and the rotational speed of the transmission. Examples are shown in Figs. 7 and 8.

5.3. Realization of High Fuel Economy

Up-shifting at a lower speed than the base vehicle with manual transmission greatly enhances fuel economy. However, if the engine is run at an excessively low speed, the drive force is insufficient. As a result, the accelerator needs to be pressed further, and the fuel economy deteriorates. Additionally, if the shift is performed in a shorter amount of time, less acceleration is lost, enhancing the fuel economy. However, the associated shift shock is larger. For this reason, the optimum shift schedule and shift time that can simultaneously realize fuel economy and product marketability are set. By optimizing the shift schedule and the shift time, it is possible to improve the fuel economy compared to that of the base transmission by 3%. As a result, top class fuel economy is achieved when compared to vehicles with equal displacement engines (Fig. 9).

5.4. Realizing Easy Driving and Sporty Driving Feel

In auto mode the following characteristics were evaluated: the responsiveness and smoothness at starting, the change in torque corresponding to accelerator pedal movement, and the responsiveness and smoothness of shifting. In manual mode, it was

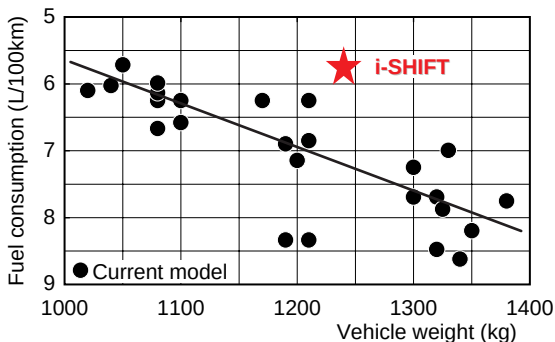


Fig. 9 Relationship between vehicle weight and fuel economy

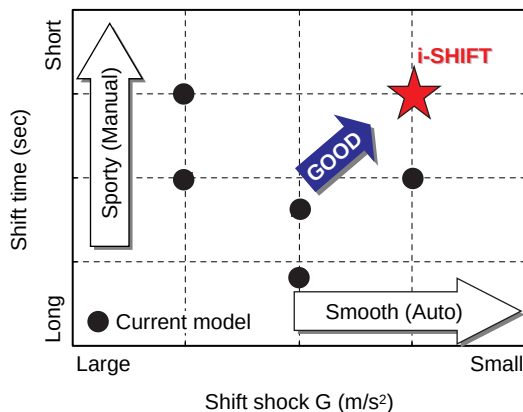


Fig. 10 Easy-to-drive and sporty feeling

confirmed that the direct cruising sensation and sporty driving feel were the same as those in a vehicle with manual transmission. As a result, it was possible to realize easy driving that responds to the driver's intention as well as a sporty driving feel (Fig. 10).

6. Conclusion

The i-SHIFT system was developed, with the following results:

- (1) The fuel economy was 3% better than that of the base manual transmission.
- (2) The use of an electric motor actuator made it possible to create a simple, lightweight and compact system.
- (3) In auto mode, easy driving including smooth starting and gear shifting was realized and the intention of the driver was incorporated into the control logic.
- (4) In manual mode, direct and sporty cruising was realized.

References

- (1) Morita, T., Okamuro, T., Ando, K., Matsumoto, H.: Development of New Model CIVIC Sedan, Honda R&D Technical Review, Vol. 18, No. 1, p. 1-7
- (2) Matsuda, K., Manita, M., Yokomizo, K., Iwasaki, T., Haginiwa, M.: Development of Lightweight and Compact 6-speed Manual Transmission, Honda R&D Technical Review, Vol. 18, No. 1, p. 28-33
- (3) Fujimoto, S., Udou, H., Kitajima, K., Kobayashi, S., Okazaki, K.: Development of a New Manual Transmission for a 3-Liter Class Vehicle, Honda R&D Technical Review, Vol. 14, No. 2, p. 37-44

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Development of Hydraulic Servo Brake System for Hybrid Vehicle

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ABSTRACT

A new brake system, able to make efficient use of regenerative brakes while maintaining excellent brake feel, has been developed to increase the fuel economy of hybrid vehicles. A hydraulic servo was used as a base to enable mechanical operation of the service brakes; solenoid valves and brake fluid pressure sensors were added to this base to make it possible to control brake line pressure as demanded. The use of a stroke simulator in the hydraulic servo prevents brake feel from being affected by the control of the brake pressure.

In addition, high-accuracy brake pressure control that functions cooperatively with the regenerative brakes is enabled -- resulting in stable braking effectiveness. Fitted in a hybrid vehicle, this system has been shown to regenerate 46% more energy than the previous system.

1. Introduction

Hybrid vehicles are attracting increasing attention due to their ability to contribute to the protection of the global environment. In hybrid vehicle systems, the kinetic energy of the vehicle during deceleration is captured as regenerative energy and used to drive the motor, thereby reducing engine workload and increasing fuel economy. Making effective use of regenerative braking and increasing the regenerative energy are therefore important issues. However, if regenerative energy is simply increased, the regenerative brakes will apply more braking force than intended by the driver. In previous brake systems, regenerative brakes have not been used to the limit of their potential so that acceptable brake feel can be maintained. To make effective use of regenerative brakes while achieving brake feel matching the driver's expectations, it is

necessary to use cooperative control of the regenerative and hydraulic brakes to ensure that the total braking force matches the braking force demanded by the driver^{(1),(2)}.

Honda has developed a new brake system that realizes cooperative control of the regenerative and hydraulic brakes, and contributes to increased fuel economy in a hybrid vehicle. This system uses a hydraulic servo to operate the service brakes and a function that enables control of brake line pressure is added. Because the hydraulic servo is therefore important to the functioning of the service brakes, it was also necessary to redesign the service brakes to enhance performance and reliability.

This paper discusses the configuration of the newly developed brake system along with the element technologies used to enhance performance and reliability of the service brakes and realize cooperative control of the regenerative and hydraulic brakes.

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2. Development Concept

Brake-by-wire systems have recently come into use to enable a variety of brake control functions to be realized^{(3),(4)}. These systems allow brake line pressure to be controlled as demanded and also make it possible to realize cooperative control of regenerative and hydraulic brakes. However, because the service brakes are electronically controlled, it is necessary to ensure electrical reliability, and there are numerous concerns related to this area⁽⁵⁾⁻⁽⁷⁾.

The basic concept of the new system was to ensure reliability by using mechanical control of the service brakes. With this purpose, a mechanical hydraulic servo was used to operate the service brakes. And a configuration in which the output pressure of the hydraulic servo is controlled electronically by solenoid valves was selected, providing the ability to handle a variety of control functions.

3. Development Goals

The major elements of the new system are the service brakes and the brake line pressure control. The following development goals were established for each of these elements:

- (1) Service brakes
To realize excellent brake feel and ensure a high level of reliability.
- (2) Brake line pressure control
To develop a cooperative control system for regenerative and hydraulic brakes that would enable the regenerative brakes to be used efficiently while maintaining comfortable brake feel.

The development of a cooperative control system for the regenerative and hydraulic brakes required the following issues to be addressed.

The amount of torque that can be regenerated by a hybrid vehicle motor varies with vehicle speed and the state of charge of the battery. To use the regenerative energy to the limit of its potential while maintaining comfortable brake feel would require accurate control of the braking force to match dynamic changes in torque regeneration, as indicated in Fig. 1.

In standard brake systems, there is a fixed relationship between brake line pressure and pedal stroke. The pedal stroke therefore changes if control is applied to the brake line pressure, and brake feeling is affected. It is therefore necessary to maintain a fixed relationship between pedal force and stroke, even while cooperative control is applied to change the brake line pressure.

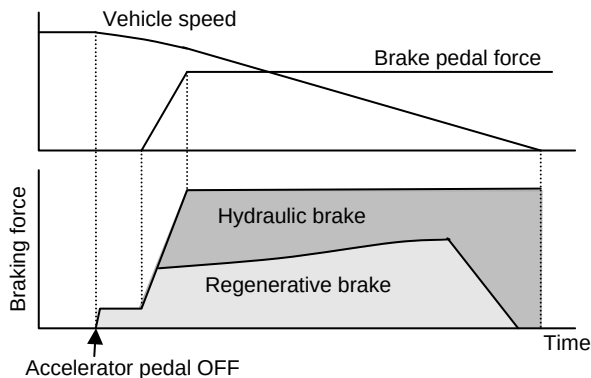


Fig. 1 Cooperative control of regenerative and hydraulic brakes

4. System Outline

The system is made up of the three components shown in Fig. 2.

- (1) Servo unit: Generates brake line pressure in response to brake pedal operation and controls brake line pressure as demanded.
- (2) Power unit: Functions as a power source for the servo unit. Generates high hydraulic pressures using an electronically controlled hydraulic pump; pressure is stored in an accumulator.
- (3) Pedal stroke sensor: Detects the stroke of the brake pedal.

4.1. Servo Unit

Figure 3 shows the configuration of the servo unit. A hydraulic servo section, consisting of a stroke simulator and a regulator valve positioned in a straight line, is connected to a master cylinder to enable mechanical control of the service brakes. Four solenoid valves and three pressure sensors are built into the hydraulic servo section, and an electronic control unit (ECU) is integrated. This enables brake line pressure to be controlled as demanded. The servo unit will be discussed in greater detail beginning in the next section.

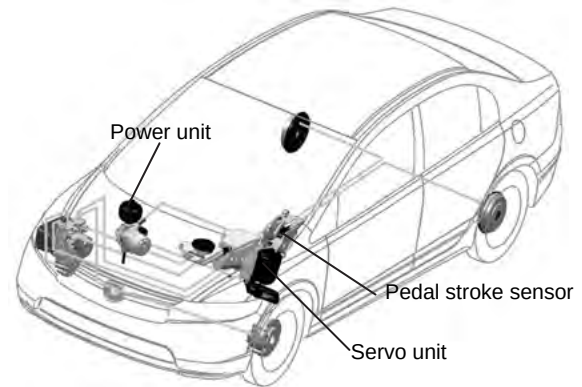


Fig. 2 System configuration

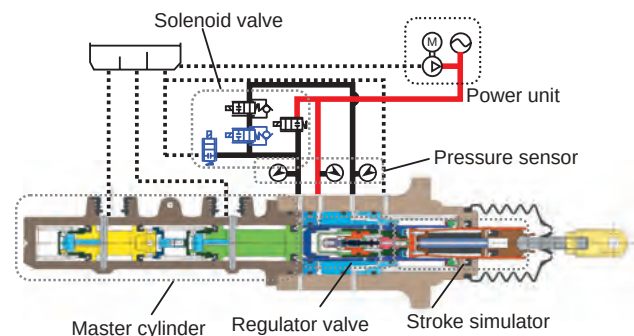


Fig. 3 Structure of servo unit

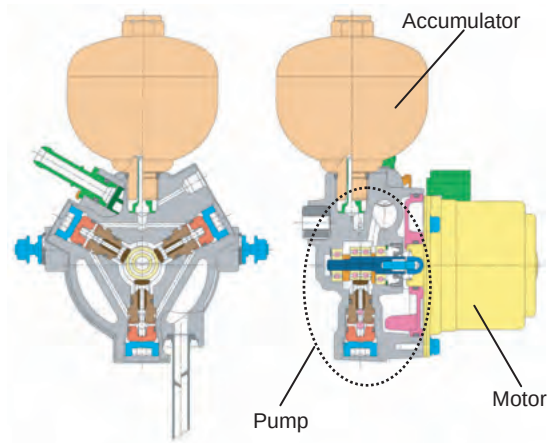


Fig. 4 Structure of power unit

4.2. Power Unit

As shown in Fig. 4, the power unit contains a 3-cylinder radial piston-type hydraulic pump driven by an electric motor. The pressure generated by the hydraulic pump is stored in an accumulator.

The power unit and the servo unit are connected by a pipe. The pressure in the accumulator is supplied to the servo unit; the regulator valves provide a sealing function. A pressure sensor is positioned in the pathways to monitor accumulator pressure in the servo unit. On the basis of the measurements provided by this sensor, the ECU drives the motor so as to maintain accumulator pressure within a specified range.

4.3. Pedal Stroke Sensor

A pedal stroke sensor is used to detect the amount of brake pedal operation by the driver, since the pressure sensors alone are unable to detect a failure in the event of a mechanical breakdown in the servo unit.

The pedal stroke sensor is a rotary sliding resistance-type sensor. It is mounted to the brake pedal bracket, and converts the pedal stroke to rotary motion by means of a link.

5. Performance and Reliability of Service Brakes

The performance and brake feel of the service brakes are dependent on the characteristics of the hydraulic servo. To achieve high reliability, it is necessary to use a design that considers performance during system malfunctions. This section will discuss the element technologies used to enhance the performance and reliability of the service brakes (one of the development goals) and the achieved level of performance.

5.1. Configuration of Hydraulic Servo

The hydraulic servo is composed of a stroke simulator, which generates a stroke in response to pedal input and transmits the input force to the regulator valve, and the regulator valve itself, which is a mechanical servo valve that adjusts the pressure supplied by the power unit proportionally to the pedal input.

The pressure adjusted by the regulator valve (the regulator pressure) is applied to the back wall of the master cylinder (the power chamber), which operates the master cylinder piston, generating brake line pressure.

5.2. Stroke Simulator

As shown in Fig. 5, a rubber element (stroke damper) and a spring are positioned in line inside the stroke simulator. Pedal input causes the spring elements to be compressed in sequence, generating a pedal stroke. The relationship between pedal force and pedal stroke is determined by the characteristics of these spring elements and is not affected by the brake line pressure. Pedal feel therefore remains constant, regardless of the control being applied to the brake line pressure.

The internal spring is used to ensure a loss stroke in initial input, therefore the displacement of this spring is small. The pedal stroke is determined by the amount of compression of the stroke damper; the characteristics of the rubber used are therefore important elements in the brake feel achieved.

The hardness and the level of hysteresis of the rubber, which determine its basic characteristics, were adjusted to achieve excellent brake feel. To ensure stable brake feel, a rubber material was selected that displays little change in hardness with changes in temperature.

In standard brake systems, the relationship between pedal force and pedal stroke is nonlinear; the amount of change of the stroke becomes lower as the pedal force increases. Because the spring characteristic of the rubber is basically linear, this causes brake feel to decline. The cylinder (control piston) into which the stroke damper is inserted is therefore tapered. When the stroke increases,

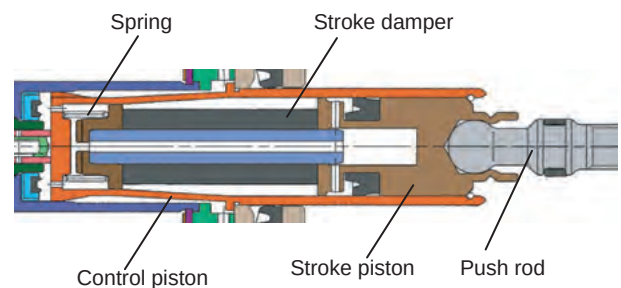


Fig. 5 Cross-section of stroke simulator

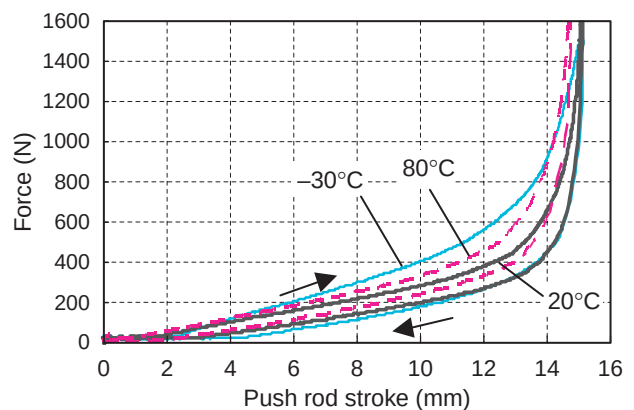


Fig. 6 Characteristic of stroke simulator

the end of the stroke damper fills the cylinder progressively, providing a nonlinear spring characteristic.

Figure 6 shows the characteristic of the stroke simulator at a range of temperatures. These results demonstrate that the relationship between stroke and input force has a low level of temperature dependence and that the tapered shape has resulted in the achievement of a nonlinear characteristic.

5.3. Regulator Valve

The regulator valve, shown in Fig. 7, is composed of an inlet valve that introduces the pressure from the accumulator into the regulator chamber when force is applied to the brake pedal and an outlet valve that releases pressure from the regulator chamber when the brake pedal force is reduced. When the brakes are not in operation, the inlet valve is sealed to maintain the pressure in the accumulator while the outlet valve remains open to release pressure from the regulator. When the brakes are operated, the input force on the pedal is transmitted to a reaction piston by the stroke simulator. The reaction piston moves forward, closing the outlet valve and opening the inlet valve. The pressure introduced through the inlet valve produces reaction force to the reaction piston; when the reaction piston reaches a position at which the pedal input force and the reaction force are in equilibrium, the inlet and outlet valves are closed, determining the regulator pressure. When the brakes are released, the reaction piston moves back until the outlet valve opens, and the regulator pressure is relieved.

Because an input force is required to oppose the accumulator pressure when the inlet valve is open, if the diameter of the inlet valve seal is too large, the threshold force at which brake line pressure begins to be produced is also high, and the brakes can appear not to be functioning. On the other hand, if the seal is small, the channel that introduces accumulator pressure is restricted, and pressure rises relatively slowly during panic stop compared to the normal brake system. A two-step seal system was used to address this issue. During the initial input, the low-diameter seal opens under a low load, resulting in a reduced threshold force. When the input is quick and there is a delay in the response of the regulator pressure, the shaft stroke increases to open the large-diameter seal, ensuring high output pressure response.

Figure 8 shows the relationship between pedal force and output pressure and Fig. 9 compares the pressure rise response at quick pedal input in the newly developed and previous systems. The two-step seal system ensures that the pedal force at which output pressure begins to be generated is the same as it was for the vacuum servo system. In addition, it enables an enhanced pressure rise response compared to that of the vacuum servo system.

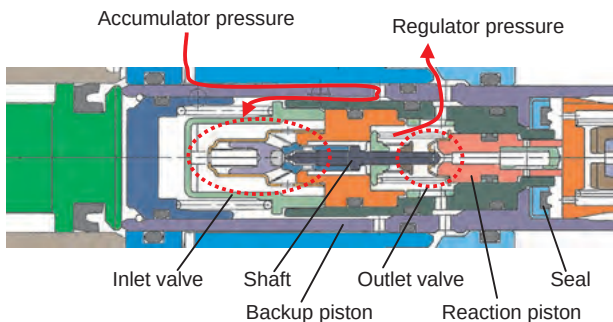


Fig. 7 Cross-section of regulator valve

5.4. Performance during Power Unit Malfunction

The hydraulic servo ceases to function when accumulator pressure is lost due to a malfunction. In the event of a malfunction, the pedal input is therefore transmitted directly to the master cylinder via the stroke simulator and a backup piston. This force is used to operate the master cylinder with no servo, enabling the brakes to function as effectively as the brakes in a vacuum servo system during a malfunction.

Under these circumstances, when the stroke simulator generates a stroke, the pedal stroke is longer than that produced by a vacuum servo. To control the stroke generated by the stroke simulator, a seal is positioned on the backup piston where it makes contact with the simulator. This seal blocks the channel for escape of brake fluid from the interior of the stroke simulator. The interior of the simulator fills with brake fluid, resulting in a restricted stroke.

Figure 10 shows the pedal stroke characteristic during a malfunction of the power unit. Restriction of the simulator stroke has ensured that the pedal stroke maintains the same level as the stroke in a vacuum servo system.

5.5. Reliability of Electrical System and Configuration of ECU

Because loss of power unit function would mean a decline in braking performance in a hydraulic servo system, the reliability of the drive circuits of the power unit motor was an important issue in the design of the ECU. In the newly developed system, the power

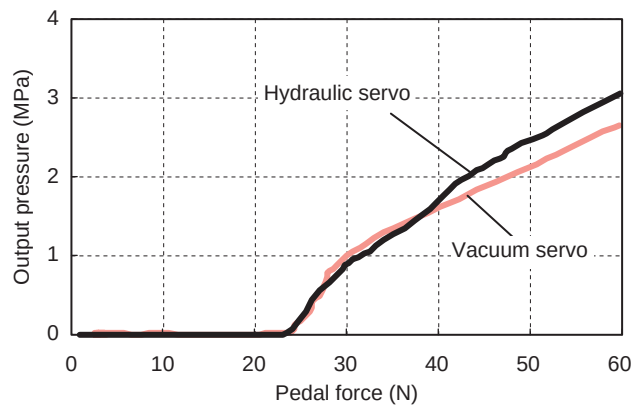


Fig. 8 Input and output characteristics

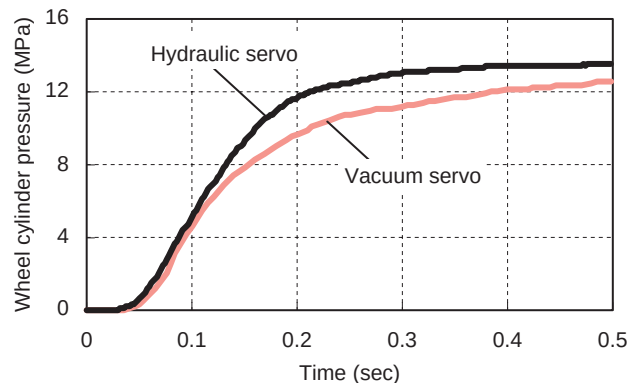


Fig. 9 Wheel cylinder pressure response at quick pedal input

source, CPU and power elements of the motor drive circuits are all made redundant. Motor function is therefore maintained in the event of a single failure, ensuring a high level of reliability.

In addition, the pressure sensor that monitors accumulator pressure is a two-output type, increasing the ability to detect a malfunction.

A fail-safe FET has been used in the power supply line for the solenoid valves to enable the power supply to be stopped in the event of a malfunction. The service brakes can be operated mechanically, and therefore brake performance is not lost even if the solenoid valve stops functioning. Figure 11 shows the configuration of the ECU hardware.

6. Control of Brake Line Pressure

To control brake pressure, a target value for pressure in the power chamber is determined on the basis of the requirements of each of the control functions; the solenoid valves are operated to produce the target brake pressure based on feedback from the fluid pressure sensors in the power chamber.

As indicated above, the use of a stroke simulator has ensured that the relationship between pedal force and stroke is not affected by control of brake pressure. Therefore, excellent brake feel can be achieved if brake pressure control is able to accurately trace the target pressure. Sliding mode control, known to possess excellent robustness⁽⁸⁾, is used in the newly developed system to ensure accurate brake pressure control.

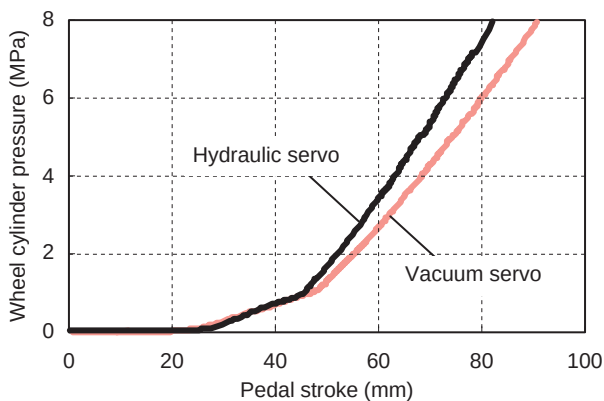


Fig. 10 Pedal stroke characteristics at power unit malfunction

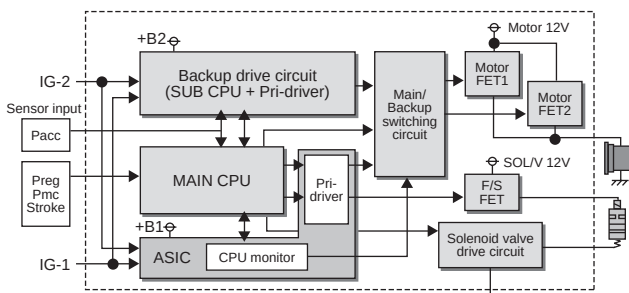


Fig. 11 ECU hardware block diagram

6.1. Solenoid Valve

Cooperative control of regenerative and hydraulic braking requires the application of control to reduce the pressure in the power chamber against the pressure in the regulator, which represents the force required for the operation of the brakes by the driver. On the other hand, control must sometimes be applied to increase pressure in order to generate brake line pressure without relation to the brake pedal operation or in excess of the brake pedal operation. To enable these different pressure control requirements to be realized, two solenoid valves for pressure decrease and two solenoid valves for pressure increase have been positioned between the regulator chamber and the power chamber in the servo unit. Pressure increases and pressure decreases control both function by isolating the power chamber from the regulator valve output and controlling the pressure in the power chamber.

Control is applied to two solenoid valves when using cooperative control of the regenerative and hydraulic brakes: a normal closed valve (NC/V) for pressure decrease that releases pressure from the power chamber into the reservoir and a normal open valve (NO/V) for pressure decrease that isolates the power chamber from the regulator output to maintain its state of reduced pressure. When applying control to increase the brake pressure in excess of the brake pedal operation, control is applied to another two solenoid valves: an NC/V for pressure increase, which introduces pressure stored in the accumulator into the power chamber, and an NO/V for pressure increase that isolates the power chamber from the regulator output to maintain its state of increased pressure. Figure 12 shows the layout of these solenoid valves. All of these solenoid valves are able to be used for linear pressure control.

6.2. Sliding Mode Control

Figure 13 shows an outline of sliding mode control. The present difference between control target pressure and control pressure, $e(n)$, and the previous difference between these parameters, $e(n-1)$, are used as state variables. A function termed the switching line is defined on the phase plane between these variables used as coordinate axes as

$$\sigma(n) = e(n) + s \cdot e(n-1) = 0 \quad (1)$$

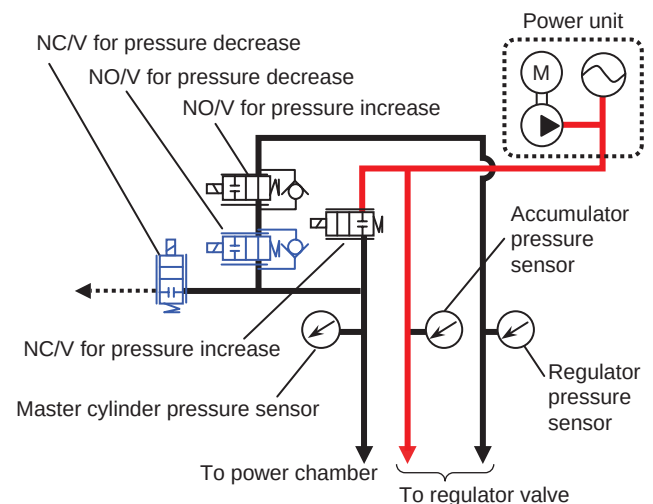


Fig. 12 Layout of solenoid valve

Control is applied to constrain the status level, representing the function of the state variables, to the switching line. If the gradient of the switching line, s , is set between 0 and -1 , the status level is constrained to the origin point of the phase plane, i.e. where the difference between target pressure and pressure = 0, and the control pressure matches target pressure.

The sum of three inputs is used as the control input to constrain the status level to the switching line; $U_{eq}(n)$, the equivalent control input enabling the status level to be constrained to the switching line; $U_{rch}(n)$, the reaching law input enabling the status level to be reached on the switching line; and $U_{adp}(n)$, the adaptive law input that controls modeling error and external disturbances to enable the status level to be maintained on the switching line⁽⁹⁾. These control inputs are determined as discussed below.

A discrete-time first-order lag model is used to define the pressure response of the solenoid valves that are subject to control.

$$P_{mc}(n+1) = a \cdot P_{mc}(n) + (1-a) \cdot U(n) \quad (2)$$

$P_{mc}(n)$: Control pressure

$U(n)$: Control input

a : Model parameter

Terming the control target pressure $P_{ref}(n)$, the difference $e(n)$ and the switching function $\sigma(n)$ are defined by the following equations:

$$e(n) = P_{ref}(n) - P_{mc}(n) \quad (3)$$

$$\sigma(n) = e(n) + s \cdot e(n-1) \quad (4)$$

Given that the condition for continued constraint of the status level to the switching line is

$$\sigma(n+1) = \sigma(n) \quad (5)$$

the equivalent control input $U_{eq}(n)$ can be determined by solving Equations (2) – (5) for $U(n)$.

$$U_{eq}(n) = \{P_{ref}(n+1) - a \cdot P_{mc}(n) - (1-s) \cdot e(n) - s \cdot e(n-1)\} / (1-a) \quad (6)$$

The reaching law input $U_{rch}(n)$ makes the status level reach the switching line, i.e. the input at which $\sigma(n) = 0$, and is a proportional element.

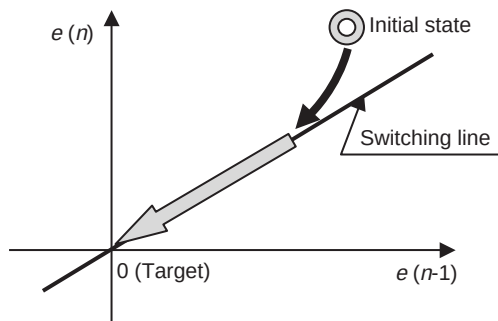


Fig. 13 Sliding mode control outline

$$U_{rch}(n) = F \cdot \sigma(n) \quad (7)$$

F : Feedback gain of reaching law

To reduce steady-state error, the adaptive law input $U_{adp}(n)$ is defined as an integral element of $\sigma(n)$ as follows:

$$U_{adp}(n) = G \cdot \sum_{i=0}^n \sigma(i) \quad (8)$$

G : Feedback gain of adaptive law

The control input $U_{in}(n)$ is the sum of the three inputs discussed above:

$$U_{in}(n) = U_{eq}(n) + U_{rch}(n) + U_{adp}(n) \quad (9)$$

6.3. Cooperative Control of Regenerative and Hydraulic Brakes

The distribution of braking force between regenerative and hydraulic brakes is determined so as to enable the regenerative brakes to be utilized as much as possible up to the limit of capacity for torque regeneration by the hybrid motor under a range of driving conditions. The control target for brake line pressure is determined by subtracting the pressure corresponding to regenerated torque from the figure provided by the fluid pressure sensor in the regulator chamber. The regulator pressure corresponds to the braking force required by the driver.

Brake line pressure control isolates the output from the regulator via the NO/V for pressure decrease, and decreases the pressure in the power chamber via the NC/V for pressure decrease. When the amount of regenerated torque decreases, pressure from the regulator is introduced to the power chamber via the NO/V for pressure decrease, thus increasing power chamber pressure. If this power chamber pressure control does not have sufficient ability to achieve the target pressure, the braking force required by the driver will not be produced.

Sliding mode control is applied to the NC/V when the pressure in the power chamber is reduced and to the NO/V when the pressure in the power chamber is increased. Fine-tuning of the control parameters for each solenoid valve has resulted in the achievement of highly responsive and stable pressure control, even under conditions such as temperature changes, which could cause modeling errors. The application of sliding mode control has ensured comfortable brake feel under a variety of driving conditions.

Figure 14 shows braking pressure during cooperative control of regenerative and hydraulic brakes. The control target pressure, representing pressure corresponding to regenerated torque

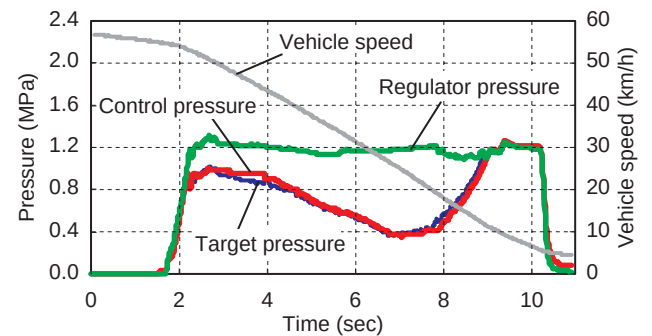


Fig. 14 Braking pressure during cooperative control

subtracted from regulator pressure, increases and decreases to match changes in the amount of torque that is able to be regenerated. The power chamber pressure, i.e. the control pressure, is able to trace this target pressure accurately.

6.4. Linear Control of Pressure Increase

Linear control of pressure increase via the two solenoid valves discussed above (the NC/V for pressure increase and the NO/V for pressure increase) has been enabled to respond to the requirement to generate braking line pressure in excess of the braking pressure represented by driver pedal input. When increased pressure is required, the NO/V for pressure increase isolates the output from the regulator, and the NC/V for pressure increase introduces pressure from the accumulator into the power chamber. To reduce the pressure in the power chamber, the NO/V for pressure increase is opened, and pressure is released into the regulator chamber.

As is the case for cooperative control of the brakes, sliding mode control is used for pressure control, ensuring excellent control performance.

7. Improved Fuel Economy

Fitted in a hybrid vehicle, the new system results in increased fuel economy in US LA-4 city mode. In the previous system, which was not provided with cooperative control, it was necessary to limit torque regeneration to maintain brake feel, and the ability of the hybrid motor to regenerate torque could not be utilized to its full capacity. The use of the newly developed system has enabled more efficient capture of regenerative energy, and the test vehicle regenerated approximately 46% more energy than the previous system during the test. Using this energy to drive the motor reduces the workload on the engine, resulting in a contribution to increased fuel economy.

8. Conclusion

A highly reliable hydraulic servo system able to ensure excellent brake feel has been used as a base in the development of a brake system that realizes a variety of brake control functions, including cooperative control of regenerative and hydraulic brakes. The main element technologies are as follows:

- (1) Fine-tuning of the characteristics of the stroke simulator and the use of a two-step inlet valve in the regulator resulted in the achievement of excellent brake feel.
- (2) High reliability has been ensured by making the power unit motor drive circuits redundant and using two-output-type sensors.
- (3) Highly accurate brake line pressure control has been achieved by applying sliding mode control to feedback control using solenoid valves.
- (4) Cooperative control of regenerative and hydraulic brakes has increased energy regeneration by 46%, resulting in a contribution to increased fuel economy.

The system discussed in this paper was developed to respond to the requirements of hybrid vehicles, but the control of brake line pressure as demanded that has been achieved can also respond to requirements for a variety of other braking functions, which will increase in the future. The potential of this system is not limited to hybrid vehicles; it can also contribute to enhanced brake performance and an increased range of brake function in other vehicle types.

References

- (1) Ogura, M., Aoki, Y., Mathison, S.: The HONDA EV PLUS Regenerative Braking System, EVS14 (1997)
- (2) Otomo, A., Sakai, A., Takasu, T., Nakamura, K.: Development of Regenerative Braking System for Hybrid Vehicle, SAE Paper, No. 1999-10-0062
- (3) Jonner, W., Winner, H., Dreilich, L., Schunck, E.: Electrohydraulic brake system -- The first approach to brake-by-wire technology, SAE Paper, No. 960991
- (4) Nakamura, E., Soga, M., Sakai, A., Otomo, A., Kobayashi, T.: Development of Electronically Controlled Brake System for Hybrid Vehicle, SAE Paper, No. 2002-01-0300
- (5) Reuter, D. F., Lloyd, E. W., Zehnder II, J. W., Elliott, J. A.: Hydraulic Design Considerations for EHB System, SAE Paper, No. 2003-01-0324
- (6) Song, W., Zheng, Y., Milot, D.: Pressure Control Calibration Time Reduction for EHB Systems, SAE Paper, No. 2004-01-0257
- (7) Lii, N., Sturm, S., Coombs, T. A.: A Study on Sensor Fusion for Fault Tolerant Brake-by-Wire Driver Input Design with Dissimilar Sensors, SAE Paper, No. 2006-01-1025
- (8) Nonami, K., Den, H.: Sliding Mode Control, Corona Publishing Co., Ltd., 263p. (1994)
- (9) Yasui, Y., Akazaki, S., Ueno, M., Iwaki, Y.: Adaptive Sliding Mode Control for Secondary O2 Feedback, HONDA R&D Technical Review, Vol. 11, No. 1, p. 65-71

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Development of Electric System for Four-stroke Engine Trials Motorcycle “RTL250F”

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ABSTRACT

In order to reduce vehicle mass, a kick-start system without battery was selected for a trials competition motorcycle with four-stroke engine. The integrated throttle body developed for small motorcycles contains the ECU (Engine Control Unit), which is the main component of the fuel injection system. The power used to start the fuel injection control system is generated by the AC generator when the kick-starter is operated. A new control system was adopted for trials motorcycle, taking into consideration drivability requirements for slight and rapid throttle operation. This control system performs calculations twice for each combustion stroke and has a logic to control the fuel injection quantity and timing. Therefore, the above control system adds many enhancements to the drivability of trials motorcycle. Furthermore, users can set data inside the ECU as required.

1. Introduction

The concern of environmental issues has grown; accordingly, emissions regulations for motorcycles have become increasingly severe. To reduce emissions from commercial motorcycles, two-stroke engines are giving way to four-stroke engines and electronically controlled fuel injection systems are being increasingly used instead of carburetors. Even in the motorcycle race world, emissions regulations are being tightened, and the use of two-stroke engines tends to be prohibited by race regulations in each category. In world trials championships, the use of motorcycles

fitted with two-stroke engines was scheduled to be completely banned from 2006 due to changes in the FIM regulations (subsequently, the regulations were amended to allow two-stroke engines to be used until the end of 2008). The RTL250F and RTL250F-W were fitted with four-stroke engines from the 2005 model, prior to the amendment of the regulations. To realize the same drivability as that achieved with a two-stroke engine, an electronically controlled fuel injection system (PGM-FI system)⁽¹⁾ was adopted. This paper describes the control of the fuel injection system, which functions as part of a battery-less system.

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Fig. 1 External view of RTL250F

2. Outline of Functions

2.1. Outline of the RTL250F

Figure 1 shows an external view of the RTL250F⁽²⁾. To ensure that the RTL250F has the same competitive performance as the two-stroke RTL250R, the various parts of the machine were made more lightweight, the center of gravity was lowered, and the mass of the machine was concentrated.

The engine was based on that of the commercial Motocrosser CRF250R, but changes were made to several parts to lower the center of gravity of the machine and make it more compact. The maximum output of the engine was controlled to 14 kW against 31 kW for the CRF250R. Also, the torque characteristics were made flat and priority was given to increasing the torque at low speeds. Because power output was increased in the range of partial throttle opening at low engine speeds, even though the full throttle torque was less than that of a two-stroke engine, a torque feeling that was at least the same as that of the two-stroke engine was obtained.

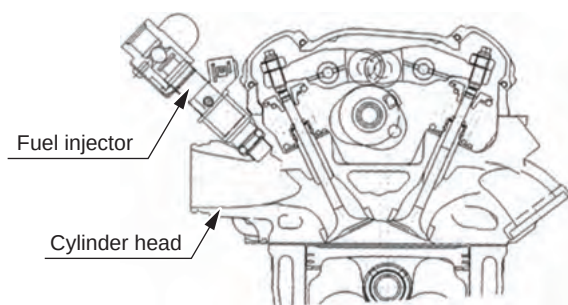


Fig. 2 Fuel injector directly installed in cylinder head

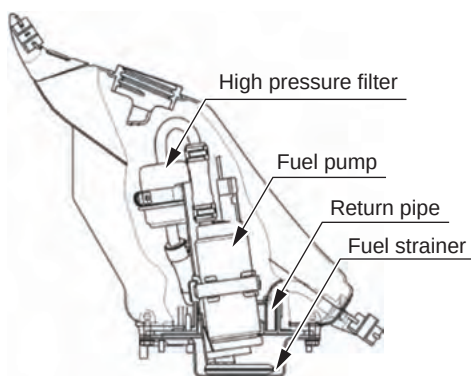


Fig. 3 Fuel tank assembly

To raise the throttle response of the PGM-FI system, the injector was mounted directly on the cylinder head (Fig. 2), while to ensure a stable supply fuel even during intense vehicle body motion when running in trials, the fuel pump was mounted inside the fuel tank (Fig. 3). The throttle body, which was based on the ECU-integrated type for small motorcycles, was changed to an air adjustment screw type that permits the user to adjust the idling speed. In addition, a decompression cam was used and the kick ratio was modified to realize smooth start-ability.

In contrast to the Japanese model of the RTL250F, which is intended exclusively for competitive use, the European model can run on public roads. The European model conforms to Euro-2 emissions regulations through use of a four-stroke engine and a PGM-FI system.

2.2. Outline of the RTL250F-W

The RTL250F-W was developed based on the RTL250F, with the goal of winning the World Championship for Trial. Although the basic design of the engine and the body is the same as that of the RTL250F, the materials used for many parts have been changed to reduce mass, the engine specifications have been changed to provide enhanced torque at low engine speed, and the road grip performance on slippery roads has been enhanced. Like the RTL250F, the ECU is integrated with the throttle body, and additionally has a built-in data logger based on the ECU for the RC211V four-stroke road racer. The built-in data logger enables the operating condition of the engine to be observed as data, and also enables the ECU to be set in a short period of time. The RTL250F-W displayed high competitive performance during the 2005 World Championship for Trial, and took both second and third place.

3. Battery-less System

The RTL250F electrical system consists of an AC generator, pulse generator, regulator rectifier, capacitor, fuel pump, fan motor, fan motor relay, bank angle sensor, water temperature sensor, ignition coil, fuel injector, ECU, and engine stop switch. The ECU contains an intake negative pressure (PB) sensor, air intake temperature (Ta) sensor, and throttle position sensor (TPS). The electrical system of the European model also includes a headlight, tail and stop lights, turn signal lights, a speedometer, horn, and light switches as safety equipment to enable the motorcycle to be ridden on public roads. The circuit diagram is shown in Fig. 4.

Normally, an engine that has a fuel injection control system is started by operation of the ECU, fuel pump, injector, and ignition coil using electrical energy stored in the battery. However, in the case of a battery-less system, these operations must be performed using electrical power generated by the AC generator when the crankshaft of engine is rotated by kick-starting. The AC generator for the RTL250F has a diameter of 147 mm, which is the largest generator used on a 250 cm³ displacement motorcycle. This is to ensure that the generator provides the large moment of inertia demanded by a trials motorcycle. A 3-phase generator is used to provide adequate electrical power at low engine speeds. It produces an output current of 7.5 A at a rotational speed of 1 000 r/min (Fig. 5). The electrical power generated by the AC generator is rectified and regulated to 14.4 VDC by means of a three-phase full-wave commutator type regulator rectifier, smoothed by a capacitor, and then supplied to each electrical load. The regulator rectifier and the capacitor are mounted on a bracket above the crankcase near the AC generator in order to reduce electric power

loss. The fan motor relay and the bank angle sensor are also mounted on this bracket, thus helping to concentrate the mass (Fig. 6).

To permit kick-starting using a battery-less system, the ECU controls the flow of current so that when the machine is being kick-started, electrical power is supplied only to the ECU, fuel pump, injector and ignition coil, and not to the fan motor or other electrical loads, such as the safety equipment in the case of the European model (Fig. 7). After the engine has started and has been allowed

to idle for a certain period, electric power is supplied to the fan motor and other electrical loads such as the safety equipment (Fig. 8).

By using an AC generator that generates a high output at low engine speeds, as well as electrical load control by the ECU and a decompression cam, the engine can start within three cranking operations, as shown in Fig. 9.

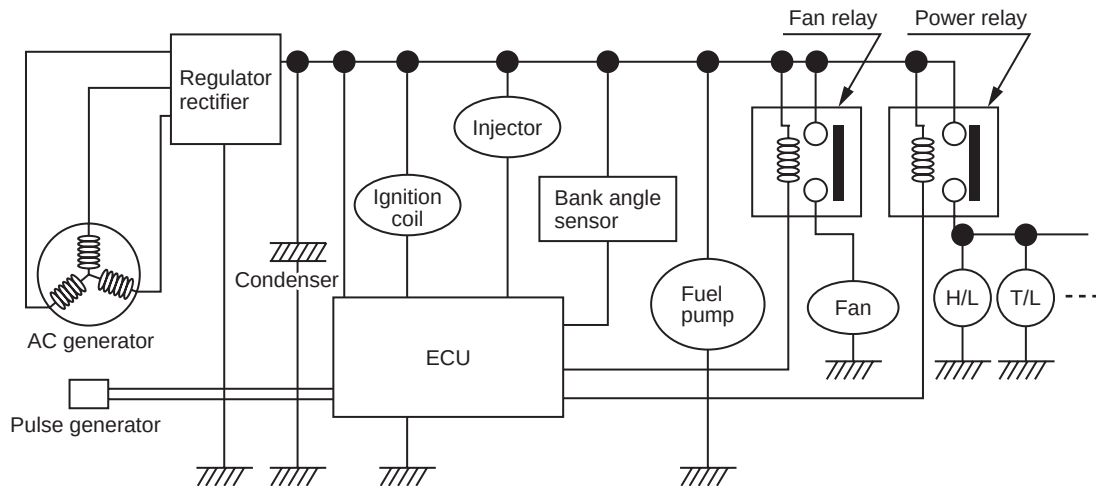


Fig. 4 Circuit of electric system

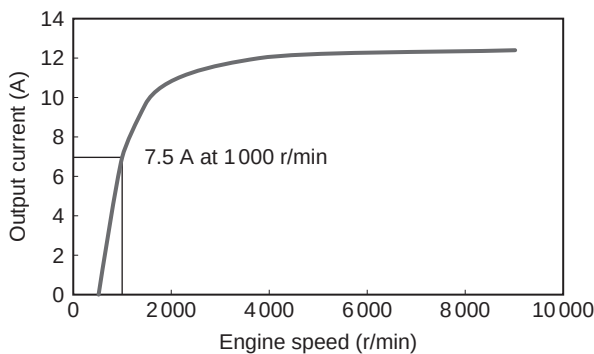


Fig. 5 AC generator characteristics

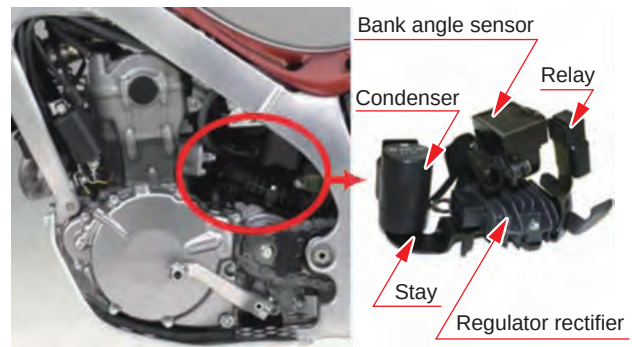


Fig. 6 External view of electrical equipments

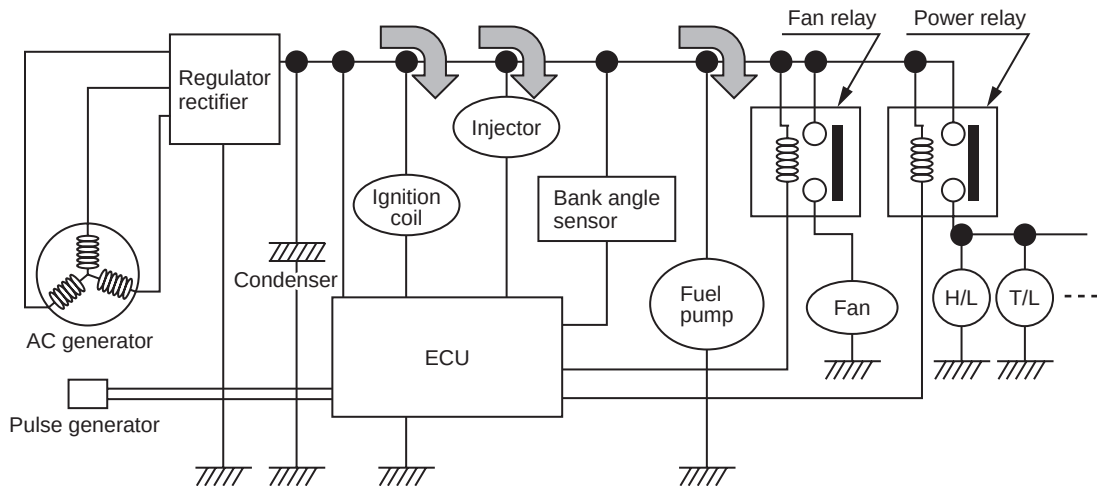


Fig. 7 Electric current mode at kick starting

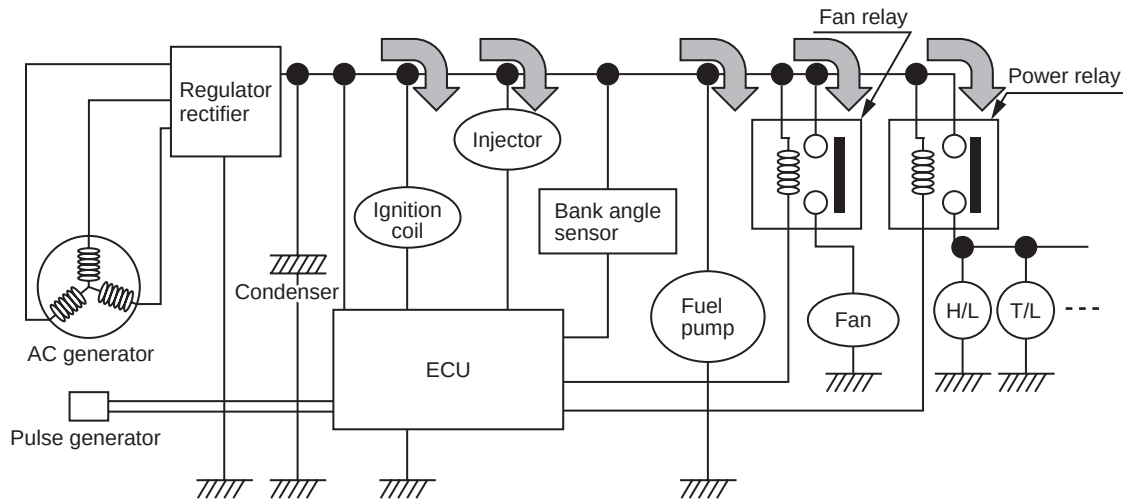


Fig. 8 Electric current mode after engine started

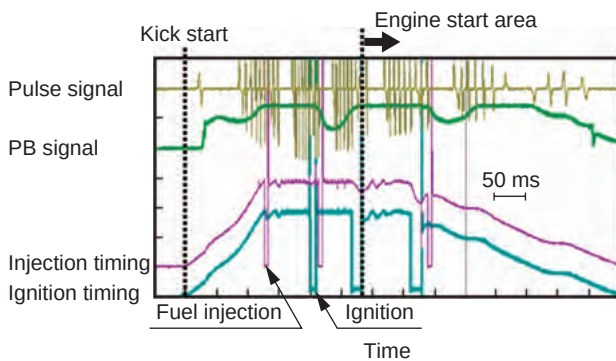


Fig. 9 Measurement data of engine startability at kick start

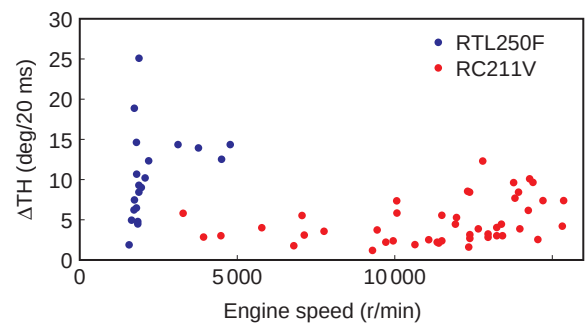


Fig. 10 Comparison data of throttle opening speed

4. PGM-FI System

The PGM-FI system determines the optimum fuel injection quantity, fuel injection timing, and ignition timing based on the following data: the crank angle input to the ECU, engine speed calculated from the crank angle, throttle opening, negative pressure of the intake pipe, intake air temperature, and cooling water temperature. It then operates the injector and energizes the ignition coil. In contrast to the use of a carburetor, fuel injection control enables the air-fuel ratio to be perfectly controlled under various conditions. In the case of the RTL250F, a test was also carried out on a carburetor specification engine. The air-fuel ratio changed remarkably when the body was subjected to violent motion during trials running. The engine stall occurred when the throttle was rapidly opened at low engine speeds, or when the motorcycle was braking downhill. Also, when the throttle was rapidly opened, engine hesitation occurred due to a delay in following the negative pressure of the intake pipe. These phenomena were solved as a result of the adoption of the PGM-FI system and also modification of the layout of the fuel pump.

In trials competitions, particular importance is placed on the response to rapid opening of the throttle at low engine speeds. The data concerning the throttle opening per unit time (ΔTH) of a trials competition and one lap of a circuit run using the RC211V four-stroke road racer were compared. As a result, it can be seen that in a trials competition the throttle is frequently opened rapidly at low engine speeds (Fig. 10). For this reason, in the case of the RTL250F, the throttle response to rapid opening was enhanced by calibrating the fuel injection twice for each 720 degrees of

crankshaft rotation angle.

In the case of conventional fuel injection control, calibrating was performed once at the timing to be optimum combustion during one 720-degree cycle of the crankshaft, consisting of the intake, compression, combustion, and exhaust strokes, and fuel injection was performed immediately after calibration. With this method, it is possible to adequately control the air-fuel ratio by setting the calibration timing and fuel injection amount for each engine speed and each throttle opening during constant speed running. However, during transient operation, particularly at low engine speeds with long stroke time, the fuel injection quantity is not calculated when the throttle is operated at the end of calibration. As a result, lower throttle response is induced at rapid throttle opening and smolder of spark plug is induced at rapid throttle closing. Figure 11 shows the timing chart at rapid throttle opening in the case of conventional calibration, and Fig. 12 shows the timing chart at rapid throttle closing. With the new control method, the number of calibrations of the fuel injection quantity during one cycle was increased to two. Fuel injection is timed to start so that fuel injection ends during the intake stroke after the end of the first calibration that begins during the exhaust stroke. In addition, a method of correcting the fuel injection amount by performing the second calibration during the intake stroke while fuel injection is beginning (hereinafter referred to as "2-calibration injection control") was devised. As a result of this method, it was possible to optimize not only the air-fuel ratio during constant engine speed, but also the fuel injection amount and the fuel injection timing when the throttle is opened or closed rapidly at low engine speeds. Figure 13 shows the timing chart for when the throttle is opened rapidly during 2-calibration

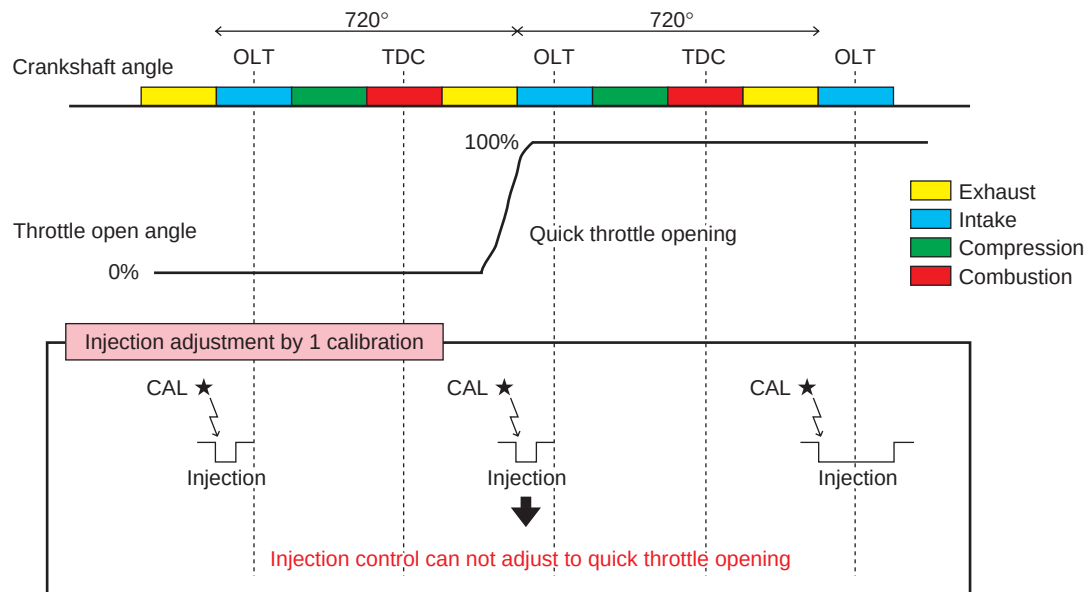


Fig. 11 Injection timing chart by 1 calibration when throttle opened

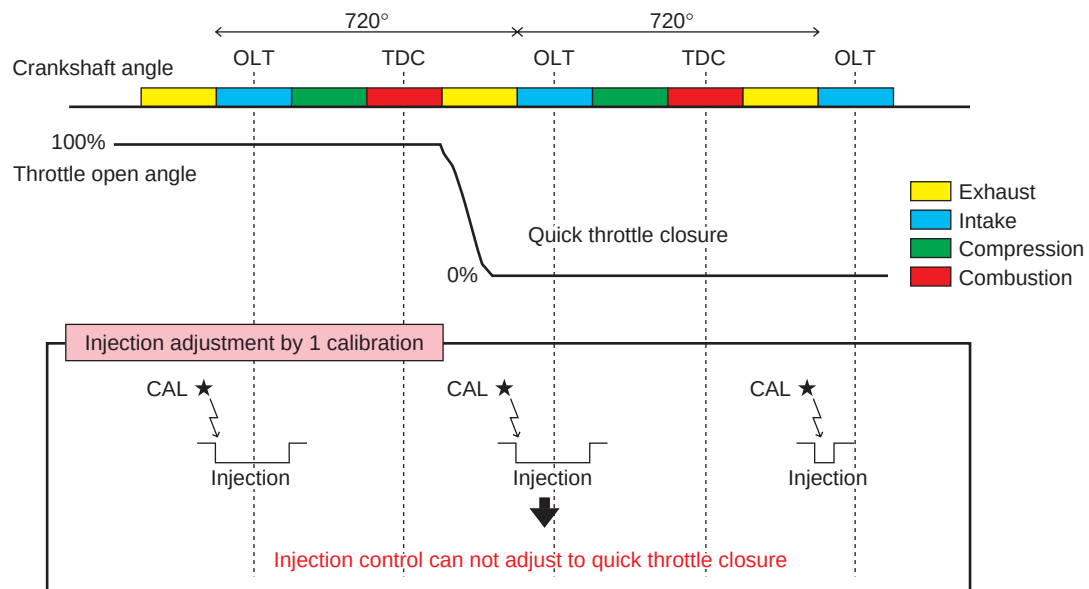


Fig. 12 Injection timing chart by 1 calibration when throttle close

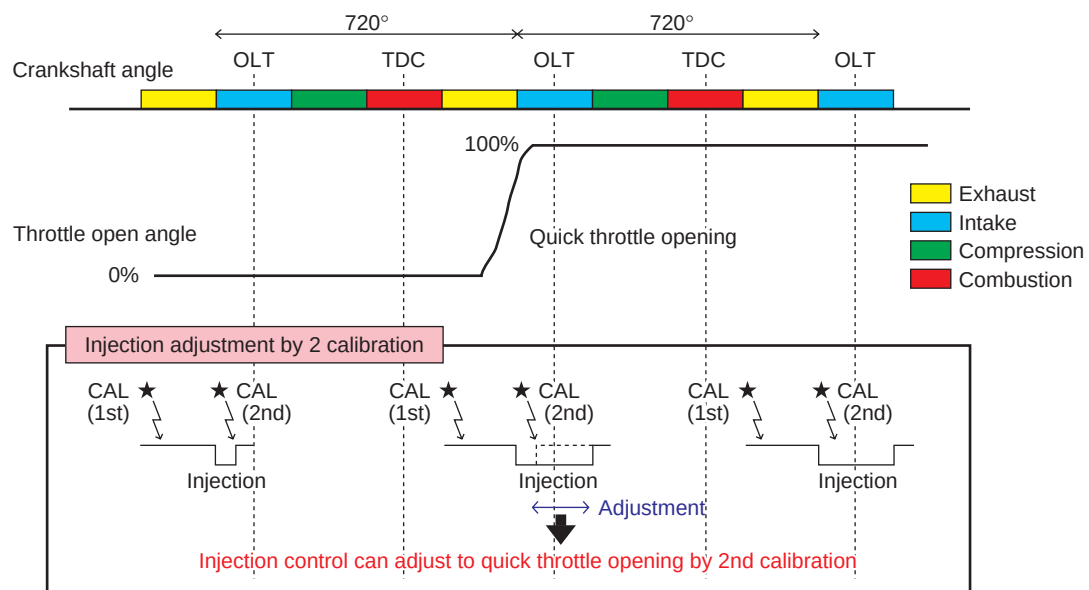
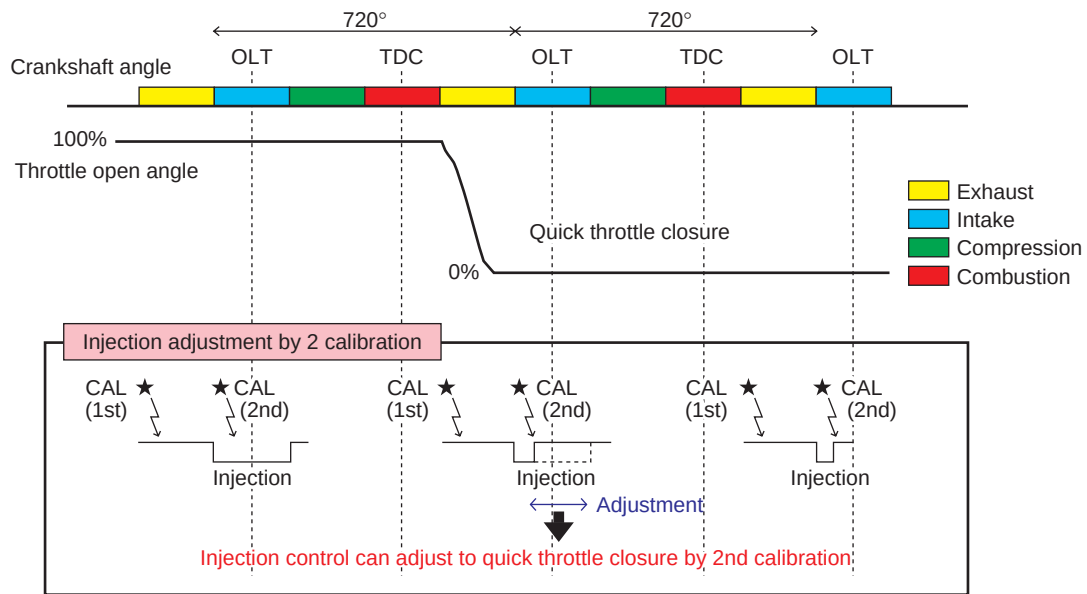


Fig. 13 Injection timing chart by 2 calibrations when throttle opened



injection control, and Fig. 14 shows the timing chart for when the throttle is closed rapidly.

With a two-stroke engine, the combustion pressure rises greatly from the beginning of opening of the throttle, so generally the throttle response is better than that of a four-stroke engine. In the case of the RTL250F, importance was placed on ensuring that the response to rapid opening of the throttle was of the same order as that of a two-stroke engine. This was done to enable a rider used to a two-stroke motorcycle to make a smooth changeover to a four-stroke engine. As a result of using 2-calibration injection control, the resulting rise in combustion pressure when the throttle was opened rapidly was comparable to that of a two-stroke engine. Figure 15 shows the results of a comparison of the combustion pressure data of the RTL250R, which is a two-stroke engine, and the RTL250F. It can be seen that the throttle response is in no way inferior to that of the two-stroke engine. Figure 16 shows the results of a comparison of the output characteristics of the RTL250R (two-stroke) and that of the RTL250F (four-stroke).

In addition, a system that could be controlled even using an ECU with a low processing speed was developed by performing 2-calibration injection control only at low engine speeds and changing

over to conventional single calibration at high engine speeds, where the calibration interval is short.

5. ECU Setting Tool

In trials competitions, races are carried out under a variety of road surface conditions, ranging from asphalt with good grip performance to river beds that have poor grip performance, and in highland regions beyond an altitude of 1 000 m. Also, users range from beginners to world championship class riders, and they have a correspondingly wide range of skill. An ECU setting tool that enables the user to change the ECU setting was developed for the RTL250F. The user can change the fuel injection amount and ignition timing within a certain range by means of a tool on a personal computer. The setting data created by the user using the ECU setting tool is stored in the EEPROM inside the ECU by serial

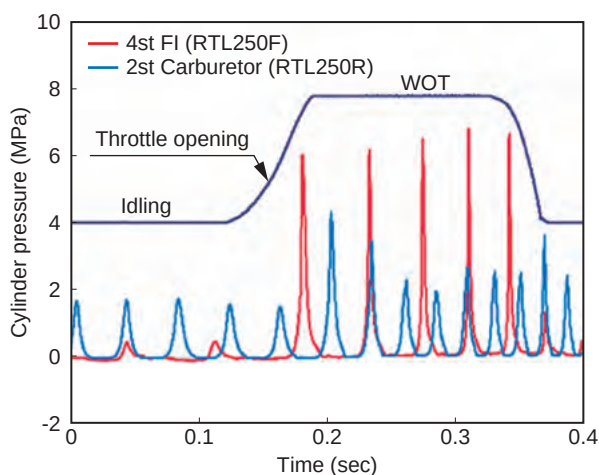


Fig. 15 Comparison data of cylinder pressure

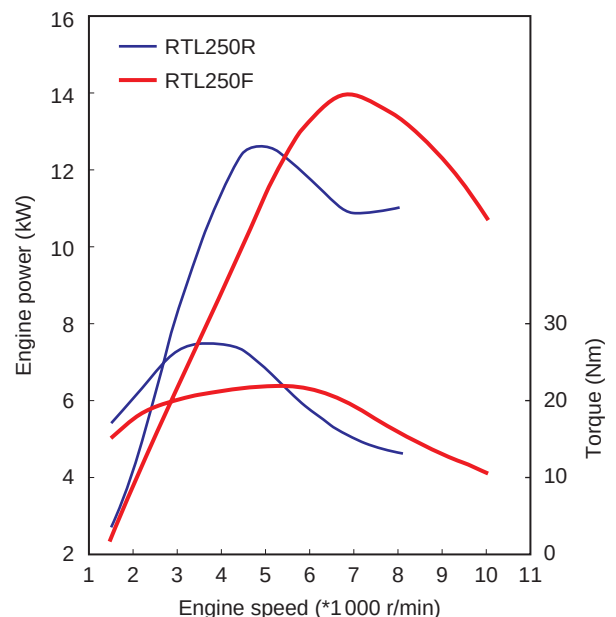


Fig. 16 Engine performance curve

communication. The ECU adds this setting data to, or subtracts it from, the fuel injection amount and ignition timing data in the flash memory, and determines the final fuel injection amount and the ignition timing. As a result, the user can find settings for various road surface conditions, altitudes and user skills. Figure 17 shows an outline of the setting tool.

6. ECU for Factory Racer

To perform the accurate and rapid ECU settings demanded by a factory racer, an ECU with a built-in data logger was developed based on the ECU for the RC211V. Compared to the ECU for the RTL250F, this ECU has a data logger function for collecting data in the flash memory, a CAN communication function for performing various measurements and driving the meters. In addition, a Bluetooth communication function for communicating

with the ECU, fan motor drive circuit, fuel pump drive circuit, multi-channel A/D input circuit, pulse input circuit, general purpose motor drive circuit, and solenoid drive circuit were added. The 32 bit CPU can cope with the increased processing required by the addition of the data logger function. There is a flash memory, in addition to an EEPROM. The flash memory can store the data (logging data) collected by the data logger function, and the EEPROM can store multiple sets of setting data. Figure 18 shows the block diagram of the ECU, and Fig. 19 shows a comparison with the ECU for the RTL250F. The device body, intake negative pressure (PB) sensor, and the throttle position sensor (TPS) are the same as those used on the ECU for the RTL250F, and the idle air control device is the same as that used on the ECU for a small motorcycle. Circuits were added along with the increasingly sophisticated functions, and the circuit board is a 6-layer 0.8 mm thick 2-stage construction flex-rigid circuit board instead of the previous 4-layer 1.6 mm thick glass-epoxy printed circuit board.

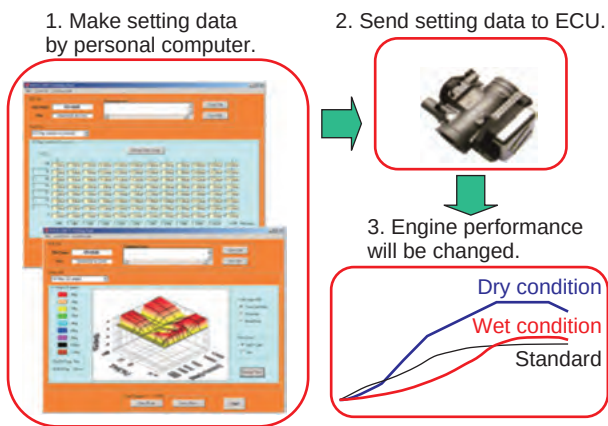


Fig. 17 ECU setting tool

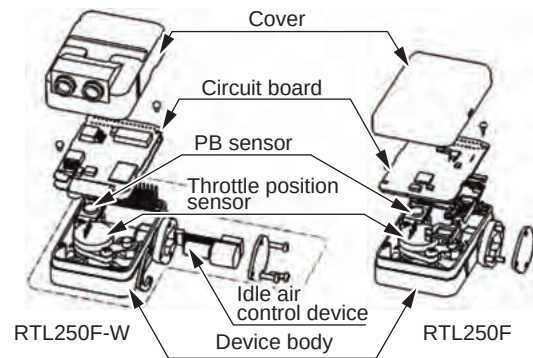


Fig. 19 ECU for factory racer

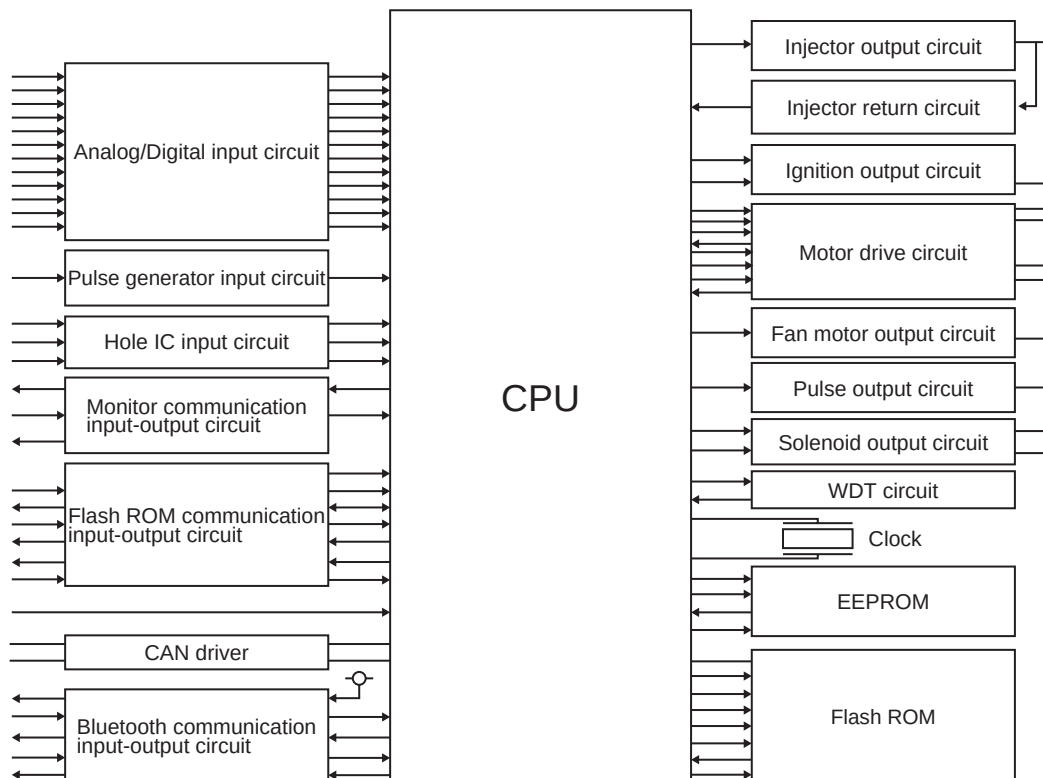


Fig. 18 ECU input-output block diagram for factory racer

The input/output circuit uses the same custom IC as that of the ECU for small motorcycles, resulting in a mere 20% volume increase and 30% mass increase. The conventional press-molded cover has been replaced by a pre-cut aluminum one, enabling the mass increase to be kept to a minimum despite the complicated shape needed by the flash memory communication connector and the Bluetooth communication connector. To obtain the optimum intake air temperature, the intake air temperature (Ta) sensor has been moved from inside the ECU to inside the air cleaner box.

The data logger function enables the collection of various data, including crank angle, engine speed, throttle opening, temperature of each part, fuel injection amount, ignition timing, and countershaft rotational speed. The collected logging data is sent to a personal computer by wireless communication using the Bluetooth communication function, and then subjected to data analysis. Based on the analyzed data, the optimum fuel injection amount and ignition timing are set, the mechanic staff creates the setting data, and the setting data is sent to the ECU by means of the Bluetooth communication function. As a result, it is possible to set the fuel injection control system in a short period of time. Figure 20 shows samples of logging data.

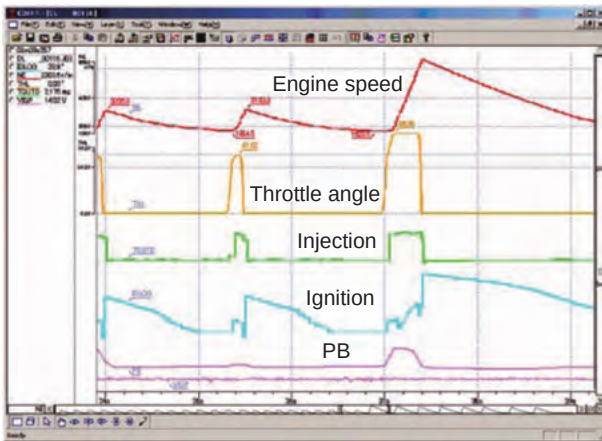


Fig. 20 Logging data sample

7. Conclusion

An electrical system that has the following features was developed for a four-stroke engine trials motorcycle.

- (1) A battery-less system that enables the motorcycle to be kick-started was developed.
- (2) The use of 2-calibration injection control enabled the fuel injection amount to be adjusted optimally in response to rapid throttle operation.
- (3) With the commercial RTL250F, a function that enables the user to freely set the fuel injection control system using a personal computer was achieved. In the case of the RTL250F-W factory racer, the built-in data logger of the ECU enables the fuel injection control system to be set in a short period of time.

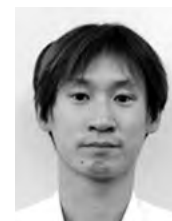
References

- (1) Tamaki, K., Kokubu, S., Akamatsu, S.: Control Device of Electronically Controlled Fuel Injection System of Air-cooled Engine for Small Motorcycle, Honda R&D Technical Review, Vol. 16, No. 1, p. 145-152
- (2) Takigawa, T., Ishikawa, E., Kakinuma, H., Kobayashi, F.: Presentation of "RTL250F" Four-stroke Engine Trials Motorcycle, Honda R&D Technical Review, Vol. 17, No. 1, p. 1-7

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Development of Motorcycle Navigation System

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ABSTRACT

For the first time, a navigation system has been installed on the Gold Wing as a standard motorcycle equipment. The system has been installed without affecting the overall concept of the Gold Wing, which includes excellent driving performance and a flowing external appearance. Components were designed smaller and lighter to achieve this. By making route guidance easy to understand and the Human-Machine Interface (HMI) easy to use, the appeal of long-distance touring model has been further enhanced. All components were designed with heat and vibration resistance as well as waterproof to allow proper function for withstanding the severe driving conditions of a motorcycle. A compact vibration-resistance semiconductor memory was applied for the map storage media.

1. Introduction

In 1981, Honda announced a navigation system for automobiles using autonomous navigation, and since then, various manufacturers of automobiles and appliances have worked to develop the possibilities of such navigation systems. Various manufacturers have commercialized navigation systems, utilizing their own concepts and technology based on the Global Positioning System (GPS). From the mid- to late 1990s, the map storage medium shifted from CD-ROM to DVD-ROM, thereby significantly increasing its usefulness. During this period, user interest rapidly increased, and the development of navigation systems by various manufacturers accelerated.

In recent years, this development has gone beyond the fields of navigation. For example, vehicles have been equipped with devices installed with Hard Disc Drives (HDD) that have common map and music memories; devices with telematic service functions that communicate with web servers provided by automobile

manufacturers to obtain various information; and devices providing visual entertainment using finely detailed 3D graphics. By installing such advanced navigation systems, the appeal of the vehicles themselves has been greatly enhanced.

For motorcycles as well, demand for similar value-added technologies has been increasing, mainly for touring models. In motorcycles, unlike automobiles, the installation space for components in a motorcycle body is small, and thus compact, lightweight systems are required. For this reason, the reality is that navigation systems for automobiles cannot be used on motorcycles as they are, and there have been no motorcycle models equipped with navigation systems as standard equipment.

After-market navigation systems have been sold that include compact portable navigation systems that can be installed on the handlebars of motorcycles. However, users who have installed these after-market navigation systems on their own motorcycle expected a navigation system that is simple and easy to use in a touring with a unified design. In consideration of user's demands, we have

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decided to develop a navigation system that can serve as a guideline for motorcycles, based on the Gold Wing. This paper will describe the technical features.

2. Development Goals

During development, the Gold Wing manufactured for the North American market was selected as the target model. The reasons for this are that it is a long-distance touring motorcycle; it was anticipated that installing a navigation system would directly increase the product appeal of this model; an audio system, vital for voice guidance, is standard on the base model; and due to its comparatively large vehicle body, there is ample capacity for installation of the components.

While maintaining the overall concept⁽¹⁾ of this model, the goal was to install a simple navigation system that would be easy for the customer base of the Gold Wing to use and that has reliable guidance performance.

The development targets were:

- (1) A compact, lightweight system that does not adversely affect the driving performance or the appeal of the external appearance of the motorcycle (Gold Wing)
- (2) A highly reliable system that functions under motorcycle driving conditions
- (3) A display that is easy to see even under motorcycle driving conditions
- (4) A switch layout that does not hinder the motorcycle driving and is easy to use
- (5) Sound with sufficient volume to be heard even under motorcycle driving conditions
- (6) A route guidance performance matching the driving conditions and high maneuverability of a motorcycle

- (7) An HMI that matches Honda automobiles in terms of look and feel and that has simple specifications that can be easily understood by Gold Wing users

3. System Overall Configuration

In this navigation system, components were designed to be as small and lightweight as possible without affecting the handling performance and maneuverability of the Gold Wing. The layout of the navigation system is shown in Fig. 1.

The meters, visor, and switches were designed so as not to tarnish the external appearance of the Gold Wing. The GPS antenna was placed inside the meter visor without exposing it. The display was integrated with the speedometer after selecting a size that was balanced when viewed as a whole, and the display was covered using a meter cover with an integrally designed visor. The switches were designed to match with original switches installed on the Gold Wing. Figure 2 shows the external appearance of the navigation system.



Fig. 2 Navigation system appearance

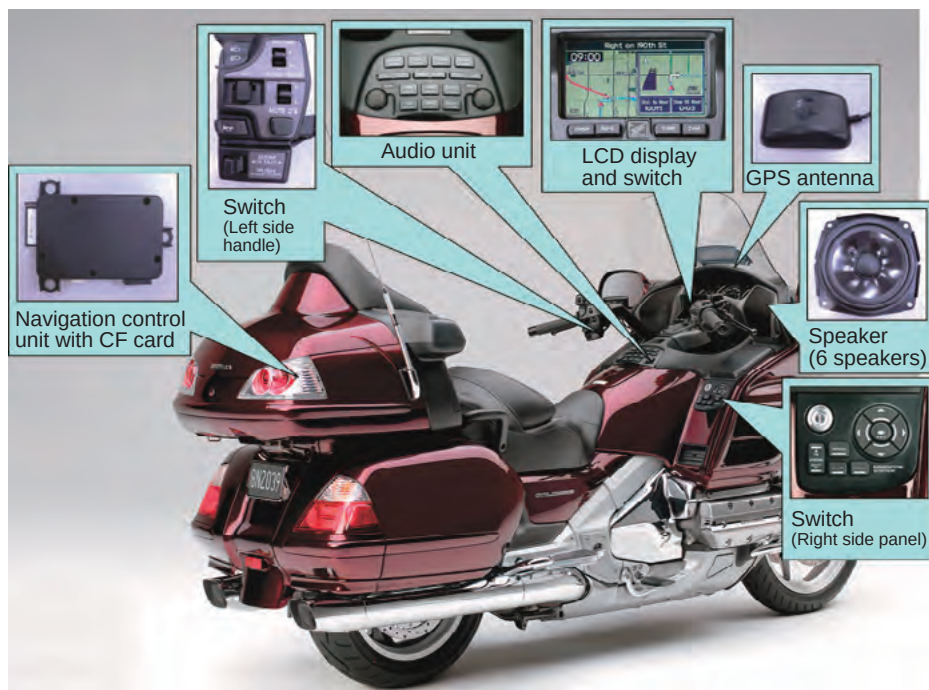


Fig. 1 Navigation system layout

4. Features of System

4.1. Display

4.1.1. Display installation position

After sufficient test driving, a position below the combination meter was selected as the location of the display because it did not affect driving.

4.1.2. Visibility of the display

Because the motorcycle does not have a roof, the rider must view the display in an open-air driving environment. Thus, in comparison to an automobile, the level of external light hitting the display fluctuates drastically between the sunshine and shade locations, and differences are great between day and night, thereby greatly influencing the ease of viewing. This display has a clear lens for eliminating the rain and dust. The surface of this clear lens was curved to avoid the reflection of the sunlight and the sky. Also, a semi-transparent and semi-reflective thin film transistor (TFT) liquid crystal display (LCD), which is less subject to the fluctuation level of external light, was used for the display module. Furthermore, driving tests were carried out to verify that the luminance of the LCD backlight would provide suitable visibility under various external light levels during day and night. Luminance for day and night modes was set up, and these modes were linked to the time function of the GPS. Each mode switches automatically. In addition, a manual control of the luminance levels was adopted to the system's specification for driving in dusk or rainy night.

4.1.3. Heat resistance reliability of the display

Motorcycles are also used under scorching sun, and under such conditions, the ambience temperature in the space between the clear lens and the display rises. Thermal storage on the liquid crystal surface caused by sunlight and heat generation by the cold-cathode tube (CCT) used in the backlight generate extreme temperature rise in the LCD module.

When the temperature of the liquid crystal exceeds operating limits, neither the reflected light nor the light of the backlight can pass through the polarizing filter, and the liquid crystal surface darkens, obscuring the image.

At the initial stage of development, darkening of the display occurred during test drives under direct sunlight in mid-summer. As a countermeasure, a heat sink was provided to reduce the temperature of the display surface. This heat sink comprises two pillars and a radiating unit. The pillars contacted on the LCD module, and about 300 square protrusions were used to establish a sufficient heat-radiating area in the radiating unit. Furthermore, the temperature of the liquid crystal unit is detected using a thermistor, and when the temperature exceeds a tolerance limit temperature, operation of the CCT is limited. As a result, blackening can be prevented with high reliability even while driving under scorching sun. Figure 3 shows the layout of the combination meter.

4.2. Control Switches

The following were concrete targets when designing the switches:

- (1) Switch operation in a driving posture should be easy
- (2) Mis-operation and interference with other switches should be avoided
- (3) Click and operation feeling should be clear when switching
- (4) Display should provide good visibility
- (5) Design should match the motorcycle appearance

Functions such as menu operation that should be carried out

during stopping and functions such as voice repeat and map enlargement that will be carried out during driving were separated, and each functional switch was arranged in suitable locations. The left, right, up, and down arrow switches, enter switch, and others needed for menu operation were concentrated on the switch panel on the right side of the vehicle. The switch shape, size, and stroke were designed to be operable even while wearing winter gloves. The switches that will be operated while driving were arranged on the left handle switch box. Similarly, the switches were designed to be easily operated even by gloved hands. The navigation panel switches are shown in Fig. 4, and the left handle switches are shown in Fig. 5.

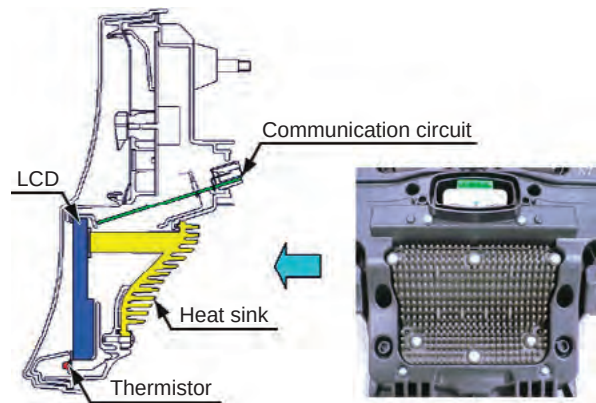


Fig. 3 Combination meter inside layout and aspect



Fig. 4 Navigation panel switches



Fig. 5 Left handle switches

4.3. Control Unit

4.3.1. Features of the control unit

Under motorcycle driving conditions, the influence of heat, water, vibration, and noise on electronic components must be given sufficient consideration. To mitigate these influences, the control unit was placed in the rear trunk and insulated from vibration. The rear trunk is distant from heat sources such as the engine and is unlikely to get wet. In order to block noises such as from the ignition of engine, motor and actuators, noise filters were provided at the input and output sections of the control unit circuit, and shielded wires were used for the communication lines between the control unit and the meters. Figure 6 shows the layout of the navigation control unit.

To make the control unit resistant to sudden shocks while driving, the hardware was built using semiconductor components with no rotating devices. Rotating devices such as CD/DVD-ROMs or HDDs normally used for map data storage are not used at all. Instead of them, compact flash (CF) memory, which uses no rotating parts, is used. Thereby, the reliability of access to data that is unaffected by heat and vibration is provided, and furthermore dark current is reduced. The reduction of dark current in a motorcycle is important because the battery charge is small compared to that of an automobile.

The control unit functions as navigation and also displays vehicle conditions. Therefore, external signals such as audio condition, outside air temperature, suspension preload level, and the operation key input were concentrated in a microcomputer in the meter as information, and are input into the control unit.

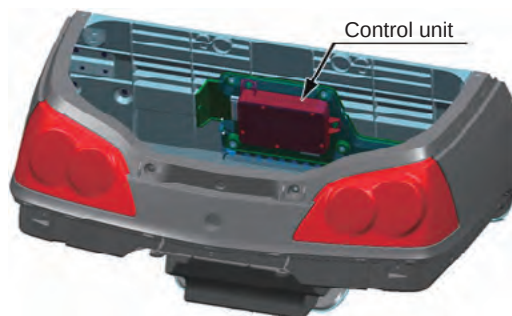


Fig. 6 Navigation control unit layout

• Function displays

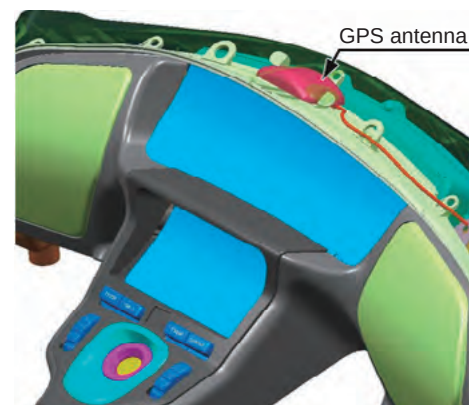
- Audio condition displays (Radio receiving: frequency, CD: playback elapsed time, CB wireless: two-way transmission conditions)
- SPA (Suspension Preload Adjustment: vehicle height adjustment) level adjusting condition
- Meter function displays (odometer, trip meter, time)
- Trunk open-closed state display
- Outside air temperature display
- Opening or ending ceremony display

The navigation system block diagram including the control unit is shown in Fig. 7. The GPS antenna location and specifications are shown in Fig. 8.

In addition, the system control program is stored on a built-in flash memory in the control unit body. The design enables subsequent version upgrades of the software to be loaded, and thereby the control unit is compatible with future system software upgrades.

4.4. Software

The system program provides a present position display as the basic functions of the navigation system, a facility search (such as hotels, restaurants, Honda dealers) for setting the destination based on the present position, and a route search to the destination based on other search results. The program has functions for route



Receiving system: 12 channel parallels
 Receiving frequency: 1575.42 MHz (L1-band)
 Receiver sensitivity (acquisition): -138 dBm
 Receiver sensitivity (tracking): -145 dBm
 Co-ordinate system: WGS-84 standard
 Measuring cycle: 1 time/1 sec
 (Estimate measuring cycle: 0.5 sec after measured)

Fig. 8 GPS antenna location and specifications

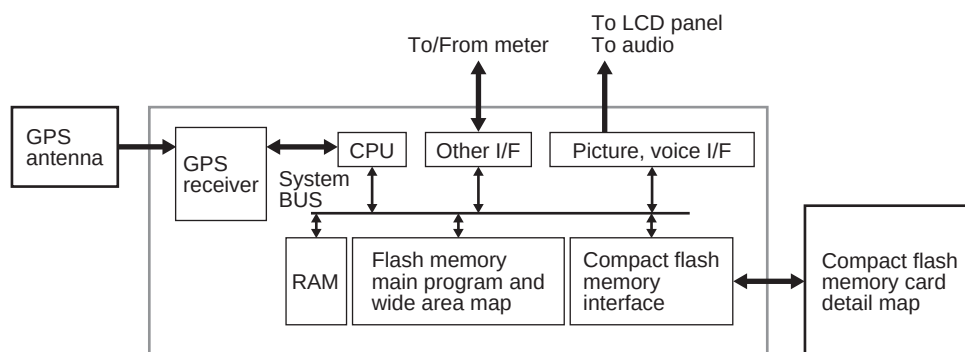


Fig. 7 Navigation system block diagram

information such as enlarged intersection maps, voice guidance to the destination, and automatic rerouting when the vehicle has departed from the guide path. Software update function is provided; the system is automatically updated by inserting a compact flash containing new system software.

In addition, predictive positioning every 0.5 seconds is executed as simple predictive navigation for interpolating the signal reception cycle from satellites (1 sec/cycle).

Hereby, the movement of own vehicle icon in an intersection pop-up is smoothed. Furthermore, even when satellite reception is not possible, the program updates the vehicle position display by carrying out map matching every 30 seconds based both on the position information of own vehicle received up to that time and the velocity pulse of own vehicle.

To realize this predictive navigation, specifications were established in which the velocity pulse serves as the velocity reference for the vehicle, so the display position update is not executed using only the position information data from GPS.

4.5. Sound Specifications

In consideration of open air high-speed driving conditions, an audio system that is more powerful than that of the base model was installed. Speakers were changed to large-diameter speakers with a resin cone, and the installation angle was modified to provide clearer sound. As a measure against noise from wind turbulence, the voice band was intensified to make the navigation voice guidance easier to hear. A sound characteristics are shown in Fig. 9.

Furthermore, the audio is muted during navigation voice output to make the guidance easier to hear. The audio system that automatically changes the volume depending on vehicle velocity is standard on the base model, and voice guidance for navigation has also been similarly linked to vehicle velocity. The volume for navigation itself is adjusted according to audio volume, but fine adjustments to increase or decrease the volume over that of the audio are possible. Auto volume bass step characteristics are shown in Fig. 10 and auto volume treble step characteristics are shown in Fig. 11.

5. Guidelines and Evaluation of the Basic Functions for Route Guidance

5.1. The Basic Functions for Route Guidance

As described in Section 2, considerations must be made that the navigation hardware does not adversely affect either the driving performance of the vehicle or the appeal of the external appearance. And, for software as well, considerations must be made that it does not influence motorcycle driving performance. Test drives were

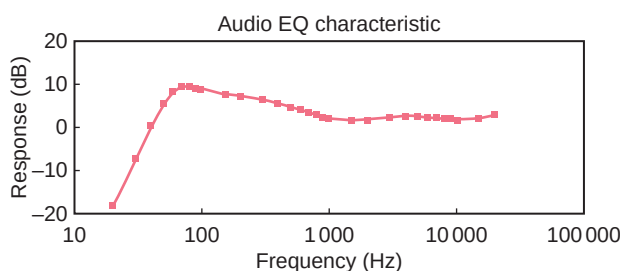


Fig. 9 Sound characteristic

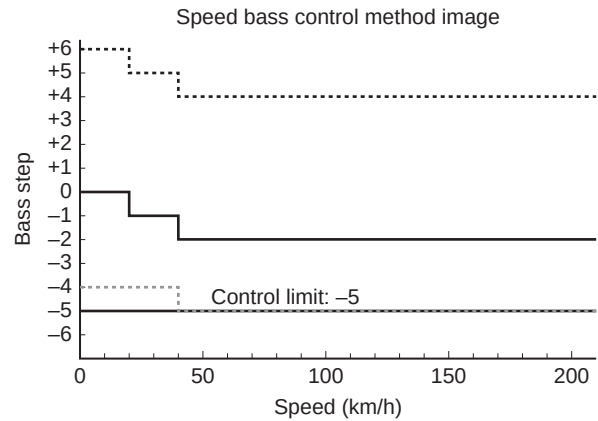


Fig. 10 Auto volume bass step characteristics

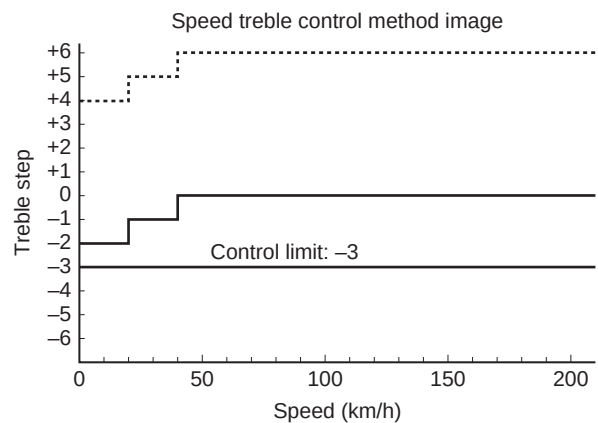


Fig. 11 Auto volume treble step characteristics

carried out on public roads under various conditions, and guidelines for basic route guidance functions of the navigation system for a motorcycle were set as shown below to refine the route guidance specifications.

- (1) When deviating from the route, a rerouting is calculated promptly and route guidance is resumed.
- (2) A simple image composition is provided so that the rider does not have to continuously look at the display, and intersection guidance is performed using audio synchronizing with pop-up screens. In the guidance, exact instructions are given by voice so that the rider understands the next action, and the display provides information of the distance up to the next intersection.
- (3) Guidance at an intersection is delivered at an appropriate time that is neither too slow nor too fast.
- (4) Voice guidance provides the minimum necessary words, is not bound by grammar or polite language, and gives priority to ease of understanding.

During development of the navigation system, detailed specifications were incorporated following the guidelines described above, and guidance timing was optimized. Figure 12 shows an example of route guidance modification 1, and Fig. 13 shows an example of route guidance modification 2.

5.2. Countdown Bar

To correctly show exactly where the driver should turn at an intersection, a 10-stage countdown bar in a pop-up display screen

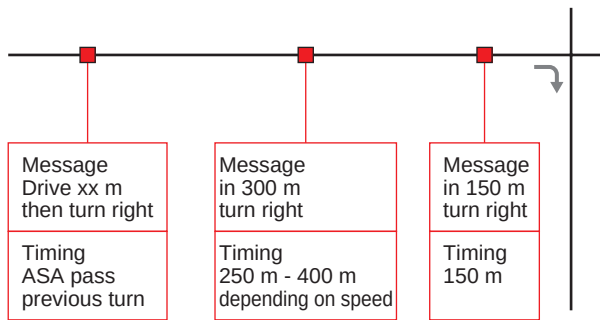


Fig. 12 Example of route guide modification 1

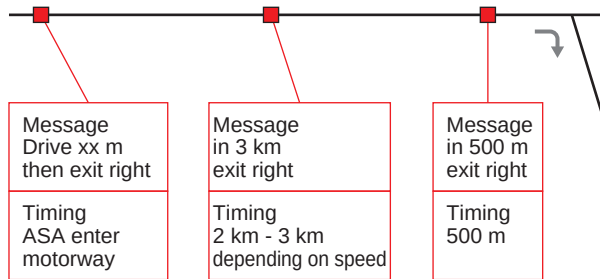


Fig. 13 Example of route guide modification 2

is used to provide the driver with a sense of distance. Using this countdown bar display, the driver can intuitively recognize the intersection or interchange where he or she should turn. An example of the countdown bar display is shown in Fig. 14.

5.3. HMI

During HMI development, the HMI for the 2003 model of the Honda US ACCORD was used as a reference. By using an HMI that is familiar to the North American GL customer base, it gave a unity in look and feel as a Honda product. Additionally, an extremely simple screen composition was adopted to make the information easily understood. An example of display format is shown in Fig. 15.



Fig. 14 Display example of countdown bar



Fig. 15 Example of HMI modification

5.4. Evaluation of the Motorcycle Navigation System

5.4.1. Evaluation of GPS performance

Because this navigation system displays the present position based on data from GPS satellites, satellite signal reception performance requires high stability. There are issues that are unique to motorcycles such as reception level reduction due to the vehicle banking during cornering or blocking of electromagnetic waves by the rider. These issues were thoroughly examined in the initial stages of development, and the reception system was arranged so as to minimize all such influences.

In order to enhance the capability for a production, the evaluation of GPS performance was confirmed by test drives that focused on the reception limits of GPS satellites.

The performance evaluation tests for satellite positioning limits of GPS were performed.

The results are shown below:

- (1) High latitude threshold
By comparing the displayed intersection position and the actual intersection position, it was possible to confirm that there was normal position precision even at high latitudes. It was also possible to confirm that the vehicle position did not depart from the road.
- (2) Verification of the influence of high-rise buildings
In order to confirm the poor reception and the multipass of signal due to high-rise buildings during driving, the system performance was verified. The verification was performed by driving according to the route guidance that passes through the valley between high-rise buildings. During route guidance, positioning error increased at the center of an area of high buildings and the update cycle of the own vehicle position display delayed. However, it was confirmed that it was possible to reach the destination by guidance using navigation text and displays of the street names at intersections.

Because the positioning of this navigation system uses only the GPS satellites, it was considered that the deterioration occurs in the route guidance reliability by the skipping of the own vehicle icon when the GPS positioning performance deteriorates. To correct this, a function was added that informs the rider about any decrease

Table 1 GPS satellite positioning status display

GPS satellite reception conditions	Display
Low reception levels	“POOR SATELLITE RECEPTION” displayed in the navigation text bar Background: Yellow
Lowest reception level	“LOST SATELLITE RECEPTION” displayed in the navigation text bar Background: Red Voice guidance one time

in position precision by changing the display depending on the satellite reception level. Table 1 shows the GPS satellite positioning status display. If the above conditions continue, the background color of the bar on the upper part of the display turns yellow for POOR and red for LOST, and normal text display continues.

This function was verified in an area of valley between high-rise buildings where reception from the satellites was less reliable. As a result, it was confirmed that the current specifications provide easily understood displays.

6. Conclusion

Targeting users of the Gold Wing, an intuitive, easy to use, and attractive navigation system has been developed. The structural components, such as the LCD and switches, that are a part of the appearance are designed to provide a feeling of unity with Honda motorcycles.

Because each of the mounted functional components that form the system is compact and lightweight, the high driving performance of the Gold Wing is unaffected. High reliability is ensured in terms of the motorcycle travel environment by using a high vibration-resistance CF, a semi-transparent and semi-reflective LCD module, heat accumulation measures for the LCD, and innovations in the shape of the lens and the visor. In addition, switch shapes that can be easily manipulated during travel and an auto volume that is easily heard during high-speed travel were used. The navigation system provides a good usability by announcing intersection guidance in suitable timing for a traveling motorcycle, based on the basic performance guidelines of the motorcycle’s route guidance system.

Acknowledgements

We would like to acknowledge the people of GARMIN Ltd., Panasonic Automotive System Company, and Nippon Seiki Co., Ltd. for their cooperation and help in the development of this navigation system.

References

- (1) Aoki, M., Tosaka, T., Minami, H., Ishihara, H., Kano, K.: Introduction of Gold Wing GL1800, Honda R&D Technical Review, Vol. 13, No. 2, p. 13-26

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Development of Heaterless Oxygen Sensor for Small Motorcycles

Mamoru URAKI*

Tatsuo HAYASHI*

ABSTRACT

An oxygen sensor for the oxygen feedback system of small size motorcycles was developed. Small size, low cost and low power consumption are required for the oxygen sensor of small size motorcycles. To achieve this heaterless, thin protector, cap and body ground structures were adopted. As a result, the sensor was reduced by 57% in volume, by 43% in mass and by 79% in the projected area of sensing unit compared with conventional sensors. Utilizing a heaterless design cost reduction by reducing parts and lower power consumption were realized. In addition, the engine output and drivability were enhanced due to the miniaturization of the sensing unit, and the flexibility of sensor installation was enhanced. Moreover, the EURO-3 mode emission control limit was achieved by using the developed sensor.

1. Introduction

Concern for global environmental issues has recently increased, also the reduction of motorcycle exhaust emissions has been increasingly required, and countries around the world are enforcing new emissions legislation. From 2006, EURO-3 regulations have been scheduled for motorcycles in the EU, and manufacturers are required to respond.

Air-fuel ratio feedback systems⁽¹⁾ are already being used in automobiles and large motorcycles to reduce emissions. It is essential to also use oxygen feedback systems in small motorcycles.

However, the following issues arise if the oxygen sensors⁽²⁾ used in automobiles and large motorcycles are used without modification in small motorcycles:

- (1) It is difficult to install the sensors in the short, small-diameter exhaust pipes of small motorcycles.
- (2) Small motorcycles have small-diameter exhaust pipes; engine power and throttle response are reduced by exhaust resistance due to a sensor element protruded into the exhaust flow.
- (3) The use of a heater to warm the sensor element increases the range of mounting options, but also necessitates greater electricity consumption, thus increasing friction in the generator and reducing engine output and fuel economy.

A heaterless oxygen sensor for small motorcycles has been developed to resolve these issues.

This paper describes the aims of the newly developed sensor, its features and its achieved performance.

* Motorcycle R&D Center

2. Development Goals

The following three development aims were established based on a consideration of the requirements for an oxygen sensor that would be used in small motorcycles:

- (1) Miniaturization
- (2) Low power consumption (heaterless element)
- (3) Low cost

The oxygen sensor measures the amount of oxygen using an element protruding into the exhaust pipe; a large element generates exhaust resistance. This exhaust resistance contributes to reduce engine output and throttle response.

In the exhaust system of small motorcycles, low-diameter exhaust pipes are used, and the effect on exhaust resistance due to installed oxygen sensor is greater. It is therefore particularly necessary in small motorcycles to reduce the size of the sensor element in order to minimize exhaust resistance.

From the viewpoint of cost reduction, a reduction in the number of parts and the amount of materials used, and miniaturization are required.

The element of an oxygen sensor must be activated to measure the amount of oxygen. To activate the element, it is necessary to maintain the surface temperature of the element at approximately 300 °C or above⁽³⁾.

The oxygen sensors used in large motorcycles use heaters to increase the temperature of the element. The reason for this is that the distance from the exhaust port to the catalyzer, near which the oxygen sensor is located, is long. The high-temperature gas discharged from the cylinder flows through the exhaust pipe into the catalyzer, but the exhaust gas temperature at the catalyzer becomes low in the case of longer exhaust pipe. The low-temperature gas reaching the sensor is unable to increase the temperature of the element sufficiently to maintain activation. The oxygen sensors used in large motorcycles are positioned downstream from the joint of exhaust pipes, and because of space restrictions, almost all sensors are positioned upstream of the catalyzer. Oxygen sensors in vehicles in which the distance from the exhaust port to the sensor position is long are required to use heaters to maintain the activation of the element.

The heater must be fitted inside the body of the sensor; this increases the size of the unit, increases its power consumption and also increases cost by increasing the number of parts used.

Therefore, the sensor was positioned close to the exhaust ports to enable activation of the element by the high-temperature exhaust gas instead of a heater, and the heaterless structure was used for oxygen sensor of small motorcycles. This heaterless structure enabled the miniaturization of the sensor body and reduced power consumption and cost.

The reduction in power consumption achieved by the heaterless structure against an oxygen sensor with a heater led to a reduction in friction in the generator in the vehicle in which the sensor was fitted. Therefore significant benefits in terms of fuel economy, engine output and cost reductions were obtained.

3. Features of the Developed Oxygen Sensor for Small Motorcycles

Figures 1 and 2, respectively, show an external view and a structural diagram of the developed sensor. Table 1 shows a comparison of the major specifications of the developed and conventional sensors. The volume and mass of the sensor and the projecting area of the sensing section were reduced by 57%, 43%

and 79%, respectively.

The unique features of this sensor are the cap structure, which resembles the spark plug in a gasoline engine, and the ceramic sleeve (outer tube).

In a conventional sensor, the sensor signal wire, the earth wire and the heater wire run from the sensor lead through the rubber parts (cap, grommet, etc.) and are caulked into the metallic parts (the sensor body). This configuration was changed to the cap structure, which is removable; this cap structure not only reduced the size of the sensor body, but also reduced the working space required when the sensor is mounted. This increased the flexibility of mounting the unit.

In addition, because it was no longer necessary to caulk the lead wire outlet, the structure of the outlet was changed from metal to an alumina ceramic sleeve. Alumina ceramic is a material with high strength and high insulating ability even under high temperatures. In consideration of their characteristics, the alumina ceramic was employed to ensure a balance between the strength of the body component and the insulating function. This also reduced the number of parts used and simplified the production process.



Fig. 1 External view of developed heaterless O₂ sensor

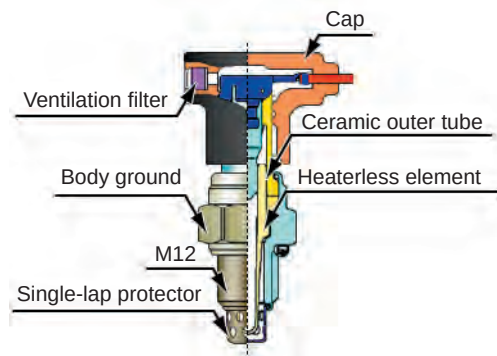


Fig. 2 Structure of heaterless O₂ sensor

Table 1 Specifications of sensors

Sensors	Developed	Conventional	Reduction rate
Flange height (mm)	40	55	27%
Screw radius	M12	M18	33%
Protector	Outer diameter (mm)	10.4	29%
	Projection area (mm ²)	206.5	79%
Parts number	15	26	—
Volume (cm ³)	6.4	14.8	57%
Mass (g)	38	67	43%
Heater	Unattached	Attached	—

In addition, one feature of this sensor is its miniaturized size⁽²⁾ compared with conventional sensors, which allows it to be mounted near the exhaust port, where the temperature of the exhaust gas is highest (Fig. 3).

Miniaturizing the body of the sensor increases flexibility in mounting the unit, and also contributes to cost reduction. Making the sensing element smaller also contributes significantly to preventing any deterioration in engine output and throttle response.

The key structures of the sensor are as follows:

- (1) Thin protector structure
- (2) Heaterless structure
- (3) Cap structure
- (4) Body ground structure

These structures were employed to achieve the development aims described in Chapter 2.

3.1. Thin Protector Structure

The outer shape of the protector for the element that protrudes into the exhaust flow to enable the sensor to determine the amount of oxygen in the exhaust gas were reduced as much as possible (the projecting area of the sensing section was reduced by 79% against conventional sensors).

If the area protruding into the exhaust pipe is large, it generates exhaust resistance, and this affects engine output and throttle response.

Figure 4 shows exhaust pressure against vehicle speed for different protector projecting areas measured using a 50 cm³ scooter. The 0 km/h in Fig. 4 shows the idling state. The exhaust pipe of the test vehicle was 16.8 mm in diameter. "Without sensor" in the figure means that no sensor was mounted in the exhaust pipe.

Figure 4 shows that in a small motorcycle with a small exhaust pipe, the projecting area of the sensor has a significant effect on increasing exhaust pressure at the maximum speed of 57 km/h. The projecting area of the protector in the new sensor was reduced to 43.6 mm², thus reducing exhaust resistance.

The (sensor of) heaterless structure, discussed in Section 3.2., requires high temperatures to activate the sensor element by using the heat of the exhaust gas. This includes antithetical requirements that the protruding area into exhaust pipe of the sensor should be reduced for reducing the exhaust resistance, but on the other hand, the protruding area should be increased for activating the sensor element.

Condensed water adhering to the sensor element may break the ceramic element by thermal shocks. Therefore, in a conventional sensor, a double protector structure is used to reduce contact with the exhaust gas because the element activation temperature is

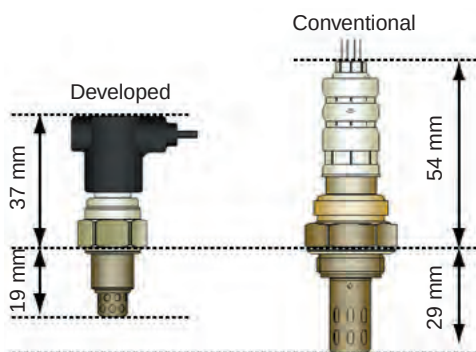


Fig. 3 Comparison of sensor dimensions

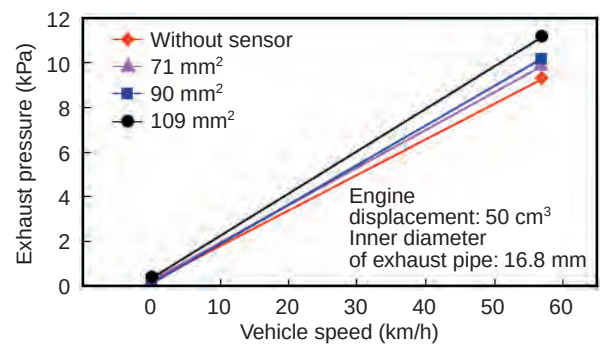


Fig. 4 Relationship between sensors projection area in exhaust pipe and exhaust pressure by vehicle speed

ensured by heating with a heater, and to reduce the adhesion to the element of condensed water in the exhaust pipe. The use of a heaterless structure⁽⁴⁾ in the newly developed sensor reduced the temperature of the element in driving conditions under which condensed water is produced, thus the incidence of thermal shock due to adhesion of condensed water was reduced. This allowed the use of a single protector with a thin structure.

In addition, this single thin protector structure enables more exhaust gas to contact the element than the conventional double protector structure, despite the reduced projecting area of the sensor, and thus contributes to the use of a heaterless structure. Figure 5 shows a comparison of protector structures in the conventional and newly developed sensors.

3.2. Heaterless Structure

As described in the section on development aims, conventional oxygen sensors use a heater to activate the element. These heaters require approximately 6.6 W of power consumption.

Figure 6 shows the results of measurements of the exhaust gas temperature at different points on the exhaust pipe at various vehicle speeds, using a 50 cm³ scooter.

Positioning the oxygen sensor close to the exhaust port in the upstream of exhaust pipe enables high-temperature exhaust gas to contact the sensor, and thus the element is maintained in the activated state required for sensing even without the heater. This makes the heaterless structure possible.

To position the sensor close to the exhaust port, it is necessary to determine changes in the amount of residual oxygen from the exhaust port to the catalyzer in the exhaust pipe. In the case of long exhaust pipes or complexly shaped exhaust pipes with multiple cylinders, the amount of residual oxygen in the exhaust gas is significantly different at the sensing section and at the inlet of the

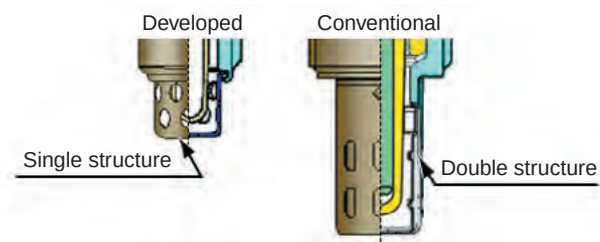


Fig. 5 Comparison of protector structures

catalyzer due to afterburning, and that would affect oxygen feedback control. However, in the case of a small motorcycle with a short, small-diameter exhaust pipe, the difference in A/F values calculated from changes in residual oxygen is less than 0.1. The effect of this difference on oxygen feedback control can be ignored.

The total length of the sensor element was shortened (-16 mm against the conventional sensor) allowing heat to be more effectively conducted to the sensor element body, and reducing its overall size. Figure 7 shows a comparison of element structures.

The use of a heaterless structure enabled the elimination of the heater body and the reduction of the heater wiring and heater support components. These are effective methods for miniaturization of the sensor body, reduction of electric power of the element heater, and cost reduction.

3.3. Body Ground Structure

The body ground structure is shown in Fig. 8. The number of parts was reduced by using the body ground structure instead of conventional lead wires. Use of the body ground configuration required torque management of the screw when mounting the sensor and ensuring of proper grounding of the sensor.

The required thread length for the sensor body screws was ensured for mounting on the aluminum cylinder head.

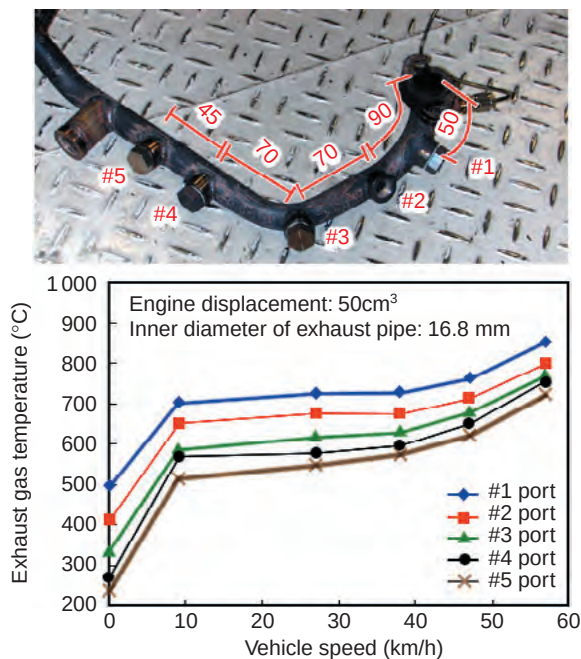


Fig. 6 Measurement result of exhaust gas temperature in each port

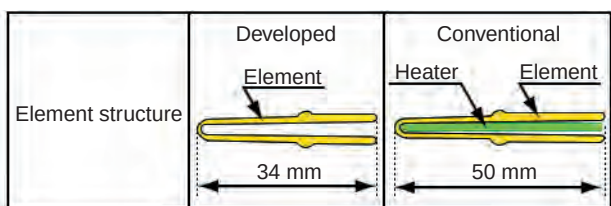


Fig. 7 Comparison of element structures

The body component of a conventional sensor was constructed using a metal outer casing⁽²⁾, and ceramic separators were used to insulate the sensor output wire, the ground wire and the heater wire. The body component of the newly developed sensor was constructed using a ceramic sleeve made of an alumina ceramic, and excellent insulation performance ensured both strength and insulation, simplified the structure, reduced the number of parts, and simplified the production process.

3.4. Cap Structure

The cap structure is shown in Fig. 9. A dividable cap structure for the body and the signal wire outlet was employed to downsize the sensor body.

The L-shaped cap structure enables the signal wires to bend at a right angle, thus increasing the ease of mounting the sensor in the confined space of a small motorcycle exhaust pipe (Fig. 10).

The dividable structure does not require a special tool to mount the body of the sensor to the vehicle, thus contributing to increased productivity.

A waterproof ventilating filter (made by PTFE) is mounted in the cap to ensure that the required amount of oxygen reaches the interior of the sensor. Just in case any gas that affects sensing performance is produced inside the sensor, the unit has a self-cleaning function.

The water proofing property of the cap provides the same level of waterproof pressure as a standard waterproof coupler, ensuring that the toughness required for mounting on motorcycles was achieved.

The sensor mounting is exposed in high temperature as the sensor is mounted on the exhaust pipe; therefore fluorine rubber was used for the exterior of the cap. The cap can resist temperatures of up to 230 °C.

The use of the body ground structure and the cap structure simplified the configuration of the conducting terminals between the sensor element and the lead wire, thus contributing to a reduction in the size of the sensor body.

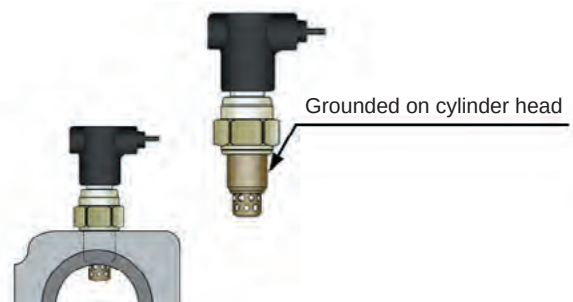


Fig. 8 Structure of body ground

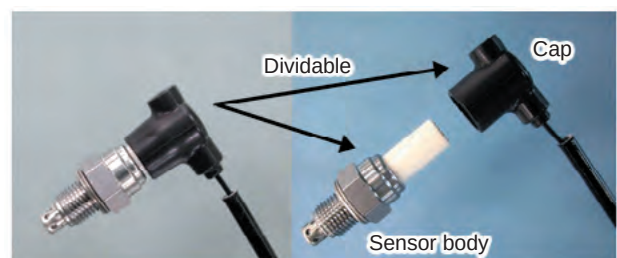


Fig. 9 Structure of cap

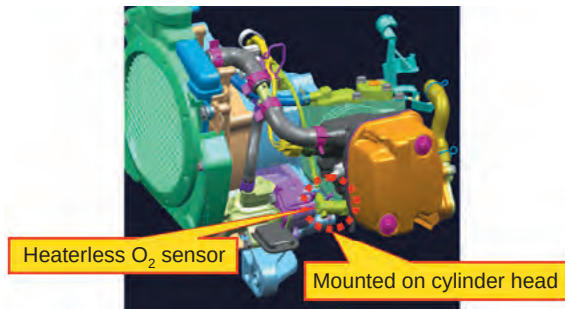


Fig. 10 Example of O₂ sensor mounted on scooter engine

4. Results of Complete Vehicle Tests

A complete vehicle (a 100 cm³ class scooter) equipped with the newly developed oxygen sensor with heaterless structure, satisfied EURO-3 mode emissions requirements.

Changes in the temperature of each section of the sensor during EURO-3 emissions measurements are shown in Fig. 11, and the results of the emissions measurements are shown in Table 2.

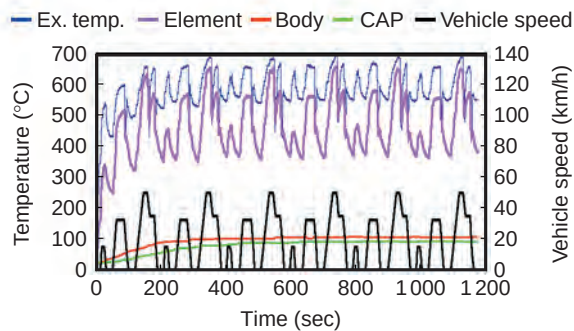


Fig. 11 Measured temperatures during emission measurement test

Table 2 Emission results

Emission	CO	HC	NOx
Result value	1.03	0.14	0.05
Limit value	2.0	0.8	0.15

5. Conclusion

An oxygen sensor with a compact sensing section has been developed for use in small motorcycles.

- (1) The sensor body has been miniaturized by using a heaterless, thin protector, cap and body ground structures; the volume of the new sensor was reduced by 57%, its mass by 43% and the projecting area of the sensing section by 79% over those of a conventional sensor.
- (2) The compact heaterless sensor has a minimal effect on engine output and fuel economy; a complete vehicle with the new sensor installed satisfied EURO-3 emissions requirements in mode tests.

Acknowledgments

We wish to sincerely thank Mr. Keisuke Makino and Mr. Tatsuya Okumura of NGK Spark Plug Co., Ltd. for their assistance in this development project.

References

- (1) Nishio, K.: Sensors for Automobile Engine Control, SANKAIDO PUBLISHING Co., Ltd, p. 77-100 (1999)
- (2) Nakahara, Y., Takami, A.: Automobile Combustion Control Sensors (4), Internal Combustion Engine, Vol. 30, No. 11, p. 72-82 (1991)
- (3) Nakahara, Y., Takami, A.: Automobile Combustion Control Sensors (6), Internal Combustion Engine, Vol. 31, No. 3, p. 69-76 (1992)
- (4) Nakahara, Y., Takami, A.: Automobile Combustion Control Sensors (5), Internal Combustion Engine, Vol. 30, No. 12, p. 73-84 (1991)

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Tatsuo HAYASHI

Development of Moquette Fabric for Enhancing Quality of Assembled Car Seat

Tomoyuki UEMURA* Rie HAYASHI*

ABSTRACT

A moquette fabric has been developed for car seats that reduces the incidence of grinning and wrinkles. This new fabric has had a huge effect on the quality of assembled seats. The pile yarn structure of the moquette fabric has been enhanced to prevent grinning by structuring the pile yarn so that highly crimped yarn is in the center. This reduces exposure of the foundation material when the fabric is folded. The fitting of the pile fabric to the seat has also been enhanced by increasing the stretchability of the foundation material by 50-100%. Furthermore, the bending resistance of the fabric has been decreased to under 50 mm by decreasing the amount used of highly permeable back coating material with an extra low Tg (Glass transition temperature). As a result, the seats assembled using this newly developed fabric are higher in quality, with no grinning or wrinkles.

1. Introduction

The demand for more luxurious automobile interiors has been increasing year by year. As seats are very important in terms of car interiors, demand is particularly strong for higher quality assembled seat appearance.

The various types of seat covering materials used include fabric, and synthetic and genuine leather. Of these, the most widely used material is fabric, so enhancing the appearance of seats covered in fabric is equivalent to enhancing the majority of seats used in Honda vehicles. Of fabric seat covering materials, moquette fabric has been extensively used as a high-quality material because it can be woven into a wide variety of patterns and also dyed in a wide array of colors. However, it is difficult to make further enhancements to assembled seat appearance made from conventional moquette fabric. Accordingly, there was a strong demand for an enhanced moquette fabric that could be used to make more luxurious seat covers and thus meet design requirements.

This paper describes the goals in developing a new moquette fabric for enhancing seat coverings, the features of the newly developed material, and the development technology and its effectiveness.

2. Construction of Moquette Covering Material

The general construction and the specifications of moquette are shown in Fig. 1 and Table 1, respectively. Moquette is a pile material made by center-cutting woven material of a double-faced construction⁽¹⁾. Various kinds of pile yarns such as regular yarn and textured yarns are used to make a variety of patterns. Both the warp and the weft of the ground yarn are made of twisted yarn consisting of polyethylene terephthalate (hereafter called PET) and rayon to provide strength and flame retardant properties. The resin back-coating material is acrylic resin applied in layers to prevent the pile yarn from dropping out.

* Automobile R&D Center

Table 1 Construction of current material

Category		Specifications	
		Material	Note
Pile yarn		PET	Regular yarn
		PET	Textured yarn
Ground yarn	Warp	PET/rayon	Twisted yarn
	Weft	PET/rayon	Twisted yarn
Backcoating		Acrylic resin	80-100g/m ²

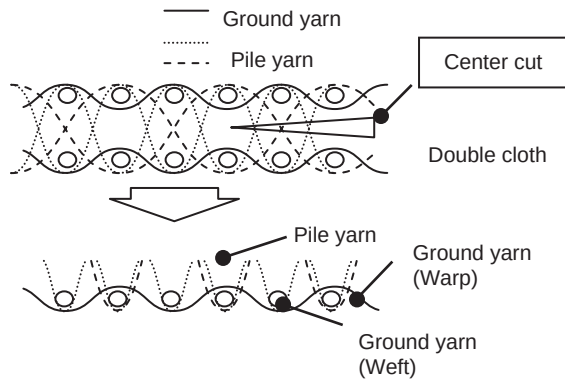


Fig. 1 General construction of moquette

3. Development Goals

When the abovementioned moquette covering material is assembled to the seat, grinning and wrinkling occur. The causes of these phenomena were analyzed, and their relationship to the physical properties of the covering material and the kind of covering material was studied. The results are shown in Fig. 2.

Grinning is a phenomenon whereby the ground yarn becomes visible when the covering material is bent, as shown in Fig. 3. It can be seen that grinning is related to the type of pile, including its shape and length, and also that wrinkles are related to the stretchability of the covering material and its stiffness. In the

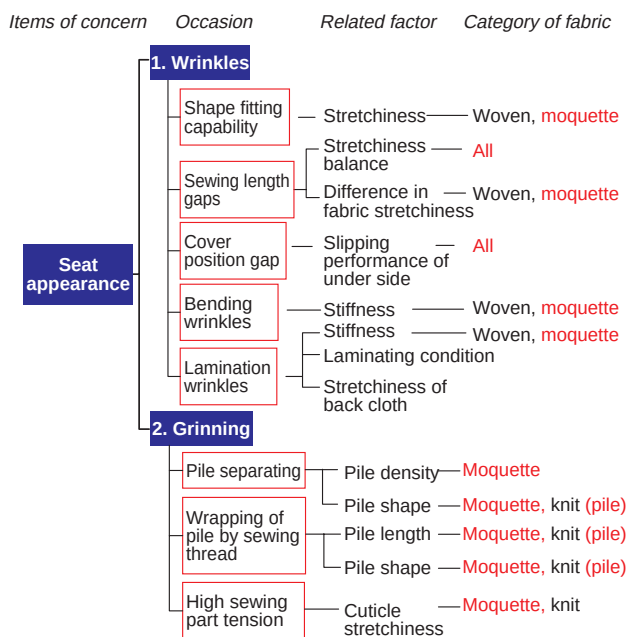


Fig. 2 Diagram of characteristic causes of grinning and wrinkles

Table 2 Grinning evaluation index

Level	Target level
	Criteria
5	Excellent
4	Good
3	Acceptable
2	Marginal
1	Not Acceptable

development of the seat fabric, a quantitative method of evaluating grinning was established and then a target was set, while for wrinkles a target was set to produce stretching and stiffness similar to knit, which produce relatively few wrinkles.

3.1. Grinning Evaluation Method and Material Development Target

Grinning consists of two phenomena, “grinning that occurs at curved parts of the seat” and “grinning that occurs at sewn parts.” An evaluation method was established for each of these.

3.1.1. Grinning at curved parts of the seat

The covering material was wrapped around a jig perpendicular to the warp, weft and bias directions, respectively, as shown in Fig. 4, and the grinning condition was evaluated in each case. Table 2 shows an outline of the grinning evaluation index. The development target was a level of 4 or higher at which grinning was not noticeable.

3.1.2. Grinning at sewn parts of the seat

The covering material sewn in the warp, weft and bias directions, respectively, was wrapped around a jig, as shown in Fig. 5, and from the observed grinning condition 5 levels were created. Likewise, the grinning condition was indexed according to the



Fig. 3 Grinning in moquette

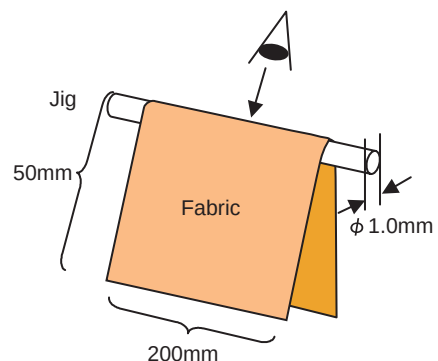


Fig. 4 Grinning test

5 levels of Table 2. The development target was a level of at least 4 or higher at which grinning at sewn parts was not noticeable, as in the case of grinning that occurs at curved parts of the seat.

3.2. Setting a Development Target for Wrinkles

The stretchiness was obtained using B: Constant load method stipulated in 8.15.2 of JIS L1096, and the stiffness was obtained using the 45° cantilever method stipulated in 8.19.1⁽²⁾.

The target stretchiness and the stiffness of the developed material were the actual values for knit material which does not produce wrinkles. These were a stretchiness of 10% or higher and stiffness of 50 mm or less, and are shown in Figs. 6 and 7.

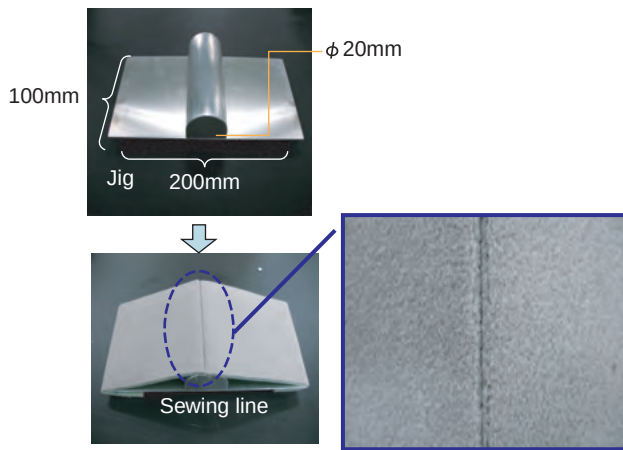


Fig. 5 Test of grinning caused by sewing line

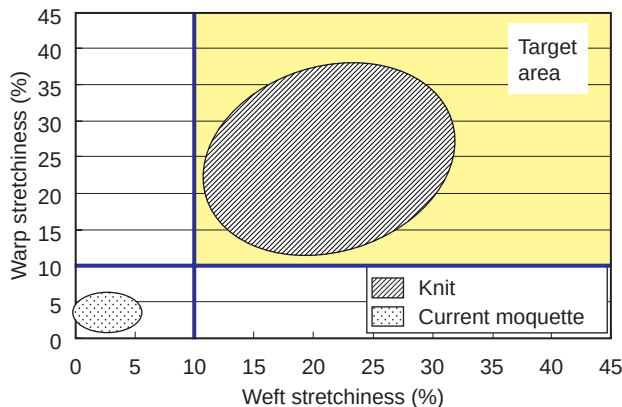


Fig. 6 Comparison of stretchiness in moquette and knit

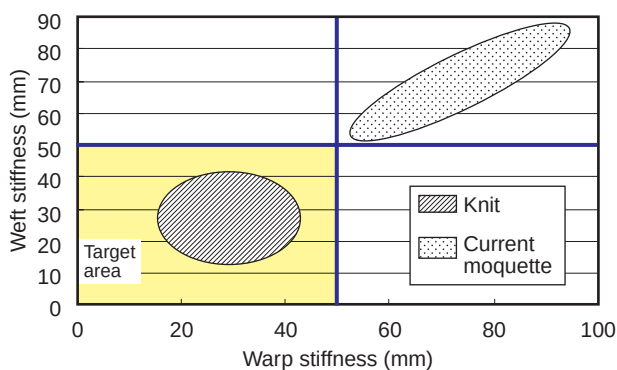


Fig. 7 Comparison of stiffness in moquette and knit

4. Technology Used in the Developed Covering Material and Its Effectiveness

4.1. Technology Used to Reduce Grinning, and Its Effectiveness

The pile construction was modified to achieve the target grinning level enhancement.

When moquette covering material of a general construction is used, gaps always occur at the bottom part of the pile, as shown in Fig. 8. When covering material is used on a flat surface, only the top part of the pile is visible, so the ground yarn cannot be seen.

However, when it is used on a curved surface, the difference in the top and bottom peripheral lengths of the covering material causes the top part of the pile to open up, enabling the ground yarn to be seen through the gaps. This phenomenon is known as grinning.

Although it is possible to reduce the gaps at the bottom part of the pile by increasing the density of the pile yarn, there is a limit to how much the density can be increased. In addition, the feeling of the moquette will be lost, so this is not an appropriate solution⁽³⁾. A method involving the use of crimped yarn for the entire pile also reduces visible gaps, but if the entire pile is made of this yarn, the intrinsic texture of the moquette will again be lost, and crimped yarn existing on the surface causes issues such as dust adherence.

A seat intended for use in an automobile always has sewn parts. The fabric used at sewn parts shows greater curvature than that on a curved surface, and at sewn parts, wrapping of the pile yarn occurs, as shown in Fig. 9. Given these factors, simply increasing the density, the traditional method studied up to now, is inadequate as a countermeasure.

To address these issues, it is necessary to cause only the bottom of the pile to swell. As a method of doing this, the pile yarn was woven into a dual-layer pile. Dual-layer pile yarn consists of several types of yarns, and the difference in shrinkage or crimping between the yarns is utilized. One layer of yarn is shrunken when center cutting during the moquette manufacturing process or the heating process to create a height difference⁽⁴⁾.



Fig. 8 Grinning in moquette

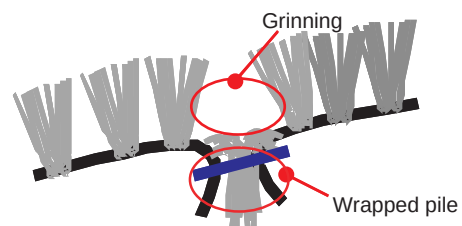


Fig. 9 Illustration of wrapped pile

Figure 10 shows a typical construction of dual-layer pile.

The pile yarn construction used in the developed covering material used a highly crimped yarn as the core yarn, and with a regular yarn as the sheath. To realize this construction, the twisted yarn construction was formed into the covered construction shown in Fig. 11. When the center cut is made, the highly crimped yarn shrinks while drawing in the regular yarn in the vicinity. The shrunken highly crimped yarn is placed at the bottom part of the pile. As a result, only the bottom part of the pile swells, thus reducing the gaps.

Figure 12 shows the cross-sectional structure of the covering material using this construction. By using highly crimped yarn, the bottom part of the pile swells, and adjacent parts of the pile fill up the gaps, thus preventing exposure of the ground yarn.

As shown in Fig. 13, this construction causes the bottom of piles adjacent to the sewn part to swell, preventing exposure of the ground yarn and wrapping of the pile, which occurred with the conventional construction.



Fig. 10 Typical construction of dual-layer pile

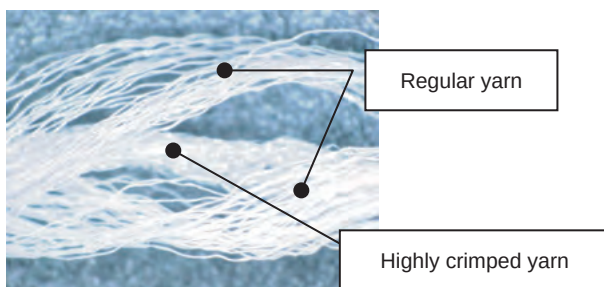


Fig. 11 Grandrelle yarn

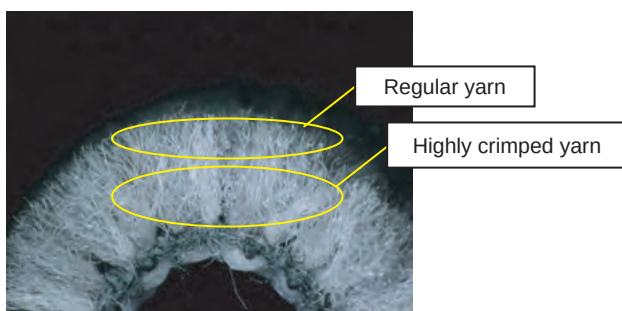


Fig. 12 Two-phase pile

Table 3 Grinning test results

Mixed yarn Covered yarn	Regular	Lightly crimped yarn	Highly crimped yarn
1dtex regular	Level 2	Level 3	Level 4
2dtex regular	Level 2	Level 3	Level 4
3dtex regular	Level 2	Level 3	Level 4

Yarns that were crimped to different extents were used as the core, and then combined with regular yarns of different sheath thickness to form the covering material. The grinning level evaluation results are shown in Table 3. It was found that only covering material that used highly crimped yarn as the core yarn achieved the level 4 target.

4.2. Wrinkle Reduction Technology and Its Effectiveness

To achieve the target stretchiness and stiffness of the covering material for minimizing wrinkles, the ground yarn construction and back coating were modified.

4.2.1. Ground yarn construction

Ground yarn for moquette covering material is often made of mixed PET and rayon yarn, with an eye to such characteristics as ability to resist flames.

However, the stretchability of both PET and rayon as yarn is low, and when it is used to make a seat, its shape conformity is low. Yarns that are well known as having high stretchability include polyurethane elastic yarn and PET elastomer⁽⁵⁾; however, all of these yarns need to be enhanced with respect to strength or other properties when used on their own.

To achieve the target stretchability of at least 10%, elastic yarns and other yarns with stretch effectiveness were studied.

Accordingly, moquette was fabricated using different yarn materials. As a result, as can be seen from Fig. 14, the target stretchiness was achieved by using urethane elastic yarn twisted with mixed PET and rayon yarn for the warp. Likewise, it can be seen in Fig. 15 that the target stretchiness was achieved by using crimped PET yarn for the weft.

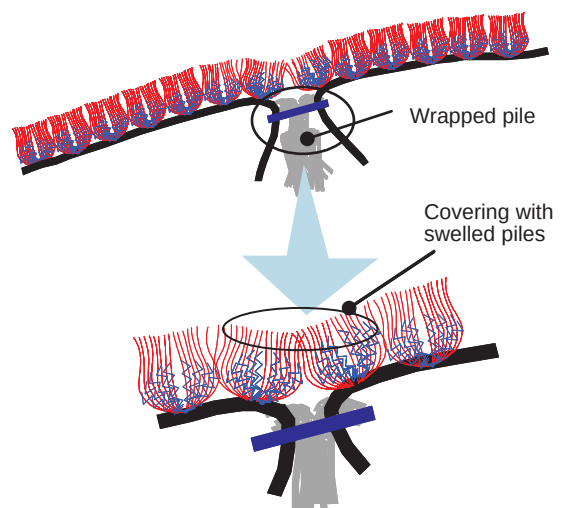


Fig. 13 Mechanism for covering grinning

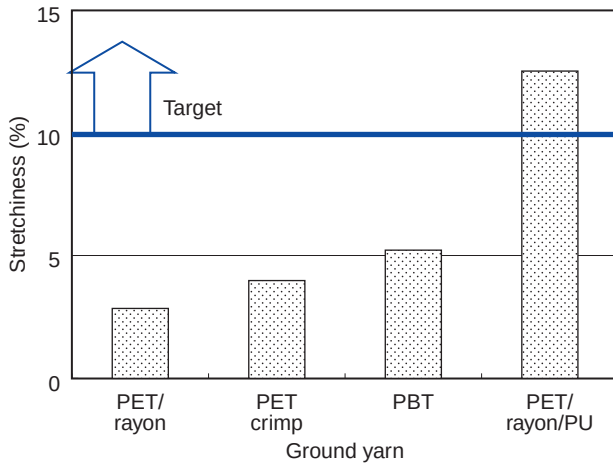


Fig. 14 Comparison of warp stretchiness in grand yarn

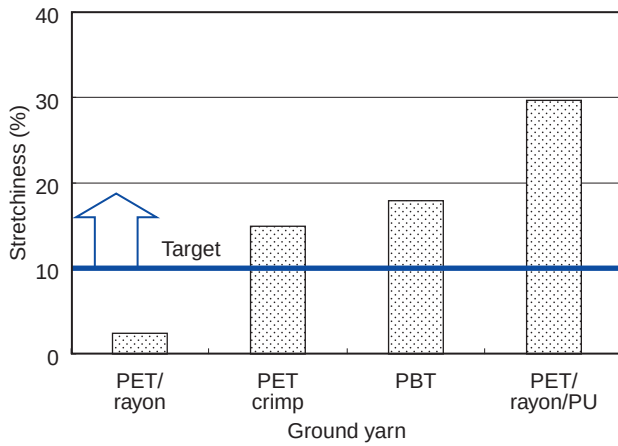


Fig. 15 Comparison of weft stretchiness in grand yarn

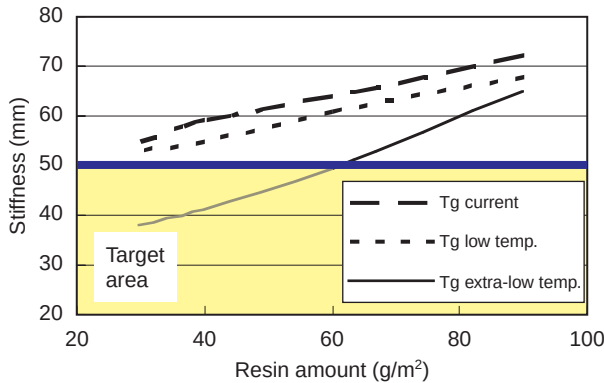


Fig. 16 Relationship between Tg and stiffness

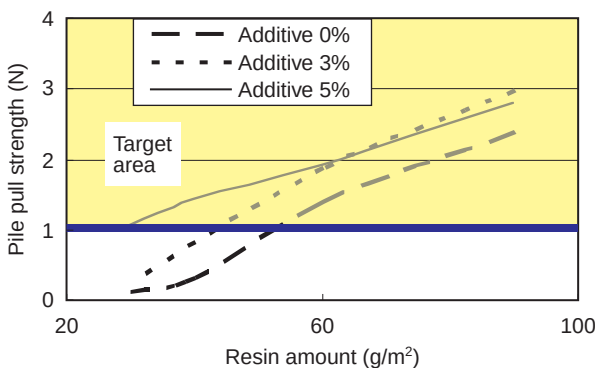


Fig. 17 Relationship between additive and pile pull strength

Table 4 Specifications for newly developed moquette

Category	Current moquette		Newly developed moquette	
	Material		Material	Note
Pile yarn	PET Regular		PET Regular	Textured yarn
			PET Highly crimped	
Ground yarn	Warp	PET/rayon	PET/Rayon/PU	Textured yarn
	Weft	PET/rayon	Crimped PET	
Backcoating	Acrylic resin		Acrylic resin	Max. 50g/m²
Stretchiness	1-5%		10-15%	
Stiffness	Min. 60mm		40mm	
Grimming	Level2		Level4	

4.2.2. Back coating

The back of the moquette is coated with back-coating material to prevent the pile yarn from falling out⁽⁶⁾. However, if the stiffness of the resin material used for the back coating is high, the stiffness of the covering material will increase, causing wrinkles. Accordingly, the coating amount and the stiffness of the resin material were modified to soften the back-coating material.

As shown in Fig. 16, generally there is correlation between the glass transition point of resin material (hereafter Tg) and stiffness: When Tg is lowered, the softness increases. The softness also increases when the amount of resin applied is reduced, but there is a corresponding tendency for the pile yarn pull strength, which is the purpose of the back-coating resin, to fall. Accordingly, a study was carried out with a view to reducing the application amount by raising the permeability of the back coating to widen the adhesion range of the yarn.

Figure 17 shows the correlation between the additive blend and the pile pull strength. From the results of the above studies, the glass transition point of back-coating material was set to extremely low temperature specifications. Addition of 3-5% of additive when blending the resin enabled the softness and pile pull strength to be balanced.

4.3. Application to Seat

The development specifications of this research are shown in Table 4.

The pile used highly crimped yarn and regular yarn to achieve a grinning level of 4. The specifications of the ground yarn were changed, enabling a stretchiness of at least 10% to be achieved. Also, the back-coating material was modified, realizing a stiffness of no more than 50 mm.

A seat was fabricated using these development specifications, and their effectiveness was confirmed. Compared to the conventional specifications, the development specifications produced no wrinkles or grinning at sewn parts, and the resulting seat had a good finish, as shown in Fig. 18.

5. Conclusion

A moquette covering material that was free from wrinkles or grinning and had a luxurious appearance was developed for use on seats.

- (1) A suitable construction to prevent grinning utilized the difference in height between pile yarns resulting from the combined use of yarns with different degrees of crimping. This construction achieved a level of at least that of knitted covering material.
- (2) A warp using PET, rayon and polyurethane elastic yarn, and a weft using PET crimped yarn were suitable to increase the stretchiness. The use of these materials enhanced the stretchiness by at least 10%.

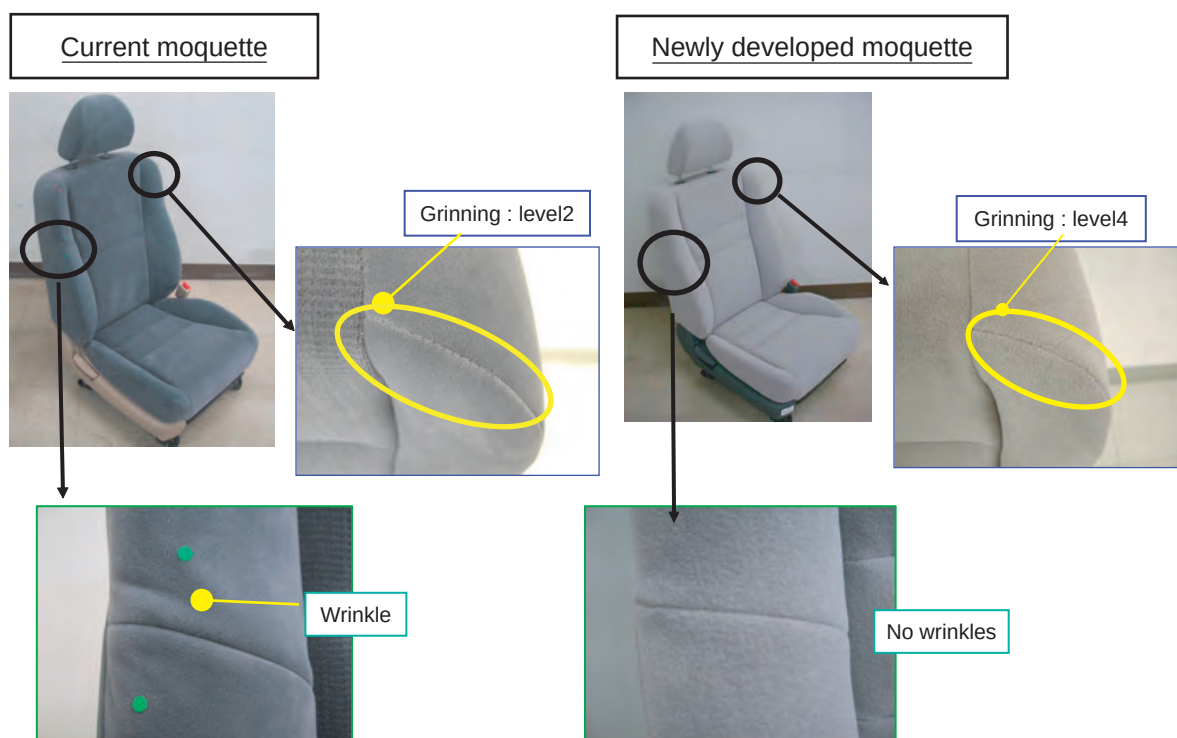


Fig. 18 Effect of newly developed moquette on seat assembly

- (3) Adjustment of the type and quantity of additives decreased the stiffness without reducing the pile pull strength or other physical properties, enabling even a low strength resin to be used.

References

- (1) Yoshikawa, K.: Atarashii Senni no Chishiki, Nagae Syobou, p. 108-111 (1972) (in Japanese)
- (2) Nihon Senni Center: Senni Shikenhon no Subete, Nihon Senni Center, p. 62-74 (1982) (in Japanese)
- (3) Nihon Senni Center: Senni Shikenhon no Subete Ouyou Hen, Nihon Senni Center, p. 63-76 (1982) (in Japanese)
- (4) The Society of Fiber Science and Technology, Japan: Senni Binran Kakouhen, Maruzen Co., LTD., p. 115-391 (1982) (in Japanese)
- (5) The Society of Fiber Science and Technology, Japan: Senni Kakou Binran Genryouhen, Maruzen Co., LTD., p. 287-310 (1982) (in Japanese)
- (6) Masuda, T., Shiozawa, K.: Senni Kakou Gijutsu, Chijin Syokan Co., LTD., p. 8-65 (1971) (in Japanese)

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Development of Low-friction Manual Transmission Fluid

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ABSTRACT

Efforts have been made for some time to lower the viscosity of automotive lubricants as a way to improve fuel economy. Manual transmission fluid (MTF) in particular is expected to increase product appeal by enhancing shift feel at low temperatures. Various studies have also been conducted to lower the viscosity of lubricating oil.

The authors of this study optimized the base oil blend and Viscosity Index Improvers (VII), which are the constituent elements of MTF, with the objective of reducing viscosity at low temperatures without lowering durability at high oil temperatures, which is an issue when attempting to lower the lubricating oil viscosity. The results confirmed that the new oil reduces friction in terms of temperature conversion by 10 °C, compared to that of the current oil, and effectively enhances shift feel at low temperatures. The developed oil was also confirmed to have durability equivalent to that of the current oil, even at high oil temperatures.

1. Introduction

In recent years, as measures for responding to environmental issues have gained public attention, the enhancement of fuel consumption in automobiles, designed to reduce CO₂ emissions, has become a major issue. In line with this, there has been increased demand for drive train lubricating oils that help enhance fuel economy performance^{(1)–(3)}. With manual transmissions (MT), there are several related issues. One is an increase in shift load, caused by the increase in agitation and rotation resistance that accompany an increase in oil viscosity at low temperatures. Another issue is the deterioration in shift feel due, for example, to double bumping.

This means that a manual transmission fluid (MTF) with low friction performance is required.

In response to these issues, Honda developed a low viscosity MTF using high viscosity index (HVI) mineral oils. This low viscosity MTF has been in mass production since 1994. However, because of the increase in friction elements due to the increased use of multiple gears and multicone synchronizer rings, the need for a lower friction MTF has increased. Generally, lowering the viscosity of the lubricating oil is an effective way to reduce friction in the transmission. However, when the viscosity of the oil is drastically reduced, its durability at high temperatures becomes an issue^{(4),(5)}.

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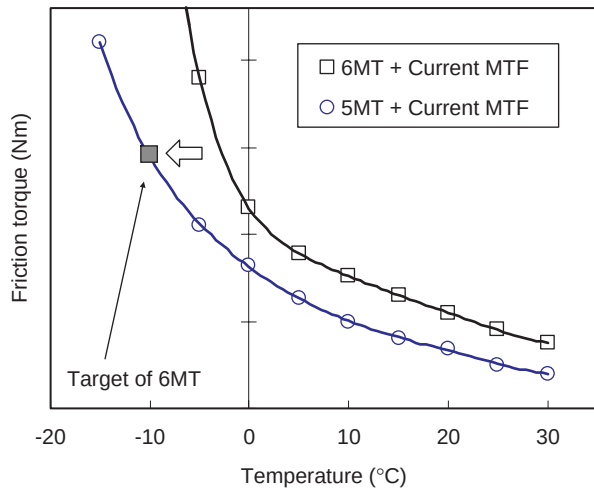


Fig. 1 Characteristic of MT friction torque

Research was conducted on developing an MTF with reduced viscosity at low temperatures that maintains its durability at high oil temperatures. This was accomplished by blending base oils and optimizing the viscosity index improvers (VII). The results are reported in this paper.

2. Development Goals

For this research, the principle focus was on maintaining the function and durability of the MT while lowering the friction of the MTF. The following development targets were set:

(1) Enhancing shift feel at low temperatures

The target of development was to provide a shift feel at an oil temperature of -10°C that was equivalent to the shift feel with the current oil at 0°C . As shown in Fig. 1, this development target was set because to attain a shift feel in a 6-speed MT, which has many friction elements, that was equivalent to that in a 5-speed MT, which has few friction elements, the amount of friction reduction necessary was slightly less than 10°C in the low temperature region of 0°C or less.

(2) Ensuring transmission function and durability

With respect to the durability of the gears and the bearings, and the function and durability of the synchronizer, which is unique to MTs, the development target was set to ensure performance equal to that provided by the current oil.

3. Method of Attaining the Targets

Based on the development targets for the function and durability of the MT, the targets for the MTF were set as shown in Table 1, and research on base oils and VIIs was carried out.

3.1. Selection of the Base Oils and VIIs

Base oils can be broadly classified into two types: synthetic oils and mineral oils. Among these, it is known that when compared in terms of identical base oil viscosities, the viscosity indexes of the synthetic oils are higher than those of the mineral oils. Esters and poly- α -olefins (PAOs), which have a structure identical to that of mineral oils, are representative synthetic oils. Because esters affect rubber components and the solubility of the performance additives, it would be difficult to use them in vehicles that have already been sold. Therefore, base oil was investigated using PAOs. Research

Table 1 MTF development target

Properties	MTF development target
Low-temperature viscosity (-10°C)	Equal to current MTF kinematic viscosity at 0°C
High-temperature viscosity (100°C)	Equal to current MTF kinematic viscosity at 100°C
Shear stability	Equal to current MTF

Table 2 Formulation of tested oil

Test oil	Base oil	Viscosity Index Improvers (VII)
Current MTF	Mineral oil High-viscosity	Low molecular weight type
A	PAO High-viscosity	High molecular weight type
B	PAO High-viscosity	Low molecular weight type
C	PAO Low-viscosity	High molecular weight type
D	PAO Low-viscosity	Low molecular weight type

was carried out using two types of VII: one with high molecular weight and the other with low molecular weight.

As shown in Table 2, test oils were made by varying the blend of the base oil and the VIIs, and formulations that fell within the temperature targets were studied. The results of the temperature characteristics for the test oils are shown in Fig. 2. Formulation C fell within the targeted viscosity range. Formulations B and D both used a low molecular weight VII. The low temperature viscosity of Formulation B deviated greatly from the target values. While Formulation D fell within the target values for low temperature viscosity, viscosity at high temperatures decreased. Formulation A did not satisfy any of the target values.

Shear stability was also confirmed for the above formulations. The fluctuation rates for viscosity during the shear stability test are shown in Fig. 3. The shear stability tests were carried out using ultrasonic tests. Only Formulation D fell into the target area for shear. Formulation C, which satisfied the viscosity target in Fig. 2, exhibited the greatest viscosity fluctuation, and did not satisfy the target values. Formulations B and A also did not satisfy the target values. Formulations A and C, which exhibited high viscosity

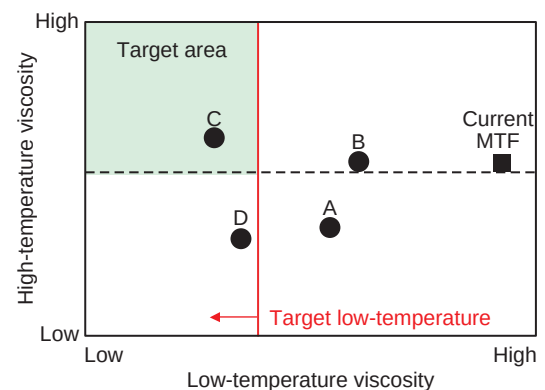


Fig. 2 Viscosity characteristics of tested oil

fluctuation due to shear, used high molecular weight VIIs, and from these results it can be understood that the reduction in viscosity can be lessened by using a low molecular weight VII. When considering lubrication function in a drive train, an MTF that exhibits little fluctuation in viscosity over time is preferable in terms of maintaining performance. Thus, the low molecular weight VIIs were used for the formulation specifications.

It can be understood from the above results that a formulation that satisfies the target range including fluctuation in viscosity due to shear is not possible. To attain the targeted low temperature viscosity, it is necessary to use a base oil with a low viscosity and to reduce the kinematic viscosity at 100 °C with respect to the initial target. Because the development target for an MTF at high temperatures was to achieve a durability equivalent to that of the current MTF, it was decided to examine a procedure in which durability would not deteriorate even when the kinematic viscosity at 100 °C was reduced. A low viscosity base oil and a low molecular weight VII were used as a base for this procedure. As shown in Fig. 4, when selecting low viscosity base oils, there are mineral oils, such as HVI mineral oils, that have viscosity indexes equivalent to those of PAOs. Considering availability and cost, investigation of the base oils was carried out using mineral oils.

3.2. Setting High Temperature Durability Targets

Although the development target at high temperatures was to maintain durability equivalent to the present durability, reducing the kinematic viscosity at high temperatures may shorten the fatigue life of gears and bearings. Therefore, in the present research, durability was evaluated in terms of gear fatigue life.

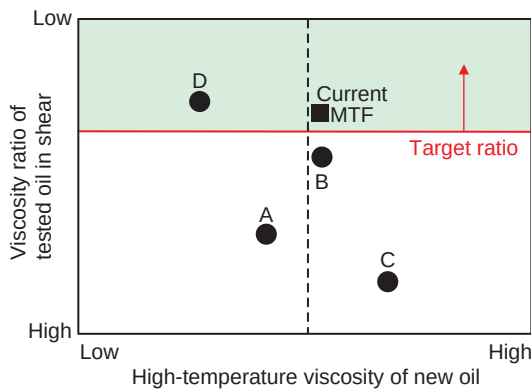


Fig. 3 Viscosity decrease after shear test

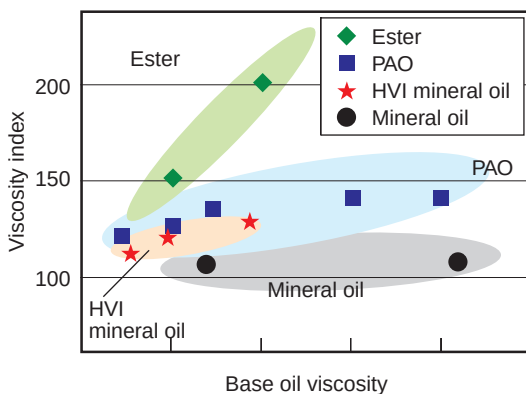


Fig. 4 Viscosity index characteristics with differences in base oil viscosity

3.3. The Mechanism of Gear Fatigue Life

One form of damage to a gear is pitting, and pitting occurrence life serves as a measure for evaluating fatigue life. There are various theories that explain the mechanism behind pitting. In the oil penetration theory⁽⁶⁾, as shown in Fig. 5, oil pockets that form a closed system occur between two surfaces when two gears that have fine asperities mesh. In this situation, the oil penetrates into micro-cracks present in the gear surface, the cracks propagate due to the high oil pressure generated when the gears mesh, and finally surface peel, that is, pitting, occurs.

Therefore, in order to extend the pitting life, it is thought that interposing an oil film to relatively reduce metal-to-metal contact, and suppressing the propagation of micro-cracks are important. Based on the above idea, parameter studies were carried out.

3.4. Development and Testing of Gear Durability

To help ensure the durability of a gear, the influence of the viscosity of the base oil and of the high viscosity base oil blend were clarified.

The IAE gear tester shown in Fig. 6 was used to test gear durability, and the durability was evaluated in terms of time until

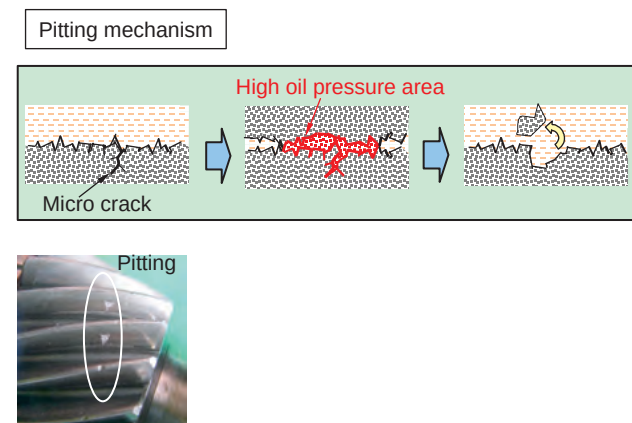


Fig. 5 Mechanism of pitting on gear

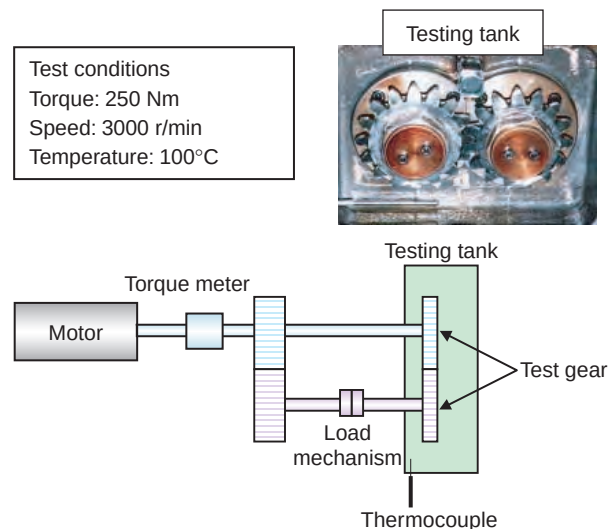


Fig. 6 IAE gear tester and durability test conditions

Table 3 Test piece specifications for durability test

	Current T/P	New T/P
Number of teeth drive/driven	15/16	30/32
Module	5.00	2.65
Face width (mm)	5.2	12.7
Material	SNCM415	SCr420
Heat treatment	Carburizing	Carburizing
Tooth surface finishing	Grinding	Shaving
Tooth surface PV (N/mm ² ·m/s)	9660	11900

pitting occurrence. As shown in Table. 3, a gear test piece (T/P) was specially prepared, taking into account the dimensions of an actual gear.

(1) Influence of the base oil viscosity

The durability of the gears was confirmed for test oils E, F, and G, which have the same kinematic viscosity as the product, by varying the base oil viscosity using a single grade HVI mineral oil formulation. As a result, it was understood that the fatigue life of the gear was extended as the base oil viscosity increased (Fig. 7).

(2) Influence of a high viscosity base oil blend

Test oil H was a blend of high viscosity base oils and had the same base oil viscosity and product kinematic viscosity as test oil G, which was a 100% HVI mineral oil. Both test oils were used to examine the durability life of the gear, and it was found that the test oil H extended pitting life (Fig. 7).

This method of blending high viscosity base oils is widely and empirically known. There have been a variety of reports about the mechanism by which it works^{(7),(8)}, and in fact it is thought that depending on various conditions, multiple phenomena are complexly interlinked. Under these circumstances, as shown in Fig. 8, we considered that because the oil film is ensured due to the three-dimensional structure of the high viscosity base oil, the number of contacts of micro-structures between surfaces that accompany sliding is reduced.

From the results of the above tests, a base oil viscosity that could maintain durability was set by preparing a high viscosity base oil while setting the base oil viscosity to the highest possible value that satisfied the low temperature viscosity.

3.5. Results of Confirming High Temperature Durability of the Developed MTF

The high temperature durability of the low viscosity MTF formulated based on the results of the tests described above was compared to the current MTF, and confirmed. The results are shown in Fig. 9. The developed MTF has a life equivalent to the pitting occurrence life of the current MTF, and a duration life equivalent to current oils was obtained even after lowering the viscosity.

In addition, duration testing of a gear was carried out for the MTF recovered after completing the duration bench test using a transmission. As a result, it was confirmed that the durability was 1.5 times the current oil degradation life, and was better than current oils after duration.

It was thus confirmed that the developed MTF is a formulation that helps ensure an oil film that does not reduce the durability even at a low viscosity.

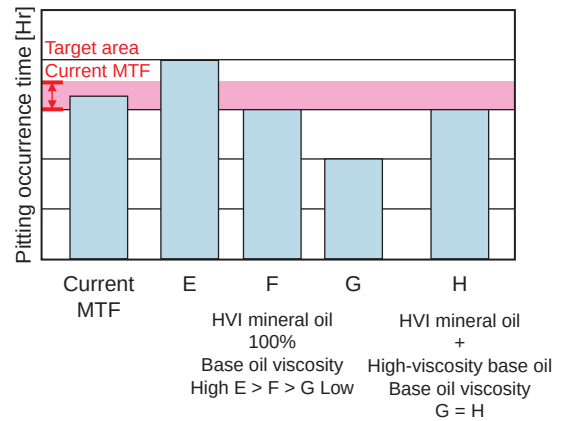
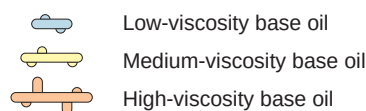


Fig. 7 Pitting durability characteristics by different oil specifications

Molecular structure image



Oil film performance

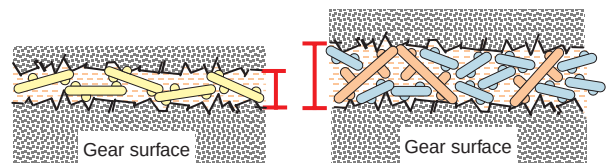


Fig. 8 Mechanism of oil film performance enhancement

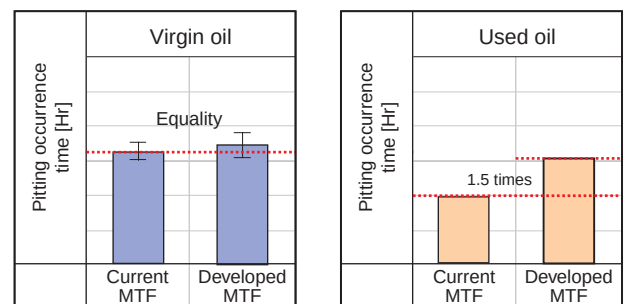


Fig. 9 Comparison of pitting durability by MTF

4. Transmission Test Results

4.1. Friction Reduction Effect

By lowering the viscosity, agitation resistance in the transmission was reduced, and a reduction in friction torque can be expected.

The effect of reducing friction during low temperatures by reducing the viscosity of the developed oil was confirmed by a bench test using a compact vehicle-class 6-speed MT. Under test conditions of 25 km/h speed, no-load, and neutral gear, the change in friction torque with respect to the transmission oil temperature

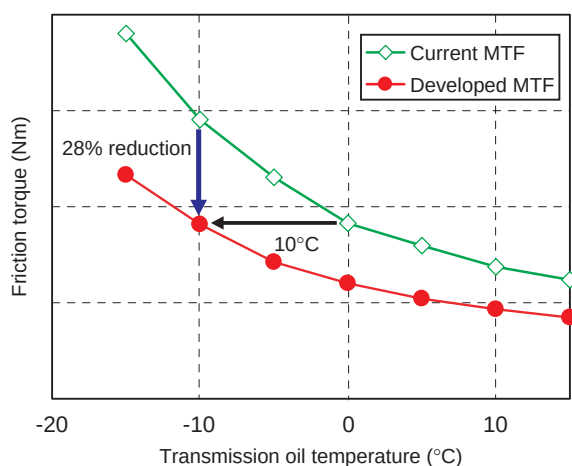


Fig. 10 Comparison of transmission friction torque

was measured. The results of the measurement are shown in Fig. 10. The developed oil was compared to the current oil, and a 28% reduction in friction torque and a friction reduction effect of about 10 °C in terms of the oil temperature were confirmed.

4.2. Shift Feel Enhancement

In MT, an increase in friction torque due to an increase in viscosity at low temperatures may lead to a reduction in the double bump feel and an increase in shift operation load. These are factors related to a deterioration of shift feel.

The effect of enhancing shift feel by lowering the viscosity of the developed oil was confirmed in a compact vehicle-class 6-speed MT vehicle. In the test conditions, the speed was fixed at 20 km/h, and the double bump load when shifting from first speed to second speed and shift operating load when shifting from second speed to first speed were measured.

As a result, in comparison to the current oil, it was confirmed that for the developed oil, at 0 °C, the double bump load was reduced by 18%, the shift operating load was reduced by 15%, and a reduction effect of about 10 °C in terms of oil temperature was confirmed.

4.3. Durability at High Temperature

Durability at high temperatures was the main issue when lowering the viscosity of the lubricating oil. In addition to the evaluation of the gear unit described above, durability evaluation using a transmission was carried out. It was confirmed that the newly developed MTF attains characteristics that are equivalent to or better than the current oils, satisfies in-house standards for the metal fatigue life of the gear and bearings, and has sufficient durability under field conditions.

In addition, in relation to the synchronizer ring, which is an important structural element unique to MTs, it was confirmed that the influence on synchronization performance and friction characteristics equals those of the current oil.

5. Conclusion

- (1) By blending high viscosity base oils with low viscosity base oils and optimizing the VIIs, durability at high temperatures was achieved and the viscosity was lowered at low temperatures.

- (2) The results confirmed that at 0 °C the friction torque in terms of temperature conversion was reduced by 10 °C in comparison to the current oil, and thereby the shift feel was enhanced.
- (3) It was confirmed by the results of unit testing and engine testing that the durability at high temperatures, which is an issue when lowering the viscosity, maintained a performance equal to the current oil.

References

- (1) Nagashima, T., Saka, T., Sasaki, T., Ohmori, I., Furumoto, M., Tanaka, H.: Development of Automotive Lubricants Based on High Viscosity Index Base Stock, HONDA R&D Technical Review, Vol. 7, p. 123-130
- (2) Tanaka, H., Nagashima, T., Sato, T., Kawauchi, S.: Effect of 0W-20 Low Viscosity Engine Oil on Fuel Economy, HONDA R&D Technical Review, Vol. 11, No. 2, p. 143-148
- (3) Yamane, T., Nagashima, T., Yoshida, J., Hirao, Y.: Development of Long Life Continuously Variable Transmission Fluid, HONDA R&D Technical Review, Vol. 12, No. 2, p. 153-158
- (4) Ito, M., Kurosawa, O., Matsuoka, T., Matsui, S.: The Evaluation of the Fuel-Economy Performance of Low-Viscosity Drive-Train Lubricants and the Development of Oils with Improved Fatigue Life, Power train & Fluid Systems 2004-01-3029 (2004)
- (5) Matsui, S.: Research Trends of Fuel Economy ATF, ENEOS Technical Review, Vol. 47, No. 1, p. 24-29 (2005)
- (6) Nakajima, A.: Effects of Lubricating Oil on Rolling Contact Fatigue Damages, Tribologist, Vol. 48, No. 10, p. 789-795 (2003)
- (7) Hirano, F.: Effect of Molecular Weight Distribution of Mineral Oil on Tribofailure of Spur Gears, American Society of Mechanical Engineers (1980)
- (8) Murakami, T., Sasaki, T., Yamamoto, Y.: Effect of Molecular Weight Distribution of Mineral Oils on Lubricating Performance under Sliding Contact Conditions, JSLE INTERNATIONAL TRIBOLOGY CONFERENCE (1985)

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Development of Automotive Shredder Residue (ASR) Recyclate Application

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ABSTRACT

Recently, with increased concerns about global environmental issues, European car manufacturers have been using thermoplastic recyclate resin and natural raw materials for application to automotive components in order to make effective use of the limited oil resources.

In Europe, where the demand for application of recyclate is growing stronger, attention is being turned to ASR (Automotive Shredder Residue) coming from disused cars. Polypropylene, separated and extruded from automotive shredder residue, has been used to develop recyclate with an excellent balance between rigidity and impact performance, that meets the requirements for under body components. This material has been applied to 2006 model year CIVIC in Europe.

Cost has been reduced with these technologies, whilst making effective use of oil resources.

1. Introduction

The End-of-Life Vehicles (ELV) Directive came into force in Europe in 2000 and sets overall targets of 80% recycling and 85% recovery by 2006, and 85% recycling and 95% recovery by 2015⁽¹⁾. Currently the reuse and recycling levels in Europe vary between 80-85%. The recycling mainly includes ferrous and non-ferrous metals and some plastic parts which can be dismantled. The remaining automotive shredder Residue (ASR) is mainly going to landfill. To achieve the ELV directive the total amount of waste that can be disposed of should be reduced to 5%. New ways to treat and recycle ASR are fundamental in achieving this. Rising landfill cost in Europe also contributes to treatment of the ASR becoming more important.

These new ways of recycling aim at more cost effective ways of recycling plastics from automotive waste. One way is to look at the plastic content of ASR and technologies that can be applied to recover and recycle that waste.

Traditional ways of recycling are mainly aimed at pre-shredder recycling, where parts are dismantled before recycling and is only economical viable for large plastic mouldings. In post shredder recycling plastics are separated and recovered after the shredder process (from ASR) and is economical viable for recovery of plastics from large and smaller parts. This paper considers Honda's activities in Europe developing ASR recycled polypropylene grades for mass production application to 2006 model year CIVIC in Europe.

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2. History of Honda Recycling in Europe

Honda is at the forefront of environmental awareness and aims to minimize the environmental impact of its activities. To this end Honda promotes activities to enhance the recyclability of cars and the use of recycled materials. Recycled materials are used instead of virgin materials in order to reduce use of natural resources. Materials that were wasted are once again used in the production for new parts. For example, in the case of bumpers consistently polypropylene has been used which can be easily recycled. Honda has applied plastic recyclates recovered from repair bumpers for over ten years in Japan and Europe^{(2), (3)}.

3. Automotive Shredder Residue

3.1. Automotive Shredder Residue Composition

Automotive Shredder Residue (ASR), is the either organic or inorganic residue from which metals have been removed during the shredder process. The composition of ASR is mainly resin,

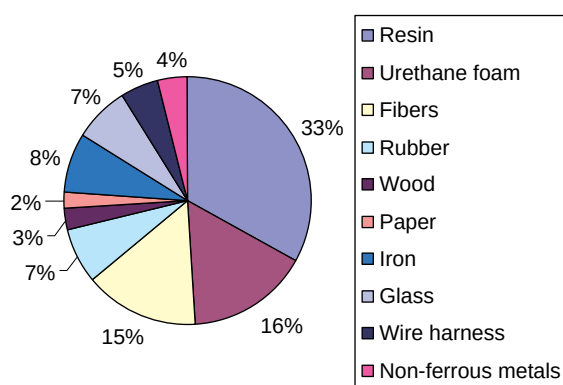


Fig. 1 Automotive Shredder Residue (ASR) composition

polyurethane foam, fibres, rubber and glass (Fig. 1) and comprises up to 25 percent of every vehicle, which means that, without the ability to effectively recycle such materials these potentially end up in the solid waste stream. The increase in the amount of residue is especially attributable to the plastics content, as its use has significantly accelerated due to weight and cost reduction measures. The separation of thermoplastics from ASR on an industrial scale can be described in three basic steps (Fig. 2)⁽⁴⁾:

- (1) Shredder process
- (2) Post shredding sorting
- (3) Plastic separation

The first step is the shredder process where the ferrous metals are removed. The remaining non-ferrous (NF) mix and fluff are fed into the Shredder Residues Treatment Line (SRTL) where the ASR is separated.

3.2. ASR and Plastics Separation

The ASR separation process starts with the metal reclaiming operation, which grinds the ASR to around 25 mm bits. These go through a large trommel, or rotating cylindrical screen, for classification into a light thermoplastic fraction and a heavy thermoset fraction, next air classification removes textiles and PUR foam. The remaining thermoplastics and thermoset rubbers go through eddy-current separators to remove copper. The thermosets can be used as fuel in the operation, further maximising process efficiency. The plastics are then put through a series of four modified dense media (DM) separation drums containing liquids of different densities⁽⁵⁾. In this continues process the plastics are separated in four different fractions, including the concentrate of thermoplastics which is treated as a separate stream⁽⁵⁾. The following thermoplastics are being recovered from the pre-concentrate:

- Olefins: Polypropylene (PP), Polyethylene (PE) and talc-filled Polypropylene (PP-T);
- Styrenics: Polystyrene (PS) and Acrylonitrile Butadiene Styrene (ABS).

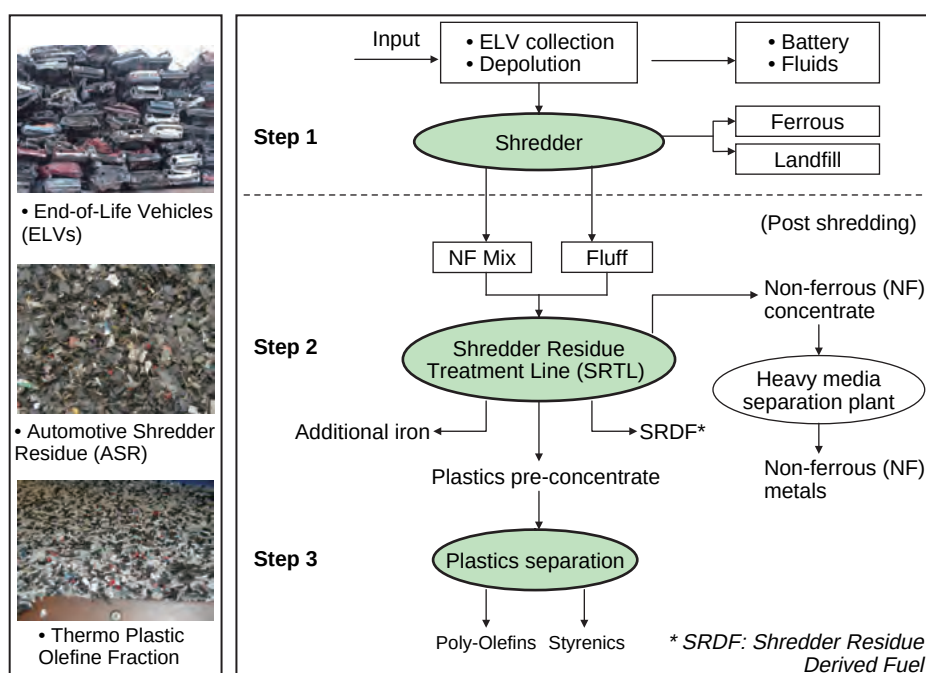


Fig. 2 Automotive Shredder Residue (ASR) Separation

The high accuracy in density separation of the thermoplastics pre-concentrate is achieved with a hydraulic static separator using heavy media. The pre-concentrate is subjected to a static dense media dissociation that uses large vats, where materials between 0.9 and 1.5 cm³ density are separated with an accuracy of 0.002 g/cm³. Heavy fractions are removed by augers from the bottom while light fractions are skimmed off the top. Compared to dynamic separators such as cyclones, there is practically no mix produced.

PP is isolated from PE and PS while ABS and PP-T are separated from the remaining plastics of 1 to 1.1 g/cm³, which are landfilled⁽⁶⁾.

In the final compounding operation, fast moving conveyors bring the mix of plastic chips from the ASR resource to continuously feed the compounding extruders (Fig. 3)⁽⁶⁾. At this stage before extrusion the ASR recycle material can be mixed with other treated material from recycling sources like bumpers or bottle caps to achieve the required target material specification, but in all cases 100% recycle grades are produced. The material is pelletized, packaged and shipped to injection moulders.

4. Concept of Development

4.1. Objectives

The direction of Honda in Europe reflects the current European trend to continue the increase in application of plastic recyclates.

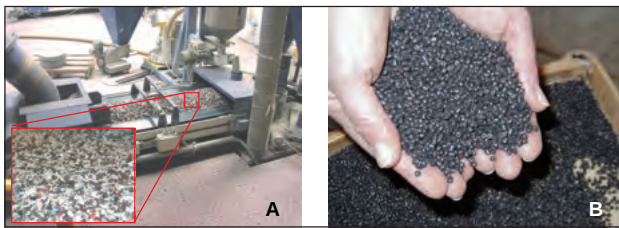


Fig. 3 Mixed polypropylene from ASR resource (A) feed compounding extruders and final product after pelletizing (B)

The objective in this development is to:

- increase the application of polypropylene recycle;
- introduce the application of ASR derived recycle;
- achieve material cost reduction.

Target parts for this development are Inner Fender and Floor Under Cover for European 2006 model year CIVIC (Fig. 4). These parts combined represent a significant weight and therefore opportunity to introduce ASR recycle and achieve a considerable cost reduction.

Both parts are under body components and have similar requirements with regard to impact performance, while other requirements like for example for stiffness are different and more specific dependent on part shape and assembly method.

Honda ASR recycle grades have to be developed for both applications based on achieving part performance suitable for the EU market. A material specification based on 100% recycle application specific for Inner Fender and for Floor Under Cover had to be established. The recycle grades for both parts had to achieve a balance of impact performance and flexural modulus required to meet part performance. Sufficient mouldability (melt index) is also required especially considering large surface area of Floor Under Cover and complex shape of Inner Fender.

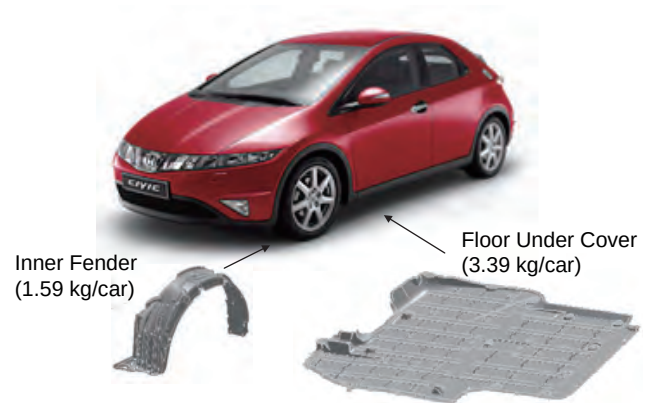


Fig. 4 Floor Under Cover and Inner Fender for 06 model year CIVIC

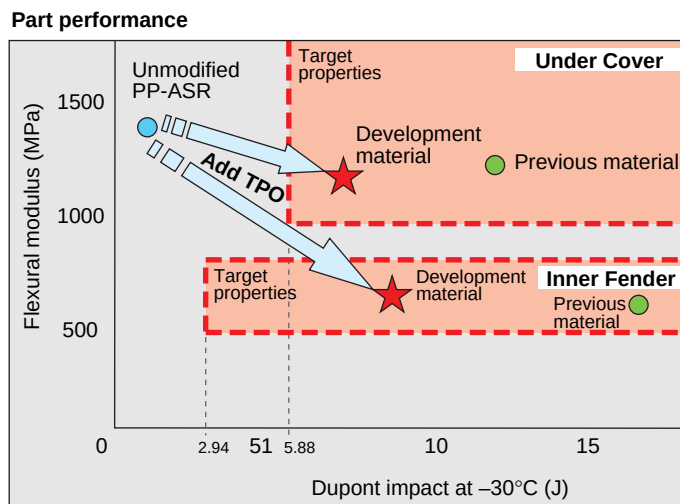


Fig. 5 Development target properties for Inner Fender and Floor Under Cover based on part performance

Table 1 Floor Under Cover and Inner Fender development material (part evaluation)

Item	Unit	Development			
		PP-ASR	Material A	Material B	Material C
				Floor Under Cover	Inner Fender
Recyclate content		100% PP ASR	90% PP ASR 10% TPO	70% PP ASR 30% TPO	60% PP ASR 40% TPO
Melt index	g/10min	4.25	6.3	7.3	8.6
Flexural modulus	MPa	1420	1340	999	540
Dupont impact strength (−30°C)	J	>1.47	>3.92	>9.8	10.8

Table 2 Deflection at high temperature of Floor Under Cover

Test condition	Splash Shield 2001 model year CIVIC	Floor Under Cover 2006 model year CIVIC
	Honda Bumper Recyclate	Material B
80°C	28mm max.	17mm max.
100°C	—	26mm max.

4.2. Under Cover and Inner Fender

Under Cover is applied for the first time to 2006 model year CIVIC to enhance aerodynamics, because of the large surface area of Floor Under Cover part stiffness has to be sufficient. Impact performance is important for under body parts to resist stone impact, especially at low temperature.

In Europe Honda has no experience with the application of recyclate for Inner Fender. Similar to Floor Under Cover good impact performance is required. Part stiffness, however has to be lower to allow enough flexibility during the assembly process.

The flexural modulus for Inner Fender was also too high to allow for sufficient part flexibility. Minimum flexural modulus, however, is required for Floor Under Cover to guarantee part stiffness which is especially critical at higher temperatures.

A specific grade with target properties had to be developed for both applications by compounding the PP-ASR with TPO recyclate to enhance the properties to target level (Table 1).

5. ASR Recyclate Development

5.1. Properties of ASR Recycled Polypropylene

The polypropylene derived from ASR recycling was evaluated first. This material was the basis from which the grades for both Inner Fender and Under Cover were developed. Secondly this material was enhanced in order to meet the specific target requirements. The correct balance of minimum impact performance and flexural modulus had to be achieved. Actual part performance was confirmed and compared to existing mass production parts, after which the recyclate material specifications could be fixed. Noticeable is the relative high rigidity and low impact performance of unmodified PP-ASR compared to the current grades applied for Inner Fender and Splash Shield. The target was to modify the PP-ASR to meet the requirements for each specific part and achieve a 100% recyclate application (Fig. 5).

5.2. Compounding of ASR Recycled Polypropylene (TPO Application)

Part requirement for impact performance was evaluated by Dupont impact at −30 °C. Unmodified PP-ASR could not meet this requirement for either application.

5.3. Part Test

Confirmation of part performance for 2006 model year CIVIC was required after the basic material composition had been established. For both applications part stiffness and impact performance were evaluated on actual parts in ASR recyclate and previous mass production material, HD/LD PE for Inner Fender and Honda bumper recyclate for Splash Shield, allowing a direct comparison between current and developed technology to be made.

Part stiffness of Floor Under Cover in Material B has been tested at high temperature and compared to mass production Splash Shield and the maximum deflection evaluated to specification for under body applications. The result for Floor Under Cover meets the development target and is better then achieved for Splash Shield on previous model (Table 2).

The flexural modulus, critical for Inner Fender, was achieved by compounding PP-ASR with 40% TPO (Material C) and has resulted in flexural modulus comparable with HD/LD PE. Evaluation of the Inner Fender on actual vehicle shows that sufficient part flexibility has been achieved.

Impact performance has been evaluated and compared with competitor OEM parts for both Inner Fender and Floor Under Cover by testing at −30 °C on actual parts. The resulting impact performance for both parts in ASR recyclate is similar to this of competitor OEMs and is acceptable for the European market (Table 3).

Table 3 Dupont impact (-30°C) of Floor Under Cover and Inner Fender

		Test result		
		Honda	Competitor A	Competitor B
Dupont impact (-30°C) (J)	Floor Under Cover	5.8 (>3.43)	2.94 (>1.96)	5.8 (>2.94)
	Inner Fender	9.8 (>5.8)	3.92 (>2.45)	6.68 (>4.9)

Table 4 Heavy metal analysis ASR recyclate grades

Element	ELV Directive limits (2000/53/EC)	Total heavy metal analysis ppm total element per material	
		Material B	Material C
Antimony	Not applicable	0.5	1.0
Arsenic	Not applicable	<0.5	<0.5
Cadmium	100 ppm	10	1.5
Chromium	1000 ppm	4.5	2.5
Lead	1000 ppm	15	10.0
Mercury	1000 ppm	<0.5	<0.5

5.4. Process Control

The source of ASR recycled polypropylene is ELV components where different polypropylene grades are applied. The properties of these input materials (i.e. ASR-PP, TPO) used for compounding the ASR recyclate grades can vary depending on the source material. It is important that properties of the compounded materials meet specification for each production batch and are stable over a long period to guarantee mass production quality. In the mass production process the input material and ratios are varied to control the material specification. The following procedure is followed:

- (1) The quality of the input materials is confirmed after the separation process.
- (2) The compounding ratio and input components to meet specification are established and fixed for batch production.
- (3) Input materials and additives are mixed at the extrusion line.
- (4) Material is extruded and pelletized, quality controlled online (melt index).
- (5) Final specification confirmation before shipment (certificate of conformity).

For each batch production this process is repeated. The recyclate grades (compounding ratio) for Inner Fender and Under Cover can therefore vary for each batch production to guarantee the specification.

For example shrinkage of the recyclate has to be stable to allow for assembly and fit and finish. Shrinkage data has been evaluated for the ASR grade over a one year period (Fig. 6). The result shows it is controlled within $\pm 0.25\%$ which is sufficient tolerance for mass production.

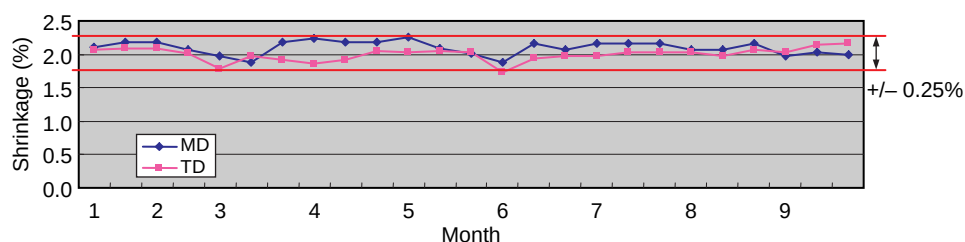


Fig. 6 Shrinkage variation over time

5.5. Contamination

The ASR and plastic pre-concentrate separation process has shown to be highly accurate (high tolerance). The level of contamination in produced single plastics streams is low. A purity of minimum 99.8% is achieved for the polypropylene grades recycled from ASR. Annex II of Directive 2000/53/EC limits the use of heavy metals lead, hexavalent chromium, mercury and cadmium to a maximum concentration per material⁽¹⁾. Especially in the light of this Directive it is important that no banned substances are present in the recyclate grades so contaminated materials are not introduced to new models.

Separate analysis was done of development grades, which confirms that all heavy metal contaminations are well within the limits set in the Directive (Table 4).

6. Cost Benefit

The cost reduction achieved for the developed ASR recyclate grades compared to existing mass production materials is shown in Fig. 7.

For Floor Under Cover a cost reduction of 60% per kg has been achieved by applying ASR recyclate compared to the Honda Bumper recyclate. Honda Bumper recyclate is continued to be applied for new CIVIC Cover Assy Front Bulkhead Cover. The cost reduction for Inner Fender compared to the previously applied material is 12%. The cost of ASR recycled plastics is relatively stable and independent of fluctuations in oil prices.

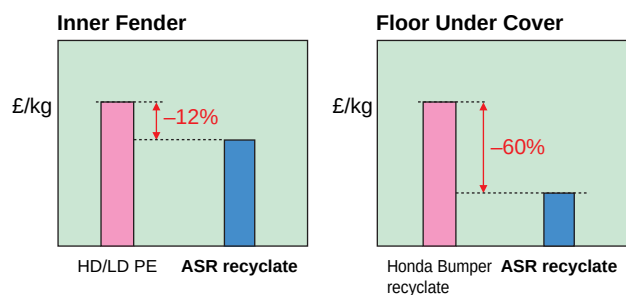


Fig. 7 Material cost reduction achieved for ASR recyclate application

7. Conclusion

This development has shown that 100% recyclate grades from ASR could be developed successfully for Inner Fender and Floor Under Cover and applied for 06 model year CIVIC in Europe.

- (1) Plastics can be separated with high accuracy from ASR, leaving low contamination;
- (2) By modifying polypropylene recycled from ASR with recycled TPO the balance of properties could be achieved for Inner Fender and Under Body application;
- (3) Significant cost reduction has been achieved; 60% material cost down for Floor Under Cover and 12% for Inner Fender;
- (4) Total recyclate application on 2006 model year CIVIC has been increased;
- (5) The compounding process applied takes into consideration the varied quality of input materials and is based to guarantee material specification.

This development has shown the potential of ASR recycled polypropylene. It opens the way to look at other applications of plastics recovered for ASR contributing to more cost reduction and less dependence on natural resources.

References

- (1) Directive 2000/53/EC of the European Parliament and of the Council on End-of Life Vehicles, Official Journal of the European Communities L269/34 (2000)
- (2) Hirawaki, S., Hikita, K., Uemura, T.: Expansion of Applicable Scope for Use of Recycled Bumper Materials by Modification of Properties, Honda R&D Technical Review, Vol. 15, No. 1, p. 227-232
- (3) Takeuchi, A., Makuta, M., Ohgane, H., Hamabe, K., Aoki, O.: Development of the Bumper-to-Bumper Recycling Technology, SAE Technical Paper Series, 960282 (1996)
- (4) François, O.: Mass-Balance in Post-Shredding Technology: Results of a Trial Based on the Shredding of 201 ELVs, IARC (2003)
- (5) ESR International L.L.C.: Dense Medium Separation (2005)
- (6) Feraudy de, H.: Automatic and Industrial sorting for ASR thermoplastics at Galloo Plastics SA at Halluin Plant, IARC (2005)

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T. Yamaguchi

Research on Automatic Inspection of Spot Welding Using Ultrasonic Testing Methods

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ABSTRACT

This paper discusses research on inspection procedures and techniques for the identification of spot welding indentations ("spot indentations" below). The goal was to increase the reliability of ultrasonic testing methods by focusing on the procedures that are used by experienced inspectors.

A mechanism that is able to apply a uniform pushing force while optimizing the inclination of the probe with respect to the area being inspected has been developed. With this new system, the center of the spot indentation can be identified even when its outline is difficult to determine.

The system was fitted onto an industrial robot ("robot" below), and tested on the inspection of automotive parts. The results indicated that the system achieved the same level of accuracy as that of an experienced inspector.

1. Introduction

High-strength steel sheets are increasingly being used to respond to requirements for automotive weight savings. Spot welding is the preferred technique for joining these sheets; furthermore, 4000 to 5000 spot welds are used in each vehicle. Tensile shear strength, an indicator of spot weld strength, is shown to increase proportionally with the tensile shear strength of the base, high strength steel material. Both destructive and non-destructive inspection can be used to ensure spot weld quality. At present, the random application of chisel tests is the most common. However, as tensile strength increases, destructive chisel tests are beginning to require significantly more effort.

The main non-destructive inspection methods are radiography, ultrasonic testing, magnetic particle inspection, die penetrant and magnetic induction. As Table 1 shows, ultrasonic testing is the most applicable of the non-destructive tests employed to detect welding defects in thin sheets utilized in automotive applications⁽¹⁾. The use of ultrasonic testing as a non-destructive inspection method is particularly prevalent in Europe⁽²⁾. In Japan, non-destructive inspection per this method has aroused intense interest as a result of its successful application at companies capitalized by or involved in technological cooperation with overseas manufacturers.

Table 1 Applicability of nondestructive test methods in 6 mm sheets

Object	Metal steel	Nonferrous metal
Main nondestructive test methods		
(a) Radiographic test	Not applicable	Not applicable
(b) Ultrasonic flaw detection	Applicable	Applicable
(c) Magnetic inspection	Conditionally applicable	Not applicable
(d) Penetration test	Not applicable	Not applicable
(e) Magnetic induction test	Not applicable	Conditionally applicable

Presently, preparations are being made for full-scale introduction and use of this method. Most automotive manufacturers in Japan use automated spot welding, performed by six-axis robots. However, at this time, spot welds are still manually inspected using ultrasonic testing methods⁽³⁾. The authors of this paper conducted research on automating the process, focusing on the procedures incorporated by experienced inspectors to increase reliability

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through the introduction of a quantitative inspection method that is not reliant on the labor or experience of a human operator.

2. Judgment of a Nugget Using Ultrasonic Testing Methods

This section discusses comparative judgment using ultrasonic waveforms (termed “echo” below)⁽⁴⁾. The rough crystalline grains in a nugget of the type shown in Fig. 1 (a), which form a good spot weld, cause a fixed level of echo attenuation. This is judged as an appropriate waveform, and the nugget is therefore determined to be of passing quality. Cross-sections of actual nuggets were taken to verify these results. Figure 1 (b) shows the echo for a nugget smaller than the diameter of the probe. Here, an intermediate echo appears between the echoes from the back wall. As shown in Fig. 1 (c), when the nugget is thin (i.e., the spot weld is not thick enough), the level of attenuation is lower and attenuation takes longer than in Fig. 1 (a). When there is absolutely no weld between the two sheets, the echo is reflected only from the bottom of the upper sheet, as shown in Fig. 1 (d). All of these echoes, other than the pattern shown in Fig. 1 (a), are judged to be inappropriate. Ultrasonic testing is, therefore, a method that makes judgments on the basis of comparisons with an ideal standard echo.

3. Issues in Achieving Automation

3.1. Applying the Probe

Applying the probe effectively when inspecting spot welds using ultrasonic testing methods is a difficult operation that requires experience on the part of the inspector. The research team, therefore, decided to use a robot to study the method used by an experienced inspector so they could determine the reasons for this skill requirement, and outline the issues involved in automation.

The six-axis robot shown in Fig. 2 was used in tests in which the probe was applied to the center of spot indentations with weld

samples being fixed. After testing the allowance for angle of inclination [Fig. 3 (a)] and the allowance for offset [Fig. 3 (b)], the possibility of the robot reproducing the positioning shown in Fig. 3 (c) was verified. Results are shown in Table 2. The robot displayed an excellent ability to reproduce the target positioning with respect to a spot indentation, indicating sufficient ability to detect conformity and non-conformity. However, while the nugget was parallel to the sheet surface of the sample used in these tests, as shown in Fig. 4 (a), under actual work conditions, the angle of the welding gun, the inclination of the sheets, and the contact angle of the welding electrodes can produce a nugget that is on an angle with the sheet surface, as shown in Fig. 4 (b). To inspect the nugget



Fig. 2 Six-axis robot

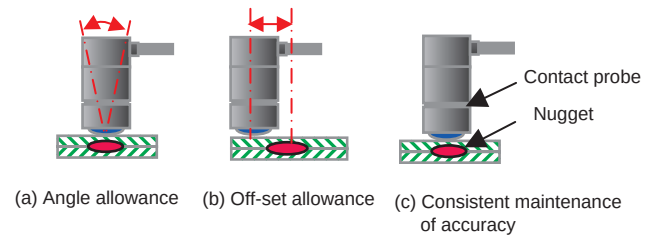


Fig. 3 Test of position and inclination

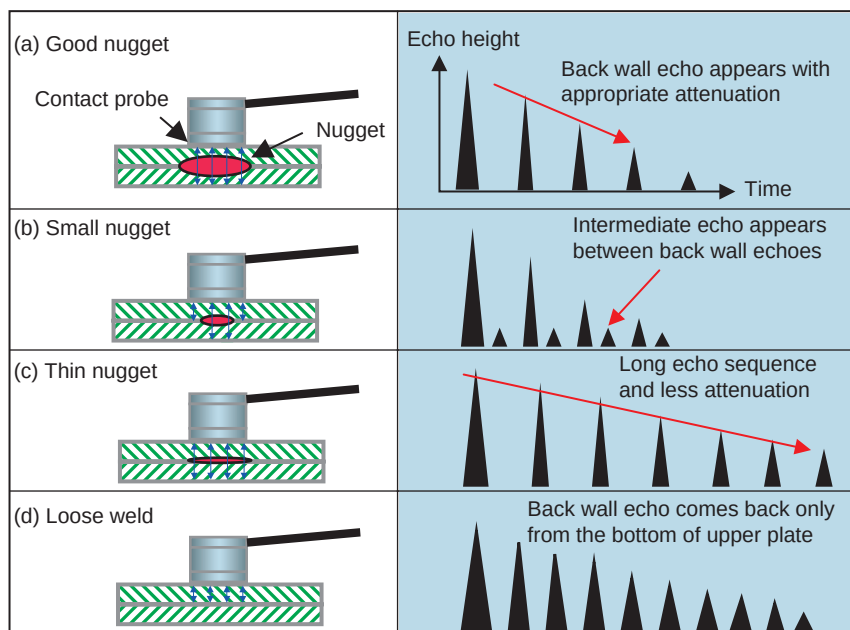


Fig. 1 Ultrasonic detection of spot welding

Table 2 Robot inspection test

Item	Result
(1) Angle allowance	$\pm 1^\circ$ below
(2) Off-set allowance	$\pm 1\text{mm}$
(3) Consistent maintenance of accuracy	Possible

under these conditions, the tip of the probe must be angled normally, relative to the nugget, as shown in Fig. 5. The levels of accuracy indicated in (1) and (2) in Table 2 are required. When the test described above was conducted on nuggets that were actually angled, no echoes were produced. In actual welding, the geometry of the sheets affects the formation sequences of the spot welds, and the nugget at each weld point might, therefore, have a minute inclination. An experienced inspector is able to alter the contact angle of the probe in response to the inclination of the sheets or the nugget, but the robot has difficulty in judging the inclination of the nugget between the steel sheets from the external appearance of the weld, and in adjusting the probe's orientation to allow contact with the nugget accordingly. This is a reason skill is required on the part of inspectors, and presents a significant issue in achieving automation.

3.2. Identification of Spot Indentations

Following the inspection method development discussed in the previous section, the following technology is required for automation of the inspection procedure; the spot indentation imaging and identification through binary image processing based on the morphology and color (hereafter called identification of spot indentations). Inspection becomes possible by identifying multiple spot indentations in sequence and positioning the probe at the center of each indentation. A method was tested using a CCD camera to image the spot indentation so that its center could be determined from its outline. In the case of normal steel, the color change produced by spot welding temperatures enabled the outline of the spot indentation to be clearly imaged, as shown in Fig. 6 (a), and its center could therefore be located immediately. However, as

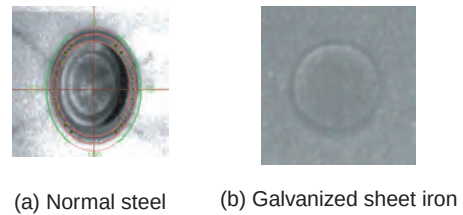


Fig. 6 Spot indentation image by CCD

Fig. 6 (b) shows, in the case of galvanized sheet iron, which is frequently used in automotive applications, the spot indentation could not be clearly imaged. This is assumed to be because the galvanized sheet iron does not change color, and the spot indentation is therefore not clearly visible, making it difficult to identify its outline. However, normal sheet steel and galvanized sheet iron are sometimes used together in automotive applications, and an inspection technology must therefore be able to identify spot indentations on both types of sheeting.

4. Experimental Verification

4.1. Study of Mechanism Response to Nugget Inclination

A mechanism was developed to enable the angle of probe inclination to be freely altered, allowing it to adopt arbitrary angles in relation to the surface of the spot indentation. This was done by applying a rocking motion in response to minute inclinations to the work surface, spot indentation and nugget. A standard robot could not reproduce the probe angles used by a human inspector, so an XY slider table and urethane rubber were, therefore, used at the end of the robot arm (Fig. 7). After the probe has made contact with the spot indentation, it inclines 3 deg. from perpendicular and describes a full circle, divided into 12 sections, around the indentation. A single spot indentation can be inspected in 6 seconds (Fig. 8).

Test pieces were prepared that would produce "good" echoes (echoes judged to be appropriate), and tests were conducted to study the mechanism response to the inclination of the work surface and the nugget. First, the test piece was fixed in place, and the inclination angle of the mechanism was freely altered in relation to it. Tests

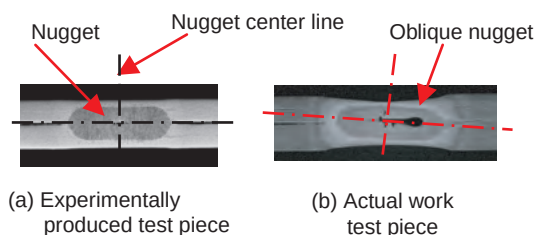


Fig. 4 Comparison of nugget test pieces

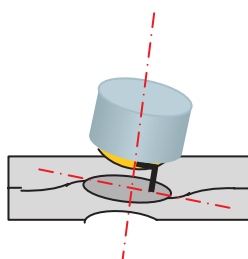


Fig. 5 Correct inclination of contact probe

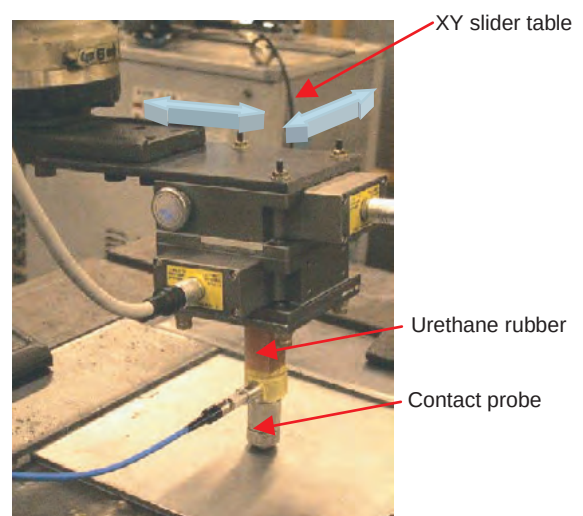


Fig. 7 Mechanism enabling response to change in nugget angle

were conducted to determine whether the developed probe-rocking motion enabled the nugget to be traced, and an appropriate echo could be produced. Results indicated that the desired echo was produced at an angle of inclination of 2 deg. (Fig. 9). The rocking mechanism was therefore judged to be able to respond to the angle of inclination of the work surface and the nugget. Next, to judge the performance of the mechanism in relation to the angle of inclination of the work surface, tests were conducted with the work inclined at angles of 45 and 90 deg. (Fig. 10). The same echo patterns were generated as when the probe was perpendicular to the work. These results indicated that automated inspection was possible even under conditions close to those found on actual production lines.

4.2. Identification of Spot Indentations on Normal Sheet Steel and Galvanized Sheet Iron

A technology was developed to identify spot indentations in both normal sheet steel and galvanized sheet iron. To enable CCD

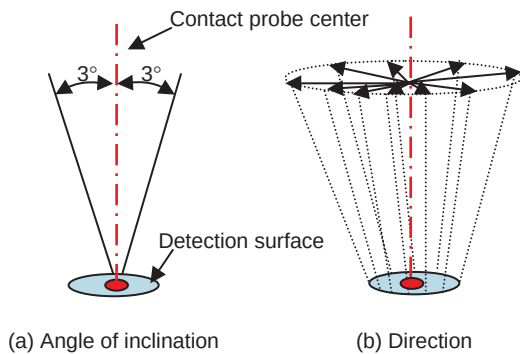


Fig. 8 Angle of inclination and direction

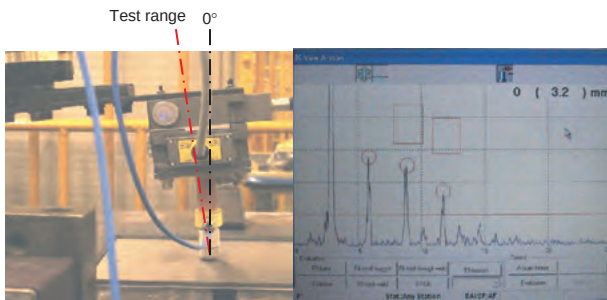


Fig. 9 Test : Change in nugget angle

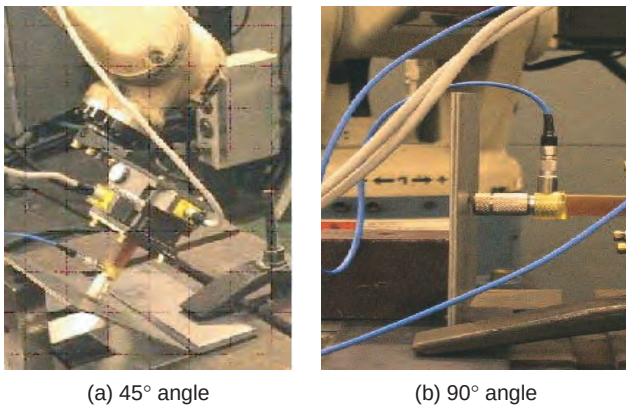


Fig. 10 Test : Change in inclination of work piece

camera imaging to accurately identify the outline of a spot indentation that does not change color, the angle of the illumination source (a red LED) and camera were adjusted to produce a shadow and a reflection of slight spot indentations on the surface of the galvanized sheet iron (Fig. 11). Orientation angles of between 5 and 30 deg. for the LED and between 30 and 60 deg. for the camera were tested (Fig. 12).

These tests indicated that the combination of the LED and the camera produced images that allowed the center of the spot indentation to be identified (Fig. 13). The center of a spot

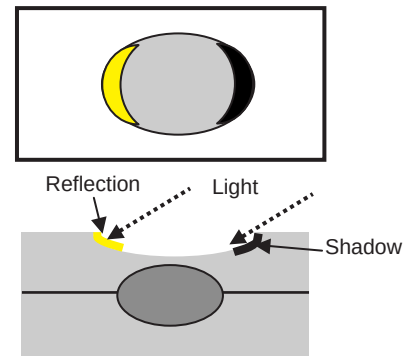


Fig. 11 Method of imaging of spot indentation outline

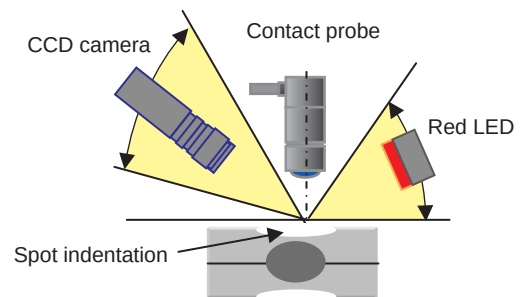


Fig. 12 Imaging test



Fig. 13 Imaging equipment

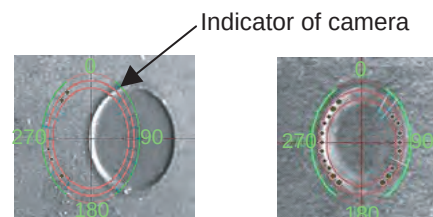


Fig. 14 Identification of spot indentation by developed system

indentation could be identified when this mechanism was fitted to a robot arm and operated at a feed rate of 300 mm/sec (Fig. 14). These trials demonstrated that the new system was able to identify a spot indentation regardless of the material.

5. Component Trials

The layout of the system, which tests the knowledge obtained from the experiment mentioned above using actual parts, is shown in Fig. 15. Figure 16 shows the operational flowchart of the system. A test piece was prepared with 20 spot indentations in a

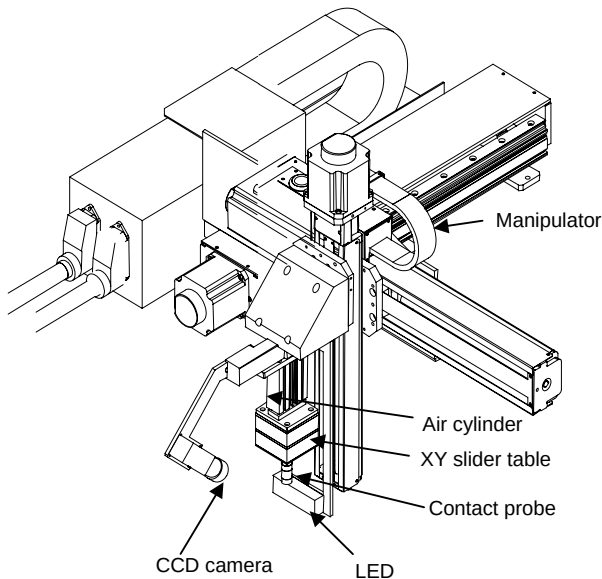


Fig. 15 Developed automatic ultrasonic flaw detection system

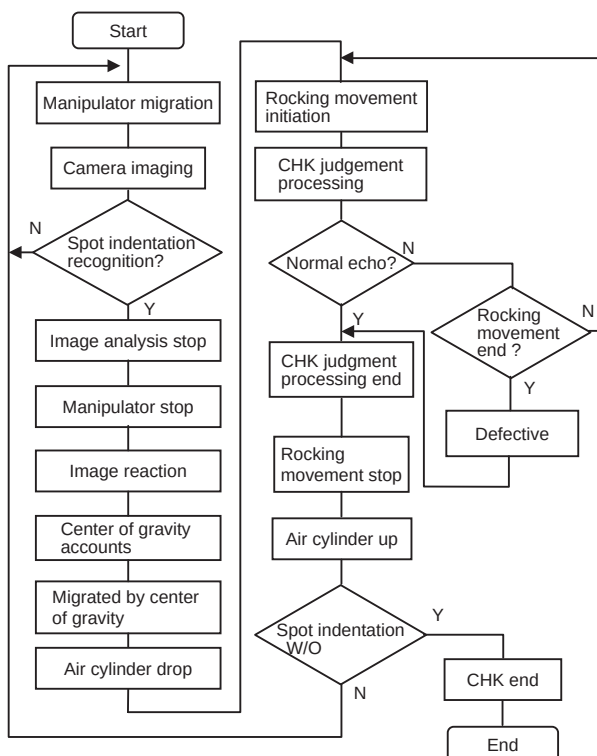


Fig. 16 Operational flowchart of developed system

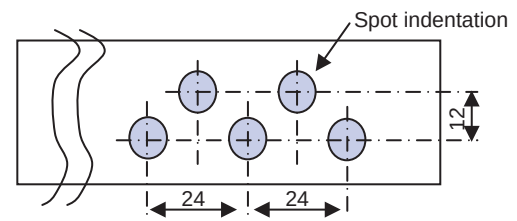
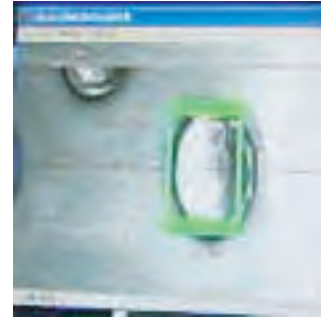


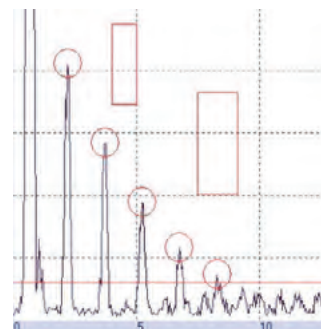
Fig. 17 Test piece



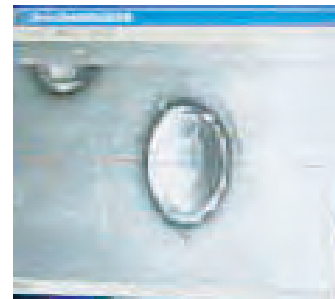
(a) Identify spot indentation



(b) Contact probe drop



(c) Back wall echo



(d) Identify next spot indentation

Fig. 18 Test results

zigzag die center distance, enabling the system to conduct a continuous inspection for flaws (Fig. 17), and test was conducted. Results showed that the system was able to image a spot indentation using the CCD camera and identify its center [Fig. 18 (a)]. Following this, it was able to contact the center of the indentation with the probe [Fig. 18 (b)] and trace the nugget by means of the probe's rocking motion, enabling an echo to be obtained [Fig. 18 (c)]. The system was able then to lift the probe and image the next spot indentation [Fig. 18 (d)]. The system repeated this process to detect all 20 of the spot indentations on the test piece. From these results, it was confirmed that the new system was effective in enabling automatic ultrasonic inspections of spot welds.

The system was fitted to a six-axis robot in combination with an automatic copying mechanism using a laser sensor (Fig. 19) and tested on a real component (Fig. 20). The results showed that the system was able to image spot indentations sequentially and produce echo patterns with simple teaching, indicating that the probe had traced the nugget.

These results clearly indicate that the system can be applied to real automotive parts.

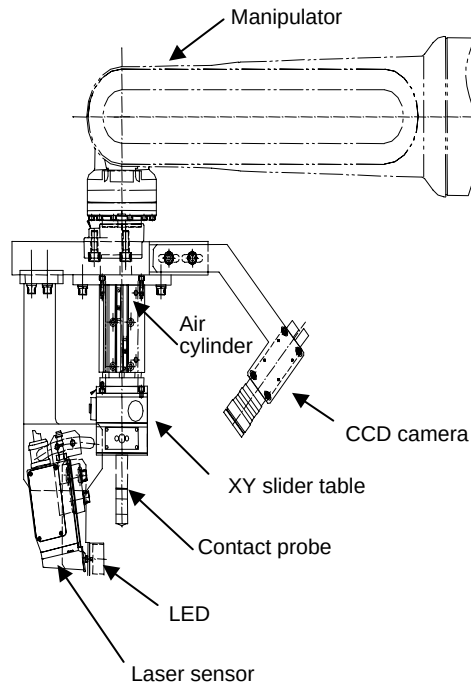
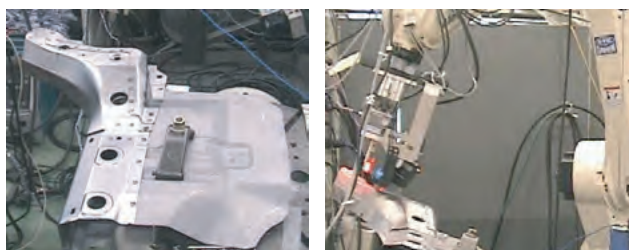


Fig. 19 Developed automatic ultrasonic flaw detection system fitted on six-axis robot



(a) Body component

(b) Test by developed system

Fig. 20 Application of developed system

6. Conclusion

Research was conducted on a technology for automated ultrasonic inspection of spot welds. The following are the significant findings of this research that suggest the possibility of automation in the future:

- (1) The research demonstrated that automated inspection was possible, even when the nugget was angled, if the probe was positioned at the center of the spot indentation and a rocking motion was used.
- (2) Even when working with materials such as galvanized sheet iron and stainless steel, on which it is difficult to identify spot indentations, the system was able to identify the outline and determine the center of indentations via CCD images of the shadows and reflections they produce, enabling automatic application of the probe to their centers.
- (3) A system was developed using the mechanisms in (1) and (2) with an industrial robot. The results of tests to identify a sequence of spot indentations indicate that this system can be used in automated ultrasonic inspection of spot welds.

References

- (1) JSNDI: Hihakaishikennhouhou no Gaiyou to Tsukaiwake, Hihakaikennsabinrann, THE NIKKAN KOGYO SHIMBUN, LTD., p. 9-10 (1985) (in Japanese)
- (2) Okada, S.: Jidoushasanngyou ni okeru Saikinn no Teikouyousetsugijutsu, Welding Technology, Vol. 51, No. 3, p. 76 (2003) (in Japanese)
- (3) Ishikawa, S.: Chouonnpa Tannshou Souchi wo Riyou shita Spot Yousetsubu Kensa, Welding Technology, Vol. 53, No. 3, p. 117 (2005) (in Japanese)
- (4) Ishikawa, S.: Chouonnpa Tannshou Souchi wo Riyou shita Spot Yousetsubu Kensa, Welding Technology, Vol. 53, No. 3, p. 114-115 (2005) (in Japanese)

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Study of Small-sized Scooter Hybrid Vehicle

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ABSTRACT

In the application of hybrid technology to motorcycles, some issues including limited space for mounting a hybrid system and small regenerative energy available during deceleration are present. In order to get the maximum effect of improving fuel economy by solving these issues, we have developed a series-parallel hybrid system. This system uses a direct rear wheel driving motor that can absorb the regeneration energy during deceleration more efficiently. Additionally an idle stopping system by use of a starter AC generator, an electronically controlled fuel injection system, and an electronically controlled belt CVT (continuously variable transmission) system were adopted. Utilizing this small-sized hybrid system the experimental scooter of 50 cm³ displacement achieved an equivalent body size as the mass produced scooter with 50 cm³ engine. Furthermore, this equivalent body size was achieved while keeping a utility space under the seat. Moreover, an improvement of 60% in fuel economy under ECE 40 mode was realized.

1. Introduction

In recent years, environmental protection technology and fuel economy enhancement technology have been widely researched as a result of concerns about the global warming and the depletion of petroleum resources. Also in motorcycles, the use of three-way catalysts and fuel injection systems is being extended. The authors addressed realizing higher level of fuel economy for a motorcycle with a hybrid system⁽¹⁾, similar to the ones used on some automobiles. A series-parallel hybrid system using a direct rear wheel driving motor was developed for a scooter with 50 cm³ of displacement, which is the smallest displacement available in a commercial production motorcycle. As a result, a 60% increase in fuel economy was obtained in ECE40 mode. This paper describes the overview of the study.

2. Issues and Goals

The main issues related to the application of hybrid technology to lightweight and compact motorcycles are as follows.

- The available regenerative energy during deceleration is small.
- The available space for installation of a hybrid system is limited.

A 50 cm³ scooter was selected for research because it seemed challenging to address the above issues due to its small vehicle body and low vehicle mass. A fuel economy enhancement target of at least 50% in the ECE40 mode was set.

The major goals of the research are as follows:

- (1) Achieving the world's highest fuel economy for a scooter with a 50 cm³ engine and a belt type continuously variable transmission (CVT).
- (2) Achieving the same drivability as that of the base vehicle.

* Motorcycle R&D Center

- (3) Achieving the same utility space under the seat as that of the base vehicle.
- (4) Achieving the same body size as that of the base vehicle.

3. Examination of Hybrid Systems

Hybrid systems consist of the following three types:

- (1) Series-hybrid system
A system in which the engine is used only as the motive power for the generator and in which the vehicle is operated by a motor.
- (2) Parallel-hybrid system
A system in which the vehicle is powered mainly by the engine and a motor is used as an auxiliary source of motive power to provide assistance during start and acceleration.
- (3) Series-parallel hybrid system
A system consisting of a combination of a series-hybrid system and a parallel-hybrid system, which operates the engine and motor according to various situations in such a way as to produce maximum efficiency.

To achieve the target fuel economy enhancement, the energy consumption of the base vehicle, the 50 cm³ scooter, was analyzed and a hybrid system was studied. Figure 1 shows the result of calculating the energy balance when the vehicle is running in ECE40 mode. The top part of Fig. 1 shows the vehicle speed in ECE40 mode, the middle part the energy balance of the base vehicle, and the bottom part the energy balance of the hybrid vehicle. In Fig. 1, the output necessary for deceleration is treated as negative output.

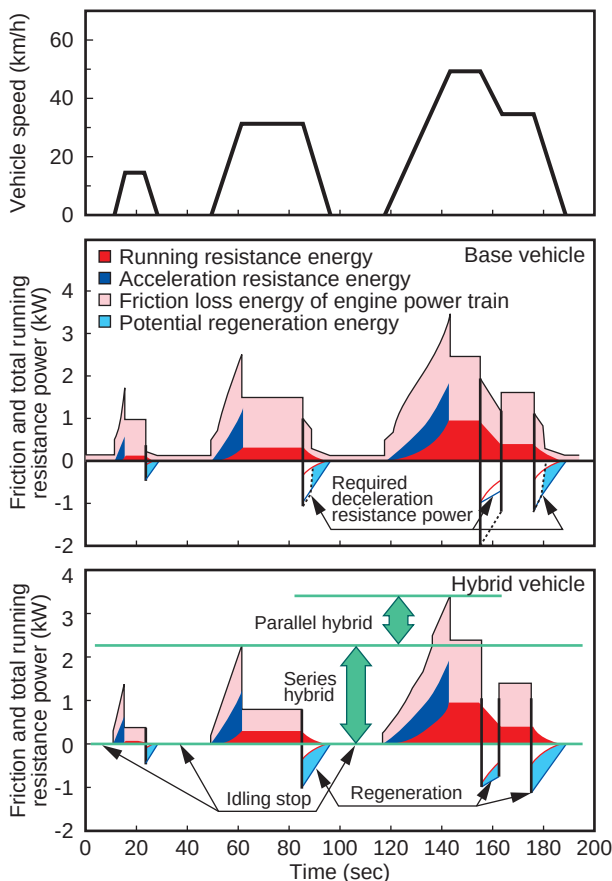


Fig. 1 Energy balance in ECE 40 mode

The remaining energy after subtracting the running resistance and the deceleration energy due to mechanical loss in the engine and the drive train from the necessary energy for deceleration, becomes the potential regeneration energy. It was found that for the base vehicle, the ratio of the mechanical loss in the engine and the drive train to the total energy necessary for running the vehicle was high, and also that this loss reduction was effective in raising the fuel economy. It was also found that the deceleration resistance due to engine friction was large compared to the resistance output necessary for deceleration, and that the potential regeneration deceleration energy could be increased by disengaging the engine from the drive train when the vehicle was decelerating. Given these facts a series-parallel hybrid system was selected as the suitable system, which can enhance fuel economy under various running conditions as indicated in the hybrid vehicle energy balance diagrams of Fig.1. In this system an idling stop, where fuel is not wastefully consumed when the vehicle is stationary is employed. Series-hybrid operation is performed in the low load running range to reduce mechanical loss in the engine and the drive train. Parallel-hybrid operation is performed in the high load running range. During deceleration the engine is disengaged from the drive train, and the deceleration energy is recovered efficiently.

4. Hybrid System Configuration

4.1. System Configuration

Figure 2 shows the configuration of the hybrid system. A two-motor hybrid system consisting of the starter-AC generator mounted on the crankshaft and the driving motor mounted on the belt-converter driven shaft was adopted. By installing a one-way clutch on the driving motor shaft, the engine is disengaged from the drive train, allowing more energy to be regenerated during deceleration. The operating revolution set point of the centrifugal starting clutch is 4 200 r/min. When the engine is running at or below 4 200 r/min, the engine is disengaged from the drive train, permitting series-hybrid operation in which the engine is used as a generator. A drive-by-wire (DBW) system is used to freely control the engine output and engine speed. The battery is a nickel-hydrogen battery, which is already being used in hybrid automobiles and features stable output characteristics that do not depend upon the charging condition. Prototype interior permanent magnet (IPM) type DC brushless motors, which have a maximum output of 1.2 kW, were used for the starter-AC generator and driving motor.

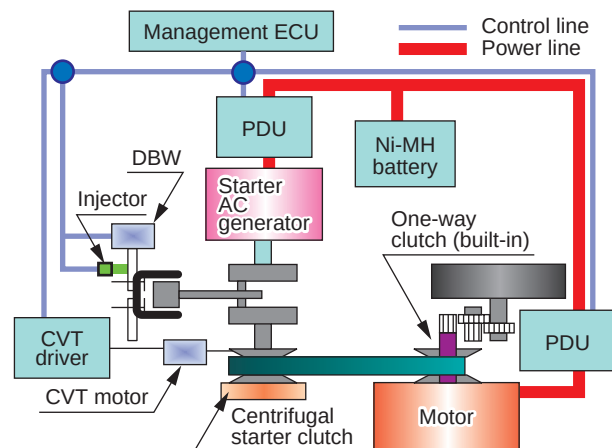


Fig. 2 Series-Parallel hybrid system

4.2. Control System

In a hybrid system it is necessary to perform cooperative control of the engine output, generator power, motor output, CVT ratio, and so forth, according to the running conditions and the condition of the battery. Therefore, the system used consisted of a management ECU, power drive unit (PDU) for the starter-AC generator, a driving motor PDU and CVT driver, and integrated control was performed by the management ECU, as shown in Fig. 2. The management ECU contains drivers for the FI and DBW. Based on the results of energy management and drive power control computations the management ECU drives the FI and DBW to control the engine output, and simultaneously assigns a target ratio and torque to the CVT driver and each PDU to control the ratio of the electronically controlled CVT as well as the output from the starter-AC generator and the driving motor.

4.3. System Installation

Figure 3 shows the components layout of the hybrid system installed on the vehicle. The battery packs were installed inside of the front cowl on both sides, and the PDUs on both sides under the seat. The layout was considered for the longitudinal and lateral mass balance so as not to affect the running stability. The PDU was mounted directly on the frame to release heat thereby eliminating the heat release fins and keeping the body compact. The FI and DBW control sections were incorporated into the management ECU. The ECU itself was miniaturized and installed rearward of the seat. As a result of these modifications the vehicle body size and the utility space under the seat were the same as those of the base vehicle.

4.4. Major Specifications

Figure 4 shows the appearance of the vehicle equipped with a hybrid system, and Table 1 shows the major specifications. Figure 5 shows the utility space under the seat.

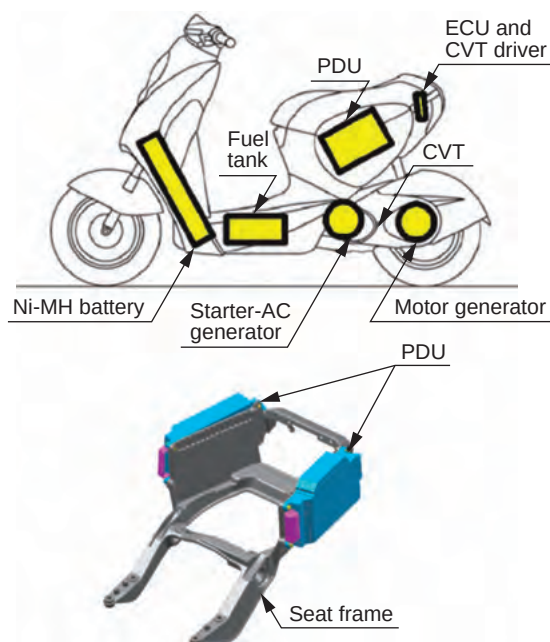


Fig. 3 Hybrid system layout

5. Fuel Economy Enhancement Technology

5.1. Series-parallel Hybrid System

Figure 6 shows the engine operation conditions when a vehicle equipped with a conventional engine is running at constant speeds of 30 km/h and 50 km/h. The brake specific fuel consumption (BSFC) at 30 km/h was larger than the BSFC at 50 km/h, indicating the fuel consumption per engine output was larger at 30 km/h. In the case of a vehicle equipped with a conventional engine, because the engine is coupled to the rear wheel via a transmission, the engine speed is determined by the vehicle speed and the gear ratio, hence it is difficult to greatly change the engine operation range. On the other hand, in the low load running range, such as when the vehicle is traveling at 30 km/h, the engine is disengaged from the drive train and series-hybrid operation is carried out, permitting the engine to be run at low speed under high load running conditions and realizing low fuel consumption (Fig. 6).

In contrast, in the high load running range, such as when the vehicle is traveling at 50 km/h, the vehicle travels in a range where the engine operation efficiency is relatively high. Therefore, the improvement level due to the use of a series-hybrid system is small. Additionally, to use a series-hybrid system it is necessary to increase the output of the motor and the generator, which imposes limitations on the mass and size of the motor and AC-generator. For these reasons it was thought that a series-parallel hybrid system that switches between series and parallel-hybrid operation during which the engine is the main motive power source would not only be effective for low fuel consumption, but also would make the system light and compact. In addition, when the vehicle is traveling under high load, the shift characteristics of the electronically controlled CVT used during parallel-hybrid operation were set to the high ratio side, and the vehicle was run at a lower engine speed to enhance the engine operation range, as shown in Fig. 6.

5.1.1. Range of switchover between series and parallel-hybrid operation

Figure 7 shows the range of series and parallel-hybrid operation. The series-hybrid operation range was set within the range of the maximum generator power, below the operating revolution of the centrifugal starting clutch and the maximum output of the driving motor. At the same time, the fuel consumption rate should be lower than that during parallel-hybrid operation in that range.



Fig. 4 Developed scooter equipped with hybrid system

Table 1 Major specifications

Items			Modified	Base
Vehicle	Overall width	(mm)	630	630
	Overall height	(mm)	1 010	1 010
	Wheelbase	(mm)	1 190	1 190
	Seat height	(mm)	710	710
	Luggage space capacity	(Litter)	22	22
	Mass	(kg)	93	81
	Maximum velocity	(km/h)	60	60
Motor	Type		DC brushless	-
	Maximum power	(kW)	1.2	-
	Maximum torque	(Nm)	6	-
Starter AC generator	Type		DC brushless	-
	Maximum power	(kW)	1.2	-
	Maximum torque	(Nm)	6	-
Engine	Type		Water-cooled gasoline engine	Water-cooled gasoline engine
	Displacement	(cm ³)	49	49
	Valve train		SOHC 4-valve	SOHC 4-valve
	Maximum power	(kW/r/min)	3.5 / 8 000	3.5 / 8 000
	Maximum torque	(Nm/r/min)	4.2 / 7 000	4.2 / 6 000
Battery	Type		Ni-MH	Pb
	Energy capacity	(Ah)	5	6
	Voltage	(V)	36	12



Fig. 5 Luggage space

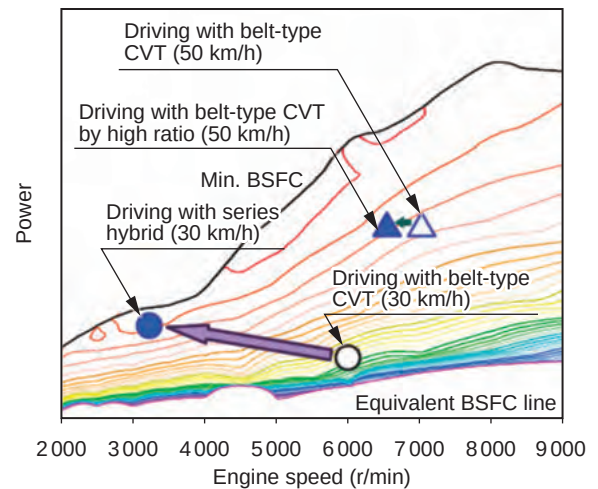


Fig. 6 Changes of engine operation conditions with series and parallel hybrid

5.1.2. Efficiency enhancement of series-hybrid operation

To enhance the energy transfer efficiency during series-hybrid operation, real-time generating control and BSFC bottom trace control were performed⁽²⁾.

Real-time generating control is a control method for reducing loss of electric charge and discharge by generating exactly the amount of electrical energy required to operate the vehicle in real time, so as to minimize charging and discharging of the battery.

BSFC bottom trace control is a control method for minimizing fuel consumption per unit generator power when generating electricity based on real-time generating control, by generating electricity along the generating BSFC bottom line (hereafter referred to as “engine control line”). The BSFC bottom line traces the engine operation points at which the BSFC becomes as small as possible with respect to the required generator power as indicated in Fig. 8.

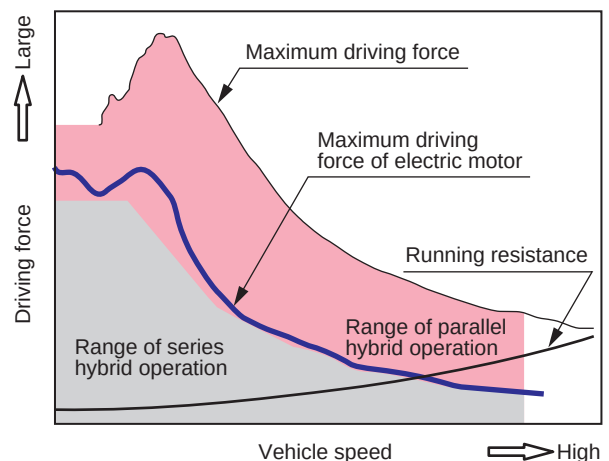


Fig. 7 Range of series or parallel hybrid driving

5.2. Recovering Deceleration Energy

A one-way clutch was used to disengage the engine and the drive train during deceleration. This allows the deceleration energy that is consumed as the engine brakes in a conventional vehicle to be recovered. Figure 9 shows the deceleration energy recovery characteristics. When the throttle is closed, the regeneration energy during deceleration is recovered.

In addition, to recover more deceleration energy the quantity of regenerative energy during braking was recovered by motor braking regeneration corresponding to the rear brake stroke.

The quantity of regenerative energy during deceleration was set to match the demand deceleration of ECE40 mode. Also, the quantity of regenerative energy during braking was set such that the upper limit was the maximum quantity of regenerative motor energy.

5.3. Other Technologies for Reducing Fuel Consumption

5.3.1. Idling stop

An effective method of reducing fuel consumption is to stop the engine whenever the vehicle comes to a stop. In this research, to reduce fuel consumption, idling stop was performed using the starter-AC generator, which is already being mass-produced.

5.3.2. Technology for reducing engine fuel consumption

Aiming at the following two points, the engine was modified, and fuel economy of the vehicle was enhanced.

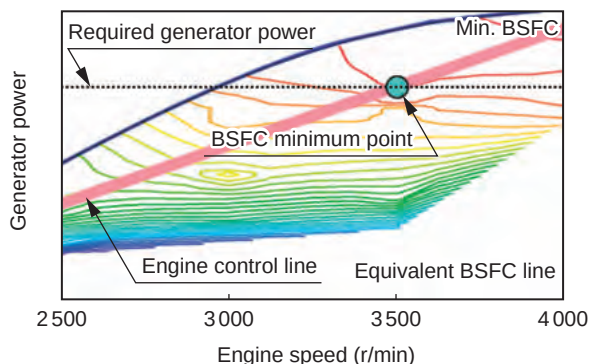


Fig. 8 Engine control line

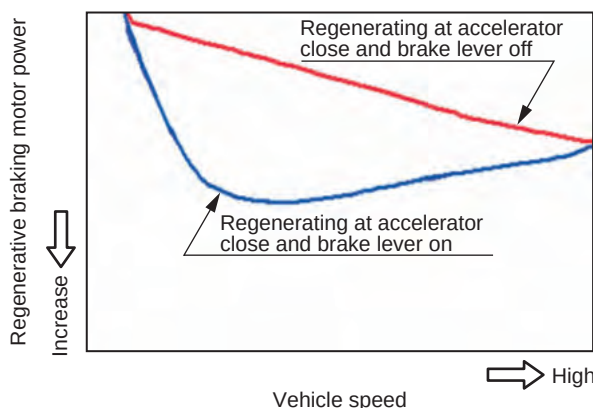


Fig. 9 Regeneration characteristics

(1) Expansion of the stoichiometric air fuel ratio range

The acceleration compensation fuel injection quantity was reduced by means of motor assistance during acceleration. Also, an aluminum sleeve cylinder was adopted to increase the cooling performance. As a result, the stoichiometric air fuel ratio range was expanded, enabling fuel consumption to be reduced while maintaining the same exhaust gas emission potential.

(2) Reduction of mechanical friction

As well as improved combustion, reducing mechanical friction is also important in enhancing fuel economy performance. The following technologies were used to reduce friction.

- Ion-plating low tension piston ring
- Molybdenum disulfide (MoS_2) shot piston⁽³⁾
- Low friction oil seal
- Low viscosity oil

Using a low tension piston ring is an effective method of reducing mechanical friction. However, there is a trade-off between oil consumption and blow-by gas discharge. In this research, a return hole was designed in the oil-ring groove and the dimensions were modified to reduce the tension on the oil ring.

Molybdenum disulfide shot utilizes Wonder Process Craft (WPC) treatment technology to apply high purity molybdenum disulfide to the sliding surface of the piston. This greatly reduces the sliding resistance while maintaining adequate effectiveness.

As a result of these technologies, it was possible to reduce the friction in the overall engine by at least 25% compared to that of a conventional 50 cm³ engine (Fig. 10). In addition, the engine output characteristics were changed to increase the torque at low and medium speed. As a result, the required engine speed during series-hybrid operation was lowered, enabling a reduction in friction torque and ensuring extra power.

6. Validation using Actual Vehicle

6.1. System Operation Condition

Figure 11 shows changes in system operation mode running in ECE40 mode, electric power of the motor, generator power, state of charge of the battery (hereinafter referred to as "SOC"), and the engine speed. When the vehicle is stopped, idling stop is implemented, and when the vehicle is started, EV operation is performed. During series-hybrid operation, generator power corresponding to the electric power to drive the motor is generated by real-time generating control. Also, it can be seen in Fig. 12 that the engine operation range during series-hybrid operation is on the engine control line.

When idling stop or EV operation is implemented, the SOC decreases. But because the deceleration energy and SOC feedback during real-time generating control are recovered, the SOC reverts to the startup value at the end of mode operation.

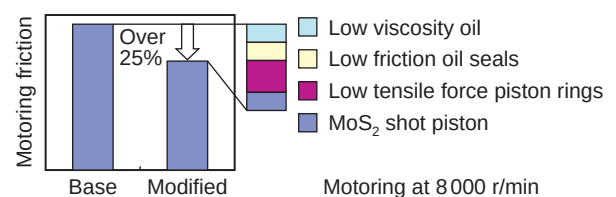


Fig. 10 Effects of engine friction reduction

6.2. ECE40 Mode Fuel Economy

Figure 13 shows the breakdown of the fuel economy enhancement effect during operation in ECE40 mode. “Parallel-hybrid” in the diagram indicates the fuel efficiency measured when the engine friction is reduced, the air-fuel ratio is maintained stoichiometrically, idling stop is implemented and parallel-hybrid operation is performed even in the low load operation range. It was found that the fuel economy was 46% higher than that of the base vehicle. “Series-parallel hybrid” in the diagram indicates the fuel efficiency measured when series-hybrid operation was performed in the low load operation range, in contrast to the abovementioned parallel-hybrid operation. The mode fuel economy for series-parallel hybrid operation reached 84 km/L, which was a 60% increase compared to that of the base vehicle.

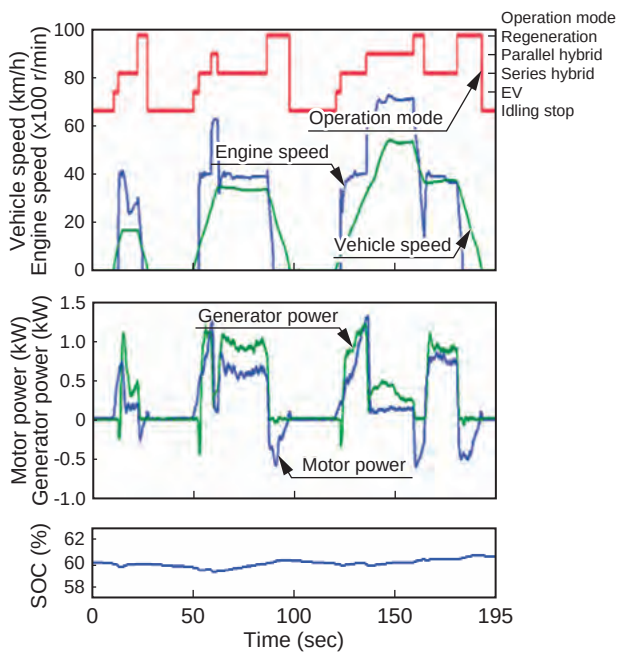


Fig. 11 System operation diagram (ECE40 mode)

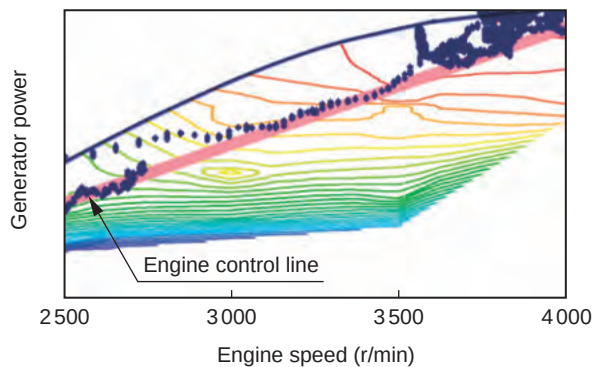


Fig. 12 ECE40 series hybrid operation region

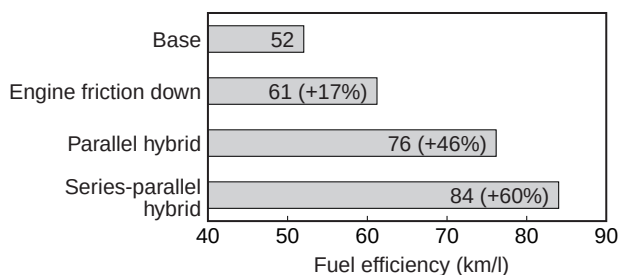


Fig. 13 ECE40 mode fuel efficiency

7. Conclusion

Tests were performed using a prototype vehicle equipped with a series-parallel hybrid system. Fuel economy enhancement was targeted for each driving condition, when the vehicle was stopped, when cruising at low and high loads, and when decelerating. The following results were obtained.

- (1) The fuel economy in ECE40 mode enhanced by 46% compared with that of the base vehicle as a result of parallel-hybrid operation, in which engine friction was reduced, the air-fuel ratio was maintained stoichiometrically, idling stop was implemented, and optimum ratio control was performed by electronically controlled CVT.
- (2) Series-hybrid operation was performed in the low load operation range, and real-time generating control and BSFC bottom trace control were performed, resulting in a fuel economy of 84 km/L. This was an additional 10.5% enhancement compared to (1) above. This is a 60% enhancement in fuel economy compared to that of the base vehicle.
- (3) The hybrid system was miniaturized and its installed position was optimized. As a result, the size of the vehicle body and the utility space under the seat were the same as those of the base vehicle.

References

- (1) Fukuo, K., Fujimura, A., Saito, M., Tsunoda, K., Takiguchi, S.: Development of the Ultra-Low-Fuel-Consumption Hybrid Car - INSIGHT, HONDA R&D Technical Review, Vol. 11, No. 2, p. 1-8
- (2) Hasegawa, Y., Aoyagi, S., Yonekura, T., Abe, H.: Energy Efficiency Improvement of Series Hybrid Vehicle, HONDA R&D Technical Review, Vol. 12, No. 1, p. 79-84
- (3) Ogiwara, H.: Technology for Applying Solid Lubricant to the Piston Skirt of Internal Combustion Engine and Its Effects, Honda R&D Technical Review, Vol. 14, No. 1, p. 85-92

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Development of 1.8L CNG Engine for 2006 CIVIC GX

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ABSTRACT

Honda has developed the 2006 CIVIC GX, a natural gas vehicle that it has been mass-producing since 1998. For this model, Honda has developed and fitted a new Compressed Natural Gas (CNG) engine. This new engine was designed to further enhance power. The increased engine displacement and use of a variable-length air intake manifold have achieved a 15% increase in power. In addition, engine fuel consumption has been reduced by 4% through a combination of a reduction in pumping loss due to the inlet valve delayed closure, and a charging stroke injection that harnesses the properties of gaseous fuel. As a countermeasure for tail pipe emissions, a cylinder head integrated with the exhaust manifold and a close-coupled two-bed catalyst have been used. As a result, the amount of precious metal used was reduced by approximately 30%, and is in compliance with the Advanced Technology Partial-Zero Emission Vehicle (ATP-ZEV) Standards. Despite the increased weight of the larger vehicle design, the CIVIC GX achieved a fuel economy increase of 15% in FTP highway mode, and an equivalent fuel economy in combined mode.

1. Introduction

In recent years, various alternative fuels have been examined from the perspective of reducing air pollution. The reduction of carbon dioxide has also become a very important issue from the perspective of reducing global warming. CNG engines are internal combustion engines that have excellent environmental performance, with virtually zero Non Methane Hydro Carbon (NMHC) emissions. Honda launched the CIVIC GX⁽⁴⁾ in 1998 to proactively address environmental issues. The 2001 CIVIC GX (D17A engine)⁽³⁾ was the first in the world to comply with the Advanced Technology Partial-Zero Emission Vehicle (ATP-ZEV) regulations. The new CNG R18A engine was developed for the 2006 CIVIC GX to maintain environmental performance while further enhancing driving performance and fuel economy. This report gives a

description of the fuel economy enhancement technology^{(1),(2)}, which uses inlet valve delayed closure.

2. Development Goals

The development goals for the CIVIC GX CNG engine were to achieve an environmental performance better than that of previous models, and to further enhance driving performance and fuel economy. The specific targets were defined as follows:

- (1) Achieve the best exhaust emission gas performance of internal combustion engines to comply with ATP-ZEV, the strictest regulations in the world
- (2) Increase engine output
- (3) Enhance fuel economy

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3. Outline and Specifications

3.1. CNG Fuel Characteristics

CNG is a gaseous fuel that contains alkanes extracted from oil and gas fields, a small amount of oxygen, and inert components. The primary alkane extract is methane, but others including butane are also used. Table 1 shows the characteristics of gasoline and methane, the main components of CNG. The characteristics of CNG compared to gasoline as an internal combustion engine fuel are as follows:

- (1) An increased Research Octane Number (RON) and a larger compression ratio are required because combustion during partial loads is poor.
- (2) It is a gaseous fuel, so there is a reduction of 10% or more in charging efficiency for the fuel volume.
- (3) It is a gaseous fuel, so the energy density is low.
- (4) Carbon dioxide emissions are reduced by about 30%.

The heating value of an internal combustion engine fuel is the heating value of the charged mixture. The characteristics of methane, the main component, are as follows.

- (1) The heating value by weight is higher than for gasoline.
- (2) The heating value when mixed with air at its stoichiometric ratio is equivalent to the heating value when gasoline is mixed with the same amount of air at its stoichiometric ratio.
- (3) Since there is a reduction of 10% or more in charging efficiency, there is also a similar reduction in the heating value for each cycle.

A comparison with the gasoline heating value is shown in Table 2.

3.2. Engine Outline

Table 3 shows a comparison with the previous D17A CNG engine and with the base R18A gasoline engine. The main changes from the gasoline specifications are as follows.

- (1) Compression ratio change
- (2) Use of a CNG fuel supply system

The main changes from the previous CNG engine are as follows.

- (1) Use of the i-VTEC inlet valve delayed closure valve train to enhance fuel economy
- (2) Use of a variable-length air intake to increase output
- (3) Change to LAF+O₂ A/F control

Figure 1 shows the overall view.

Table 1 Comparison of methane and gasoline fuel characteristics

Fuel		CH ₄	Gasoline
RON		130	91
A/F	Weight	17.2	14.5
	Volume	9.52	8.3 × 10 ³ (Liquid)
Charging efficiency (%)		90.5	100
Exhaust gas at stoich.	CO ₂ (%)	9.5	13.5
	H ₂ O (%)	19.0	12.5

Table 2 Comparison of methane and gasoline heating value

Fuel		CH ₄	Gasoline
Lower heating value	(MJ/kg)	50.1	42.7
	(MJ/air m ³)	3.80	3.80
	(MJ/air+fuel m ³)	3.42	3.80

Table 3 Engine specifications

Engine code	R18A		D17A
Fuel	CNG	Gasoline	CNG
Cylinder configuration	In-line 4-cylinder		In-line 4-cylinder
Bore × Stroke (mm)	81 × 87.3		75 × 94.4
Displacement (cm ³)	1799		1668
Compression ratio	12.5	10.5	12.5
Valve train	SOHC i-VTEC Inlet delayed closure		SOHC VTEC-E 1 intake valve inactive
Number of valves	4 per cylinder		4 per cylinder
Valve diameter (mm)	IN	32	30
	EX	26	26
Cylinder offset (mm)	12		0
Intake manifold	Variable intake system		Conventional
Fuel control	LAF+O ₂ sensor		O ₂ +O ₂ sensor



Fig. 1 R18A i-VTEC CNG engine

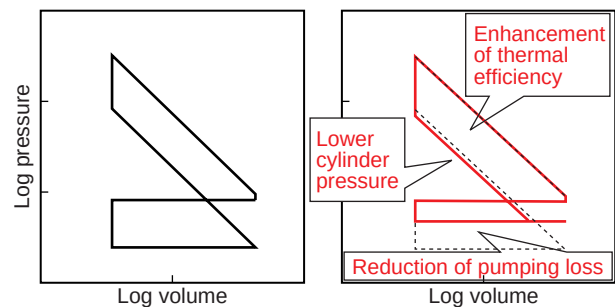
4. Fuel Economy Improvement Technology

4.1. Fuel Economy Improvement Effect by Inlet Valve Delayed Closure

Pumping loss reduction technology based on inlet valve delayed closure was introduced for this engine. The aim of this technology is to enhance the fuel economy. Figure 2 shows the conceptual diagrams.

There are three main reasons for the effectiveness of inlet valve delayed closure in improving fuel economy:

- (1) Reduction of pumping loss
- (2) Enhancement of thermal efficiency due to the reduction of the effective compression ratio and the extended expansion ratio
- (3) Reduction of mechanical loss due to a lower cylinder pressure caused by the reduction of the effective compression ratio



(a) Normal closure

(b) Delayed closure

Fig. 2 Comparison of P-V diagram between normal and delayed closure

4.2. Pumping Loss Reduction

The pumping loss is expressed as follows:

$$W_p = \left(\frac{P_{BA}}{k-1} \frac{\epsilon}{\epsilon-1} \right) \left(1 - \left(\frac{1}{\epsilon} \right)^{k-1} \right) - \frac{P \left(\frac{P_{BA}}{P} \right)^{\frac{1}{k}} \frac{\epsilon}{\epsilon-1}}{k-1} \left(1 - \left(\frac{1}{\epsilon} \right)^{k-1} \right) + P \left(\left(\frac{P_{BA}}{P} \right)^{\frac{1}{k}} \frac{\epsilon}{\epsilon-1} - \frac{1}{\epsilon-1} \right) - P_{BA} V \quad (1)$$

W_p : Pumping loss

ϵ : Compression ratio

V : Stroke volume

P : Air pressure

P_{BA} : Intake pressure

κ : Specific heat ratio

Here, if the following relationship is assumed,

$$P_{BA_d} = \frac{\epsilon}{\epsilon_d} P_{BA}$$

W_{pd} : Pumping loss during inlet valve delayed closure operation

P_{BA_d} : Intake pressure during inlet valve delayed closure

ϵ_d : Compression ratio during inlet valve delayed closure

Pumping loss during inlet valve delayed closure operation can be expressed as:

$$W_{pd} = \left(\frac{\epsilon}{\epsilon_d} \frac{P_{BA}}{\epsilon-1} \right) \left(1 - \left(\frac{1}{\epsilon} \right)^{k-1} \right) - \frac{P \left(\frac{\epsilon}{\epsilon_d} \frac{P_{BA}}{P} \right)^{\frac{1}{k}} \frac{\epsilon}{\epsilon-1}}{k-1} \left(1 - \left(\frac{1}{\epsilon} \right)^{k-1} \right) + P \left(\left(\frac{\epsilon}{\epsilon_d} \frac{P_{BA}}{P} \right)^{\frac{1}{k}} \frac{\epsilon}{\epsilon-1} - \frac{1}{\epsilon-1} \right) - \frac{\epsilon}{\epsilon_d} P_{BA} V \quad (2)$$

When Eqs. (1) and (2) are compared, it can be seen that the pumping loss reduction ratio during inlet valve delayed closure operation is determined by the effective compression ratio and intake pressure.

4.3. Thermal Efficiency Enhancement Due to Extended Expansion Ratio

This section will examine the thermal efficiency⁽⁵⁾ of the extended expansion ratio cycle caused by the inlet valve delayed closure. The thermal efficiency of a regular Otto cycle is:

$$\eta = 1 - \left(\frac{1}{\epsilon} \right)^{k-1}$$

η : Thermal efficiency

The thermal efficiency during inlet valve delayed closure operation is expressed as follows.

$$\eta_d = 1 - \left(\frac{1}{\epsilon} \right)^{k-1} + \frac{T_a C_V}{Q} \left(1 - \left(\frac{\epsilon_d}{\epsilon} \right)^{k-1} \right)$$

η_d : Thermal efficiency during delayed closure operation

Q : Heating value

T_a : Intake temperature

C_V : Specific heat at constant volume

The thermal efficiency ratio is expressed as follows.

$$\frac{\eta_d}{\eta} = 1 + \frac{T_a C_V}{Q} \frac{1 - \left(\frac{\epsilon_d}{\epsilon} \right)^{k-1}}{1 - \left(\frac{1}{\epsilon} \right)^{k-1}}$$

This is a slightly increased thermal efficiency. Figure 3 shows the results of the calculation. If the ratio of the effective compression ratio and the compression ratio is 0.7, then the thermal efficiency is increased by 1.5%.

4.4. Mechanical Loss Reduction

The reduction of the effective compression ratio due to inlet valve delayed closure operation lowers the cylinder pressure, and so is effective in reducing mechanical loss. The cylinder pressure ratio is expressed as:

$$\frac{P_{cd}}{P_c} = \left(\frac{\epsilon_d}{\epsilon} \right)^k$$

P_c : Pressure after compression

P_{cd} : Pressure after compression during delayed closure operation

This is a reduction. This is likely to reduce mechanical loss in areas such as the bearings.

The characteristics of inlet valve delayed closure operation are summarized below:

- (1) Reduction of pumping loss
- (2) Increase in thermal efficiency
- (3) Reduction of mechanical loss
- (4) Reduction of effective compression ratio
- (5) Reduction of charging efficiency

The above are determined by the delayed Inlet Valve Closure (I.V.C.). As a result, a thermal efficiency increase effect is achieved through the pumping loss reduction. Also, a mechanism that compensates for the accompanying reduction in charging efficiency is needed.

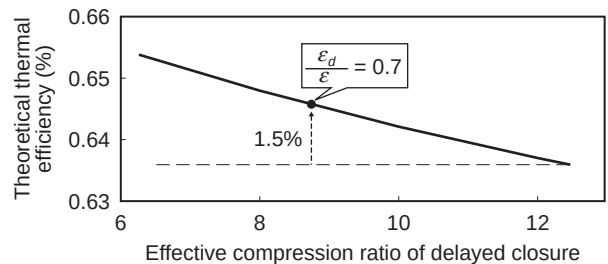


Fig. 3 Comparison of theoretical thermal efficiency between conventional and delayed valve timing

4.5. i-VTEC Inlet Valve Delayed Closure Mechanism

This engine uses Variable Valve Timing and Lift Electronic Control (VTEC) to achieve both high output and inlet valve delayed closure operation. In conditions such as acceleration, which require high output, the inlet valve is activated with a cam that generates the maximum output (this is referred to as high output cam). When the engine load is light, such as during cruising, the engine switches to a cam that delays the closure timing of one of the inlet valves (this is referred to as delayed closure cam). At the same time, pumping loss is reduced by opening the electric control throttle, which improves the fuel economy. The i-VTEC mechanism integrates control of this electric control throttle and VTEC, and enables the engine to switch optimally between the high output cam and delayed closure cam in accordance with the driving conditions.

Figure 4 shows the i-VTEC inlet valve delayed closure valve train. The i-VTEC switching is performed by connecting and separating the rocker arm, through the hydraulic movement of the synchro piston built into it. In this engine, the switching mechanism is only used in one of the rocker arms in two inlet valves.

When the high output cam operates, synchro piston B is moved hydraulically, which disconnects synchro piston A, and activates the inlet valve with cam profile A.

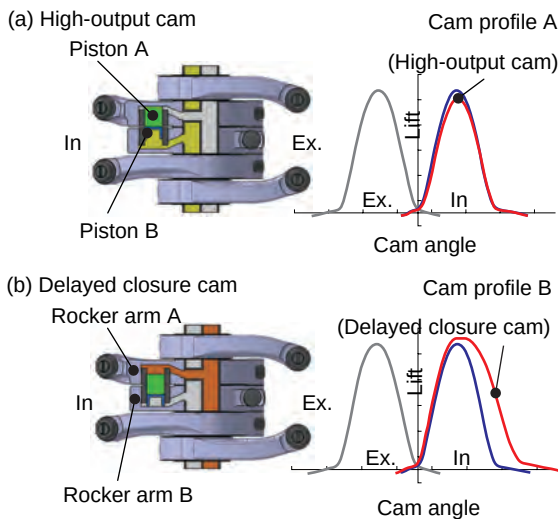


Fig. 4 Rocker arm and cam profile

5. Engine Performance

5.1. Determining the Delayed I.V.C.

It is preferable to delay the I.V.C. as much as possible from the perspective of minimizing pumping loss. On the other hand, the delay is accompanied by a reduction in output. This reduces the operable range with delayed closure and increases the switching frequency. As a result, a delayed closure compression ratio must be determined that delivers the required output for the upper limit of engine speed when cruising.

The delayed closure operating range for this engine has been set so that the speed upper limit during normal use is 120 km in 5th, and the required output is 130% of output when cruising. These settings are equivalent to those in the gasoline specifications of this engine. The delayed closure cam I.V.C. uses the following to estimate the output change from the actual gasoline measurements:

- (1) Reduced charging efficiency due to CNG injection
- (2) Heating value
- (3) Compression ratio change

As shown in Fig. 5, the estimate is 86 degrees, which is earlier than the 94 degrees for gasoline.

The BSFC reduction rate in this case is estimated to be about 5%, compared to about 6% for gasoline. The results of the test show that the output is approximately equal to the assumed value, but that the BSFC reduction rate is lower than the value assumed from the pumping loss reduction. Figure 6 shows the BSFC and ISFC changes for each load. It verifies that the ISFC is increased when the load is low. The reason a result equivalent to gasoline was not obtained may be that the 10% to 90% mass burnt during the delayed closure was too long. An analysis of this reason is performed below.

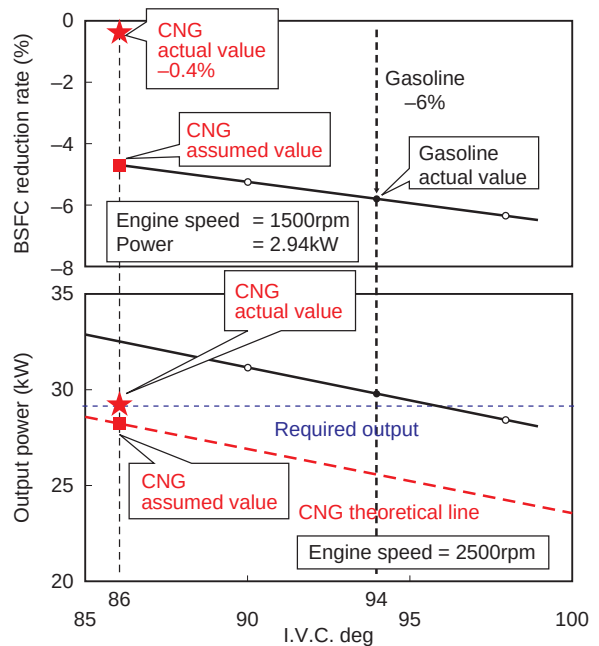


Fig. 5 Relationship between I.V.C. and output power and charging efficiency

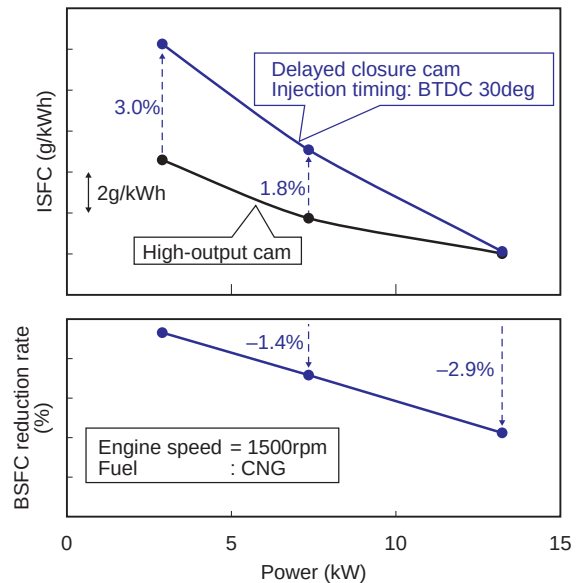


Fig. 6 Comparison of ISFC between conventional and delayed valve timing

5.2. Fuel Economy Improvement for Delayed Closure Operation

Figure 7 and Table 4 show a comparison of the mass burnt ratio between CNG and gasoline combustion with the high output cam. The compression ratio for CNG increases from 10.5 to 12.5, so that the main 10% to 90% mass burnt (Mass Burnt Fraction 10 - 90%) is approximately equivalent to that of gasoline.

Figure 8 and Table 5 show a comparison of the mass burnt ratio for CNG between the high output cam and the delayed closure cam. With the delayed closure cam, it is likely that the main 10% to 90% mass burnt will be extended and the thermal efficiency decreased. As shown by Fig. 9 and Table 6, when the delayed closure operation is performed with gasoline, the extension of the main 10% to 90% mass burnt is short. To enhance the ISFC, it is likely that a shortening of the 10% to 90% mass burnt during delayed closure operation will be required.

As a method for shortening the main 10% to 90% mass burnt, a flow increase was attempted in this engine by harnessing the characteristics of gaseous fuel injection. Figure 10 shows the relationship between injection timing and BSFC. By combining the delayed closure with the charging stroke injection, it was possible to verify a BSFC reduction of about 5%. However, it was not possible to verify a BSFC reduction with the high output cam.

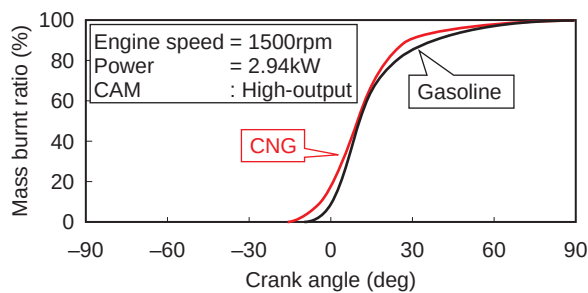


Fig. 7 Comparison of mass burnt ratio between CNG and gasoline

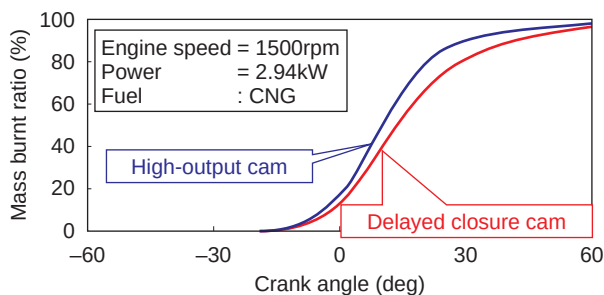


Fig. 8 Comparison of mass burnt ratio between conventional and delayed valve timing

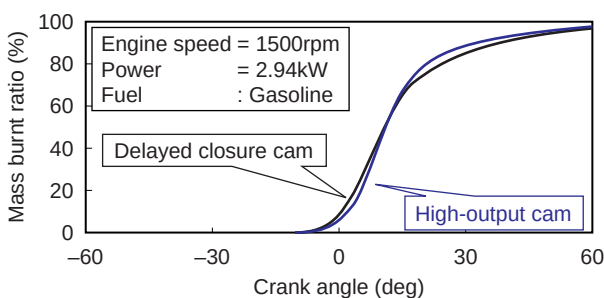


Fig. 9 Comparison of mass burnt ratio between conventional and delayed valve timing

Table 4 Comparison of IG to 10% mass burnt and 10% to 90% mass burnt between CNG and gasoline

Fuel	CNG	Gasoline
IG to 10% mass burnt (deg.)	33	38
10% to 90% mass burnt (deg.)	33	30

Table 5 Comparison of IG to 10% mass burnt and 10% to 90% mass burnt between conventional and delayed valve timing

Fuel	CNG	
Inlet closure	High-output	Delayed
IG to 10% mass burnt (deg.)	33	40
10% to 90% mass burnt (deg.)	33	43

Table 6 Comparison of IG to 10% mass burnt and 10% to 90% mass burnt between conventional and delayed valve timing

Fuel	CNG		Gasoline	
Inlet closure	High-output	Delayed	High-output	Delayed
Injection timing	Exhaust	Exhaust	Exhaust	Exhaust
IG to 10% mass burnt (deg.)	33	40	38	46
10% to 90% mass burnt (deg.)	33	43	30	37

Figure 11 and Table 7 show the mass burnt ratio in order to verify from the combustion that the BSFC has been enhanced. In the exhaust stroke injection, it was verified that the main 10% to 90% mass burnt that was extended by about 10 degrees was shortened by about 8 degrees, and it was understood that it recovered to a main 10% to 90% mass burnt approximately equivalent to the high output cam. It was understood that even in comparison with gasoline, the 10% to 90% mass burnt extension was shorter for the CNG charging stroke injection. Figure 12 shows the reduction effect for BSFC and ISFC when combined with charging stroke

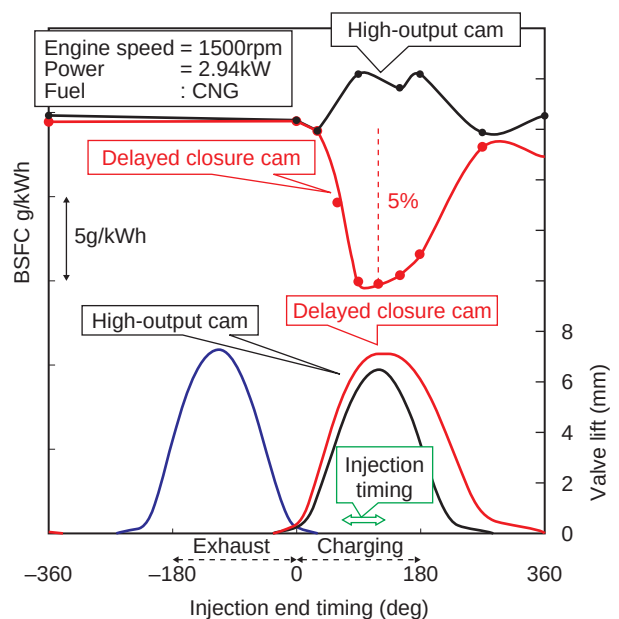


Fig. 10 Comparison of injection timing characteristics between conventional and delayed valve timing

injection. It was verified that by combining the delayed closure with the reduced charging stroke injection ISFC, a BSFC reduction equivalent to the predicted value was obtained.

Possible reasons for the enhancement are as follows:

- (1) The kinetic energy of the fuel injection that reaches the speed of sound at the injection nozzle is converted to the activation of the flow inside the cylinder.
 - (2) The 10% to 90% mass burnt was shortened by stratifying the mixture and making it richer in the vicinity of the spark plug.
- These are issues to be examined in the future.

5.3. Actual Measurement Results for Pumping Loss and Mechanical Loss

Figure 13 shows the relationship between intake negative pressure and motoring friction when the engine speed is 1500 rpm. Even with the same intake pipe negative pressure, there are differences in the motoring friction values between the high output cam and the delayed closure cam. This is likely to be because the mechanical loss in areas such as the bearings was reduced. This reduction was due to the lowering of the cylinder pressure during the compression stroke, caused by the reduction in the effective compression ratio. Figure 14 shows the motoring friction under the conditions of 1500 rpm and 2.94 kW. As shown by Table 8, the BSFC reduction effect due to friction reduction caused by the delayed closure cam was about 3.5%. It can be estimated that the

Table 7 IG to 10% mass burnt & 10% to 90% mass burnt

Fuel	CNG		Gasoline	
	High-output	Delayed	High-output	Delayed
Inlet closure	High-output	Delayed	High-output	Delayed
Injection timing	Exhaust	Charging	Exhaust	Exhaust
IG to 10% mass burnt (deg.)	33	40	38	46
10% to 90% mass burnt (deg.)	33	35	30	37

Table 8 Comparison of friction between conventional and delayed valve timing

	Power (kW)	Friction (kW)	Indicated power (kW)	BSFC reduction (%)
High-output cam	2.94	3.18	6.12	—
Delayed closure cam	2.94	2.97	5.91	3.5

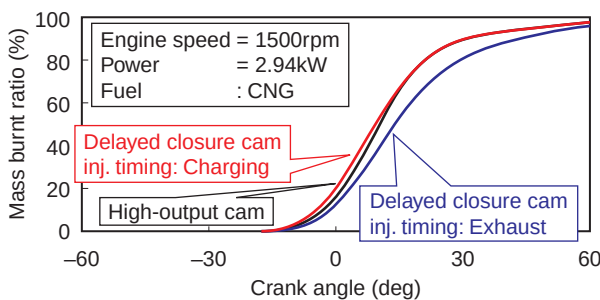


Fig. 11 Comparison of mass burnt ratio between conventional and delayed valve timing

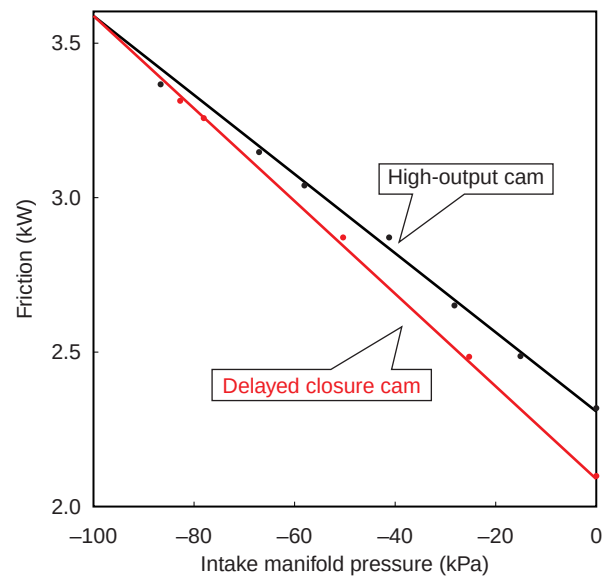


Fig. 13 Comparison of friction between conventional and delayed valve timing

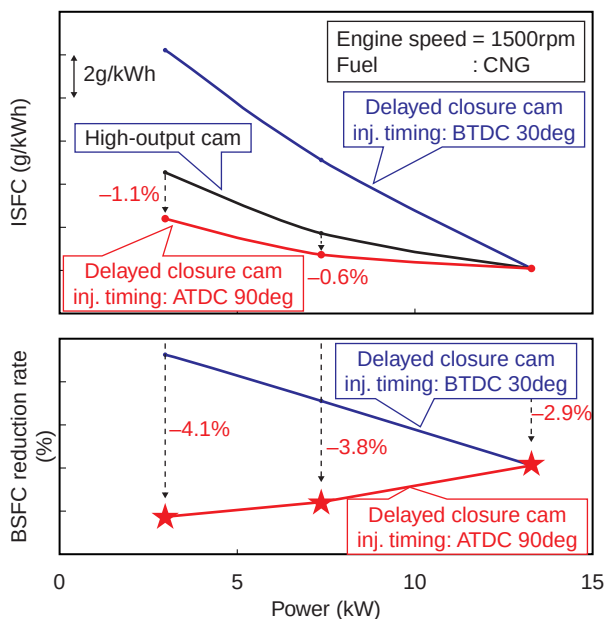


Fig. 12 Comparison of SFC between conventional and delayed valve timing

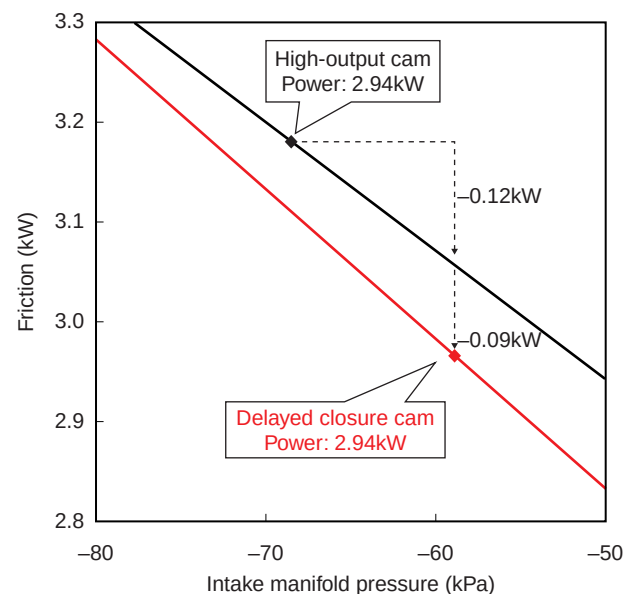


Fig. 14 Comparison of friction between conventional and delayed valve timing

effect of the pumping loss reduction due to the increase in throttle angle is about 2%. It was verified that by performing inlet valve delayed closure operations, a motoring friction reduction was achieved that exceeded the effect from the pumping loss reduction.

5.4. Engine Performance

This engine uses a new variable-length air intake manifold as shown in Fig. 15. An increase in the charging efficiency is achieved by closing the bypass valve in low to medium speed ranges and opening it in high speed ranges. Figure 16 shows the output characteristics for a wide open throttle. As shown in Table 9, there was an increase of 7% in the converted output value for engine displacement. Combined with the 8% increase in the engine displacement, this made a 15% increase in output.

As shown in Fig. 17, an excess output of 30% or greater was achieved for the output with delayed closure when the vehicle speed was 120 km/h or less. This means that delayed closure operations are possible for cruising up to 120 km/h and for slow acceleration. This enhances the fuel economy while driving and increases the product appeal.

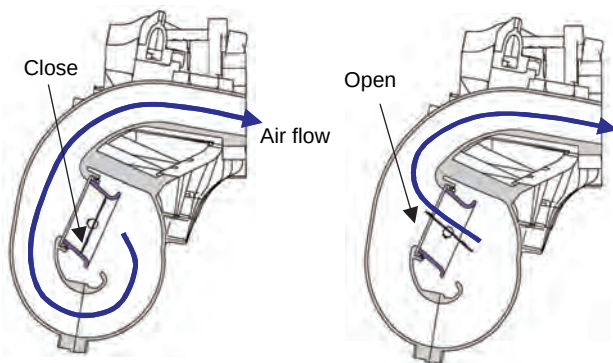


Fig. 15 Motion of bypass valves

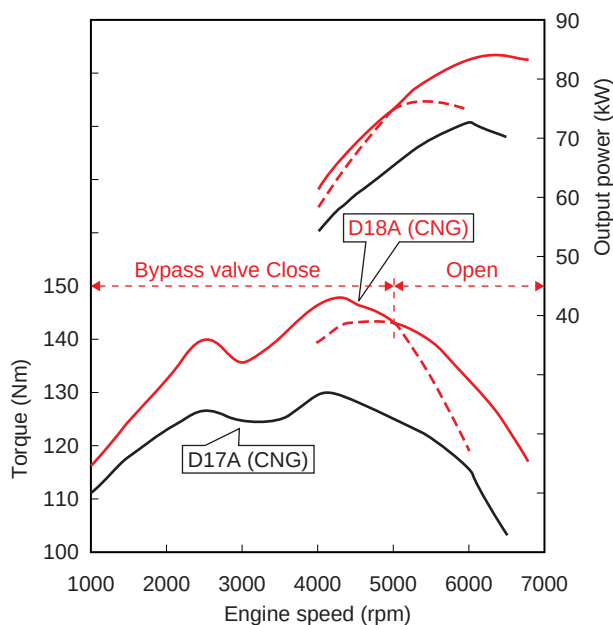


Fig. 16 Engine performance

Table 9 Comparison of engine power between D18A and D17A

	D18A (CNG)	D17A (CNG)
Max. power (kW/rpm)	84/6300	73/6000
Max. torque (Nm/rpm)	148/4300	130/4200

5.5. Actual Vehicle Performance

5.5.1. Exhaust emission performance

The 2006 model uses a cylinder head that is integrated with the exhaust manifold and close-coupled two-bed catalyst, and this is shown in Fig. 18. By concentrating the catalytic converter in a close-coupled position where a high temperature can be maintained, the catalyst precious metal amount is reduced by 30% compared to the previous model. This achieves exhaust emission levels that comply with the ATP-ZEV regulations.

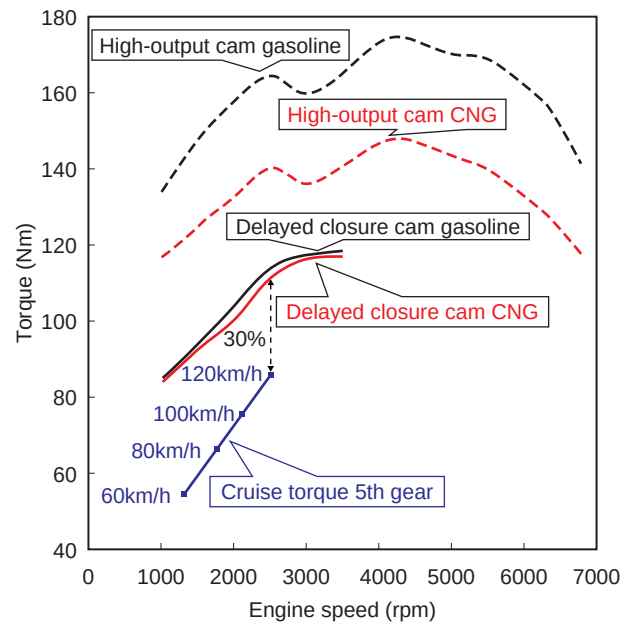


Fig. 17 Operating region of delayed closure cam

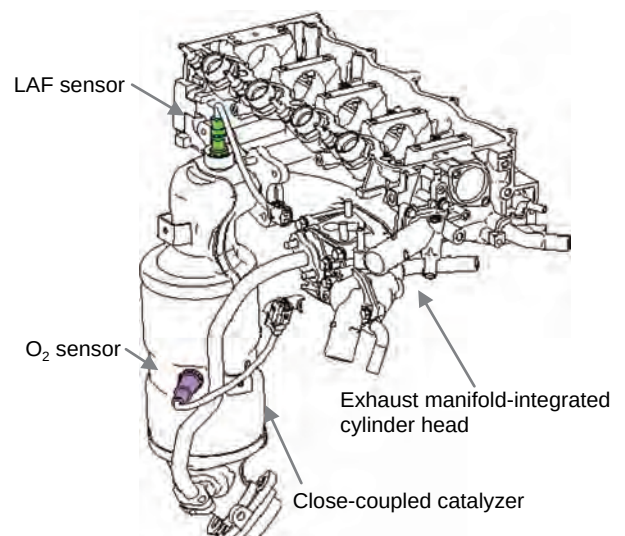


Fig. 18 Exhaust and catalytic converter system

5.5.2. Achieved fuel economy values

In addition to the previously mentioned fuel economy enhancement effect of delayed closure, the use of a variable-length air intake increases the maximum torque by 14%. This creates a high transmission ratio, and lowers the engine speed during cruising to enhance the fuel economy. Table 10 shows the achieved fuel economy values for the 2006 and 2001 models. The fuel economy values in the table for CNG are values converted to a gasoline equivalent (GGE: Gasoline Gallon Equivalent). For a comparison of driving conditions, Fig. 19 shows FTP city mode, and Fig. 20 shows highway mode. The delayed closure frequency is a high 55% in highway mode, where cruising is frequent. Despite the increased weight accompanying the larger vehicle design, a fuel economy enhancement of 15% was achieved. Even in combined mode, a fuel economy equivalent to the 2001 model was achieved.

Table 10 Achieved fuel economy of NGV (Label FE)

	Label FE (mile/GGE)				
	City	HWY	Comb	I.W. (lb)	Transmission
2006 model	28	39	32	3250	5AT
2001 model	30	34	32	3000	CVT

6. Conclusion

A CNG engine was developed that uses a variable valve train with inlet valve delayed closure. Cause analysis was performed to enhance the thermal efficiency. As a result, the increase in the 10% to 90% mass burnt during low loads that accompanies the reduction in the effective compression ratio was enhanced with charging stroke injection.

The following technical enhancements have been achieved in comparison with the 2001 model.

- (1) An enhancement in engine fuel economy of 4% has been achieved. This was done by reducing the pumping loss with inlet valve delayed closure, and by enhancing combustion through charging stroke injection that harnesses the characteristics of gaseous fuel.
- (2) Output for engine displacement has increased by 7% compared with the 2001 model engine, and when combined with the additional engine displacement amount there is a total output increase of 15% compared with the previous model.
- (3) As a countermeasure to reduce exhaust emissions, a cylinder head integrated with the exhaust manifold and a close-coupled two-bed catalyst are used. This also reduces the amount of precious metal by 30% and achieves compliance with the ATP-ZEV regulations.
- (4) Despite the increased weight accompanying the larger vehicle design, a fuel economy enhancement of 15% was achieved in highway mode, and a fuel economy equivalent to the previous model was achieved in combined mode.

7. Afterword

CNG is likely to be useful not only as a countermeasure for emissions, but also for reducing carbon dioxide which is a major contributor to global warming. Such countermeasures have been increasingly used worldwide in recent years. In addition, vast amounts of methane hydrate reserves have been confirmed. The household charging machines launched in 2005 are likely to address the issue of the low number of charging facilities. In the future we aim to further spread the engine among general users, and take action to enhance its appeal as a product.

References

- (1) Seko, K., Taga, W., Torii, K., Nakamura, S., Akima, K., Sekiya, N.: Development of 1.8L i-VTEC Gasoline Engine for 2006 Model Year Honda CIVIC, Honda R&D Technical Review, Vol. 18, No. 1, p. 8-15
- (2) Seko, K., Hayashi, A., Nakajima, M.: Achievement of Enhanced Fuel Economy via i-VTEC Intake Valve Closure Delay Mechanism, Honda R&D Technical Review, Vol. 18, No. 1, p. 88-93
- (3) Toda, T., Matsuki, M., Ohmori, T., Yamamoto, Y.: Development of a 1.7L CNG Engine for the 2001 Civic GX, Honda R&D Technical Review, Vol. 14, No. 1, p. 49-56

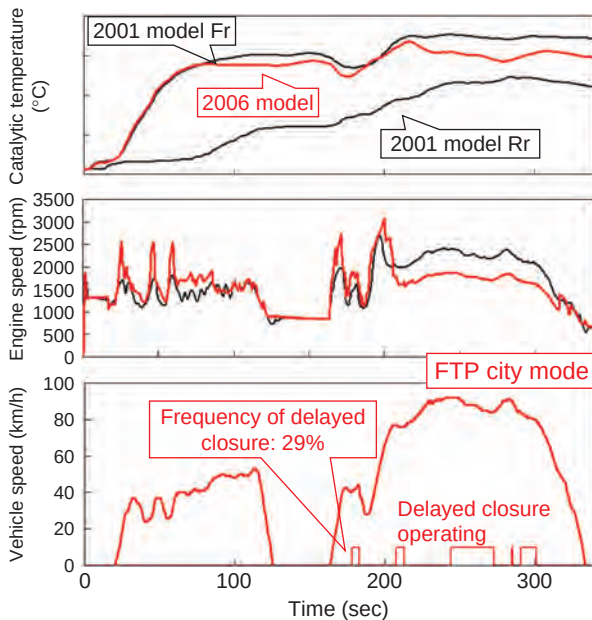


Fig. 19 Engine speed and catalyst temperature during FTP city mode

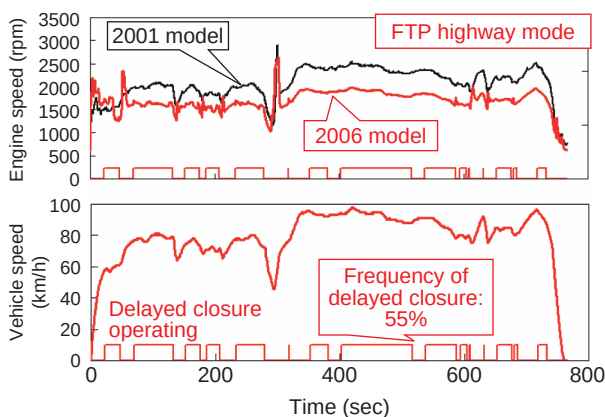


Fig. 20 Engine speed during FTP highway mode

- (4) Arai, S., Muraishi, T.: HONDA Compressed Natural Gas Vehicle, HONDA R&D Technical Review, Vol. 9, p. 17-24
- (5) Watanabe, S., Kono, S., Kurata, M., Koga, H.: Research on Extended Expansion General-Purpose Engine—Theoretical Analysis of Multiple Linkage System and Improvement of Thermal Efficiency—, Honda R&D Technical Review, Vol. 18, No. 1, p. 81-87

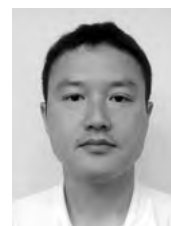
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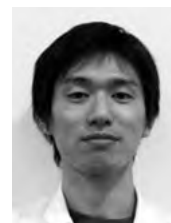
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Accurate Idle Speed Control for Gasoline Engines with Continuously Variable Valve Actuation

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ABSTRACT

A gasoline engine with continuously variable valve actuation uses the valves instead of the throttle to control the suction air amount, thus reducing pumping loss. However, just a slight change in valve lift causes the suction air amount to vary greatly, which makes it difficult to accurately control idle speed when using conventional idle speed control.

Accordingly, in this research a new method of controlling the idle speed was proposed. This new idle speed control features coordination between suction air flow control and ignition timing control, each of which uses a two-degree-of-freedom sliding mode algorithm. Large deviations are corrected by air flow control, while small deviations are corrected by ignition timing control, thus realizing fine control of engine speed, which is necessary for reducing the idle speed.

As a result, it was possible to reduce the idle speed of a gasoline engine equipped with a continuously variable valve actuation, as well as increase the idle speed stability.

1. Introduction

Concern about global warming has increased in recent years, giving rise to social demands for reduced CO₂ emissions and increasing the importance of enhancing automobile fuel efficiency. As a result, gasoline engines with continuously variable valve actuation are garnering attention as a technology that increases fuel efficiency⁽¹⁾⁻⁽³⁾.

A gasoline engine with continuously variable valve actuation uses the valves instead of the throttle to control suction air amount, which reduces pumping loss. On the other hand, the airflow amount becomes more sensitive to even slight changes in the valve lift compared to conventional gasoline engines with throttles. This makes it a challenge to accurately control the engine speed using suction air flow, such as for idle speed control.

Conventional idle speed control consists of the ignition timing control and suction air control based on independent PI control. Therefore, the ignition timing and suction air controls interfere with

each other, resulting in overshoot with respect to changes in the target value.

To resolve this issue, conventional idle speed control uses a method that stops feedback and performs preset open-loop control when the target value changes in the downward direction or the engine speed drops. However, robustness (stability) with respect to parts deterioration is an issue with this method.

Therefore, it is necessary to construct a closed-loop idle speed control that does not produce overshoot when the target value changes or when recovering to the target idle speed after deviation occurs for some reason.

Sliding mode control⁽⁴⁾, which is a type of robust control, is one method that has recently been applied to engines^{(5),(6)} and other devices⁽⁷⁾.

Therefore, this study designed a new idle speed control based on cooperation between two types of sliding mode control to lower the idle speed of a gasoline engine with continuously variable valve actuation.

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2. Accurate Cooperative Two-Input Idle Speed Control

Accurately Cooperative Two-input Idle speed Control (ACTIC) was devised to lower the idle speed of a gasoline engine with continuously variable valve actuation.

In ACTIC, the control of suction air and ignition timing applied a two-degree-of-freedom sliding mode algorithm to accurately control the engine speed. In addition, cooperative control is realized between the suction air and ignition timing controls to prevent the two types of control from interfering with each other. An overview of this two-degree-of-freedom sliding mode control and cooperative control is described below.

2.1. Two-Degree-of-Freedom Sliding Mode Control

Two-degree-of-freedom sliding mode control is a sliding mode control that can separately designate the target value response speed and the deviation convergence speed. The target value response speed is designated using a target value filter, and the deviation convergence speed is designated using a switching function.

Labeling the engine speed as N_E and the set point of idle speed as N_r , the target value N_f used by sliding mode control is calculated using the target value filter shown in Eq. (1).

$$N_f(k) = -Pole_f N_f(k-1) + (1 + Pole_f) N_r(k) \quad (1)$$

The N_f behavior can be designated by the filter ratio $Pole_f$ of the target value filter. Therefore, the target value response behavior in sliding mode control can also be designated by using this N_f as the control target value.

With sliding mode control, the deviation $E_{NE}(k)$ between the target value $N_f(k-1)$ and the engine speed $N_E(k)$ is obtained using Eq. (2). The switching function $\sigma_{NE}(k)$ is defined as shown in Eq. (3).

$$E_{NE}(k) = N_E(k) - N_f(k-1) \quad (2)$$

$$\sigma_{NE}(k) = E_{NE}(k) + Pole E_{NE}(k-1) \quad (3)$$

Figure 1 shows a phase plane with $E_{NE}(k)$ and $E_{NE}(k-1)$ as the respective coordinate axes. The straight line where the switching function σ_{NE} becomes equal to zero on this phase plane is called the switching line.

Sliding mode control performs control so that the combination of $E_{NE}(k)$ and $E_{NE}(k-1)$ (hereafter the “deviation state”) is restricted to the switching line, and the deviation state is then converged along the switching line to the phase plane origin. In other words, the deviation $E_{NE}(k)$ is converged to zero. Here, the switching line is expressed by Eq. (4) based on the relationship between $E_{NE}(k)$ and $E_{NE}(k-1)$. In addition, $Pole$ is set within the range given by Eq. (5).

$$E_{NE}(k) = -Pole E_{NE}(k-1) \quad (4)$$

$$-1 < Pole < 0 \quad (5)$$

Therefore, the deviation convergence speed can be designated by $Pole$, which is the switching line gradient.

2.2. Cooperative Control

To achieve cooperation between suction air control and ignition timing control in ACTIC, the target value filter that designates the target value response speed and the switching function that specifies the deviation convergence speed are shared by both controls. In

addition, cooperative control is achieved by using the following characteristics of suction air control and ignition timing control.

Ignition timing control

- No dead time and little response delay
- Small N_E change width
- High N_E control resolution

Suction air control

- Dead time and large response delay
- Large N_E change width
- Low N_E control resolution

The two controls achieve cooperation by using these characteristics to switch the initiative control as follows.

- Tracking to large target changes and reducing large deviations
→ Suction air control is the initiative control.
- Tracking to small target changes and reducing small deviations
→ Ignition timing control is the initiative control.

3. Control Design

Figure 2 shows the ACTIC system diagram. The details of ignition timing control, suction air control and cooperative control are described below.

3.1. Ignition Timing Control

Ignition timing control comprises a sliding mode controller consisting of the reaching law input $U_{rch_ig}(k)$ and adaptive law input $U_{adp_ig}(k)$. The forgetting integration ratio λ is introduced to the adaptive law input $U_{adp_ig}(k)$ in order to gradually return to the pre-decided ignition timing when idling after converging to the target value. This avoids the need to maintain the ignition timing in an excessively retarded state, and suppresses the corresponding drop in fuel efficiency. The calculation formulas are as follows.

$$U_{sl_ig}(k) = U_{rch_ig}(k) + U_{adp_ig}(k) \quad (6)$$

$$U_{rch_ig}(k) = K_{rch_ig} \sigma_{NE}(k) \quad (7)$$

$$U_{adp_ig}(k) = K_{adp_ig} \sigma_{NE}(k) + \lambda U_{adp_ig}(k-1) \quad (8)$$

K_{rch_ig} , K_{adp_ig} : Feedback gain

λ : Forgetting integration ratio ($0 < \lambda < 1$)

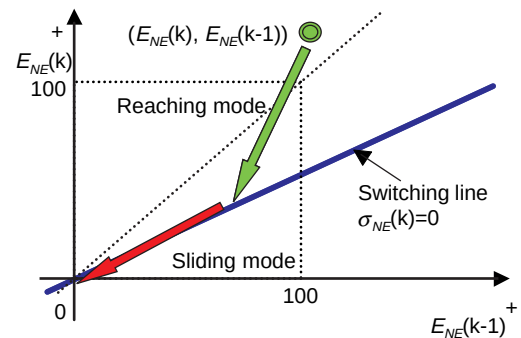


Fig. 1 Sliding mode control

3.2. Suction Air Control

Suction air control consists of a sliding mode controller that is calculated based on a controlled object model that represents the input/output relation, and an adaptive disturbance observer that corrects the controlled object model representing the input/output relation. The details of the controlled object model, sliding mode controller and adaptive disturbance observer are described below.

3.2.1. Controlled object model

The controlled object model that expresses the relationship between engine speed $N_E(k)$ and control input $U_{sl_ar}(k)$ is defined as shown in Eq. (9).

$$\begin{aligned} N_E(k+1) = & \quad (9) \\ & a_1 N_E(k) + a_2 N_E(k-1) + b_1 U_{sl_ar}(k) + b_2 U_{sl_ar}(k-1) + c_1(k) \\ & a_1 a_2 b_1 b_2 : \text{Model parameters} \\ & c_1(k) : \text{Estimated disturbance value} \end{aligned}$$

The model parameters a_1 , a_2 , b_1 and b_2 are predetermined by system identification. In addition, the estimated disturbance value c_1 is calculated consecutively by the adaptive disturbance observer.

3.2.2. Adaptive disturbance observer

The adaptive disturbance observer is calculated based on the deviation $E_{dov_ar}(k)$ between the $N_E(k)$ and the $N_{E_hat}(k)$ calculated using the controlled object model prescribed by Eq. (9). In other words, the c_{1_ar} that results in the minimum difference between N_{E_hat} and N_E is estimated by the fixed gain method used by adaptive control. The calculation formulas are as follows.

$$\begin{aligned} N_{E_hat}(k) = & \\ & a_1 N_E(k-1) + a_2 N_E(k-2) + b_1 U_{sl_ar}(k-1) \\ & + b_2 U_{sl_ar}(k-2) + c_{1_ar}(k-1) \end{aligned} \quad (10)$$

$$E_{dov_ar}(k) = N_E(k) - N_{E_hat}(k) \quad (11)$$

$$c_{1_ar}(k) = c_{1_ar}(k-1) + \frac{P_{ar}}{1 + P_{ar}} E_{dov_ar}(k) \quad (12)$$

P_{ar} : Fixed gain

The adaptive disturbance observer absorbs the modeling error of the controlled object model prescribed by Eq. (9) by estimating the disturbance element c_{1_ar} . This achieves a high robustness with respect to disturbances and parts deterioration.

3.2.3. Sliding mode controller

The sliding mode controller used by suction air control consists of the reaching law input $U_{rch_ar}(k)$ and the equivalent law input $U_{eq_ar}(k)$. The calculation formulas are as follows.

$$U_{sl_ar}(k) = U_{rch_ar}(k) + U_{eq_ar}(k) \quad (13)$$

$$U_{rch_ar}(k) = K_{rch_ar} \sigma_{NE}(k) \quad (14)$$

$$\begin{aligned} U_{eq_ar}(k) = & \\ & \frac{1}{b_1} \{ (1 - a_1 - Pole) N_E(k) + (Pole - a_2) N_E(k-1) \\ & + N_f(k) + (Pole - 1) N_f(k-1) - Pole N_f(k-2) \\ & - b_2 U_{sl_ar}(k-1) - c_1(k) \} \end{aligned} \quad (15)$$

K_{rch_ar} : Feedback gain

$U_{eq_ar}(k)$ has a function that constrains the deviation state onto the switching line prescribed by the switching function. The estimated disturbance value c_{1_ar} is inserted to add a disturbance compensation function to this $U_{eq_ar}(k)$.

Ignition timing control performs disturbance compensation using $U_{adp_ig}(k)$, which is based on the integration value of the switching function. In contrast, suction air control performs disturbance compensation by $U_{eq_ar}(k)$, which uses the $c_{1_ar}(k)$ estimated by the adaptive disturbance observer. c_{1_ar} estimates the disturbance using the aforementioned fixed gain method, so the disturbance

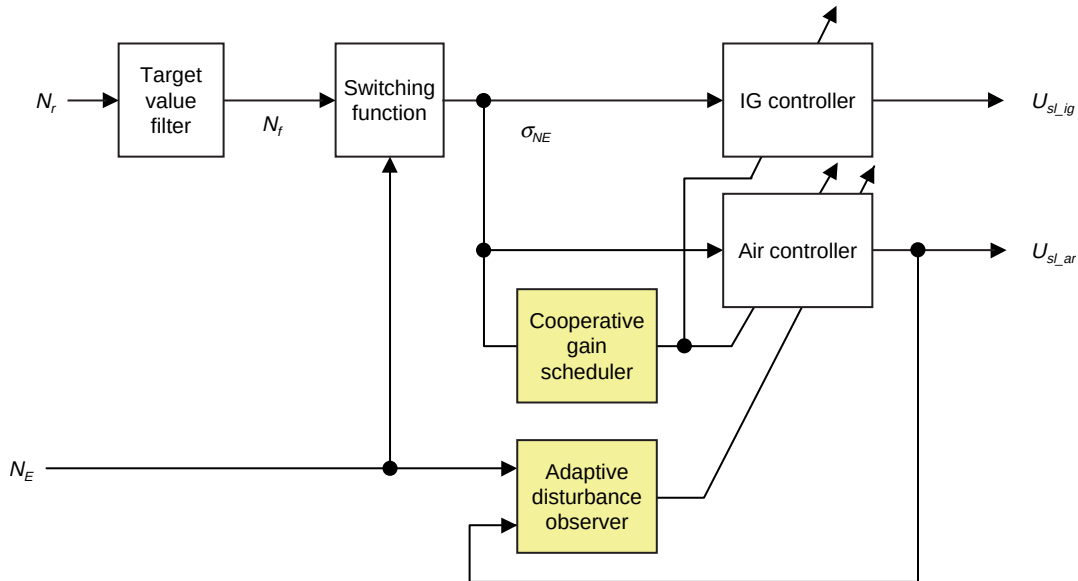


Fig. 2 System diagram of ACTIC

compensation responsiveness is better than for $U_{adp_ig}(k)$, which performs disturbance compensation using an integrator.

3.3. Cooperative Gain Scheduler

ACTIC achieves cooperative control by determining the initiative control and performing the appropriate switching with a cooperative gain scheduler.

As shown in Fig. 3, the initiative control is judged to be ignition timing control in areas with a small switching function value σ_{NE} , and suction air control in areas with a large switching function value σ_{NE} .

In addition, a low suction air control gain is set in the area where ignition timing control is the initiative control, and a low ignition timing control gain is set in the area where suction air control is the initiative control. The initiative control is switched by changing the control gain in this manner. Figures 4 and 5 show the relation between the ignition timing control gain and the suction air control gain.

4. Vehicle Test Results

The stability, responsiveness and disturbance rejection capability of ACTIC and conventional idle speed control were compared. In addition, the behavior after engine start and fuel consumption were compared with and without forgetting integration to illustrate the effects of the ACTIC forgetting integration. The results are described below.

4.1. Stability

Figure 6 compares the stability of ACTIC and conventional idle speed control in the unloaded condition with an idle speed set point of 600 rpm. These results show that ACTIC has higher stability than conventional idle speed control.

4.2. Responsiveness and Disturbance Rejection Capability

To compare responsiveness and disturbance rejection capability, the engine speed behavior was measured when the idle speed set point was changed from 600 rpm to 720 rpm while simultaneously

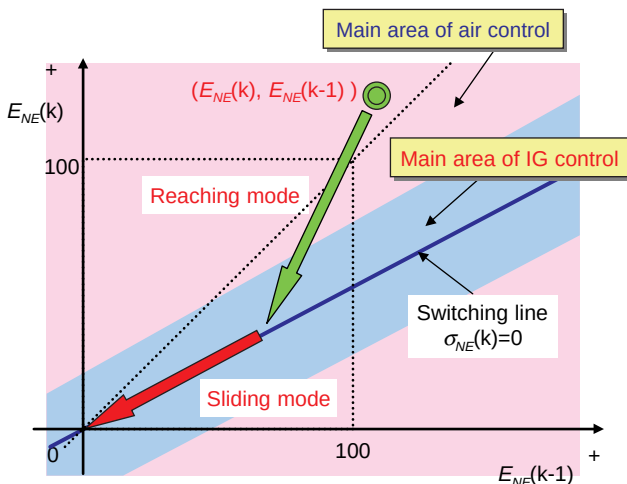


Fig. 3 Main controller switching

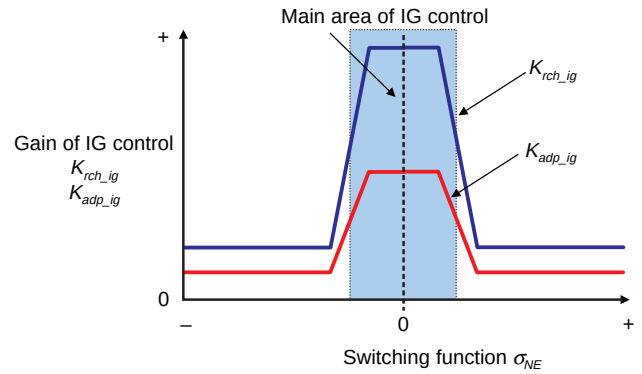


Fig. 4 Gain scheduling of ignition timing control

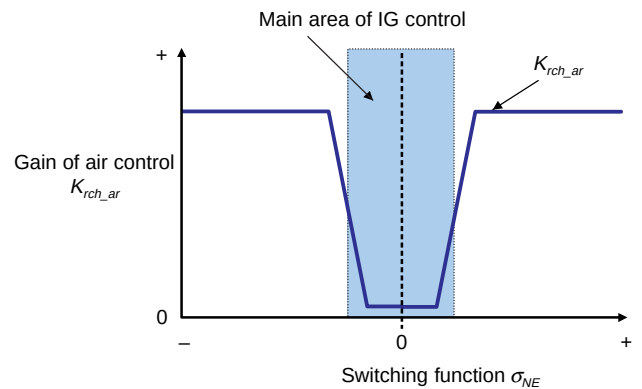


Fig. 5 Gain scheduling of suction air control

applying a load input (air conditioning load). Figure 7 shows the results.

Conventional idle speed control was able to suppress the drop in engine speed caused by the air conditioning load (disturbance). However, the engine speed overshoot the set point of idle speed, resulting in wasteful over-revving. This overshoot is caused by interference between ignition timing control and suction air control.

Comparatively, ACTIC was able to suppress the drop in engine speed caused by the air conditioning load (disturbance) in the same manner as conventional idle speed control. In addition, ACTIC also reduced engine speed over-revving due to overshoot compared to conventional idle speed control.

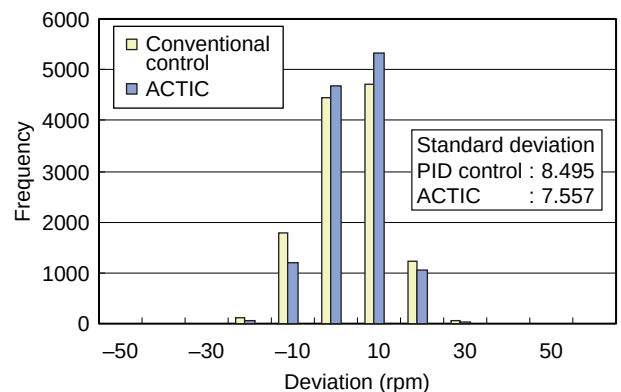


Fig. 6 Comparison of stability with idle speed control

These results show that ACTIC reduces interference between suction air control and ignition timing control using cooperative control, and realizes higher target value responsiveness and disturbance rejection capability than conventional idle speed control.

4.3. Forgetting Integration

The behavior after engine start was measured with and without forgetting integration to confirm its effects. Figure 8 shows the engine speed, ignition timing and valve lift behavior after engine start until the engine speed settled down to the idle speed set point of 600 rpm.

Forgetting integration caused the ignition timing to return to the preset ignition timing, and the valve lift also moved toward a lower lift at the same time.

Next, Fig. 9 compares the standard deviation of the engine speed after converging to the idle speed set point of 600 rpm with and without forgetting integration.

These results show that the idle speed stability decreases without forgetting integration. The idle speed control stability decreases because the ignition timing is excessively retarded, causing the torque fluctuations to increase.

In addition, Fig. 10 compares the fuel consumption after converging to the idle speed set point of 600 rpm with and without forgetting integration.

The fuel consumption increases without forgetting integration compared to that with forgetting integration. This is also caused by excessive retardation of the ignition timing.

These results confirm that the introduction of forgetting integration can suppress the drop in fuel efficiency when idling and also avoid reduced stability due to excessive ignition timing retard.

5. Conclusion

A new idle speed control (ACTIC) that cooperatively controls the ignition timing and suction air flow was introduced to lower the idle speed in a gasoline engine with continuously variable valve actuation. Vehicle test results provided the following information.

- (1) Stability is enhanced at an idle speed of 600 rpm.
- (2) There is no interference between ignition timing control and suction air control, creating enhanced responsiveness to target speed changes and disturbance rejection capability.
- (3) The forgetting integration of ignition timing control can suppress excessive retardation of the ignition timing, which enhances both idle speed stability and fuel efficiency.

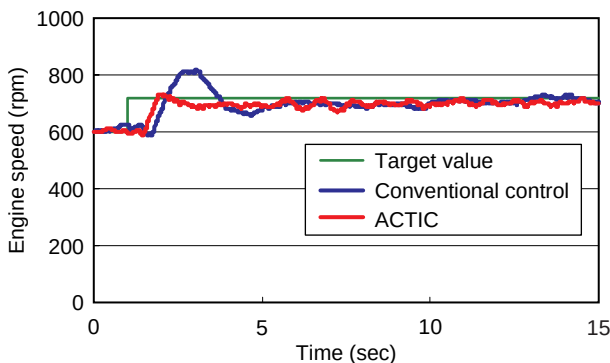


Fig. 7 Idle speed control response when disturbance is input

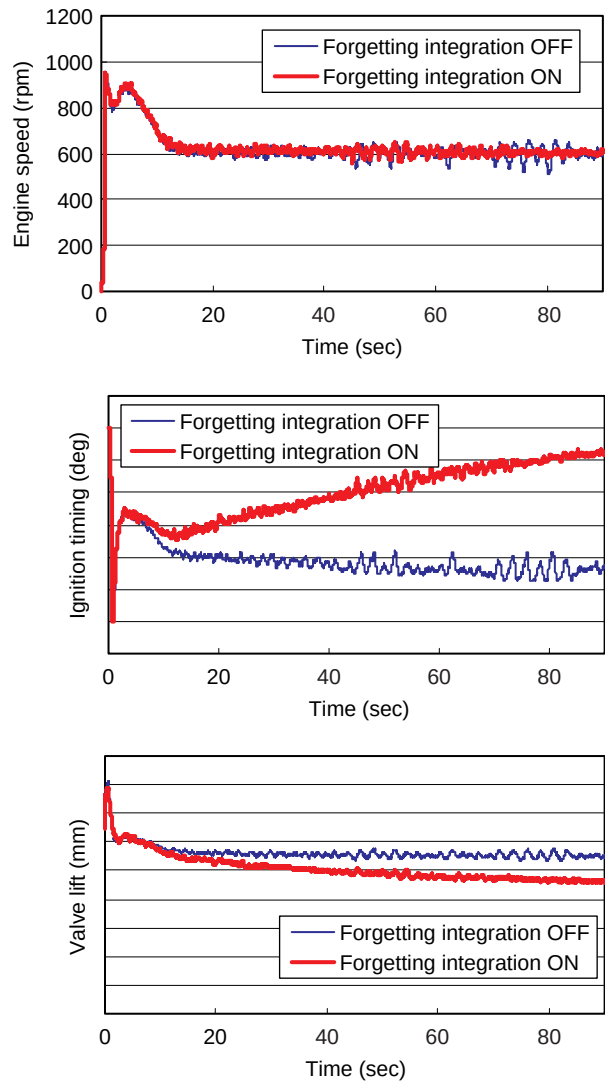


Fig. 8 Influence of forgetting integration on ignition timing and inlet valve lift behavior

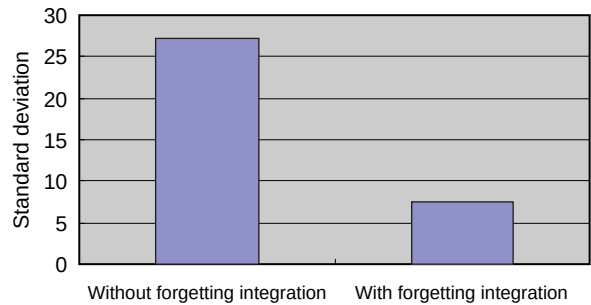


Fig. 9 Influence of forgetting integration on idle speed stability

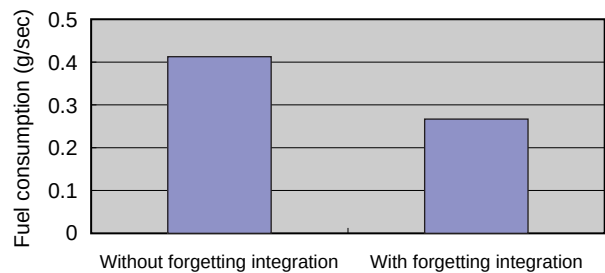


Fig. 10 Influence of forgetting integration on fuel consumption

References

- (1) Yamane, K.: Fully Variable Valve Actuation System from BMW, VALVETRONIC, Engine Technology, Vol. 5 (2003)
- (2) Arinaga, T.: A Study of Friction Characteristics of Continuously Variable Valve Event & Lift (VEL) System, SAE Paper 2006-01-0222 (2006)
- (3) Himsel, F.: INA EcoValve A Continuously Variable Valve Lift (CVVL) System, SAE Paper 2004-01-1391 (2004)
- (4) Nonami, K., Den, K.: Sliding Mode Control: Design Theory of Non-linear Robust Control, Corona Publishing Co., Ltd., 263p. (1994)
- (5) Yasui, Y., Akazaki, S., Ueno, M., Iwaki, Y.: Secondary O2 Feedback Using Prediction and Identification Type Sliding Mode Control, SAE Paper 2000-01-0936 (2000)
- (6) Yasui, Y., Shimojo, K., Saito, M., Tamagawa, Y.: Rapid Engine Speed Control for AMT Using Two-Degree-of-Freedom Sliding Mode Algorithm, SAE Paper 2005-01-1592 (2005)
- (7) Hashimoto, E., Ishiguro, T., Yasui, Y., Akazaki, S.: High Reliability Electronic Throttle System Design, SAE Paper 2003-01-0708 (2003)

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Proposal of New Bearing Simulation Rig Tester for High Efficiency Engine Bearing Development

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ABSTRACT

To select bearing specifications for an engine using a bearing rig test, it is necessary to achieve sliding conditions that are near the actual operating conditions of the engine. A new bearing rig tester with low resonance and high rigidity permits high frequency excitation and has a mechanism to correct misalignment between the axle and the bearing. This rig tester can reproduce load fluctuations under the actual operating conditions of the engine. The test equipment makes it possible to clarify for the first time the mechanism that causes the difference in the break-in performance of lead alloy and aluminum alloy bearings. It was possible to evaluate the small end bearing fitted on an actual connecting rod, and the sliding characteristics caused by different surface treatment methods for the small end bearings. In addition, it was possible to measure the friction coefficient of the bearings with high repeatability, and efficiently evaluate the bearing material.

1. Introduction

As automobile engines have become smaller and lighter, bearing widths have decreased. Simultaneously the engine output, performance and combustion pressure have all increased. These and other factors impose a greater load on the bearings every year. Bearing materials have also changed, particularly with the introduction of eco-friendlier lead-free bearings⁽¹⁾⁻⁽³⁾. To develop an efficient engine, it is important to carry out conformance tests on many parts before they are used in an actual engine.

Because the function of a bearing is greatly affected by conditions in the vicinity of the housing, methods such as elastohydrodynamic lubrication (EHL) analysis have been used to analyze the causes of bearing housing deformation^{(4),(5),(9),(10)}, changes in the viscosity of the lubricant caused by the temperature^{(6),(8)}, shaft profile changes⁽⁷⁾,

and other factors. The more severe the sliding conditions become, the greater the necessity to take into account many factors including those mentioned above. Also, it is still difficult to account for transient deformation, which is necessary for expressing the break-in characteristics; transient changes in plasticity accompanying wear; and transient state changes accompanying wear. This means that it is still necessary to rely on actual engine tests to determine most of these factors. On the other hand, to develop the bearing vicinity more efficiently using conventional static load bearing testers, it is necessary to bring the sliding conditions, including bearing housing deformation, closer to those of an actual engine.

This paper describes the increased efficiency in evaluating bearing rig sliding performance using a newly designed bearing tester, as well as its influence in the selection of the actual engine specifications.

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2. Bearing Rig Tester

2.1. Conventional Tester

Figure 1 is a configuration diagram⁽¹¹⁾ of the standard static load bearing tester used until now.

Because this tester has been designed to produce a static load, it cannot generate a dynamic load. Figure 2 shows an example of calculating the connecting rod small end bearing load when the engine is operating, and the static load applied. In addition, it shows the position of the connecting rod small end of the piston pin versus load.

The magnitude and direction of a dynamic load change over time, and the thickness of the oil film between the piston pin and the connecting rod small end bearing changes too. However, in the case of a static load, these parameters remain constant with time. The oil film is maintained under a static load due to the wedge action (effect due to pressure generated by the relative velocity between the shaft and the bearing in the sliding direction). In the case of a dynamic load, the direction and magnitude of the load change, resulting in a squeeze action (effect due to pressure generated by the velocity of the shaft and the radial direction of the bearing). As a result of this effect, a comparatively thick oil film is maintained at the maximum load, and the main direction and magnitude of heat generated by oil shear due to the change in the load direction are dispersed, and the temperature of the sliding face also falls. The oil film pressure of the sliding face generated by the

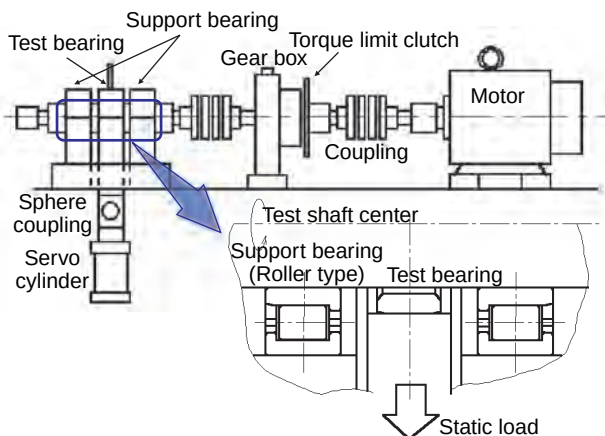


Fig. 1 Conventional system

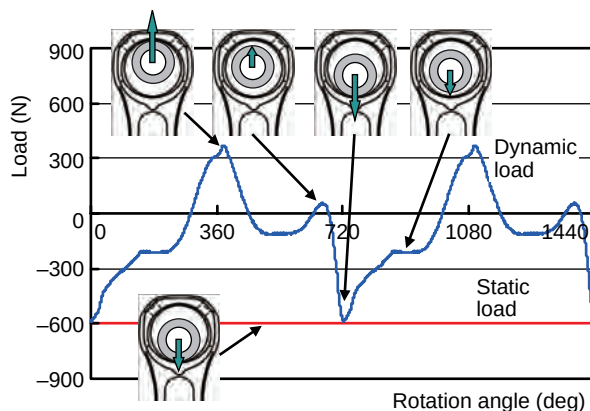


Fig. 2 Differences in sliding parts by load type

Table 1 Performance of new bearing rig tester

Item	Performance
Bearing slide conditions: Similar to actual engine	• Use of various entity connecting rods • Synchronization of X, Y direction loads • Generation of various load waves • Frequency: 300 Hz Load: ± 10 kN • Synchronization of bearing load with shaft angle
Tester protection from seizure	• Construction that can separate torque meter from motor torque
Repeatability of test results	• Adjusting mechanism of bearing-shaft tilt • Roller-type support bearing • Multi-point measurement of bearing temperature

wedge and squeeze actions causes elastic deformation of the piston pin and connecting rod small end bearing. This deformation is relatively large compared to the oil film thickness that is formed, so it changes the sliding conditions. Also, the lower the rigidity of the housing, the greater the deformation, and hence the greater its effect.

Given the above factors, the only way to make the sliding conditions closer to those of an actual engine is to use a dynamic load, and so it was necessary to develop a new tester.

2.2. New Bearing Tester

Table 1 shows the functions and features of the new tester. None of these functions are available in commercially available testers. To get the full performance out of the exciter, it is necessary to study not only the exciter but many other parts. Almost all parts are being newly designed with reference to previous research projects⁽¹²⁾⁻⁽²⁴⁾ to enable the tester to be used for as wide a range of tests as possible.

2.2.1. Overall configuration of the tester

Figure 3 shows the overall configuration diagram of the developed tester.

The torque meter was designed to withstand high rotational speeds, and the gear box with torque limiter clutch can reliably protect the rotational system from the abrupt generation of friction. As a result, the torque meter can be left installed up to the highest speed test conditions, enabling friction to be measured at all times.

If plain bearings were used as support bearings, heat generated in the support bearings would be transmitted to the test section, affecting the results. Also, when plain bearings undergo temporal change as a result of wear, the friction and oil flow rates change, causing the temperature of the test bearings to change as well. To prevent this and other issues from occurring, roller bearings are used. In the new tester, a double row of roller bearings is used to ensure rigidity when a dynamic load test is being performed.

The actuator and servo valve, which generate the excitation pressure, are larger than those used in conventional static load tests, so the frame surrounding the actuator is being completely reevaluated.

The new tester also has a new control system that adjusts the tilt angle of the shaft and the bearing when tests are being performed.

2.2.2. Study of the tester frame

To ensure satisfactory excitation at high frequencies, it is essential that the frame is rigid to minimize displacement and nullify the effects of resonance. The target excitation frequency is 300 Hz, so it is necessary to keep the resonant frequencies above this value. Figure 4 shows the FEM model of the frame that was studied, and the resonant frequencies obtained from calculation.

Generally, the exciter has a columnar frame. However, a study

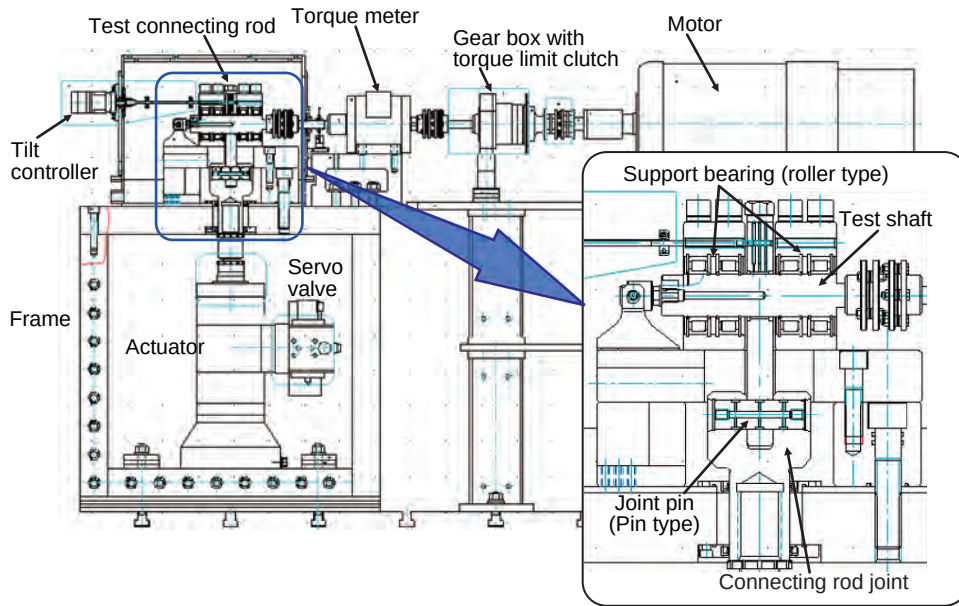


Fig. 3 Structure of new bearing rig tester

has shown that with the columnar construction, it is not possible to achieve the target frequency by either making the columns thicker or increasing the number of columns. Accordingly, a box structure using a sheet thickness that is highly rigid, albeit at the expense of maintainability, was used. Also, the frame structure of the newly fabricated tester was designed in such a way that the frame for the horizontal direction exciter can be joined to the frame for the vertical direction exciter, to enable the test bearing to be excited not only in the vertical direction but also in the horizontal direction as well.

2.2.3. Study of the bearing tester

To excite the test bearing at a high frequency and under a large load, it is necessary to minimize deformation under load of both the frame and the tester. This is because hydraulic oil pump capacity and servo valve flow characteristics are limited. The main items implemented for this purpose are as follows:

- (1) Use of a double row of support roller bearings and an increased interference setting for the outer race
- (2) Use of a more rigid support bearing housing

- (3) Elimination of the ball joint between the connecting rod and connecting rod joint and minimization of the clearance between them
- (4) Use of an increased number of clamping bolts for each part

Figure 5 shows photographs of the conventional tester and the new tester, and also the time system waveform of load versus output from the displacement sensor mounted on the actuator when these testers are used. As a result, it was found that with the conventional tester, the ± 100 kN-converted displacement was 1.60 mm, which is excessively large. It was found that there were two causes for this displacement; deflection of the various parts, and looseness due to clearances. By enhancing the rigidity of the parts concerned and minimizing the various clearances, it was possible to reduce the amplitude to a ± 100 kN converted value of 0.59 mm.

		Column-type frame	Box-type frame
FEM model			
Resonance frequency (Hz)	Front and rear	280.4	389.3
	Right and left	264.6	437.0
	Twist	222.3	750.0
	Above and below	480.6	787.1

Fig. 4 Resonance frequency difference by frame type

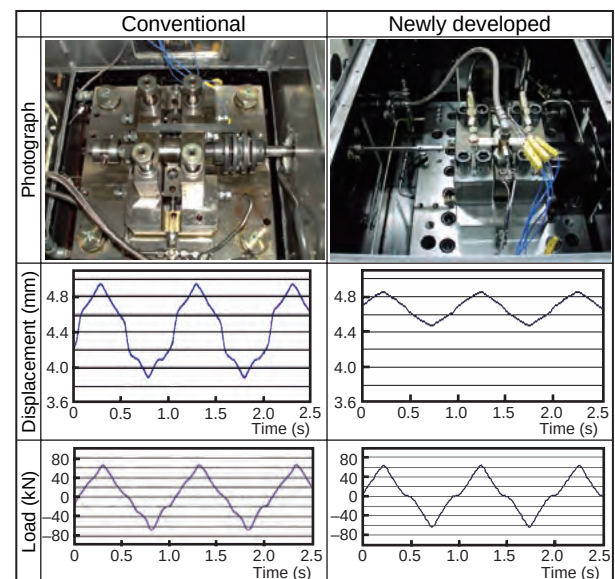


Fig. 5 Displacement of exciter piston by load

2.2.4. Tilt control system

The results of the bearing test are affected by a number of factors.

Compatibility of data is important when performing a comparison of the bearing itself, such as a comparative evaluation of the basic performance of the bearing material. In such a case, it is necessary to remove influencing factors other than the bearing, such as the rigidity of the housing, and so on, before carrying out the tests. The use of a highly rigid connecting rod such as that shown in Fig. 6 is effective.

However, when a highly rigid connecting rod is used, misalignment of the connecting rod and the shaft greatly affects the results of the test. It is thought that this is due to the fact that with a low rigidity connecting rod, parallel contact is achieved by a self-fitting process resulting from elastic deformation and bending of the stem of the connecting rod. With a highly rigid connecting rod, this self-fitting effect is minor. To address this issue when using the previously used static load tester, a ball joint was used at the connection point between the exciter and the connecting rod. It was found, however, that as the load increased, the effectiveness of these bearings was reduced and the displacement under load increased. Use of the ball joint was therefore discontinued.

Figure 7 shows the test bearing temperature measurement points and the connecting rod tilt adjustment mechanism when a highly rigid connecting rod is used.

A feed mechanism driven by a step motor is installed on the connecting rod, three thermocouples T_R , T_M and T_F are installed in the width direction of the bearing. T_M measures the temperature of the rear face in the middle of the bearing, and T_R and T_F measure the temperatures of the rear faces of the ends of the bearing.

When misalignment occurs, a greater amount of heat is generated on the side where clearance between the shaft and the bearing is smaller, hence the temperature rises. Accordingly, the temperatures of T_R and T_F at the ends of the bearing were measured, and both temperatures were maintained at a constant level, enabling the test shaft and bearing to make parallel contact with each other.

Figure 8 shows the conditions for the test performed using this adjustment mechanism, and Fig. 9 shows the differences in temperature when the misalignment adjustment mechanism was installed and when it was not installed.

When there was no misalignment adjustment mechanism, the difference between T_R and T_F increased, and in the zone where the temperature difference widens and the effect of misalignment becomes severe, the temperature waveform became irregular. It can be seen that when there is a misalignment adjustment mechanism, T_R and T_F are maintained at constant levels. It was found to maintain constant contact between the shaft and bearing, it is necessary to adjust the feed amount together with the load.

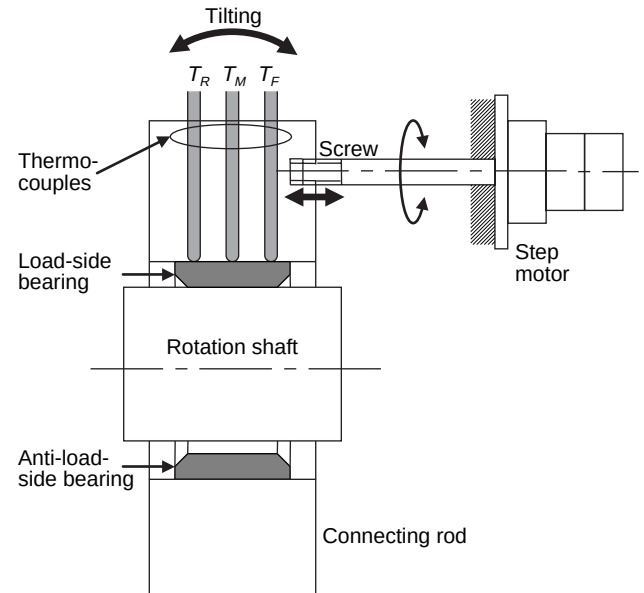


Fig. 7 Temperature measuring points and tilt control system of bearing

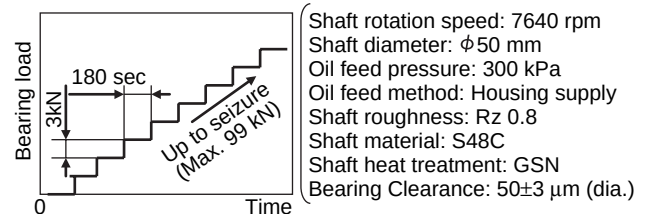


Fig. 8 Test pattern and example conditions in static load testing

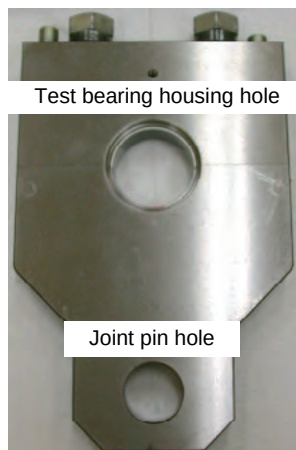


Fig. 6 Rigid-type connecting rod

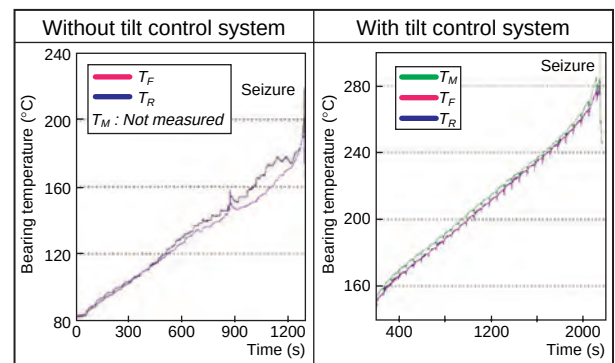


Fig. 9 Comparison of bearing temperature by tilt control system

3. Results of Tests using the New Bearing Tester

3.1. Results of Seizure Test

Figure 10 shows the results of changing the bearing material, and Fig. 11 shows the construction and surface hardness of the test bearing.

From the temperature results, it was seen that in the case of an aluminum alloy bearing, the temperatures of T_R and T_F at the ends of the bearing tended to be higher than the temperature of T_M at the middle of the bearing. It is thought that this is due to the fact that the sliding conditions at the ends were more severe than that at the middle of the bearing. With a lead alloy bearing, on the other hand, the temperature differences between T_R , T_M and T_F were not as large as in the case of an aluminum alloy bearing. Also, in a comparison of an aluminum alloy bearing and a lead alloy bearing, it was found that the lead alloy bearing seized more quickly.

To explain this fact, the temperature difference D_T was defined according to the following equation.

$$D_T = T_M - \frac{T_F + T_R}{2} \quad (1)$$

D_T : Temperature difference

T_M : Bearing middle temperature

T_F : Bearing front temperature

T_R : Bearing rear temperature

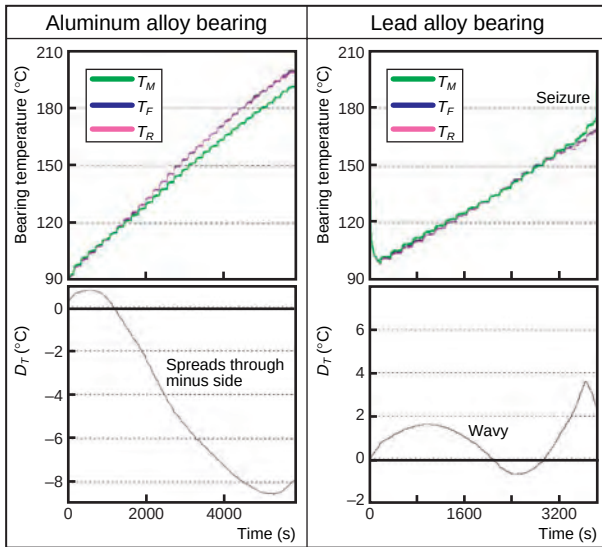


Fig. 10 Typical temperature pattern of aluminum alloy and lead alloy bearings

Bearing type	Aluminum alloy bearing	Lead alloy bearing
Construction	Aluminum alloy 200 μm Bonding layer 50 μm Back steel Reminder	Over-layer 15 μm Lining 200 μm Back steel Reminder
Surface hardness	50 (HV5)	13-17 (HVM10)

Fig. 11 Seizure test sample construction and hardness

The D_T of an aluminum alloy bearing was initially a positive value, and subsequently it tended to shift markedly to the negative. In contrast, the D_T of a lead alloy bearing did not increase significantly and tended to be wavy.

Figure 12 shows the form of wear of the bearing resulting from the test. The bearing wear width B_W after the test is defined as shown in Eq. (2).

$$B_W = B_A - B_B \quad (2)$$

B_W : Bearing wear width

B_A : Original effective width prior to test

B_B : No wear width after test

Figure 13 shows the relationship between the bearing wear width B_W and the bearing load.

The results demonstrated that the wear of a lead alloy bearing is greater than an aluminum alloy bearing when both are subjected to the same load. It can be said that the sliding face formed on the lead alloy bearing is caused mainly by wear, while that sliding face formed on the aluminum alloy bearing is caused mainly by elastic deformation. The sliding face is formed by the break-in

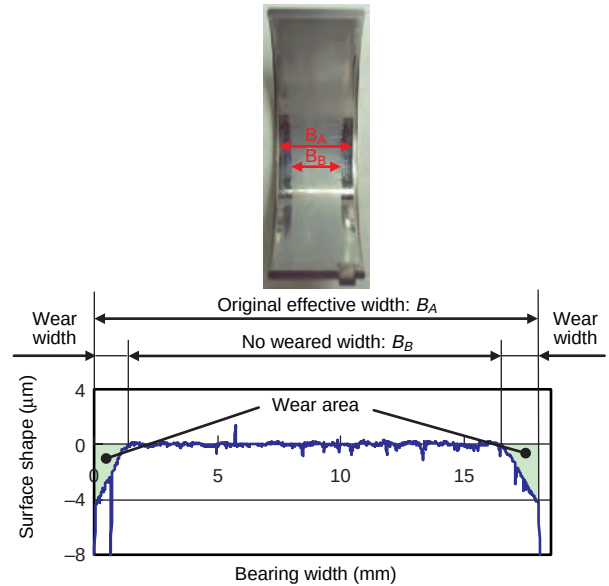


Fig. 12 Bearing shape features after testing

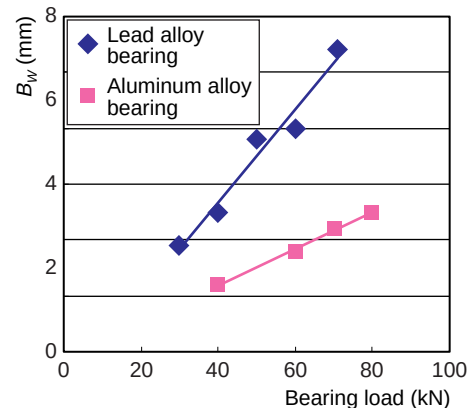


Fig. 13 Wear width of bearing by testing load

action. It is thought that the break-in form of a lead alloy bearing is due to wear, while an aluminum alloy bearing is largely due to elastic deformation. The break-in form was shown to differ greatly depending on the bearing material. It was also demonstrated that by measuring the temperature of the bearing it is possible to study one part of the break-in process of a bearing.

3.2. Friction Coefficient Test Results

Because the new tester measures temperature to provide the ideal conditions for maintaining the shaft and bearing in parallel, a test was performed to obtain the coefficient of friction of the bearing.

Figure 14 shows the test conditions. The test method used involved applying a 25 Hz amplitude load to gradually reduce the rotational speed, thereby making the sliding conditions more severe. Figure 15 shows the results obtained by repeating the test conditions described above.

As the number of rotational measurements increases, the friction decreases and stabilizes. It was found that if this process is repeated four or five times, the friction stabilizes, and a break-in condition is obtained. Figure 16 shows a graph of continuous data converted into a non-dimensional coefficient of friction, which was obtained by measuring the friction when different bearings (A, B and C) from the same lot were installed in the tester. To calculate the friction graph, the load was set to the maximum value of the

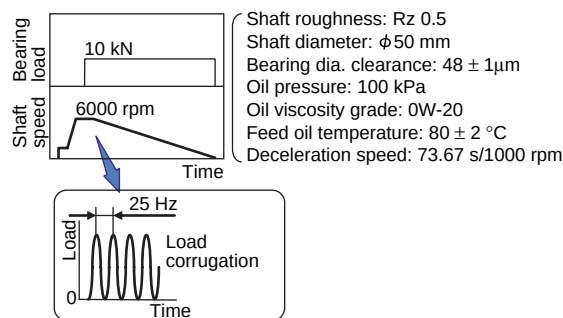


Fig. 14 Coefficient of friction test conditions

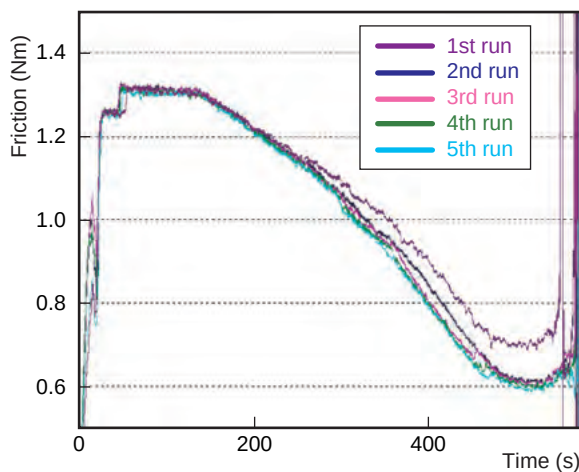


Fig. 15 Results of repeated friction test (using same bearing)

amplitude load, and the oil viscosity was measured using the mean value of the bearing rear face temperature for each sampling time. It was confirmed that even when the bearing installed in the tester was changed over, the same results were obtained, and by measuring the temperature and maintaining the shaft and bearing parallel to each other, it was possible to evaluate the coefficient of friction with high reproducibility.

3.3. Results of Test with the Bearings fitted on an Actual Connecting Rod

The new tester can perform basic tests using a rigid connecting rod as well as under sliding conditions that approximate those in an actual engine. The following is a description of the results of tests performed using actual engine parts.

In the engine used for this test, the connecting rod small-end bush was eliminated to reduce the weight of the reciprocating parts. When the bush was eliminated, the piston pin was surface-treated to provide better lubrication against the piston pin. In selecting this surface treatment, a test was carried out using the new bearing tester. Figure 17 shows the installed condition of the test connecting rod on the tester, and Fig. 18 shows the load condition. The piston pin was rotated at a constant speed of 2500 rpm by a motor. The load was applied from the big end of the connecting rod from 0 kN to a maximum compressive load of 40 kN at a frequency of 100 Hz, and then excited to the minimum tensile load of 20 kN over a period of 20 seconds. After 100 total cycles, the test was deemed complete. Oil was dripped into the hole at the small end of the connecting rod by a step-motor-driven oil pump.

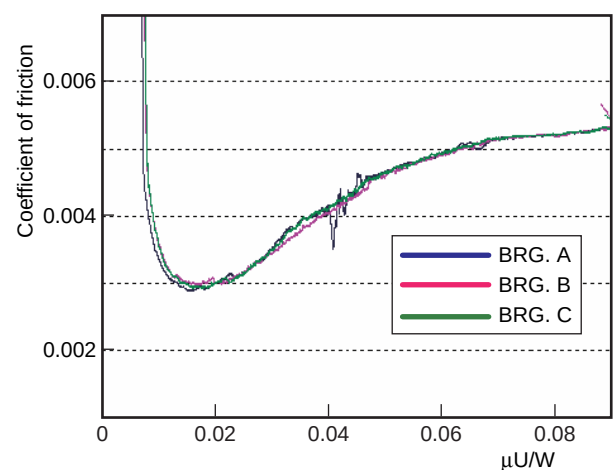


Fig. 16 Friction coefficient curve

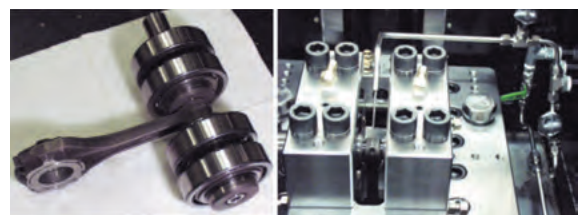


Fig. 17 Connecting rod (left); rod connected to test equipment (right)

Figure 19 shows a pattern diagram of the friction waveform obtained. The peak value of the stabilized friction is defined as F_m , and the peak value of the friction that increased midway is defined as F_p .

Figure 20 shows the types of surface treatment to which the piston pin and the hole in the small end of the connecting rod were subjected and photographs of the piston pin after the test. The surface treatments used were selected on the basis of high film strength, resistance to separation, and/or good lubricity. Figure 21 shows the measured friction results.

The condition of the lubrication was judged based on the following points:

- (1) Small amount of steady-state friction.
- (2) Small increase in friction due to scuffing.

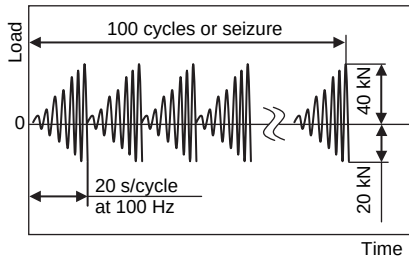


Fig. 18 Testing load conditions

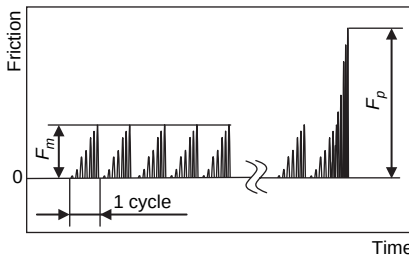


Fig. 19 Friction feature and definition of F_m and F_p

Regarding the results using different specifications, A was considered to be strong because of the nature of the film. However, it had large F_m and F_p values and exhibited low lubricity. B showed stable initial friction, but had a relatively large F_m value, and the friction was no longer stable after scuffing occurred. C showed little friction under steady-state conditions, and exhibited superior scuffing performance. In the case of D and E, scuffing occurred intermittently. F showed the smallest F_m value of all the lubricants tested, and exhibited good lubricity. The piston pin and connecting rod that had been surface-treated with F were tested in an actual engine. They met the standards of the actual engine test, so F was selected for use in commercial automobiles. This method was effective when applied to determine the bearing specifications for the engine.

Type	Surface treatment		Photograph after testing
	Piston pin	Piston pin boss	
A	Titanium nitride	Carburizing	Could not be pulled off
B	Prototype ferrous oxide coating (Fe_3O_4)	Carburizing	
C	Sulfurizing	Carburizing	
D	Prototype ferrous oxide coating (Fe_3O_4)	Sulfurizing after carburizing	
E	Prototype ferrous oxide coating (Fe_3O_4)	Shot blasting after carburizing	
F	Ferrous oxide coating (Fe_3O_4)	Carburizing	

Fig. 20 Surface treatment and surface status after testing

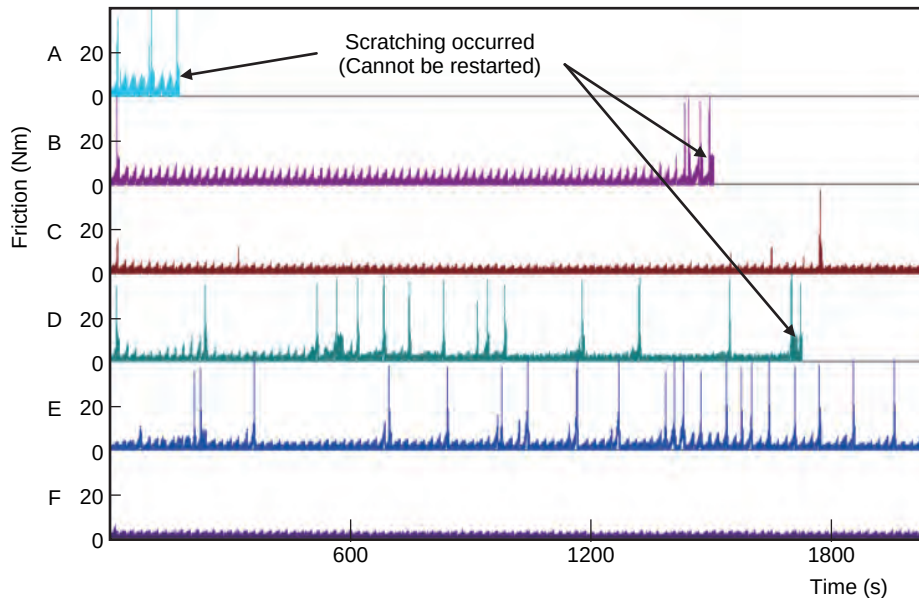


Fig. 21 Difference in friction by surface treatment

4. Conclusion

A unique dynamic load tester was devised that is capable of reproducing the sliding condition of an actual engine bearing and increasing testing efficiency by means of rig acceleration test, both issues for conventional static load bearing testers. The following results were obtained.

- (1) As a result of performing a comparative seizure test on lead and aluminum alloy bearings, it was found that the form of break-in of lead alloy bearings was due to wear, whereas the form of break-in of aluminum alloy bearings was largely due to elastic deformation.
- (2) It is possible to study the bearing break-in process by measuring the bearing temperature during break-in.
- (3) By measuring the temperature and maintaining the shaft and the bearing parallel alignment to each other, it was possible to evaluate the coefficient of friction with high reproducibility.
- (4) This method reproduced sliding conditions that approximate those of an actual engine, indicating that it is an effective method to select bearing specifications.

References

- (1) Ono, A., Takayanagi, S., Ishikawa, H., Tanaka, T., Nakahara, T.: Development of Lead Free Bearing Material with Multi Layer for Heavy Loaded Engines, Aachener Kolloquium Fahrzeug-und Motorentechnik 2005, p. 1079-1100 (2005)
- (2) Kanayama, H., Kawakami, S., Gouhara, C., Fuwa, Y., Michioka, H.: Development of new Engine Bearings with Overlay Consisting of Solid Lubricants, 1997 JSAE Annual Congress Proceedings, No. 976, p. 141-144 (1997)
- (3) Tomikawa, T., Fuwa, Y., Shimura, Y.: Advance of Lead Free of Sliding Bearing for Gasoline Engines, Transactions of Society of Automotive Engineers of Japan, Vol. 35, No. 4, p. 123-126 (2004)
- (4) Oh, K. P., Goenka, P. K.: The Elastohydrodynamic Solution of Journal Bearings Under Dynamic Loading, Transactions of ASME, Vol. 107, p. 389-395 (1985)
- (5) Oh, K. P.: The Numerical Solution of Dynamically Loaded Elastohydrodynamic Contact as a Nonlinear Complementarity Problem, Transactions of ASME, Vol. 106, p. 88-95 (1984)
- (6) Fantino, B., Frene, J., Parquet, du J.: Viscosity Effects on the Dynamic Characteristics of an Elastic Engine Bearing, SAE Paper, 852074, (1985)
- (7) Alderson, A. L., Webster, W. D. Jr.: The Effects of Crankshaft Lobing on Connecting Rod Bearings, SAE Paper, 881286 (1988)
- (8) Mitsui, J., Yamada, F.: Jyanaru Suberi Jikuu no Yumaku Tokusei ni Kansuru Kenkyuu, Transactions of The Japan Society of Mechanical Engineers, Vol. 45, No. 390, p. 238-247 (1979) (in Japanese)
- (9) Okamoto, Y., Kitahara, K., Ushijima, K., Aoyama, S., Jones, G. J., Xu, H.: A Study for Wear and Fatigue on Engine Bearings by Using EHL Analysis, 1999 JSAE Annual Congress Proceedings, No. 46-99, p. 9-12 (1999)
- (10) Sato, K., Makino, K., Machida, K.: A study of Oil Film Pressure Distribution on Connecting Rods Big Ends, SAE Paper, 2002-01-0296 (2002)
- (11) Machida, K., Makino, K., Satoh, K.: Analysis of Engine Plain Bearings Seizure by the Bearing Tester, HONDA R&D Technical Review, Vol. 12, No. 1, p. 39-44
- (12) Natsumeda, S., Someya, T.: Negative Pressures in Statically and Dynamically Loaded Journal Bearings, 13th Leeds-Lyon Symposium on Tribology, Paper III(ii), p. 65-72 (1986)
- (13) Middleton, V., Dudley, B. R., McCallion, H.: An Investigation into the Performance of Dynamically Loaded Journal Bearings: EXPERIMENT, Proc. Inst. Mech. Engrs. 1966-67, Vol. 181 Pt 3B, p. 9-27 (1966)
- (14) Mori, Y., Miyata, S., Abe, Y., Fujita, E.: Jyanaru Jikuu ni Okeru Yumaku Fu-renzoku no Kenkyuu, Transactions of The Japan Society of Mechanical Engineers, p.658-666 (1966) (in Japanese)
- (15) Mitsui, J.: Jyanaru Suberi Jikuu no Yumaku Tokusei ni Kansuru Kenkyuu, Transactions of The Japan Society of Mechanical Engineers, Vol. 48, No. 428, p. 565-576 (1982) (in Japanese)
- (16) Someya, T.: Introduction to the Theory of Bearings in Internal Combustion Engines, Journal of Society of Automotive Engineers of Japan, Vol. 27, No. 7, p. 806-815 (1973)
- (17) Norose, S., Lin, J., Sasada, T.: Jyanaru Jikuu no Yaketsuki Kikou, Kikai no Kenkyuu, Vol. 39, No. 4, p. 485-488 (1987)
- (18) Desaki, T., Goto, Y., Kamiya, S.: Development of the Aluminum Alloy Bearing with Higher Wear Resistance, 1999 JSAE Annual Congress Proceedings, No. 46-99, p. 17-20 (1999)

- (19) Masuda, T., Ushijima, K., Hamai, K.: A Measurement of Oil Film Pressure Distribution in Connecting Rod Bearing With Test Rig, Tribology Transactions, Vol. 35, p. 71-76 (1992)
- (20) Yamamoto, K., Matsuhisa, H., Wada, M., Sakamoto, M., Tanaka, T.: Development of Three Layers Copper-Lead Bearings for Higher Speed Automotive Engines, SAE, 910161 (1991)
- (21) Hashizume, K., Kumada, Y., Kimura, Y.: Performance of Microgrooved Bearings in Automotive Engines, 1995 JSAE Annual Congress Proceedings, No. 955, p. 117-120 (1995)
- (22) Fukuoka, T.: Performance Test Method and Testing Machine of Journal Bearings, Journal of Japan Society of Lubrication Engineers, Vol. 29, No. 8, p. 573-577 (1984)
- (23) Esaki, J.: Dynamic Characteristics and the Test Apparatus of Oil Film in Journal Bearings, Journal of Japan Society of Lubrication Engineers, Vol. 29, No. 8, p. 579-584 (1984)
- (24) Fujisawa, Y., Tsuji, M., Machida, K., Narishige, T.: Studies on Increase Plane Bearing Performance, 1993 JSAE Annual Congress Proceedings, No. 931, p. 89-92 (1993)

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CFD Air Flow Analysis for Air-cooled Motorcycle Engines

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ABSTRACT

In order to develop an air-cooled engine, cooling air flow was analyzed by CFD (Computational Fluid Dynamics). Cooling performance of engine is necessary for precise temperature control. However, stable radiation is difficult for motorcycles because of the driving wind. Cooling fins are thin and complicated; therefore plenty of work time is necessary to create a boundary fitted mesh. In addition, since low Reynolds (Re) number area coexists with high Re number area in a calculation domain, applications of the wall function are limited. Therefore, we employed the Partial Cells in Cartesian coordinate method, which expresses complex shapes according to turbulence boundary conditions in the Cartesian coordinate. Analysis results showed good agreement with wind tunnel test results. Furthermore, useful knowledge in engine developments was obtained by examinations of local heat transfer coefficients calculated by Karman's analogy.

1. Introduction

Recently, even as motorcycle user requirements diversify, the demand for motorcycles using simple air-cooled engines remains strong.

The major issue in terms of cooling performance in these air-cooled engines is the fact that the heat radiation must depend on the airflow that passes over the engine when the vehicle is in motion. Ensuring sufficient overall heat radiation and achieving an appropriate balance between the cylinders in multi-cylinder engines are important factors in cooling performance. The intensive pursuit of cooling performance is simultaneously a pursuit of functional beauty. Previously, when developing air-cooled engines, it has been difficult to quantitatively analyze airflow behavior at the initial stages of the engine design process, and designers have relied on making the airflow visible using wind tunnels and taking temperature measurements. The importance of reducing emissions and enhancing fuel economy has recently increased in the field of motorcycles too, and precise engine temperature control has become an important issue. In addition, an efficient and adaptable development process that can respond to the requirement for large item small volume production is desirable from a business perspective.

The shapes of the cooling fins and other components of air-cooled engines are fine and highly complex, and a considerable

amount of operation time are required to generate normal boundary-fitted meshes such as tetra mesh. For this reason, there is no choice but to simplify detail forms. In addition, significant differences in velocity exist within the same calculation domain; for example, the front of the engine, which is directly struck by the air, is a high Reynolds (Re) number area, and the rear of the engine, where there is virtually no airflow, is a low Re number area. This necessitates a turbulence model which is applicable from laminar flow areas to high- Re areas. Therefore, the Partial Cells in Cartesian coordinate method^{(1),(2)} was used for CFD analysis method. The Partial Cells in Cartesian coordinate method is a computational fluid dynamics method using Cartesian coordinates that involves almost no expenditure of operation time on mesh generation for flow fields around complex forms. This method is provided with a turbulence model applicable to flows with a wide range of Reynolds numbers.

This research was intended to analyze the cooling air behavior around a motorcycle under actual driving conditions. Analyses were conducted using a motorcycle equipped with an in-line 4-cylinder 750 cm³ engine, set in a wind tunnel. The behavior of the airflow between the cooling fins was analyzed; in addition the distribution of heat transfer coefficients on the surface of the engine was calculated to quantitatively determine cooling performance. Furthermore, an optimization of cooling performance was examined based on obtained findings of the analyses.

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2. Air-cooled Engine Design and CFD Analysis Items

Figure 1 shows the relationship between air-cooled engine design and the subjects of CFD analysis.

To enable effective use of cooling air, visualization of the airflow, the shape of the cooling fins and the air channels, and the position of the oil cooler are included as CFD analysis items. Visualization of air flow in a wind tunnel shows only the behavior of the air. By contrast, this CFD analysis aims at the examination under actual driving conditions and the calculation of the heat transfer coefficient of each part of the engine for quantitative study of cooling performance. In addition, possibilities to connect to the calculation of the temperature distribution of the whole engine and the thermal stress analyses were considered.

When designing a new engine, CFD analysis will include the shape of the oil cooling jacket, layout of the oil pathway, optimization of amount of cooling oil, and the shape of the exhaust ports for minimizing heat source.

3. Calculation Method

3.1. Partial Cells in Cartesian Coordinate Method

In terms of design man-hours, the major issue in CFD analysis is the mesh generation, which in particular represents an issue when the analysis subject is a motorcycle. The Partial Cells in Cartesian coordinate method is one of Cartesian coordinate method that was developed to significantly reduce the operating time required for mesh generation when analyzing flow fields around complex forms. Figure 2 compares the concept of input data between the Partial Cells in Cartesian coordinate method⁽¹⁾ and the boundary fitted coordinate (BFC) method. Because the mesh in the BFC method is required to correspond to the shape data, a calculation mesh must be generated every time when the shape is changed. By contrast, the shape data and the mesh data are mutually independent in the Partial Cells in Cartesian coordinate method, and there is therefore no requirement to regenerate and modify meshes when the shape is changed.

When the mesh lines and the surface shape of analysis subject do not match, fluid cells lacking a part of hexahedron are produced in vicinity of the wall. These are called partial cells. By this definition, partial cells can exist in an enormous number of shapes, as shown in Fig. 3 (a). Discretization equations based on the finite volume method (FVM) were derived to allow calculating according to the shape in partial cells of any shape. In addition, because there are no fitted meshes to the boundary, an original turbulence model in vicinity to the wall⁽²⁾ that considered the turbulent flow behavior in low Reynolds number areas close to the wall has been used to calculate the turbulent flow, while the k - ϵ model and the wall function are fundamental. Cells divided by the wall, as shown in Fig. 3 (b), are not considered. The reason is based on the fact that the state inside the cell is expressed by one value. When this situation is predicted, the nearby meshes are more finely regenerated.

3.2. Calculation Procedure

Figure 4 shows the calculation procedure. The shape data is formulated with numerical wind tunnels by 3D-CAD software (for example CATIA) and written to a file as STL data. The STL data format is commonly used with optical molding devices; it presents shapes as forms resembling triangular aggregates, and is supported by almost all CAD systems. However, because the format was originally used in CAM fields, it lacks precision in the field of numerical simulations, and the content of the data will differ in

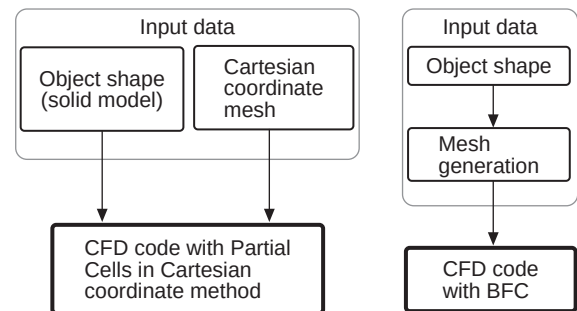


Fig. 2 Comparison of data flow between Partial Cells in Cartesian coordinate method and BFC

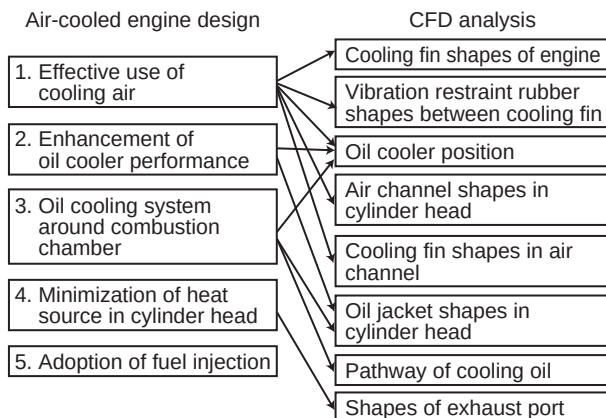


Fig. 1 Relation between air-cooled engine design and CFD analysis

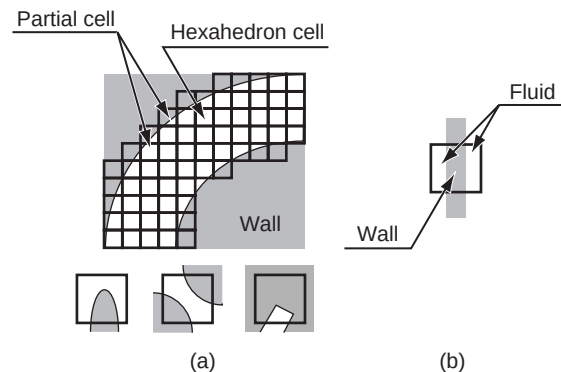


Fig. 3 Examples of partial cells (a), and partial cells divided by wall (b)

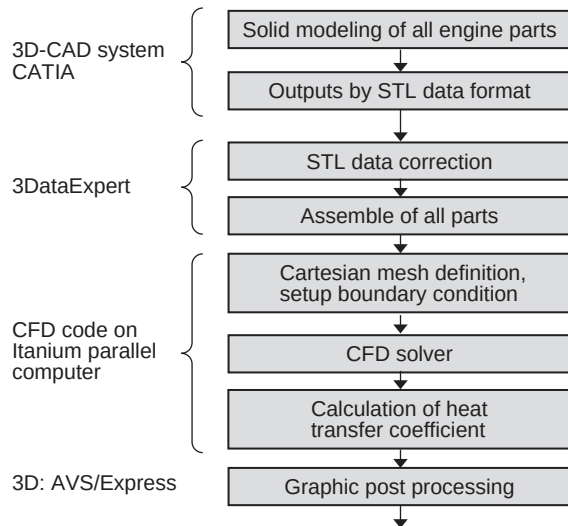


Fig. 4 Procedure of air cooling engine analysis by CFD code

each CAD system. For example, there may be gaps between certain triangles and contiguous triangles, and data correction is therefore required. In this project, a commercially available software program⁽³⁾ was used to correct the problems and also to assemble components. Following parts assembly, the Cartesian mesh was defined, the inlet boundary conditions were established, and calculations were started. When the calculations were completed, the results were used to calculate heat transfer coefficients, and the results of these calculations were represented graphically.

3.3. Formulation of Shape Data

Figure 5 (a) shows the solid model of the engine used in this analysis. Because it is not necessary to simplify forms to generate the mesh when using the Partial Cells in Cartesian coordinate method, the fillets and other shapes around the cooling fins can be created as correct as possible. In addition to engine shape data, frame shape data were created as required to enable analysis of behavior under actual driving conditions. The parts positioned upstream from the engine (the front tire, front forks, headlight, fuel tank, oil cooler, etc.) have a strong effect on airflow around the engine. Because the bottom of the tank has a particularly significant effect on the airflow in the space between the tank and the head cover, small parts and the wire harness were also created in as much detail as possible. The fins of the oil cooler core were simplified to reduce the amount of generated data, and were created with a distance of 1.5 mm separating the fins and a fin thickness of 0.25 mm. Downstream parts such as the rear tire and seat were not absolutely necessary for the purposes of this project, but were created to enable evaluation of aerodynamic performance. Because the Partial Cells in Cartesian coordinate method involves almost no expenditure of time on mesh generation, the formulation of this shape data represented the major commitment of operation time in the analysis.

Figure 5 (b) shows the completely assembled STL data. Fine facets are used for areas with complex shapes, such as the area around the cooling fins, while large, long and/or narrow facets are used for parts like the frame pipes. This has limited the number of facets to 1.8 million, enabling the data to be processed by a computer with 2 GB of memory.

3.4. Calculation Domain and Boundary Conditions

Figure 5 (c) shows the wind tunnel that forms the calculation domain and the boundary conditions. For analyses of aerodynamic performance, it is necessary to use a large wind tunnel in order to eliminate the effect of the side walls. However, because the intent of the present study was only to analyze the airflow around the engine, a small wind tunnel (5.0 m × 2.0 m × 2.0 m) was used to reduce calculation time.

Boundary conditions were set as an airflow velocity corresponding to vehicle speed. A 5% turbulence intensity was used and the length scale was set at 0.05 m, at the entrance of the wind tunnel, and fixed walls were used for the floor and side walls. Calculations were conducted for a maximum vehicle speed of 210 km/h and 60 km/h to correspond to wind tunnel test conditions.

3.5. Calculation Grid

From considerations of installed memory and calculation time, the number of grids in the entire calculation domain was set at 133 × 128 × 139, to total approximately 2.37 million grids. Because a calculation grid size of approximately 2.3 mm was used close to the fins, while a fin pitch of approximately 10 mm and a fin thickness of 1–4 mm were set, two or three grids were used between fins. The thickness of the tubes in the oil cooler was approximately 2 mm with an approximate 10 mm distance between them. Two to three grids were used between the tubes, as in the case of the cooling fins. The type of divided cells shown in Fig. 3 (b) occurred in the core section, but this was allowable because the quantities of the state of multiple flow domains in the same cell can be regarded as identical. Figure 5 (d) shows the grid for the area around the oil cooler and engine fins. A maximum distance between grids was set 40 mm close at the front tire, 85 mm close at the rear tire, and 250 mm close at the entrance and exit of the wind tunnel.

The calculation time was approximately 7 hours using a 16 CPU parallel computer. While the calculation time for the method used here was almost the same as that for tetra mesh CFD software when the total number of meshes was the same, given the fact that complex forms could be calculated with a comparatively lower number of meshes, the actual calculation speed was several times faster. This method makes it possible to complete calculations within one night, and has therefore become an effective design support tool.

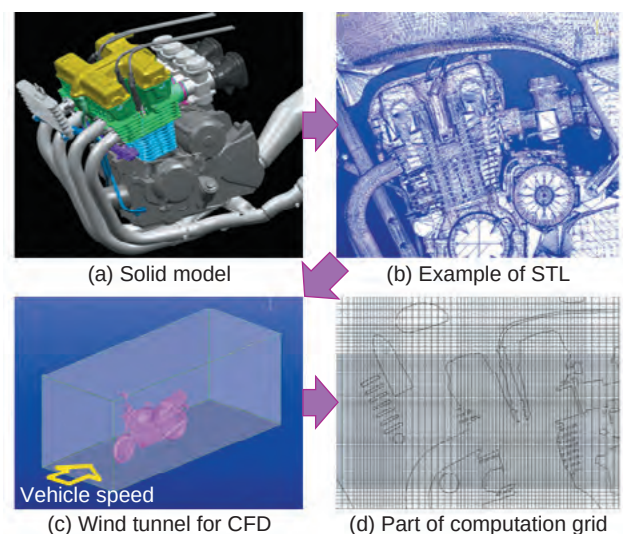


Fig. 5 Calculation flow chart

3.6. Calculation of Heat Transfer Coefficients Using Karman's Analogy

The heat transfer coefficient, α , is defined by the following Newtonian equation:

$$q = \alpha (T_w - T_f) \quad (1)$$

Here, q is the amount of heat transfer per unit area, T_w is the temperature of the wall, and T_f is the temperature of the fluid. The latter parameter is established at a sufficient distance from the wall.

The heat transfer coefficient is a function of an enormous kind of parameters, including all the effects on heat transfer, such as the shape of the solid walls, the type of fluid, and the state and velocity of the flow. It is therefore challenging to determine heat transfer coefficients for complex forms experimentally, and the use of CFD analysis is considered to have particular value in this area. In addition, turbulence has a significant effect on the state of the flow.

Normally, heat transfer coefficients are determined by solving energy equations for the flow field. This makes boundary conditions for heat at the walls necessary. If changes in the temperature of the walls are also considered, thermal conductivity calculations are required for the solids in the simulation model. Because the heat transfer coefficients are determined by the fluid properties and flow field state, when changes in the temperature of the fluid cause changes in these fluid properties and/or flow field state, coupling calculation between fluid and solid able to be solved for the mutual dependence of flow and temperature become necessary, and this requires enormous calculation resources. However, in most cases the fluid temperature changes that result from the cooling airflows in air-cooled engines can be ignored. To increase calculation efficiency, Eq. (2), which uses Karman's analogy⁽⁴⁾ (derived from the relationship between fluid friction and heat transport), is used to obtain the heat transfer coefficient.

$$\frac{1}{\alpha} = \frac{1}{\rho C_p} \sqrt{\frac{\rho}{\tau_w}} \left[\frac{U_s - U_2}{\sqrt{\tau_w/\rho}} + 5P_r + 5\ln(5P_r + 1) \right] \quad (2)$$

where α is the heat transfer coefficient, ρ is density, C_p is isopiestic specific heat, τ_w is the wall shear stress, P_r is the Prandtl number, U_s is the mean velocity at the outer edge of the boundary layer, and U_2 is the mean velocity in the boundary layer between the logarithmic and transition areas.

4. Calculation Results and Discussions

4.1. Rendering Flow Visible

Figure 6 shows the pressure distribution on the vehicle surface and flow lines around the vehicle, providing an overall image of the calculation results. No partial cells were used to display the stream lines because a commercially available software program was used; post-processing using a normal Cartesian coordinate mesh was used. Because of the extremely rough nature of mesh that was used, other than in the area around the oil cooler and the engine, accuracy in calculating flow inclusions to the tires and the back wake was not a consideration. The space between the front fender and tire was blocked in to prevent the generation of divided cells of the type shown in Fig. 3 (b) in the calculations using rough meshes.

Figure 7 (a) shows the velocity magnitude in a cross-section of the center of the vehicle. Most of the airflow over the front fender

strikes the engine block, passes under the oil pan and flows to the rear. Figure 7 (b) shows the flow velocity vector at an offset of 57 mm to the left of the center of the vehicle. While airflow towards the oil cooler is visible, the oil cooler is mounted in relation to the frame, positioning it diagonally to the flow, and there is therefore room for enhancement from the perspective of cooling performance. The airflow passed through the oil cooler is broad in three dimensions, but some flows go over the handles and the tank to the upward. There is little flow over the head cover, and as per predictions no strong flow can be observed at the rear of the engine. Figure 7 (c) shows the turbulence intensity at an offset of 57 mm to the left of the center of the vehicle. As this figure indicates, turbulence is very strong behind the front tire. Because the tire and the floor are both fixed in this calculation, turbulence can be predicted to be even stronger when the vehicle is actually running.

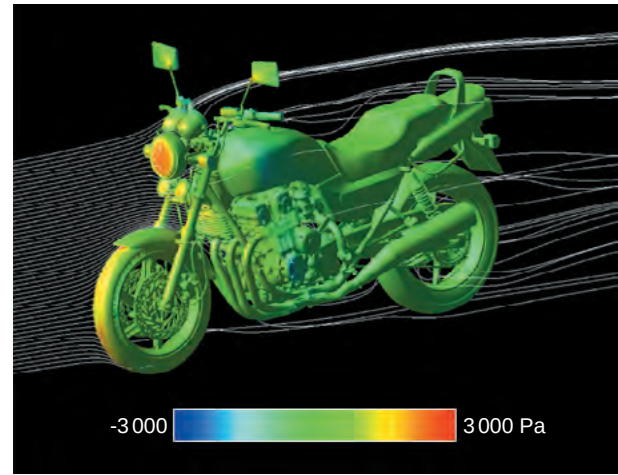


Fig. 6 Stream line around motorcycle

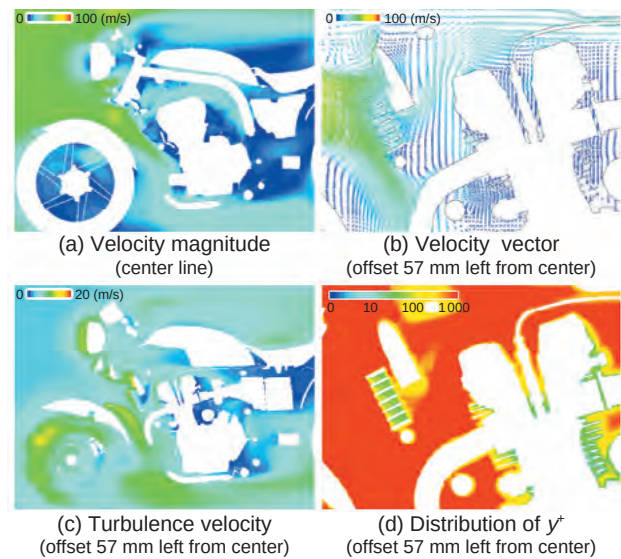


Fig. 7 Section-cut post processing

Figure 7 (d) shows the distribution of the dimensionless distance from wall y^+ , as determined using Eq. (3), to verify the accuracy of the calculating of the turbulent flow close to the walls.

$$y^+ = \frac{\rho l C_\mu^{1/4} k^{1/2}}{\mu} \quad (3)$$

where ρ is density, l is distance from the wall, C_μ is the turbulence model constant (0.09), k is kinetic turbulence energy and μ is the viscosity.

Figure 8 shows the results from the previous figure represented in three dimensions. The two-dimensional representation in Fig. 7 shows the calculation results for the main spaces, but the three-dimensional representation in Fig. 8 enables the calculation results for the areas close to the wall to be read. Figure 8 (a) shows the pressure distribution on the surface of the vehicle. These pressure results are used to determine the hexagonal component force when evaluating aerodynamic performance. Figure 8 (b) shows the flow velocity near the wall, and Fig. 8 (c) shows the turbulence intensity near the wall. Strictly speaking, because the flow velocity on the physical wall is zero, this represents the flow velocity and the turbulence intensity in the partial cells contiguous to the wall. Because the size in each partial cell is different, the distance in millimeters from the wall of the represented values differs according to each partial cell size. For this reason, caution must be exercised

when treating these results as flow analysis results. In addition, cooling air analysis must provide a quantitative estimate of the ultimate magnitude of heat radiation and the temperature that will be reached by each part of the engine. Equation (2) is therefore used to determine heat transfer coefficients for the engine surface from flow velocity and turbulence energy. The results are shown in Fig. 8 (d). The red sections in the figure (the cylinder head, the side of the head cover on the exhaust side, the front of the crank case cover, etc.) represent areas at which the heat transfer coefficient is close to 360 W/m²K and the heat transfer coefficient is high, i.e., the areas displaying excellent heat radiation.

4.2. Airflow around the Cylinder Fins

Post-processing using the particle trace method was conducted for the area around the cylinder fins to visualize the airflow. The results are shown in Fig. 9. The particle trace method places particles in the flow and traces their locus. This method corresponds to using smoke in wind tunnel tests to make the streak visible, and results for the two methods correlate for steady-state flows. The flow velocity at the entrance of the wind tunnel was set at 60 km/h to correspond to a wind tunnel test. Angular fins were used in this engine, and the airflow striking the edge of the cylinder block fin was observed to deflect from the edge without having flown along the fin or to the rear of the engine. At middle of cylinder block, the

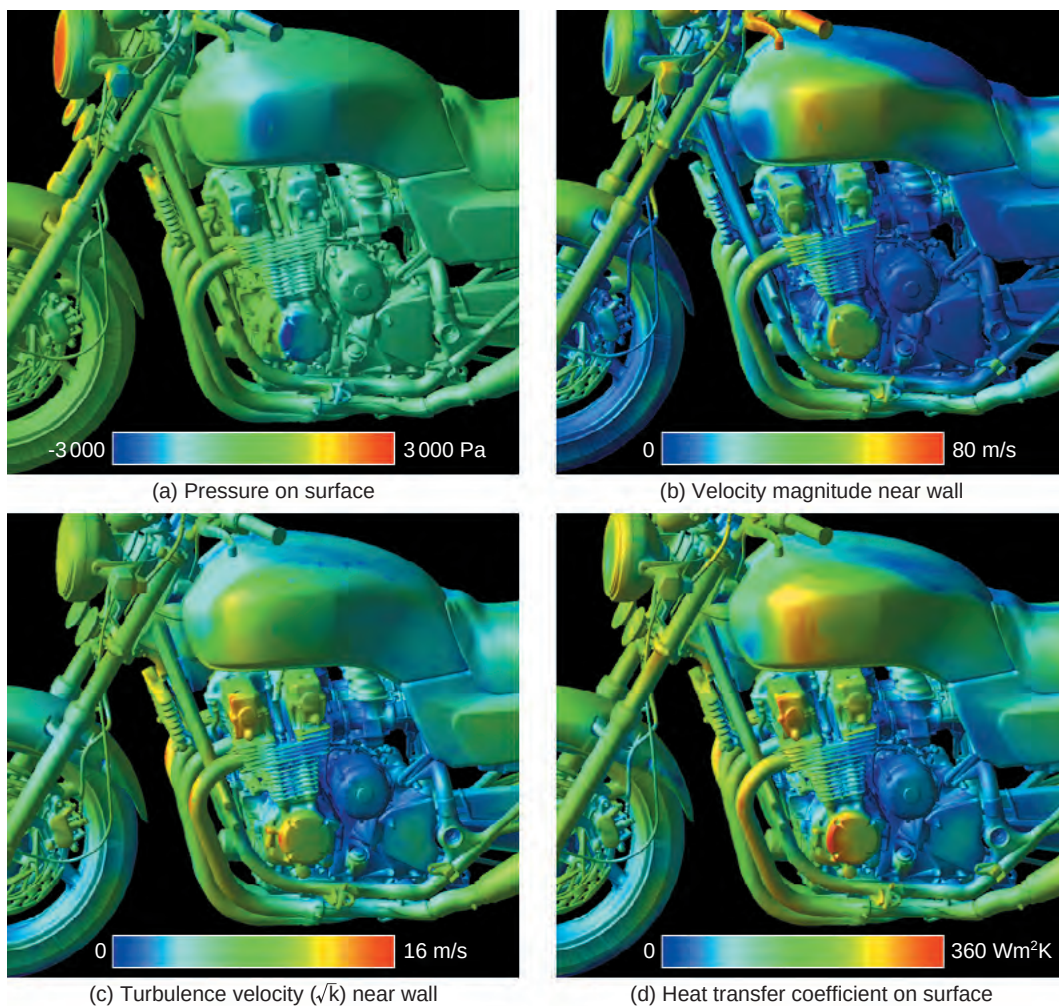


Fig. 8 Post treatment derived from CFD results

place is below ignition plug, Fig. 10 shows velocity vector between cylinder fins by close-up indication. The reverse direction of flow was observed in a backward domain from the cylinder center. The heat transfer coefficients shown in Fig. 8 (d) also indicate that heat radiation from the rear half of the cylinder fins is low. There is therefore a margin for enhancement from the perspective of cooling performance.

4.3. Comparison with Wind Tunnel Tests

Wind tunnel tests were conducted to verify the CFD results. Figure 11 shows the result of these tests. The installation conditions of the wind tunnel resulted in the tests being conducted at a flow velocity of 60 km/h. As Fig. 11 shows, the smoke used in the tests was deflected from the edges of the cylinder block fins and flowed away from the vehicle rather than flowing between the fins. This result corresponded with the results of visualization using CFD, and then the accuracy of CFD was confirmed by actual vehicle tests.

4.4. Study of Optimum Fin Shape

A variety of studies of the fin shapes were conducted. The results indicated that the change from angled fins to the rounded fins shown in Fig. 12 would enable the achievement of the flow characteristics close along the fins shown in Fig. 13. These studies therefore enabled the development of a fin shape displaying enhanced cooling performance.

4.5. Airflow above the Spark Plugs

Precise heat control in the middle cylinders is an issue for in-line 4-cylinder engines. In particular, the temperature of the spark plug seats, which are mounted above the combustion chamber of

the middle cylinder, reaches temperatures that are tens of degrees higher than those at the end cylinders. The airflow around the spark plugs was therefore analyzed. Figure 14 shows the visualization results of the airflow around the spark plugs using the particle trace method. The airflow over the head cover is drawn laterally from the side of the spark plugs by the wind when the vehicle is in operation, but the flow from the head cover to the area around the spark plugs is limited, especially, with virtually no airflow to the area around the spark plugs at the middle cylinders. As a result, and also indicated by the heat transfer coefficients shown in Fig. 8 (d), the heat transfer coefficient at the head surface around



Fig. 11 Flow visualization by wind tunnel with smoke

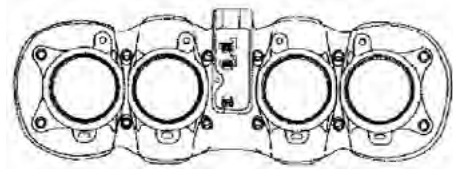


Fig. 12 Overhead view of cylinder block of prototype engine

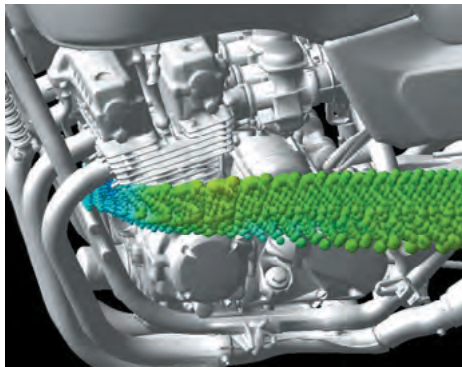


Fig. 9 CFD results by particle trace method

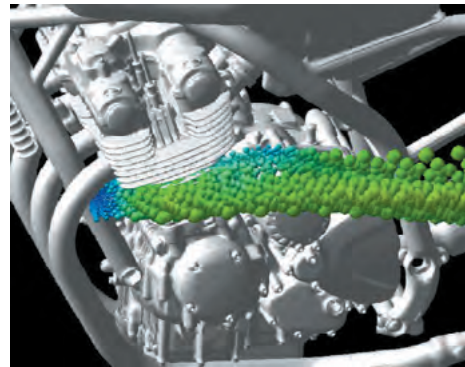


Fig. 13 Flow in vicinity of rounded-fins cylinder block

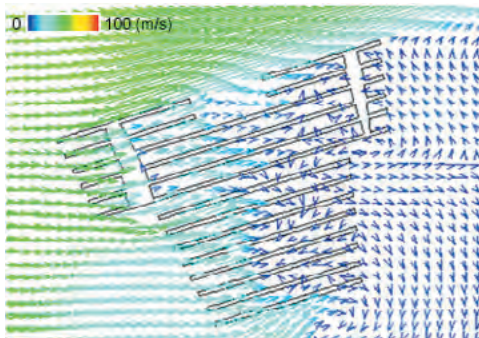


Fig. 10 Velocity vector around fins

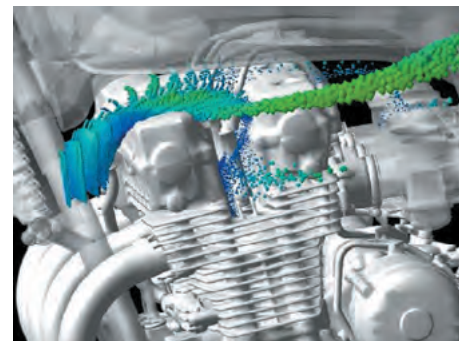


Fig. 14 Flow analysis around ignition plug

spark plugs of the middle cylinder is low. It is necessary to establish an air inlet enhancing the cooling performance of the area around the middle cylinder spark plugs, and this will require a design that is coordinated with the configuration of the interior of the engine.

4.6. Heat Transfer Coefficients in Each Section and Future Prospects

Figure 15 (a) shows the heat transfer coefficients over the entire motorcycle. Figure 15 (b) shows a view of the front of the engine, and Fig. 15 (c) shows a view of the front of the engine from below and diagonally. For greater clarity, the frame parts (tires, exhaust pipe, etc.) have been removed from Figs. 15 (b) and (c), and only the engine is shown. As the figures show, while there is good heat radiation from the sides of the engine, the heat transfer coefficients of the central section of the engine, and particularly the area around the cylinders, are unexpectedly low. By contrast, there are numerous areas displaying high heat transfer coefficients at the bottom of the engine (the oil pan, etc.), indicating the significance of the oil pan's effect on engine cooling. In addition, the heat transfer coefficient of the area around the oil filter that is attached to the bottom of the front of the engine is quite high, demonstrating that the filter also contributes to cool the engine.

Figure 15 (d) shows the frame parts and the engine viewed from the bottom, and gives a good idea of heat transfer in parts other than those on the engine. For example, the figure indicates that

cooling air does not necessarily flow uniformly over the entire core area of the oil cooler. Results for heat radiation from the exhaust pipe will be useful in predicting the exhaust gas temperature in the pipe when the vehicle is running. The figure also indicates that it is possible to study brake disk cooling using CFD.

These analyses of heat transfer coefficients can connect to thermal analysis of the entire engine. Conventionally, thermal analyses link fluids and solids and solve energy equations. However, in cases in which different coolants coexist, such as air-cooled engines that also use oil cooling, the amount of calculation required makes it difficult to link all the elements and solve the energy equations. The method using Karman's analogy discussed here, by contrast, is simple and can be used for analyses during the design process, making it an effective method of examining design proposals. In addition, the method also enables consideration of heat transfer on the inner walls of the intake and exhaust ports, enabling analysis under conditions that resemble those encountered in actual vehicle operation. Moreover, if heat transfer between the combustion gas and the cylinder heads is solved by coupling those in combustion simulations, present-day computer capabilities make it difficult to solve cooling air by additionally coupling it in the simulation. Using Karman's analogy for heat transfer between the cylinder head and the coolant can be considered an effective method of calculating the relationship between combustion and cooling when conducting combustion analyses.

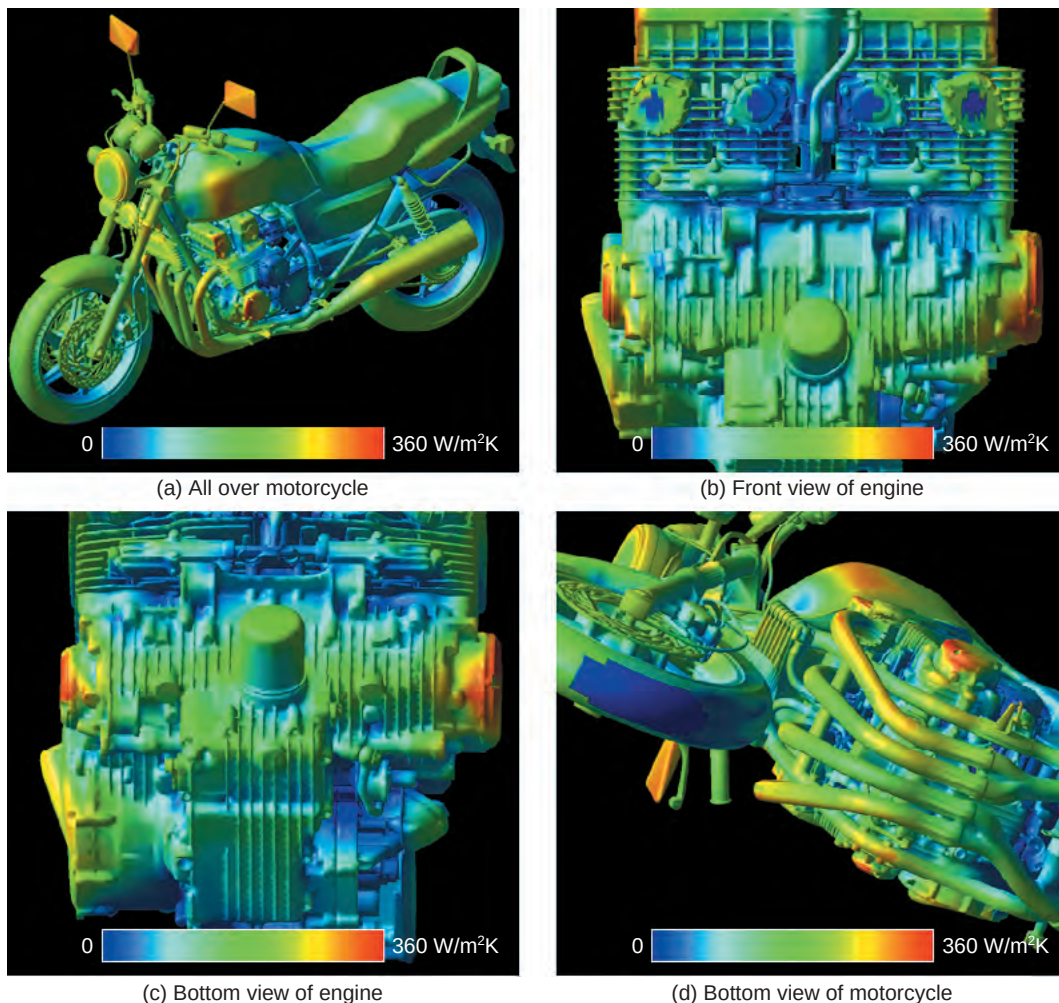


Fig. 15 Heat transfer coefficient at 210 km/h

4.7. Study of Cooling Performance of Auxiliary Equipment

The reason for the high heat transfer coefficients around the oil filter is the fact that the airflow between the front forks is bent downwards and deflected as airflows go over the front fender, and thus a comparatively strong flow strikes the engine, as indicated in the velocity magnitude distribution shown in Fig. 7 (a). Positioning a heat converter where the oil filter is currently located would be one means of making effective use of the airflow generated while the vehicle is running. A proposed engine design is shown in Fig. 16, and an example of analysis results used in studying this design is shown in Fig. 17. The installation angle of the heat converter affects the amount of the airflow through the core, but this effect can also be studied on the desktop by CFD. Due to limited space constraints, a high-capacity heat radiator cannot be mounted, but the method would be effective in preventing the oil cooler from blocking airflow to the area around the spark plugs. These discussions were conducted by mean velocity based on the calculation condition under steady-state flow, but given that strong turbulence was observed at the rear of the front fender in Fig. 7 (c), there is a possibility that an unsteady airflow is being generated.

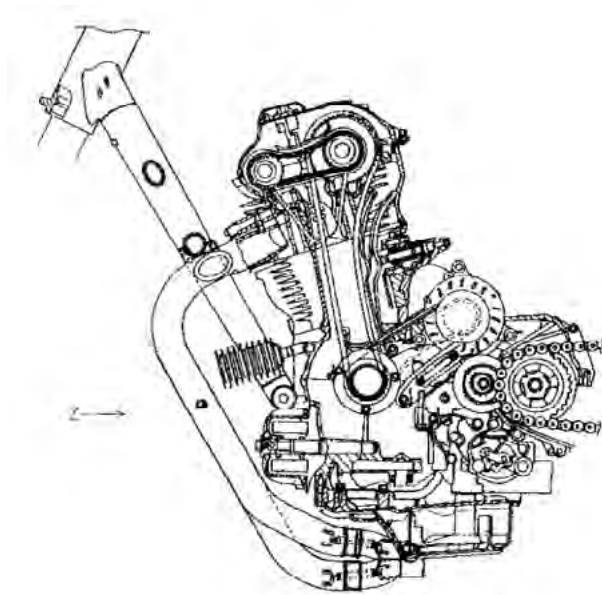


Fig. 16 Cross section of prototype engine

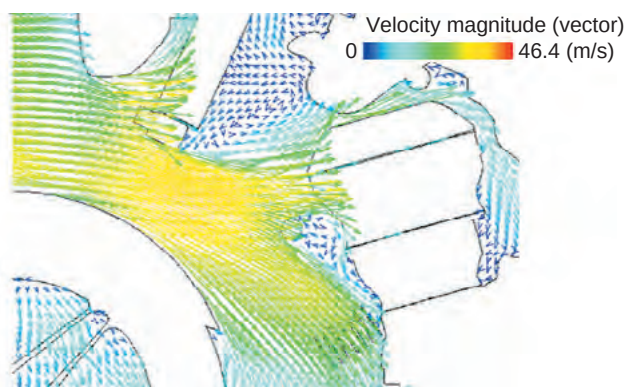


Fig. 17 Velocity vector investigation at oil cooler location

5. Conclusion

An analysis of cooling performance in an air-cooled engine was conducted using the Partial Cells in Cartesian coordinate method. The following conclusions were drawn from this research:

- (1) Analysis of airflow behavior including fine shapes around cooling fins of air-cooled engines became available in a short time using the Partial Cells in Cartesian coordinate method, and practical use of CFD as design support tool became possible.
- (2) The visualization results by the Partial Cells in Cartesian coordinate method correlated well with the visualization results in wind tunnel tests.
- (3) The calculation of the distribution of heat transfer coefficients using Karman's analogy has enabled local studies of heat radiation characteristics, and it has become possible to quantitatively evaluate design specifications. This method can also be used to calculate the temperature distribution in the engine and analyze thermal stress.
- (4) Many and useful knowledge pertaining to cooling fin shapes and the heat transfer coefficient in each part of an air-cooled engine were provided for engine developments.

Acknowledgements

We wish to express our gratitude to all concerned of Honda R&D Sun Co., Ltd. for their cooperation in three-dimensional solid modeling.

References

- (1) Takahashi, Y., Fukuzawa, K., Fujii, I.: Numerical simulation of flow in intake ports and cylinder of multi-valve S.I. engine using PCC method, *Proceedings of International Symposium on COMODIA 94*, p. 529-534 (1994)
- (2) Takahashi, Y., Nomura, T., Ishima, T., Obokata, T.: Numerical analysis on steady state in-cylinder-flow by Partial Cells in Cartesian Coordinate method and verification with PIV measurement, *Transactions of the Japan Society of Mechanical Engineers*, Vol. 71, No. 706, B (2005-6), p. 1694-1701 (2005)
- (3) 3DataExpert, <http://www.deskartes.com>, 2006/1/16
- (4) Tachibana, F., Akiyama, M., Okamoto, Y., Morishita, T.: *Fundamentals of Heat transmission*, Corona Publishing Co., Ltd., p. 44 (1965)

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Characterization of Dielectric Energy Storage Mechanism and Achievement of Continuous Production Method for High Capacitance-targeted Ultracapacitors

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ABSTRACT

To achieve high energy density activated carbon for use in ultracapacitors for automobiles, an activated carbon with twice the specific capacitance per unit volume compared to an isotropic activated carbon was developed. The hypothesis was that higher capacitance is related to the space-charge polarization of the inner edge plane. As well, the preparation of graphitizing carbon domains was researched. The high graphitizing activated carbons derived from new aromatic compounds demonstrate higher capacitance. Activated carbons obtained by conventional mass production have batch processing issues, such as contamination by metals generated by etched reactors, variation in capacitance within production batches, and generation of potassium metals. To resolve the above issues, we investigated a continuous mass-production process, and have successfully developed a new KOH activation process.

1. Introduction

The elimination of emission gases, including CO₂, is an important issue from a global warming standpoint, and fuel economy enhancements are required for motor vehicles to meet this goal. One means of enhancing fuel economy would be through the expanded use of electrical powerplants. This increases the importance of an electric power source with peak power buffer that can absorb load fluctuations. Attention is focused at present on the electric double-layer capacitor, which is a low resistance, highly reliable energy storage device. The market for this device has mainly been for small power supplies for backup power in cell phones, etc. In recent years, its use in uninterruptible power supplies

(UPSs) has grown. Expectations for automotive uses include energy storage for fuel cell vehicles (FCVs) and hybrid electric vehicles (HEVs), electrical assist power supplies for electrical auxiliaries, and separated power supplies for electronic devices. Capacitor strengths lie in the ease of energy regeneration due to high input/output density, and high reliability with no limits to cycle life. The most sought after properties in expanding capacitors use are to maintain their high power density characteristics, their main strength, and to increase energy density. In this development project, energy storage theory was verified and an activated carbon was developed as a new high energy density storage material. Mass production techniques were also developed, and a continuously operating plant process was verified. The results are reported below.

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2. Capacitor Features

The electric double-layer capacitor is composed of cathode- and anode-activated carbon electrodes, and a non-aqueous electrolyte. There are 1-2 nm pores inside the activated carbon. The ions are absorbed as charged ions, and desorbed by a discharge. The advantages of a capacitor over a secondary battery are clear. In a secondary battery there exists an active material chemical change, including the thermodynamic losses accompanying chemical energy conversions, and deterioration over time. The discharge power density of batteries is related to electrode reaction speeds, so the amount of power discharged is limited. In addition, the power density is temperature-dependent, thus the batteries can only be used under a limited environment. Because of this it is necessary to control the charge and discharge current of a battery by using a current limiting device. In contrast, a capacitor maintains an electric potential difference that is generated by an externally applied voltage. Therefore, intrinsic high energy efficiency, rapid charge and discharge is possible, strengthening its repetitive charge and discharge characteristics regardless of environmental temperature, in addition to being highly reliable. Figure 1 shows a comparison of power and energy densities for a capacitor and secondary batteries. Although capacitors have many desirable characteristics, including very high power density, they have very low energy density. To make capacitors useful for automotive applications (and use their high reliability and high power density), it is crucial to create capacitors with higher energy densities.

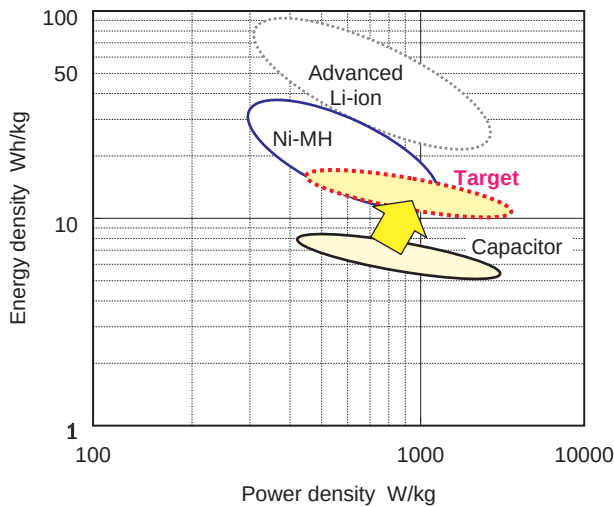


Fig. 1 Ragone plot of capacitor and secondary batteries at 25°C (initial profiles)

3. Development Goals

The development target for the capacitor's energy density was set at 10 Wh/kg, which is approximately three times that of conventional small capacitors. New material technologies were essential to meet this target. The main section explains the technologies that are used to create high capacitance in the activated carbon, as well as the mass production technologies involved.

3.1. Hypothesis of the Theory of New High-Capacitance

Material Methods used to produce high energy density capacitors include raising the rated voltage and raising the capacitance of the activated carbon electrode. The rated voltage for many electric double-layer capacitors is 2.5 V. Though in theory the application

of a higher voltage is possible, this method in practice is restricted, due to a drop in reliability. To raise the rated voltage, a new nonaqueous electrolyte that is unlikely to produce electrochemical degradation is needed. The authors of this study focused on giving the activated carbon a higher capacitance. An electrical charge of electric double-layer is formed between different electrochemical potentials. That is to say, it is formed on the interface between the carbon surface and the electrolyte. Referring to the capacitor's general equation (1), with an electric double-layer capacitor the dielectric existence ϵ or distance between electrical charges d cannot clearly be defined⁽¹⁾. Accordingly, to raise capacitance C , it is important to expand the electric double-layer interface S . It is well known that the capacitance of conventional activated carbon electrodes is extremely large when an electrode's specific surface area is around 2000 m^2g^{-1} . If the area is expanded further, the activated carbon density drops, and the electrode becomes bulky, so raising the capacitance per unit volume is not possible⁽²⁾. However, in 1995 a new activated carbon was discovered that had higher capacitance, despite its smaller specific surface area

$$C = \epsilon_0 \epsilon \frac{S}{d}$$

Eq. 1 General formula of capacitance

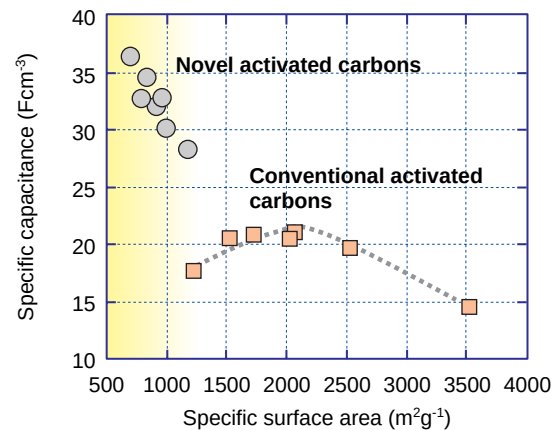


Fig. 2 Relationship between capacitance per unit volume (Fcm^3) and specific surface area (m^2g^{-1}) of activated carbons

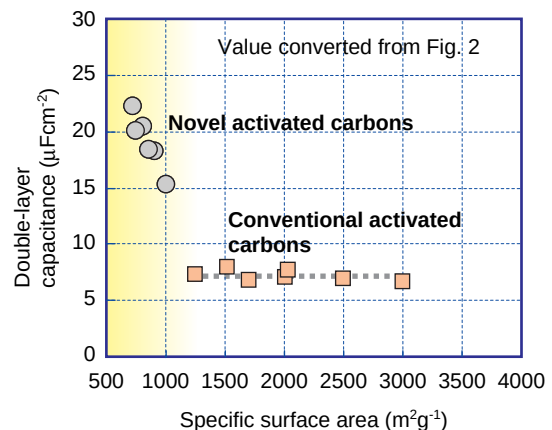


Fig. 3 Relationship between capacitance per unit surface area (μFcm^2) and specific surface area (m^2g^{-1}) of activated carbons

(Fig. 2)⁽³⁾. This activated carbon was obtained by activating a graphitizing carbon to increase its porosity, using potassium hydroxide. Figure 3 shows the results of a cause analysis for increasing material capacitance. The relationship between specific surface area (m^2g^{-1}) and capacitance per unit specific area (μFcm^{-2}) of the newer activated carbon is clearly different from that of conventional activated carbon. As the specific surface area decreases, i.e., the activated carbon density increases, the electric double-layer interface capacity (μFcm^{-2}) also increases. From these results, it was hypothesized that solid dielectrics exist inside of this activated carbon, like those existing in ceramic and aluminum electrolytic capacitors, and capacitance is increased. Specifically, it was assumed that an area of high dielectric constant exists from the edge plane towards the bulk when electric double-layers are formed between the edge surface of the fine graphite crystals, known as the edge plane, and the electrolyte⁽⁴⁾. This reasoning is based on the reports that were published in the 1970s by J. P. Randin and E. Yeager, among others, on their highly oriented graphite electrode research. They studied the differences in graphite fracture surfaces and reported that edge plane capacitance is particularly large, and moreover, that capacitance is voltage-dependent. From these results, they suggest that capacitance was increased due to space charge polarization, and that the increase rate was related to the proportions of edge plane sp^2 - and sp^3 -bonds that were present. They used a semiconductor model to show that capacitance is greater when the sp^2 -bond ratio is higher⁽⁵⁾. The same phenomenon occurs with the porous carbon electrodes, such as those made of activated carbon. To achieve higher capacitance, it was necessary to form an electric double-layer interface with many edge planes with the carbon structure of activated carbon like the structure of graphite.

3.2. Mass Production Issues in Alkaline Activation

The standard activated carbon production method involves carbonizing organic raw materials such as cellulose, and activating them so that the materials become porous by means of oxidation through high temperature water vapor. However, the new material has a graphitizing structure, so gas activation is difficult. Therefore, a production method known as alkaline activation was adopted, which can make the material porous by mixing it with a metal

hydroxide such as potassium hydroxide and creating a reaction at 600 °C or higher.

There are two issues that need to be taken into consideration when using the alkaline activation method for mass production⁽⁶⁾ (Fig. 4-1). The first is that when a reaction involving molten alkaline at high temperatures is used, there is corrosion of the reaction furnace material. Nickel (Ni), with good corrosion resistance, was used, but with continuous operation was found to produce corrosion deterioration. The results of an analysis of this process are shown in Fig. 4-2. The thermal history of furnace causes the grain growth of furnace material (Ni), and molten potassium hydroxide (KOH) permeates the grain boundaries. In the presence of molten KOH, Ni, and activated carbon, electrolytic corrosion accelerates, and Ni particles separate. When these Ni particles become electrodisolution in the activated carbon electrode, as shown in Fig. 5, the electrolytic reduction that occurs at the counter electrode creates a conductive path, leading to a large voltage drop (self-discharge). That is to say, because of the Ni contamination,

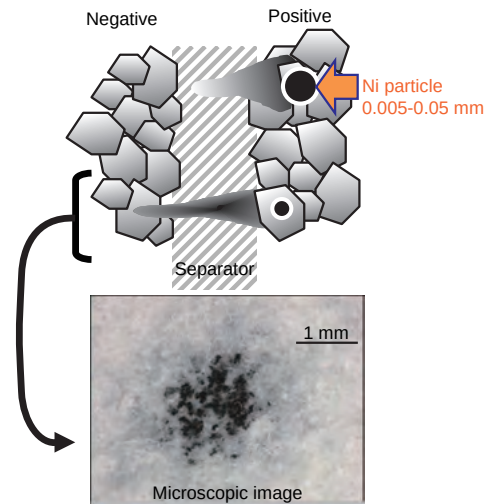


Fig. 5 Model of self-discharge caused by Ni particle and microscopic image of dotted separator

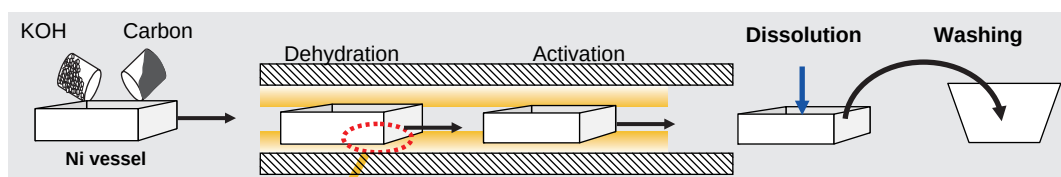


Fig. 4-1 Diagram of conventional process

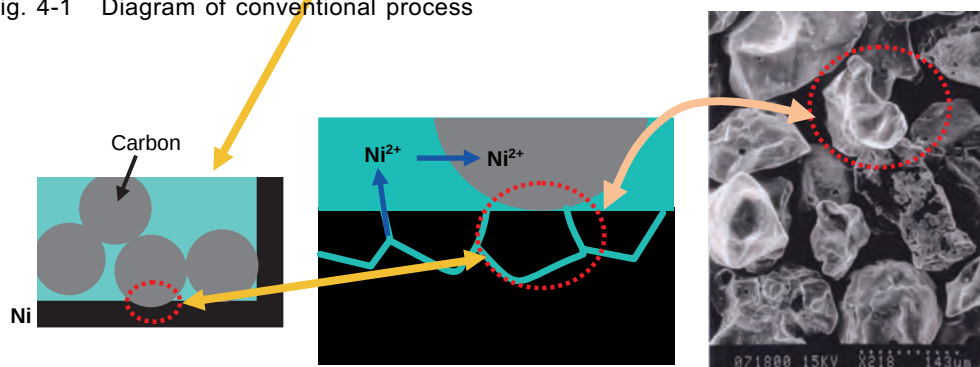


Fig. 4-2 Mechanism of vessel corrosion in conventional KOH activation process

there is a decrease in the capacitor cell yield, producing a reduced lifespan in the production equipment. The second issue is that with high temperature heating, metal potassium is created from the KOH as a by-product, which remains in the reaction system, making it difficult to continue safely operating the system on a long-term basis, and difficult to increase manufacturing quantities. This means batch production is used, which is an impediment to cost reduction efforts. It was necessary to create new mass production methods to address these issues.

4. Means of Resolution

4.1. Technology for Carbocyclic Arrangement for Higher Capacitance

First, the standard graphitizing carbon material production method is explained. When the heavy residues of petroleum or coal are heated, thermal condensation (at approx. 300 °C) multicycles the aromatic compounds, forming condensed polycyclic aromatic compounds. The condensed polycyclic aromatic compounds form stacked aggregates that lead to optical anisotropy (mesophase pitch). This state is the starting point for a graphite structure. The mesophase pitch is further heated at 700 °C for it to be carbonized and pulverized. The resulting graphitizing carbon powder reacts (is activated) with the KOH, creating activated carbon. Mesophase is a molecule group (domain) of condensed polycyclic aromatic compounds, and when the domain grows large, graphene sheets of carbocyclic rings are formed, and graphite material develops. However, if these formations develop too much, the crystallite enlarges, destabilizing KOH impregnation and reaction. As such, since the activation reaction does not occur, activated carbon is not produced. To form more edge planes for higher capacitance, a divided graphite crystalline structure is required. This would necessitate the formation of carbocyclic rings only in the stacked (C axis) direction. However, no carbonaceous production methods like this have been reported, up to now. The authors of this study carbonized various organic materials and characterized their structures. Among them, particularly activated carbon obtained from pitch, which is made from PVC (polyvinyl chloride) resin, showed higher capacitance than that of the activated carbon obtained from mesophase pitch. Focusing on these results, it was assumed that rather than the aromatic rings, which are the main structure of organic raw materials, the adjoining side change

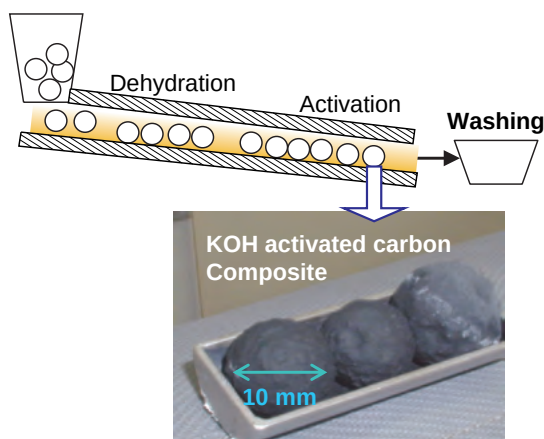


Fig. 6 View of newly developed activation process and picture of activated KOH-carbon composite

molecular structure is more important in developing domain of carbocyclic stack in the C axis direction. A new base organic molecular structure was developed.

4.2. Continuous Alkaline Activation Technology

In addition to addressing the issues in the manufacturing process of furnace material corrosion and by-products such as metal potassium, high quality, low-cost activated carbon for automobile applications must be manufactured. To reduce material characteristic variations, carbon and alkaline should be mixed and heated uniformly. To reduce cost, fast reactions need to occur. As a result of studies, a new continuous production method was created, as shown in Fig. 6⁽⁷⁾. In the conventional activation reaction, an excess amount of solid KOH was mixed in, causing a reaction in the molten KOH. With the new production method, the carbon powder and KOH are mixed in advance in a stoichiometric composition, and after being molded into spherical shapes (composites), they are heat-treated. Changing the composite molding condition and size can change the physical properties of the activated carbon. One feature of this production method is that because the KOH is impregnated with the carbon powder before the high temperature heat-treatment, meltdown of KOH is prevented, thereby avoiding equipment corrosion. Moreover, because the amount of KOH can be reduced, by-products are also reduced. Reaction of the composite on a moving bed, e.g. a rotary kiln, allows a uniform reaction process, enabling activated carbon with a uniform characteristic to be produced.

5. Results

5.1. High-capacitance Activated Carbon Generation Process

We proposed the use of condensed polycyclic aromatic compound pitch (aliphatic pitch: ALP) having a low softening point and plenty of aliphatic groups, which are suitable in obtaining

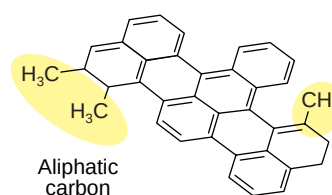


Fig. 7 Design concept of new aromatic compound, ALP

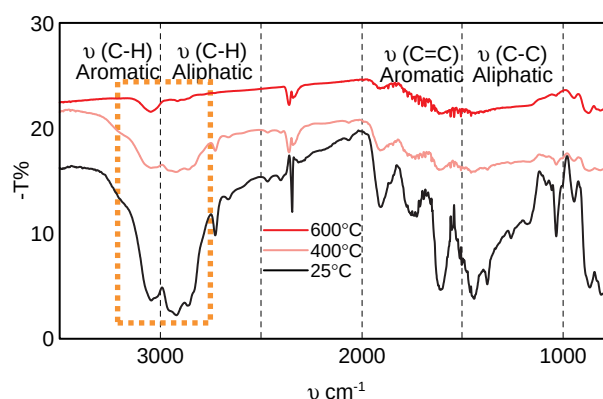


Fig. 8 ALP FT-IR profiles during carbonization process

graphitizing carbon compounds with dispersed fine graphite crystals (Fig. 7)⁽⁸⁾. ALP derived from methylnaphthalene monomer was heat-treated, and the ALP based carbon precursor forming process was analyzed using Fourier transform infrared spectroscopy (FT-IR). Focusing on changes around 3000 cm^{-1} in Fig. 8, the ALP aliphatic -CH peak is greatly reduced due to dehydrogenation, and the formed graphitizing carbon compounds have residual aromatic -CH. These results are shown in the model in Fig. 9. Due to the steric effect of the methyl group, which is an ALP side chain, there is stacking in the layer direction⁽⁹⁾, and because of the presence of naphthene rings, the integral molecular structure is weakened, stacking carbocyclic flat surface structure development in the carbonization process, and at the same time, the aromatic -CH group remains, which is the reactive site. In the scatter plot of Fig. 10, the horizontal axis is the graphitizing carbon atomic ratio H/C, while the vertical axis is the real density of the carbonization. The relationship between the capacitor capacitances obtained is shown. In the carbonization process, dehydrogenation and carbocyclic compound creation and shrinkage occur in turn, so the relationship between the H/C ratio and real density is fixed and the conventional reaction is described by the black line. However, when using ALP, in heat-treatment at comparatively low temperatures, high real density and high H/C (new relationship), new graphitizing carbon compounds can be formed, so activated carbon that used this compound was shown to have a capacitance that is 25% higher, as shown with the dotted line.

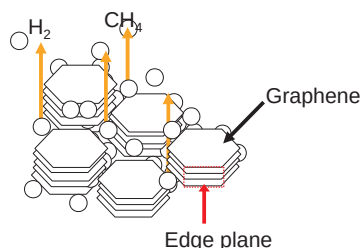


Fig. 9 Model of stacking graphene and graphitizing carbon domain during carbonization

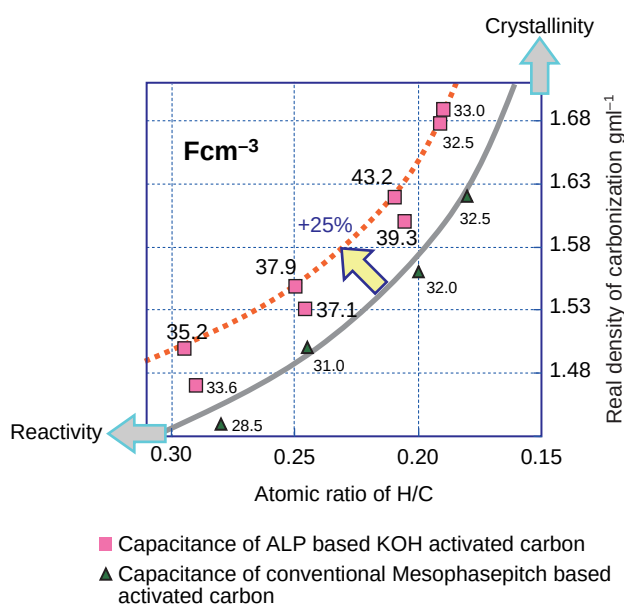


Fig. 10 Scatter plot of capacitances derived from several carbon sources

5.2. Plant Continuous Operation using the New Production Method

The activation furnace (rotary kiln) for the new production method is shown in Fig. 11. The composite is introduced into it from the left side of the diagram, passes through the electrical furnace heating zone, and is discharged after cooling. As shown in the photo in Fig. 6, the activated composite is discharged without any changes to shape. Figure 12 shows the capacitor cell performance results using the activated carbon produced in the activation furnace shown in Fig. 11, and Fig. 13 shows the associated tolerance of self-discharge. Because the heat-treatment process is stabilized and variations in the physical properties of the activated carbon pieces are reduced, the activated carbon capacitance is raised 10% or more, with the accompanying self-discharge yield also enhanced. An advantage of this production method in terms of cost is that it is possible to reduce the amount of KOH used, and thus reduce the metal potassium that is produced as a by-product. This makes it possible to have a continuous manufacturing process. Moreover, Ni, a high-cost metal, is not consumed in the process. With the conventional production method, the activated carbon adheres to the reaction vessel, so it was necessary to have a process where water was poured in and the activated carbon scraped out. However, with the new method, the composite is moved to the washing process in its spherical shape, allowing a continuous process more easily than before. Even if the production quantity expands, the system can be easily adjusted by extending the reaction zone, allowing a reduction in capital expenditures.

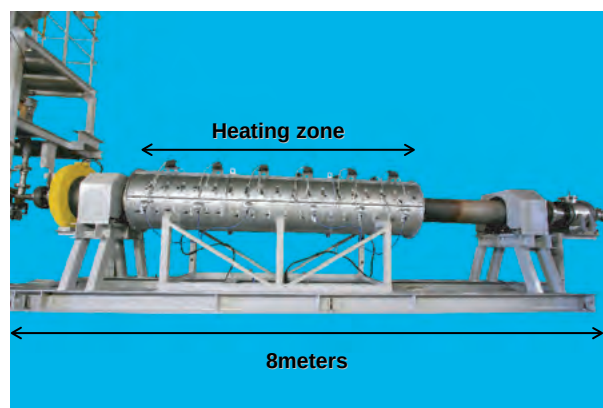


Fig. 11 Picture of newly developed mass production plant

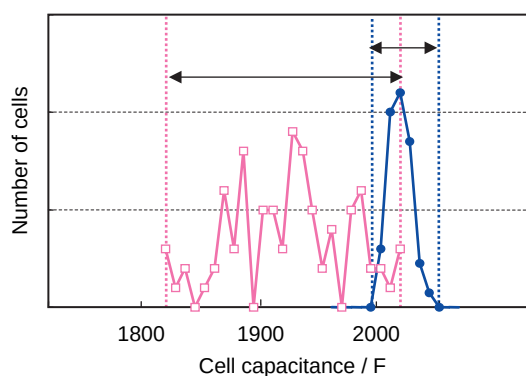


Fig. 12 Histogram of capacitance variation

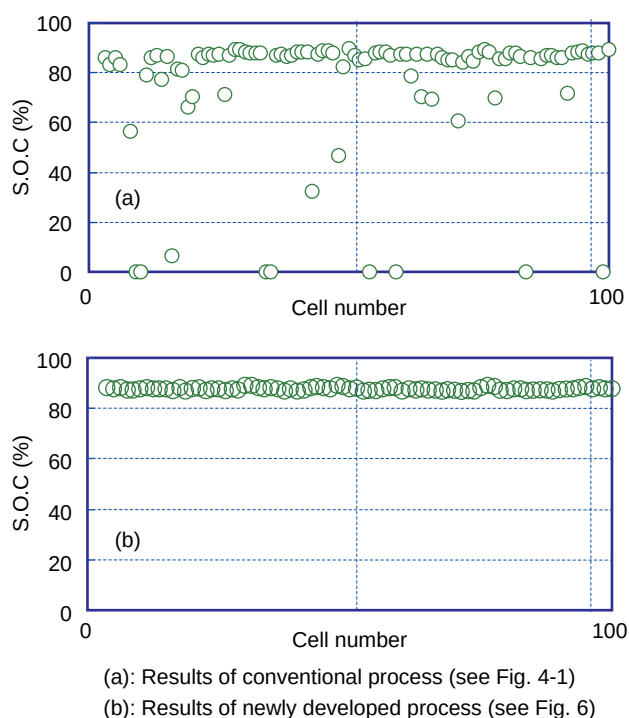


Fig. 13 Comparison of self-discharge characteristics after 72hr at 25°C

6. Considerations

6.1. The Possibilities for Graphitizing Activated Carbon

When designing carbonaceous compounds with a dispersed graphite structure such as ALPs, it is very important to focus on

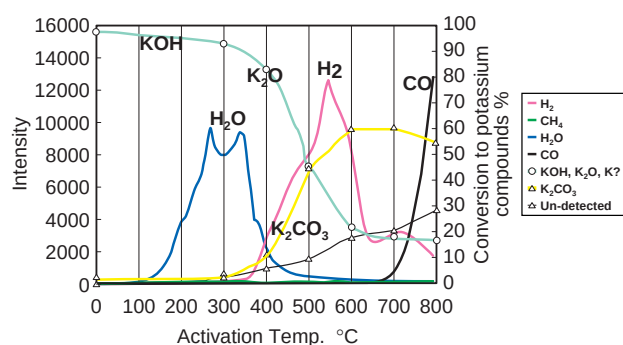
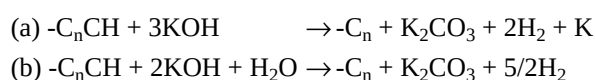


Fig. 14 TDS-MASS and KOH conversion profiles during activation process

the thermal motility of the polycyclic aromatic molecules during the carbon precursor forming process. Specifically, it is possible to suggest that a low-cost organic raw material with a mixed aromatic-aliphatic structure can be searched. In addition, there are additional issues involved when making such compounds with even higher capacitances. If the graphite structure is developed to raise capacitance, this results in a higher carbon bulk density, and pore volume (mlg⁻¹) is reduced. Therefore, the electrolyte ions that would be necessary for charge balance at the electric double-layer interface are in a saturated state at the space, and the activated carbon swells. To overcome this issue in the future, it is important to develop electrolyte ions or an ionic-liquid having high charge balance efficiency properties.

6.2. Proposal of an Alkaline Activation Reaction Mechanism

Until now, few studies had been conducted on the KOH activation reaction mechanism^{(10),(11)}. Therefore, in modifying the procedure for mass production, a detailed analysis of a KOH activation reaction mechanism was carried out, and those results are shown in Fig. 14. From 300 °C KOH dehydration occurs, and at around 400 °C, KOH is consumed and converted to potassium carbonate, at the same time, generating hydrogen. At approximately 600 °C, activation is complete, and at 700 °C or higher, CO desorption on the carbon surface occurs. These results are summarized in Fig. 15. KOH permeates between the graphite crystallite, and reacts with the aromatic -CH, with dehydrogenation, after which the edge plane oxidation occurs. Next, disproportionation of KOH forms potassium oxides, while removing the edge carbon atoms to form K₂CO₃, leading to pore creation. If this reaction is optimized, reaction Eq. 2-(b) is obtained. Conventionally, the reaction is Eq. 2-(a), where hydrogen and metal potassium generation is caused by the excess KOH. However, by introducing a reaction that reduces KOH, the quantity of KOH used is reduced by 15%, thereby reducing by-product metal potassium formation, allowing the formation of an activated carbon continuous manufacturing process.



(a) Conventional KOH activation process
(b) Lower KOH activation

Eq. 2 Comparison of chemical equations

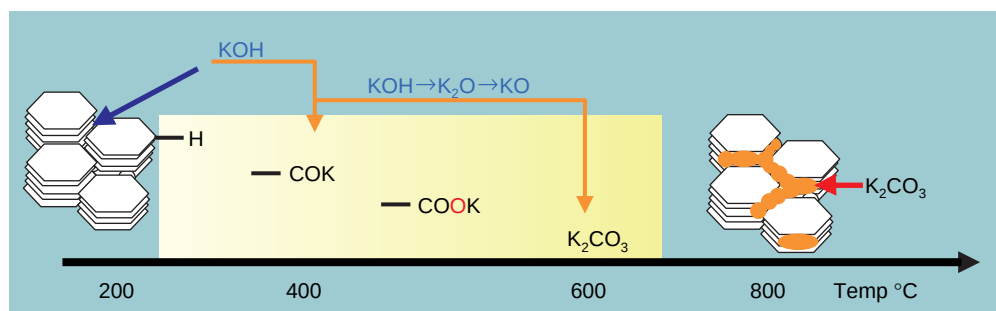


Fig. 15 Mechanism of KOH activation process

7. Conclusion

To increase the energy storage capacity of an electric double-layer capacitor, a high-capacitance activated carbon material was developed, and a mass production process was created.

- (1) A high-capacitance energy storage theory was introduced using activated carbon for the capacitor, and a carbon structure was derived for this purpose.
- (2) As a result of the development of a new organic compound that can produce a high-capacitance carbon structure, capacitor energy storage was increased to twice that of the conventional structure, achieving a practical energy density (wh/kg) equivalent to that of a secondary battery.
- (3) To obtain high quality, low-cost activated carbon, a continuous mass production method was created that resolves the issues associated with conventional alkaline activation production methods. A mass production demonstration plant was built, its operation verified, and its effectiveness confirmed.
- (4) The reaction mechanism of the production method required for this mass production as well as the new reaction mechanism were clarified.

References

- (1) Ue, M.: Denki Kagaku Capacitor you Dennaieki, DENKI KAGAKU (Presently Electrochemistry), Vol. 66, No. 9, p. 904-911 (1998) (in Japanese)
- (2) Morimoto, T., Hiratuka, K., Sawada, Y., Kurihara, K.: Electric double-layer capacitor using organic electrolyte, Journal of Power Sources, Vol. 60, p. 239-247 (1996)
- (3) U. S. 5877935
- (4) Fujino, T., Oyama, S., Oki, N., Noguchi, M., Sato, K., Endo, M., Kim, Y. J.: A characterization of alkaline-activated mesophase-based carbon for electrochemical capacitor, Mol. Cryst. Liq. Cryst., Vol. 388, p. [481]/67-[488]/74 (2002)
- (5) Randin, J. P., Yeager, E.: Differential capacitance study on the edge orientation of pyrolytic graphite and glassy carbon electrodes, Electroanalytical Chemistry and Interfacial Electrochemistry, Vol. 58, p. 313-322 (1975)
- (6) Morimoto, T.: Development and Industrialization of Electric Double-Layer Capacitors, Tanso, No. 214, p. 202-209 (2004)
- (7) JP W2003/089371
- (8) U. S. 6660583
- (9) Mochida, I., Korai, Y., Kun, C. H., Watanabe, F., Sakai, Y.: Chemistry of synthesis, structure, preparation and application of aromatic-derived mesophase pitch, Carbon, Vol. 38, p. 305-328 (2002)
- (10) Maeda, T., Kim, Y. J., Koshiha, K., Ishii, K., Kasai, T., Endo, M., Nishimura, Y., Kawabuchi, Y.: Activation Method of a Mesophase Pitch-based Carbon Fiber, Tanso, No. 201, p. 7-11 (2002)
- (11) Kim, Y. J., Horie, Y., Matsuzawa, Y., Ozaki, S., Endo, M., Dresselhaus, M. S.: Structural features necessary to obtain a high specific capacitance in electric double layer capacitors, Carbon, Vol. 42, p. 2423-2432 (2004)

■ Authors ■



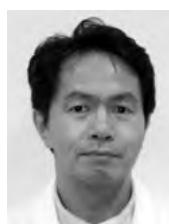
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Study of Durability Prediction Considering Wear Properties for Multiple Plate Clutches

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ABSTRACT

To increase the durability of multiple-plate clutches used in automatic transmissions, attention was focused on the wear history of the facing material. Measurements have confirmed that correlations can be observed between initial wear and disk contact pressure when the clutch is engaged, and between steady wear and plate temperature. Next, simulation technology was developed to quantify the disk contact pressure and plate temperature. When simulated contact pressure distribution and temperature distribution were used to establish correlations with durability wear, good proportional relationships were found in both cases. It was also found that when clutch specifications and driving conditions were varied, the gradient of the correction also varied, but the correlation remained proportional as long as the same facing material was used. The gradient was ranked as a wear property specific to the facing material. By determining the wear property of the facing material, it was possible for the first time to predict wear condition in a test of the clutch as an independent part, for any arbitrary set of clutch specifications and driving conditions.

1. Introduction

Recent increases in engine output have led to demands for lightweight and compact automatic transmissions that have a high torque capacity⁽¹⁾. One feature of Honda transmissions is a compact design that locates the clutch for each speed on three or four parallel shafts⁽²⁾. A smooth shift response with minimal shift shock is achieved by directly controlling the clutch piston pressure that contributes to the response when shifting gears. The parallel shaft mechanism, however, means that the clutch diameter is limited, so torque capacity is achieved simply by increasing the number of disks. In addition, increasing the difference between the inner and outer disk diameters reduces the amount of heat generated per unit disk area, which is considered an effective means for enhancing durability. This makes it important to control disk contact pressure distribution and plate temperature distribution during clutch

engagement. Experience has shown that failure to create an appropriate design can result in abnormal wear and burning of the facing materials⁽³⁾. However, this empiric rule is greatly influenced by the clutch structure, wear properties, operating conditions and other factors⁽⁴⁾, so quantization is difficult. For this reason, clutch durability has been verified up to now solely by long-term durability testing.

Therefore, this paper is focused on the facing material wear history, and clarifies that disk contact pressure distribution and plate temperature distribution both have qualitative correlations to the amount of clutch wear. Simulation technology was developed, allowing the disk contact pressure distribution and plate temperature distribution to be quantified, and making it possible for the first time to predict wear history. This predicted wear history can then be used to set specifications, including clutch durability, in the initial design stages. The details are reported below.

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2. Study Goals

Figure 1 shows the main section of the 5-speed automatic transmission used for the analysis. As shown in this figure, the clutches and gears for each speed are located on the parallel shafts, and the shifting process allows shifting from clutch to clutch by controlling the oil pressure supplied from the oil holes provided inside the rotating shaft. Focusing on the clutch structure (Fig. 2), the guide is fixed to the shaft via a spline, and the plates that mesh on the outer teeth rotate in sync with the guide. The disk has differential rotation from the plate before engagement starts, and this differential rotation reaches zero when engagement is finished. During this engagement process, the facing materials affixed to the disk transmit motive power by friction, but at the same time are damaged by heat generation⁽⁵⁾. When this damage is repeated over a long period, wear and burning may appear. Therefore, the goals of this study were set as follows.

- (1) To analyze the disk wear mechanism
- (2) To establish technology that can predict wear amounts in the initial stages of clutch design

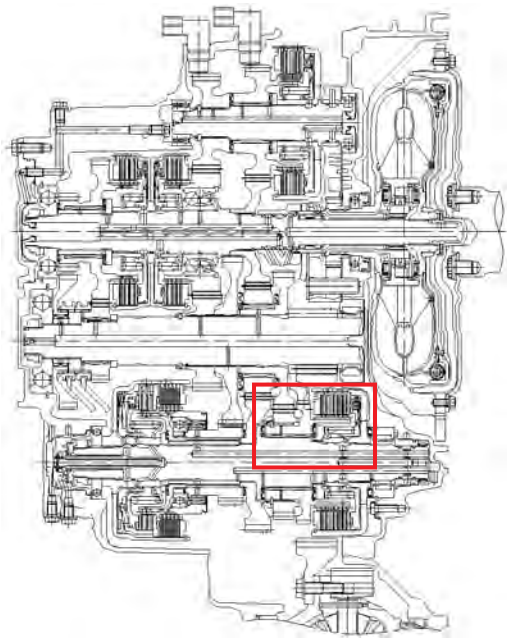


Fig. 1 Main section

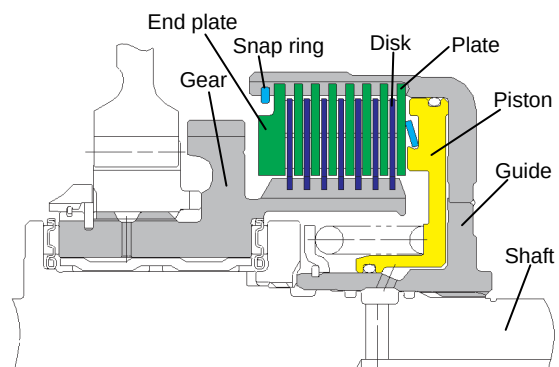


Fig. 2 Clutch detail

3. Analysis of Disk Wear Mechanism

Figure 3 shows the wear history obtained using a clutch pack durability test apparatus (Fig. 4). The horizontal axis shows the number of engagement cycles, and the vertical axis shows the wear amount. The wear amount was obtained by measuring the thickness of each disk and totaling the decrease in thickness. Note that the wear amount is known to be greatly influenced by the surface roughness of the disks and plates as well as by the amount of lubricant used, but these parameters were treated as constant values in this study. The figure clearly shows that the wear pattern is divided into an initial wear area in which wear progresses at a low number of engagement cycles, a steady wear area thereafter in which wear progresses at a gradual rate, and a transient wear area between the other two areas. Therefore, the wear patterns in each area were investigated.

3.1. Correlation between Initial Wear and Contact Pressure

The facing material structure consists of various fibers and inorganic matter bonded together by phenol resin with gaps, called pores, maintained between the various fibers. The facing materials ensure anti-wear performance while the clutch is engaged by absorbing and discharging lubricating oil through these pores. Facing materials generally have a Young's modulus of an order up to three digits smaller than that of steel or other materials, so there is significant deformation relative to the contact pressure load. In addition, the compression-displacement properties of the porous structure exhibit a response such as that shown in Fig. 5. Looking at the initial response, the displacement increases significantly under even the slightest contact pressure load. This indicates that large forces are not required to crush the aforementioned pores. This also means that there is little restorative force to return the porous structure to its original shape after a load is removed. Based on the above, the initial wear phenomenon is thought to be due to the yielding of the facing material under contact pressure loads.

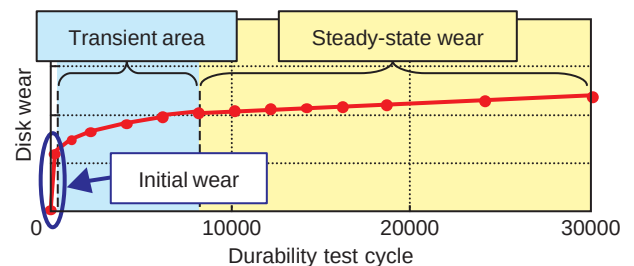


Fig. 3 Disk wear behavior

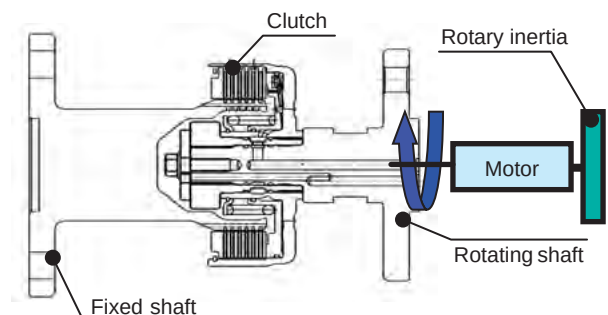


Fig. 4 Clutch fatigue test apparatus

Therefore, the correlation between the initial wear rate per cycle and the disk contact pressure was investigated using the clutch specifications shown in Fig. 6. Three-disk specifications were used to facilitate the clasp of pressure scaling paper between the disks. The disk contact pressure values in this figure are a visual representation of the color shade produced by the pressure scaling paper in the form of a contour chart. In addition, the initial wear rate was obtained by measuring the wear amount at two locations in the radial direction of each disk, and dividing the mean amount by the number of cycles. The wear amount was measured by conducting durability tests with operating conditions that varied the piston pressure F between two levels and the work energy Q/A between two levels. The resulting graph shows that the initial wear rate has a proportional relationship to the disk contact pressure and work energy (Fig. 7).

3.2. Correlation between Steady Wear and Temperature

The steady wear rate is smaller than the initial wear rate. That is to say, when operation enters the steady wear area, the damage per cycle becomes extremely small, and the facing material exhibits the wet clutch characteristic of maintaining long-term durability. However, even though there is only a slight increase, the wear curve still increases in the steady area (Fig. 3), so the influencing factors were investigated. As mentioned above, the facing materials include phenol resin, and have characteristics such that the weight decreases little by little as the temperature rises. In addition, it is known from thermogravimetry (TG) analysis and other methods that the weight decreases significantly at a certain temperature.

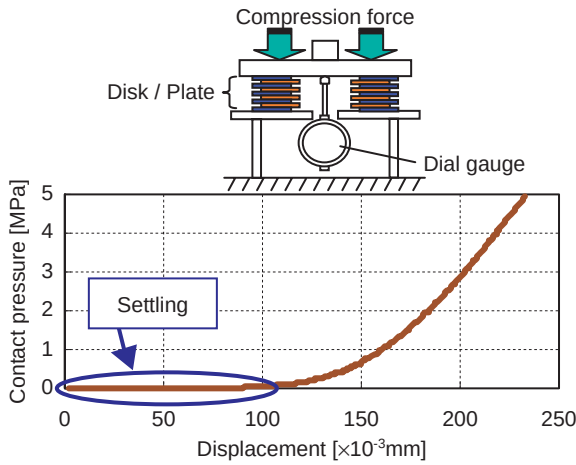


Fig. 5 Compression-displacement property

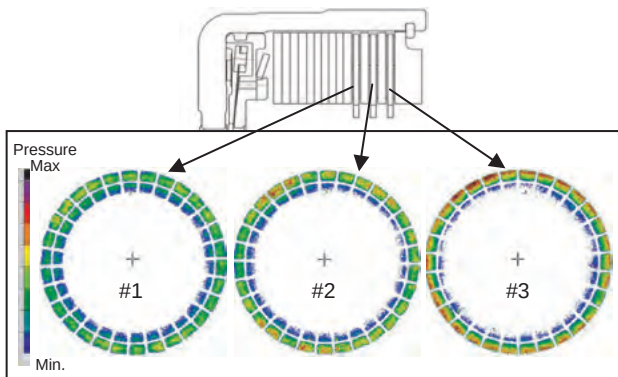


Fig. 6 Measured 3-disk clutch pressure distribution

Therefore, temperature was thought to have some relationship with steady wear, and the steady wear amounts obtained as a result of the durability tests with three-disk specifications were organized according to the maximum temperature during engagement. The temperatures were measured using thermocouples embedded in each plate (Fig. 8). As operating conditions, the piston pressure was fixed and the work energy was varied between three levels. The results showed an excellent correlation between the steady wear rate and the maximum temperature under high work energy conditions, but the wear amounts sampled in the lower work energy areas were smaller, so somewhat ambiguous results were obtained (Fig. 9).

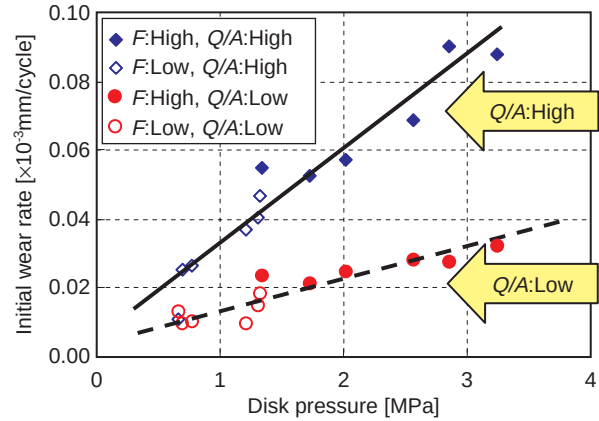


Fig. 7 Correlation of initial wear and disk pressure

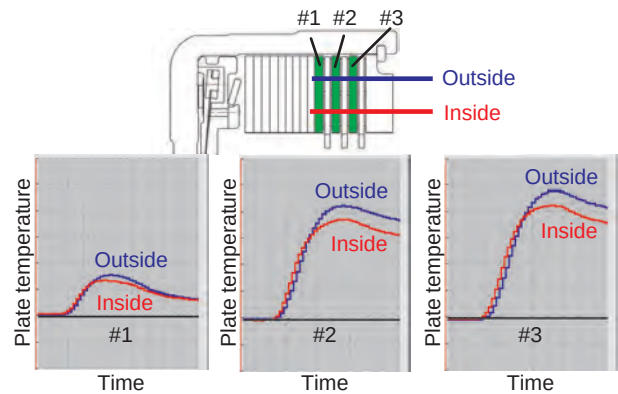


Fig. 8 Measured 3-disk clutch plate temperature

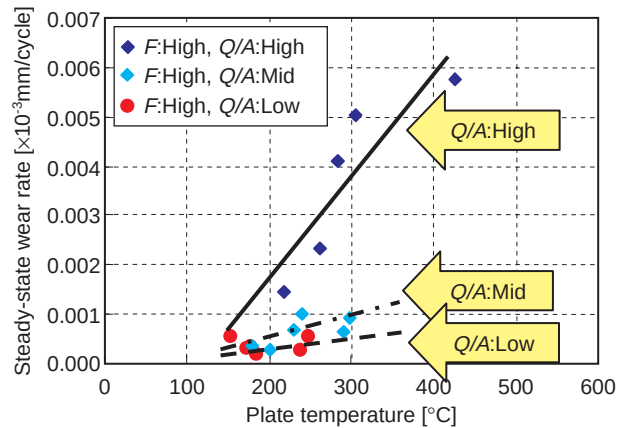


Fig. 9 Correlation of steady state wear and plate temperature

4. Development of Simulation Technology

Analysis of the disk wear mechanism showed a correlation between the initial wear rate and the contact pressure, and the steady wear rate is thought to be related to the maximum temperature. Therefore, the contact pressure distribution and the temperature distribution were quantified using structural analysis.

4.1. Disk Contact Pressure Simulation

Figure 10 shows the structural analysis model used to simulate the contact pressure distribution.

The analysis model is a cyclic, symmetric elastic model, and the angle was selected so that an integer multiple of this model produces the complete clutch shape. The disks are composed of a core plate and facing materials, so contact pressure analysis needed to accurately reproduce the facing material area. Therefore, the facing materials were modeled using shell elements and were attached to the disk surface. In addition, the compression-displacement properties of the facing materials were introduced to the analysis as a property of the contact surface. Figure 11 shows the contact pressure distribution results for typical facing material segment shapes. The contact pressure distribution of each disk changes according to the piston pressing position, the snap ring receiving position, the number of disks, the disk thickness, and other factors, so the distributions are non-uniform.

Next, the simulation results were compared with contact pressure values measured using a tactile sensor to verify the simulation accuracy (Fig. 12). The compared positions were the two locations at which the wear amount was measured for each disk. As shown in Fig. 13, the resulting difference was found to be within $\pm 15\%$.

4.2. Plate Temperature Simulation

The temperature distribution was simulated using the heat transient analysis method. However, to use this method various thermal physical property values must be identified. These values are the thermal conductivity that represents the rate of heat transfer within the same material, the heat transfer coefficient that determines the rate at which heat is transferred and received via a contact surface, and the thermal diffusion coefficient that represents heat dissipation due to lubrication and the air. Therefore, temperature measurements and simulations were performed to identify each of these thermal physical properties individually. The identification method for one of these thermal physical properties, the heat transfer coefficient, is described below.

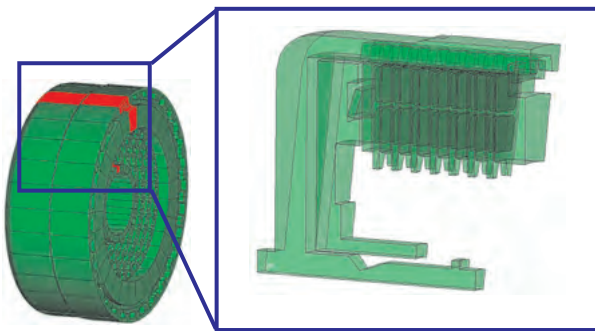


Fig. 10 Analysis model for contact pressure distribution

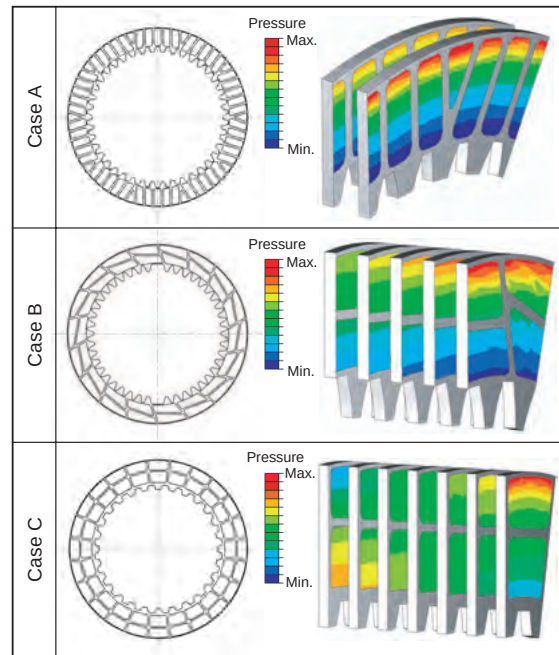


Fig. 11 Results of contact pressure distribution analysis

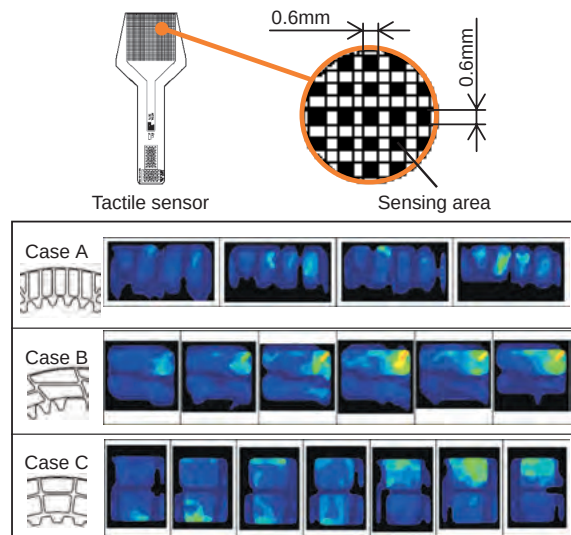


Fig. 12 Contact pressure distribution measurements

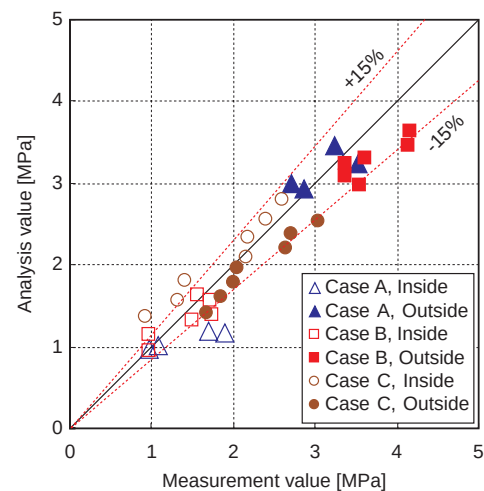


Fig. 13 Accuracy of contact pressure distribution analyses

Figure 14 shows the test apparatus used to obtain the heat transfer coefficient. Plates and wet disks were stacked alternately on top of the testing table, and pressure was then applied to reach the specified contact pressure. In addition, the uppermost plate was heated using a heater, and the temperature of the middle plates was measured using thermocouples. The measurement results shown in Fig. 15 clearly indicate that the temperature rises more slowly for the lower plates.

A heat transient analysis model that includes the testing table was created to simulate this heat transfer, and the heat transfer coefficients were adjusted to provide the same temperature history for each plate. The dotted line in Fig. 15 shows the temperature history ultimately obtained from these calculations. The measurement and simulation results match closely, thus verifying the accuracy of the identified heat transfer coefficient.

The temperature distribution during clutch engagement was simulated using the three thermal physical property values identified in this manner. At this time, it is necessary to apply the heat flux to the entire disk surface as a load condition. The heat flux q_i at an arbitrary position is calculated by the following equation.

$$q_i = r_i \times \mu_i \times p_i \times \omega$$

r : Radial position, μ : Dynamic friction coefficient,

p : Contact pressure, ω : Slip velocity, i : Position variable

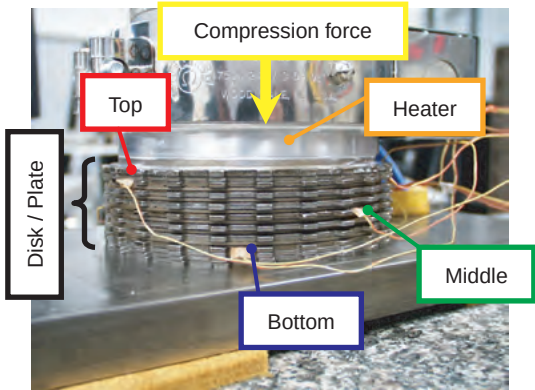


Fig. 14 Measurement apparatus for identifying heat transfer coefficient

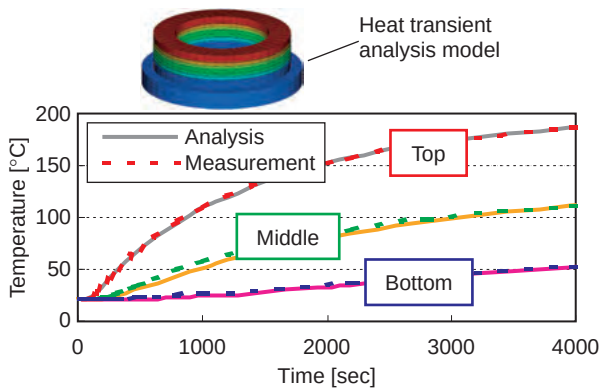


Fig. 15 Identification of heat transfer coefficient

Here, the dynamic friction coefficient used the values obtained by the SAE #2 function test, and was treated as a function of the contact pressure, slip velocity and temperature (Fig. 16).

In addition, the contact pressure used the results of the contact pressure distribution simulation. Figure 17 shows that the maximum temperature of each plate during the engagement process when varying Q/A could be accurately reproduced. However, detailed examination shows that the temperature error increases for plates #6 and #7, counted from the piston.

Therefore, the plate temperature rise error was investigated, and was found to be approximately $\pm 20\%$ (Fig. 18). This is thought to be because the same thermal diffusion coefficient that represents the effects of heat dissipation due to the lubricating oil was used for all the plates in the simulation. It is thought that this error can be reduced in the future by clarifying the flow of lubricating oil inside the clutch.

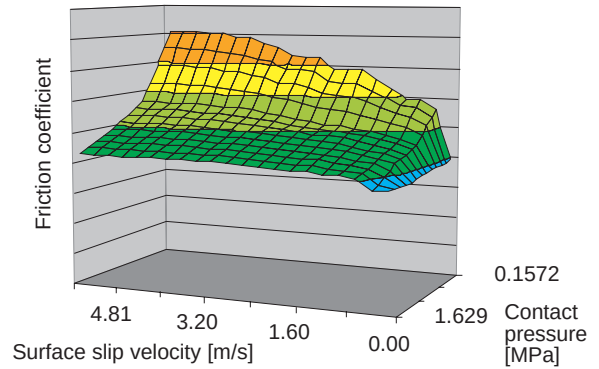


Fig. 16 Dynamic friction coefficient

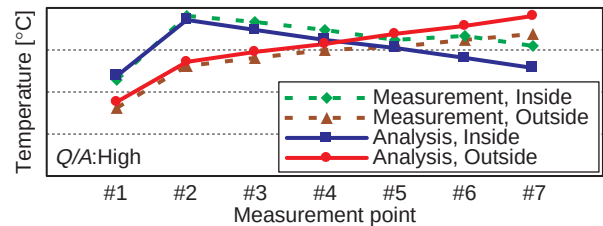
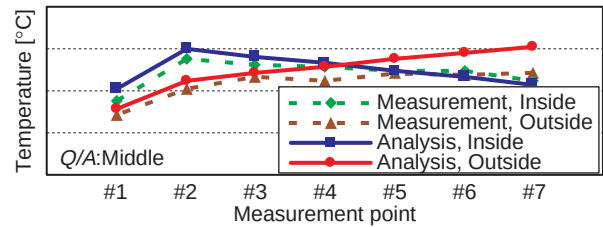
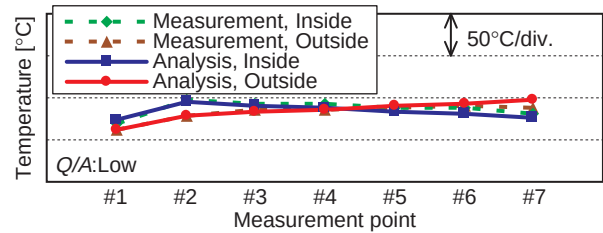


Fig. 17 Plate temperature distribution

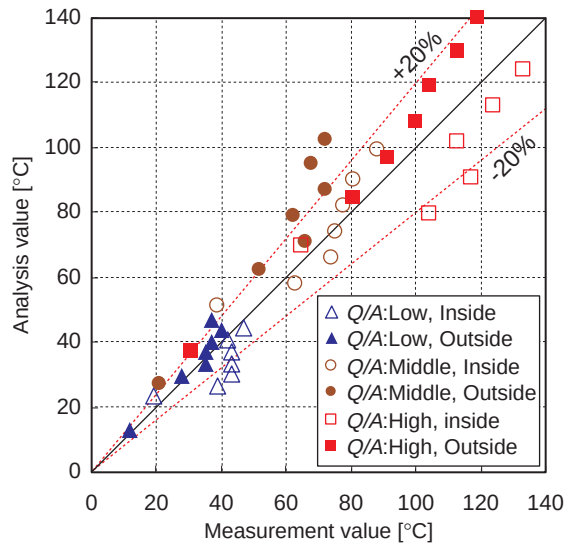


Fig. 18 Accuracy of temperature distribution analyses

5. Development of Wear Amount Prediction Technology

5.1. Wear History Calculation Method

The development of simulation technology enabled quantization of the disk contact pressure distribution and the plate temperature distribution during engagement. Therefore, the correlation with the durable wear amount (dotted line in Fig. 19) was investigated once again using actual clutch specifications.

The results yielded the proportional relationship between the initial wear rate and the disk contact pressure shown in Fig. 20, and the proportional relationship between the steady wear rate and the plate temperature shown in Fig. 21. This substantiates the results of the previous wear mechanism analysis.

The important point here is that proportional relationships were obtained, and since the slope (K_i and K_s) of each relationship is known, the wear histories can be reproduced using the following simple equations.

$$W_i = P \times K_i \times N$$

$$W_s = T \times K_s \times N$$

W_i : Initial wear amount, W_s : Steady wear amount

P : Contact pressure, T : Temperature, N : Number of cycles

In addition, the average value of these wear histories is used as the wear amount W_T in the transient area that lies between the initial wear area and the steady wear area.

$$W_T = (W_i + W_s) / 2$$

The solid lines in Fig. 19 are the wear histories calculated using the above equations. They match closely, both qualitatively and quantitatively, with the actual measurements. This shows that when the results of durability tests are available, the wear histories can be expressed by obtaining the slopes K_i and K_s . However, a remaining issue was that wear amounts could not be predicted unless the durability test results were available.

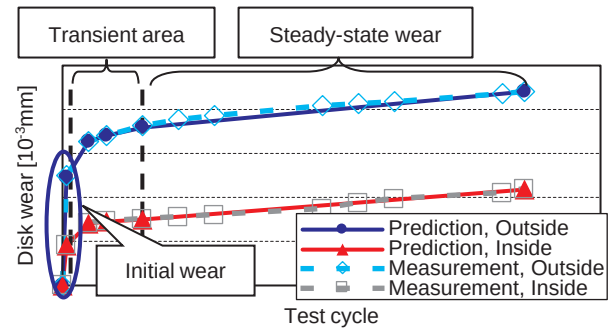


Fig. 19 Wear prediction for clutch disk

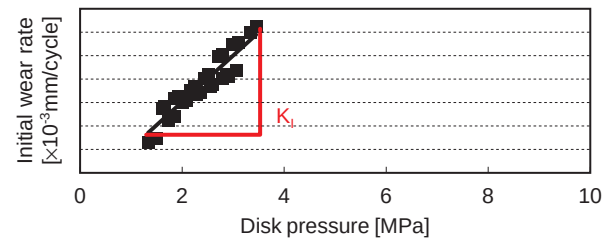


Fig. 20 Relationship between initial wear rate and pressure

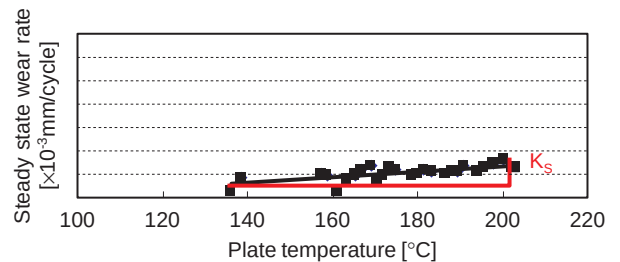


Fig. 21 Relationship between steady-state wear rate and temperature

5.2. Facing Material Wear Property Maps

Therefore, these relational equations were applied to various durability test results using different clutch specifications and Q/A. In these calculations the segment shape was allowed to differ, but only the facing materials used the same materials in all cases. The results clearly showed that while the slopes K_i and K_s differed in each durability test, the slopes changed in the order of the work energy. Figures 22 and 23 show this relationship in the form of 3-D graphs. These graphs include the durability test results using ten different sampling clutch specifications. These two graphs show 3-D maps of the specific wear properties of the facing materials.

The ability to obtain facing material wear property maps made it possible to obtain the slopes K_i and K_s , which are needed to calculate the wear amounts for arbitrary clutch specifications and operating conditions by interpolating the data from these wear property maps.

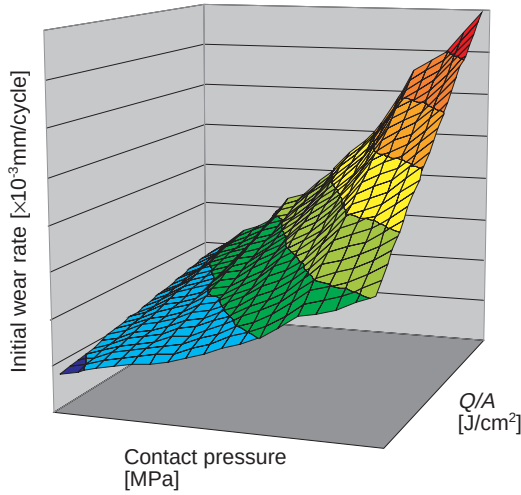


Fig. 22 Initial wear properties map

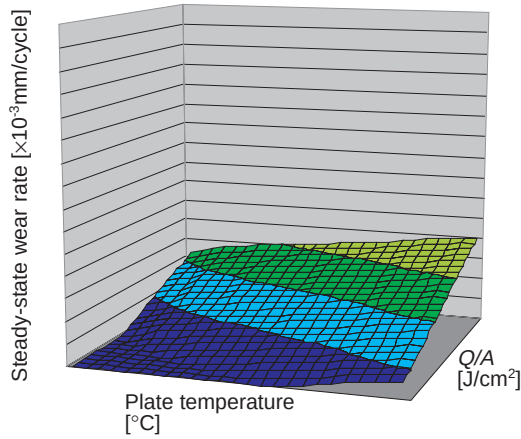


Fig. 23 Steady-state wear properties map

6. Application to the Initial Stages of Clutch Design

Changing the piston shape to increase the effective clutch diameter is often considered a means of increasing the torque capacity without increasing the clutch size. Figure 24 shows three typical formats for the piston pressing position. Viewed simply in terms of the effective clutch diameter, the torque capacity increases in the order of inner, flat and outer pressing types. However, even though the torque capacity increases, it may not be possible to use these methods if the wear amounts would also increase accordingly.

Therefore, the newly developed technology for predicting wear history was applied to these three piston formats. Figure 24 shows that the disk contact pressure distribution varies greatly depending on the piston pressing position. The inner pressing piston exhibits an even contact pressure distribution over the entire disk, but the contact pressure tends to decrease on the inside of the disk and shift towards the outside as the piston format shifts towards the flat and outer pressing types. Also, the plate temperature distribution basically shows the same tendency as the contact pressure distribution, in which the maximum temperature tends to increase as well (Fig. 25).

Next, Fig. 26 shows the results of predicting the wear history using the simulated disk contact pressure distribution, plate temperature distribution, and the facing material wear properties. The two lines in each graph represent the wear amounts in the center of the facing material inner segment as well as in the center of the facing material outer segment. The wear amounts were compared using the disks with the greatest wear amounts. The number of cycles until the maximum allowable wear amount was reached was determined for each specification. The inner pressing type was found to have the longest life, followed by the flat pressing type and outer pressing type, in that order. In addition, judging by the number of cycles required by this clutch, the outer pressing type specifications cannot be used. Examination of the gap between the two lines shows that the inner pressing type uses the entire disk surface, but there is little wear on the inside of the disk for the outer pressing type, indicating that this area is not used.

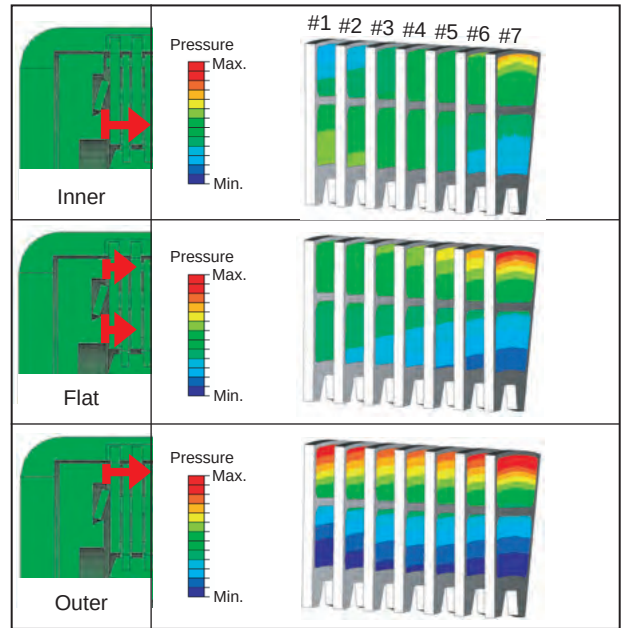


Fig. 24 Difference of contact pressure distribution

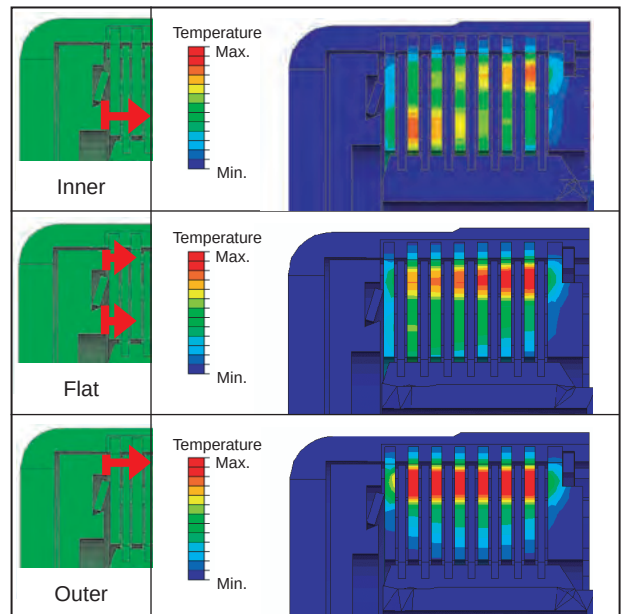


Fig. 25 Difference of temperature distribution

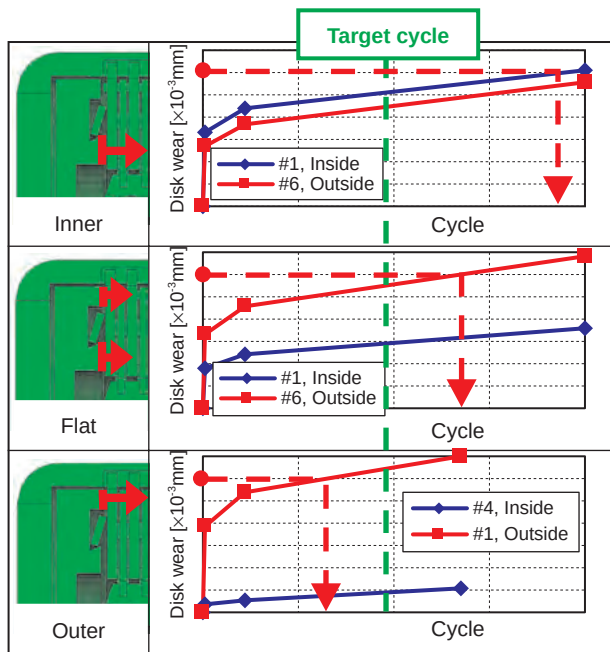


Fig. 26 Difference of amounts of wear

Thus as shown above, the newly developed technology makes it possible to set specifications, including clutch durability, in the initial design stages.

7. Conclusion

Detailed analysis of disk wear amounts in a clutch pack yielded the following information, which is needed to set specifications, including clutch durability, in the initial design stages.

- (1) Contact pressure was found to be the main factor for the initial wear amount, and maximum temperature for the steady wear amount.
- (2) Analysis technology was developed for simulating the disk contact pressure distribution and plate temperature distribution, and the disk surface status during engagement was quantified.
- (3) Wear amounts could be predicted for optional clutch specifications and operating conditions by obtaining the unique wear properties of the facing materials.

8. Afterword

The multiple plate clutch structure has existed for a number of decades, but the manifestation of the wear properties^{(6),(7)}, vibration issues, lubricating oil flows⁽⁸⁾ and other characteristics have not yet been completely explained. Multiple plate clutches are a typical tribological mechanism, so complete quantization is difficult, but the authors hope to continue applying practical engineering approaches in the future.

References

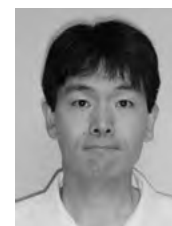
- (1) Kato, Y., Shibayama, T.: Mechanism of Automatic Transmissions and Their Requirements for Wet Clutches and Wet Brakes, *Tribologist*, Vol. 39, No. 12, p. 6-11 (1994)

- (2) Uchida, Y., Miyamoto, O., Otani, T., Maruyama, T., Sakai, K., Tanimoto, S.: Development of New Compact 4-speed Automatic Transmission for Honda LIFE, *Honda R&D Technical Review*, Vol. 16, No. 2, p. 79-86
- (3) Inoue, M., Arai, Y., Azegami, T., Hirao, Y.: Research on Reduction of Unusual Noise in Wet Multiple Disk Clutch for 4WD, *Honda R&D Technical Review*, Vol. 15, No. 2, p. 133-138
- (4) Kato, Y., Akasaka, R., Chiba, N., Mori, Y.: Analysis on the Friction Characteristics of a Paper-Based Facing for a Wet Clutch of an Automatic Transmission, *Tribologist*, Vol. 39, No. 12, p. 71-78 (1994)
- (5) Natsumeda, S.: On the Engagement Mechanism of a Friction Plate, *NSK Technical Journal*, No. 655, p. 32-38 (1993)
- (6) Haj-Fraj, A., Pfeiffer, F.: Modeling of Wet Clutches in Automatic Transmissions, *The 2nd International Conference on Recent Advances in Mechatronics*, p. 9-15 (1999)
- (7) Miyoshi, T.: Friction Characteristics and Porosity of Friction Plate, *NSK Technical Journal*, No. 655, p. 25-31 (1993)
- (8) Miura, T., Sekine, N., Azegami, T., Murakami, Y.: Friction Property of Segment Bonded-type Wet Clutches for Automatic Transmission, *HONDA R&D Technical Review*, Vol. 10, p. 142-149

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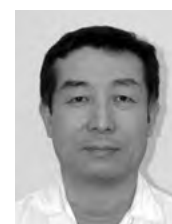
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Establishment of Analysis Method for Weight Reduction Design of Motorcycle Frame Parts

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ABSTRACT

In the present development of motorcycle frame parts, basic skeleton layouts are designed by repeating some examinations in order to achieve weight reduction considering rigidity and strength targets. Various structural optimization technologies were studied to reduce the man-hours required for the repeated examination. Instead of expanding the design area of the topology optimization technology, an analysis was established by using a method of virtual rigidity enhancement. The analysis obtains ideas easily and rapidly for weight reduction in consideration of rigidity.

1. Introduction

Weight reduction increases both marketability and product appeal, and is constantly demanded in motorcycle development. In addition, demands for weight reduction have also increased in recent years from the viewpoint of enhancing fuel efficiency in relation to environmental issues.

In response to these demands, the use of CAE-based structural optimization technology for parts has become quite common, and is yielding the desired effects⁽¹⁾.

However, advanced CAE operation skills are required, and in many cases this work must be performed by expert analysis engineers. In addition, creating analysis models, calculating them and evaluating the result take much time, therefore it is difficult to conveniently reflect results in development. Motorcycle frames, in particular, must have efficient engine and auxiliary layouts while maintaining the required strength and rigidity, so the investigation of basic skeleton layouts in the initial development stages often

depends on the know-how of experienced designers.

Therefore, analysis technology is needed that can conveniently provide ideas for reducing weight at the basic skeleton layout investigation stage. A new analysis technology for lightweight design that applies topology optimization technology has been developed to meet these demands. This paper describes an outline of this analysis technology for developing motorcycle and all terrain vehicle (ATV) chassis parts, and presents examples of analysis.

2. Development Goals and Target Requirements

Various performance evaluation items such as rigidity, strength and vibration must be investigated when designing motorcycle parts. This study aimed to develop a weight reduction technology that considers rigidity, which contributes significantly to the overall structural balance.

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Conventionally, repeating actual vehicle tests based on past experience, or CAE analysis was performed as shown in Fig. 1. However, iterative process must be reduced to increase developmental efficiency. Therefore, a target process was set as shown in the lower half of Fig. 1. For designers to examine ideas by themselves within this target process, the analysis time must be relatively short, operation must be simple and the results used for evaluation must be easy to understand.

Therefore, the following target requirements were set.

- (1) Ideas for shapes to reduce weight in consideration of rigidity should be obtained.
- (2) The total man-hours spent on analysis should be one working day (10 hours) or less.
- (3) The analysis technology should be easy to use for designers in the future.

3. Theory Establishment

3.1. Comparison of Various Structural Optimization Methods

Conventional analysis technology for lightweight design consists of thickness optimization, shape optimization and topology optimization.

Thickness optimization performs optimization using thickness as the design variable. However, the use of solid meshes for structural analysis is increasing with the recent spread of 3D data product drawings in the industrial world. Solid meshes express thickness using shape, so the concept of thickness does not exist. This means that thickness cannot be set as a design variable, so thickness optimization analysis is not possible. Shape optimization requires numerous calculations when there are ten or more dimensions to be set as design variables, so it is not suitable for analyzing entire models. In addition, the large number of modeling man-hours needed to define the design area is an issue for topology optimization. Therefore, as an alternative, the concept of increasing the Young's modulus hypothetically was introduced in topology optimization, and the feasibility of new structural optimization analysis method was investigated. These details are described below.

3.2. Conventional Topology Optimization

On the basis of the load and constraints set for given space, as shown Fig. 2, topology optimization is a method for obtaining the optimum layout, as shown in Fig. 3. This determines how the materials should be distributed to obtain the strongest (most rigid) structure.

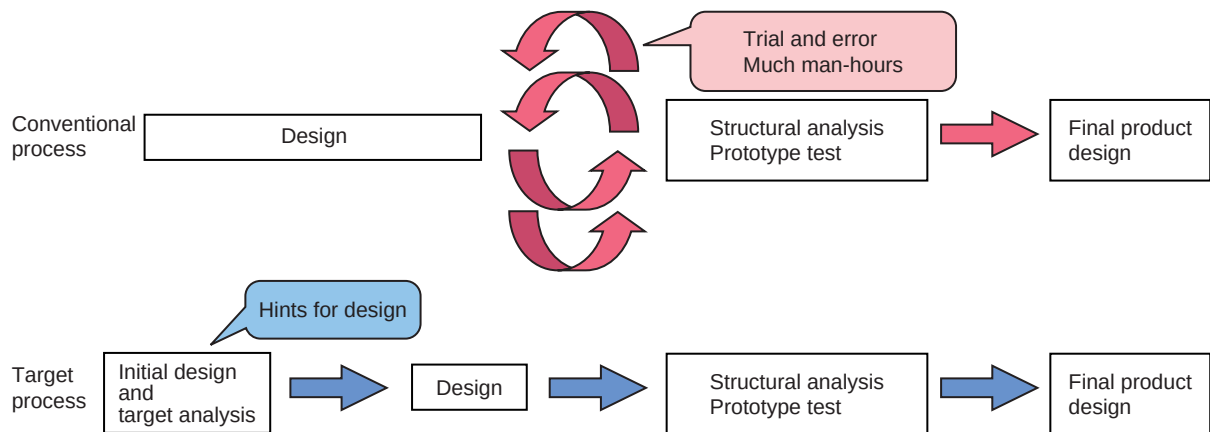


Fig. 1 Target process of design process

Method	Thickness optimization	Shape optimization	Topology optimization
Image			
Design variable	Thickness	Dimension	Density
Merit	Low analysis costs (Shell model) Easy to set design parameters Fast calculation time	Effective for finding ideal shapes	Low analysis costs Easy to set design parameters Fast calculation time
Demerit	Difficulty in analysis by solid mesh models	Large amount of man-hours Difficulty in design parameters setting Difficulty in analysis of complex shapes Long calculation time	Difficulty in design area setting of complex shapes

Table 1 Comparison of various structural optimization methods

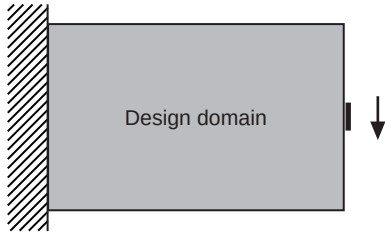


Fig. 2 Design domain for topology optimization analysis

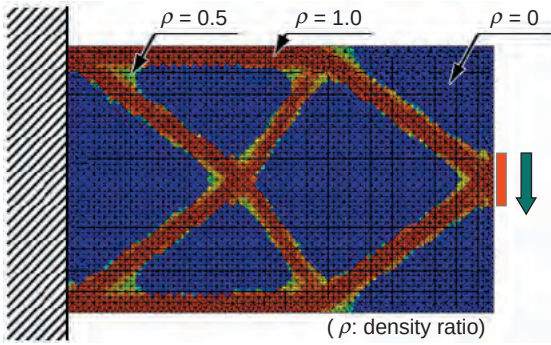


Fig. 3 Topology optimization using density approach

The methods used in topology optimization include the homogenization and density methods. The density method is described here.

In Hooke's law used in structural analysis, if it is hypothesized that the strain is the displacement as shown in Eq. (1), the strain decreases as the Young's modulus increases. Therefore, it can be said that the displacement is small.

$$\varepsilon = \sigma/E \quad (1)$$

ε : Strain

σ : Stress

E : Young's modulus

The density method assumes that the Young's modulus of the material is proportional to the power of the density as shown in Eq. (2), and obtains the optimum topology, using the density ratio of each mesh as design variables^{(2),(3)}.

$$E = \rho^N E_0 \quad (2)$$

E : Young's modulus used in the calculations

ρ : Density

N : Multiplier

E_0 : Set Young's modulus

As shown in Fig. 3, the Young's modulus used in the calculations fluctuates from 0.0 to E_0 by varying the density of each mesh from 0.0 to 1.0. Optimization consists of using each mesh as a design variable and performing iterative calculations to obtain an optimum solution that maximizes or minimizes the objective function while satisfying the constraints.

3.3. Issues with Conventional Design Area Setting Functions

If the density of the initial shape is considered to be 1.0, the rigidity cannot be increased above that of the initial shape.

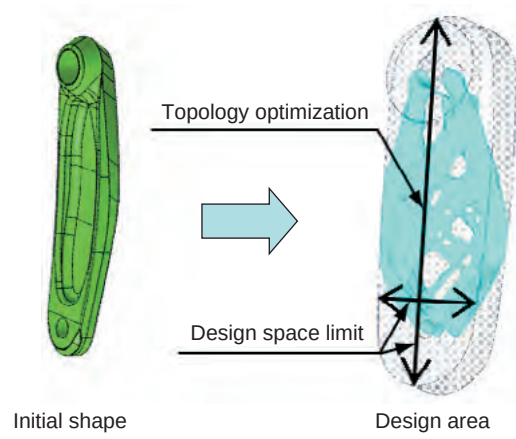


Fig. 4 Set design area

Therefore, the design area is expanded before performing topology optimization, as shown in Fig. 4. However, the number of modeling man-hours is quite high, except for simple shapes, so work to expand the design area does not satisfy the target requirement for no more than 10 man-hours for analysis. In addition, because the engine and auxiliary units are mounted on a motorcycle frame, expanding design area is difficult, and the effect of optimization cannot be obtained easily.

3.4. Topology Optimization Method with an Expanded Concept

To resolve these design area setting issues, the method of setting a high Young's modulus hypothetically, as shown in Eq. (3), was devised as an alternate method to expanding the design area. Here, the Young's modulus is multiplied by k .

$$E_{01} = kE_0 \quad (3)$$

Next, the Young's modulus multiplied by k is substituted into Eq. (2), and Eq. (4) is obtained.

$$kE = \rho^N kE_0 \quad (4)$$

By varying the density from 0.0 to 0.1, the Young's modulus used in the calculation becomes 0.0 to kE_0 , and the rigidity also becomes a multiple of k . For example, doubling the Young's modulus reduces the displacement by 1/2 and makes it possible to increase the initial rigidity hypothetically, as shown in Fig. 5. This creates some margin in the rigidity, allowing use of the topology optimization method. This is the result of increasing the rigidity hypothetically, so it has no meaning as an absolute value. However, the results are output with shading of density, so areas with high density can be judged as locations that require strengthening, and

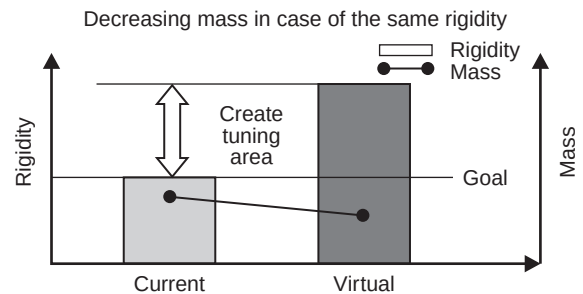


Fig. 5 Increasing virtual rigidity

areas with low density as locations that require weakening. In other words, the results can be understood as the sensitivity relative to the target rigidity. The calculations performed are for topology optimization, but the results are viewed as sensitivity, so this method was named topology sensitivity analysis. Weight reduction can be achieved by strengthening locations with high sensitivity and weakening locations with low sensitivity. Of course, analysis can be also performed using different conditions to efficiently increase the rigidity with a minimum increase in mass, and this is often necessary in intermediate development stages. That is, the mass and rigidity can be optionally set as constraints and objective functions for optimization. Previously, comprehensive countermeasures were considered while judging from stress diagrams, deformation diagrams and other structural analysis results, so it was challenging and took a long time to make judgments. However, the newly developed analysis method enables judgment directly from the density distribution, so judgment errors are reduced and the time is also shortened.

3.5. Comparison of Stress Countermeasure Method and Topology Sensitivity Analysis Method

When investigating an increase in rigidity, the usual method is to reinforce areas with high stress when a load is applied. While this method is not wrong, it is not an appropriate method when considering both rigidity and mass. On the other hand, topology sensitivity analysis is a topology optimization method, that is, an optimization issue, so this method sets an objective function, constraints and design variables as shown below, and then performs optimization by iterative calculations⁽⁴⁾.

Objective function: Min. $\{f(x)\}$

Constraints: $g_i(x) \leq 0; i = 1, 2, \dots, k$

Design variables: $x = (x_1, x_2, \dots, x_n)$

Therefore, this method obtains a more optimum shape than the stress countermeasure method. This means that the topology sensitivity analysis method is more effective than the stress countermeasure method from the viewpoint of both rigidity and mass.

4. Verification

Two examples of verification results for the analysis method explained in section 3.4. are described below.

4.1. Verification of Plate Shape Analysis

This verification example compares the countermeasure method that focuses on high stress areas with the countermeasure method based on topology sensitivity. The calculation conditions for topology sensitivity are as follows.

- (1) Design variables: Entire model
- (2) Constraints: Equivalent rigidity
- (3) Objective function: Minimization of mass

Figure 6 shows the analysis conditions and analysis results for each method. The upper right shows the stress distribution obtained from static analysis, and the lower right shows the topology sensitivity analysis results. Red areas indicate high sensitivity, and blue areas indicate low sensitivity. When adding a patch as a countermeasure, the greatest effect is achieved by applying the patch to the red area for both analysis methods. Figure 7 shows the patch shapes derived from each analysis result. The patch mass is equivalent for comparison purposes. Figure 8 shows the results, and indicates that the newly developed analysis method effectively

increases the rigidity by approximately 20% for the same mass compared to the stress countermeasure method. Although the mass is the same in this example, targeting the same rigidity would allow the mass to be reduced by the amount of increase in rigidity.

4.2. Verification of Frame Part Shape Analysis

This verification example shows the results of confirming the effects of applying reinforcing patches to areas with high, low and medium sensitivity in the analysis results. The calculation conditions are as follows. In this example two constraints are set.

- (1) Design variables: Entire model
- (2) Constraints: Increased torsional rigidity, equivalent lateral rigidity
- (3) Objective function: Minimization of mass

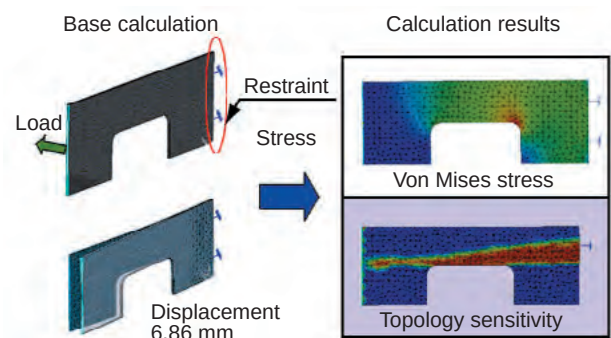


Fig. 6 Verification plate model

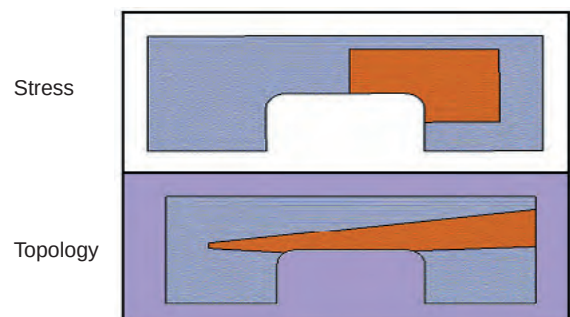


Fig. 7 Comparison of additional patch shapes

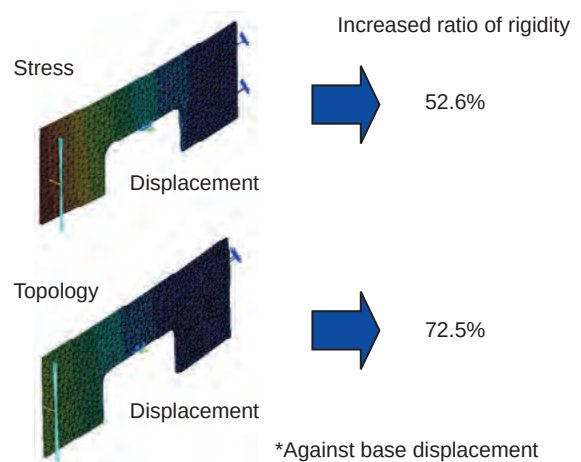


Fig. 8 Comparison of displacement

Figure 9 shows the analysis results, and Fig. 10 shows the patches for the high, medium and low sensitivity areas derived from Fig. 9. The patch mass was set to the same mass for comparison purposes. Table 2 summarizes the results for each patch, and shows that the countermeasure efficiency is greatest in areas with high sensitivity. In addition, applying patches to high sensitivity areas caused the lateral rigidity to decrease slightly. Generally, when one rigidity is increased, the remaining rigidity also increases, but these results showed a drop in lateral rigidity. In terms of sensitivity, however, this is the same as the direction set by the constraints. This example indicates that the newly developed analysis method is also effective for tuning the various rigidities in terms of direction.

5. Examples of Application

5.1. Example of Frame Parts Analysis

This example describes the results of analysis performed to

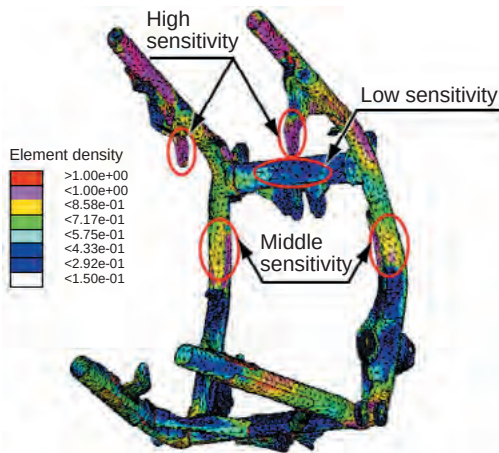


Fig. 9 Result of topology optimization sensitivity for conventional pipe frame

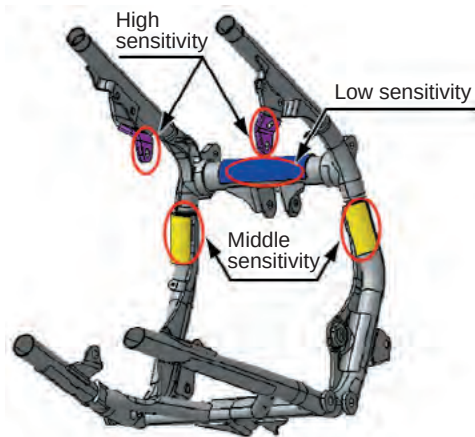


Fig. 10 Example of additional patches

Table 2 Verification effect of each patch

Sensitivity	Low	Middle	High
Torsional rigidity	0.03%	1.21%	4.31%
Lateral rigidity	0.32%	0.84%	-2.01%
Effect	Non	Small	Large

obtain frame rigidity with the aim of minimizing the increase in mass. This analysis used a sports type motorcycle frame, and the calculation conditions are as follows.

- (1) Design variables: Entire model
- (2) Constraints: Increased torsional rigidity, equivalent lateral rigidity
- (3) Objective function: Minimization of mass

Figure 11 shows the analysis results. In these results, red and purple areas indicate locations that require strengthening, and blue areas indicate locations that require weakening. Concretely, measures include tuning the thickness and diameters, adding and deleting ribs, patches, and others. In some cases a change of the skeleton layout is also considered. However, manufacturing technique requirements, strength, cost and other factors are also examined. Previously, parts that required strengthening or weakening were not able to be found out, and there were cases in which misjudgment was made, however, this analysis method is effective for understanding the key locations. Finally, static analysis is performed with the proposed countermeasures input, and the effects are confirmed. Figure 12 shows the example proposed countermeasures. The results confirmed a 10.7% increase in efficiency of the ratio of torsional rigidity to mass.

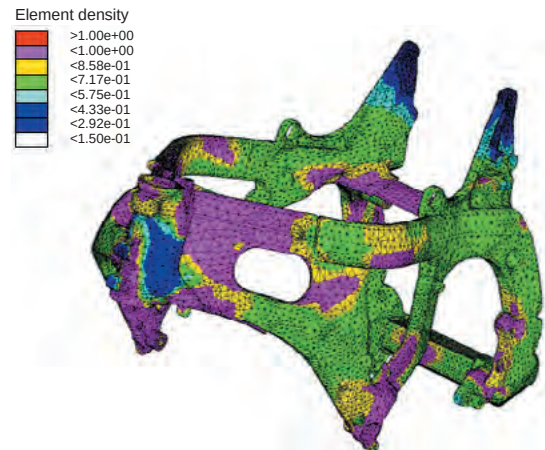
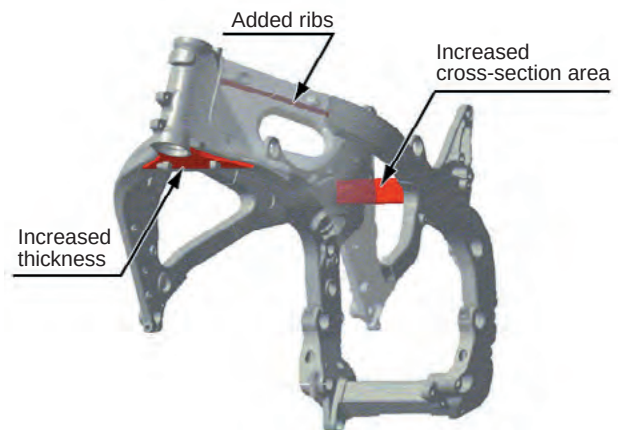


Fig. 11 Result of topology optimization sensitivity for box section frame



	Base	Modified
Torsional stiffness/Mass	100	110.7
Lateral stiffness/Mass	100	102.6

Fig. 12 Modification by using topology optimization sensitivity results

5.2. Other Analysis Examples

The devised topology sensitivity analysis can be applied not just to frames, but to any part that requires rigidity. Figure 13 shows an example of analysis results for a seat bottom plate. Normally, when investigating ribs, the ideal method is to increase the design area and then perform topology optimization using the conventional method. However, depending on the development stage, there are many cases in which the design cannot be changed due to the relation between the layout and the proposed countermeasure support time. The newly developed analysis method is effective in these cases. The design changes shown in Fig. 13 are concretely investigated based on the analysis results within the range over which design changes are possible. For example, ribs are added and thickness is increased in locations that require strengthening, and ribs are taken out and thickness is reduced in locations that require weakening, and others.

Figure 14 shows the analysis man-hours for the results shown in Fig. 11 and Fig. 13. Analysis can be performed within one day (10 hours) in both cases, although this assumes the existence of a static analysis model. There are cases in which the analysis results show little effect for parts that were designed to the optimum shape. However, an approximate 10-20% reduction in weight can normally be achieved by maintaining the same rigidity, or the rigidity can be tuned to suppress the increase in mass.

In addition, the newly developed analysis method is based on existing static analysis models, and analysis can be performed without changing the model, simply by making the settings for topology sensitivity analysis. Therefore, simplification allows designers to use the new method by themselves without requiring the assistance of expert analysis engineers.

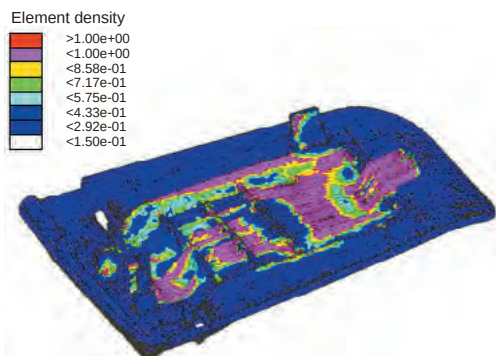


Fig. 13 Result of topology optimization sensitivity for seat bottom

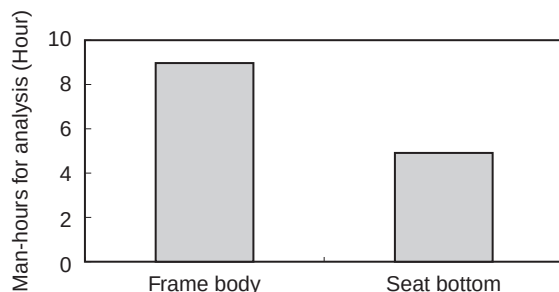


Fig. 14 Total man-hours for analysis

6. Conclusion

It was found that the sensitivity relative to the rigidity of each location of an existing shape can be obtained by increasing the Young's modulus hypothetically instead of expanding the design area in topology optimization.

Construction of this analysis technology yielded the following results.

- (1) Ideas for shapes to reduce mass in consideration of rigidity could be obtained.
- (2) The total man-hours required for analysis were within one day (10 hours), and investigations could be carried out in real time in accordance with the development flow.
- (3) An approximately 10-20% weight reduction was made possible assuming the same rigidity for the applied parts.

The newly developed analysis method allowed target development requirements to be met, and it is thought that this new technology can be applied to the design of not only motorcycle parts but also various mechanical parts. In the future, the authors intend to enhance the functions, simplify operations, and promote faster speed to allow designers to use the newly developed method by themselves.

Acknowledgements

The authors wish to express their deep thanks to all related parties at Altair Engineering, Inc. for their enormous cooperation in the development of this new analysis method.

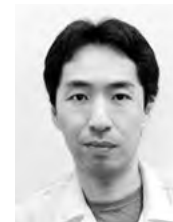
References

- (1) Nishikata, E.: Approach To Weight Reduction of Motorcycles, Journal of Society of Automotive Engineers of Japan, Vol. 59, No. 11, p. 72-76 (2005)
- (2) Fujii, D., Suzuki, K., Ohtsubo, H.: Topology Optimization of Structures Using the Voxel Finite Element Method, Transactions of JSCES, Paper No. 20000010 (2000)
- (3) http://www.jsme.or.jp/dsd/A-TS12-05/minutes/12/Ishii_2004-09-27.pdf
- (4) Kawamo, K., Yokoyama, M., Hasegawa, H.: Fundamentals and Applications of Optimization Theory, CORONA PUBLISHING CO., LTD., p. 2-3 (2000)

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Development of Exhaust Orifice Noise Optimization Technology Using Light-CAE

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ABSTRACT

A prediction optimization technology for exhaust orifice noise and exhaust gas flow rate has been developed as an easy-to-use development support tool for designers.

The accuracy of predicted exhaust orifice noise was increased by using the Lighthill analogy and by correcting for airflow noise using empirical values. Combining this optimization technology with a Light-CAE system enables the relationship between the design parameters and the target values of orifice noise and exhaust gas flow rate to be represented by an approximation function. It also allows designers to carry out analyses in real time. As Light-CAE systems can be easily used in the initial design phase, this technology is positioned as a product development support tool.

1. Introduction

Computer Aided Engineering (CAE) is more and more widely used in various stages of product development. In the development of exhaust systems, specifically, detailed CAE analysis prior to testing a prototype can enable quantitative prediction of a variety of performance parameters, including vibration, strength, silencer attenuation and exhaust gas flow rate. An increasing number of designers now seek a CAE development support tool that enables such performance predictions to be made quickly, easily and with comparative accuracy at the primary design phase. Recent dramatic advances in hardware and software have made the realization of such a tool a possibility.

The authors therefore developed a light CAE system enabling analyses to be conducted easily, even by designers without specialist CAE knowledge. The new system enables designers to conduct real-time analyses, using an optimization technology to represent the relation between target values for exhaust orifice noise and exhaust gas flow rate and design parameters (variables) as approximation functions. This paper discusses the details of this development project.

2. The Need for Light CAE

“Light,” CAE⁽¹⁾ is the opposite of “powered” CAE. Powered CAE is widely used as an alternative to testing. It is used to provide highly accurate quantitative simulations of physical phenomena covering a broad range of testing procedures, and can be used for tests ranging from analyses of body skeletal structure to detailed structural analyses of brackets. However, powered CAE requires extensive computer resources, and normally also necessitates the expenditure of a considerable amount of time on the generation of meshes for detailed three-dimensional models.

There is no clear definition of the boundaries between light CAE and powered CAE, and they are relative to each other. Figure 1 shows a comparison between the two forms of CAE analysis, with the CPU performance required to enable predictions to be made in a day of run-time (number of floating-point operations per second: FLOPS) on the horizontal axis and the degree of freedom in the model at which the required accuracy can be obtained (number of coordinate axes per grid point: DOF) on the vertical axis.

This graph clearly indicates that considerably fewer performance prediction tools are used in light CAE than in powered CAE.

* Automobile R&D Center

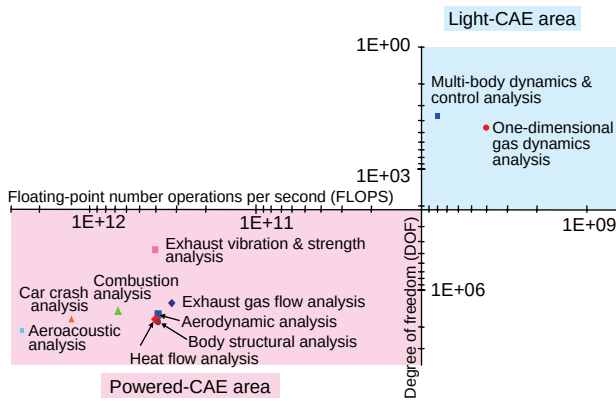


Fig. 1 Map of Light-CAE & powered-CAE analysis tools

The initial stage of development consists almost entirely of primary design, also termed conceptual design. At this stage, component parameters such as shape, dimensions and materials have not yet been defined in technical drawings. Detailed design procedures are carried out downstream from the initial design procedures, after these parameters have been defined. Designers use CAE in the primary design process to enable them to decide specifications for components; the time required for CAE analyses may represent up to 30% of the man-months expended in primary design. This expenditure of time is due to the fact that specifications must be analyzed in detail at this stage to prevent the necessity of changing them repeatedly during the later development process.

This situation called for the development of a light CAE system that was easy for designers to use. A development support tool that would reduce the time required from model production to analysis while enabling sufficient examination of component specifications using the relation between design parameters and target values in real-time.

3. Light CAE System

Figure 2 shows an outline of the light CAE system.

The system is broadly divided into three parts: Exhaust gas dynamic calculation, optimization and real-time analysis. The optimization module is the core of the system; it automatically seeks optimal solutions to enable it to determine optimum functions for the relation between design parameters and exhaust orifice noise and exhaust gas flow rate target values. The one-dimensional gas dynamic calculation module (light CAE) automatically performs repeated calculations, and the optimum function is automatically determined using a predetermined optimization algorithm. The optimum functions are fed into a Visual Basic for Application (VBA) program in the real-time interface that uses Microsoft Excel's macro command function. This allows exhaust orifice noise and exhaust gas flow rate corresponding to multiple design parameters input by designers to be plotted on graphs in real-time.

3.1. One-dimensional Gas Dynamic Calculation

Exhaust gas dynamic calculations based on gas dynamics theory⁽²⁾ are used to simultaneously calculate and analyze exhaust orifice noise and exhaust gas flow rate. This method allows changes in the pressure, velocity and temperature of the moving gas upstream of the exhaust pipe to be calculated for a series of time

steps. These variables are used as input conditions for time history calculations using the Navier-Stokes equation and governing equations for conservation of mass and energy. A time history calculation of the pressure, flow rate and temperature of the gas in the exhaust pipe is generated.

The authors used the one-dimensional gas dynamics CAE WAVE produced by Ricardo as a light CAE tool for these calculations. WAVE provides a model of exhaust noise source upstream from the exhaust pipe, generating the dynamic motion of the gas from engine specifications (bore/stroke, compression ratio, valve dynamic lift, etc.), and allows upstream noise to be calculated with downstream orifice noise and exhaust gas flow rate. However, exhaust orifice noise is made up of pulsation noise and flow noise, and its sources are the compression wave at the end of the exhaust pipe and changes in the pressure of the vortex rings in the exhaust jet⁽³⁾. Flow noise predictions cannot be made using one-dimensional gas dynamics CAE. The effectiveness of corrective calculations using Lighthill analogy equations (to be discussed below) in increasing the accuracy of predictions of orifice noise made by means of CAE using one-dimensional gas dynamics software was therefore attempted⁽⁴⁾.

3.1.1. Calculation of pulsation noise

For pulsation noise calculations, the sound power generated by pressure at the end of the exhaust pipe is attenuated in proportion to the square of the distance from the end of the pipe. The sound pressure up to a specified measurement point (a microphone) is given by Eq. (1)⁽⁵⁾.

$$P(t) = \frac{\rho}{4\pi R} \frac{dS(t - R/c)}{dt} \quad (1)$$

P : Sound pressure of pulsation noise (Pa)

S : Volumetric flow rate (m^3/s)

R : Distance from sound source to microphone (m)

c : Speed of sound (m/s)

t : Time (s)

ρ : Air density (kg/m^3)

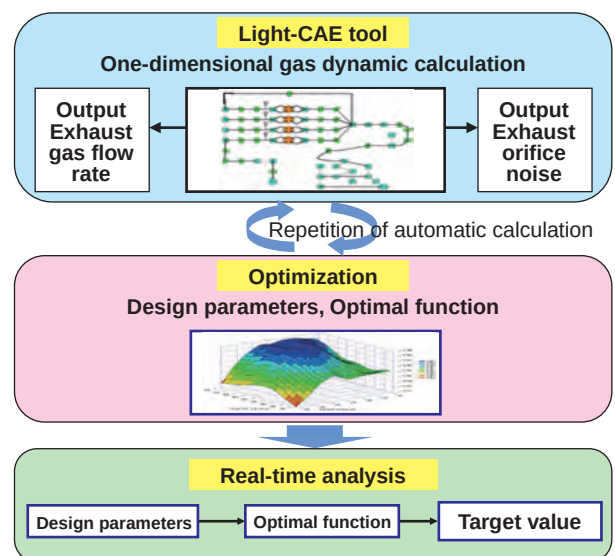


Fig. 2 Flow diagram of Light-CAE system

Figure 3 shows a comparison between measurement results for exhaust orifice noise at a microphone and calculations using Eq. (1). At high engine speeds (3000 rpm and above), the calculation results fall below the measurement results.

This is considered to be because the calculation results represent only the pulsation noise component, without factoring in the flow noise component.

3.1.2. Calculation of flow noise

In the case of exhaust orifice noise, Lighthill analogy equations⁽⁶⁾ enable the sound intensity of flow noise, Ei , to be expressed by Eq. (2).

$$Ei = \frac{\rho}{r^2 c^3} L^2 U^6 \quad (2)$$

Ei : Sound intensity of flow noise (W/m²)

ρ : Density (kg/m³)

r : Distance from sound source (m)

c : Speed of sound (m/s)

L : Typical length of sound source (m)

U : Flow velocity (m/s)

Here, L refers to the typical length of the source of flow noise, i.e. the distance from the generation of the main vortex rings to the point at which they become attenuated.

The relation between sound intensity, Ei , and the sound pressure level of flow noise, Ep , is given by Eq. (3).

$$Ep = \sqrt{Ei \times \rho \times c} \quad (3)$$

Ep : Sound pressure level of flow noise (Pa)

Based on the results of tests using a high-speed pulsation flow generator⁽⁷⁾, Lighthill analogy equations (2) and (3) were used to determine the relationship between flow velocity (U) and the length of a sound source generating a specific sound pressure of flow noise at the microphone (Ep) (Fig. 4).

Figure 4 shows that U becomes greater as L decreases. This is caused by the increase in the high frequency component as the spaces between the multiple vortex rings, that are the source of flow noise, became narrow. In addition, the results indicate that L is largely unaffected by the pulsation cycle, and is almost entirely dependent on the flow velocity.

In other words, it is possible to predict the sound pressure of flow noise when U and L are known.

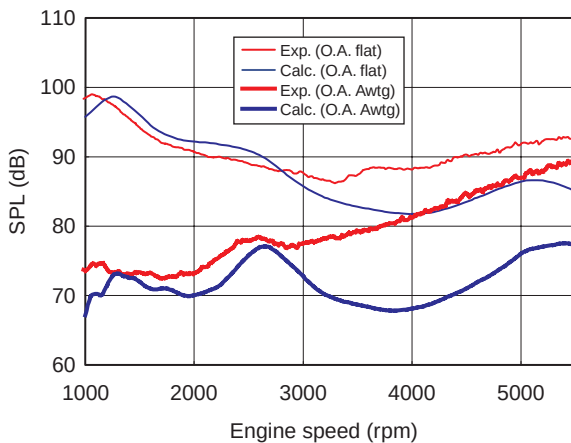


Fig. 3 Comparison of orifice noise level in combustion

3.1.3. Calculation of exhaust orifice noise

Figure 5 shows a comparison of calculation and measurement results for the sound pressure level (pulsation noise + flow noise) close to the exhaust orifice (microphone position: 50 cm/45 deg behind the exhaust orifice) of a 4-cylinder, 1.3L internal combustion engine under fast acceleration. Results are plotted against engine speed on the horizontal axis. Calculation results were obtained by inputting L and U into Eqs. (2) and (3).

Figure 5 indicates that the accuracy of predictions of exhaust orifice noise is increased by calculating both pulsation noise and flow noise.

3.1.4. Calculation of exhaust gas flow rate

The flow rate (Q) in the exhaust system can be quantitatively determined by positioning a flow tester that uses a blower fan to generate a laminar flow upstream. The flow tester is used to find the flow rate when the difference between the pressure upstream and the pressure of the flow downstream escaping to the atmosphere is equivalent to a fixed water head pressure (600 mmAq). The exhaust system model used in gas dynamic calculations (excluding the exhaust noise source model) and the light CAE tool were used to calculate the results of such a test. Figure 6 shows a comparison between calculation results and the experimental results.

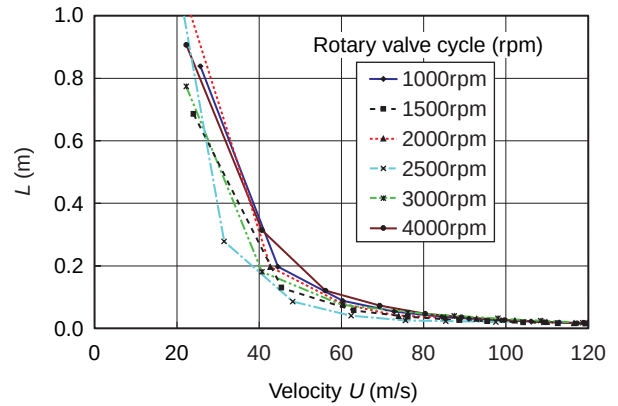


Fig. 4 Relationship between L (noise source length) and flow velocity

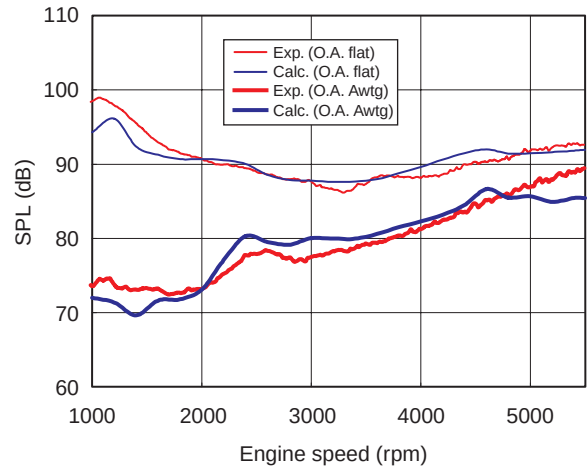


Fig. 5 Comparison of orifice noise level in combustion engine

3.2. Optimization Technology

Optimization in this system refers to the expression of the relation between the design parameters, i.e. the required shapes and dimensions when the designers formulate the design drawings (length and diameter of pipe, length and diameter of silencer, etc) and the target values for exhaust orifice noise and exhaust flow rate as optimum functions. Response surface models (RSM)⁽⁸⁾ are used to determine these optimum functions.

3.2.1. Response surface models

Response surface models are statistical approximation models based on regressive analysis and used in engineering applications. The nth optimum function is estimated using the least squares method to minimize the error in target values against design parameters.

To enable response surface models to be used, the system uses iSIGHT, produced by Engineous, to control the light CAE and automatically search for optimum functions.

Figure 7 shows the approximation function for the relationship between exhaust orifice noise and the design parameter pipe length as determined using an RSM. Figure 8 shows the approximation function for the relationship between exhaust gas flow rate and the design parameter pipe diameter as determined using an RSM. The points marked with squares in both figures show results obtained using light CAE in isolation.

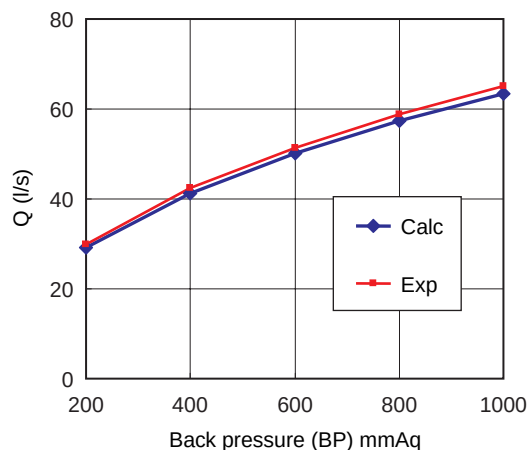


Fig. 6 Calculated and experimental accuracy of flow tester

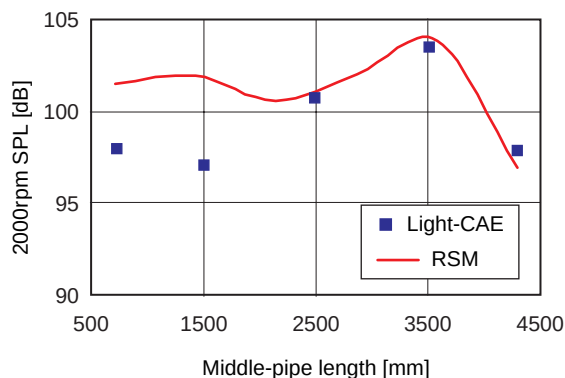


Fig. 7 Relationship between RSM approximation error and Light-CAE (Orifice noise)

3.2.2. Method of enhancement of approximation accuracy

Figures 7 and 8 show approximation error on a regression curve against results determined using CAE in isolation. Exhaust orifice noise shows strong nonlinearity against many design parameters, and significant approximation errors are often observed between regression curves and light CAE in the case of this parameter.

The following countermeasures are available in this case:

- (1) Increase the order of polynomial equations for the regression curve
- (2) Increase the number of searches for learning data
- (3) Restrict the range of design parameters

The response used in the present research is discussed below.

Design parameters in the form of two different exhaust pipe lengths were established as horizontal and vertical axes, and searching points representing searches for learning data by light CAE in these design parameter spaces were plotted between them (Fig. 9).

As this graph shows, using light CAE to search for learning data across the entire design parameter space means increasing the number of calculations required for optimization, and is therefore impracticable. The algorithm shown in Fig. 10 was used to resolve these issues.

This algorithm does away with the requirement to search for study points across the entire design parameter space. Instead, it increases the number of searches for learning data across the space

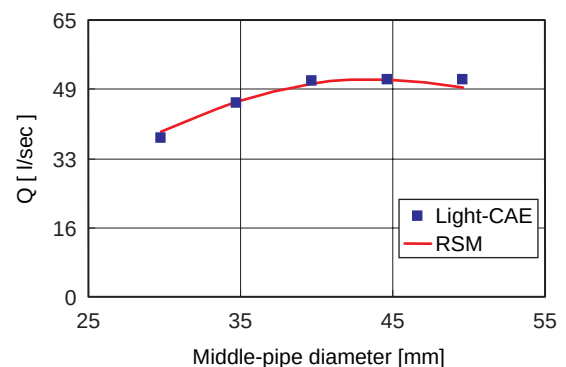


Fig. 8 Relationship between RSM approximation error and Light-CAE (Exhaust gas flow rate)

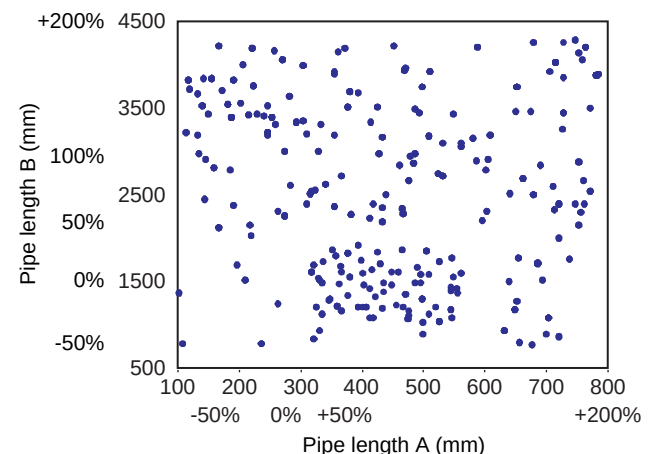


Fig. 9 Search map of optimal study data (Before enhancement)

of important design parameters to a range $\pm 50\%$ from an indicated median point (Fig. 11).

Figure 12 shows a comparison between approximation functions for the relationship between the design parameter pipe length and exhaust orifice noise, as determined by an RSM following enhancement of approximation accuracy, and the relationship determined by light CAE in isolation.

This graph shows a reduction in the degree of error between approximation functions obtained using an RSM and results obtained using light CAE than that shown in Fig. 7.

3.2.3. Method of real-time analysis

The optimum functions expressing the relationship between design parameters and target values for exhaust orifice noise and exhaust gas flow rate obtained using RSM are high degree polynomials made up of enormous amounts of coefficient data.

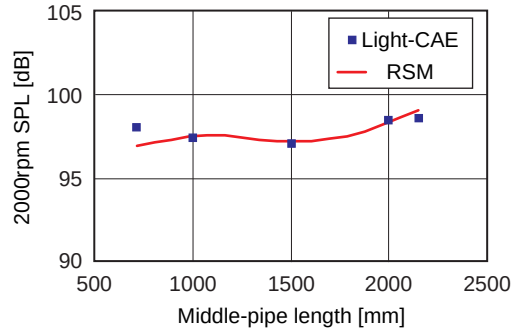


Fig. 12 Relationship between RSM approximation error and Light-CAE (Orifice noise)

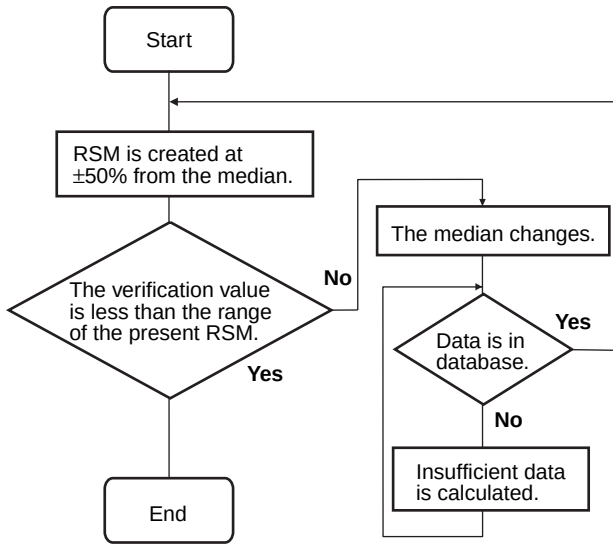


Fig. 10 Enhancement algorithm in RSM approximation accuracy

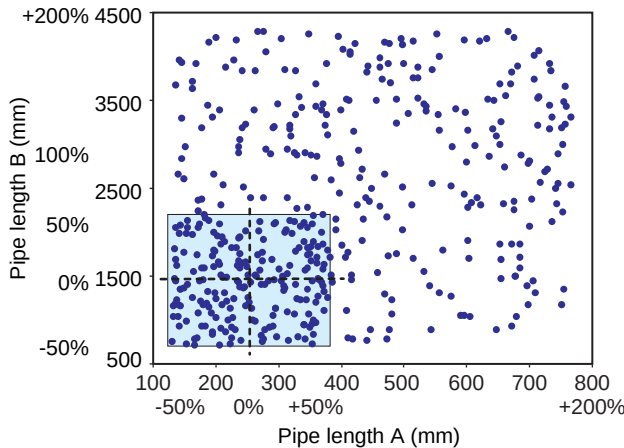


Fig. 11 Search map of optimal study data (After enhancement)

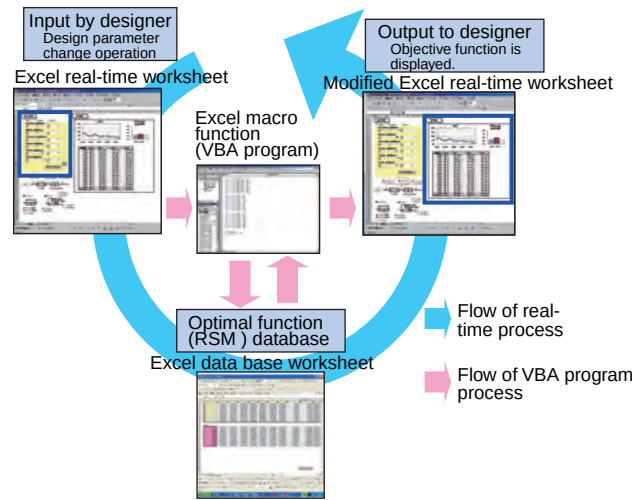


Fig. 13 Block diagram of real-time process

These optimum functions can be used to instantaneously and continuously display the relationship between design parameters and target values. Figure 13 shows a block diagram of real-time analysis.

This process enables designers to analyze in real-time whether input design parameters enable the achievement of specific target values.

4. Examples of Application as a Development Support Tool

The technology discussed in the preceding sections was developed as a practical development support tool for designers. The following points were stressed in this development:

- (1) Simplicity of use, such that designers did not require specialized knowledge of CAE operation.
- (2) Reduction of pre-processing time by enabling designers to choose from models pre-recorded as parametric models.
- (3) Post processing of results using Microsoft Excel, and presentation in a diagrammatic form that is easy for designers to understand.

4.1. Input Interface

To provide a development support tool that was easy for designers to use and that did not announce itself as a light CAE tool, a specialized interface was developed using Sun Microsystems' Java as the software language (Fig. 14). Figure 15 shows an example of the menu sheet, in this case for the basic layout of an exhaust system. The diameters, lengths, etc. of pipes in the exhaust system (composed of the engine, catalyzer, silencer, etc.) can be freely input as parametric numerical values, enabling design parameters to be set.

Figure 16 shows the silencer menu sheet. In this case again, design parameters can be set by freely inputting the silencer diameter, total length, length and diameter of inner pipes as parametric numerical values.

When all the conditions required for calculations have been input, design parameters are displayed as a pull-down menu. This enables designers to set the range of the parameters, median point and upper and lower limits, following which the calculations are executed (Fig. 17). E-mail addresses can also be input for forwarding of calculation results.

4.2. Real-time Analysis Interface

A VBA program using Excel's macro command function is used to display the calculation results. In the worksheet shown in Fig. 18, design parameters are shown as the slide bars at the left, exhaust

orifice noise is shown as the line graph in the upper middle, and exhaust gas flow rate is shown as the bar graph at the upper right.

As designers input values in the design parameter slide bars, they are able to view real-time changes in the graphs of exhaust orifice noise and exhaust gas flow. This enables them to easily analyze the sensitivity and prepotency of the design parameters against target values.

This real-time analysis is enabled by the ability to determine optimum functions for the relationship between parameters and target values provided by the combination of light CAE and optimization technology.

5. Conclusion

A light CAE system that is easy for designers to use in the initial stage of design has been developed.

- (1) The accuracy of light CAE in predicting exhaust orifice noise was enhanced by correcting the associated flow noise using test results based on Lighthill analogy.
- (2) Optimization technology and light CAE were combined to allow the relationship between design parameters and target values for exhaust orifice noise and gas flow to be expressed as approximation functions, enabling designers to execute analyses in real-time.

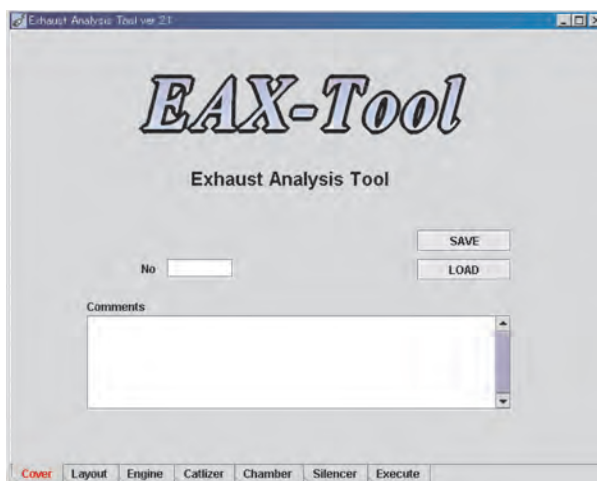


Fig. 14 Opening worksheet

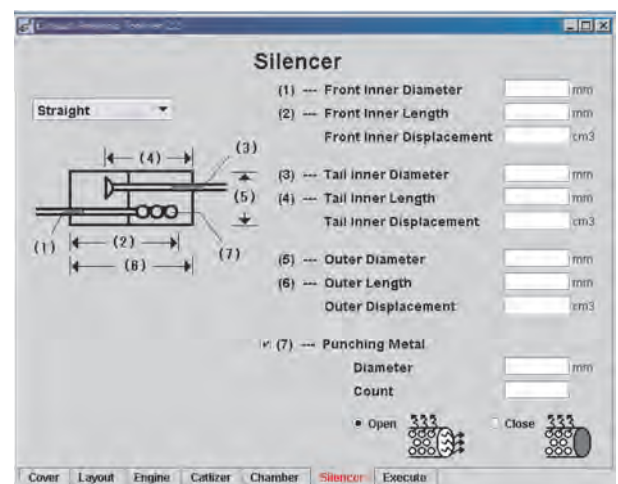


Fig. 16 Silencer system layout menu

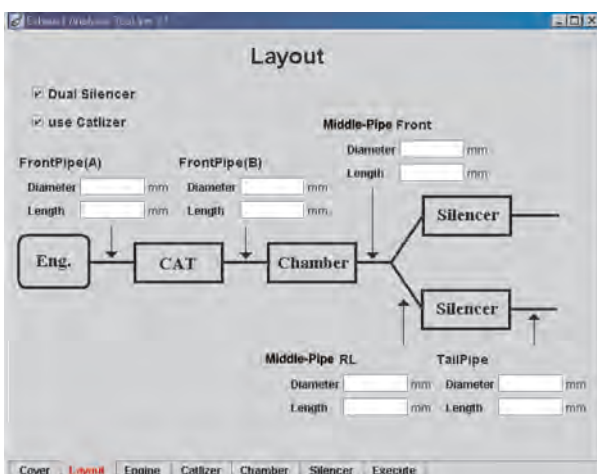


Fig. 15 Exhaust system layout menu

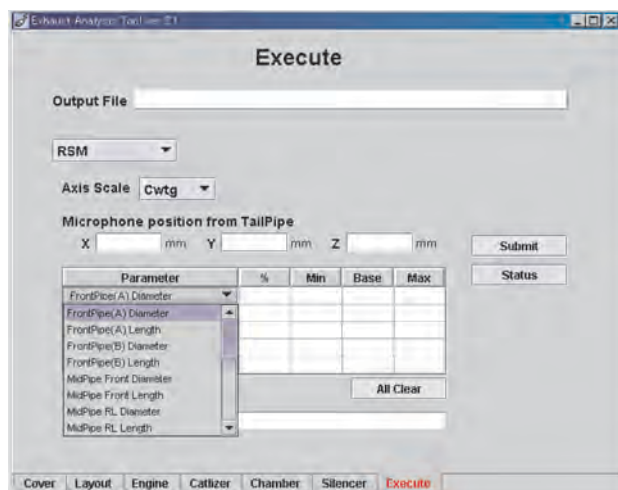


Fig. 17 Execution worksheet

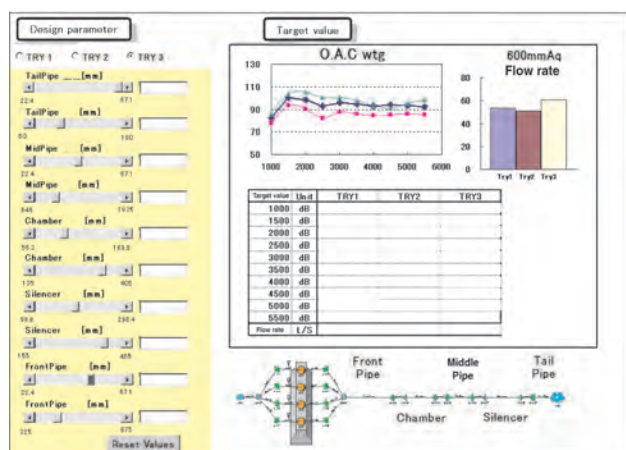


Fig. 18 Real-time analysis worksheet

- (3) The technologies discussed in (1) and (2) above were packaged as a light CAE system (calculation time: less than 1/10 that of power CAE) to provide designers with a development support tool.

References

- (1) Kuriyama, T.: Strategic CAE to Shorten Development Time - Power CAE and Light CAE, Journal of Society of Automotive Engineers of Japan, Vol. 51, No. 4, p. 79-85 (1997)
- (2) Shapiro, A. H.: The Dynamics and Thermodynamics of Compressible Fluid Flow, Vol. 1, The Ronald Press Company (1954)
- (3) Sakurai, M.: Numerical Simulation of Exhaust Pulsating Flow & Orifice Noise, Review of Automotive Engineering, Vol. 27, No. 1, p. 151-154 (2006)
- (4) Sakurai, M., Kasajima, Y.: Computational Simulation of Exhaust Pulsating Flow & Orifice Noise (Third Report), JSAE Annual Congress Proceedings, No. 175-06, p. 10-14 (2006)
- (5) Ricardo-WAVE User's Manual
- (6) Goldstein, M. E.: Aeroacoustics, p. 311 (1991)
- (7) Sakurai, M., Tanaka, K., Kasajima, Y.: Clarification of Mechanism of Exhaust Orifice Noise via Exhaust Pulsating Flow Simulation, Honda R&D Technical Review, Vol. 17, No. 2, p. 77-83
- (8) Box, G.E.P., Wilson, K. B.: On the Experimental Attainment of Optimum Conditions, Journal of the Royal Statistical Soc., B-13, p. 1-45 (1951)

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Study of Base Number Estimation Method for Engine Oil Deterioration Monitoring System

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ABSTRACT

In this research project, technology for estimating the base number (BN) deterioration, used as an index for judging engine oil deterioration, was developed. A detailed investigation using an NOx bubbling unit was carried out, and as a result it was found that the speed at which the BN falls is determined mainly by heat and NOx. In addition, a detailed analysis was carried out, and it was found that the reduction speed can be expressed using the differential rate law. A study of the possibility of estimating reduction of the BN in an actual engine was carried out using a BN deterioration equation. The results of this study showed that it was possible to adequately estimate the BN deterioration. This indicated that it is possible to estimate the reduction of the BN on-board by the engine control unit (ECU).

1. Introduction

Engine oil cleans, cools and inhibits rust and corrosion, in addition to its functions as a lubricant for various engine parts, and a sealant for the combustion gas. Although all of these functions are important, reduction of its detergent properties (detergency) in particular can result in sludge and deposits inside the engine, which can lead to serious issues such as oil circuit blockages. Engine oil additives that are closely related to detergency include the metallic detergents represented by salicylates. These detergents offer resistance to high temperatures and are said to be indispensable for minimizing deposits on hot parts of the engine, such as those near the piston and lubricated parts of the turbocharger^{(1),(3)}. These detergents also protect the engine by neutralizing organic acids, generated as a result of fuel combustion, and the nitric acid that is

generated as a byproduct of NOx in the blow-by gas⁽²⁾. The base number (BN) indicates the amount of basic substances contained in engine oil, and is a representative index of the amount of residual basic metallic detergent. The BN is also used as an index of the oil's ability to neutralize acid, and is an important indicator of the degree of deterioration of engine oil. However, even if the BN is used as an index of the useable life of the oil, special equipment is necessary to quantitatively measure the BN. For this reason, it is hard for an automobile user to know when to replace the oil using the BN as an index. The main aim of this research was to devise a method of estimating the degree of deterioration of the BN by utilizing data acquired from an engine sensor, such as the engine water temperature sensor, without having to actually measure the BN. Another aim was to utilize this data to study the feasibility of using an on-board BN deterioration detection system.

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2. Relationship between BN and Oil Deterioration

2.1. Relationship between BN and High Temperature Deposits

The BN is often used as an index of oil deterioration. In this research, a validation was carried out concerning the effect of BN on minimizing deposits, using the hot tube test (JPI-5S-55-99) and an engine test. The hot tube test involves passing a mixture of oil and air through a heated glass tube, and evaluating the deposits formed inside the glass tube. Figure 1 shows an outline diagram of the hot tube test. Its effectiveness in minimizing deposits was evaluated by investigating the relationship between the deposit generation temperature and the BN. In the hot tube test, the deposit generation temperature was defined as follows: First, the oil under evaluation was tested at various temperatures; the deposit amounts at each temperature were then measured. The amount of deposits versus temperature was plotted, and the calibration curve method was used to estimate the deposit generation temperature. Also a test using an actual engine with fresh oil was conducted. Mid-way through the test the engine was disassembled and the deposits on the piston were then visually confirmed. The BN at this time was

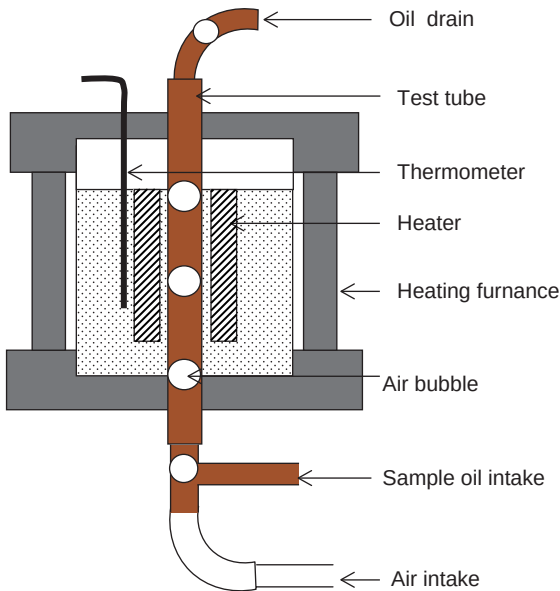


Fig. 1 Hot tube test device

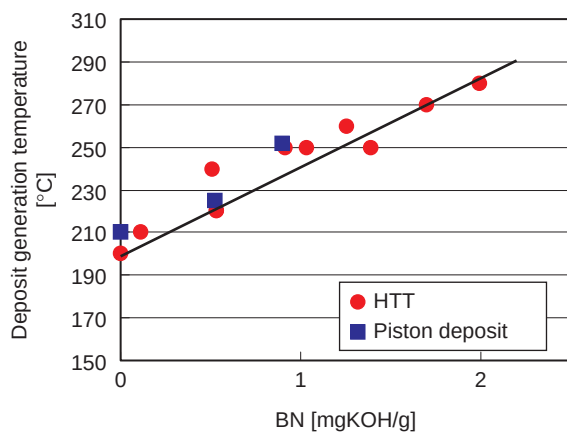


Fig. 2 Relationship between BN and deposit generation temperature

Table 1 Sample lists for deposit temperature preparation

Sample no.	Mileage [km]	BN [mgKOH/g]	Oil	Test
1	4337	2	Honda genuine	Hot tube test
2	4470	1.39	↑	↑
3	5000	1.70	↑	↑
4	6892	0.53	↑	↑
5	7985	0.11	↑	↑
6	9156	2.3	↑	↑
7	10301	3.2	↑	↑
8	12274	3.4	↑	↑
9	13585	1.7	↑	↑
10	16175	1.7	↑	↑
11	18616	1.5	↑	↑
12	25032	0	↑	↑
13	—	0	↑	Engine
14	—	0.52	↑	↑
15	—	0.9	↑	↑

measured to examine the relationship between the deposits and piston temperature. A previously measured value was used as the piston temperature. The oil used is shown in Table 1.

As shown in Fig. 2, a strong correlation between the BN and the deposit generation temperature was seen. A metallic detergent blended with oil basically has the same structure as that of alkaline soap. When the alkali concentration of alkaline soap falls, its effectiveness weakens. It is thought that the detergency of engine oil also weakens as a result of the reduction of the BN, and that this is the reason the deposit generation temperature falls when the BN is reduced. Thus, it can be seen that there is a clear relationship between the BN and high temperature deposits. Generally, the usable range of the BN is between 0.5 and 1.0 mgKOH/g, although this will vary depending on the performance of the engine oil. The present research results indicate that the deposit generation temperature within the usable range of the BN is around 230 °C to 250 °C. This coincides roughly with the maximum temperature of the engine parts which need lubricating oil, and can be considered appropriate as a limit value.

2.2. Relationship between the BN and Metal Wear

Figure 3 shows the relationship between the BN and the metal content of the oil. All of the oil used in the analysis was collected

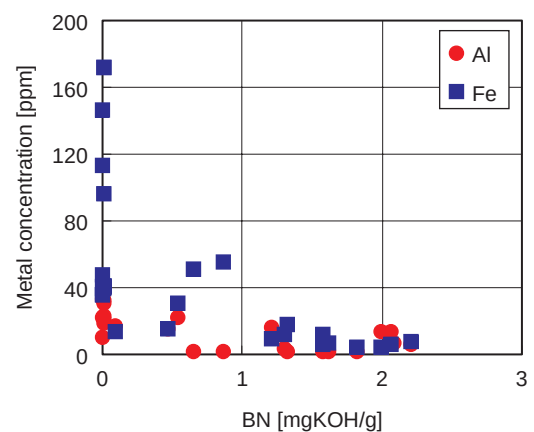


Fig. 3 Relationship between BN and metal concentration

Table 2 Sample lists for metal concentration research

Sample no.	Mileage [km]	BN [mgKOH/g]
1	1931	1.58
2	2000	1.58
3	2000	2.06
4	2982	1.82
5	3000	1.99
6	4000	1.21
7	4280	2.21
8	4532	1.30
9	5000	0.54
10	5038	1.62
11	6001	1.32
12	6371	2.08
13	7833	0.00
14	7895	0.09
15	8983	0.65
16	9182	0.00
17	9200	0.47
18	10552	0.01
19	10901	0.01
20	13937	0.01
21	15000	0.00
22	15648	0.01
23	15825	0.86
24	20000	0.00
25	24630	0.01

from users, and had been used for different travel distances and travel times, and under different conditions. The oil used in the analysis is shown in Table 2. It can be seen that when the BN falls, the metal concentration in the oil tends to rise. This is thought to be that when the BN falls, its ability to neutralize acid is reduced, and as the oil becomes highly acidic, metal dissolves in it. It can thus be said that the BN is an important index of oil deterioration from the viewpoint of metal corrosion and wear.

3. Derivation of an Equation for Calculating the BN Deterioration Rate Based on Empirical Values

3.1. Cause of Deterioration of the BN

Of the factors responsible for the reduction of the BN, the first is the neutralization reaction due to acidic matter generated by oxidation in the oil itself. The second is an external factor such as NO_x, SO_x or an organic acid generated during combustion when the engine is running. It is thought that these acidic matters mix with the oil^{(4),(5)} and react with basic substances such as detergents, causing the BN to fall. The third factor is the occurrence of pyrolysis at high-temperature parts, such as the piston, which causes the BN to fall. To analyze the effect of each external factor, a deterioration test of the engine oil was carried out using an NO_x bubbling test. Figure 4 shows an outline of the NO_x bubbling test apparatus. NO_x gas, fuel composition and water content were used as the external deterioration factors. The test conditions are shown in Table 3. NO gas was used to represent NO_x as investigations of NO_x composition in the crankcase showed that it consisted mostly of NO. NO gas was mixed with air and the mixture blown into the oil. The air was first blown through water to adequately moisturize it, and then mixed with the NO gas. The fuel component used was the residue formed by distilling white gasoline at 150 °C. This procedure collects relatively large molecules so as to simulate the

Table 3 NO_x bubbling test conditions

	Test condition
NO _x (ppm)	0 to 2776
Water	Steam with air
White gas (wt%)	5
Oil temp (°C)	25, 80, 150
Oil 1	Honda genuine-1
Oil 2	Honda genuine-2
Oil 3	Special

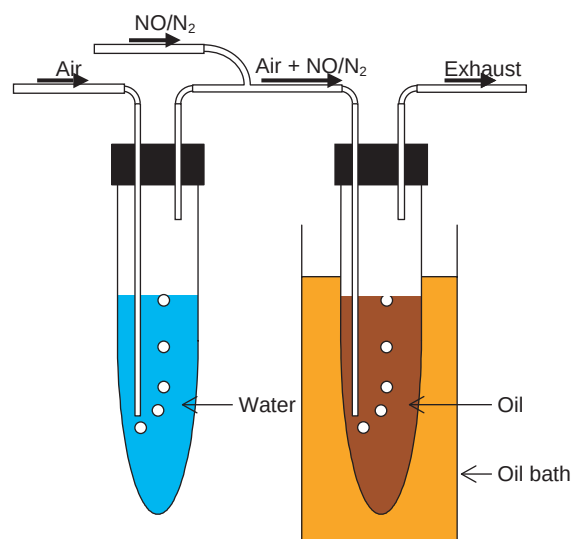
actual unburnt fuel, as well as to oxidize the fuel to a certain extent to simulate the condition of unburnt fuel. The SO_x component depends greatly upon the actual fuel, and in this research the use of sulfur-free fuel was assumed so SO_x was not taken into consideration. The method of measuring the BN is described under JIS and ASTM guidelines, and all BNs used in this report were measured using the hydrochloric acid method based on the method described in JIS K 2501.

3.2. Deterioration of the BN due to Heat

The following are conceivable causes of a reduction in the BN when heat alone is applied to the engine oil.

- (1) Alteration and pyrolysis of the basic additive itself
- (2) A reaction that neutralizes the carboxylic acid generated by the oxidation of the engine oil itself
- (3) Generation of acid due to the oxidation of the engine oil additives (e.g. generation of sulfuric acid due to the oxidation of additives containing sulfur)

NO_x and other causes of deterioration were removed from the engine oil by passing it through an NO_x bubbling apparatus, and the BN deterioration rate when only heat and air blown through the engine oil was investigated. The BN deterioration rate was analyzed, and an attempt was made to indicate the rate as a BN deterioration rate equation. The differential method⁽⁶⁾ was used as the analysis method. As shown in Fig. 5, the order of reaction was estimated to be approximately 2, based on analysis. Here, if it is assumed that the order of reaction with respect to the BN is 2, a deterioration rate equation is defined as Eq. (1).

Fig. 4 NO_x bubbling device

$$-\frac{d[BN]_{Heat}}{dt} = k_1[BN]^2 \quad (1)$$

When the temperature dependence of the BN deterioration rate is measured, and an Arrhenius plot of the deterioration rate coefficient k_1 is made, a linear relationship is obtained as shown in Fig. 6, so it is considered that the BN deterioration rate is that of an Arrhenius type.

3.3. Deterioration of the BN by NOx

As shown in Fig. 7, when NOx is blown into the engine oil, the BN deterioration rate increases. Because the BN deteriorates even when NOx is not blown through the engine oil, it is natural to assume that the deterioration rate term due to NOx can be treated as an independent term in Eq. (1). Here, if the deterioration rate term due to NOx is defined as A_{nox} , the following equation can be assumed.

$$-\frac{d[BN]_{Heat,NOx}}{dt} = k_1[BN]^2 + A_{nox} \quad (2)$$

The second term of Eq. (2) can be obtained empirically by performing tests with and without NOx, respectively, and obtaining the difference between the resulting deterioration rates. Figure 8 shows the relationship between A_{nox} and BN. As a result of these tests, it was found that A_{nox} increases proportionally to the BN. Here, it can be seen that the inclination of A_{nox} has increased due to

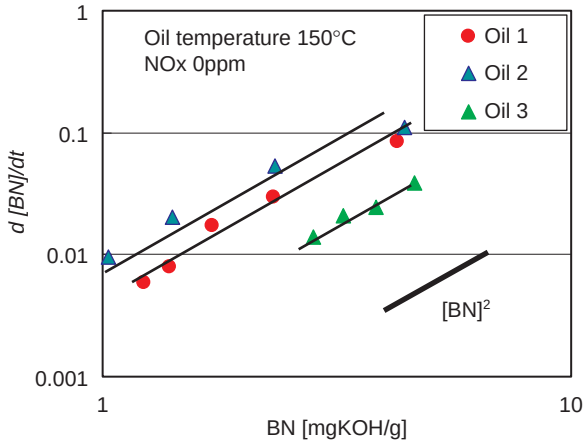


Fig. 5 BN dependence of BN deterioration velocity

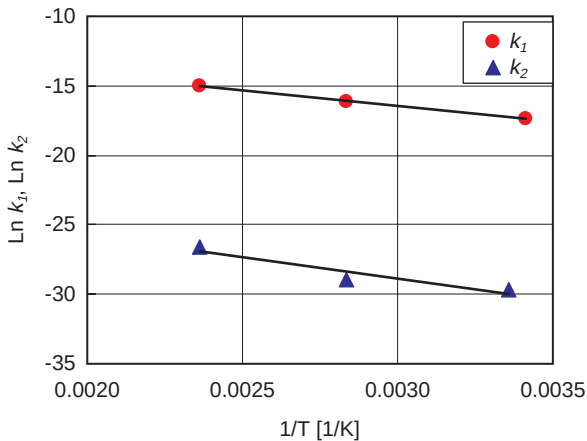


Fig. 6 Arrhenius plot for reaction rate constant k_1 , k_2

the NOx concentration. Figure 9 shows the magnitude of the inclination versus the NOx concentration. It was found that the inclination of A_{nox} is proportional to the square of NOx. In other words, it can be said that A_{nox} is proportional to the BN, and is also proportional to the square of NOx. From the above relationship, it can be considered that if the proportional constant is defined as k_2 , A_{nox} can be represented by Eq. (3).

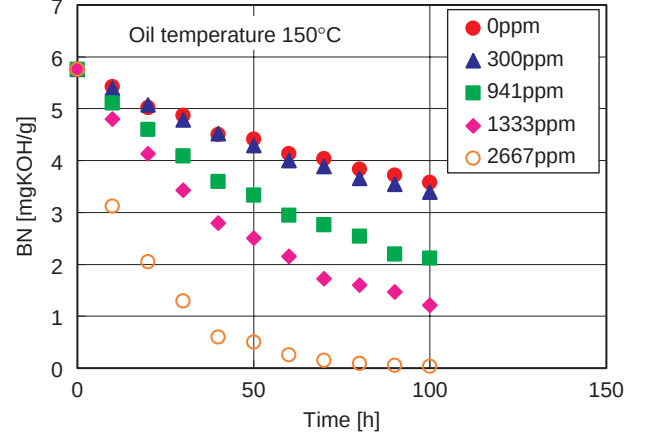


Fig. 7 NOx-dependence of BN degradation rate

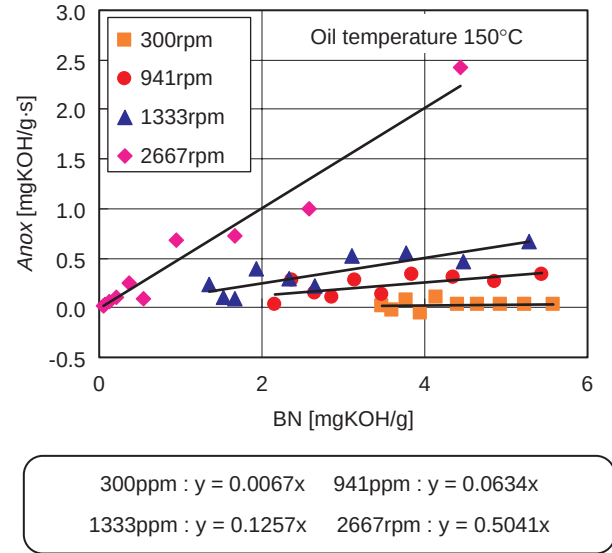


Fig. 8 BN dependence of A_{nox}

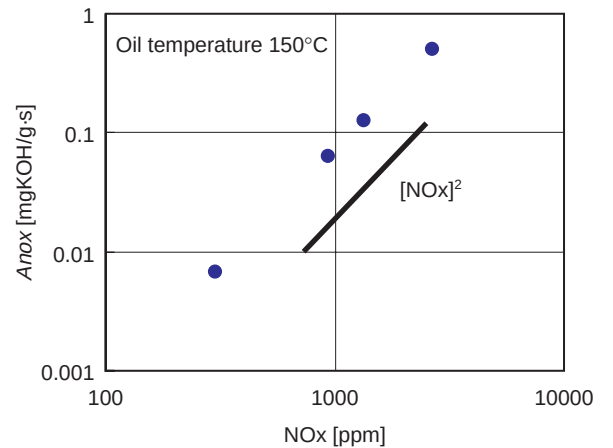


Fig. 9 NOx dependence of A_{nox}

$$A_{nox} = k_2[BN][NOx]^2 \quad (3)$$

In addition, it was found that A_{nox} is also dependant on temperature. In other words, the proportional constant k_2 depends upon temperature, and similar to k_1 , an Arrhenius plot of k_2 produced a linear relationship, as shown in Fig. 5.

3.4. Deterioration of the BN due to the Fuel Composition and Water Content

It is said that generally oil deterioration is greatly affected by unburnt fuel and the water content. It is well known that the existence of water in the engine oil accelerates the dissolution of NOx ions in the oil^{(4),(5)}, and it is thought that it also has a large effect on the reduction of the BN as well. When the oil temperature is low, unburnt fuel is liable to become mixed with the oil⁽⁷⁾. As a result, it is thought that organic acid due to unburnt fuel is liable to be produced, affecting the reduction of the BN. To validate this assumption, the effect of water content and fuel were studied in this research. As a result, it was found that the effects of water content and fuel were either nil or negligible, as shown in Fig. 10. It is thought that the fuel oxidized becomes an organic acid. The organic acid reacts with the base component in the oil to generate a salt, and is neutralized. Generally, an organic acid is weakly acidic compared to hydrochloric acid, for example, and in the BN measurement using the hydrochloric acid method, the salt itself is titrated. Consequently, it is thought that almost none of the organic acid contributes to the deterioration of the BN. Previously, regarding the effect of water content, the existence of water resulted in the precipitation of needle crystals from an Mg-type detergent, causing the BN to drop markedly. This issue has been solved as a result of progress made in additive technology. Also, it was thought that the addition of water would cause the amount of NOx in the oil to increase, which in turn would cause the BN to fall, but no such effect was found in tests performed during this research. There is a report stating that if there is a certain amount of water in the oil, the amount of NOx that dissolves in oil will become a rate controlling factor⁽⁴⁾; it is conceivable that this was an influencing factor.

From the results of the above tests, it is possible to express the BN deterioration rate using the following general equation (k_1 and k_2 are Arrhenius type deterioration rate coefficients).

$$-\frac{d[BN]}{dt} = k_1[BN]^2 + k_2[BN][NOx]^2 \quad (4)$$

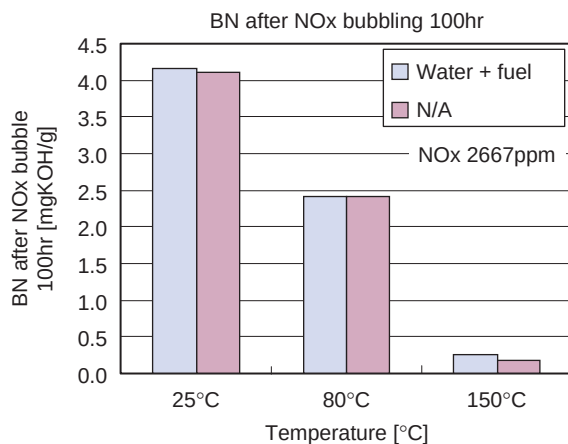


Fig. 10 Water and fuel influence on BN degradation

4. Validation of BN Deterioration in an Actual Vehicle

4.1. Deterioration Rate Equation for Validation using an Actual Vehicle

Using deterioration rate Eq. (4) for the BN obtained from the results of oil tests, a validation test was carried out to see if it was also possible to indicate the BN deterioration due to engine operation. The oil temperature and NOx, which constitute necessary information for carrying out a validation on a vehicle, were measured or estimated using the following method.

(1) Oil temperature

The oil temperature was measured at the oil drain bolt using a thermocouple.

(2) NOx

The concentration of NOx inside the crankcase, which is considered to come into contact with the oil most frequently, was used as a reference for the NOx. Because it is difficult to measure NOx inside the crankshaft of an actual engine using an on-board device, the NOx concentration was estimated from the driving conditions.

When Eq. (4) is applied to an actual engine, the values of contact efficiency between air or NOx and oil are dependent on whether an oil test or an engine test is carried out, so it was difficult to conceive that the deterioration rate coefficient obtained from the oil test could be used as is. Accordingly, the oil was allowed to deteriorate in an actual engine, and the contact efficiency coefficient η was applied to Eq. (4) so that it matched the BN data obtained in this case.

$$\frac{d[BN]}{dt} = \eta \times (k_1[BN]^2 + k_2[BN][NOx]^2) \quad (5)$$

4.2. Results of Validation using an Actual Vehicle, and Estimated Accuracy

Honda genuine oil was used for the vehicle validation. Eq. (5) was used to calculate the BN, and each coefficient used was set for use with Honda genuine oil. Figure 11 shows the correlation between the measured BN analysis values and the value of the BN obtained from calculation, in each mode. The measured values agreed well with the simulation values, indicating that it was possible in practice to estimate the BN without issue.

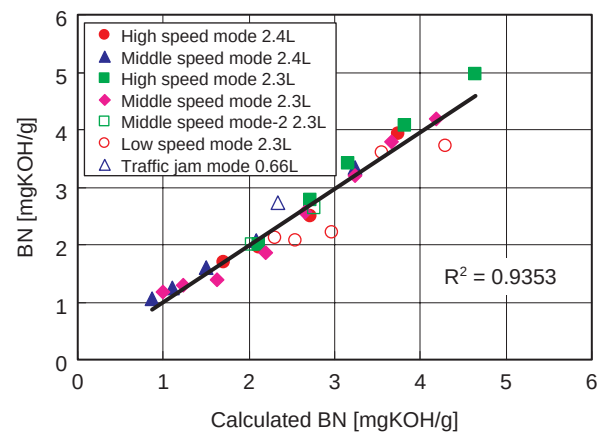


Fig. 11 Correlation between calculation results and actual BN measurement values

5. Considerations

The BN deterioration rate equation proposed in this research was derived from the results of tests using various types of engine oil. For this reason, although there was some apprehension concerning cases where the equation was applied to various kinds of oil on the market, it was found that it could be used with all of the oils tested to present, and so the equation is considered to be highly versatile. It is thought that this is because the basic phenomenon of the deterioration mechanism remains unchanged even when the type of additive or chemical structure that indicates the BN changes. It is known that the deterioration rate coefficient and the BN concentration of the engine oil differ significantly with the type of engine oil, thus affecting the life of the oil. When using the deterioration rate equation to detect the BN deterioration on board, it is desirable to determine the composition for each kind of engine oil and to incorporate the results into the deterioration rate coefficient, as well as to set the BN working limit value for each kind of oil used. Conceivable methods of setting the BN deterioration rate coefficient and the working limit value include methods in which the user enters the kind of oil used manually, or developing a sensor for automatically identifying the kind of oil used. It is also desirable to specify the kind of oil to use, which is frequently being done these days by the European automobile manufacturers. By using a combination of these methods, the BN estimation accuracy should increase, enabling the oil replacement intervals to be increased, and also enabling the construction of a highly effective system for users.

6. Conclusion

In this research, the effectiveness of methods of estimating the deterioration of the BN, which can be used as an index of engine oil deterioration, was determined. Oil tests were carried out, and a detailed analysis was performed. As a result, it was found that there was regularity in the deterioration of the BN, and a general equation describing the rate of BN deterioration was proposed. Use of this equation to calculate the BN deterioration in an actual vehicle showed that the BN deterioration could be adequately estimated using this method. This indicates that it is possible to realize an on-board type BN deterioration estimation system using the abovementioned algorithm.

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References

- (1) Sakurai, T.: Sekiyuseihintenkazai, SAIWAI SHOBO, p. 287 (1973) (in Japanese)
- (2) Konishi, S., Kamita, T.: Principle and Application to Lubricating Oil, Corona Publishing Co., Ltd., p. 144 (1992)
- (3) Inoue, K., Yamanaka, Y.: Change in Performance of Engine Oils with Degradation, SAE Paper 902122
- (4) Murakami, Y., Aihara, H., Kuniya, J.: Analysis of Mechanism Intermixing Combustion Products in Engine Oil, Transactions of The Japan Society of Mechanical Engineers (B edition), Vol. 53, No. 496, p. 3814-3821 (1987)
- (5) Murakami, Y., Aihara, H.: Effects of NO_x and Unburned Gasoline on Low Temperature Sludge Formulation in Engine Oil, SAE Paper 910747
- (6) Kawaguchi, S., Nishimoto, K.: Butsuri Kagaku Josetsu, Kagaku-Dojin Publishing Company, Inc, p. 97-98 (1985) (in Japanese)
- (7) Murakami, Y., Aihara, H.: Analysis of Mechanism Intermixing Combustion Products in Engine Oil (Continued Report), Transactions of The Japan Society of Mechanical Engineers (B edition), Vol. 55, No. 519, p. 3571-3580 (1988)

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Technical Papers Submitted to Out-of-company: January, 2006 to June, 2006

(English Papers only)

Date	Conference	Title	Author	Organization
2006.1.30	JSAE/Review of Automotive Engineering Vol.27, No.1	A New Steering Control Method and its Evaluation	Yasuharu Oyama Shoichi Sano	Tochigi R&D Center Tokyo Denki University
		Analysis of Vibration Behavior of Full Vehicle during Brake Judder	Tetsuya Yamaguchi	Tochigi R&D Center
		Evaluation Method for Road Load Simulation Using a Tire Model	Hitoshi Haga	Tochigi R&D Center
		Numerical Simulation of Exhaust Pulsating Flow & Orifice Noise	Masato Sakurai	Tochigi R&D Center
		Development of Simulation Model for Active Control Engine Mount	Kosuke Sakamoto Tatsuhide Sakai	Tochigi R&D Center Tochigi R&D Center
		Development of New Intelligent Tourer: 2004 Model INSPIRE	Shigehiro Yasuki Yasuaki Asaki Hiroshi Tashiro Hideki Matsuoka Taku Kono	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
2006.2.1-2	SAE/SAE Hybrid Vehicle Technologies 2006 Symposium (San Diego, CA, USA)	The New 2006 CIVIC HYBRID technical presentation	Tatsuro Horie Tsutomu Kishi Osamu Hasegawa	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
2006.4.1	JSAE/Review of Automotive Engineering Vol.27, No.2	Development of Compact Lightweight Muffler Valve for 2005 Model Year LEGEND	Hidegori Hamashima	Automobile R&D Center
	The Institute of Electrical Engineers of Japan (IEEJ)/IEEJ Transactions on Industry Applications Vol.126-D, No.4	High Speed Analysis of 3-phase High Frequency Voltage Input Sensorless Control Fed by PWM Inverters with Magnetic Saturation	Nobuyuki Imai Shigeo Morimoto Masayuki Sanada Yoji Takeda	Automobile R&D Center Osaka Prefecture University Osaka Prefecture University Osaka Prefecture University
2006.4.3-6	SAE/SAE 2006 World Congress (Detroit, USA)	Real-time Piston Temperature Measurement using Induction Telemetry System	Toshiyasu Watanabe Masatoshi Suzuki Shingo Ueda	Motorcycle R&D Center Motorcycle R&D Center Motorcycle R&D Center
		Development of IMA Motor for 2006 Civic Hybrid	Shoei Abe Masashi Murata	Automobile R&D Center Automobile R&D Center
		Development of the Intelligent Power Unit for 2006 Civic Hybrid	Toshiyuki Matsuoka Masami Nakamura Kunio Hasegawa	Automobile R&D Center Automobile R&D Center Automobile R&D Center
		PCCI Operation with Fuel Injection Timing set close to TDC	Tomohiro Kanda Takazo Hakozaiki Tatsuya Uchimoto Junichi Hatano Naoto Kitayama Hiroshi Sono	Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center
		Study on Ignition Timing Control for Diesel Engines Using In-Cylinder Pressure Sensor	Mamoru Hasegawa Yuichi Shimasaki Satoshi Yamaguchi Makoto Kobayashi Hideki Sakamoto Naoto Kitayama Tomohiro Kanda	Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center
		Study on Impulse Charger for Enhancement of Volumetric Efficiency of SI Engine	Hitoshi Takahashi Takeya Harada Toshihiro Yamaki Toshihiro Oikawa	Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center
		The Influence of Pelvis Design on the Lateral Pelvic Impact Response of the Polar-II Pedestrian Dummy	Damien Subit Johan Ivarsson Jeff Crandall Yuji Kikuchi Yukou Takahashi	University of Virginia University of Virginia University of Virginia Automobile R&D Center Automobile R&D Center
		Development of a Finite Element Model for a Pedestrian Pelvis and Lower Limb	Yuji Kikuchi Yukou Takahashi Fumie Mori	Automobile R&D Center Automobile R&D Center PSG Co., Ltd.

2006.4.3-6	SAE/SAE 2006 World Congress (Detroit, USA)	Development of a Prototype Safety Analysis System to Assess and Forecast Vehicle Safety	Hideki Suzuki Yoichi Sugimoto Mike Van Auken John W. Zellner	Automobile R&D Center Automobile R&D Center Dynamic Research, Inc. Dynamic Research, Inc.
		Evaluation Method for Road Load Simulation Using a Tire Model and an Applied Example	Hitoshi Haga	Automobile R&D Center
		Development of a Lightweight Sound Package for the New Model Civic in 2006	Yoshihiro Noguchi Kazuhiro Misaji Hiroko Tada Takayuki Doi	Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center
		Ignition and combustion simulation in HCCI engines	Toshihiro Ohashi Xiaofeng Yang Toru Takabayashi Yasuhiro Urata Shunichi Kubota Hideki Katsuyama	Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center
		Advanced Development of Speech Enabled Voice Recognition Enabled Embedded Navigation Systems	Koichi Kojima Kenneth White Harvey Ruback Jan Sedivy Keisuke Kondo	Automobile R&D Center IBM Corporation IBM Corporation IBM Corporation Automobile R&D Center
		Development of a Nighttime Pedestrian Recognition Assistance System	Takayuki Tsuji Nobuharu Nagaoka Hideki Hashimoto Izumi Takatsudo	Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center
		Application to Engine Development of Friction analysis by Piston Secondary Motion Simulation in Consideration of Cylinder Block Bore Distortion	Kenji Sato Makoto Ito Shinsuke Koda Kinya Fujii	Automobile R&D Center SANNO TEC Co., Ltd. Automobile R&D Center Automobile R&D Center
		Development of a New Low Fuel Consumption 1.8L i-VTEC Gasoline Engine with Delayed Intake Valve Closing	Kazuhiro Akima Kazuyuki Seko Wataru Taga Kenji Torii Satoshi Nakamura	HONDA R&D AMERICAS, INC. OHIO CENTER Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center
		Comparison of Pressure Losses due to Catalysts with Different Specifications Using Simple Theoretical Analysis	Masaru Enomoto	Automobile R&D Center
		Research and Analysis of ISG Belt-drive System for Idling Stop System	Kazuma Okuda Yasuyuki Komatsu Youichi Nakahara	Automobile R&D Center Automobile R&D Center Automobile R&D Center
		Development of Hybrid System for 2006 Compact Sedan	Takaaki Iijima	Automobile R&D Center
		Advanced Emission Control System for ULEV Application	Katsushi Watanabe Wataru Taga Tomotaka Hirota Kenji Tanikawa Kazuhiro Nagashima Geng Zhang Hideaki Muraki	Automobile R&D Center Automobile R&D Center Johnson Matthey Japan Inc. Johnson Matthey Japan Inc. Johnson Matthey Japan Inc. Johnson Matthey Japan Inc. Johnson Matthey Japan Inc.
		Development of Fuel Cell Vehicle with Next-generation Fuel Cell Stack	Yoshikazu Murakami Kenji Uchibori	Automobile R&D Center Automobile R&D Center
		Development of High-strength Piston Material with High Pressure Die Casting	Akito Tanihata Naoko Sato Koji Katsumata Takashi Shiraishi	Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center
		Application of 590/780MPa Grade TRIP Steel to Body-in-White	Takahiro Tanae James Dykeman	Automobile R&D Center HONDA R&D AMERICAS, INC. OHIO CENTER
		Development of a light weight high dampening liquid applied sound deadener material	Naoki Tsuruta Yoshihiro Noguchi Hiroshi Suzuki	Automobile R&D Center Automobile R&D Center Nihon Tokushu Toryo Co., LTD.

2006.4.3-6	SAE/SAE 2006 World Congress (Detroit, USA)	Cogging Torque Target and Design for EPS	Atsuhiko Yoneda Takashi Miyoshi Yasuo Shimizu	Automobile R&D Center Automobile R&D Center Automobile R&D Center
		Control for Moment of Motor Inertia on EPS	Yasuo Shimizu Yoshihiro Oniwa	Automobile R&D Center Automobile R&D Center
		A Study of Worm Gear to Size Reduction and Performance Enhancement for EPS	Katsuji Watanabe Yasuo Shimizu Yasuhiro Terada	Automobile R&D Center Automobile R&D Center Automobile R&D Center
		Development and Validation of the Finite Element Model for the Polar-II Upper Body	Akihiko Akiyama Yukou Takahashi Masayoshi Okamoto Yuji Kikuchi Jaeho Shin Sang-Hyun Lee Jason Kerrigan Kurosh Darvish	Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center University of Virginia University of Virginia University of Virginia University of Virginia
		Application of Technique for Prediction of Stress in Elements of Metal Pushing V-belt by Simulation of Dynamic Behavior	Toshihiro Saito Koji Miyamoto	Automobile R&D Center ITO CORP.
2006.4.10-13	Eindhoven University of Technology/13th International Conference on Deformation, Yield and Fracture of Polymers (Kerkrade, Netherlands)	Analysis on Mechanical Behaviour of Polypropylene by Coarse-Grained Molecular Dynamics Model	Hiroyuki Mae	Automobile R&D Center
		Fracture Simulation of Elastomer Blended Polypropylene Based on Elastoviscoplastic Constitutive Equation with Craze and Tensile Softening Law	Hiroyuki Mae	Automobile R&D Center
2006.5.7-10	ASME (American Society of Mechanical Engineers)/2006 Internal Combustion Engine Division, Spring Technical Conference (Aachen, Germany)	SIMULATION OF ACCESSORY DRIVE BELT CONCERNING TO SLIPPAGE AND TRANSVERSAL VIBRATION	Hiroshi Takagishi Hiroshi Yoneguchi Atsushi Nagakubo	Automobile R&D Center Automobile R&D Center PSG Co., Ltd.
2006.5.8-10	SAE/SAE 2006 Government/Industry Meeting (Washington, D.C., USA)	Development of 2005 Model Year ACCORD Hybrid	Masaaki Kaizuka	Automobile R&D Center
2006.5.8-11	ASME/ASME Turbo Expo 2006, Power for Land, Sea and Air (Barcelona, Spain)	Aerodynamic Characteristics of Supercritical Outlet Guide Vanes at Low Reynolds Number Conditions	Toyotaka Sonoda Heinz-Adolf Schreiber	Aircraft Engine R&D Center DLR (German Aerospace Center)
2006.5.9-10	Automotive Circle International/Euro Car Body Forum "15th European Automotive Lightweight Engineering Conference" (Frankfurt, Germany)	Development of Aluminum Automotive Parts Technologies for Mass Production	Noboru Hayashi Kei Kiyama	Automobile R&D Center Honda Engineering Co., Ltd.
2006.5.17-19	Advanced automotive battery/AABC 2006 (Baltimore, USA)	Development of NiMH Batteries for HEV	Kazuhiko Aitaka	Automobile R&D Center
2006.6.14-16	American Automatic Control Council/ACC (American Control Conference) (Minnesota, USA)	Cooperated Control of Multi Stage Injection System for Direct Injection Gasoline Engine	Ikue Kawasumi Yuji Yasui Kosuke Higashitani Hisashi Ito	Automobile R&D Center Automobile R&D Center Automobile R&D Center Automobile R&D Center
		Development of an Engine Idle Speed and Emission Controller	Masaaki Nagashima William S. Levine	Automobile R&D Center University of Maryland

Editor's Postscript

The most important part of a technical paper is the introduction. It's no exaggeration to say that a paper is evaluated based on the introduction. It clearly (and in comparison with competitors) states the background of the research involved, the results that have been achieved in similar research, and any problems that exist, with citations of reference sources from Japan and other countries included throughout. It informs the reader about the purpose of the research, its original features, and its necessity. This is the most important function of the introduction. The fact of the matter is, however, that extremely few papers achieve this level of style in the first drafts. I am not bothered at all when the introductory section is somewhat long because of its importance. Rather, I think that it should be longer. When a writer is able to write an introductory section which covers all those points, then I can expect the paper to have a solid structure overall. The rest of the sections, to put it very bluntly, are like supplements or appendices to the introduction. If you are going to be submitting a technical paper to the *Technical Review*, then I hope you will think about putting a bit more into your introduction when you write it. I believe that doing so is the first step toward writing an international-level technical paper that shows that you are a first-rate engineer or researcher with true originality.

(Sonoda)

