

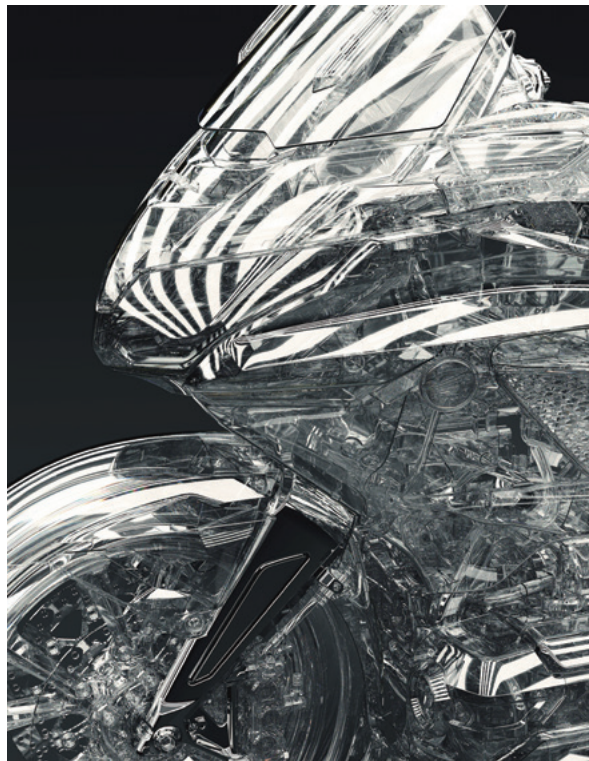
Honda R&D



Technical Review

October 2018

Vol.30 No.2



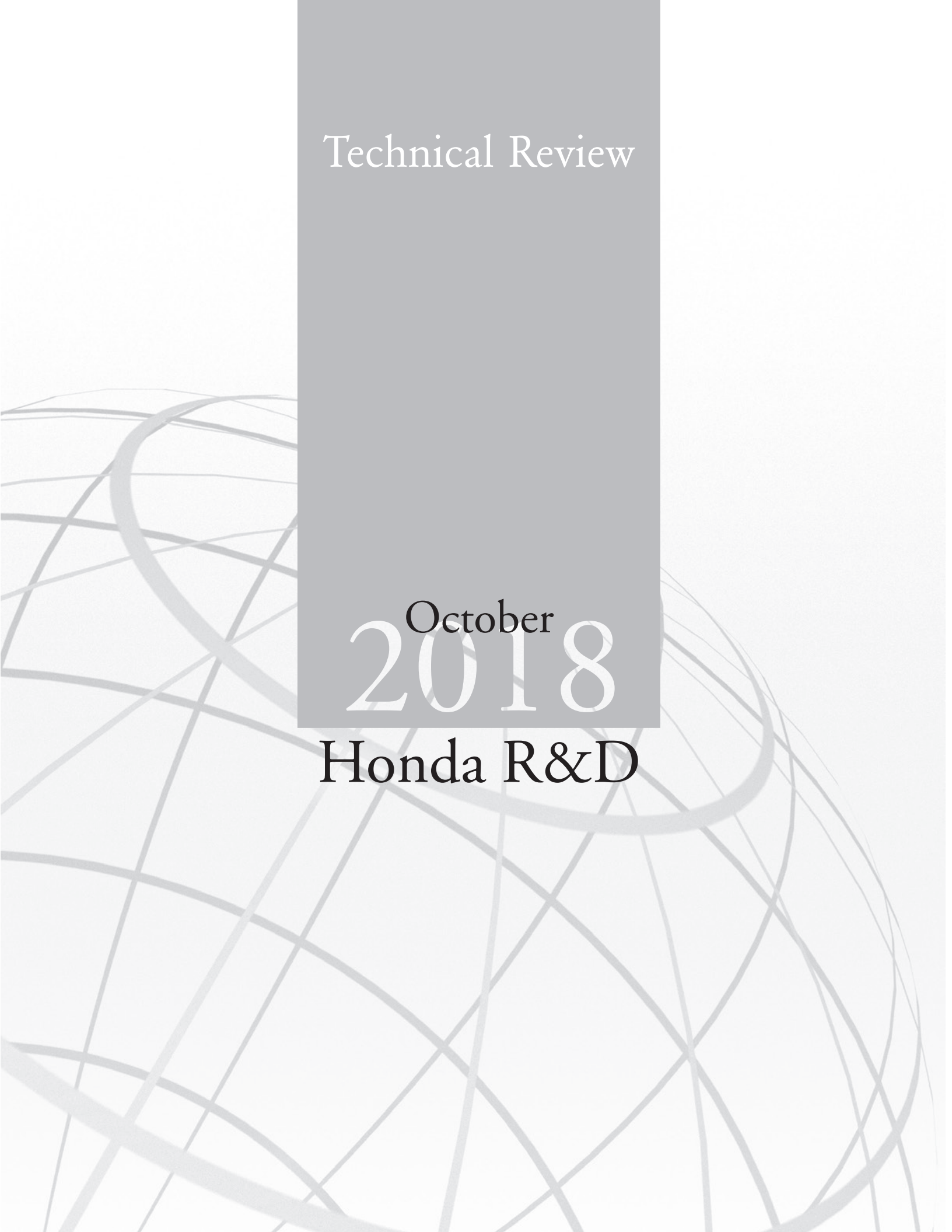
Comment on the Cover

The cover design for this issue takes the new model Gold Wing as its motif.

The image represents the fusion of mechanical elements, made up from large numbers of parts, with a beautiful styling design.



Power Products R&D Center
Michihito OKADO

The background features a light gray wireframe sphere on the left side. A solid gray vertical bar is positioned in the center, serving as a backdrop for the text.

Technical Review

October
2018

Honda R&D

CONTENTS

Introduction of new technologies

Development of 2018 Gold Wing	Kihoko KAITA, Yutaka NAKANISHI	1
Development of Horizontally-opposed Six-cylinder Engine for 2018 Gold Wing	Hiroshi YOKOTA, Satoru NOJIMA, Yutaka SONODA	16
Development of 7-speed Dual Clutch Transmission for 2018 Gold Wing	Taku HIRAYAMA, Yukio YOKOGAWA, Yasushi FUJIMOTO	31
Development of Ride Modes for Various Conditions for 2018 Gold Wing	Atsushi KUSUDA, Yukio YOKOGAWA, Kazuho NAKAI	38
Development of Double Wishbone Type Front Suspension for 2018 Gold Wing	Naoki KUWABARA, Junpei MISHIRO	47
Development of Robotic Lawnmower Miimo HRM3000	Susumu OKUBO, Kenta KAWANISHI, Makoto YAMAMURA Takuya KANISAWA, Dai HYAKUSAWA	53
Technology for Enhancing Thermal Efficiency of Gasoline Engine by Pre-chamber Jet Combustion	Hiroki KOBAYASHI, Kiminori KOMURA, Keitaro NAKANISHI Atsushi OHTA, Hiroki NARUMI, Noriyuki TAKEGATA	60
Development of Evaluation Method for Waviness Property on Hypoid Gear Tooth Surface	Takeshi HORI, Naoto SHUKUSHIMA	68
Intelligent Power Unit for the 2018 Model Year Accord Hybrid	Jun MACHIDA, Naoyoshi MADOKA, Shogo HAYASE	75
Initiative to Evaluate Effects of Atmospheric Impurities on PEM Fuel Cell Performance	Keiko YOSHIDA, Ryu OKANO	80
Method for Predicting Vibration Transmission of Car Air-conditioning Hose in Light of Prototype-less Development	Masahiro FUKAZAWA, Shingo UNIGAME	87

Technical papers

Analysis of Capacity Fading for High-power Lithium Ion Rechargeable Batteries

..... Pu QIAN, Tomohiro KINOSHITA, Hiroshi SAKAI
Terumi FURUTA, Mitsumoto KAWAI 93

Durability Evaluation of Polymer Electrolyte Membrane for Fuel Cells by Molecular Weight Analysis

..... Atsushi HIRAIDE, Akira NAKAHARA 103

Technique for Simulating Vehicle Handling in Very Low Speed Range for Model Based Development

..... Takayuki TOYOSHIMA, Yoshitomo AZEKATSU, Yoshiaki KOBAYASHI
Naofumi IKEDA, Hiroki TAKANO, Isao OBATA 110

CFD Calculation of Cavitation in Outboard Motors. Ayaka NANGO, Yuka IGA 119

Development of 2018 Gold Wing

Kihoko KAITA* Yutaka NAKANISHI*

ABSTRACT

The Gold Wing is a grand tourer type motorcycle of Honda, which was first released in 1975. In the development of the new Gold Wing, we endeavored to create the motorcycle with the most enjoyable maneuverability throughout all the models appeared in the chronicle of the Gold Wing by applying the latest technologies while inheriting their DNA of reliability and safety conscience. To achieve this goal, a light weight and highly dense packaging was set as the most crucial objective, and its packaging design was reviewed throughout the entire vehicle. For the power source, a light weight compact engine, which is water-cooled four-stroke horizontally-opposed six-cylinder engine with 1,833 cm³ displacement, was developed. As its transmission, a dual clutch transmission, which has seven-speed and reverse gears with slow speed forward and backward moving functions, was applied first in the Gold Wing series. A double wishbone type suspension uniquely designed by Honda was developed for its suspension. Thanks to those advanced technologies, the downsizing was realized compared to the previous model by 40 kg in weight, 55 mm in vehicle length, 40 mm in vehicle width and 25 mm in vehicle height, and that made it possible to realize the motorcycle with the most enjoyable maneuverability throughout all the models appeared in the chronicle of the Gold Wing.

1. Introduction

Ever since the introduction in 1975, the Gold Wing, which is a Honda motorcycle model belonging to the grand tourer category, has kept evolving on the basis of inputs from the customers and the market. Approximately 795,000 units have been produced cumulatively. Primarily for the North American market, the first generation Gold Wing was born with a concept “King of motorcycle,” meaning the motorcycle sitting atop all the motorcycles beyond the category or brand classification. To maintain dignity and elegance appropriate for “King” such as majestic presence and composed manner of running, we aimed at a model featuring quietness and comfort. To achieve aforementioned goal, the basic packaging arrangement of the Gold Wing, which is composed of a water-cooled four-stroke horizontally-opposed cylinder engine featuring low vibration, excellent quietness and low center of gravity and coupled with shaft drive system and an under-seat fuel tank, was applied. With its high reliability, sense of security, low vibration and high quality of running, the first-generation Gold Wing was appreciated by the riders fond of long-distance touring like the one across the American Continent. They customized Gold Wing by adding a large fairing, a rear trunk and panniers, and enjoyed motorcycle

trip with the partner. Responding quickly to such needs of the market, the second-generation Gold Wing GL1100 Interstate was introduced in 1980. To enhance performance when carrying a passenger or heavy loads, the engine displacement was increased to 1,100 cc to add torque in low to middle engine speeds. In addition to the standard model, the large fairing, the rear trunk, and the panniers were provided as standard equipment in the Interstate model. The Gold Wing Aspencade was introduced in 1982. This model was equipped with the audio system, CB radio, passenger seat back, etc., serving as a benchmark of modern-day Gold Wing. Moreover in the third generation (1984), the engine displacement was increased to 1,200 cc. In the fourth generation (1988), the number of cylinders was increased employing the horizontally-opposed six-cylinder configuration, and the engine displacement was further increased to 1,500 cc. In the most recent fifth generation (2001), while the horizontally-opposed six-cylinder configuration was maintained, the engine displacement was increased to 1,800 cc⁽¹⁾. While undergoing minor changes such as the application of antilock brake system (hereafter as ABS), world’s first motorcycle airbag, cosmetic restyling, the GL1800 have been appreciated by the customers over the last 17 years.

Undergoing those changes, the Gold Wing has kept

* Motorcycle R&D Center

evolving as a grand tourer motorcycle. It was a matter of course that the body size grew large and the weight increased as a result of evolution. Meanwhile in recent years, the F6B (2013), the F6C (2014) were developed on the basis of the fifth generation GL1800 to strengthen the model line up having the Gold Wing in its core. The fifth generation GL1800 is sold in more than 70 countries and territories. The environment in which the product is used is globally diversifying.

Keeping in mind the aforementioned backgrounds, hearings from the customers in various regions were conducted to set concept of the sixth generation Gold Wing. The hearings revealed some changes of customer demands for the Gold Wing. Especially such comments as “being grand and luxurious impair user-friendliness in some situation,” “the comfortableness sometimes makes me feel boring” called for in-depth rethinking of the long-lasting (more than 40 years) theme “The King” itself. Accordingly at the start of product development, we liberated ourselves from the preconception “The King” which used to be a precondition, considered what the fundamental attractiveness of the motorcycle was, and redefined the value that the sixth generation Gold Wing should offer to the customers.

The total update was conducted by employing the latest technologies into the sixth generation, 2018 Gold Wing GL1800 while maintaining the aura and high-quality image of Gold Wing models of the past generations.

The aims of product development and the technical means to achieve the goals are discussed in this paper.

Figure 1 shows sixth generation Gold Wing.

Figure 2 shows previous generation Gold Wings.

Figure 3 shows fifth generation Gold Wing, F6B and F6C.

Figure 4 shows vehicle size and weight.



The Gold Wing Tour DCT Airbag (2018)

Fig. 1 Appearance of 2018 Gold Wing Tour



Fig. 2 Appearances of successive Gold Wings



Fig. 3 Appearances of fifth generation Gold Wing series

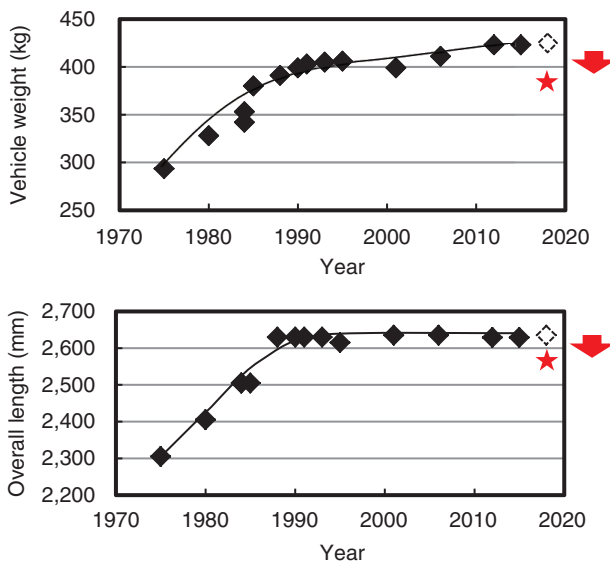


Fig. 4 Weight and overall length of successive Gold Wing series

2. Aims of Product Development

2.1. Concept

The concept of this model was set as “The Honda Premium Tourer-More exciting and fulfilling riding experiences for rider and passenger.”

When hearing in the market in order to set the concept, such voices are addressed from the baby-boomer generation, in which majority of the users of the previous models belongs to, that while the customers who accumulate ages along with the Gold Wing want to continue riding a Gold Wing, they want mitigation in weight and a better slow speed maneuverability. In the meantime, the product should also appeal to men and women in the Generation-X, who are the target users of the next generation, to stay as a long-loved motorcycle. They also addressed such voices that “I also want casual use of the motorcycle for a trip to escape from everyday life using a short spare time in daily life.” From such changes of users and usage, we considered it was

necessary to leap from the gorgeous & luxurious, massive and heavy line and to regain youthfulness.

The core concept of Gold Wing that has lasted over the past 40 years is the fun of travelling accompanied by the partner. We aimed at a motorcycle with the most enjoyable maneuverability throughout all the models appeared in the chronicle of the Gold Wing by brushing up the core concept and applying latest technology while maintaining reliability and sense of security, which are brought about by the unique horizontally-opposed six-cylinder engine and what the Gold Wing is known for.

To realize the aim, the theme in various fields has been set as per the following.

(1) Complete vehicle and chassis

Light weight and high-density packaging of complete vehicle to allow wide variety of applications from daily use to a long touring

(2) Power unit

Light weight and compact horizontally-opposed six-cylinder power unit, smooth yet aggressive when required

(3) Electrical systems, control systems

Advanced technologies to realize premium feel

(4) Styling

Taut lines with contemporary edge

Based on the above themes, studies were conducted on what kind of technologies, specifications and equipment should be applied, and practical actions were taken accordingly.

2.2. Types and Specifications

In this product development project, two types were developed simultaneously to allow customer in a broader range of generation and preference to enjoy the new model. The model with the conventional rear trunk was designated as “Gold Wing Tour,” and the one without the rear trunk as “Gold Wing.” The overview of the models is shown in Fig. 5.

The manual transmission (hereafter MT) and the dual clutch transmission (hereafter DCT) are provided to each model.

In the Gold Wing Tour, the type with an airbag is also provided as “Gold Wing Tour DCT Airbag.”

With these combinations, a maximum of five types are provided for a destination country or region. Furthermore, designating derivative models and various equipment specifications to satisfy the needs of each destination country or region, a total of 48 derivative models have been developed for a total of 18 destination countries and regions.

Unless otherwise specified, the Gold Wing Tour DCT Airbag type for the USA is referenced in this paper.

2.3. Specifications and Equipment

Employing new technologies suitable to create premium image, the equipment specifications were determined at this occasion of full model change. The major equipment specifications are listed in Table 1. The bold letter cells in the table indicate new mechanisms and/or new technologies.

2.3.1. Engine

Aiming at light weight, compactness, more power output and fuel economy, the 1,833 cm³, horizontally-opposed six-cylinder engine has been developed (Fig. 6). Characteristics such as the adequate torque in lower engine speeds, the relaxed manner when cruising, the strong acceleration from cruising state, low vibration, quietness had to be enhanced from the previous model. To achieve the goal, the bore/stroke ratio was reviewed, and the Unicam valve train⁽²⁾ using the finger follower rocker arm in the intake

side just like in CRF450 (motocrosser) was employed. The power output and fuel economy were enhanced by the modifications such as the higher compression ratio, the redesigning of combustion chamber from the two-valve bath-tub type to the 4-valve pent roof type, etc. (Fig. 7). The details are discussed in the separate paper titled “Development of Horizontally-opposed Six-cylinder Engine for 2018 Gold Wing”⁽³⁾.

In the starting and charging system, the integrated starter generator (hereafter ISG) having the functions of starter motor and alternator is employed first time in a large-size motorcycle, thus quietness at engine start-up has been enhanced. By combining the ISG and the DCT, idling stop function enabling smooth start and stop is also provided. (not in USA type and MT type)

2.3.2. Transmission

Two types of transmission, the MT and the DCT are provided. The new Gold Wing is equipped with 6-speed manual transmission compared to the 5-speed for the previous model. The gear ratio from the first to the fifth gear is almost identical to the previous model to maintain adequate traction. The addition of the higher ratio of sixth gear helps ensure quietness while cruising as well as improving fuel consumption. The cam-type assist slipper clutch is employed in the manual type. With the assist system employed, the clutch pull is lighter by approximately 20%. The slipper system reduces shocks when engaging the clutch after down shifting.

Meanwhile in the DCT employed first time in the Gold Wing, the forward gears are increased to 7 speeds. While maintaining the same ratio as the sixth gear of the MT in the seventh gear, a gear is added in the lower gears. With the closer lower gear ratio, the differences of engine speed before and after shifting are reduced, which lowers noises and shocks when shifting. The shift timing is adjusted to the purpose of each of the four ride modes. The details are discussed in separate paper titled “Development of Ride Modes for Various Conditions for 2018 Gold Wing”⁽⁴⁾.

The comparison of gear ratio distribution for new model and previous model converted to the vehicle speed is shown in Fig. 8.

The reverse system, which is unique only in the Gold Wing, has been modified from the type driven by the starter motor to the one driven by the engine. (only in DCT type. For the MT type, the reverse utilizes the starter motor just like the previous model.) With the chain used in the reverse power train, the compact and light weight construction has been achieved without adding an idle shaft. Moreover, as the direction of driving force transmission from the two clutches is such that it is in the reverse direction from the clutch for odd-number gears and in the forward direction from the clutch for the even-number gears, the system permits forward /reverse switch only by altering clutch engagement. Using this system, the walking speed mode has been developed to allow forward and reverse travel at the



The Gold Wing Tour DCT Airbag



The Gold Wing

Fig. 5 Appearances of sixth generation Gold Wing series

Table 1 Comparison of specifications and equipment between developed and previous models

Item		Developed model		Previous model
		Gold Wing	Gold Wing Tour (DCT, airbag)	GL1800 (airbag)
Engine	Cylinder arrangement	6 horizontally opposed		6 horizontally opposed
	Cooling system	Liquid cooled		Liquid cooled
	Valve train	4 valve OHC (Unicam)		2 valve OHC
	Bore × Stroke (mm)	73 × 73		74 × 71
	Displacement (cm ³)	1,833		1,832
	Throttle bore	$\phi 50 \times 1$ with TBW		$\phi 40 \times 2$
	Starter	Integrated starter generator		Motor
	Generator			Excitation formula
	Transmission	Manual 6 speed	Dual clutch 7 speed + Reverse	Manual 5 speed
	Clutch system	Mechanical with assist slipper	Hydraulic control	Hydraulic assist
	Reverse	Motor driven ※1	Engine driven	Motor driven
Body	Frame body	Aluminum twin tube		Aluminum twin tube
	Windscreen	Electrically adjustable (low height)	Electrically adjustable (standard)	Manually adjustable
	Airbag	—	equipped	equipped
	Drive type	Shaft		Shaft
	Front suspension	Double wishbone type		Telescopic type
	Rear suspension	Swing arm type (PRO-LINK/PRO-ARM)		Swing arm type (PRO-LINK/PRO-ARM)
Electrical equipment and control	Lighting devices	LED		Bulb
	LCD display	7 inch multi display		6.5 inch multi display
	Apple CarPlay ※3	adopted		—
	Ignition	Honda SMART Key system (electrically)		Mechanical key
	Steering lock system			Mechanical key + remote controller
	Rear trunk and pannier lock system			
	Cruise control	equipped		equipped
	Mode select	4 modes		—
	Honda Selectable Torque Control	※2	equipped	—
	Hill Start Assist	equipped		—

※1 No reverse mode for US model

※2 Only equipped for Japan model

※3 Apple CarPlay is trademark of Apple Inc.



Fig. 6 Appearance of sixth Gold Wing's engine

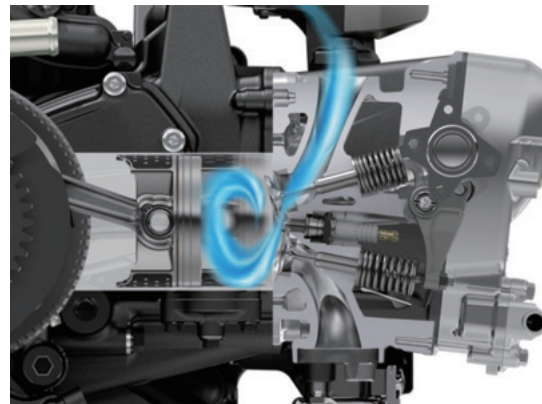


Fig. 7 Internal view of engine

very slow speed by operating the switch on the handlebar. This system makes back and forth maneuver and parking easy, enhancing convenience. The DCT gear arrangement is shown in Fig. 9.

The details are discussed in the separate paper titled “Development of 7-speed Dual Clutch Transmission for 2018 Gold Wing”⁽⁵⁾.

2.3.3. Front suspension

The Honda’s unique double wishbone type suspension for motorcycles has been developed for the front end. With this suspension design employed, as well as realizing the highly dense packaging as mentioned later in section 3.2.1., an enhancement of riding comfort is intended. In the conventional telescopic type front suspension, the suspension itself is used as the structural members.

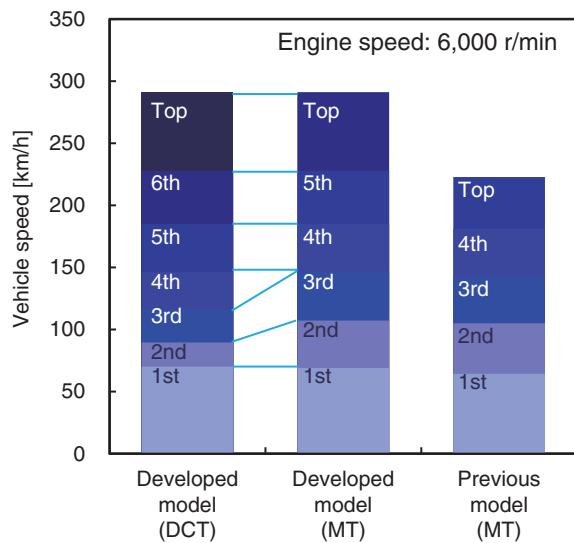


Fig. 8 Vehicle speed on each gear position at 6,000 r/min

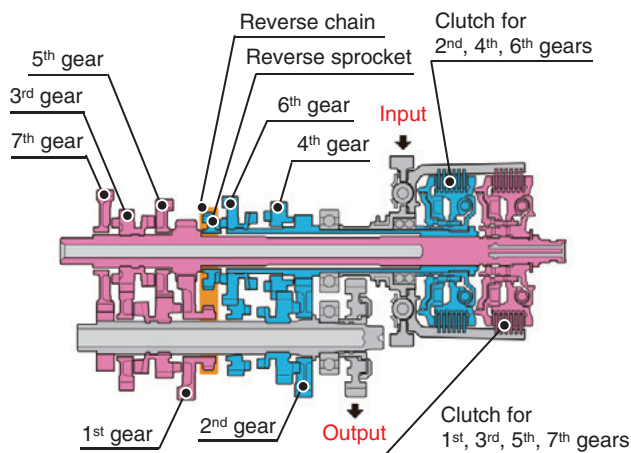


Fig. 9 Gear arrangement of dual clutch transmission

Accordingly, large side forces occur in the outer and inner fork tubes, causing an increase of sliding friction.

In the newly developed double wishbone type, the function to absorb shocks is separated from the steering function of the fork holder. Both of the said components are held by the upper and lower arms (Fig. 10). With a roller bearing incorporated in all the pivots of the linkage, the friction at the time of suspension stroke or when steering is reduced. Consequently, the sliding friction of the cushion has been reduced, and the shocks transmitted from the road surface to the handlebar have been reduced by approximately 30% from the previous model. With the shock unit separated from the steering system, the steering moment of inertia has been reduced by more than 40% from the previous model, which enhances steering agility.

Consequently, the agile handling, when doing frequent steering in a city associated with changing vehicle leaning directions and/or lane changes, was realized, as well as smooth ride without large shocks when doing acceleration/deceleration or passing over a bump/ditch on a road surface.

The details are discussed in the separate paper titled “Development of Double Wishbone Type Front Suspension for 2018 Gold Wing”⁽⁶⁾.

2.3.4. Electrical systems & control systems

The system that allows the rider to select a ride mode that best suits the conditions by controlling five variables has been developed. In this system, the integrated control is conducted by linking variables such as the engine power response and the traction at the time of rear wheel slip, which are controlled by the throttle by wire system (hereafter TBW), the gear shift timing (DCT type only), the suspension damping force characteristics and the front/

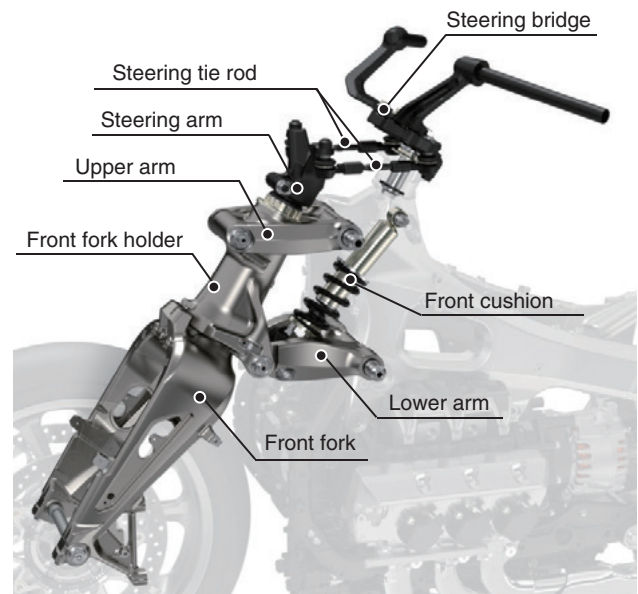


Fig. 10 Configuration of double wishbone type front suspension

rear brake force distribution. The details are discussed in the separate paper titled “Development of Ride Modes for Various Conditions for 2018 Gold Wing”⁽⁴⁾.

The hill start assist system, which is the first in Honda motorcycles, is employed to enhance sense of security and convenience when starting on a hill. When rider’s squeeze of brake lever at the speed and load above the prescribed level is detected when the motorcycle is stationary, the hill start assist mode starts functioning. As the rider starts to launch the motorcycle, the brake force is maintained for a prescribed time period even after releasing the brake lever to help ensure sense of security when starting from a hill.

Furthermore, the cruise control is equipped just like the previous model. Apple CarPlay (Apple CarPlay is a trademark of Apple Inc.), firstly applied in motorcycles, 7-inch multi-function display and Bluetooth compatible audio system were employed as the built-in infotainment system. The Honda SMART Key system is applied to allow ignition switching and handlebar locking without using the hard key (Fig. 11). This system is linked with the locks of the side pocket, panniers, and rear trunk for convenience.

The appearance picture of the whole instrument panel is shown in Fig. 12.



Fig. 11 Honda SMART Key



Fig. 12 Instrument panel

Table 2 Comparison of dimensions and other properties between developed and previous models

		Developed model		Previous model
		Gold Wing (MT)	Gold Wing Tour (DCT, airbag)	GL1800 (MT, airbag)
Curb mass	kg	356	383	423
Overall length	mm	2,475	2,575	2,630
Overall width	mm	925	905	945
Overall height	mm	1,340	1,430	1,455
Wheelbase	mm	1,695		1,690
Seat height	mm	745		740
Ground clearance	mm	130		125
Caster angle		30°30′		29°15′
Trail	mm	109		109
Fuel tank capacity	L	21		25

3. Configuration of Complete Vehicle

3.1. Main Dimensions of Complete Vehicle

The main dimensions of the complete vehicle have been set to achieve the stability that allows relaxed riding and the maneuverability that allows positive control of the motorcycle. The stability of the motorcycle is maintained by extending the wheel base to 1,695 mm, which is longer than the previous model by 5 mm, and employing the less-steep caster angle. The comparison of the dimensions and other properties between developed and previous models is shown in Table 2. The main dimensions of developed model are shown in Fig. 13.

Pursuing a light weight and highly dense packaging, the complete vehicle has been downsized. The size of this model and the previous model is compared in Fig. 14. The overall length is shorter by 55 mm, overall width by

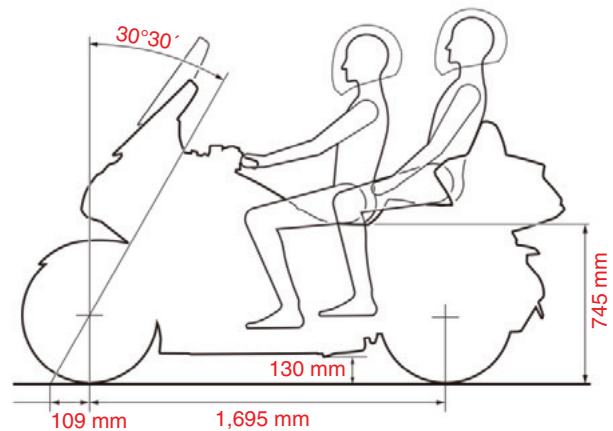


Fig. 13 Main dimensions

40 mm, and overall height by 25 mm from the previous model (comparison by the highest grade models with airbag for the USA).

3.2. Weight of Complete Vehicle

Aiming at a curb weight of complete vehicle less than the previous model by 30 kg, the target was set at realization of a weight as low as F6B (refers to the trunk-less type of previous model. Compared to the previous Gold Wing with trunk, curb weight less by 28 kg), on which anyone can sense the difference when riding, in the type with the trunk.

By means of the technologies mentioned later in section 3.3.2., the engine weight was reduced by 6.2 kg in the MT type and 3.8 kg in the DCT type. The weight was reduced by 18.2 kg in the body parts by means of the latter-mentioned technologies. In the electrical department, a total of 13.1 kg was reduced by the reduction of number of wire harness and sensors by the use of multiple communications, the combining of control units, the optimum layout of various parts, etc. In addition to those, the amount of fuel, oils, coolant, etc. were reviewed. Thus, the curb weight of complete vehicle has been reduced by 40 kg from the previous model (comparison by the highest grade models with airbag for the USA).

3.3. Packaging of Complete Vehicle

The packaging of complete vehicle and the newly-employed technologies are addressed in this section. These contributed to the downsizing and weight reduction of complete vehicle. Moreover, the weight distribution on the front wheel is increased to 46.5% from 44.9% in the previous model. The inertia moment of motorcycle is reduced by approximately 15% from the previous model both in the yaw and roll directions. Consequently, the motorcycle with the most enjoyable maneuverability

throughout all the models appeared in the chronicle of the Gold Wing was realized.

The packaging is shown in Fig. 15.

3.3.1. Reduced length and width of motorcycle by double wishbone-type front suspension

Generally in motorcycles, it is preferable to locate the engine as close to the front as possible to shorten the overall length. However, the mounting location of the engine depends on the relative position of the front wheel. When the motorcycle has a conventional telescopic type front suspension, the front wheel moves toward the upper rear part of the motorcycle as shown by the green lines in Fig. 16. Meanwhile in the newly developed double wishbone type suspension, the trajectory of front wheel stroke can be shifted to the more upright location by setting of the geometry of two links as shown by the red lines in Fig. 16. This suspension design allows 24 mm reduction of longitudinal travel of the front wheel when the front cushion is fully compressed. Consequently, the mounting location of

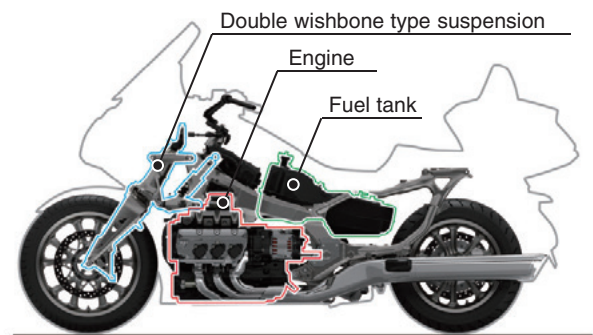


Fig. 15 Packaging

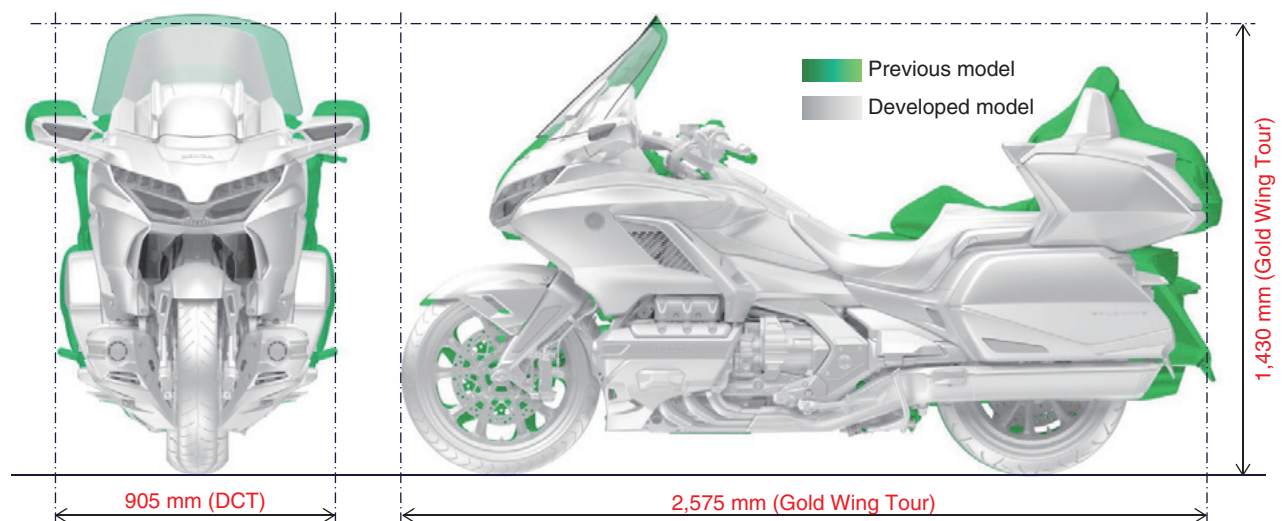


Fig. 14 Comparison of vehicle size

the engine was able to be moved forward.

The employment of the double wishbone type front suspension design also permits reduction of the width of complete vehicle. In the telescopic type front suspension, the right and left fork pipes that hold the front wheel in between turn around the steering axis when steering. Since no parts can be laid in the trajectory of fork pipe movement, the parts have to be located outside of the trajectory of two fork pipes. Meanwhile in the double wishbone type, as the stanchion above the fork that holds the front wheel in between can be a single shaft, the space necessary for turning of the steering can be reduced. Accordingly, the side radiators located outside of the steering space and the inner areas of the fairings that work as air guides were able to be gathered to the locations closer to the center of the motorcycle. Coupled with the latter-mentioned optimum layout and efficiency enhancement of the cooling system as discussed latter in Chapter 5, the maximum width of the fairing has been reduced by approximately 90 mm from the previous model.

3.3.2. Lightweight and compact design of engine and its mounting location

To contribute to realization of the target as the vehicle “a light weight and highly dense packaging,” the key goal of the engine design was lightweight and compact configuration. While there were factors that could increase weight such as the change from two-valve per cylinder to four-valve design, the employment of seven-speed DCT, the goal of weight reduction was set at 6 kg in the MT type, and 3.5 kg in the DCT type from the previous model. It was also targeted to reduce the longitudinal engine length from the front end of the front cover up to the rear end of the left cylinder head by 30 mm from the previous model to enhance maneuverability of the complete vehicle as well as shifting the riding position toward the front of motorcycle.

Measures for the engine weight reduction are such as the downsizing of basic engine dimension, the change from

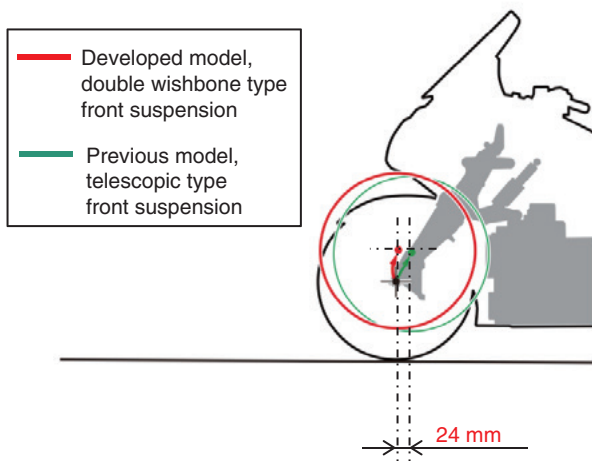


Fig. 16 Trajectories of front axle

the previous two throttles bore design to the single throttle bore type, the elimination of starter motor and related parts due to the employment of the ISG. The measures for weight reduction extend further to the abolishment of parts exclusively for reverse function which became unnecessary due to the addition of the walking speed mode in the DCT type. With all those measures taken, despite the weight increase factors mentioned above, the engine weight has been reduced by approximately 6.2 kg (manual transmission) and approximately 3.8 kg (DCT type).

The longitudinal length of the engine from the front end of the front cover up to the rear end of the left cylinder head has been reduced by 29 mm by the modification of cylinder bore from $\phi 74$ to $\phi 73$, the reduction of length of crankshaft by the application of high-strength alloy steel (Fig. 17).

3.3.3. Seating positions of rider and passenger

Generally in a motorcycle having horizontally-opposed cylinder engine, as the cylinder heads stick out sideways, how to maintain spaces for rider's feet is an issue. With the engine downsized and mounted toward the front, rider's seating position can be relocated by 36 mm toward the front from the previous model. And the relocation of rider and passenger more toward the front allows relocation of the panniers and the rear trunk, which are located in the rear end of motorcycle, more toward the front. As a result, the vehicle inertia moments are reduced while maintaining adequate space for the rider and the passenger. These contribute the further enhancement of stability from an increase of weight on the front wheel. Furthermore, sporty

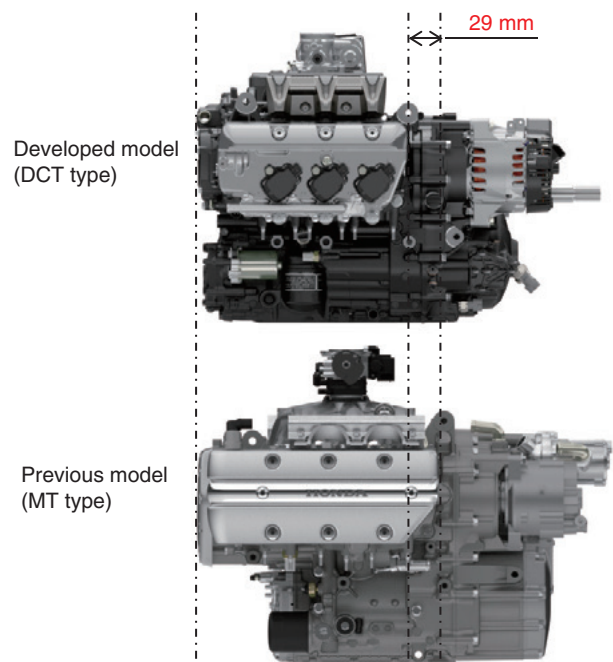


Fig. 17 Comparison of engine lengths between developed and previous models

maneuverability is realized by moving rider's seating position close to motorcycle's center of gravity. In addition, the rider's seating position located closer to the windscreen than the previous model allows reduction of windscreen size and enhancement of wind protection for the heads of the rider and the passenger. The details of wind protection are discussed later in Chapter 5.

In association with the frontward shifted seating configurations of the rider and the passenger, fulfilling both requirements for having the space for the airbag system installation and for keeping the space for fuel filler mouth installation became another issue. To deal with these issues, airbag module was newly developed for this model and its longitudinal length was shortened from 160 mm to 125 mm and its weight was reduced from 4.2 kg to 3.4 kg. As a result, this issue was overcome. The comparison of the riding position of this model to the previous model is shown in Fig. 18. The triangle in the picture indicates the riding position composed of the hip point, grip point and the foot peg point. In this model, while retaining the same basic triangle as the previous model, the hip point is relocated toward the front of motorcycle.

3.3.4. Fuel tank

The traditional under-seat fuel tank of previous Gold Wings is employed with an aim at a low center of gravity. The issue in the under-seat fuel tank design is how to maintain necessary fuel capacity while ensuring easy reach of rider's feet to the ground. Since the cruising range is very important for touring models such as Gold Wing, a target was set at improvement of fuel consumption in WMTC mode by 20% and reduction of fuel tank capacity to 21 liters, which is 4 liters less than the previous model, to maintain the same cruising range as the previous model.

The means to improve fuel consumption are the reduction of complete vehicle weight and aerodynamic drag

along with the employment of fuel-efficient engine design. Accordingly, the fuel consumption has been improved by 20% in the WMTC mode^{*1} and 35% in the 60 km/h constant speed mode^{*2} from the previous model.

*1 Comparison by the highest grade types with airbag for Europe

*2 Comparison by the highest grade types with airbag for Japan

The details of aerodynamic drag reduction are discussed in Chapter 5. The fuel-efficient engine design is discussed in the separate paper titled "Development of Horizontally-opposed Six-cylinder Engine for 2018 Gold Wing"⁽³⁾.

Similarly to the previous model, the multi-layer blow-molding method is applied to the plastic fuel tank to take advantages of a high degree of freedom in the designing of shape. The front part of tank is widened to allow natural knee grip for the rider. The part where the rider sits astride is narrowed for easy reach of rider's feet to the ground when stopped. With all the above measures taken, the weight of fuel tank as a unit has been reduced by 0.6 kg under 21-liter fuel capacity. The external view of fuel tank is shown in Fig. 19.

3.3.5. Cooling system

The challenge in the reduction of width of the fairing in front of the rider was how to achieve both the engine cooling performance and the rider protection from heat (prevention of entry of excessive hot air into the rider space and control of temperature increase.) The side radiator design having a radiator in both sides of the frame body was employed for the previous model. The cooling fan was located inside of the radiator core beside the frame body to keep the rider from hot air.

In this new model, the fin pitch of radiator core is reduced from the previous model. The cooling fan is located outside of the radiator core beside the frame body and housed in the bulged part of the fairing. With the said measures taken, the width of the fairing near the radiator

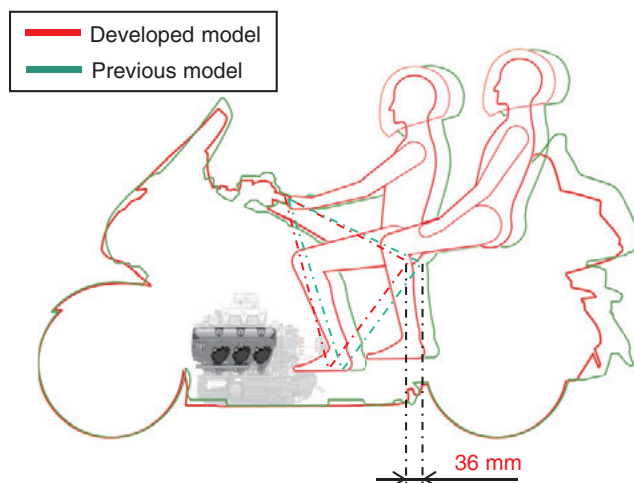


Fig. 18 Comparison of rider positions between developed and previous models



Fig. 19 Fuel tank assembly

has been shortened. Meanwhile regarding the potential rider comfort issues from heat when the cooling fans are located outside, the openings are provided in the front of fairing to take in fresh air to lower the temperature of air flowing to the rear by mixing the hot air from the radiator with the fresh air. In addition to it, the fairing shape near the rear of the radiator is designed so as to help prevent entry of the hot air after the radiator into the rider space. The air temperature distribution around the rider's and passenger's feet analyzed by Computational Fluid Dynamics (hereafter CFD) is shown in Fig. 20. It can be confirmed that the wind exhausted through the radiator flows outside the foot of the rider and the passenger.

Accordingly, the volume occupied by the radiator core (not including the retainers and attachment to frame body) has been reduced by 15% from the previous model while maintaining the same level of comfort as the previous model. Thus, the weight of the cooling system is reduced by 2.6 kg. The maximum width of the fairing in front of the rider is also reduced by approximately 90 mm from the previous model.

3.3.6. Luggage system

The capacities of the panniers and the rear trunk have been reviewed in this model. In the previous model, the 150 liter capacity was provided to allow carrying of luggage for two persons for a week-long trip presuming an across-the-continent tour. Taking into account the changes of travel environment and user-friendliness for riders and passengers in younger generations, the total capacity of the panniers and the rear trunk in this new model is set at 110 liters, which corresponds to the luggage for a three-nights-and-four-days travel. In both of the panniers and the rear trunk, in order to achieve high-quality image and sturdiness when opening, closing or touching, the outer and inner two-panel construction is employed just like in the previous model, and the overlapped area and thickness are totally reviewed. Thus, a total of 6.2 kg has been reduced from the panniers and the rear trunk.

Also, considering the need of having more luggage capacity, the trunk rack having a larger size and loading capacity than the previous model has been developed as an accessory. For the model without the trunk, the carrier that

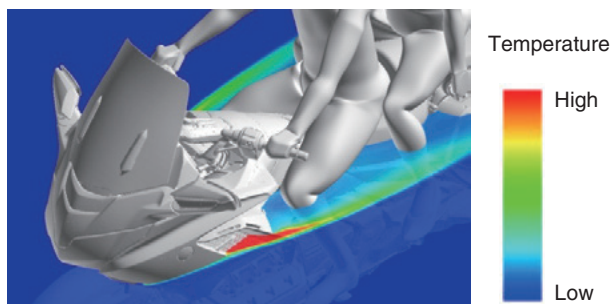


Fig. 20 Result of thermal distribution analysis by CFD

can be mounted on the cowl in the rear of the seat has been developed as an accessory.

3.4. Rider Comfort and Riding Position

3.4.1. Handlebar position

In the previous model, the target was set to achieve both a relaxed riding position suitable for a tourer motorcycle and easy handling while employing the telescopic type front suspension. Accordingly, the longitudinal offset between the front suspension pivot axis and the handlebar grip points was increased, and the space between the handlebar grips was increased in the sideways direction. Consequently the travel of the grip points was increased when the handlebar was turned.

Due to the employment of the double wishbone type front suspension in this model, where steering axis and handle bar pivot axis are separated, the degree of freedom of handle bar pivot axis position has been increased, allowing handling bar pivot axis to be positioned closer to the rider. Also, the grip points were relocated upwards by approximately 20 mm, the space between the handlebar grips in the sideways direction was reduced by 10 mm, and the hand grip angle was increased to make the rider feel more relaxed in his upper arms. This makes the grip points turn at a smaller turning radius than the previous model, which makes turning of the handlebar easy. The handle movable extents are compared in Fig. 21.

This construction enables downsizing of the handlebar and reduction of turning radius, contributing to the compact and slim appearance of the motorcycle.

3.4.2. Seat contour

In the previous model, the seat was contoured in such

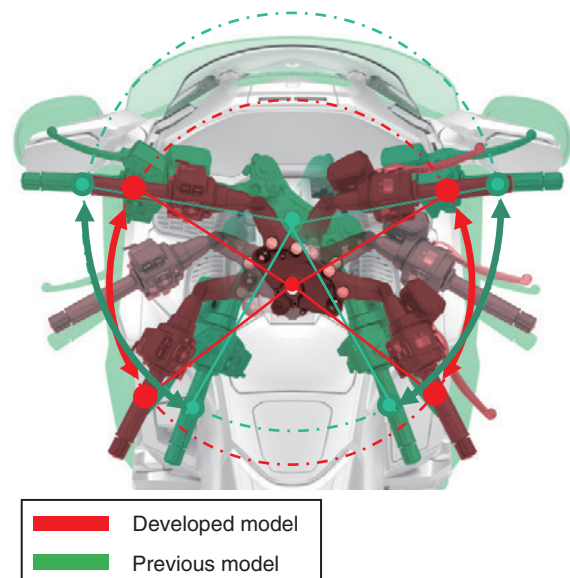


Fig. 21 Comparison of handle movable extents between developed and previous models

a manner that the space for the rider was separated from that of the passengers so that the rider could enjoy control of motorcycle with comfort and the passenger could enjoy comfortable touring. Maintaining the same concept, the basic seat contour of the previous model has been applied to this model. The seat contours of this model and the previous model are shown in Fig. 22.

In this new model, the bucket behind rider's hips is eliminated to allow active control of the motorcycle by shifting weight of rider. With the seat back contour modified, rider's shifting of hips for feet to reach the ground, which used to be necessary in the previous model, is no longer necessary. Coupled with the earlier mentioned reduction of fuel tank capacity, this contributes to easy reaching of the feet to the ground. The appearance is changed from the soft image of the previous model to the modern and edgy image to match the styling of this model.

The seat appearance shapes are shown in Fig. 22.

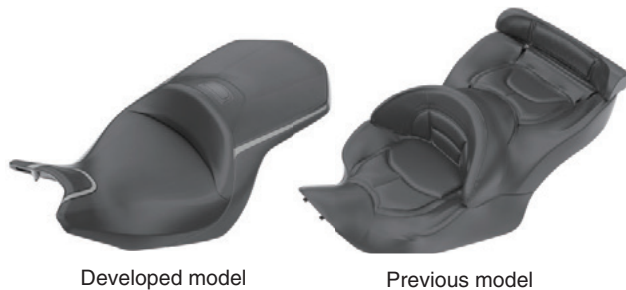


Fig. 22 Comparison of seat shapes

4. Chassis

4.1. Frame Body

While maintaining the traditional twin-tube aluminum frame design, the frame body has been redesigned with an aim at both stability and agility. In the newly designed main frame structure, as well as applying the thin aluminum die cast technologies to all parts, the component parts are upsized and integrated, reducing the number of component parts from 23 pieces in the previous model to 4 pieces. Accordingly, the number of welding of the component parts is reduced and the total welding length is reduced from 8.0 m to 3.6 m, which allows automated welding and contributes to weight reduction, high productivity and consistent quality. The application of thin aluminum die cast method allows fine adjustment of thickness necessary for each portion. The employment of the new swing arm pivot construction to be discussed in the following Section 4.2. allows frame body weight reduction by reducing the number of cross pipes, which used to be laid above and under the pivot axis.

In the rear shock unit, the pillow ball is employed in

the top and bottom pivots to absorb torsional deformation of Pro Arm component parts for enhancement of riding comfort. Coupled with the employment of the earlier mentioned double wishbone type front suspension, the smooth functioning of front and rear suspensions allows reduction of frame body stiffness from the previous model.

Thus, the frame body weight has been reduced by 4 kg while enhancing the maneuverability and the riding comfort of the motorcycle.

The frame body structure is shown in Fig. 23 and the rear cushion in Fig. 24.

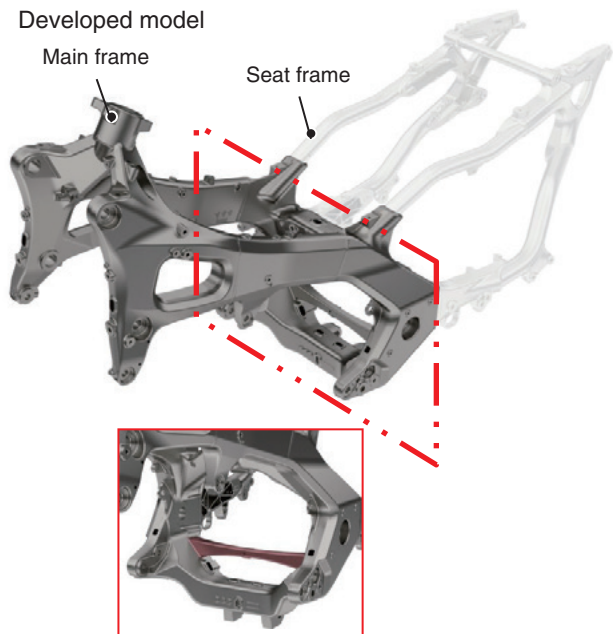


Fig. 23 Comparison of frame bodies

4.2. Swing Arm

The swing arm pivot is redesigned to reduce weight and enhance handling stability and riding comfort while maintaining the same single-sided arm construction containing the drive shaft in the right side of the motorcycle.

In the previous model, the swing arm was held by the taper roller bearings in the right and left sides. Due to the said construction, the frame body had to be stiff to withstand the preload necessary for the bearings, causing an increase of weight. In this model, the ball bearing used only in the left side locates the swing arm onto the frame body, and the needle bearing is used in the right side.

This design eliminates influence of axial retaining force of the right side pivot to the stiffness on the frame body side, enhancing degree of freedom of frame body design.

By the optimum stiffness in the frame body side from all those modifications, the handling stability and supple riding quality have been realized.

The construction of pivot is shown in Fig. 25.



Fig. 24 External view of rear cushion

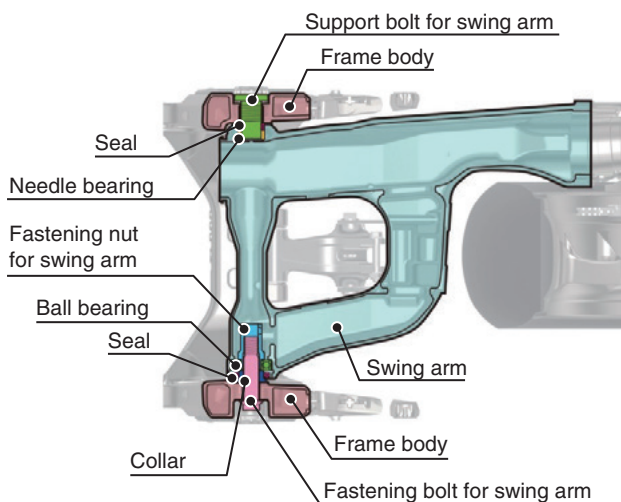


Fig. 25 Structure of swing arm pivot shaft

5. Air Management

The challenge in the air management department in this new model development was how to achieve all of the performance targets, such as the comfort of rider and passenger (wind protection, heat management), fuel consumption (reduction of aerodynamics drag), enhanced maneuverability, and compact appearance.

Generally, the streamline bordering the inside and outside of the apace protected by the wind screen draws an arc, reaches to the apex near the head of the rider, and gradually comes down. Accordingly, even when the screen size is the same, the wind protection performance is less affected by the lowering of the streamline when the wind screen is closer to the rider. In this model, both the rider and passenger seating positions are moved to the front of motorcycle by 36 mm from the previous model to maintain wind protection for the rider and the passenger.

Moreover in this model, with the employment of the electrically adjustable wind screen and the air ducts, the air is positively introduced to inside of the wind-protected area to allow the rider enjoy “nice breeze” unique in motorcycle riding.

Thus the target level of comfort has been achieved while realizing the compactness, which reduces aerodynamic drag coefficient by 13% from the previous model, contributing to the enhancement of fuel consumption and maneuverability.

Coupled with the fuel consumption improvement of the engine, the fuel consumption is improved as mentioned in Section 3.3.4. In addition to it, the weight of the fairing is reduced by 2.4 kg.

5.1. Electrically Controlled Wind Screen

The wind screen was manually adjustable in the previous model. The motor-driven, electrically controlled wind screen is employed in this model. This system provides continuously adjustable screen height by the handlebar switch to fit to the situations and speeds at running in town or on freeway. Also, the round shape of slide rail allows simultaneous altering of height and angle of the wind screen, allowing more changes of stream lines from a certain length of screen travel. Thus, the compact and light weight system has been achieved.

The trace of wind screen travel is shown in Fig. 26.

5.2. Air Management Analysis by CFD

In this new model development, the air management was analyzed by CFD.

The steady-state analysis is generally used to analyze the streamlines that border the inside and outside of the wind-protected zone. This method is also used in this new model development to clarify wind-protected zone. Meanwhile in actual riding, the air pressure fluctuates near the border due to the turbulence, causing such a phenomenon that the rider’s head is shaken side to side or longitudinal directions in some cases. As the steady-state

analysis does not permit reproducing of fluctuations of flow speed and/or air pressure, it used to be unlikely to analyze the phenomenon. Thereby in this product development, the unsteady-state analysis was introduced in addition to the steady-state analysis⁽⁷⁾, to visualize the pressure fluctuations as a physical quantity.

The distribution of average flow speed determined by the steady-state analysis is shown in Fig. 27 (a) and (b). The screen height was set at the highest within the



Fig. 26 External view of electronically controlled wind screen

adjustable range in all cases.

As shown in the picture (a), the border is above the helmets of the rider and the passenger in this new model, indicating the space for the rider and the passenger is within the wind-protected zone. Similarly in the previous model as shown by the picture (b), the border was also above the helmets of the rider and the passenger, indicating no significant difference in the wind-protected zone.

The distribution of pressure fluctuations determined by the non-steady-state analysis is shown in Fig. 27 (c) and (d).

As the comparison of pictures (c) and (d) indicates, the pressure fluctuations are less in this model than in the previous model especially at passenger's head. Actual wind tunnel tests and riding tests revealed that shaking of rider's head was less in this model than in the previous model. As such, the analysis of correlation between the simulation and the rider feedback from actual riding test allows enhancement of simulation accuracy and proper designing of details of shape.

With all the measures taken, the screen width of this model at the widest part is reduced by 185 mm from the previous model, the height from the ground up to the upper end of screen is reduced by 25 mm (Comparison at the lowest position within the adjustable range) contributing to the compactness and the comfortableness necessary for a tourer motorcycle.

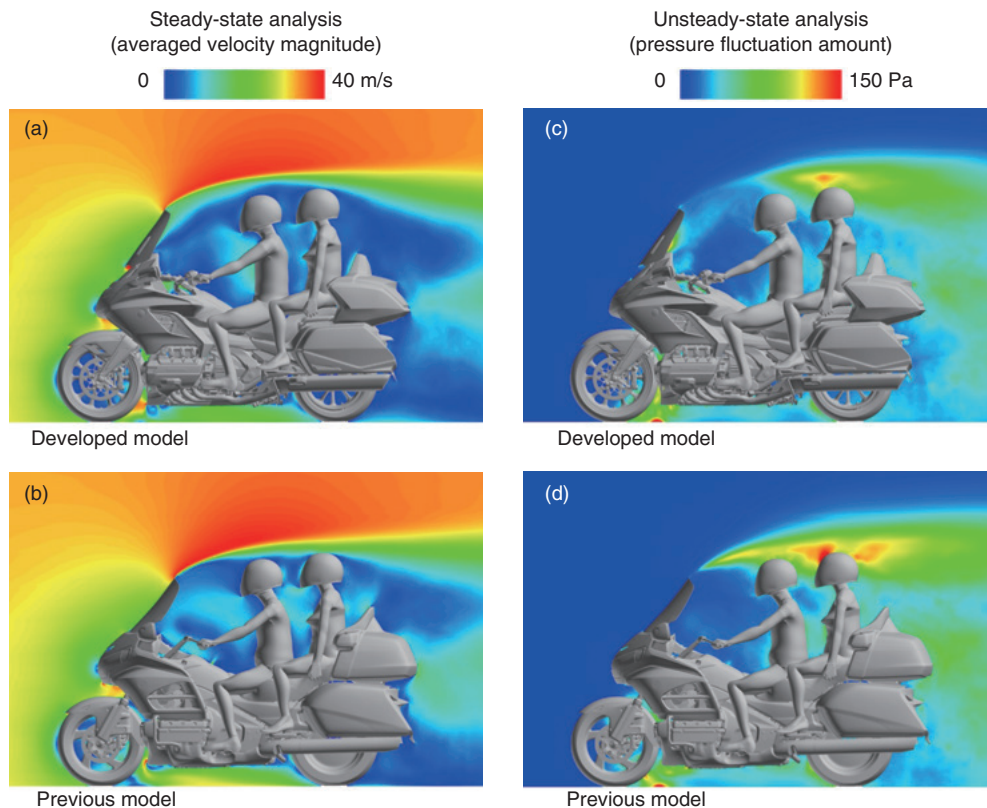


Fig. 27 Results of verification for wind protection performance by CFD analysis

6. Conclusion

In the newly developed Gold Wing, a lot of latest technology is introduced while maintaining the legacy and tradition continued in the Gold Wings of past generations. The curb weight of complete vehicle is reduced by 40 kg. Regarding the compactness of complete vehicle, the overall length is less by 55 mm, the width by 40 mm, and the height by 25 mm from the previous model.

Thus created is the modern-edition premium grand tourer that the owner can enjoy together with the passenger a high quality travel full of “fun to ride, fun to control,” which is the essential attractiveness of a motorcycle.

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
- (2) Hoshi N., Shimura Y., Okuma T., Nomura A., Goto K.: Introduction of 4-stroke motocrosser CRF450, Honda R&D Technical Review, Vol.14 No.1, p.21-30
- (3) Yokota, H., Nojima, S., Sonoda, Y., Osaki, Y., Kontani, M., Yamasaki, S.: Development of Horizontally-opposed Six-cylinder Engine for 2018 Gold Wing, Honda R&D Technical Review, Vol.30 No.2, p.16-30
- (4) Kusuda A., Yokogawa Y., Nakai K.: Development of Ride Modes for Various Conditions for 2018 Gold Wing, Honda R&D Technical Review, Vol.30 No.2, p.38-46
- (5) Hirayama T., Yokogawa Y., Fujimoto Y.: Development of 7-speed Dual Clutch Transmission for 2018 Gold Wing, Honda R&D Technical Review, Vol.30 No.2, p.31-37
- (6) Kuwabara N., Mishiro J.: Development of Double Wishbone Type Front Suspension for 2018 Gold Wing, Honda R&D Technical Review, Vol.30 No.2, p.47-52
- (7) Nishiwaki D., Ozawa H.: Analysis of non-steady-state flow that causes head shake of motorcycle rider, JSAE 2018 spring convention, technical paper presentation, No.15-18, p.1-5 (2018) (in Japanese)

■ Author ■



Kihoko KAITA



Yutaka NAKANISHI

Development of Horizontally-opposed Six-cylinder Engine for 2018 Gold Wing

Hiroshi YOKOTA*

Satoru NOJIMA*

Yutaka SONODA*

Yusuke OSAKI*

Masahiro KONTANI*

Satoru YAMASAKI*

ABSTRACT

On the basis of the concept of new Gold Wing, the new horizontally-opposed six-cylinder power unit was developed with the theme “smooth yet aggressive at times, light weight and compact flat-six power unit.” To achieve weight reduction and compactness of the power unit, the materials and the shapes of the crankshaft and the connecting rods were redesigned. To realize lightweight and compact power unit, the Unicam valve train having finger follower rocker arms for the intake valves and the single throttle body were applied. To enhance combustion efficiency, the combustion chamber was changed from bathtub type for two valves, to the pent roof type for four valves. In addition, the compression ratio was increased and the intake ports were applied with a shape generating a strong tumbling of intake air. To enhance maneuvering qualities, which was derived from the set concept of the developed motorcycle, a throttle-by-wire system was adopted. To the drive system of the Integrated Starter Generator, a damper of revised configuration and a rubber coupling were implemented and the requirements for tranquil riding were fulfilled. A comfortable exhaust sound was also provided by reassessing the design of the exhaust system. With all the measures mentioned above, weight reduction, together with downsizing, was successfully achieved by 7.1 kg for the DCT equipped model and by 9.5 kg for the MT equipped model compared to the previous model. Furthermore, the peak output power was increased by 6 kW and, yet, the fuel economy was improved by 24% (in the case of the model sold in Japanese market) in the WMTC mode (class 3-2). As mentioned above, the enjoyably deep and pulsated exhaust sound was also achieved while offering the mild and high quality sounds of mechanical operations.

1. Introduction

Ever since the first introduction of the GL1000 in 1975, the Gold Wing evolved and its model lineup was strengthened as Honda's grand tourer model for long-distance touring. From the first model, the horizontally-opposed engine configuration has been applied featuring low center of gravity, small vibration and lower mechanical noise. In the early four cylinder engines, the displacement was increased from 1000 cc to 1100 cc, then to 1200 cc. While modifying further to six cylinders, 1500 cc, and to 1800 cc, the horizontally-opposed engine configuration stays alive for more than 40 years.

In this paper, the horizontally-opposed six-cylinder (hereafter F6) power unit for the sixth generation Gold Wing - GL 1800 is introduced. This model has been totally updated and its mass production started in 2018. This power unit is based on the former F6 engine, the only

one in motorcycles, with modifications applied to reduce weight and achieve compactness by applying advanced technologies such as the latest valve train mechanism, the Throttle by Wire (hereafter TBW), the Integrated Starter Generator (hereafter ISG), the Dual Clutch Transmission (hereafter DCT), etc. Compared to the previous model, the power output is increased, the throttle response is enhanced as well as improving the fuel consumption. With its deep and enjoyably pulsated exhaust sound along with the lower mechanical noise, the new power unit is intended to be “fun to ride” for the rider.

The power unit comes in two types; the seven-speed DCT type and the six-speed manual transmission (hereafter MT) type. Unless otherwise mentioned, this paper refers to the DCT type. The details of DCT are discussed in the separate paper titled “Development of 7-speed Dual Clutch Transmission for 2018 Gold Wing”⁽¹⁾.

* Motorcycle R&D Center

2. Goals of Power Unit Development

Considering changes of the market and the customer demands, the target of complete motorcycle was set at a leap from the gorgeous & luxurious, massive and heavy line and to regain youthfulness, aiming at a “fun to ride, fun to control” motorcycle. Keeping that in mind, the new power unit has been developed with the theme “smooth yet aggressive at times, light-weight and compact flat-six power unit.” The major goals were set as per the following.

- 1) Reduce weight by 6.5 kg in DCT type, 9.0 kg in MT type
- 2) Reduce length from the front of engine to the rear end of the left cylinder head by 30 mm
- 3) Same or more power output as the previous model. Improve fuel consumption by 20% in WMTC mode (class 3-2)
- 4) Deep and pulsated exhaust sound and high-quality engine start sound

To contribute to the concept of new Gold Wing “light-weight and highly dense packaging that allows the rider to fully use the motorcycle in various conditions,” the key goal of the power unit was set at light weight and compactness. Despite the valve train modifications from the two-valve to the four-valve and the employment of seven-speed DCT, the target weight reduction of power unit was set at 6.5 kg in DCT type, 9.0 kg in MT type from target of complete motorcycle.

To shift the riding position further to the front of motorcycle to enhance maneuverability, a target was set to reduce the length from the front of engine to the rear end of left cylinder head by 30 mm.

While maintaining and evolving such features of the previous model as the adequate torque in low engine speeds, strong acceleration, low vibrations and quietness, a target was set to improve fuel consumption by 20% in the WMTC mode (class 3-2).

It was also aimed to realize deep and pulsated exhaust sound appropriate for a motorcycle with a large displacement engine along with high-quality engine start sounds suitable for a top-end model.

The overview of the power unit is shown in Fig. 1.



Fig. 1 Overview of power unit

Table 1 Power unit specifications

	Developed model		Previous model
	DCT type	MT type	
Cylinder arrangement	F6-cylinder		F6-cylinder
Bore x Stroke (mm)	73 x 73		74 x 71
Displacement (cm ³)	1833		1832
Compression ratio	10.5 : 1		9.8 : 1
Maximum power	94 kW / 5500 rpm		88 kW / 5500 rpm
Maximum torque	171 Nm / 4500 rpm		169 Nm / 4000 rpm
Valve train	Unicam - 4 valve		SOHC - 2 valve
Combustion chamber configuration	Pent roof		Bathtub
Diameter (mm) / Number of throttle body	φ 50 / 1		φ 40 / 2
Generator	Integrated Starter Generator (ISG)		Dedicated generator
Starter			Dedicated motor
Transmission	7-speed DCT	6-speed MT	5-speed MT
Reverse system	Engine drive	Motor drive	Motor drive
Engine weight (kg)	114.3	111.9	118.1
Engine length x width x height (mm)	617 x 702 x 503		633 x 706 x 513

*The engine weight includes the intake system, starting system, charging system and the transmission. The intake system consists of the throttle body and the downstream parts. Exhaust system not included.

*The engine length does not include the final shaft.

3. Power Unit Specifications and Other Major Specifications

Major specifications of the new Gold Wing (North America type) and the previous model are compared in Table 1.

While maintaining the same longitudinal, water-cooled 4-stroke, F6 configuration and 1800 cc displacement from the previous model, the engine has been totally redesigned. For weight reduction and compactness, the bore/stroke and the number of throttle bodies are modified. To enhance combustion efficiency, the compression ratio is raised, and the combustion chamber is redesigned from the two-valve bathtub type to the four-valve, pent-roof type. For easy operation of motorcycle, the seven-speed DCT type is provided in addition to the six-speed MT type. Moreover, the ISG and the engine-driven reverse system are newly employed.

4. Basic Engine Configuration

4.1. Crankshaft and Connecting Rod

To enhance motorcycle maneuverability and rider comfort, the longitudinal length of the engine was reduced. The crank shaft is compared to the previous model in Fig. 2.

As well as changing the bore from previous model's

$\phi 74$ mm to $\phi 73$ mm, the crankshaft material was changed from the steel with soft nitriding to the alloy steel having a high internal hardness. With the change of material and the choice of optimum shapes, the crank journal width was reduced by 1 mm, the crank pin width by 2 mm, and the crank web width by 1~2 mm, thus reducing the overall length of crank shaft (the length between the front and rear journals) was reduced by 22.6 mm. Consequently the length between cylinders (bore pitch) was reduced by 9 mm, and the right and left cylinder offset by 4 mm. The crank pin diameter was increased by 4 mm to deal with the increase of contact pressure in the connecting rod big end bearing from the reduced crank pin width and the lowering of crankshaft stiffness from the increased stroke. The crank pulse rotor was relocated from the front of crankshaft in the previous model to the rear of crankshaft, reducing the length from the engine front cover to the crankshaft by approximately 7 mm.

With all these measures taken, the length from the front of engine up to the rear end of left cylinder head was reduced by 29 mm from the previous model as shown in Fig. 3, allowing shifting of riding position more to the front of motorcycle to increase weight on the front wheel while maintaining rider's leg spaces, contributing to an enhancement of maneuverability of the motorcycle.

Regarding the connecting rod, the length was reduced by 7 mm by the change of material from the microalloyed steel to the high-strength carburized steel to help prevent increase of engine width from the increase of stroke as well as reducing the weight. Regarding the tightening of the rod and the cap, the nut-less design was applied using M6 bolt tightened in the plastic region while the previous model uses M8 bolt tightened in the elastic region. This design contributed to the above-mentioned reduction of crank pin

width. The connecting rod is compared to the previous model in Fig. 4.

With these modifications applied, the total weight of the crankshaft and six connecting rods was reduced by 1.1 kg.

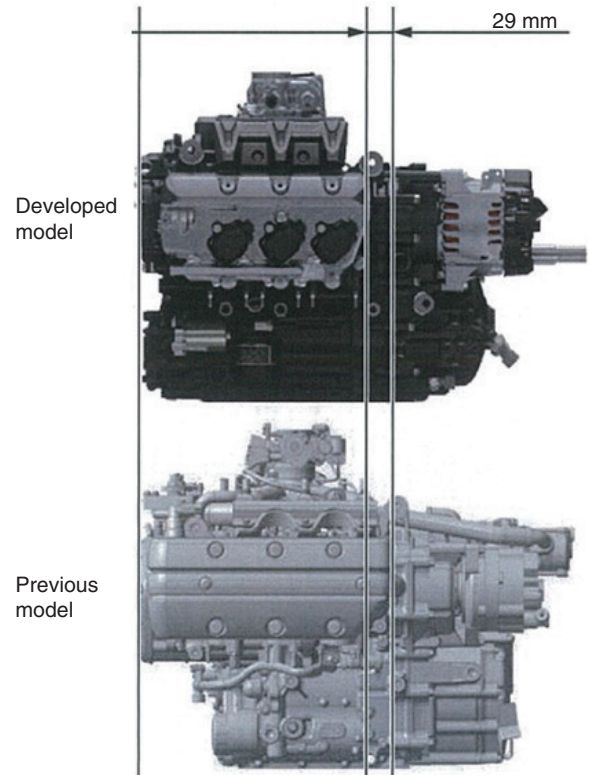


Fig. 3 Left view of power unit

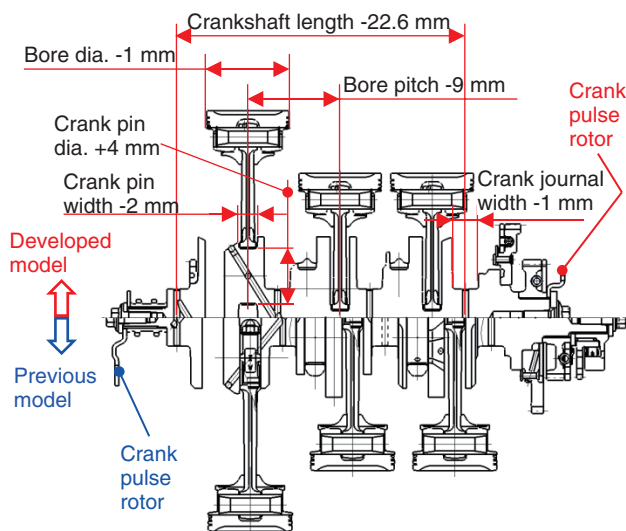


Fig. 2 Crank shaft (comparison in the left cylinder)

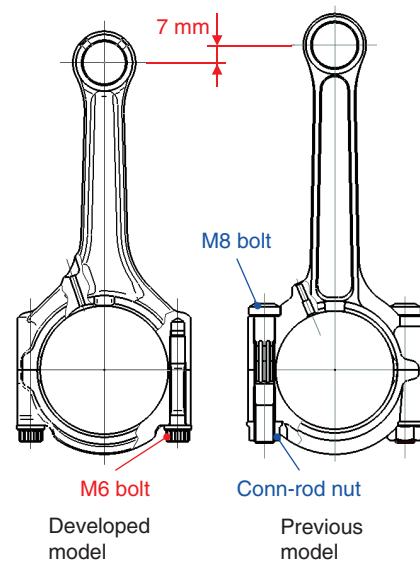


Fig. 4 Connecting rod

4.2. Valve Train and Cylinder Head

The Unicam valve train having the finger follower rocker arms in the intake side was applied for weight reduction and compactness. The construction is shown in Fig. 5.

To maintain adequate lean angle of the vehicle, which is an inherent issue in a horizontally-opposed engine, a single camshaft is laid in the upper part (intake side) just as in the previous model, applying the Unicam design in which the exhaust valve is activated by the roller rocker arm to deal with the modification from the two-valve to the four-valve design. With the finger follower rocker arm applied in the intake side, the valve lifter holding part, which is located in the intake side of cylinder head in the previous model or conventional Unicam system⁽²⁾, is eliminated. Thus, the total weight of right and left cylinder heads is reduced by 2.8 kg.

Furthermore, several weight reductions in the valve train system, such as applying the coaxial shaft which supports the inlet and exhaust rocker arms commonly, employing the thin-wall camshaft by using the high-strength material, removing material from behind the cam lobe, were conducted. With all these measures taken, a total of 1.3 kg was reduced from the valve train system despite the weight-increasing factors such as the change from the two-valve design to the four-valve system, and the addition of rocker arm parts.

4.3. Engine Weight Reduction

In addition to the measures mentioned in Section 4.1. and 4.2., the engine weight was reduced by various other means in the entire engine. Some typical examples are discussed hereafter.

The change of cylinder sleeve material from the cast iron to the composite aluminum material⁽³⁾ reduced 2.2 kg. The modified parts layout in the cooling system reduced 0.7 kg. The elimination of starter motor and related parts in the starting system and the charging system from the application of the ISG reduced 2.4 kg (discussed in Chapter 5). The

modification of the throttle body from the twin throttle type to the single throttle type reduced 1.1 kg (discussed in Chapter 7). The application of the transmission construction integrating the reverse system⁽¹⁾ allowed elimination of reverse-related parts, reducing 1.3 kg (DCT type only).

With all these measures taken, the engine weight was reduced by 3.8 kg in the DCT type from the previous model despite the weight-increase from the application of DCT and the increased number of speed from five to seven. In the MT type, 6.2 kg was saved despite the weight increase from the change to six speeds from the five speeds.

* The engine weight includes the parts in the intake system in and under the throttle body, those in the starting and charging systems, and the transmission.

4.4. Main Bearing Cap

In the Gold Wing, the iron-based sintered material was employed in the main bearing cap to reduce crankshaft knocking noises since the previous model from which the six-cylinder engine became 1800 cc⁽⁴⁾. Due to the increase of power output in the new model from the previous model, there was an issue of fretting in the bearing cap mating surfaces on the crankcase of this model.

To deal with this issue, we focused our attention on the reduction of stress amplitude on the bearing cap mating surfaces, the microscopically small slip occurring on the mating surfaces (hereafter as relative slip), which are the main causes of fretting fatigue⁽⁵⁾. It was also considered that an equalization of surface pressure distribution on the contact surfaces and a prevention of lowering of surface pressure in the outside of the bearing cap at the time of load application would be effective to reduce the said stress amplitude and the slip. It was based on such a consideration that an excessively small surface pressure tends to increase relative slip.

Effective measures to deal with the said issues include an increase of retaining stiffness such as an increase of bearing cap stiffness, use of four retainer bolts and/or an increase of journal bolt pitch. Any of such measures, however, causes an increase of engine weight and size. Accordingly, taking into account achieving weight reduction and compactness, the construction in which the bearing cap pressed in the crankcase was applied. The construction is shown in Fig. 6.

As well as press-fitting the outsides of bearing cap into the crankcase, the journal bolt was downsized from M12 to M10 by applying the high-strength material. Furthermore, the bolt pitch was increased to equalize pressures on the mating surfaces.

The analyzed surface pressure distribution on the bearing cap mating surfaces of the crankcase at the time of tightening and when the load is applied is shown in Fig. 7. The diagrams indicate surface pressure distribution on the mating surface "A" in the lower side of the crankcase shown in Fig. 6. The data at the time of load application are those taken when the relative slip in the outside is at its

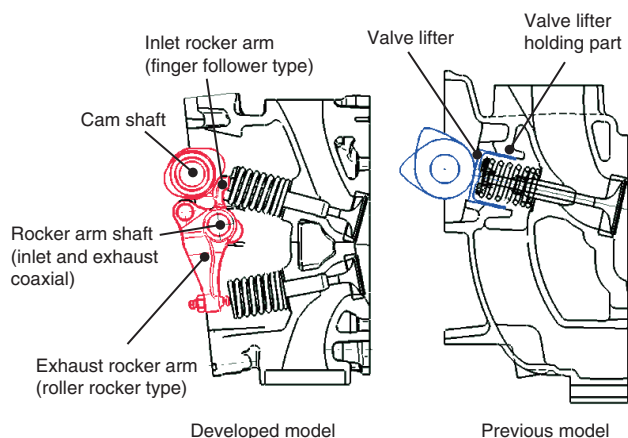


Fig. 5 Cross section of valve system

maximum. In the previous model, the surface pressure is high in the inside (journal side) and low in the outside when tightened. When the load is applied, the surface pressure lowers significantly near the outside end of the bearing cap. With the press-fitting applied, the surface pressure at the time of tightening is equalized, and the lowering of surface pressure near the outside end at the time of load application is no longer apparent.

It should be noted that while the lowering of surface pressure is seen in the inside (journal side) when the load is applied, as the relative slip in the inside is small in this bearing cap retaining design, the fretting is no longer a concern nor seen in the actual test engine.

The stress amplitude and the relative slip on the bearing cap mating surface of the crankcase are shown in Fig. 8. The stress amplitude is compared at the maximum part in the entire mating surface. Compared to the previous model, the stress amplitude and the relative slip are reduced.

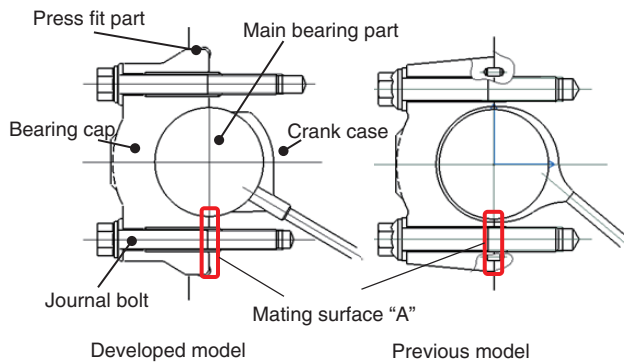


Fig. 6 Bearing cap

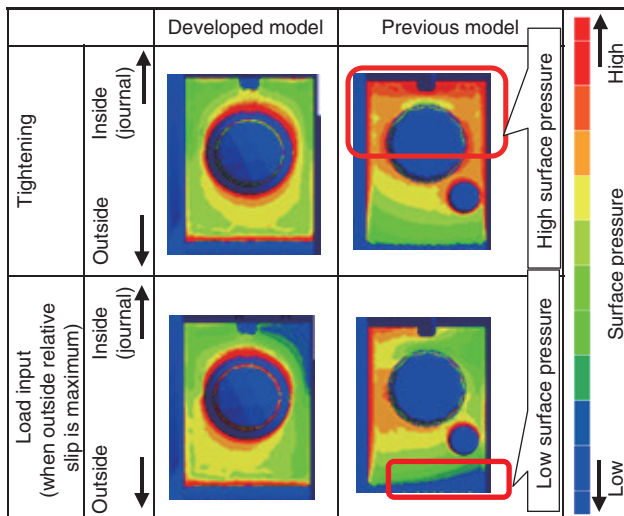
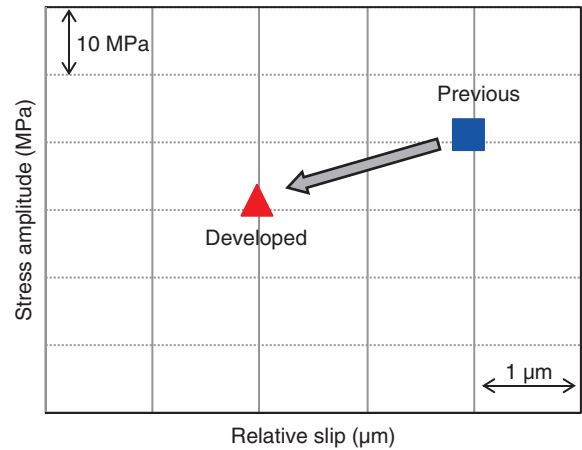


Fig. 7 Results of crank case surface pressure analysis

With all these measures taken, there is no fretting in the mating surface of the test engine, contributing to the weight reduction and compactness of the engine.



Analysis condition

Journal bolt axis power: lower limit

Combustion pressure: maximum

Press fit interference: center

Fig. 8 Results of stress and relative slip analysis

5. Integrated Starter Generator (ISG)

5.1. Mechanical Construction

In the development of the engine starting system and the charging system of the new Goldwing, the highest priority was given to weight reduction and lower mechanical noise.

To achieve the goals, the ISG, which performs as a starter motor and an alternator, was first applied in a large motorcycle. The shape of ISG is shown in Fig. 9. In the previous model, the starter motor was used in the starting system and the excitation-type alternator was used in the charging system separately. The employment of ISG permitted elimination of the starter motor, starting reduction gear, the starter magnetic switch and related electric wires, etc. In that way, the weight in the starting and charging systems was reduced by 2.4 kg (-20%).

The construction of ISG and its drive train of this model are shown in Fig. 10. The alternator, starter motor and their drive train of the previous model are shown in Fig. 11.

5.2. Starting and Charging Performance

The ISG is already applied widely in automobiles. When the ISG for automobiles is applied as it is, an electric current 2.2 times more than the previous model will be required at the time of starting, requiring a larger battery than the previous model.

To deal with the said issue, the electrical control system



Fig. 9 Shape of ISG for developed Gold Wing

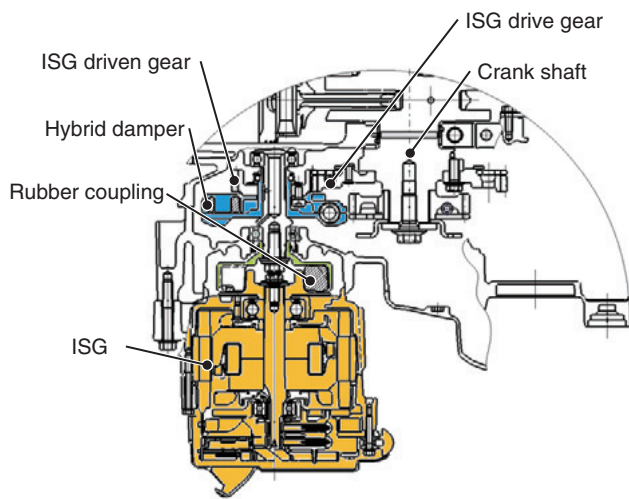


Fig. 10 Gear and couplings to drive ISG

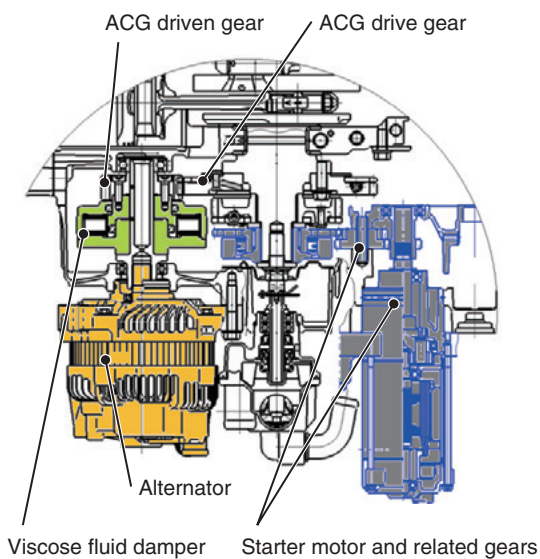


Fig. 11 Gears and couplings to drive starter motor and alternator

for starting was reviewed and the optimized, allowing reduction of required starting current to less than 1.5 times of the previous model. Thus the starting performance required for this model was satisfied without modifying the capacity and size of the battery.

Meanwhile in the charging system, the alternator capacity necessary for this model is approximately 1.2 times more than the previous model. However, as the rated alternator output of the ISG is approximately 1.7 times more than the previous model, the power exceeds the allowance of the component parts in the charging system such as the wire harness, fuses, etc. used in the previous model. This causes an increase of weight from use of large diameter wires, etc. and requires redesigning of fuses and fuse box.

To obtain alternator output 1.2 times as much as the previous model, the cooperative control was incorporated to maintain the exciting current of ISG at an appropriate level at each engine speed by commands from the PGM-FI unit. The output current characteristics of the ISG of this model and the alternator of the previous model are shown in Fig. 12. While obtaining approximately 2.9 times (a in the graph) more power output than the previous model near the idling, it is approximately 1.2 times near 4000 rpm (b in the graph), helping ensure steady power output in the wide engine speed range.

5.3. Drive System and Noises

While the ISG in automobiles is usually belt-driven, the direct drive via the gears is applied in this model. As the characteristics of a motorcycle having a longitudinal engine, the roll moment that shakes the motorcycle right and left occurs when the spinning of crankshaft is accelerated or decelerated. These shaking motions are uncomfortable for the rider when the motorcycle is standing still. To cancel the roll moment, the gear drive system was applied so that the ISG spins in the opposite direction of the crankshaft.

The helical gears were applied to drive the ISG to further reduce the gear whine noise of the previous model

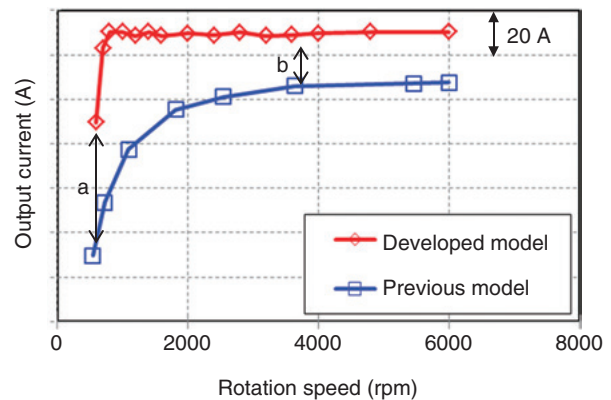


Fig. 12 Characteristics of output current to rotation speed

in which the alternator was driven by spur gears, thus enhancing high-quality feel. The comparison of noise level of the helical gear and the spur gear is shown in Fig. 13. The horizontal axis refers to the engine speed. The vertical axis represents the sound pressure measured at rider's ears. The solid lines indicate overall sound pressure level. The dotted lines indicate the sound pressure level of the order component (first order) of ISG gear meshing. In the spur gear, the order component sound pressure of ISG gear is especially high near 1700 rpm (corresponds to 80 km/h in top gear), lifting the overall sound pressure level. Meanwhile in the helical gear, there is no surge of

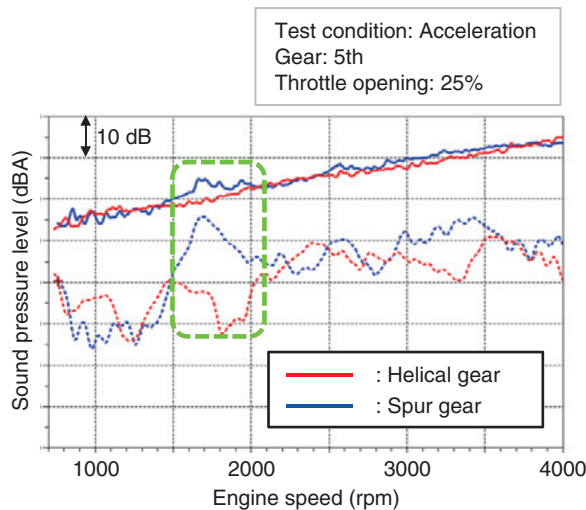


Fig. 13 Comparison of ISG gear noise

gear noise that lifts the overall sound pressure level, thus enhancing the high quality feel.

In the ISG drive train, a shock absorbing device (hereafter as damper) was necessary to reduce gear rush noise from crankshaft rotational variation. The ISG starts the engine in addition to power generation. In the viscous damper applied in the previous model, as slippage occurred from the cranking torque, it did not allow starting of the engine. In the new model, the newly-designed damper composing of the spring and the rubber was applied along with the rubber coupling. The construction of new damper is shown in Fig. 14.

The newly-designed damper is composed of the arc spring that maintains the damping stroke and the highly impact- absorbing damper rubber. The application of this construction helps ensure necessary working angle of damper and the maximum load capacity in the limited space. The rubber coupling is applied for easy coupling and reduction of coupling rush noises. As the drive shaft and the alternator were connected by the spline without using coupling in the previous model, noises occurred from the backrush in the spline connection. The application of rubber coupling eliminates the factor in occurring of rush noise.

The use of ISG for starting of engine reduces engine start noise from the previous model. In the previous model, as the starter motor torque was much lower than the required cranking torque, a large reduction gear ratio was required (total reduction ratio 28.3, internal reduction ratio 6.4).

Meanwhile in the ISG, since the power output as a starter motor was increased to approximately 4.4 times more than the previous model, the reduction gear ratio was lowered to 2.5. Accordingly, the rotating speed of ISG

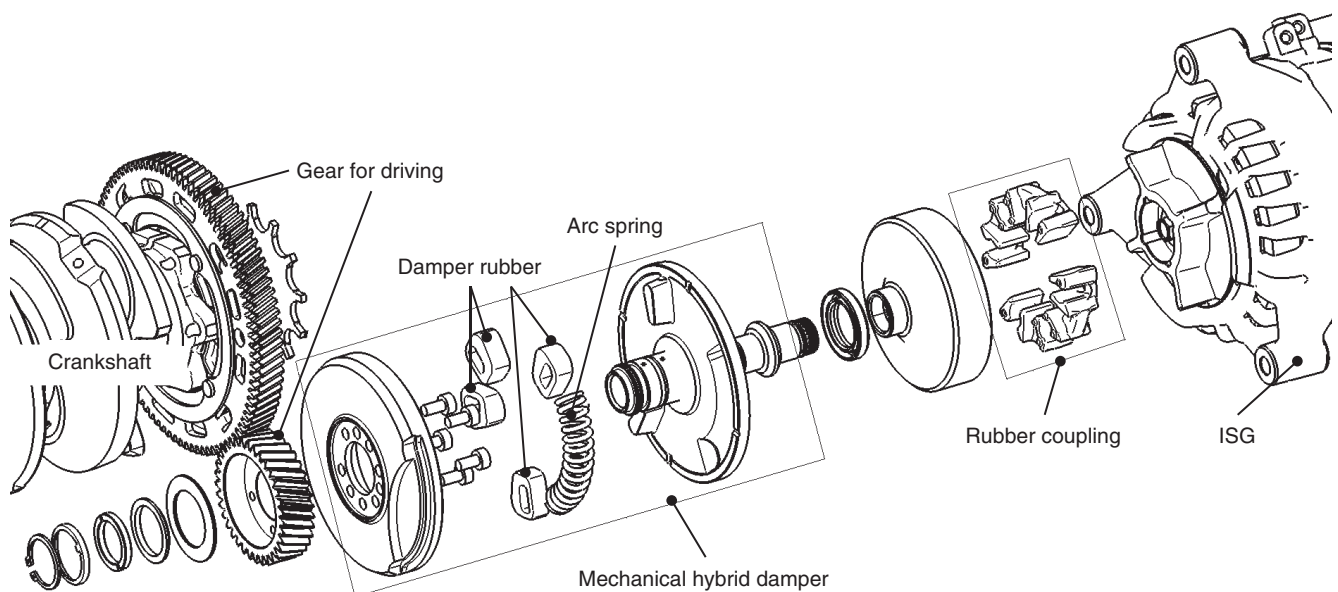


Fig. 14 Structure of ISG damper and coupling

when starting the engine was lower than the rotating speed of the starter motor and the reduction gear of the previous model. The wide and large-module helical gears were applied for starting of engine (=ISG drive gear) because the gears had to be strong enough to withstand the driving torque variation of the ISG. Accordingly, the tooth stiffness became higher than the starter gear of the previous model. Furthermore, the small reduction ratio permitted reduction only by a single stage rather than the two-stage reduction in the previous model. This also increased stiffness of the entire gear train. Consequently the sound pressure variation resulting from the deformation of the starter power train was reduced, contributing to the sound quality enhancement. The sound pressures at the time of starting in this model and the previous model are compared in Fig. 15. The horizontal axis refers to the time and the sound pressure in the vertical axis. The overall sound pressure and its fluctuation are lower in this model than in the previous model.

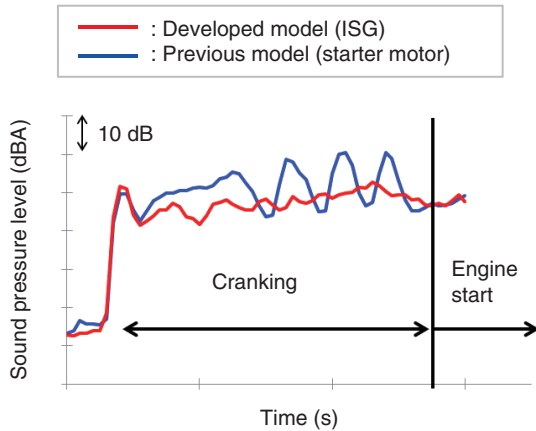


Fig. 15 Comparison of engine starting noise

6. Engine Performance

To achieve both the required engine power and the fuel consumption, the combustion efficiency was enhanced. And this has brought about the higher engine performance than the previous model.

In the cylinder head, the camshaft was located closer to the intake side in the same way as the previous model to help ensure lean angle and to maintain rider's leg space. In the previous model, an intake and an exhaust valve were laid parallel and activated by the direct valve lifting system using the SOHC design. In the new Gold Wing, the newly-applied Unicam valve train allowed the four-valve configuration while maintaining the same SOHC design having the camshaft close to the intake side just like the previous model. The volumetric efficiency was thus enhanced by the modification from the two-valve to the four-valve design while using the high-tumble intake

port design to strengthen turbulence in the cylinder. The measured volumetric efficiency is shown in Fig. 16. It is the same or higher than the previous model across the entire engine speed range. With the application of four-valve design, the combustion chamber was changed from the bathtub type of the previous model to the pent roof type, and the compression ratio was raised from 9.8 to 10.5.

The engine power output and the torque (converted on the crank shaft) are shown in Fig. 17. The indicated specific fuel consumption (ISFC) in the ordinary operation zone is shown in Fig. 18. The power output is achieved the same or higher level than the previous model across the entire engine speed zone. The ISFC is improved in the low-load zone.

The friction loss is compared to the previous model in Fig. 19. In the DCT type, despite the increase of

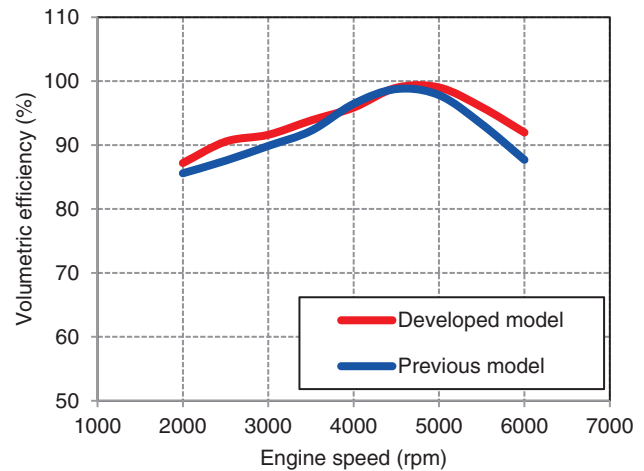


Fig. 16 Volumetric efficiency

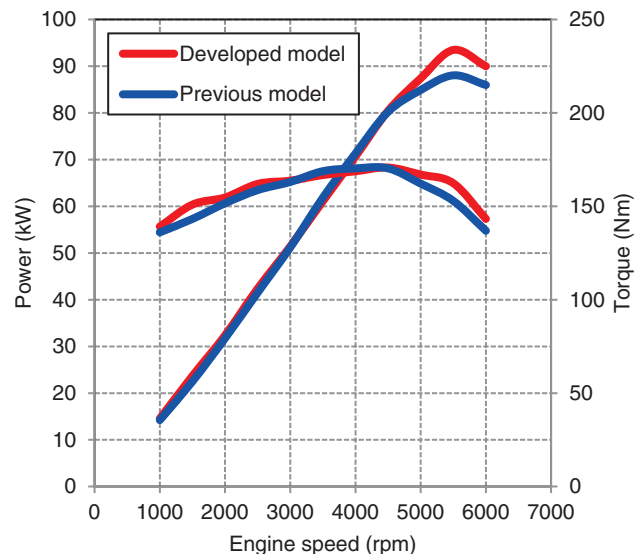


Fig. 17 Engine performance

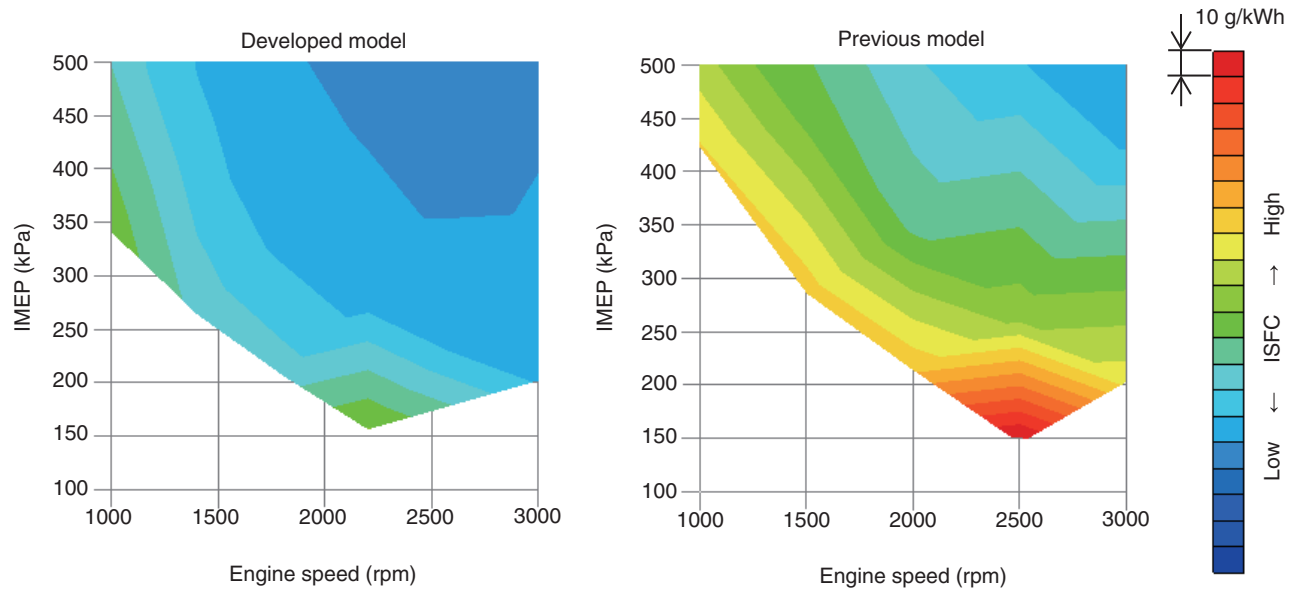


Fig. 18 ISFC map

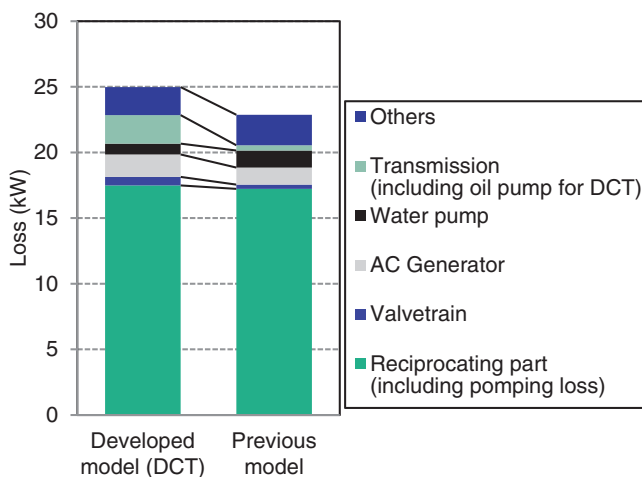


Fig. 19 Breakdown of loss at 6000 rpm

friction loss from the previous model due to the addition of exclusive oil pump for DCT, the fuel consumption in the WMTC mode (class 3-2) is improved from 14.7 km/L to 18.2 km/L (in the case of the model sold in Japanese market) due to the ISFC improvement, the weight reduction, the increase of gear step in transmission, etc.

7. Intake System

7.1. TBW

To provide the rider with high riding quality by precise control of the throttle, the TBW was applied in the throttle body. The controls using the TBW are as per the following.

- Simplified cruise control system
- Honda selectable torque control system
- Ride modes selection system
- Control of rapid engine torque increase when the throttle is opened quickly
- Walking speed mode - forward and reverse (DCT type only)
- Reduction of shift shock [moderating of inertia torque in the crankshaft system occurring at the time of gear shift (DCT type only)]
- Control of idle speed
- Maximum speed limiter
- Control of throttle opening when the idle stop is canceled

As such, the TBW allows addition of various functions in addition to the simplification of the cruise control system applied from the previous model.

7.2. Single-throttle Body

To offset an increase of weight and space with the addition of the motor to operate the TBW, the single throttle design was applied instead of the twin throttle type of the previous model.

The throttle body and the intake manifold of the previous model and the developed model are shown in Figs. 20 and 21.

In automobiles, the single throttle design is common in multi-cylinder engines. When the previous model was developed, the idea of single throttle was abandoned because the required throttle response was unattainable. In this model, the volume in downstream of the throttle valve (the volume of intake manifolds and intake ports) was reduced by 8% from the previous model. Coupled with the change of valve train from the two-valve to the four-valve

design and the improvement of combustion by the optimum valve timing, etc., the throttle response was achieved the same or better level than the previous model.

The changes of geometric opening area of throttle valve relative to the throttle valve angle are shown in Fig. 22. The changes of volumetric efficiency relative to the throttle valve angle calculated by CAE are shown in Fig. 23. For the calculation, the commercially-available one-dimensional analysis software was used (Simulation conditions: engine speed 1000 rpm, throttle twist speed 80 deg/sec).

Compared to the previous model, the rate of change of volumetric efficiency is higher in this model despite the low rate of change of geometric opening area.

The changes of driving force relative to the throttle opening angle measured from the actual motorcycle are shown in Fig. 24 (Measurement conditions: engine speed 1000 rpm, throttle opening speed 6.7 deg/sec). It has been confirmed from these findings that the throttle response is better in this model than the previous model.

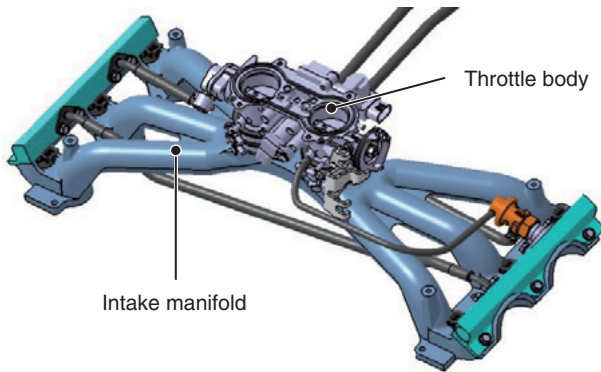


Fig. 20 Throttle body and intake manifold of previous model

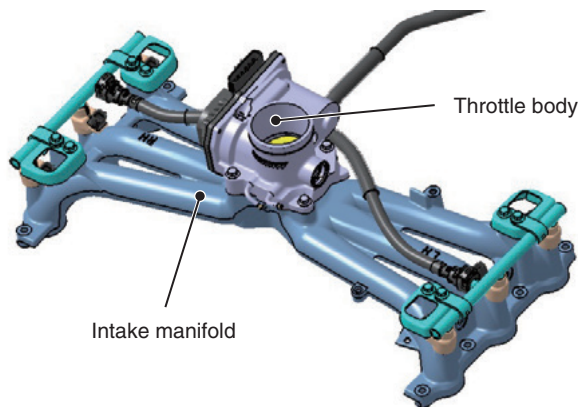


Fig. 21 Throttle body and intake manifold of developed model

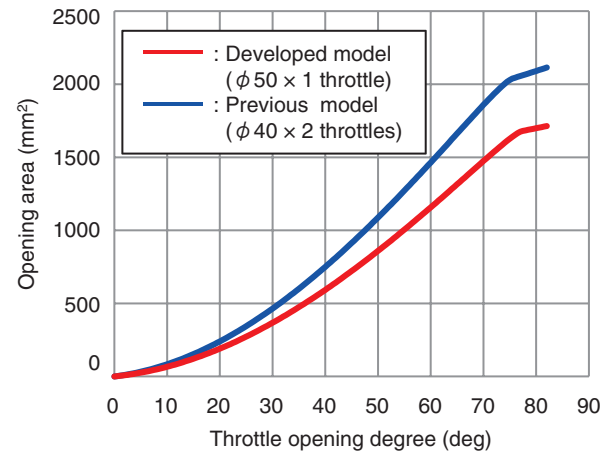


Fig. 22 Throttle opening characteristic

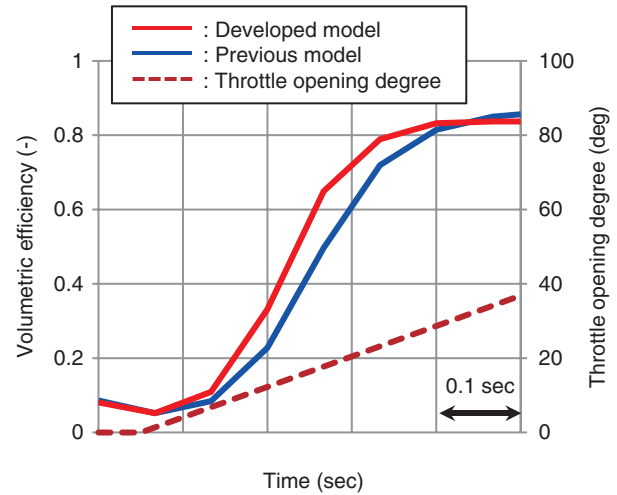


Fig. 23 Transition of volumetric efficiency in degree of throttle opening

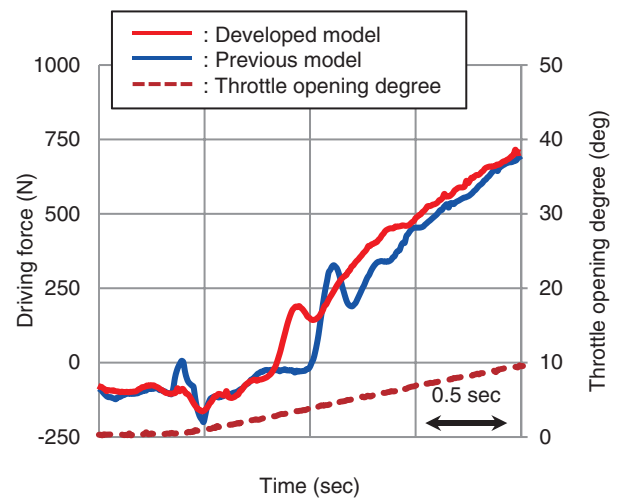


Fig. 24 Driving force (actual measurement)

7.3. Air Duct Layout

The electrical components such as the airbag module, ABS modulator, ECU were laid around the air box just like in the previous model. The double wishbone front suspension was applied in this model for realization of the concept of complete motorcycle. It is therefore necessary to contain the air box in the limited space.

Although the same 6.6-liter air box volume as the previous model was eventually maintained, the air filter surface area had to be reduced by 3%. Accordingly, the air duct layout was reviewed for effective use of the air filter surface area to achieve the maximum suction efficiency. The design specifications such as one of the two air ducts was facing to the front and the other facing to the rear were determined while analyzing flow of intake air.

The air cleaner box having two air ducts facing to the front just like the previous model is shown in Fig. 25 (Type A). The one with an air duct facing to the front and the other facing to the rear is shown in Fig. 26 (Type B).

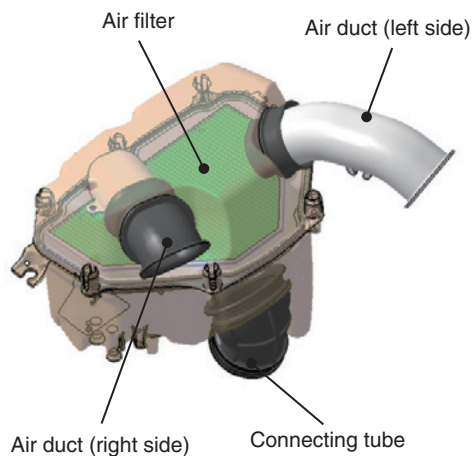


Fig. 25 Air cleaner box Type A

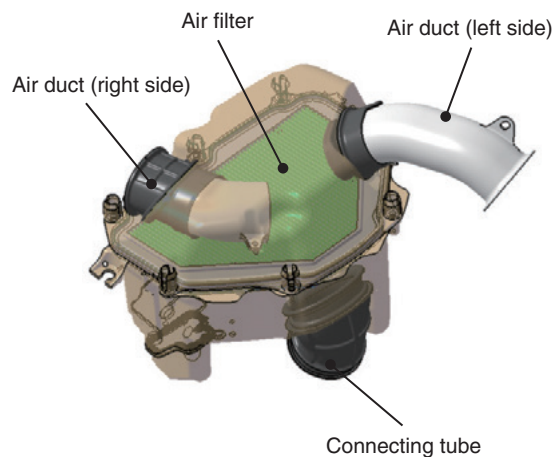


Fig. 26 Air cleaner box Type B

The average pressures on the cross section of the inlet and outlet of the air duct and the connecting tube and on the projected surface of upstream and downstream of the air filter calculated by CFD are shown in Fig. 27. Note that the commercially-available fluid dynamics analysis software was used for the calculation (conditions: steady flow when engine speed corresponds to 5500 rpm). Compared to Type A, the average pressure in various parts is higher in the Type B air box.

The internal pressure of air cleaner box (downstream of the air filter) measured from the actual motorcycle is shown in Fig. 28 (measurement conditions: engine speed 1000-6000 rpm, throttle fully open).

Also in the measurement, a higher air pressure is seen

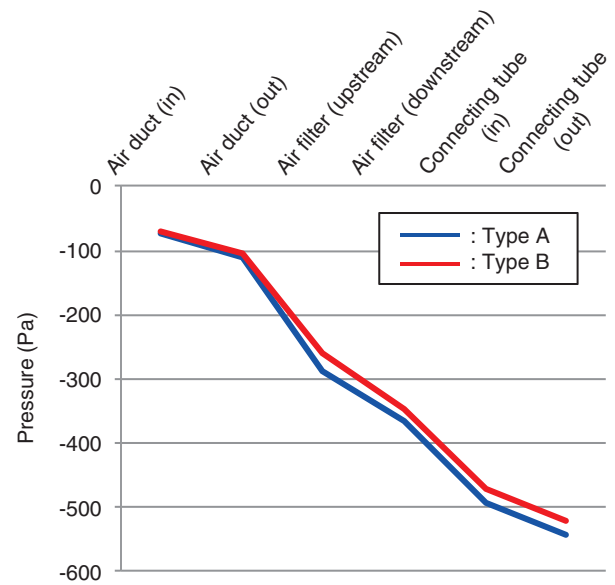


Fig. 27 Average pressure of each part

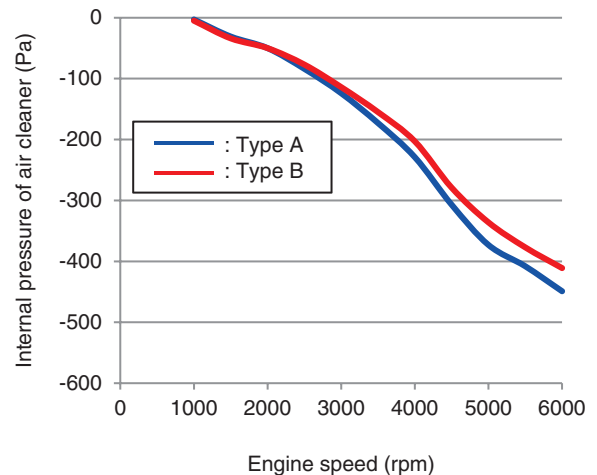


Fig. 28 Internal pressure of air cleaner box

in Type B than in type A, which is an indication of small pressure loss. From the said findings, the Type B air duct layout was applied in this model.

7.4. Reduction of Weight in Intake System

In the intake system, the weight was reduced by 1.1 kg from the previous model by the redesigning of the cruise control system, the throttle body and the intake manifold to deal with the application of single throttle system.

8. Exhaust System

In the exhaust system, a target was set to achieve exhaust sounds appropriate for the new Gold Wing as well as reducing the weight. As this new model requires high-quality image, the exhaust sounds have to be deep. A target was also set to create pulse feeling even from the F6 engine. The means to enhance deepness and pulse feeling is explained in the followings.

The internal construction of the muffler is compared to the previous model in Fig. 29. The inside of muffler is of such a construction that has a resonating chamber to attenuate sounds in addition to the same two-chamber construction as the previous model. Compared to the single tail pipe design of the previous model, two tail pipes are provided to obtain crisp sounds. The first tail pipe is open to the atmosphere from the first expansion chamber without passing the second chamber. This construction is aimed at

achievement of crisp sounds linked with the combustion by directing the high pressure exhaust gas directly to the tail pipe.

In the studies of construction, the attenuating characteristics of the muffler were set as an indicator of assessment. The acoustic characteristics were calculated from the sound pressures measured at the entrance and exit of the muffler using the test rig as shown in Fig. 30. The attenuating characteristics are compared to the previous model in Fig. 31. The horizontal axis refers to the frequency, the vertical axis represents attenuation. The red lines refer to this model and the blue lines the previous model.

To enhance deepness of exhaust sounds, it is necessary to emphasize sound pressures in low frequencies under 200 Hz. In the previous model, the construction was of a Helmholtz resonator composed of the connecting pipe that connects the first expansion chamber to the resonance chamber. The resonating frequency (the frequency of high attenuation) was near 150 Hz. Meanwhile in this model, the resonating frequency was set near 350 Hz by applying punched holes having 3 mm diameter and 5 mm pitch to the tube that connects the first expansion chamber to the second expansion chamber in addition to the tube that connects the first expansion chamber to the resonance chamber, thus controlling the attenuation under 200 Hz (green-framed part in Fig. 31). Consequently, the sound pressures in the low frequencies increased from the previous model, contributing

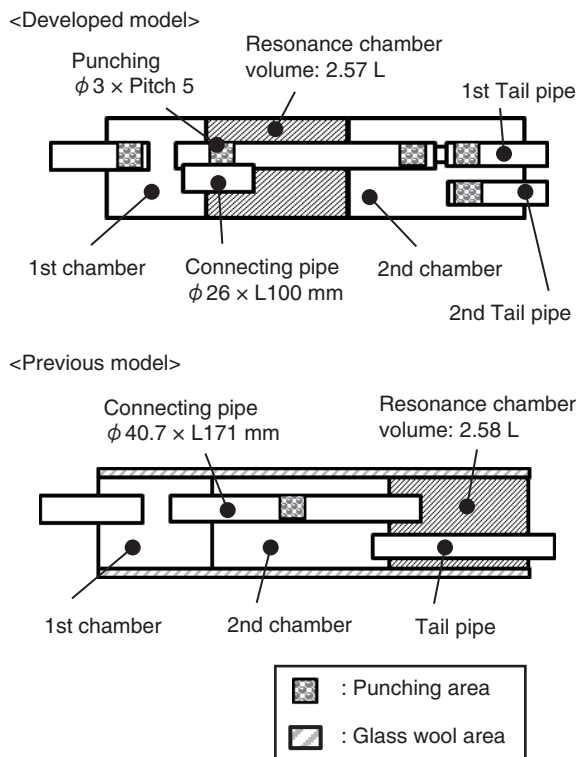


Fig. 29 Comparison of muffler structures

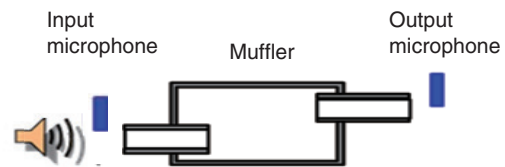


Fig. 30 Schematic diagram of test rig

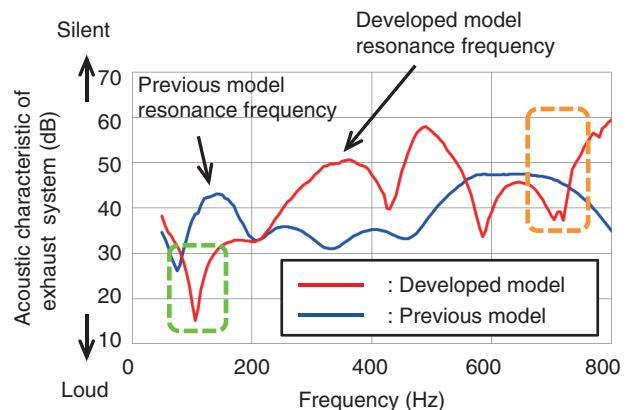


Fig. 31 Acoustic characteristic of exhaust system

to the enhancement of deepness of exhaust sound. Figure 32 shows the graph comparing, between this model and the previous model, the sound pressures in the low frequency band after adopting low pass filter under 200 Hz to the actual exhaust noise measurement results. The horizontal axis shows the engine speed and the vertical axis shows the sound pressure level of 200 Hz or less. It was confirmed that the sound pressure in the low frequency band of this model exceeds that of the previous model, proving the deepness of sound is enhanced at all the engine speed range.

Also, the increased attenuation of high frequency sound (orange-framed part of Fig. 31) permitted elimination of the glass wool in the outer canister applied in the previous model. With these measures taken, 2.1 kg has been reduced from the right and left mufflers.

It is commonly known that people feel pulse feeling when the frequency of sound pressure fluctuation is less than 30 Hz. In a six cylinder engine having evenly split combustion intervals, the frequency of exhaust sound fluctuation is three times per a full turn of the crankshaft (third order). As the frequency becomes higher than 30 Hz when the engine speed is above 600 rpm (10 Hz), the exhaust sounds are heard as continuous and constant. Accordingly in the Gold Wing in which the idle speed is 730 rpm, pulse feeling is usually unattainable.

To deal with the said issue, such an exhaust pipe construction was applied that creates difference in the sound volume by reducing the cross-sectional area of the inner pipe by 20% when compared to other pipes from the entrance of exhaust pipe up to the collector only in No.2 and No.3 cylinders (Fig. 33). It was considered that the amplitude of exhaust sound pressure variation becomes small in the cylinders having a small exhaust pipe diameter, and the sound volume appears as changing once in two full turns of the crankshaft (0.5th order).

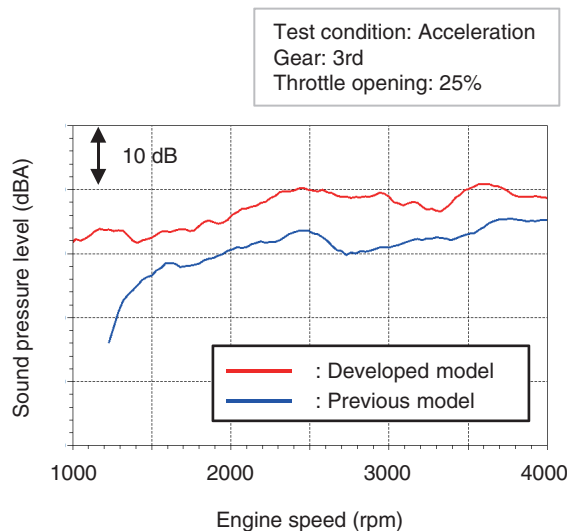


Fig. 32 Sound pressure level (under 200 Hz)

The images of exhaust sound pressure fluctuations at a constant engine speed are shown in Fig. 34. The upper diagram refers to the conventional, equal diameter exhaust pipe. The lower one is of the exhaust pipe having the cross sectional area reduced only in two cylinders.

The exhaust sound pressure level measured from the actual motorcycle is shown in Fig. 35. It can be understood from the data of developed model that the sound pressure fluctuates at the frequency of 0.5th order (16 Hz=0.06 sec interval at 2000 rpm). Meanwhile from the sound pressure data of the previous model, the sound pressure fluctuates at the third order frequency (100 Hz=0.01 sec interval at 2000 rpm). From these data, an enhancement of pulse feeling as

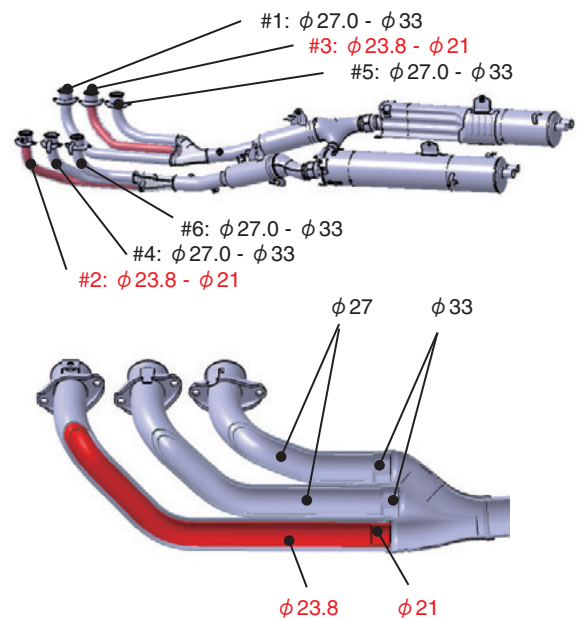


Fig. 33 Configuration of exhaust system

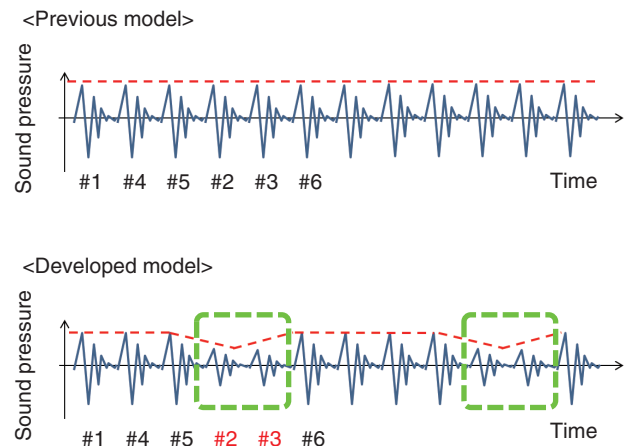


Fig. 34 Comparison of sound pressure waves (image)

intended was confirmed. As the frequency of 0.5th order sound pressure fluctuation stays below 30 Hz up to 3600 rpm, the pulse feeling is attainable in the developed model also at regular engine speeds.

In Fig. 36, the evaluation results of deepness and pulse feeling are shown using the index that Tanaka et al. have defined in their research⁽⁶⁾. Compared to previous model,

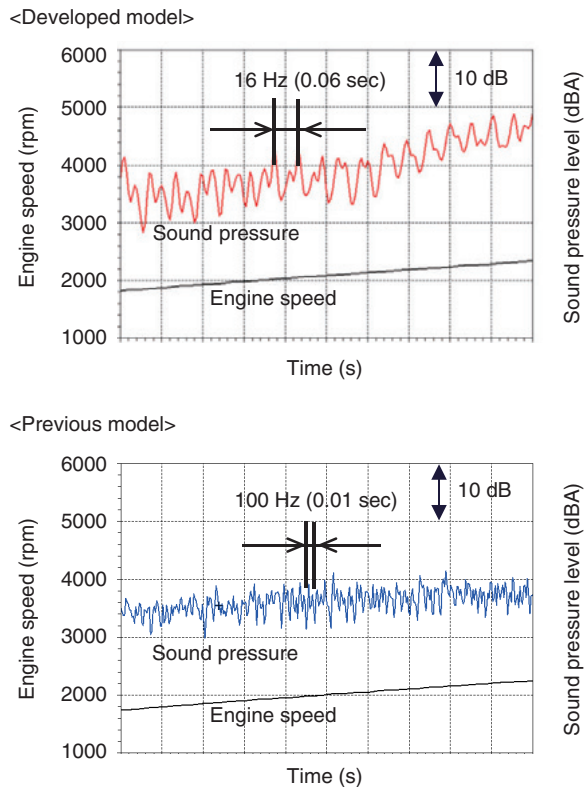


Fig. 35 Comparison of exhaust sound pressure

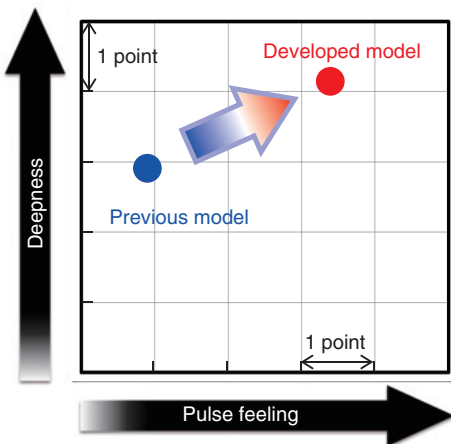


Fig. 36 Quantitative evaluation of exhaust sound

the deepness is enhanced by 1.2 points and the pulse feeling is enhanced by 2.5 points. These results illustrate that the exhaust sound desirable for the new Gold Wing was successfully created.

In addition to the earlier-mentioned muffler weight reduction, the muffler cover was downsized to balance the styling design and the function, allowing reduction of the weight of entire exhaust system by 3.3 kg. With this weight saving added to the earlier-mentioned engine weight, the weight reduction as a power unit is 7.1 kg in the DCT type and 9.5 kg in the MT type.

9. Conclusion

Through the development of the horizontally-opposed six-cylinder power unit for the new Gold Wing upon the concept “smooth yet aggressive at times, light weight and compact flat-six power unit,” the following goals have been achieved.

- (1) In the entire power unit, the weight is reduced by 7.1 kg in DCT type, 9.5 kg in MT type.
- (2) The length from the front of engine to the rear end of the left cylinder head is reduced by 29 mm, reducing the overall length, width and height of the engine for compactness.
- (3) The maximum power output is increased by 6 kW. The output is same or more in the entire engine speed range. Fuel consumption is improved by 24% (in the case of the model sold in Japanese market) in the WMTC mode (class 3-2).
- (4) The enjoyably deep and pulsated exhaust sound was also achieved while offering the mild and high quality sounds of mechanical operations.

References

- (1) Hirayama, T., Yokogawa, Y., Fujimoto, Y.: Development of 7-speed Dual Clutch Transmission for 2018 Gold Wing, Honda R&D Technical Review, Vol. 30, No. 2, p. 31-37
- (2) Hoshi, N., Shimura, Y., Okuma, T., Nomura, A., Gogo, K.: Introduction of CRF450R Equipped with 4-Stroke Engine, Honda R&D Technical Review, Vol. 14, No. 1, p. 21-30
- (3) Inui, H., Sato, E., Kaji, T., Shigezumi, S.: Development of Aluminum Powder Metal Composites Material for Cylinder Sleeves Suitable for Extrusion Process, Transactions of Society of Automotive Engineers of Japan, Vol. 46, No. 2, p. 545-550, (2015) (in Japanese)
- (4) 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
- (5) Ishihara, Y., Hamakawa, T., Satoh, K., Yamasaki, T., Shi, C., Hashimoto, H., Haneda, H., Takahashi, S.: Prediction of Fracture of Cylinder Block Main Bearing by Fretting Fatigue and Development of Verification

Method, Honda R&D Technical Review, Vol. 28, No. 1,
p. 191-196

- (6) Tanaka, K., Sugita, H., Saito, H., Sekita, M.: “Design
Method of Motorcycle Exhaust Sound Fitting to Vehicle
Concept Regardless of Engine Configurations” SAE
Technical Paper 2014-32-0121, 2014, doi:10.4271/2014-
32-0121

■ Author ■



Hiroshi YOKOTA



Satoru NOJIMA



Yutaka SONODA



Yusuke OSAKI



Masahiro KONTANI



Satoru YAMASAKI

Development of 7-speed Dual Clutch Transmission for 2018 Gold Wing

Taku HIRAYAMA*

Yukio YOKOGAWA**

Yasushi FUJIMOTO**

ABSTRACT

The Gold Wing, Honda's grand tourer motorcycle, has been highly praised for its quality feel and quietness. The 2018 Gold Wing GL1800 is offered with either the manual transmission or a newly added 7-speed dual clutch transmission, which allows for comfortable, yet sporty and enjoyable shifting. The transmission ratios are optimized for enhanced quality feel and quietness, and shifting become even quieter thanks to shift controls coordinating the clutch and throttle-by-wire and added dampers to absorb shifting noise. In addition, the reverse mechanism, a Gold Wing exclusive feature, was changed from the existing motor-driven system to an engine-driven system utilizing the architecture of the dual clutch transmission, producing a lightweight and compact engine package. Further, a new forward/reverse walking speed mode was added, capable of both forward and backward creep at very low speed with just the push of a handlebar switch. These changes have made the Gold Wing more convenient by making it easier to turn around and park.

1. Introduction

The sixth generation Gold Wing GL1800 has been newly redesigned to achieve its concept: The Honda Premium Tourer-Impress couples with more brilliance and fullness. The Gold Wing inherits the same water-cooled four-stroke horizontally-opposed six-cylinder, longitudinally-mounted engine (hereinafter the "F6") architecture from the previous model.

Transmission offerings include a manual transmission (MT) and a newly-developed dual clutch transmission (DCT). As with the existing motorcycle specifications, the transmission is set in the same case as the power unit.

The motorcycle DCT relieves riders from clutch and shifting operations to concentrate on the acceleration, cornering, braking, and other actions that make for a fun ride. It also gives the motorcycle a sporty feel with its direct and linear power transfer in response to throttle operation.

The VFR1200F⁽¹⁾ was the first mass production model equipped with a DCT, followed by subsequent updates in the NC700/750 series⁽²⁾ and CRF1000L Africa Twin⁽³⁾.

The DCT in the sixth generation Gold Wing stays true to its concept while adding new features and enhancing quality feel.

Now, for the first time in a motorcycle DCT, we applied a 7-speed gear mission for the Gold Wing DCT specifications to fit a wider variety of applications and enhance quietness. The Gold Wing's unique reverse feature now utilizes the DCT's original structure for an engine-driven reverse gear, replacing the previous model's motor-driven system for a weight reduction. (The MT model retains the motor-driven reverse system from the previous model.) Also, the Gold Wing offers a completely new forward/reverse walking speed mode, which allows riders the added convenience of creeping forward and backward at a very low speed with just the push of a handlebar switch.

The Gold Wing that requires high quality feel needs to be further reduced in transmission shifting noise and operating noises than the existing DCT. Therefore, dampers was installed on those sections responsible for tapping noises when shifting, enhancing both quality feel and quietness.

Additionally, in aims for a sportier ride in this Gold Wing, clutch control and throttle-by-wire (TBW) control was cooperated for quicker shifting.

This paper will contain detailed explanations of the unique structure and unique technology introduced for the DCT equipped on this model.

* Honda Motor Co., Ltd.

* Motorcycle R&D Center

2. Unique Technology for Gold Wing DCT

The following unique technologies were developed for applying the DCT for the Gold Wing:

- (1) Engine-driven reverse mechanism by cooperated clutch and TBW control. Also, a reverse chain was used to achieve lightweight and compact both reverse mechanism and transmission configuration.
- (2) Walking speed mode for enhancing maneuverability at walking speed in situation like parking or K-turn
- (3) Enable sporty shifting that is smooth, yet occasionally shows an aggressive side by cooperated clutch and TBW control.
- (4) First time 7-speed motorcycle DCT and reducing shifting noise for enhancing quality feel with quiet and smooth shifting while riding.

3. DCT System Configuration

An explanation of the unique basic configuration of the DCT equipped on the Gold Wing in this paper follows below.

3.1. DCT Placement

As shown in Fig. 1, the crankshaft and drive shaft are mounted longitudinally to the body as in the previous model. To make it more compact, DCT system parts were placed inside the engine.

In the interest of reducing weight, the reverse mechanism uses an engine-driven system fully mounted within the engine. By eliminating need for an electric motor or control driver, this reduces weight by approximately 1.3 kg from the motor-driven system.

3.2. Reverse Mechanism and Transmission Configuration

Just as in conventional motorcycle DCTs, a twin-shaft architecture comprising a two-element main shaft and a countershaft has been adopted for the Gold Wing transmission. The transmission has eight gears in total: seven forward gears and reverse. As seen in Fig. 1, no gears are used for the reverse mechanism, instead opting for a wear-resistant silent chain for reduced weight.

A chain-driven architecture was adopted here, eliminating the need for an idler shaft and idler gear for a more compact, lightweight reverse mechanism. When driven by a gear, the countershaft rotates counter to the main shaft during forward motion (Fig. 2-a). Driving in reverse requires an added idler shaft and idler gear between the main shaft and countershaft to rotate the countershaft clockwise (Fig. 2-b). Meanwhile, if chain-driven, both the main shaft and countershaft will rotate clockwise (Fig. 2-c).

The speed reduction of the chain alone will not provide the required reduction gear ratio, however. Thus, speed is reduced a total of three times between the main shaft and countershaft to reach the required reduction

gear ratio when in reverse. A special drive path is used for this purpose.

The power transmission paths in forward and reverse are shown in Fig. 3. The reverse path starts with the odd-number #1 clutch, proceeding through the odd-number main shaft to the first speed reduction (first gear), then

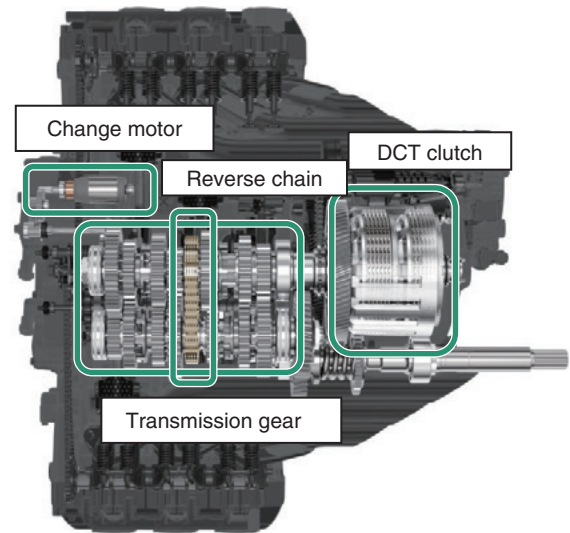


Fig. 1 Cutaway drawing of developed transmission

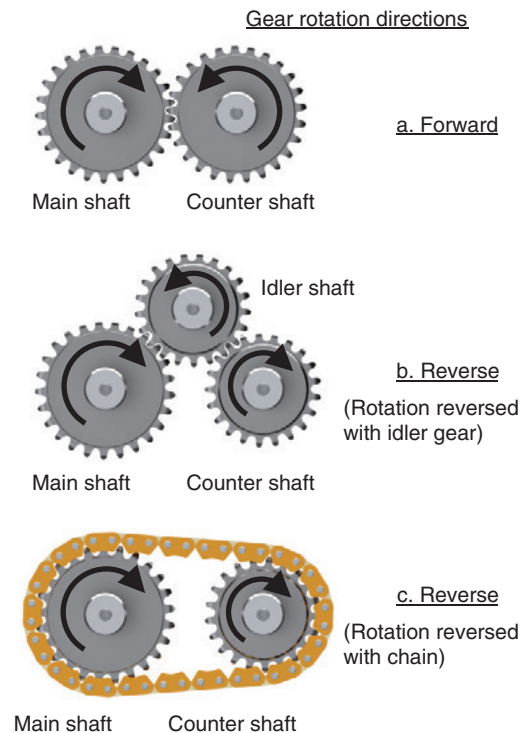


Fig. 2 Direction of shaft rotation (image)

the second speed reduction (reversing rotation with the reverse chain), then onto the even-number main shaft, and finally to the third speed reduction (second gear). For forward gears, the path goes from the even-number #2 clutch to even-number main shaft and then the first speed reduction (second gear).

Thus, the power transmission path for the two clutches proceeds through the odd-numbered gears for reverse and the even-number gears for forward motion. This structure allows for switching between reverse and forward motion simply by engaging the clutch without any need to change gears.

The transmission architecture is unprecedented, using a chain to link the two-element main shaft and the countershaft with a special layout between the gears and reverse chain of 7-3-5-1-R-6-4-2.

Additionally, as adopting a reverse chain makes assembly impossible with the main shaft and countershaft separated, the main shaft and countershaft need to be pre-mated before mounting to the engine during assembly. Therefore, the cassette transmission architecture shown in Fig. 4 was adopted to facilitate assembly.

3.3. Shift Mechanism

With the addition of seventh gear and reverse positions, the number of shift drum positions has changed from 12 in the existing 6-speed DCT to 15, as seen in Table 1. This change has shifted the shift drum angle per position from 30° in the existing DCT to 24° , not leaving enough room on the periphery of the shift drum for the full length of the shift fork lead groove. As a workaround, the shift drum outer diameter was enlarged from 42 mm to 58 mm.

Bringing the lead groove length of the outer periphery

of the shift drum per 1 position was made equivalent to the existing 6-speed DCT specification.

To keep the engine compact, the gear shift actuation motor was arranged to face the engine interior (Fig. 1).



Fig. 4 Cassette transmission architecture

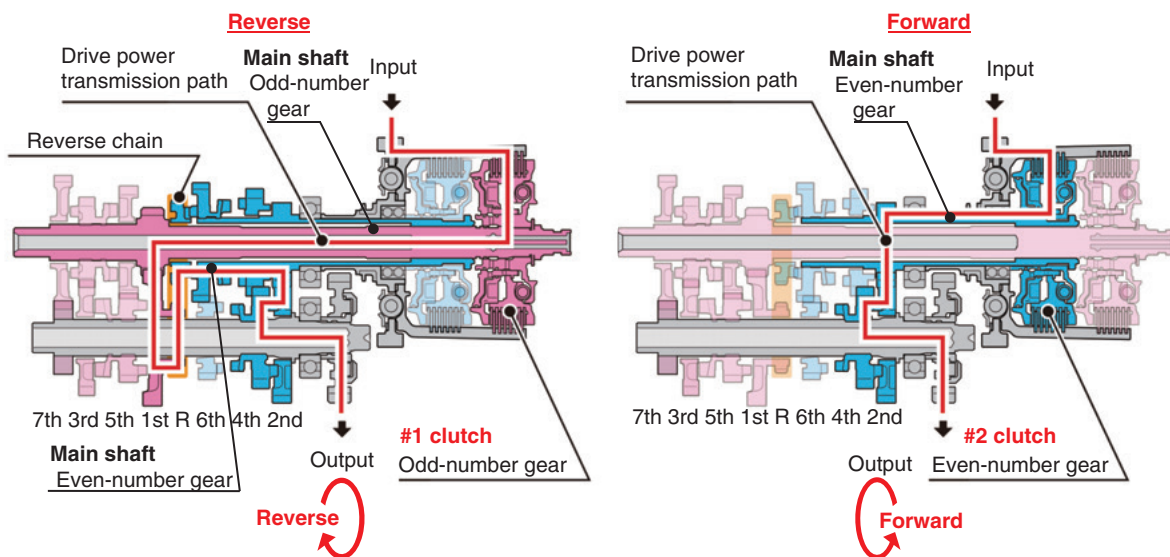


Fig. 3 Power transmission path

Table 1 Shift drum positions

Shift drum positions of developed model				Shift drum positions of other DCT models			
Shift drum position #	Gear shift conditions	Engaged gear		Shift drum position #	Gear shift conditions	Engaged gear	
		Odd-numbered gears	Even-numbered gears			Odd-numbered gears	Even-numbered gears
1	Rvs	N	R, 2	1	Neutral	N	N
2	Neutral	N	R	2	Start, 1st	1	N
3	Start, 1 st	1	N	3	1st ⇄ 2nd shift	1	2
4	1st ⇄ 2nd shift	1	2	4	2nd	N	2
5	2 nd	N	2	5	2nd ⇄ 3rd shift	3	2
6	2nd ⇄ 3rd shift	3	2	6	3rd	3	N
7	3 rd	3	N	7	3rd ⇄ 4th shift	3	4
8	3rd ⇄ 4th shift	3	4	8	4th	N	4
9	4 th	N	4	9	4th ⇄ 5th shift	5	4
10	4th ⇄ 5th shift	5	4	10	5th	5	N
11	5 th	5	N	11	5th ⇄ 6th shift	5	6
12	5th ⇄ 6th shift	5	6	12	6th	N	6
13	6 th	N	6				
14	6th ⇄ 7th shift	7	6				
15	7 th	7	N				

4. Unique technologies for Gold Wing DCT

This section will explain the developed unique technologies.

4.1. Walking Speed Mode

In DCT-equipped models, a walking speed mode has been added along with the switch in the reverse feature from the electric motor-driven system in the previous model to an engine-driven system.

Reversing on a motorcycle requires the rider to creep at very low speed while on the motorcycle. In manual motorcycles, installing an engine-driven reverse mechanism would require fine speed adjustments on the part of the rider to manipulate the throttle and clutch similar to driving forward at low speed. This made it challenging to use the engine as the power source for reverse mechanisms⁽⁴⁾.

The Gold Wing DCT specifications enabled an engine-driven reverse mechanism with TBW-linked control, where the clutch and throttle are operated simultaneously to allow for finer speed adjustment.

Also, as previously mentioned, riders can switch between reverse and forward motion simply by engaging the clutch without any need to change gears. Utilizing this feature, a switch-activated walking speed mode was developed, capable of forward and rearward creep at very low speed. Next, the below is an explanation of walking speed mode. With the transmission in neutral, once activating walking speed mode with the walking speed mode switch, riders can move forward by pressing the shift up switch or backward by pressing the shift down switch. A view of the switch cluster on the left handle grip is shown in Fig. 5. The motorcycle moves at a creep when either switch is pressed.

Walking speed mode enhances maneuverability at very low speed, for situations like parking and K-turns to change direction. For more stability when creeping with the switch activated, clutch torque transfer is finely adjusted by vehicle speed feedback control. However, if the motorcycle builds up speed going downhill, the clutch will eventually be released as vehicle speed feedback control gradually continues to reduce the clutch torque transfer. The motorcycle will ride free, unable to maintain the target speed. Thus, driving power can be used in both the advancing and opposite directions to reduce motorcycle speed by engaging either the forward clutch if moving in reverse or the reverse clutch if



Fig. 5 View of left handle grip switches

moving forward. With this in mind, controls were also implemented to engage the opposite direction clutch instead of the advancing direction clutch when vehicle speed exceeds the target speed to restrict vehicle speed from exceeding a certain level (Fig. 6).

Also, in walking speed mode, feedback control with TBW puts engine speed slightly higher than idle at approximately 1000 rpm even if engine load increases when the clutch is engaged.

Thus, walking speed mode reduces the need for the rider to make fine input with the throttle grip so that they can focus on maneuvering the motorcycle.

4.2. Sporty Shifting Control

In a DCT, the clutch connected to the previous gear is released at the same time as the clutch connected to the next gear is engaged for smooth shifting with no interruption in drive force. This Gold Wing model has four ride modes for various conditions. One of the four modes offered is SPORT. In this mode, new controls have been implemented, cooperating with TBW control to shorten shift times for a sportier ride while leaving shift shock during an up shift equivalent to that in other modes.

Clutch engagement times could also be shortened in conventional DCT controls by increasing the torque transfer from the clutch when engaged. However, shortened engagement times resulted in an increased engine speed reduction rate (crankshaft angular acceleration), increasing inertial torque in the crankshaft system. When transferred to the rear wheel, this inertial torque changes chassis longitudinal g-force and the rider feels shift shock. This made it challenging to shorten clutch engagement

times with conventional shift control as it would require restricting shock from clutch engagement to within rider tolerances (Fig. 7-a, b).

Thus, as shown in Fig. 7-c, TBW control was linked and the throttle closed when the clutch is engaged, resulting in less increase in engine torque from inertial torque generated by quicker shifting. Shift time has thus been reduced 200 ms from 300 ms, approximately 35% (at throttle opening 25 degree and engine speed 3000 rpm) with no increase in shift shock.

4.3. Enhanced Transmission Quietness Due to Additional Speed and Ratio Distribution Optimization

To reduce shift shock and enhance quietness, a seventh gear was added to the transmission with newly optimized ratios for each gear.

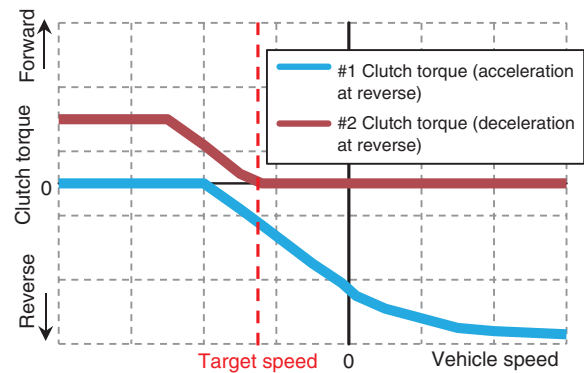


Fig. 6 Clutch torque at reverse

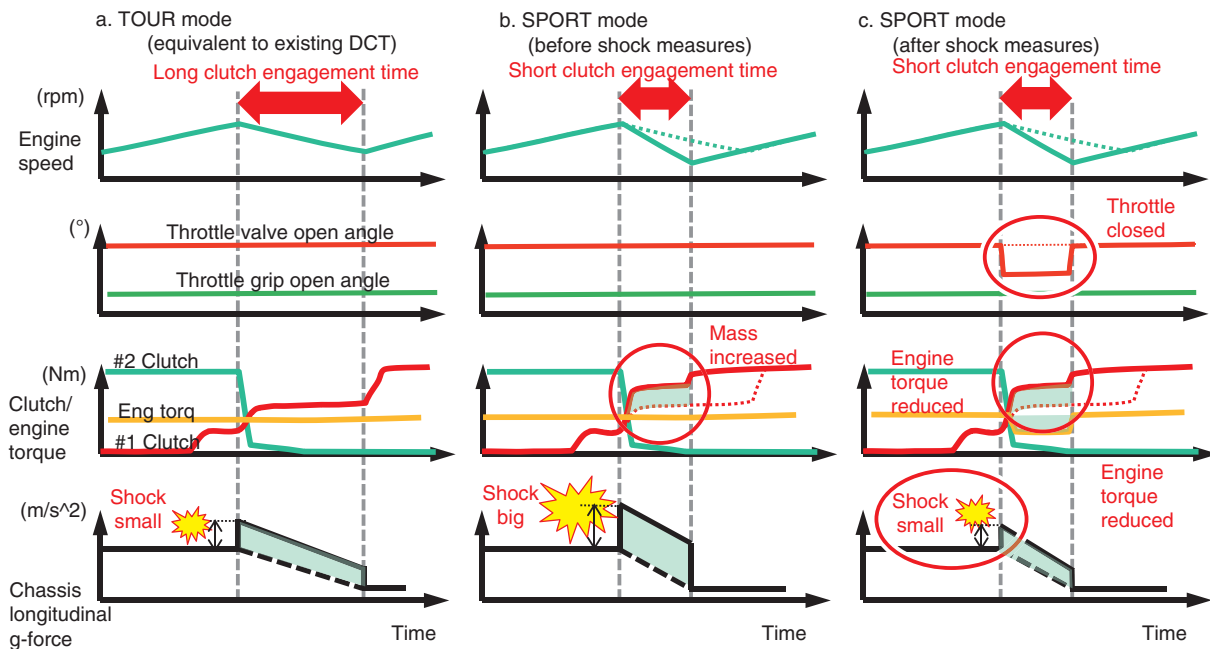


Fig. 7 Shift times and control

In the low speed range, where less driving noise makes operation noise during shifts readily audible, a gear was added for a closer ratio than the previous model. This is depicted in Fig. 8. These changes reduced shifting noise by reducing the engine speed differences between gears before and after shifting. In addition, the closer ratio reduced the change in driving force between gears before and after shifting, which mitigates shift shock and enhances quality feel.

Meanwhile, at the high speed ranges most frequently used in touring, an additional speed was added to keep engine speed low and enhance quietness when cruising.

As seen in Fig. 8, the previous model ran at approximately 2600 rpm in fifth gear when driving at 100 km/h; this has been reduced to approximately 2000 rpm in seventh gear for a reduction of 600 rpm. When cruising at 100 km/h, this gearing has thus reduced noise levels at the rider's ear by approximately 5 dB from the previous model.

4.4. Reducing Tapping Noise during Shifts

Clutch center dampers were equipped for enhanced quality feel. These dampers work to reduce the exciting force and vibration frequency when the dog clutches of each gear strike during shifts to reduce tapping noise. A structural drawing of Gold Wing clutch is given in Fig. 9.

In the process of making different sections of the Gold Wing quieter, tapping sounds during shifts which were unintrusive on the existing model became relatively more noticeable. Therefore, rubber dampers were installed for those sections causing the tapping during shifts.

During shifts, the end face of the shift fork shaft was dragging against the shift fork, resulting in a tapping noise when it hit the engine case. Dampers were attached to both ends of the shaft to muffle the tapping noise (Fig. 10).

Also, the master arm, which rotates the shift drum

during shifts, was making noise when it hit the stopper pin restricting master arm rotation. To reduce this noise, more dampers were attached to the master arm (Fig. 11).

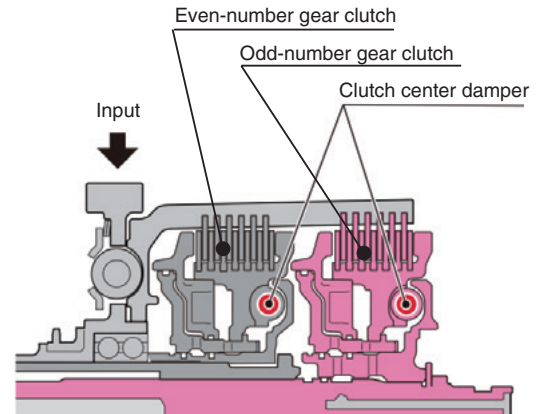


Fig. 9 Structure of dual clutch

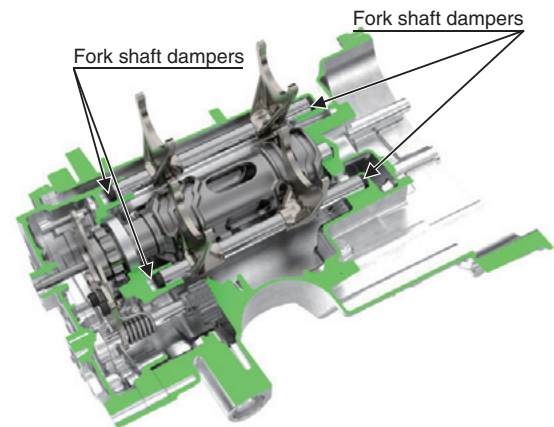


Fig. 10 Fork shaft dampers



Fig. 8 Speed diagram

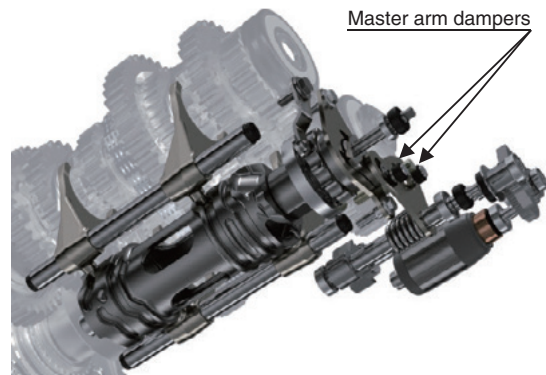


Fig. 11 Master arm dampers

5. Conclusion

This paper introduced the following developed unique technologies of DCT for the sixth generation Gold Wing.

- (1) Cooperated clutch and TBW control enabled use of an engine-driven reverse mechanism.

Also, a reverse chain was used to achieve a lightweight, compact F6 engine.

This reduces weight by approximately 1.3 kg from the motor-driven system.

- (2) Walking speed mode enabled riders to creep forward and backward at a very low speed with just the push of a handlebar switch.

This has enhanced vehicle operability at walking speed for situations like maneuvering the motorcycle in tight spaces or when parking.

- (3) Cooperated clutch and TBW control permits shift shock to be restricted with 35% quicker shift times for smooth, yet sporty shifting in SPORT mode.

- (4) A seventh gear was added, and the gear ratio was adjusted for enhanced quietness.

This gearing has thus reduced noise levels at the rider's ear by approximately 5 dB from the previous model.

Dampers were installed for sections causing the tapping during shifts, enhancing quality feel with quiet and smooth shifting while driving.

Adopting those technologies have helped to make this the motorcycle with the most enjoyable maneuverability throughout all the models appeared in the chronicle of the Gold Wing to drive, balancing sporty handling, quality feel, and quietness.

References

- (1) Watanabe, J., Arai, D., Abe, T., Tanaka, M., Ogasawara, A., Tsuchiya, M., Tsubota, R.: Development of Dual Clutch Transmission for Large Motorcycles, SAE Paper, 2010-32-0057 (2010)
- (2) Kobayashi, K., Negro M., Osuka T., Mitsubori T., Oshima T.: Development of NC700 Series, Honda R&D Technical Review, Vol. 24, No. 2, p. 23-31
- (3) Iizuka N., Yamakura Y., Imai Y., Iida A., Sugano T., Nakamura M., Koyama S., Nakamura K., Ono K.: Development of CRF1000L Africa Twin, Honda R&D Technical Review, Vol. 28, No. 2, p. 9-21
- (4) Ohta A., Yamagata T., Ito T., Nebu H., Uchibaba K., Fujita H.: Reverse System Design for Motorcycles, HONDA R&D Technical Review, Vol. 2, p. 49-58 (in Japanese)

■ Author ■



Taku HIRAYAMA



Yukio YOKOGAWA



Yasushi FUJIMOTO

Development of Ride Modes for Various Conditions for 2018 Gold Wing

Atsushi KUSUDA*

Yukio YOKOGAWA*

Kazuho NAKAI*

ABSTRACT

With its well-balanced riding comfort and sportiness, Honda's grand tourer model Gold Wing has gained good reputation for decades. In the 2018 Gold Wing GL1800, in addition to bringing the comfort and sportiness to a new height, a target was set to provide the rider with more performance benefits by offering selectable ride modes to adjust to various riding conditions such as the sport ride, the riding on suburban road, in rain, etc. To achieve the target, various factors such as the engine power response have to be altered. When the number of factors to alter increases, it is not easy for the rider to combine such factors and adjust to the riding conditions and rider needs. Accordingly developed in this model is world's industry first (according to Honda's research) new system for the motorcycle to allow the rider to select a ride mode for each condition by the integrated control of five factors (engine power response, characteristic of driving force, gear shift timing, suspension damping, front/rear brake force distribution). The newly-developed system provides the riding performance that allows the rider to control the motorcycle as the rider wants in various conditions.

1. Introduction

The Gold Wing is a grand tourer model of Honda motorcycles. The previous fifth generation Gold Wing has a good reputation by its well-balanced riding comfort and sportiness. In the sixth generation 2018 Gold Wing GL1800, a target was set at "the motorcycle most enjoyable to ride in the history of Gold Wing." Our interpretation of "the most enjoyable to ride" is to offer such characteristics that allow the rider to control the motorcycle as he wants in various types of riding such as the sport ride, the riding on suburban road, in rain, etc. As shown in Fig. 1, we aimed to expand the range of performance that riders can enjoy. To achieve the goal, it is necessary to alter variables such as the suspension setting, engine power response, etc. to adjust the characteristics of the motorcycle to the conditions and rider needs. In the case of Gold Wing, we considered it was necessary to alter a large number of variables to adjust to the wide variety of riding conditions and rider demands. When the variables to alter are increased, it is not easy to achieve a good combination of various levels of setting that best suits the riding conditions and rider needs. Accordingly developed in this model is the system that allows the rider to select a ride mode that best suits the conditions by the integrated control of five factors (engine power response,

characteristic of driving force, gear shift timing, suspension damping, front/rear brake force distribution) that are world industry first, according to Honda's research (hereafter referred to as "ride modes for various conditions"). The details of the system are discussed in this paper.

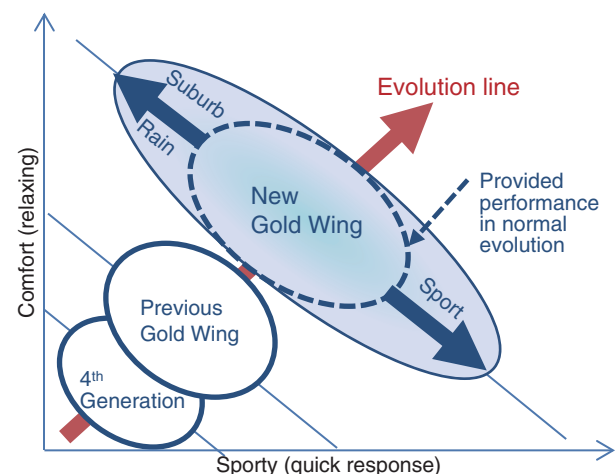


Fig. 1 Target zone

* Motorcycle R&D Center

2. Target Setting

In the sixth-generation Gold Wing, the setup in which the comfort and sportiness under various conditions are well-balanced is referred to as the TOUR mode. Using the TOUR mode as a basis, three other ride modes, SPORT, ECON, and RAIN are set to allow adjustment to specific riding conditions. With an aim at extension of performance zone that the rider can enjoy, one of these modes is selectable by the rider using the select switch when riding. The presumed conditions and the riding characteristics in each mode are shown in Table 1, the performance positioning image in Fig. 2. Using the TOUR mode as a basis, the specific aims of three modes are as per the following.

- (1) SPORT: Enhancing dynamic performance such as acceleration
- (2) ECON: Enhancing rider's comfort such as quietness, better fuel economy
- (3) RAIN: Enhancing secure feel by retention of rear tire grip on slippery surface

Table 1 Presumed conditions and riding characteristics

Mode	Condition	Characteristic
TOUR	Versatile	The most balanced in various situations (The Gold Wing)
SPORT	Winding / Solo riding	Exciting with quick response
ECON	Suburb / Congested (urban)	Comfort and relax with mild response
RAIN	Wet / Dirt	More secure in low μ road condition

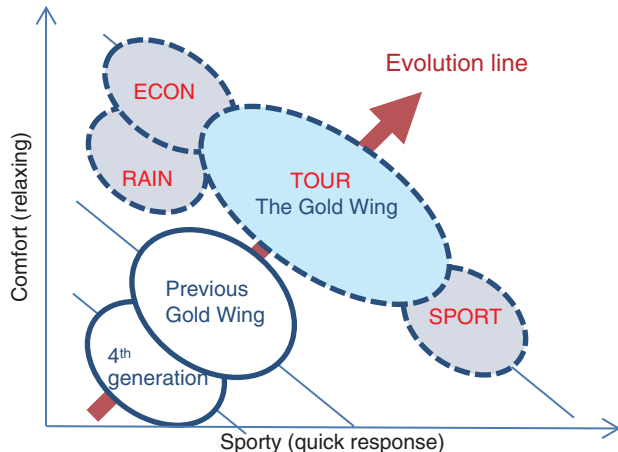


Fig. 2 Performance positioning image

3. Methods

Explained here are the methods to achieve the targets of various ride modes. While this system is applied both in the manual transmission model and the Dual Clutch Transmission (DCT) model, this paper discusses about the system incorporated in the DCT model.

3.1. Technical Measures

Choosing the variables from the typical riding style in each mode to attain the performance the rider wants, the technical measures was selected. The technical measures to adjust the five factors to the demands of each mode are shown in Fig. 3.

3.1.1. SPORT mode

When performing a sportier riding on a winding road, it is necessary to prepare for a corner entry by a quick, sure shifting of gear and/or stabilize the motorcycle by applying the brake in the deceleration zone before entering the corner. In such a condition, the gear shift timing, the front/rear brake force distribution, and the suspension damping are the variables affecting motorcycle stability.

When accelerating from the corner exit, appropriate throttle control and gear shift timing are required to help prevent tire slip for quick acceleration. In such a case, the engine power response to the throttle grip operation, the characteristic of driving force, and the gear shift timing are the factors that affect acceleration.

3.1.2. ECON mode

The ECON mode presumes riding suitable in congested roads and/or suburban roads. As starting and stopping are repeated on a congested road, a gentle response of motorcycle movement to opening/closing of throttle is required. In such a case, the engine power response when the throttle is opened or closed is the factor that affects motorcycle movement. When cruising on a suburban road at a relaxing pace under 100 km/h, quietness is important in addition to the gentle motorcycle movement. A reduction of noises when riding is therefore required. In such a case, the gear shift timing is the factor that helps reduction of engine noises.

3.1.3. RAIN mode

To obtain a higher secure feel on slippery surfaces, a gentle response of motorcycle movement when the throttle is opened or closed is required just like in the ECON mode. The stability of motorcycle is also required when the rear tire slips by an abrupt throttle operation. In such a case, the engine power response to the throttle grip operation, the suspension damping, and the characteristic of driving force at the time of slippage are the factor that affect changes of motorcycle movement and reduction of tire slippage.

In view of the above, five factors; the engine power response, the characteristic of driving force, the gear shift

timing, the suspension damping, and the front/rear brake force distribution, which are contributing variables to satisfy rider demands in each mode, have been chosen.

3.2. Control System

The overview of the system to simultaneously control the five technical measures is shown in Fig. 4. The system is configured to allow integrated control by the use of CAN for the communication among the control units that control various devices. The rider selects a mode from the right handlebar switch. The mode selection is allowed even when riding. The switching of mode is completed when the throttle grip is fully closed after operating the mode switch.

4. Setting of Control Items in Each Mode

Explained hereafter are the details of the technical measures to achieve the goals of each of SPORT, ECON, or RAIN mode.

4.1. Throttle by Wire (TBW)

As shown in Fig. 5, TBW is such a system that determines throttle opening from the throttle grip angle detected by the accelerator position sensor (APS) in the right hand switch housing, the engine speed, etc. to regulate engine power. The engine power characteristics corresponding to various throttle grip angle at the certain

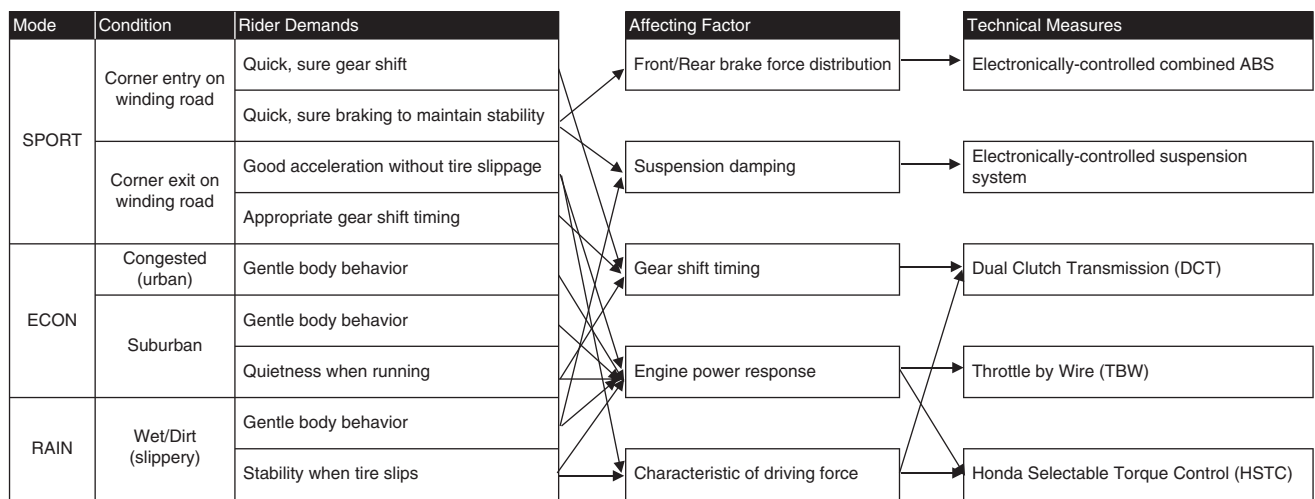


Fig. 3 Technical measures to adjust five factors to demands of each mode

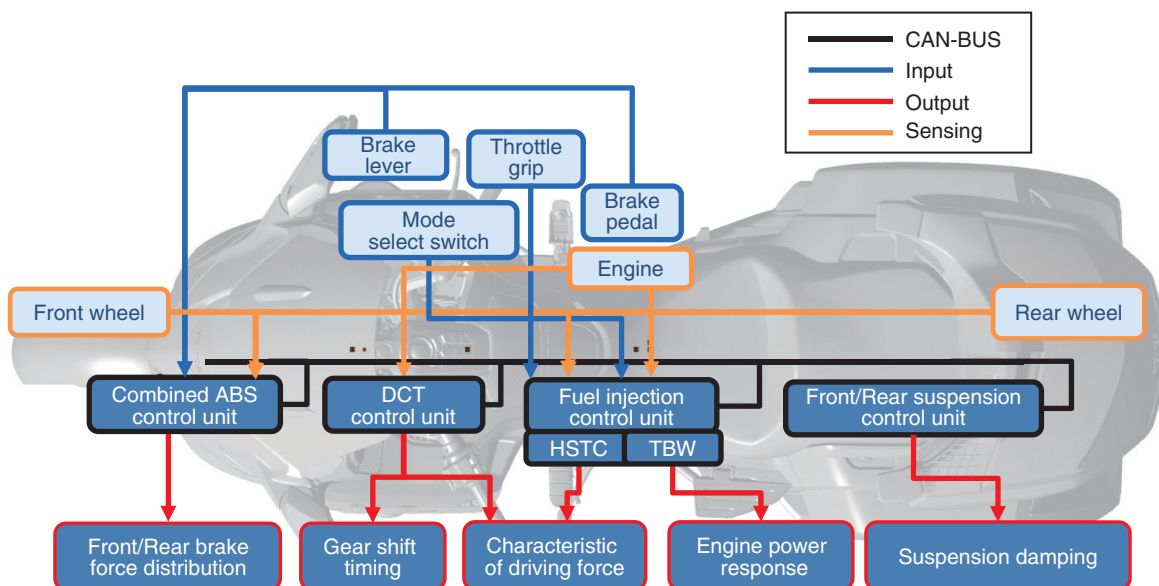


Fig. 4 Overview of system to control five factors

engine speed in each mode are shown in Fig. 6. Compared to the basic TOUR mode, the engine power rises more quickly when the throttle grip is opened in the SPORT mode to allow the rider to enjoy quicker acceleration. In the ECON mode, the engine power rises moderately to prevent abrupt changes of engine power when the throttle grip is opened. In the RAIN mode, the engine power response when the throttle grip is opened is further moderated from the ECON mode to ensure secure feel on slippery surfaces. The relationship between the throttle grip opening and the throttle valve opening to obtain the above-mentioned engine power characteristics is shown in Fig. 7.

4.2. Honda Selectable Torque Control (HSTC) System

HSTC is such a system that helps prevent excessive slipping of the rear tire by calculating rear wheel slip ratio from the vehicle speed signals detected by the wheel speed sensor located in the front and rear wheels, and regulates engine torque when the slip ratio exceeds the threshold. The engine torque is controlled by regulating the throttle

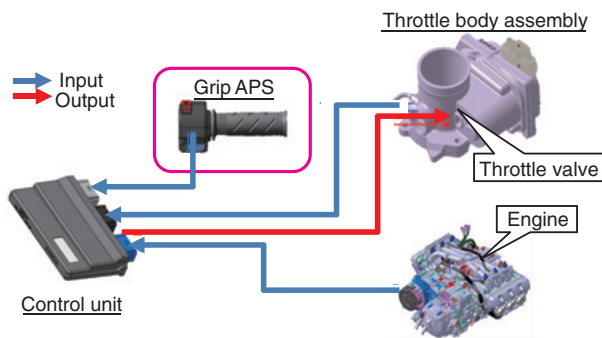


Fig. 5 TBW system

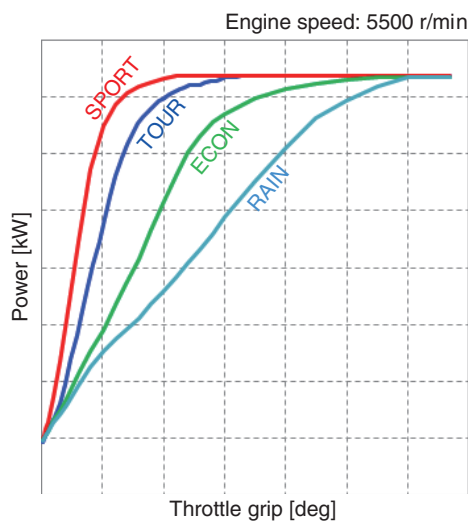


Fig. 6 Engine power characteristic in each mode

opening by TBW and adjusting fuel injection volume. This system controls engine torque in two stages; when the slip ratio exceeds the threshold, the control by TBW intervenes first, and as the slip ratio increases further and passes the next threshold, the control by fuel injection volume comes into play.

The relationship between the slip ratio and the engine torque in various modes is shown in Fig. 8. Due to the character of this model as an on-road tourer, the slip ratio threshold is set at the same level in all modes. The target of engine torque control is set differently in the SPORT mode from other modes. Compared to the TOUR mode in which the priority is given to a quick ceasing of tire slip on slippery surfaces, the target torque is set higher in the SPORT mode to allow quick acceleration even when the tire is slipping. Although the torque control setting is the same in the ECON and RAIN modes as in the TOUR mode, as the engine power response when the throttle grip is opened is set moderately as mentioned earlier, a wheel slippage is less likely.

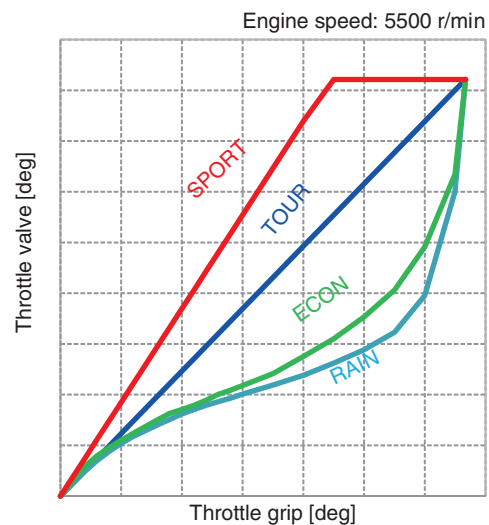


Fig. 7 TBW characteristics

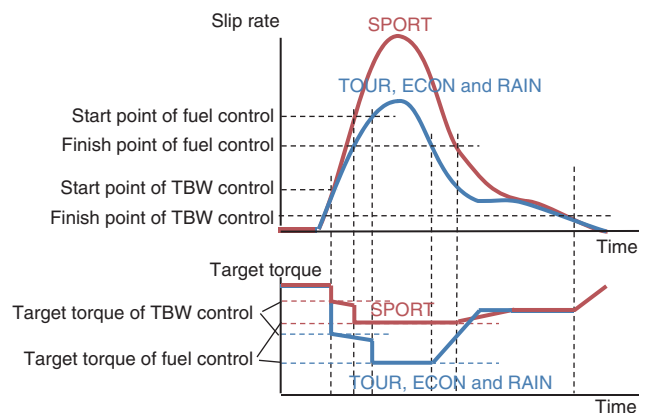


Fig. 8 Slip ratio threshold and target torque

4.3. Electronically Controlled Suspension

The electronically controlled suspension is such a system that alters damping of the front and rear suspensions by incorporating the stepping motor into the damping adjuster of the shock absorber. The damping characteristics of the front and rear shock absorbers are shown in Fig. 9. The damping force is altered only in the tension side. Compared to the basic TOUR mode, the damping force is set higher in the SPORT mode to reduce pitching motions from a hard acceleration or deceleration. In the RAIN mode, the damping force is set low to reduce changes of tire contact pressure on the road surface.

4.4. Electronically Controlled Combined ABS

The brake system applies the electronically controlled combined ABS in which the ABS is added to the dual combined brake that achieves optimum distribution of brake force in the front and rear wheels⁽¹⁾. The overview of the system is shown in Fig. 10. When the front brake lever is held, the brake force occurs in the front wheel. At the same

time, the hydraulic pressure in the rear caliper is increased by the ABS modulator having the pressure booster in it to generate brake force in the rear wheel. When the rear brake pedal is stepped, the hydraulic pressure in the rear brake master cylinder is distributed to the front and rear calipers after having the pressure regulated by the modulator.

In the SPORT mode, the rear brake force is set high when the front brake lever is held to control excessive pitching motions from the higher deceleration. The rear caliper pressure when the front brake lever is held is shown in Fig. 11.

4.5. The Difference of DCT Shift Timing for Each Ride Mode

The DCT is such a system in which the benefits of an automatic transmission are added to the manual transmission while maintaining the direct transmission of traction. Details of the DCT for this model are discussed in “Development of 7-speed Dual Clutch Transmission for 2018 Gold Wing”⁽³⁾. The control of shift timing

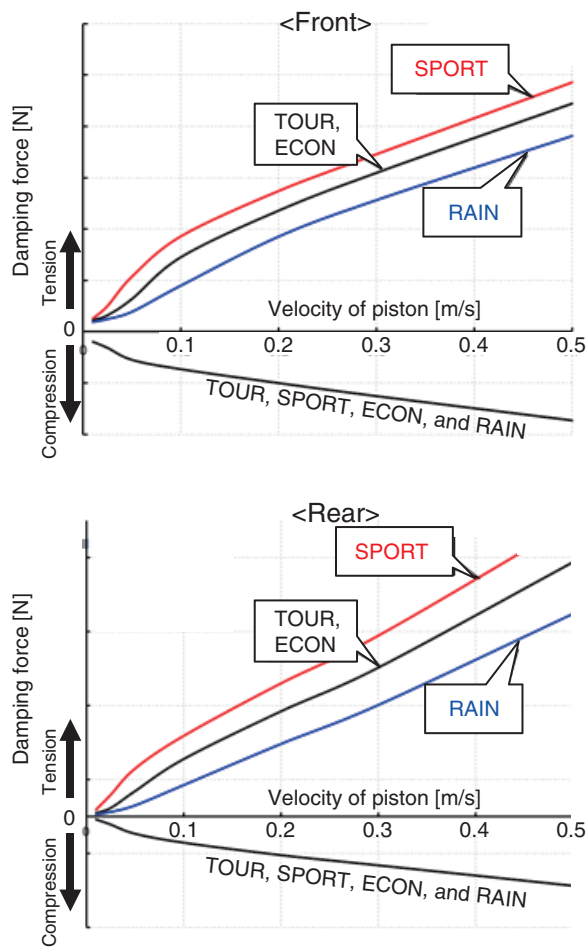


Fig. 9 Damping characteristics of front and rear shock absorbers

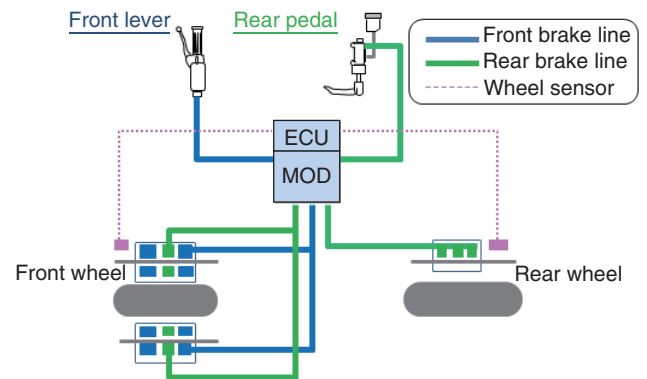


Fig. 10 Electronically controlled combined ABS

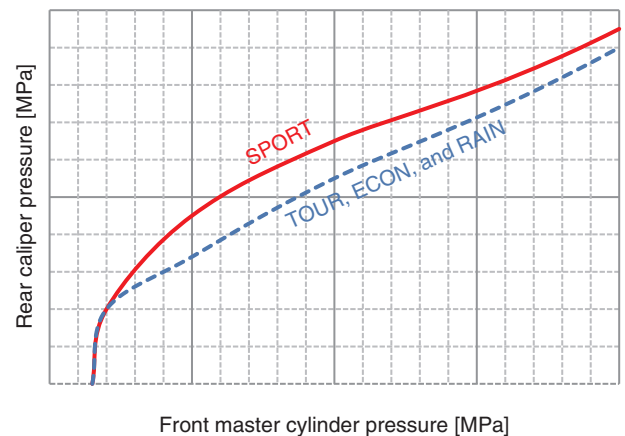


Fig. 11 Rear brake pressure when front brake lever is held

Table 2 Setting of each item in each mode

Mode	Condition	Characteristic	Engine power response (TBW)	Characteristic of driving force (HSTC)	Gear shift timing (DCT)	Suspension damping (Suspension)	Brake force distribution (Combined ABS)
TOUR	Versatile	The most balanced in various situations (The Gold Wing)	Throttle grip $\approx$ Throttle valve Response = Normal	Controls torque at rear wheel slip Control amount = Large	Optimum low speed cruise Shift timing = Normal	Offers riding comfort in various situation Damping force = Normal	Controllable brake characteristics having stable performance Rear brake pressure = Normal
SPORT	Winding Solo riding	Exciting with quick response	Throttle grip < Throttle valve Response = Quicker	Controls torque at rear wheel slip Control amount = Small	Holds gear up to high engine revolution Shift timing = Later	Controls pitching Enhances stability at acceleration/deceleration Damping force = Higher	Brake characteristics coping with frequent acceleration/deceleration Rear brake pressure = Higher
ECON	Suburb Congested (urban)	Comfort and relax with mild response	Throttle grip > Throttle valve Response = Milder	Controls torque at rear wheel slip Control amount = Large	Shift up at earliest timing Shift timing = Earliest	Offers riding comfort in various situation Damping force = Normal	Controllable brake characteristics having stable performance Rear brake pressure = Normal
RAIN	Wet/Dirt	More secure in low μ road condition	Throttle grip > Throttle valve Response = Mildest	Controls torque at rear wheel slip Control amount = Large	Shift up at earlier timing Shift timing = Earlier	Reducing changes of tire contact area on road surface Damping force = Lower	Controllable brake characteristics having stable performance Rear brake pressure = Normal

and the adaptive clutch capacity in each ride mode are discussed here.

Among the shifting characteristics, the gear shift timing is set for each ride mode as per the following. In the SPORT mode, the gear shift timing is set at higher engine speeds than in the basic TOUR mode to maintain lower gear to allow the rider to enjoy strong acceleration. In the ECON mode, the gear shift timing is set at lower engine speeds to quickly shift to the higher gear to enhance quietness and fuel economy. In the RAIN mode, the rear wheel traction is controlled by shifting the low-speed gears at the engine speeds close to the ECON mode to ensure smooth speed control by the throttle grip operation and to enhance controllability on slippery surfaces. Meanwhile in the high-speed gears, the gears are shifted at higher engine speeds close to that of TOUR to ensure an appropriate engine brake effect on slippery surfaces when the throttle grip is closed. The engine speeds to shift gears in each mode are shown in Fig. 12.

The dog clutches are applied in the DCT gears. The dog clutch has a free play in the direction of rotation. Accordingly, a big change of rear wheel torque occurs from colliding of dogs in the gears when the throttle grip is opened or closed, causing erratic motions of the motorcycle. To cope with this issue, the adaptive clutch capacity control is applied. The adaptive clutch capacity control is such that allows momentary slip of the clutch by lowering the clutch torque from the normal level when the throttle is opened or closed⁽²⁾ to reduce large torque fluctuations on the rear wheel when the dogs collide. As shown in Fig.

13, the clutch capacity in the RAIN mode is set 10% lower than in the TOUR mode. As a result, the rear wheel torque fluctuations by the throttle grip operation are reduced. Moreover, coupled with the TBW control characteristics shown in the earlier Fig. 6, the controllability of motorcycle on slippery surfaces is enhanced. Meanwhile in the SPORT mode, the clutch torque capacity is increased by 5% from the TOUR mode to achieve more direct feeling while reducing shocks from throttle operations.

The setting of control items in various modes are discussed so far.

The setting of each item in each mode is shown in Table 2.

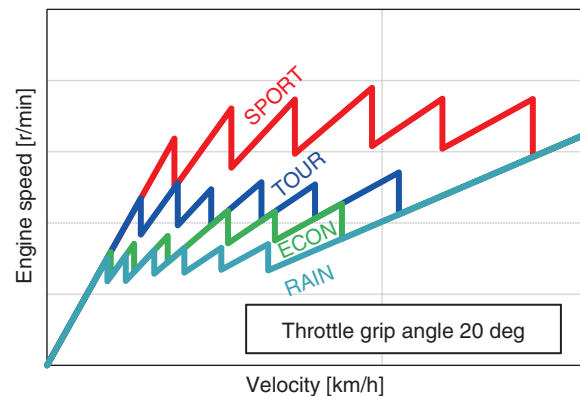


Fig. 12 Shift timing in each mode

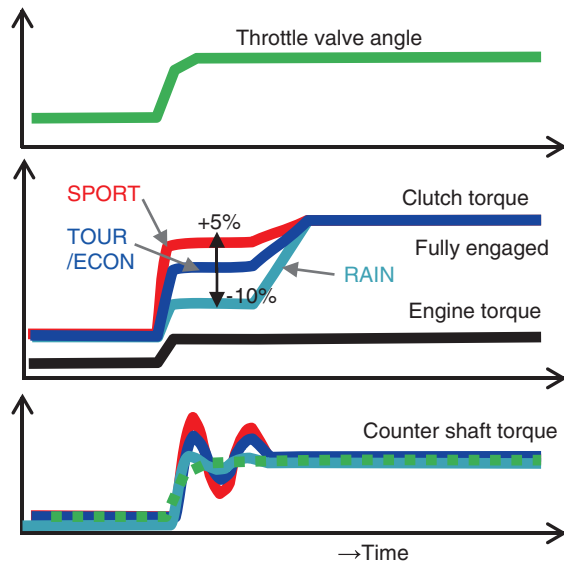


Fig. 13 Adaptive clutch torque in each mode

5. Effectiveness

Effectiveness of the control systems is discussed hereafter.

5.1. SPORT Mode; Enhancement of Dynamic Performance

The acceleration performance is one of the representative indicators of dynamic performance enhancement in the SPORT mode. The measured acceleration performance is shown in Fig. 14. When setting the test mode, conditions such as exiting a corner of a winding road, a general merge

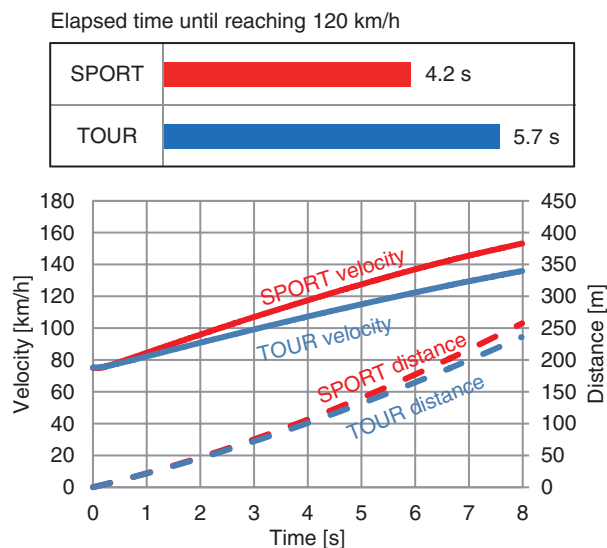


Fig. 14 Acceleration performance

into a freeway in the USA were presumed. Opening the throttle grip up to 75% in one second from the steady speed of 75 km/h (45 mph), the elapsed time until reaching 120 km/h (75 mph) was measured. In the SPORT mode, the time until reaching 120 km/h (75 mph) was shortened by 1.5 seconds from that in the TOUR mode. As shown in Fig. 15, when the throttle grip is opened in the same manner, while the throttle valve opening is restricted to 80% in the TOUR mode, it is 100% in the SPORT mode. The DCT shift pattern in the SPORT mode is such that selects a lower gear when accelerating. Due to the above-mentioned setup, the better acceleration has been achieved.

5.2. ECON Mode; Enhancement of Comfort and Economy

To clarify enhancement of comfort and economy in the ECON mode from the basic TOUR mode, the noise at rider's ears and the fuel consumption were measured. The measured noise level at rider's ears when traveling at the constant speed (75 km/h) is shown in Fig. 16. The cruising on general suburban roads in the USA was presumed for the test. The noise level at rider's ears is reduced by 3 dB (3.6%) in the ECON mode. This noise

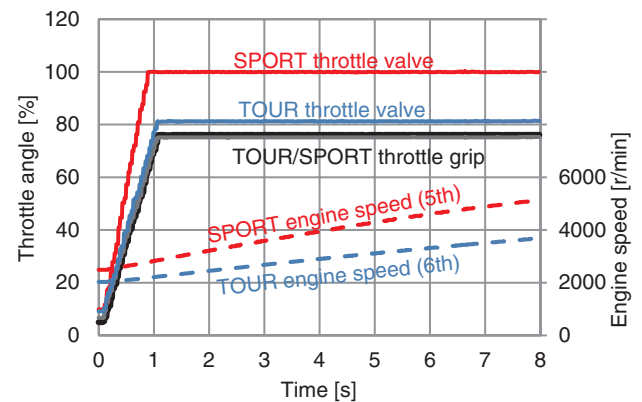


Fig. 15 Throttle angle and engine speed in each mode

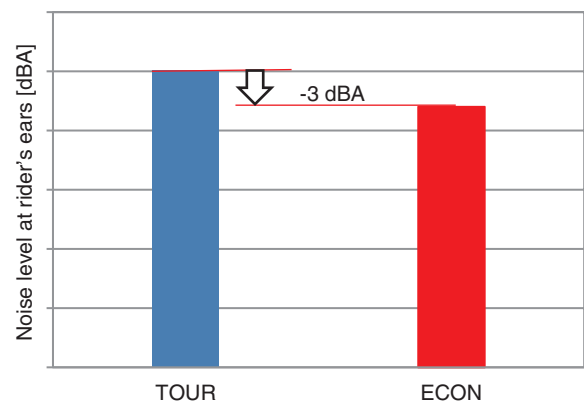


Fig. 16 Noise at rider's ears

reduction has been achieved by setting the DCT shift timing in such a manner that selects a higher gear than in the TOUR mode to lower the engine speed when running. The fuel economy measured in the test mode is shown in Fig. 17. For the tests, the US LA4 mode and the international standard WMTC mode were used. As the control of gear shift timing by the DCT is set in such a manner that allows running at higher gears when measuring fuel consumption in the ECON mode, the fuel consumption has been enhanced by approximately 3% in each mode.

5.3. RAIN Mode; Enhancement of Secure Feel

The rear tire slip ceasing is one of the representative indicators for enhancement of secure feel in the RAIN mode. The ceasing of tire slip was measured and shown in Fig. 18. The tests were conducted on a medium friction surface having a friction coefficient of 0.4, and the vehicle speed approximately 35 km/h (first gear) which is the engine speed at 80% where the maximum torque occurred, the throttle grip was opened fully in 0.3 second. The time period from the start of the torque control until the once-increased rear tire slip ratio lowered to 30% was measured. It was clarified that the tire slip ceasing time in the RAIN mode was approximately 200 msec, which was the shortest. The tire slip ceasing time in other mode is also within the level that ensures secure feel for the rider (SPORT: approx. 300 msec, TOUR/ECON: approx. 250 msec).

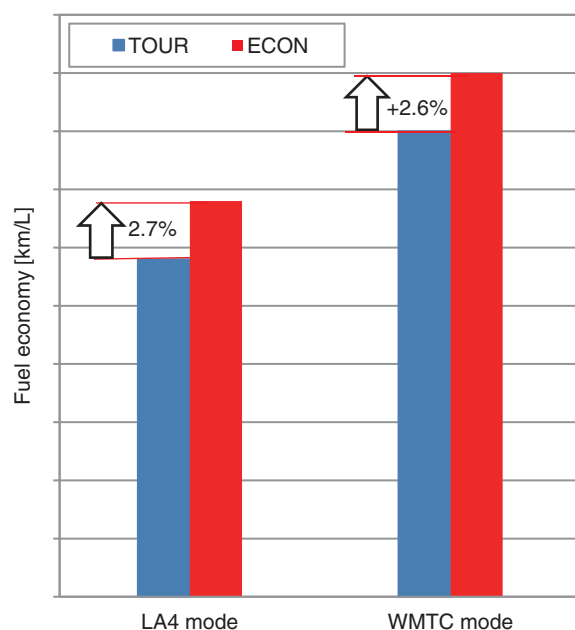


Fig. 17 Fuel consumption

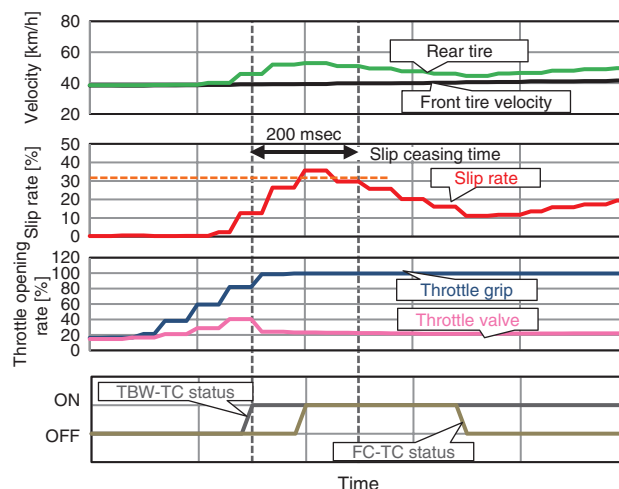


Fig. 18 Test result of tire slip ceasing

6. Conclusion

The following three, which are the aims of development of ride modes for various conditions, have been accomplished, allowing extension of performance zone that the rider can enjoy.

In the SPORT mode, the acceleration from the steady speed has been enhanced, which is an indication of an enhancement of the dynamic performance in comparison with the TOUR mode.

In the ECON mode, the noise level at rider's ears when running at the steady speed has been lowered, and the fuel economy in the test modes has been enhanced, which are an indication of enhancement of comfort and economy in comparison with the TOUR mode.

In the RAIN mode, a quick ceasing of tire slip on the medium friction surface has been confirmed to ensure secure feel for the rider in comparison with the TOUR mode.

The world's first commercial motorcycle (according to Honda's research) having the ride modes for various conditions has been realized by the integrated control of five factors (engine power response, characteristic of driving force, gear shift timing, suspension damping, front/rear brake force distribution). The newly-developed system provides the riding performance that allows the rider to control the motorcycle as the rider wants in various conditions.

References

- (1) Nishikawa, Y., Minamisato, T., Takenouchi, K., Tani, K., Takayanagi, S., Fukaya, S.: Development of ABS for Super sports Motorcycles Using a Brake-by-Wires

- System, JSAE, No. 65-09, p. 15-18 (2009)
- (2) Yokogawa, Y.: Introduction of Dual Clutch Transmission System for Motorcycles, JSAE Vol. 71, No. 9, p. 33-38, (2017)
- (3) Hirayama, T., Yokogawa, Y., Fujimoto, Y.: Development of 7-speed Dual Clutch Transmission for 2018 Gold Wing, Honda R&D Technical Review, Vol. 30, No. 2, p. 31-37

■ Author ■

Atsushi KUSUDA



Yukio YOKOGAWA



Kazuho NAKAI

Development of Double Wishbone Type Front Suspension for 2018 Gold Wing

Naoki KUWABARA* Junpei MISHIRO*

ABSTRACT

In developing the front suspension system for 6th generation 2018 Gold Wing GL1800, we aimed to realize both agile handling and high stability at the same time, and a comfortable and high quality riding experience. To this end, we developed a Honda original double wishbone type suspension system for motorcycles. By devising the arm layout, the stroke trajectory of the front axle was set in the vertical direction, and the amount of rearward movement of the tire was reduced. This made it possible to allow forward shifting of the engine position, which in turn increases the front divided load, and we were able to realize both agile handling and high stability at the same time. In addition, by using needle bearings for all pivots and at the same time having a structure in which lateral force does not act on the cushion, the sliding resistance was able to be reduced as compared with the telescopic type suspension. As a result of all these enhancements, the vertical acceleration of sprung mass was reduced by about 30% compared with the previous model in traveling roads such as an uneven road, contributing to the realization of a high quality and comfortable riding experience. Furthermore, in this system, the handlebar pivot axis is offset backward from the wheel steering pivot axis, and tie rods are used to connect between the pivots. This mechanism effectively shortened the length of the handlebars and enhanced operability in situations close to full steering.

1. Introduction

The 2018 Gold Wing GL1800 has undergone the first full model changeover, in 17 years since the launch of the 5th generation model in 2001. We aimed to develop “the motorcycle with the most enjoyable maneuverability throughout all the models appeared in the chronicle of the Gold Wing.”, by providing the feeling of high quality in the maneuverability, running performance, and comfort. To achieve this, the following two items were set as the goal.

- Realization of packaging to enable both agile handling and high stability
- Realization of comfortable and high quality riding comfort

To realize both agile handling and high stability, it was found necessary to move the engine as well as the riding position forward to increase the front divided load. Furthermore, to achieve a comfortable and high quality ride, it was required to develop a suspension system with low sliding resistance and high rigidity. Since the first generation of Gold Wing, we have been using telescopic type suspension for the front suspension system. However, to achieve the high-level goal of this development, a

completely new suspension system was required being apart from the conventional structure. As a result of comparing various types of suspension systems, the front suspension which is most suitable for this model was a double wishbone type suspension system.

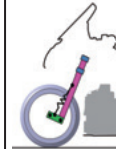
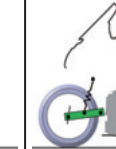
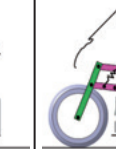
This paper introduces the double wishbone type front suspension system newly developed for the 6th generation Gold Wing GL1800.

2. Selecting the Type of Suspension

In selecting the type of suspension, we considered various factors besides the required performances including the external appearance suitable to the status of the new grand tourer as well as mounting capability of highly advanced systems. **Figure 1** shows the result of the comparative study. From the result, it became clear that the double wishbone type suspension, which adopts the needle bearing for the pivot which is advantageous for reduction of sliding resistance and high rigidity, is the most effective in achieving the goal of this model.

On the other hand, in the previous model, while a telescopic type suspension was adopted as a front

* Motorcycle R&D Center

Type		Telescopic		Bottom link		Arm link	
		Upright (previous model)	Inverted	Leading	Trailing	Hub steer	Bearing pivot DWB* link
							
Performance	Shift engine position forward	BASE	≡	≡	≡	+	+
	Reduce sliding resistance	BASE	≡	+	+	+	++
	Increase stiffness	BASE	+	-	-	-	+
	Reduce steering moment of inertia	BASE	≡	-	-	+	+
	Weight	BASE	≡	-	-	-	-
Market-ability	Innovativeness	BASE	≡	≡	≡	+	+
	Appearance	BASE	≡	≡	≡	+	+

Change from base model

Comparable:



Superior:



Inferior:



*DWB: Double wishbone type

Fig. 1 Comparison of suspension types

suspension, we aimed at compatibility between relaxed position like grand tourer and slow speed maneuverability. For this reason, the forward and backward offset amount between the handle grips and steering was long, and the width between the grips was widened. Therefore, with this structure, the amount of movement of the grip end during the steering increases. In applying this handle structure to newly developing model, the issue was that one grip point becomes very close to the rider's side and the other grip point becomes very far at full steering. To solve the issue, we devised a new system in which the handlebar pivot axis and the wheel steering pivot axis are made separate axes, the handlebar pivot axis is offset rearward, and tie-rods are used to connect between the pivots. Based on the above idea, we have selected a double wishbone type suspension system using a needle bearing for all pivots, with the handlebar pivot axis and the wheel steering pivot axis are separate axes in newly developing model.

3. Suspension Structure

Double wishbone type suspensions are fairly popular in four wheel vehicles. As this type of suspension can realize superb riding comfort and high controllability, it is popularly used for high-class automobiles, sports cars, and F1 racing machines. For example, Honda is using it for the NSX, the Legend, and other models (Fig. 2)⁽¹⁾.

The specific features of double wishbone type suspensions are listed below:

- Smooth operation due to no side force on the cushion unit;
- High structural rigidity with upper and lower arms to support the wheel; and
- High freedom of alignment design to determine the caster angle and camber angle.



Fig. 2 Double wishbone type suspension for automobiles

While inheriting these features, the double wishbone system was arranged for the newly developing model by changing the wheel to arm mounting angle by 90 degrees to create a unique configuration for use on a motorcycle. The structure has two arms to support the front fork and fork holder that move up and down. Figure 3 illustrates the structure of the suspension. This system consists of three portions, i.e., wheel steering portion, shock absorbing portion, and handlebar pivot portion.

The wheel steering portion consists of the front fork and the steering arm coupled to it. These are aluminum die casted parts of thin-walled hollow structure. By making it an integral unit to support the wheel from both sides, a highly rigid, light-weight structure was created. Unlike conventional telescopic forks, this structure does not have shock absorbers. As a result, the rotational moment of inertia of the steering system was reduced by more than 40% from the previous model.

The shock absorber portion is structured to support the fork holder corresponding to a steering head tube at the forefront of a conventional motorcycle frame by the upper and lower arms, and a cushion unit is attached to the lower arm. The point of this structure is that no bending force is applied on the cushion unit. Also, while it is generally the case with four wheel vehicles that ball joints or rubber bushings are used for the joints of links, all pivots of this structure use needle bearings and four-point contact ball bearings, it is structured to obtain high rigidity and smooth rotation without play. Furthermore, by properly arranging

the arms, the system is designed to move the front wheel vertically against the ground, or to make the stroke trajectory of the wheel vertical, to reduce the backward movement of the front tire by 24 mm from the previous model (Fig. 4).

The handlebar pivot portion consists of the steering bridge, handlebars, and tie rods. By connecting sprung mass (i.e. the handlebar pivot portion) and unsprung mass (i.e. the wheel steering portion and the shock absorber portion) with special tie rods which were reduced play, this structure provides high quality riding comfort as well as the feel of direct maneuverability.

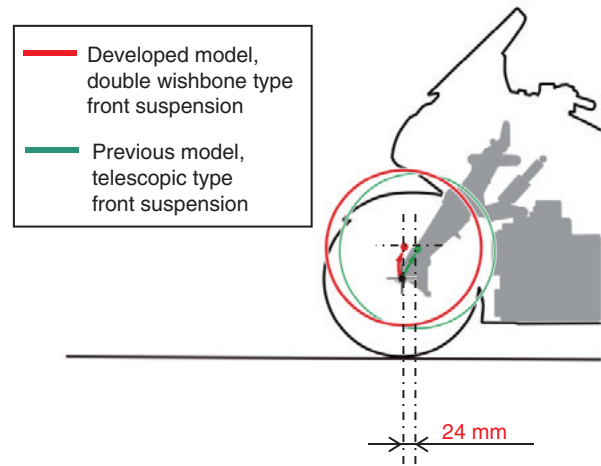


Fig. 4 Comparison of front tire stroke trajectories

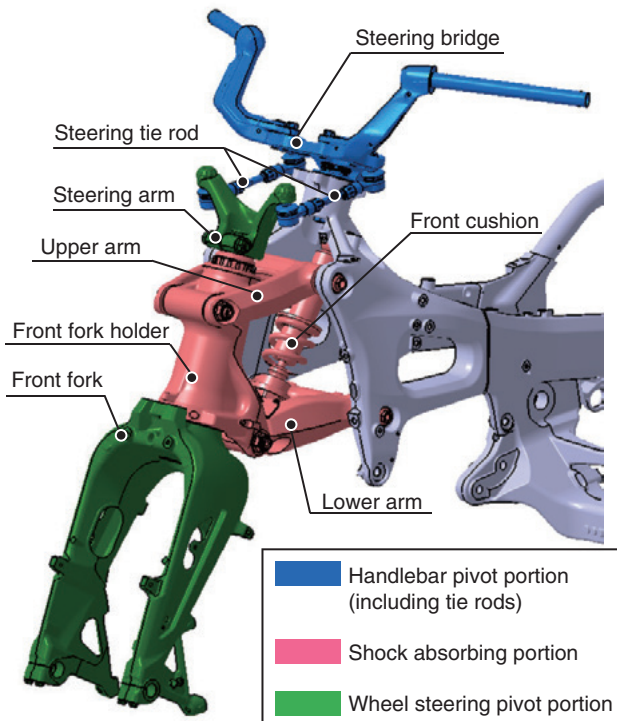


Fig. 3 Front suspension of 2018 Gold Wing

4. Packaging to Enable Both Agile Handling and High Stability

As already mentioned, because the backward movement of the front wheel was reduced by 24 mm, we were able to shift the engine mounting position forward. By doing so, the front divided load ratio was increased to 46.5% in DCT model compared to 44.9% of the previous model. In addition, while maintaining the living space of the rider and the passenger, it was possible to shift the riding position forward by 36 mm together with the compactness of the engine (Fig. 5). As a result, the aim of enhanced maneuverability and body size reduction were achieved as intended. Furthermore, by making the distance between the wind screen and the rider closer, we were able to use a smaller wind screen to gain the same or better wind protection effect compared to the previous model.

While the conventional telescopic type suspension requires some space to allow the movement at the top of the right and left forks for the steering, the new system does not require such a space because only the wheel

steering pivot is to be steered. So, the width of the front cowl can be reduced. As a result, together with the size reduction of the side radiator, we were able to reduce the front cowl width by about 100 mm around the wheel steering pivot (Fig. 6). In addition, by separating the handlebar pivot from the wheel steering pivot, we were able to lower the height of the instrument panel above the wheel, contributing to lowering the center of gravity of the motorcycle. Furthermore, being helped by the increased freedom of design to position the handlebar, we were able to increase the clearance between the rider and the handlebar in the fully steered position by 90 mm for enhanced operability (Fig. 7).

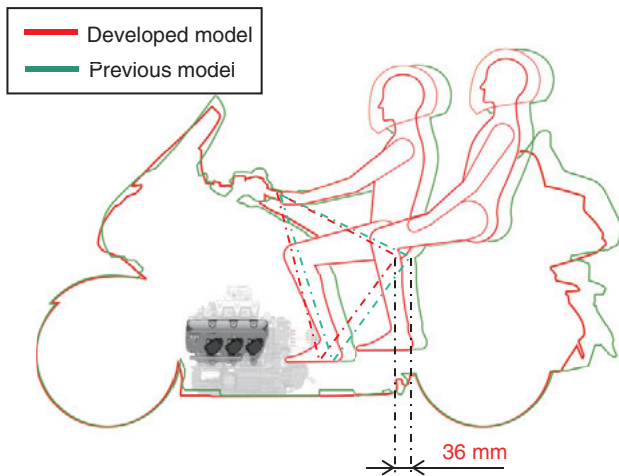


Fig. 5 Engine location and riding position

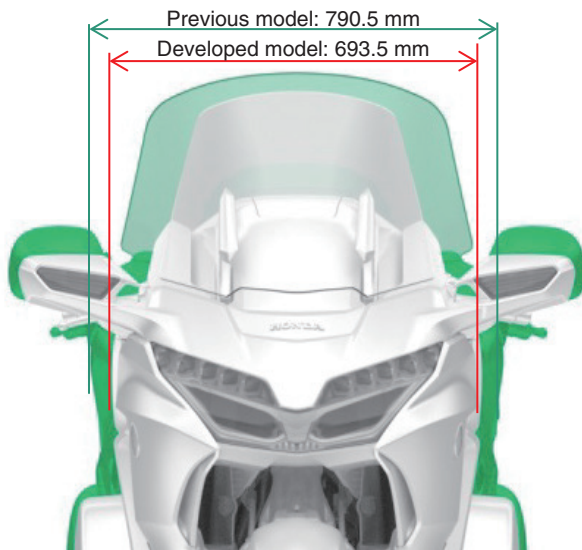


Fig. 6 Comparison of front cowl widths

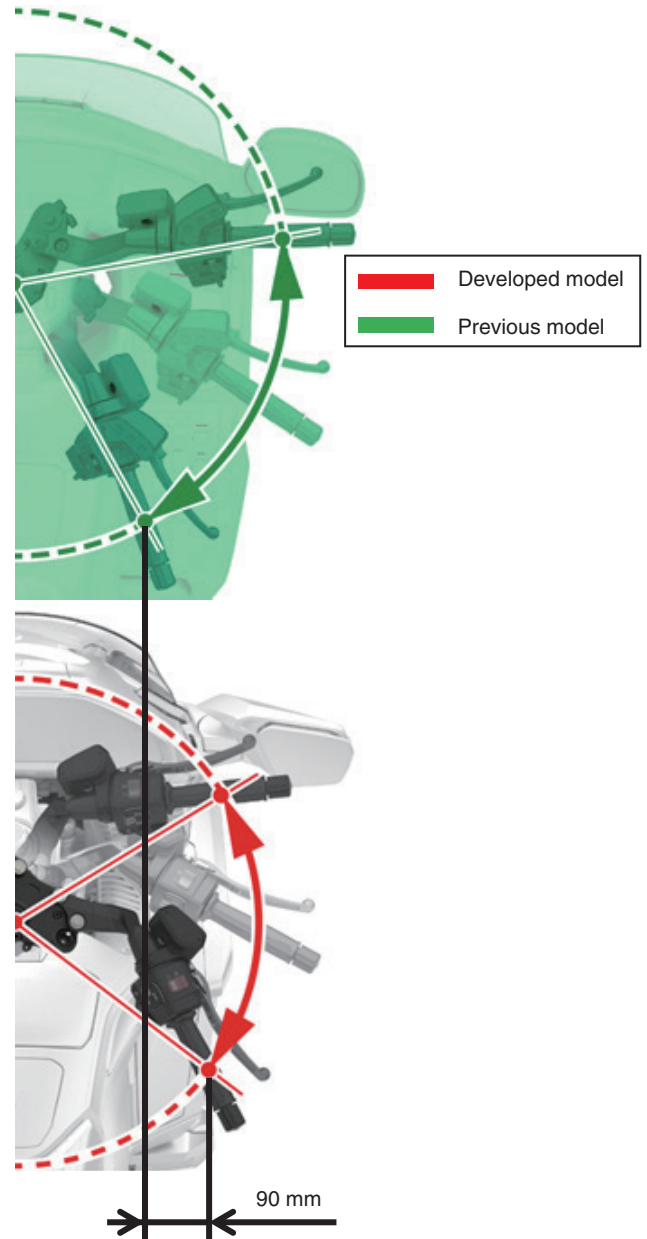


Fig. 7 Comparison of handle grip positions at full steering

5. Realization of High Quality Riding Comfort

To check the performance achieved of the double wishbone type front suspension described in Section 3, we performed actual running measurement test. We measured the vertical acceleration of the sprung mass by running the vehicle at 60 km/h on continuous steps of 10 mm - 20 mm high. Figure 8 shows the results. From the figure, we can see that the peak acceleration level is lower than that of the previous model. Next, Fig. 9 shows the average values of 10 samples of peak acceleration. From the comparison, we

can see that the vertical acceleration of the sprung mass is reduced by 30% from the previous model.

Then, to verify the mechanism of the performance enhancement, we measured the acceleration of both sprung mass and unsprung mass and the pitch angular velocity of the body by running the vehicle at 100 km/h on a single step of 20 mm high. Figure 10 shows the time-series data of the vertical acceleration of the sprung mass. The developed model shows lower peak acceleration and quicker convergence than the previous model. Figure 11 shows the time-series data of the pitching rate. We can see that the developed model shows the level of higher frequency components is lower and the overall behavior is gentle as compared to the previous model. It can be inferred

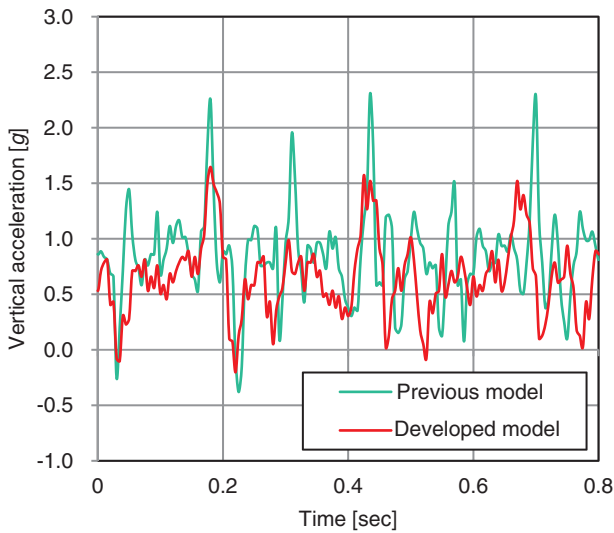


Fig. 8 Sprung mass vertical acceleration when running on continuous steps

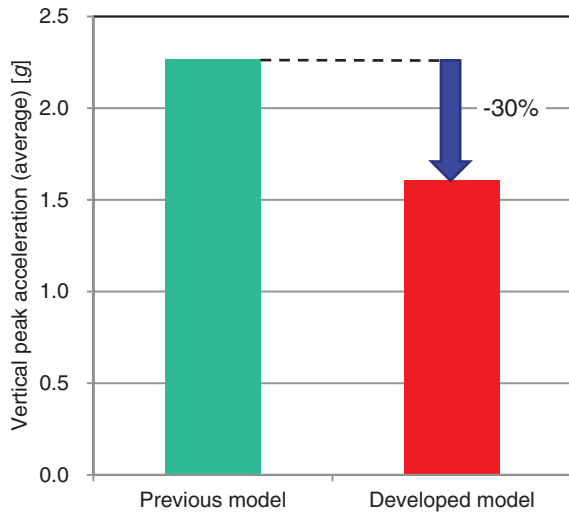


Fig. 9 Average of sprung mass peak acceleration when running on continuous steps

that these characteristics are connected to the high quality feeling. Furthermore, we performed spectrum analysis on the longitudinal acceleration of the unsprung mass. The result is shown in Fig. 12. In the case of the previous model with telescopic type front suspension, we see peaks at

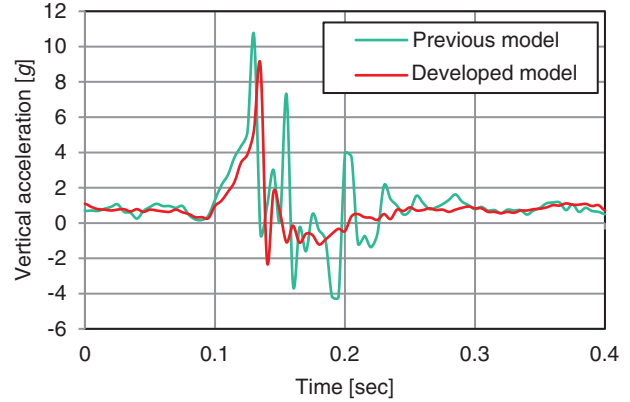


Fig. 10 Vertical acceleration of sprung mass when passing through step ($h=20$ mm)

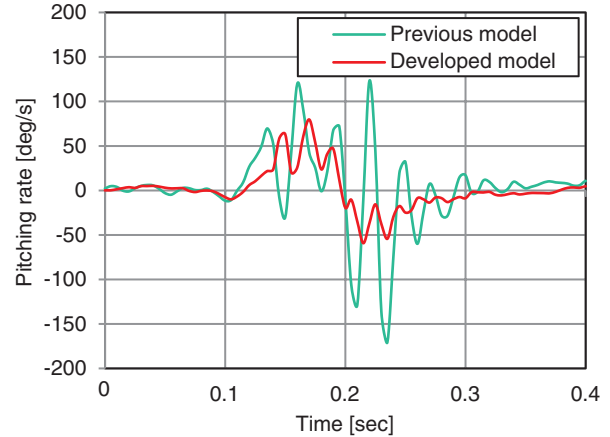


Fig. 11 Pitching rate when passing through step ($h=20$ mm)

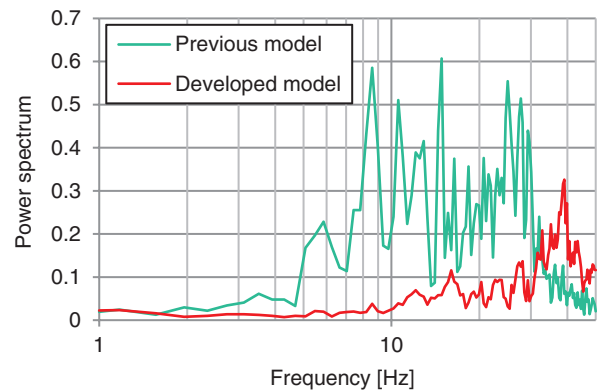


Fig. 12 Power spectrum of longitudinal acceleration of unsprung mass when passing through step ($h=20$ mm)

around 10 Hz and 25 Hz. These peaks are considered to be showing the longitudinal vibration caused by the bending motion of the front fork. On the other hand, the peak spectrum for the new model is higher, i.e., at around 40 Hz, and it shows that the bending stiffness of the suspension is higher and thus longitudinal vibration is minimized.

From these results, we can see that the suspension system of the developed model can operate smoothly while absorbing the inputs from the road surface, and thus it provides a high quality riding comfort.

6. Conclusions

For realization of packaging to enable both agile handling and high stability, and realization of comfortable and high quality riding comfort, we developed a new front suspension. We adopted a double wishbone type suspension for motorcycles with a unique structure separating the handlebar pivot axis from the wheel steering pivot axis. With this suspension structure, it became possible to change the tire stroke trajectory and to mount the engine forward. As a result, the front divided load increased, realizing both agile handling and high stability at the same time. By adopting this structure with reduced sliding resistance by adopting needle bearing for the pivot of each part of the link, the acceleration level of the sprung mass was reduced by 30%, the convergence of pitch motion was enhanced, and a high-quality riding comfort was achieved.

Reference

- (1) <http://www.honda.co.jp/NSX/webcatalog/performance/driving/>

■ Author ■



Naoki KUWABARA



Junpei MISHIRO

Development of Robotic Lawnmower Miimo HRM3000

Susumu OKUBO*

Kenta KAWANISHI*

Makoto YAMAMURA*

Takuya KANISAWA*

Dai HYAKUSAWA*

ABSTRACT

A robotic lawnmower has been developed that mows lawns efficiently and simplifies placement, cleaning, and operation. It uses an efficient method for turning within the mowing boundaries to shorten the time required for mowing.

In addition, in order to facilitate mower cleaning, the robotic lawnmower is the world's first to be given a waterproof frame that is washable with a garden hose.

Also, whereas existing robotic lawnmowers have limited options for where to place the docking station for charging, this robotic lawnmower was enabled with a system for returning to the docking station from any angle in response to user demand for as much freedom as possible in station placement.

Furthermore, the mower has a likable exterior design and can be linked with smartphones.

1. Introduction

Many people perceive mowing as a chore and want a lawnmower that will give them more leisure time.

In 2012, Honda released two robotic lawnmowers: the Miimo HRM300 and Miimo HRM500. They were then remodeled in 2015 and 2016, respectively, and the model names were changed to the HRM310K1 and 520K1 (hereinafter the conventional models).

Since their release, the conventional models have met demand for keeping lawns beautiful without expending the time or effort⁽¹⁾.

In Europe, where robotic lawnmowers are prevalent, competition is fierce. In addition to recent offerings by European manufacturers, Chinese manufacturers have entered the market. Research is also active with multiple studies⁽²⁾.

Honda has newly developed the HRM3000 as a premium model as shown in Fig. 1. This paper will

showcase the features of this model: namely, technology for mowing lawns more quickly; the washable, waterproof structure; technology increasing the freedom of docking station placement; and other new functions.

2. Development Goals

Prior to the development, Honda has conducted European market research to analyze user demands and issues with existing robotic lawnmowers.

The results showed that demand was high for the following four items:

- (1) Shortening mowing times by using the mower body for autonomous lawn mowing (hereinafter the mower) to increase time for tending to flower gardens and plants and playing in the yard with children
- (2) Simplifying the laborious process of cleaning clingy grass from the mower blade area and underside after mowing
- (3) Installing the docking station somewhere inconspicuous to help prevent theft and keep the yard landscape beautiful
- (4) Simplifying the post-purchase configuration procedure and the restarting process when the mower stops unexpectedly or has other issues

Based on the above user demands, the development goals for the HRM3000 were as follows:

- (1) Shorten mowing time from conventional models
- (2) Make mower washable with garden hose
- (3) Increase the freedom of docking station placement
- (4) Simple, easily-understood operation



Fig. 1 Robotic lawnmower Miimo HRM3000

* Power Products R&D Center

3. HRM3000 Configuration and Design

3.1. HRM3000 Configuration

The HRM3000 robotic lawnmower comes with the aforementioned docking station and mower, as illustrated in Fig. 2. The docking station emits various signals, including signals to the boundary wire for setting the mowing boundary and a station signal for giving the position of the docking station.

A structural drawing of the mower is given in Fig. 3, and a specifications comparison with the conventional model HRM520K1 is given in Table 1. The mower comprises a mowing blade motor, motors for driving each of the rear left and right wheels, a trimming height adjustment motor, a battery power source, and a control board to control mower behavior. It also comes equipped with a display for showing device status and inputting mowing settings, as well as an area sensor for sensing the magnetic fields of the boundary wire and station signals. Users can configure and operate the mower via the display or through a smartphone using Bluetooth® wireless communication with the control board. As detailed below, smartphone-only functionality has also been added.

3.2. Design

3.2.1. Basic concept

The basic concept for this model is to clearly promote Honda's brand for robotic lawnmowers. As shown in the

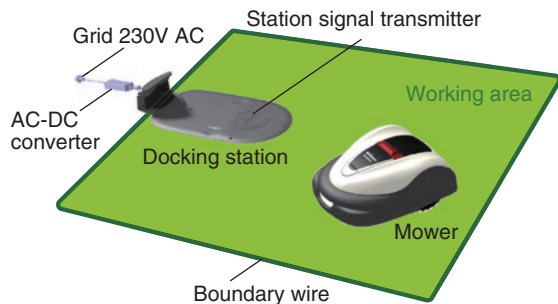


Fig. 2 Robotic lawnmower configuration

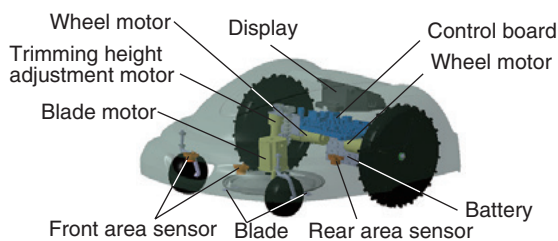


Fig. 3 Mower components

Table 1 Comparison of specifications

Model		HRM520K1	HRM3000
Overall length (mm)		645	710
Overall width (mm)		550	
Overall height (mm)		275	301
Wheelbase (mm)		300	326
Mass (kg)		11.9	13.8
Wheel diameter (mm)	Front	100	
	Rear	200	250
Forward speed (m·s ⁻¹)		0 - 0.55	
Trimming width (mm)		220	250
Trimming height (mm)		20 - 60	
Blade rotating speed (min ⁻¹)		2500	2290
Blade type, Number of blades		Swing-back type, 3 blades	
Motor performance (W)	For driving	10.8 x 2	17.6 x 2
	For mowing	28.8	40.8
	For trimming height adjust	-	3.2
Battery	Type	Li-ion	
	Capacity	22.2V x 3.6Ah	22.2V x 5.4Ah
Sound power level (dB(A))		60.5	59

basic concept illustration in Fig. 4, the product has been designed to promote the luxury of the premium model as sensible added values and to get users to develop an affinity and loyalty in using the product.

3.2.2. Mower design

As factors in increasing human affinity and loyalty to robots, Honda engineers hypothesized that movement and creature-like feel are more important than external shape. Based on this assumption, they focused on producing a creature-like feel with the smooth turning as described in Section 4.1.1. and styled the mower with organic curves reminiscent of that movement as shown in Fig. 1. For coloring, the available colors were based on the conventional models to build the overall brand appeal of Honda's robotic lawnmower line-up.



Fig. 4 Basic concept illustrations

4. New Technologies and Functions

4.1. Increased Mowing Efficiency

The mower should efficiently mow every inch of the mowing area in order to trim grass to the set height and complete the job in a short time. Therefore, the mower's trimming width was widened and driving methods were modified to enable the mower to mow more quickly.

4.1.1. Smooth turning

Mowing areas are shaped in varying levels of complexity, meaning that the mower may travel over a section multiple times depending on mower location. Such route overlap will increase mowing time. For this reason, the conventional models were designed to traverse every inch by turning at randomized angles once they reached the mowing boundary⁽¹⁾. As shown in the turning method illustration in Fig. 5(a), the mower stops once reaching the mowing boundary. The mower then transitions to its turning motion, which requires an excess of time due to its stopping. Therefore, in order to reduce mowing time, Honda engineers introduced a smooth turning motion with no stopping, as illustrated in Fig. 5(b).

With smooth turning, the mower slows down when near the mowing boundary while turning at a randomized angle and then continues moving. The mower receives the magnetic field intensity sent by the boundary wire via a front area sensor, calculates the distance to the boundary wire, and starts its turn to continue moving when it reaches the edge of the mowing boundary. This motion reduces time loss from stopping for turns and enables faster mowing times.

However, while smooth turning reduces loss from stopping time, the mower sometimes misses spots in corners or along the mowing boundary. Thus, the turning methods in Figs. 5(a) and 5(b) were combined as shown in Fig. 5(c) to make the mower traverse the entire mowing area more evenly. Hereinafter, this method is referred to as mixed turning. A time was also set for switching between turning methods. Taking this switching time and the range of turning angles as parameters, the engineers ran simulations to solve for efficient mowing values in areas of

various shapes. As given in Fig. 6, the lawn shape used in the simulations was set by European industry standards⁽³⁾. Mowing efficiency was measured as the time it takes the mower to pass over 95% of the area within the working boundary. The results confirmed that the mixed turning method shown in Fig. 7 enhanced mowing efficiency more than 20% over the conventional models.

In Fig. 8, the mowing areas are filled in for the two turning methods over a 240-minute simulation. The unmowed areas are shown in black, areas mowed only once in yellow, and those mowed twice or more in red.

4.1.2. Trimming width

The amount mowed is also important to shortening the mowing time. Given that enlarging the trimming width

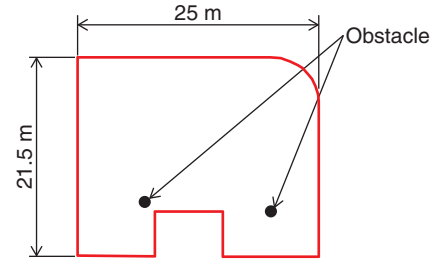


Fig. 6 496 m² working area

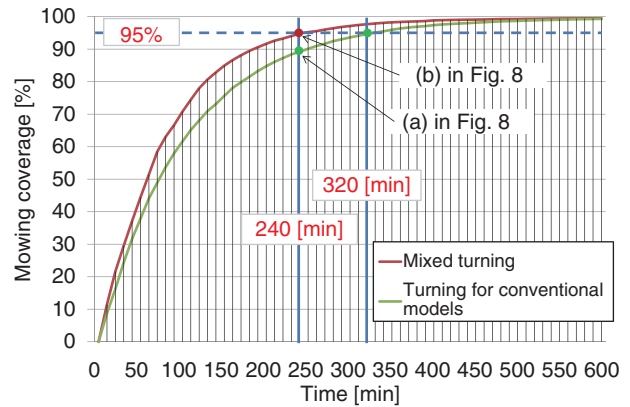


Fig. 7 Simulation results

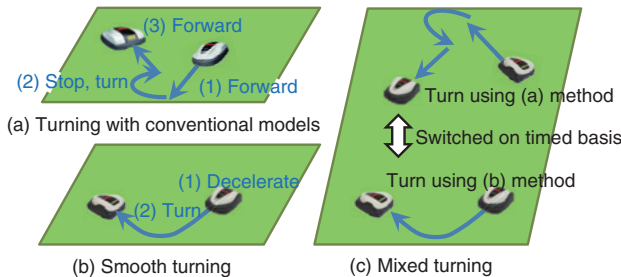


Fig. 5 Turning method

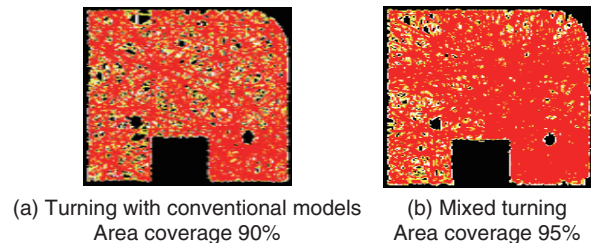


Fig. 8 Simulated mowing coverage at 240 mins.

will increase the amount mowed, trimming width was changed from 220 mm for the conventional models to 250 mm. Enlarging the trimming width also allows for the blade speed to be set lower, which reduces noise while also increasing the amount mowed. The results are given in Table 1: The sound power level is kept at or below that of the conventional models.

4.1.3. Results of increased efficiency

The HRM3000 was equipped with the parameters as solved in the simulations and run on an actual lawn. The resulting mower track as observed by motion capture is shown in Fig. 9. The times for both methods in attaining a 95% mowing area coverage matched those in the simulations from Fig. 7, illustrating that the new method can complete the task in a shorter period than the conventional method on a real lawn.

4.2. Washable Mower

If the robotic lawnmower is left to continue autonomous operation, grass trimmings will stick to the underside of the mower as seen in Fig. 10 and require removal. As seen in Fig. 11, washing with a garden hose or other running water source is effective in washing the unit quickly. However, as of 2016, neither the conventional models nor competitor robotic lawnmowers were warranted to be resistant to running water. Therefore, a boot and waterproof aeration filter were added to the conventional models for a waterproof structure on the HRM3000 as shown in Fig. 12. Furthermore, the trimming height adjustment mechanism was motorized and contained within the waterproof frame to increase water resistance.

With this structure, water resistance as stipulated by international standard IP level⁽⁴⁾ was increased from IPX4, which protects against splashes, up to IPX5, which protects against water jets. IPX4 has been the industry standard for

robotic lawnmowers to date. This makes the HRM3000 the world's first hose-washable robotic lawnmower.

4.3. Docking Station Placement

Various user requests, including theft concerns and preserving the lawn landscape, required docking station placement to be made appropriate for each lawn. Also, it



Fig. 10 Grass trimmings on mower underside

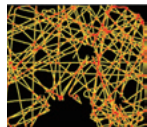


Fig. 11 Washing

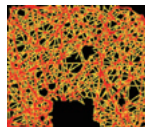
HRM3000



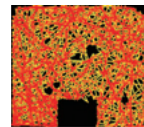
Time : 10 min
Coverage : 12.5%



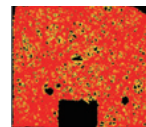
: 50 min
: 49.8%



: 100 min
: 75.5%

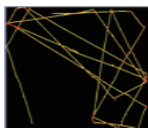


: 150 min
: 86.5%

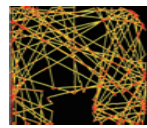


: 240 min
: 95.2%

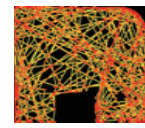
Conventional model



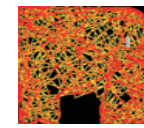
Time : 10 min
Coverage : 11.5%



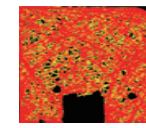
: 50 min
: 43.0%



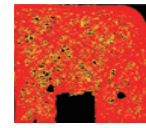
: 100 min
: 65.2%



: 150 min
: 79.7%



: 250 min
: 91.6%



: 330 min
: 95.0%

Fig. 9 Actual mowing coverage observed by motion capture

is important to help ensure that the mower returns to the docking station to charge its battery so that it can continue mowing.

For the conventional models, the docking station could only be placed somewhere parallel to the boundary wire. In response to the user demands, HRM3000 docking station placement options were expanded to include locations perpendicular to the boundary wire, recessed areas, and boundary wire corners as given in Fig. 13.

Expanding placement options for the docking station requires the mower to be able to return to the docking station from any angle. Therefore, a station signal system was applied to the docking station, enabling the mower to detect the station location from any direction.

For the station signal system, a circular wire installed on the bottom of the docking station constantly sends out a uniform magnetic signal to the surrounding area. Fig. 14 depicts the magnetic field generated by the station signal.

When the conventional models return to their docking station, they travel along the boundary wire as shown in Fig. 15. Given the weak range of the station signal's magnetic field, however, the mower can only detect the signal when near to the station and can only approach the station from one direction.

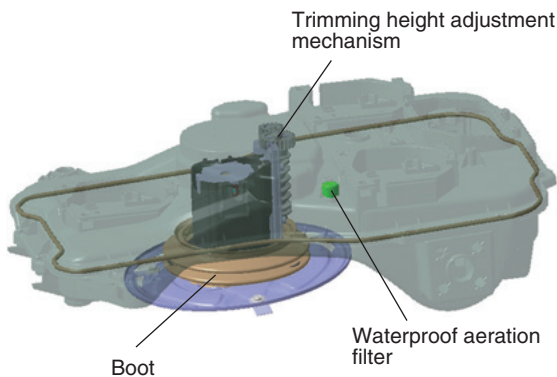


Fig. 12 Waterproof frame

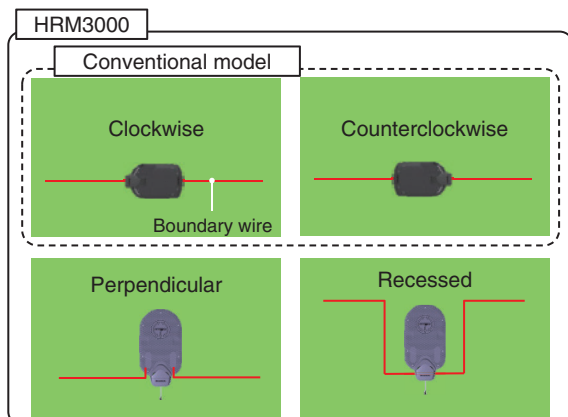


Fig. 13 Docking station placement

When the HRM3000 returns to its docking station, as shown in Step 1 of Fig. 16, it first returns to the boundary wire, then detects the station signal using the front area sensor, and then heads for the center of where the magnetic field is strongest. This allows for the docking station to guide the mower to it from any direction.

Finally, as shown in Step 2 of Fig. 16, in order for the mower to dock with the station's charging terminal, the mower's rear area sensor is used to detect the docking signal. Once the rear area sensor reaches the center of where the magnetic field is strongest, the mower returns to the charging terminal of the docking station. This station signal system has increased freedom in docking station placement.

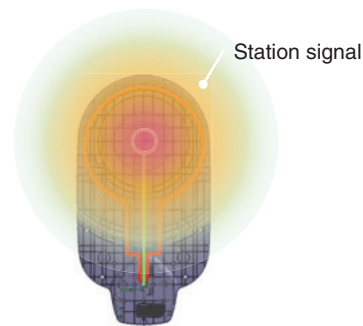


Fig. 14 Station signal magnetic field

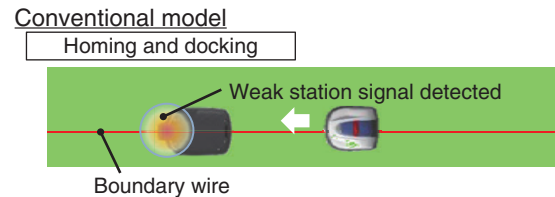


Fig. 15 Docking method of conventional models

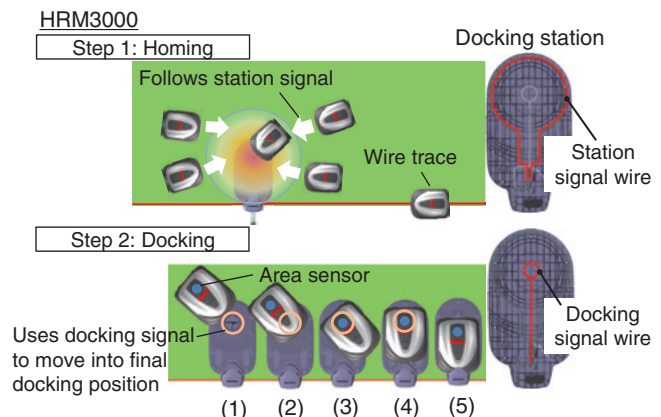


Fig. 16 Driving method when docking

4.4. Smartphone-linked Features

4.4.1. Troubleshooting

With this feature, users can check mower status and find out how to treat any troubles that may arise. As shown in the example given in Fig. 17, the screen displays operations for the user to perform and instructions when they should contact their dealer.

4.4.2. Trouble reporting

As shown in Fig. 18, with this feature, users can use their smartphone to have the mower transmit malfunction history and configuration information to the dealer product diagnostic system⁽⁵⁾ via Honda servers. With the conventional models, users had to call or otherwise tell the dealer when a mower malfunction occurred. The treatment and resolution process could be time-consuming: due to discrepancies between information from the user and reality, the dealer would sometimes have to visit the user's lawn to correctly assess the situation.

By transmitting the malfunction history and configuration information in advance to reveal the real situation, the dealer can guess the cause of and treat the malfunction before ever visiting the user's yard. The system transmits quantitative data, allowing precise information to be shared and analyzed, as well as quick response to the underlying causes.

Window shows user's next operations.

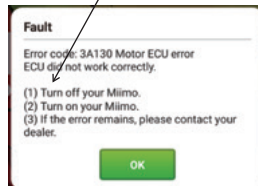


Fig. 17 Troubleshooting example

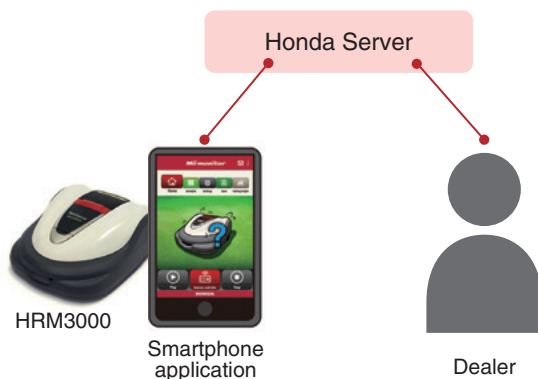


Fig. 18 Trouble reporting system

4.4.3. GUI for the smartphone application

Following with the design basic concept, the graphical user interface (GUI) was designed to be intuitive and build user loyalty.

As given in Fig. 19, current mower status display on the Home screen is animated to add movement and expression so that the user will feel a closeness with the product and intuitively understand the status. Next, as shown in Fig. 20, icons for frequently-used operations are arranged along the top portion of the Home screen, where users view frequently⁽⁶⁾, to increase convenience.



Fig. 19 Status indicators



Fig. 20 Home screen and icon design

5. Conclusion

Honda engineers have developed the following technologies to meet the demands voiced in market research of robotic lawnmower users:

- (1) Smooth turning, to allow for faster mowing
- (2) A waterproof frame, making the mower washable with water
- (3) A station signal system, for increased freedom in docking station placement

In addition, smartphone linking has been enabled to provide a product for simplified configuration and response to any troubles that arise.

References

- (1) Kawakami, T., Kobayashi, H., Yamamura, M., Maruyama, S.: Development of Robotic Lawnmower Miimo, Honda R&D Technical Review, Vol. 25, No. 2, p. 54-59
- (2) Shimamura, H., Matsui, Y., Schmitt, S.: Study of Optimal Operation Control for Robotic Lawnmower, Honda R&D Technical Review, Vol. 29, No. 2, p. 85-89
- (3) European Garden Machinery Industry Federation: EGMF ROBOTIC MOWERS PERFORMANCE EVALUATION STANDARD RLM004-1.0/2016, Clause 4.2
- (4) International Electrotechnical Commission IEC60529:2001, Clause 6
- (5) Dobashi, M., Yamamura, M., Miwa, A., Yamanaka, M.: Diagnostic System for Power Products Using Automatic Recognition Technology for Multi-protocols, Honda R&D Technical Review, Vol. 20, No. 2, p. 96-100
- (6) <https://www.nngroup.com/articles/f-shaped-pattern-reading-web-content-discovered/>, 4/17/2006

■ Author ■



Susumu OKUBO



Kenta KAWANISHI



Makoto YAMAMURA



Takuya KANISAWA



Dai HYAKUSAWA

Technology for Enhancing Thermal Efficiency of Gasoline Engine by Pre-chamber Jet Combustion

Hiroki KOBAYASHI*

Kiminori KOMURA*

Keitaro NAKANISHI*

Atsushi OHTA*

Hiroki NARUMI*

Noriyuki TAKEGATA*

ABSTRACT

A two-point increase in brake thermal efficiency was achieved over an engine with a brake thermal efficiency of 45% from a previous study by using a single-cylinder gasoline engine adopting pre-chamber jet combustion. These results were achieved with a mechanical compression ratio of 16:1 by optimizing the pre-chamber design, coating the pre-chamber inner walls with thermal insulation, and optimizing the combustion chamber shape, including compression ratio settings. The resulting configuration also achieved NOx emissions of 30 ppm and hydrocarbon emissions of 2000 ppm, which puts it in compliance with SULEV20 emission regulations. In addition, combustion stability was kept to a CPmi of 3% or lower, the rate of in-cylinder pressure increase to 300 kPa/deg or lower, and noise vibration requirements for actual vehicles were satisfied.

1. Introduction

Given the recent surge in initiatives to reduce CO₂ emissions, one of the attributed causes of climate change, the share of electric vehicles (EVs), fuel cell vehicles (FCVs), and other zero emission vehicles (ZEVs) is expected to increase. Still, ZEVs are projected to take some time to see widespread use. In the meantime, hybrid electric vehicles (HEVs) and plug-in hybrid electric vehicles (PHEVs) with internal combustion engines will likely be the mainstream for some time. In addition to power technology research on motor and battery technology, further improvement in the fuel economy of HEVs and PHEVs will require research on technology for enhancing brake thermal efficiency for internal combustion engines. In its research to date, Honda has achieved a gasoline engine brake thermal efficiency of 45%⁽¹⁾ and targets a brake thermal efficiency of 50% as the next step. Achieving a 50% brake thermal efficiency will require adopting lean burn in a high-compression engine.

Lean burning requires lean limit performance to greatly suppress knocking and to meet strict emission regulations. In its research on Homogeneous Lean-charge Spark Ignition (HLSI) combustion⁽²⁾, Honda has achieved both a high lean limit performance and low NOx emissions. However, homogeneous lean burning is known to be less effective in suppressing knocking than EGR combustion⁽³⁾. Another method, stratified combustion, is believed to suppress knocking by thinning in-cylinder end gas. With stratified

combustion, the issue to address is that NOx emissions are increased more with it than with homogeneous combustion. Stratified combustion has been researched along with Direct Injection Stratified Charge (DISC) combustion⁽⁴⁾. DISC combustion can combine high lean limit performance and low NOx emissions if adopted in conjunction with EGR. As combining with EGR will lessen the resulting increases in thermal efficiency, however, the authors saw it as necessary to use stratified combustion that realizes low NOx emissions without EGR to maximize the effect of enhancement in thermal efficiency.

Thus, in this study, pre-chamber jet combustion, a form of stratified combustion, was researched as means of achieving the performance requirements for lean burn. The engine configuration for pre-chamber jet combustion is as shown in Fig. 1. A pre-chamber is installed in the center of the combustion chamber, and that pre-chamber contains a spark plug and a dedicated direct injector (DI) to control the air-fuel ratio (AFR) at which energy from a mass-produced ignition system can achieve spark. In addition, a port injector (PI) is separately equipped to supply fuel to the main combustion chamber. Thus, stratified combustion is performed with different AFRs of air-fuel mixtures in the main chamber and pre-chamber. Furthermore, for the pre-chamber jet combustion, the ratio of the rich air-fuel mixture is determined by setting of the pre-chamber volume, allowing the dedicated DI injector to control AFR to the target values. This is what allows the configuration to

* Automobile R&D Center

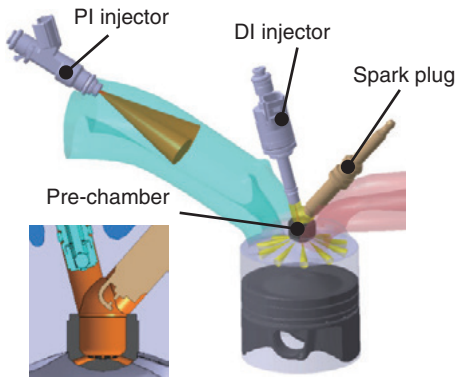


Fig. 1 Engine configuration for pre-chamber jet combustion

keep NOx emissions low with stratified combustion. In this paper, the authors will introduce the research on this pre-chamber jet combustion.

2. Setting Target Values

In setting the targets for achieving 50% brake thermal efficiency, a three-point increase will be achieved with advances in waste heat recovery, the injection system, injectors, cooling loss reductions, and other technology, and the remaining two points will be achieved with lean burn. The goal of this research is to enhance on the thermal efficiency of the previously studied engine with 45% brake thermal efficiency⁽¹⁾ (hereinafter “the previous engine”) by two or more points by incorporating lean burn. When lean burn is used, a three-way catalytic converter cannot be used to filter exhaust emissions. Thus, meeting stricter emission regulations will require a special post-treatment catalyst for lean burn and a reduction in feed emissions from the combustion chamber. To start, the feed emission reduction target values were examined. SULEV20 was set as target emission regulation, assuming that a special post-treatment catalyst for lean burn⁽⁵⁾ would be adopted to achieve this target. The authors assumed that the catalytic system will filter 99.6% of hydrocarbons (HC) and 95.9% of NOx. In the examination, mode engine rpms and load operation frequency were used as calculated from driving data in FTP-75 mode, simulating the mode through regular integration. From these calculations, as an efficient point load, the targets for feed NOx emissions at 50 ppm or less and for feed HC emissions at 2800 ppm or less were set. In addition, the values for combustion stability and rate of in-cylinder pressure rise ($dP/d\theta$) were required to satisfy noise vibration requirements for actual vehicles. Therefore, the following five performance targets for achieving maximum brake thermal efficiency, including a target for brake thermal efficiency enhancement were set. The indicator used for combustion stability is indicated mean effective pressure (CPmi).

- (1) Brake thermal efficiency target:
Increase at least two points from 45%-efficient technology
- (2) Emission targets for achieving SULEV20:
Feed NOx emissions of 50 ppm or lower
Feed HC emissions of 2800 ppm or lower
- (3) Actual vehicle noise vibration requirements:
CPmi of 3% or lower
 $dP/d\theta$ of 350 kPa/deg or less

Next, the AFR required to achieve target performance in terms of brake thermal efficiency and NOx emissions was examined. The discussion will start off with an explanation of the results from examining the AFR required for achieving the target brake thermal efficiency. Using one-dimensional simulations, the authors calculated change in brake thermal efficiency with Minimum Advanced Best Torque (MBT) driving and main burn duration, defined as 10% to 90% mass fraction burned (MFB), fixed at 25°, calculating for both EGR combustion and lean burn driving conditions. The calculations were based on performance results of the previous engine, and driving conditions were set to achieve maximum brake thermal efficiency. The calculation results are given in Fig. 2. These calculations revealed that lean limit performance would need to be an AFR of 28:1 or higher in order to increase brake thermal efficiency by two points.

Next, to clarify the AFR needed to achieve NOx emissions of 50 ppm or less, the characteristics for NOx emissions were determined at the AFR used in the previous engine. As NOx emissions correlated to maximum in-cylinder temperature, the emissions change by retard of ignition timing due to knocking. Therefore, NOx emissions were checked with driving conditions which were not affected by knocking: engine speeds of 2000 rpms and an indicated mean effective pressure (IMEP) of 470 kPa. The results, given in Fig. 3, show that in order to keep NOx emissions to 50 ppm or less, lean limit performance will need to be an AFR of 30:1 or higher. From the above two results, a lean burn capable of running at an AFR of 30:1 or higher will be necessary.

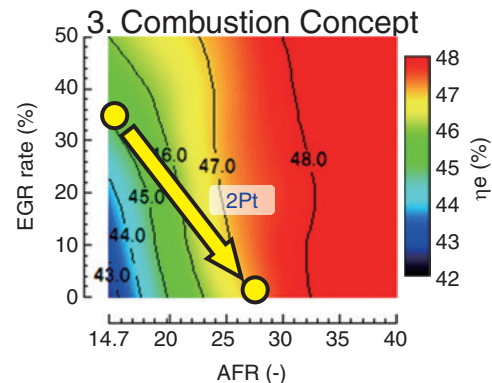


Fig. 2 Brake thermal efficiency for EGR combustion and lean combustion using 1-D simulation

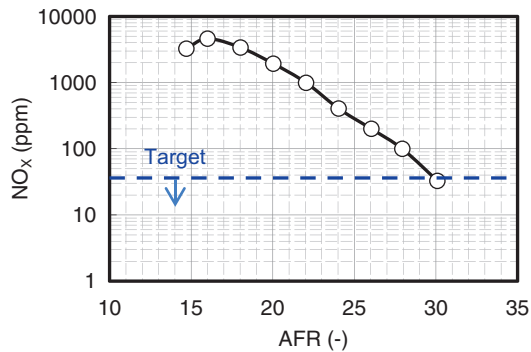


Fig. 3 Characteristics of NO_x emissions by AFR for base engine

The combustion process for pre-chamber jet combustion will now be explained through images visualizing pre-chamber jet combustion, shot from a direct bottom view, and the Computational Fluid Dynamics (CFD) calculations. The AFR is kept different between the pre-chamber and main chamber by having the PI injector supply fuel to the main chamber in the intake stroke and having the dedicated DI injector inject fuel into the pre-chamber in the compression stroke. Next, a spark is ignited by the spark plug in the pre-chamber [Fig. 4(1)], starting combustion in the pre-chamber and generating a difference in pressure between the main chamber and pre-chamber. This pressure difference is then used to force a flame to shoot out from the pre-chamber to the main chamber [Fig. 4(2)]. From this thermal energy, the air-fuel mixture in the main chamber will start combustion [Fig. 4(3)]. In order to achieve such a combustion format, the ability of pre-chamber jet combustion to maintain stable ignition performance and

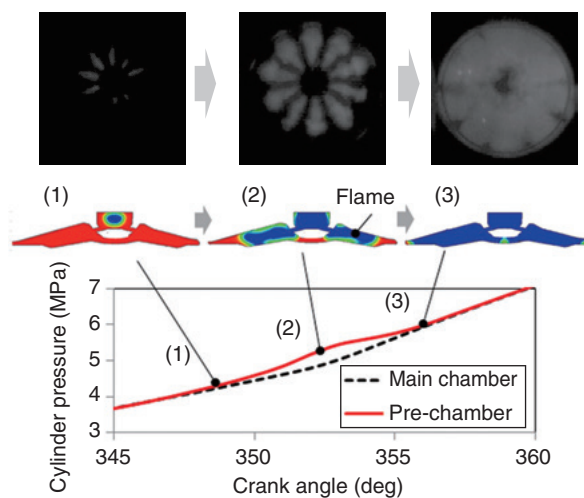


Fig. 4 Visualized direct bottom view and CFD calculations for pre-chamber jet combustion

multi-point ignition caused by the main chamber air-fuel mixture allow for rapid combustion. All this achieves a high lean limit performance and is expected to suppress knocking.

Pre-chamber jet combustion also offers advantages in terms of lowering NO_x emissions. In the past, stratified combustion secured stable ignition even with varying in-cylinder flow and DI injection cycles, necessitating an AFR and mixture distribution rich enough for fuel to ignite near the plug. In addition, high AFR mixture content was often found near the plug. These increased NO_x emissions have been an issue. In pre-chamber jet combustion, fuel is injected into the pre-chamber in the compression stroke, containing fuel within the pre-chamber without discharging it to the main chamber. Therefore, the pre-chamber volume can be configured to adjust the area where the AFR is richer than the overall in-chamber AFR. As another advantage, the air-fuel mixture in the pre-chamber can be controlled to the target AFR, simultaneously stabilizing the rich air-fuel mixture in the plug vicinity each cycle. The above characteristic allows this form of combustion to reduce NO_x emissions with stratified combustion.

4. Engine Design

The following is an explanation of the impacts of each design parameter of the pre-chamber jet combustion engine on achieving the target performance.

4.1. Evaluation Engine Specifications

The specifications in Table 1 are for the test engine used in this study. This was a single-cylinder engine bored out to a diameter of 73 mm while keeping the stroke/bore ratio and compression ratio, two important elements in the previous study, the same as in the previous engine. An Atkinson cycle, which has also been an important element, is again adopted in this study, holding the intake valve open longer. Compression ratio for the compression stroke was set to the effective compression ratio as calculated from the cylinder volume for compression at top dead center (TDC) and that when the intake valve closes. Compression ratio for the expansion stroke is shown as the expansion ratio (mechanical compression ratio).

4.2. Pre-chamber Design

The authors contemplated the pre-chamber design, which determines the thermal energy supplied to the main combustion chamber from the pre-chamber and in turn greatly impacts combustion in the main chamber in a pre-chamber jet combustion engine. As shown in Fig. 5, the main design parameters for the pre-chamber are the pre-chamber volume, number of nozzles, nozzle diameter, and the nozzle cross-sectional area as calculated from the number of nozzles and nozzle diameter. In order to arrive at the optimum design, the impacts on each performance by these parameters were confirmed. As will be explained

Table 1 Test engine specifications for pre-chamber jet combustion

Number of cylinders	1
Displacement volume (cm ³)	458
Bore (mm)	φ 73
Stroke (mm)	109.5
Stroke/bore ratio	1.5
Expansion ratio	17.0
Effective compression ratio	12.5
Aspiration	Supercharger
Fuel supply (main/pre)	(PI/DI)
Ignition energy (mJ)	60

in detail below, the pre-chamber volume and nozzle cross-sectional area largely contributed to achieve the target values.

Figure 6 is a graph showing the impacts on each performance if pre-chamber volume is changed at an engine speed of 2000 rpms, IMEP of 810 kPa, and AFR of 35:1, which are the conditions for achieving maximum brake thermal efficiency with previous engine. As the pre-chamber volume is reduced, NO_x emissions and dP/dθ both decrease, whereas the 10-90% MFB angle tends to increase. There was no decline in thermal efficiency down to a pre-chamber volume of 2.0 cm³, beyond which smaller volumes resulted in reduced thermal efficiency. Thus, a smaller pre-chamber volume results in a decrease in rich air-fuel mixture ratio, consequentially reducing NO_x emissions. Smaller volumes also reduce the amount of gas in the pre-chamber, resulting in less thermal energy supplied from pre-chamber to main chamber and in turn both reducing the dP/dθ and extending burn duration. If thermal energy supplied from pre-chamber to main chamber cannot maintain combustion stability at an AFR of 35:1, HC increases and thermal efficiency will presumably decline greatly. From the above performance results, pre-chamber volume was set to 2.0 cm³, which minimizes NO_x emissions and dP/dθ without reducing thermal efficiency.

Figure 7 is a graph showing the impacts on each performance if the pre-chamber nozzle cross-sectional area is changed at an engine speed of 2000 rpms, IMEP

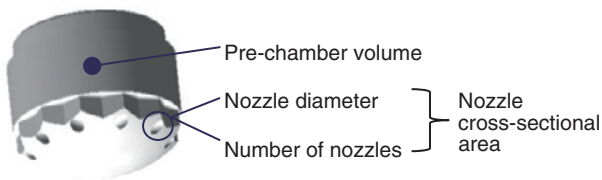
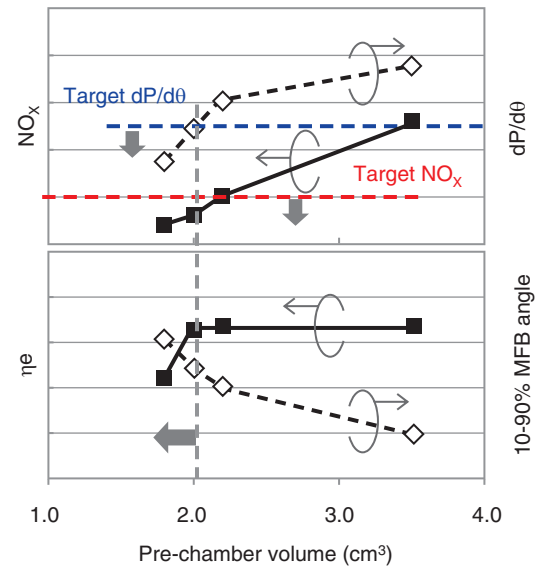
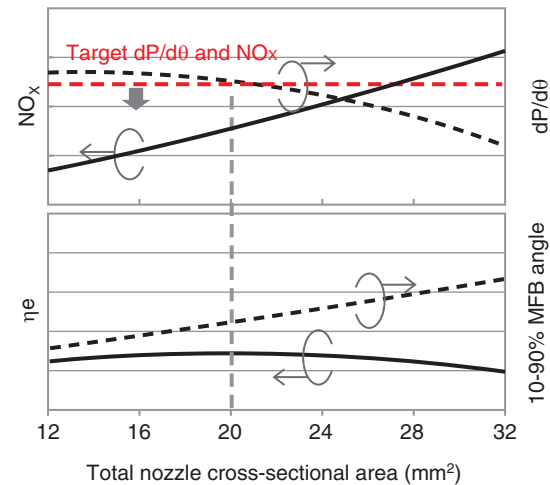


Fig. 5 Pre-chamber design parameters

of 810 kPa, and AFR of 35:1. As the pre-chamber nozzle cross-sectional area is reduced, NO_x emissions and the 10-90% MFB angle will also reduce, but dP/dθ will be increased. A pre-chamber nozzle cross-sectional area of 20 mm² maximizes thermal efficiency; thermal efficiency is reduced both above and below this value. The reason for this is that, as the pre-chamber nozzle cross-sectional area is reduced, pre-chamber pressure rises, in turn increasing the thermal energy supplied to the main chamber and keeping combustion stable. Thus, ignition timing can be retarded while keeping the same combustion stability for lower NO_x emissions by lowering the maximum in-chamber temperatures. As for the change in the thermal efficiency, if the pre-chamber nozzle cross-sectional

Fig. 6 Comparison of pre-chamber volume on NO_x and thermal efficiencyFig. 7 Comparison of NO_x and thermal efficiency by total nozzle cross-sectional area

area is too large, pre-chamber pressure rises less, in turn decreasing the thermal energy supplied from pre-chamber to main chamber. Therefore, at an AFR of 35:1, chamber combustion stability cannot be maintained and thermal efficiency drops due to an increase in HC. Conversely, if the pre-chamber nozzle cross-sectional area is too small, and the thermal energy supplied from pre-chamber to main chamber increases the pre-chamber pressure rises beyond that required to provide combustion stability at an AFR of 35:1. Presumably, this might greatly increase cooling loss in the pre-chamber and decrease thermal efficiency. In contrast, if thermal energy supplied from pre-chamber to main chamber increases, $dP/d\theta$ will rise due to the rapid combustion. From the above performance results, pre-chamber nozzle cross-sectional area was set to 20 mm^2 , which keeps NOx emissions and $dP/d\theta$ at or below the target values and maximizes thermal efficiency. Specifically, the pre-chamber was designed with 10 nozzles at a nozzle diameter of $\phi 1.6 \text{ mm}$.

Each performance for the pre-chamber after design optimizations is shown in Fig. 8. This figure relates the performance results and AFR under heavy load at an engine speed of 2000 rpm and IMEP of 810 kPa. Just as in the previous study, in their brake thermal efficiency calculations, the authors used supercharger loss, estimated using mechanical friction and one-dimensional simulations equivalent to an inline four-cylinder engine. In terms of combustion stability, the pre-chamber achieves a CPmi of 3% or lower up to an AFR of 38:1 and operates at MBT at

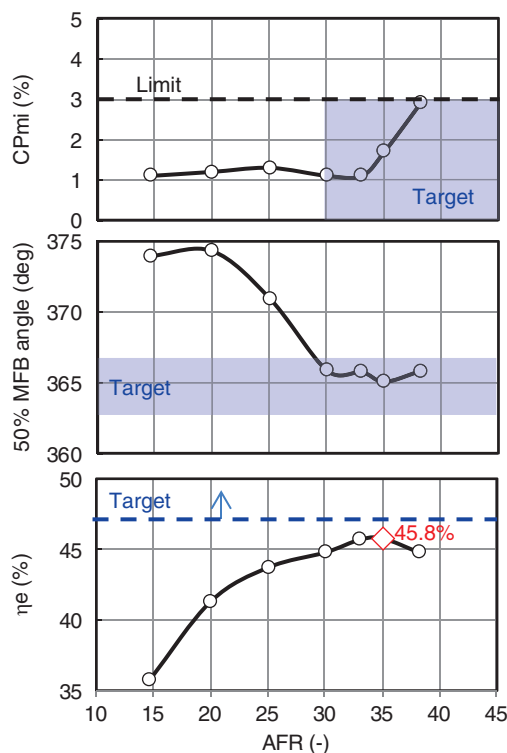


Fig. 8 Lean characteristics under heavy load

AFRs of 30:1 or higher.

4.3. Technology for Reducing Pre-chamber Cooling Loss

Given the greater specific surface area (surface-to-volume ratio) for pre-chamber combustion, cooling loss to the wall surface could be higher. Thus, to enhance thermal efficiency, the authors worked to reduce cooling loss in the pre-chamber. To this end, the area of the pre-chamber inner wall was coated with thermal insulation as shown in red in Fig. 9. This area covers 37% of the surface area of the pre-chamber. Figure 10 shows the effect of coating at an engine speed of 2000 rpm, IMEP of 810 kPa, and AFR of 35:1. Thus, the reduction in cooling loss and accompanying increase in thermal energy to the main chamber have reduced unburned loss, increasing thermal efficiency by 0.5 points. This did not change time loss as knocking was not affected; however, waste heat also increased as not all of the above loss reduction translates to thermal efficiency.

4.4. Combustion Chamber Surface-to-volume Ratio

In the pre-chamber jet combustion engine, the combustion chamber surface-to-volume ratio is increased over that of standard gasoline engines at the same compression ratio. Also, at the same pre-chamber volume, this effect increases with the compression ratio. Thus, the relationship between compression ratio and the combustion chamber surface-to-volume ratio was investigated. The thermal efficiency sensitivity map shown in Fig. 11

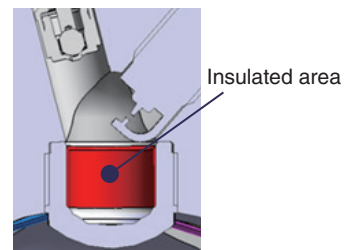


Fig. 9 Insulated area of pre-chamber

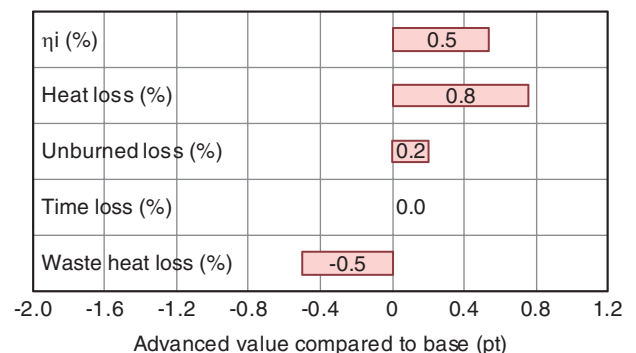


Fig. 10 Effects of coating on pre-chamber

gives the results from comparing compression ratio and surface-to-volume ratio. As step 1 in the figure, to confirm the impact of the combustion chamber's surface-to-volume ratio, the authors confirmed the performance of the combustion chamber's surface-to-volume ratio when lowering the compression ratio. It was found that, while reducing compression ratio did reduce the theoretical thermal efficiency, it also greatly reduced cooling loss and unburned loss, conversely resulting in a net enhancement in thermal efficiency. Next, as step 2 in the figure, the combustion chamber shape to further reduce its surface-to-volume ratio and enhance thermal efficiency was optimized. From the above, including the revised combustion chamber shape, the compression ratio for the pre-chamber jet combustion engine was set to 16:1.

4.5. In-cylinder Flow

In the previous study, the intake ports for the 45% brake thermal efficiency engine adopted high tumble in order to increase flame velocity. In pre-chamber jet combustion, however, the authors expected the in-cylinder flow demands to change due to the change in combustion format. Thus, an in-cylinder flow for optimum pre-chamber jet combustion was selected. Two port specifications with different tumble ratios were compared for performance as shown in Fig. 12. Port A had a tumble ratio of 0.4 while Port B had a tumble ratio of 1.8. Engine speed is 2000 rpm and IMEP is 510 kPa for these comparison results. Comparing port performance proved that differences in tumble ratio had no effect on burn duration, but the higher tumble ratio in the Port B specification reduced combustion stability. As to why burn duration remained unchanged, given that the spark plug in pre-chamber jet combustion resides in the pre-chamber, it is possible that enhanced flow does not promote initial flame kernel growth as it normally does in an SI engine. Next, in-

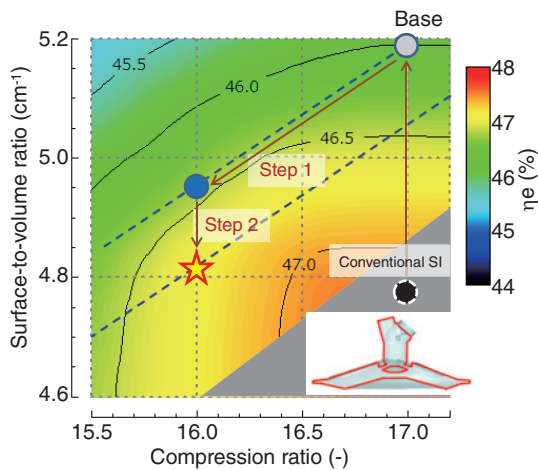


Fig. 11 Effects of surface-to-volume ratio and compression ratio on brake thermal efficiency in pre-chamber jet combustion engine

cylinder flow was analyzed using CFD to analyze the drop in combustion stability, as shown in Fig. 13. This figure shows the distribution of in-cylinder flow at a crankshaft angle of 80 deg before top dead center in the compression stroke after fuel is injected into the pre-chamber. Whereas almost no gas is exchanged between the pre-chamber and main chamber in Port A, gas is clearly exchanged between pre-chamber and main chamber in Port B due to the residual tumble. In the compression stroke, fuel is injected by the dedicated DI injector in the pre-chamber to keep it in the pre-chamber. This fuel is forced into the main chamber by residual tumble, resulting in a different amount of fuel in the pre-chamber when combustion starts in each cycle. Thus, presumably due to this variation in the thermal energy supplied from pre-chamber to main chamber from cycle to cycle, combustion in the main chamber also varies for each cycle, reducing combustion stability. From the above, Port A with the lower tumble ratio was selected for the pre-chamber jet combustion in-cylinder flow specifications.

5. Verification Results

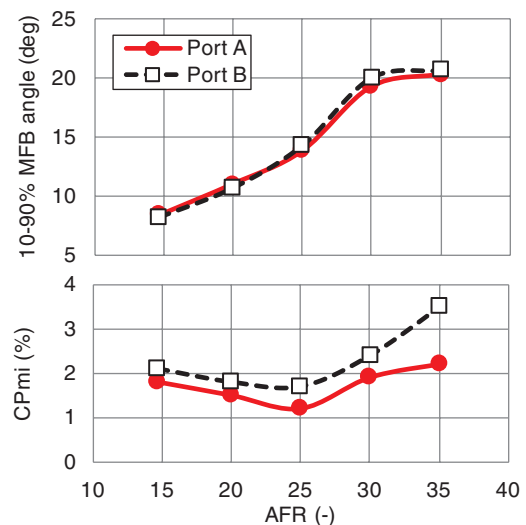


Fig. 12 Performance results for intake ports with different tumble ratios

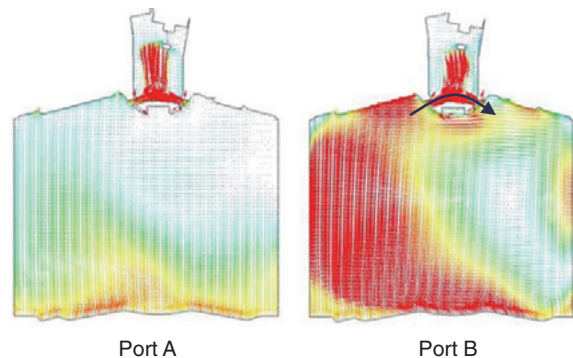


Fig. 13 In-cylinder flow with CFD at crank angle of 80° BTDC

Table 2 Final test engine specifications

Number of cylinders	1
Engine type	DOHC 4-valve
Displacement volume (cm ³)	458
Bore (mm)	φ 73
Stroke (mm)	109.5
Stroke/bore ratio	1.5
Expansion ratio	16.0
Effective compression ratio	12.5
Pre-chamber volume (cm ³)	2.0
Nozzle diameter (mm)	φ 1.6
Number of nozzles	10
Pre-chamber specification	Thermal insulation coating
Intake port	Weak flow
Aspiration	Supercharger
Fuel supply (main/pre)	PI/DI
Ignition energy (mJ)	60

The final values achieved with a single-cylinder engine applying all the supporting technology for a pre-chamber jet combustion engine described above were confirmed. The engine specifications in Table 2 are for the final specifications. Just as in the previous study, the authors calculated brake thermal efficiency using supercharger loss, as estimated using mechanical friction and one-dimensional simulations equivalent to an inline four-cylinder engine.

Figure 14 relates each performance and AFR under heavy load at an engine speed of 2000 rpms and IMEP of 870 kPa. In terms of combustion stability, the pre-chamber achieves a CPmi of 3% or lower up to an AFR of 40:1 and operates at MBT at AFRs of 35:1 or higher, achieving a two-point increase over 45% brake thermal efficiency. At the same AFR of 35:1 which increases brake thermal efficiency, the other performance targets were also achieved: NO_x emissions were 30 ppm, HC emissions were 2000 ppm, CPmi was 1.2%, and dP/dθ was 300 kPa/deg.

A comparison of brake thermal efficiency at an engine speed of 2000 rpms under each load between the pre-chamber jet combustion engine and the 45% brake thermal efficiency engine for the previous study is shown in Fig. 15. It was confirmed that brake thermal efficiency also increased outside maximum load conditions.

The details in this study were verified on a single cylinder engine. Hereafter, toward practical applications in a multi-cylinder engine for automobiles applying these achievements, two outstanding topics for future study are development of turbocharger system compatible with both

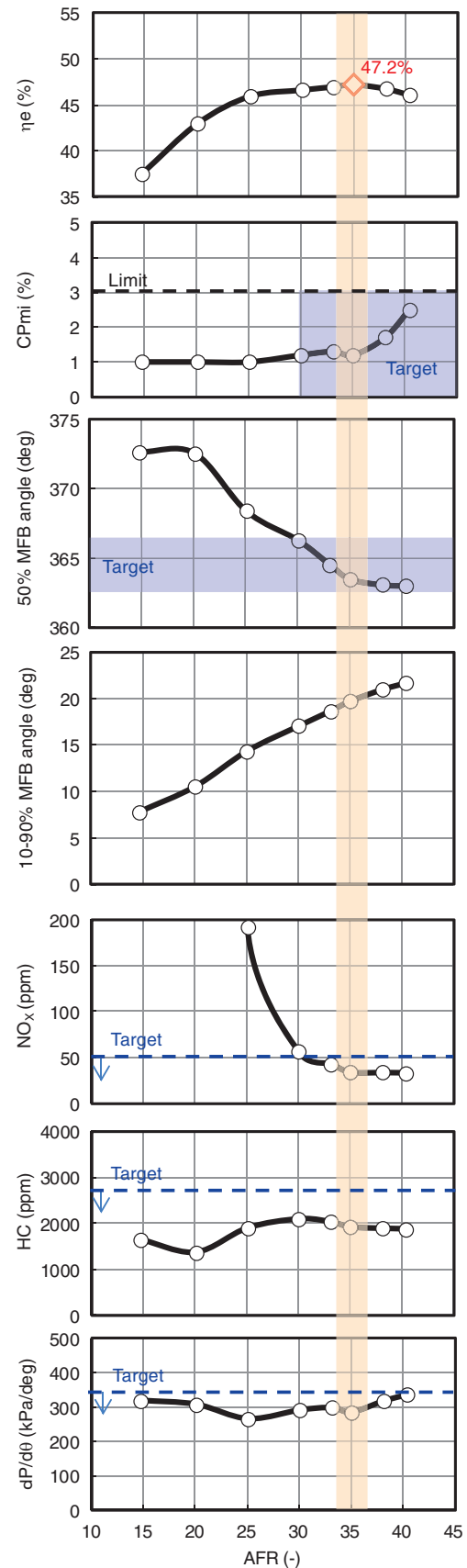


Fig. 14 Lean characteristics at high load with final test engine specifications

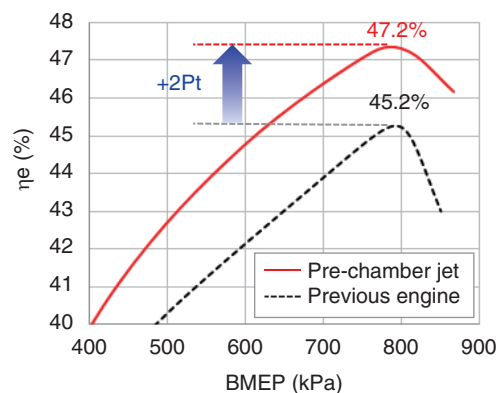


Fig. 15 Achieved brake thermal efficiency at engine speed of 2000 rpm

partial load and output point operation and development of a lean burn-specific catalyst.

6. Conclusion

Pre-chamber jet combustion was applied to a single-cylinder gasoline engine with mechanical compression ratio of 16:1, optimizing the pre-chamber design and reducing the losses caused by the addition of pre-chambers. These developments achieved the following performance at an engine speed of 2000 rpms, IMEP of 870 kPa, and AFR of 35:1.

- (1) Brake thermal efficiency target:
Increased two points from 45%-efficient technology
- (2) Emission targets for achieving SULEV20:
Feed NO_x emissions of 30 ppm
Feed HC emissions of 2000 ppm
- (3) Actual vehicle noise vibration requirements:
CPmi of 3% or lower up to AFR of 40:1
dP/dθ of 300 kPa/deg

References

- (1) Tagishi, R., Ikeya, K., Takazawa, M., Yamada, T.: Technology for Realization of Gasoline Engine Brake Thermal Efficiency of 45%, Honda R&D Technical Review, Vol. 27, No. 2, p. 1-11
- (2) Hanabusa, H., Kondo, T., Hashimoto, K., Sono, H., Furutani, M.: Study on Homogeneous Lean Charge Spark Ignition Combustion, SAE Technical Paper, doi:10.4271/2013-01-2562, (2013)
- (3) Nakata, K., Nogawa, S., Takahashi, D., Yoshihara, Y., Kumagai, A., Suzuki, T.: Engine Technologies for Achieving 45% Thermal Efficiency of S.I. Engine, SAE Technical Paper, doi:10.4271/2015-01-1896, (2015)
- (4) Kohda, Y., Suganami, T., Kobayashi, H., Ogawa, K.: Development of High-performance and Low-emission Direct Injection Gasoline Engine, Honda R&D

Technical Review, Vol. 16, No. 1, p.39-46

- (5) Takeori, H., Wada, K., Matsuo, Y., Morita, T., Konomoto, T., Murata, Y., Kimura, M., Miyauchi, A.: Study of an Aftertreatment System for HLSI Lean-burn Engine, SAE Technical Paper, doi:10.4271/2018-01-0945, (2018)

Author



Hiroki KOBAYASHI



Kiminori KOMURA



Keitaro NAKANISHI



Atsushi OHTA



Hiroki NARUMI



Noriyuki TAKEGATA

Development of Evaluation Method for Waviness Property on Hypoid Gear Tooth Surface

Takeshi HORI* Naoto SHUKUSHIMA*

ABSTRACT

Measuring and analysis methods have been discovered for quantitative evaluation of tooth surface waviness, which cause high-order sound radiation from hypoid gears. Tooth surface waviness was measured by an optical, non-contact 3D measuring device. Taking this measurement result, images were filtered by applying a 2D Fourier Transform and inverse Fourier Transform to extract the cyclical waviness components impacting gear noise frequencies. The method was used to evaluate tooth waviness of hypoid gear in an actual gearbox unit. The cyclical peak order of the waviness matched the peak order of frequencies of sound radiation from gears. In addition, upon evaluating multiple specimen gears, a quantitative correlation was found between the amplitude of peak order waviness and the pressure level of sound radiation from gears. This has newly made quantitative evaluation of hypoid gears in isolation possible.

1. Introduction

With the spreading popularity of all-wheel drive (AWD) multi-purpose passenger vehicles, typified by the SUV, Honda has developed and commercialized numerous hypoid gears for drivetrains. A hypoid gear is a type of bevel gear that turns the rotational axis at a right angle on the transfer path of rotational power from the engine or motor to the driving wheels. As illustrated with the AWD drivetrain of the Honda vehicle shown in Fig. 1, a common structure is to place gear sets at both the front and rear ends of the propeller shaft. The front end gear set is built into the power transfer unit attached to the transmission, and the rear end gear set is built into a rear drive unit mounted above the rear axle, with each transmitting driving torque.

With the increase in cabin quietness for passenger vehicles in recent years, the quality and performance demands for drivetrain components are higher than ever in terms of noise, vibrations, and harshness, collectively termed NVH. Hypoid gears have posed particular challenges in the development stage for cabin noise due to the high-frequency NVH they emit from meshing. These high-frequency meshing sounds are raspy in the normal driving range on actual vehicles. NVH generated by toothed gears transmitting power is synchronized with the mesh cycle frequency of meshed tooth surfaces of a gear pair -- namely, the driver and driven gears. With the order number being a constant multiple of the mesh cycle frequency, the frequency of high-order gear noises is known

to take on the waviness of tooth surface and be affected by accuracy control in the tooth surface manufacturing process⁽¹⁾. However, there were no conventional indicators or methods capable of quantitatively evaluating the tooth surface waviness causing this high-order gear noise. Thus, with the gears being challenging to evaluate in isolation, these waviness had been evaluated in the mass production manufacturing process using a quality pass/fail judgment based on meshing test results of completed gear sets, followed by production control. This led to increased man-hours for controls and decreased production efficiency. Additionally, much time and effort was spent on gear meshing tests for different manufacturing conditions and for parameter studies on actual unit noise tests to feed preventive measures for gear noise and waviness back into the manufacturing control criteria for individual gears.

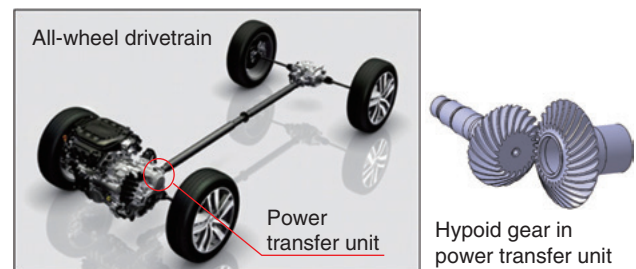


Fig. 1 Drivetrain and hypoid gear of Honda AWD vehicle

* Automobile R&D Center

2. Generating Mechanism for High-order Gear Noise

2.1. Relationship between Hypoid Gear Tooth Surface Waviness and High-order Gear Noise

Comparisons in cabin noise for development vehicles were made between vehicles with and without noticeable high-frequency noise near the 3000 Hz to 4000 Hz range. A hypoid gear set as built into the power transfer units of vehicles producing high-frequency noise was loaded into a meshing test machine assembled the same as in the actual unit and tested under equivalent loads to those in the actual units. This test confirmed a transmission error peak in the meshing order corresponding to the frequency of the high-frequency noises being generated in vehicle. Cabin noise data for the vehicles generating high-frequency noise is shown in Fig. 2. Here, the x-axis represents the meshing order number for the frequency of one meshing cycle of hypoid gear, and the y-axis represents vehicle driving speed. The color tones in the contour diagram represent the sound pressure levels. For this paper, one meshing cycle is defined as the time from the moment that two tooth surfaces on a hypoid gear mesh to the moment when they move to their adjacent tooth surfaces and repeat the same movement again. This cycle is the same for both driving and driven gears and is equal to the value of the time required for the gears to make one rotation, divided by the number of teeth. While cabin noise is being measured, the vehicle is test driven according to set mode conditions and at a constant rate of acceleration. Sound pressure data was recorded for all frequency ranges at each measured driving speed, and the sound pressure levels used to analyze the meshing order number for one meshing cycle of hypoid gear were arranged along the x-axis. Next, the data was continuously recorded while moving up the graph y-axis with the vehicle driving speed. From the graph, it was confirmed that the hypoid gear was generating a constant 7.5-order high-frequency noise for each meshing cycle at nearly all speed ranges during measurement. The gear unit meshing transmission error data is given in Fig. 3. For the meshing tester, a Gleason Works 600HTT was used in the measurements. The transmission error data was collected during gear

rotation for a fixed time and then analyzed for frequency using a Fast Fourier Transform (FFT). The meshing order numbers for one meshing cycle of hypoid gear were then graphed on the x-axis with transmission error amplitude on the y-axis. In this graph, 7.5-order transmission error peak that was the same as the high-frequency noise observed in the cabin noise data was confirmed.

This meshing transmission error, a value measured in rads or mm, represents the rotational motion for one meshing cycle as the two gears rotate. It is also equivalent to the position relative to the normal direction as determined from the geometric shape (profile) of both teeth surfaces. This transmission error is known to cyclically generate an angular acceleration on the driven gear that contributes directly to vibratory forces causing NVH^{(2), (3)}. From these results, the high-frequency noises causing issues in prototypes were identified as 7.5-order gear noises radiating from the hypoid gear tooth surface. Additionally, a stripe pattern facing the same direction at regular intervals was observed on the hypoid gear meshing teeth surfaces radiating a gear noise. Tooth surfaces from driving side pinion are illustrated in Fig. 4. Next is Fig. 5, which is a sketch of the meshing points on the tooth surface acting as the load center for power transmission. The tooth surface has a mesh trajectory (denoted as “path of contact” in Fig. 5) connecting the contact points between mesh starting point A1 and mesh ending point A2. Power is transmitted by these contact points moving at a constant speed along the path of contact. There are also two switching points on this line: B1 for the moment when the tooth switches from meshing with

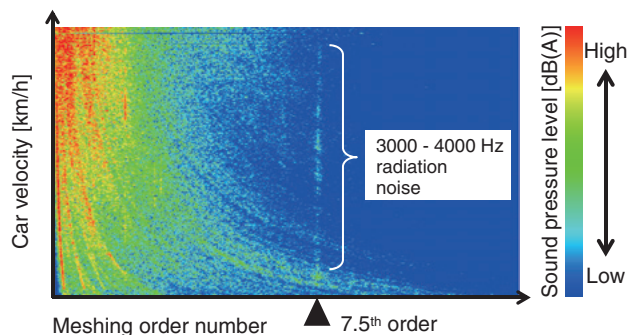


Fig. 2 Sound radiation level inside actual vehicle

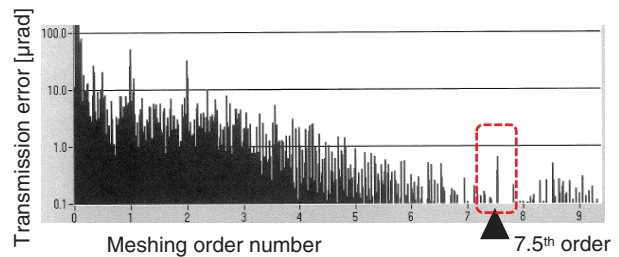


Fig. 3 Transmission error spectrum of hypoid gear

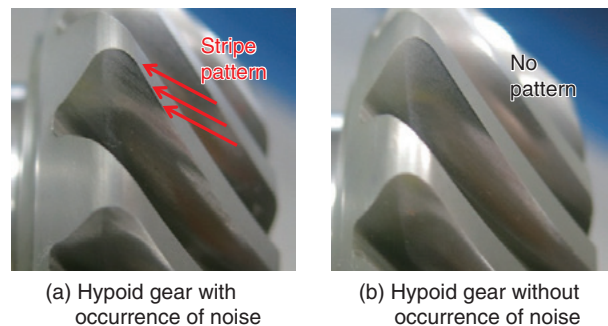


Fig. 4 Stripe pattern on tooth surface of hypoid gear

the previous tooth, and B2 for the moment when the tooth switches to meshing with the next tooth. As the gear rotates for an angle of 2π divided by the number of teeth, the mesh point will move from B1 to B2. Taking this motion as one unit, power is transmitted as this motion is repeated. From these points, it was surmised that the striped pattern found on the tooth surface in Fig. 4 crosses the path of contact at regular intervals. Next is Fig. 6, a motion graph of the transmission error properties over one meshing cycle. Here, the x-axis represents the rotational angle of the driving gear and the y-axis represents the delay amount (transmission error) as the amplitude of rotational fluctuation in the driven gear. Where the transmission error is 0 on the x-axis, the tooth surfaces of the driving and driven gears are contacting at a constant speed and there is no delay in rotation for the driven gear. The intersections on the transmission error curve match up with mesh switching points B1 and B2. This interval is equal to the angle of rotation for one meshing cycle. At points B1 and B2, a first-order cycle rotational delay is generated according to how crowned the tooth profile is at the point. The delays generated are equal to the position relative to the normal direction of the tooth surface. Thus, hypothetically, if there were a constantly cyclical waviness along the path of contact between B1 and B2, a transmission error and rotational fluctuation will be generated over one meshing cycle while rotating with a cycle order multiplied by the value of the distance moved between B1 and B2 divided by the waviness interval pitch length. Existing research has clearly shown that these tooth

surface waviness do impact specific-order transmission error^{(1),(4)}.

2.2. Purpose of Quantitative Evaluation Method for Tooth Surface Waviness

Existing research has shown a method of measuring and evaluating tooth surface waviness two-dimensionally on a measurement line of tooth profile and tooth lead shape by using a gear measuring machine or other profile measuring machine⁽¹⁾. Applying this existing method, the authors used a Klingelnberg AG gear profile measuring machine and dedicated software to measure tooth profiles on the path of contact shown in Fig. 5, then identified the resulting section curve in the normal direction. They also ran an FFT to analyze the cyclicity of the waviness on the curve. For the test gear used for measurement, the cutter alignment was adjusted during gear cutting to intentionally produce tooth surface waviness. The measurement results are given in Fig. 7. The section curve data on the path of contact of the tooth surface is shown in Fig. 7(a). Here, the x-axis represents the position in the path of contact length direction, and the y-axis represents the shape in the tooth surface normal direction. Meanwhile, the frequency analysis results for the data from Fig. 7(a) are given in Fig. 7(b), with the x-axis representing the cycle order number for one meshing cycle and the y-axis representing the amplitude in the tooth surface normal direction. In the graph, a number of cycles at different orders are present in addition to those near the 7.5-order, corresponding to the high-frequency noise generated in the vehicle. From this, it is no simple task to quantitatively evaluate the cycle order and amplitude values, which are the peaks as obtained in the measurements, as the property values of waviness.

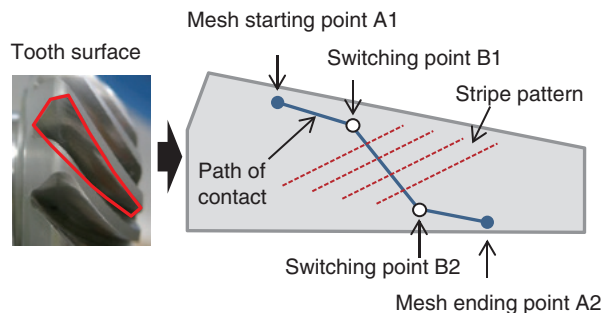


Fig. 5 Sketch of tooth surface and path of contact

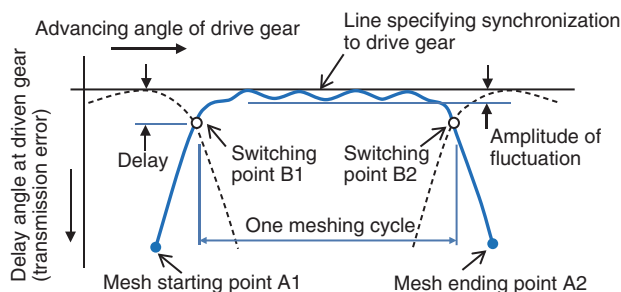


Fig. 6 Motion graph during one meshing cycle

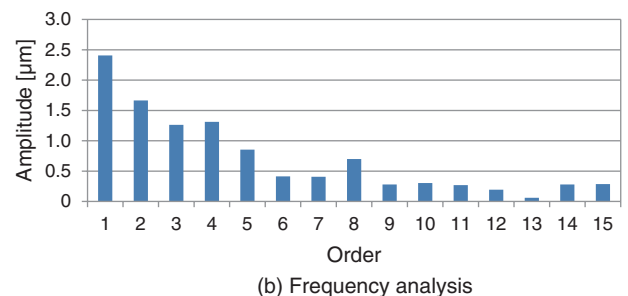
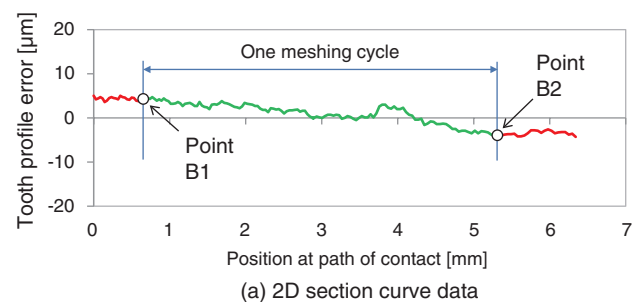


Fig. 7 Measurement results

Passing through gear cutting, heat treatment, and other steps in the manufacturing process, hypoid gears have irregularities of differing wavelengths in various directions on their tooth surfaces. The main properties, shown in the tooth surface sketch in Fig. 8, are: (1) tooth profile and crowning, (2) waviness, and (3) machining lines. Machining line is a collective term, including milling patterns generated at the cutting point and lines transferred to the tooth surface from surface properties in the cutter blade. With the above method, with all property data on the path of contact being captured in the measurement stage, the comparatively small waviness gets drowned out by other roughness levels and cannot be manifested. As this makes it challenging to evaluate waviness with the tooth surface alone using the existing method, waviness was evaluated with transmission error as measured in meshing tests of a completed gear set⁽¹⁾. For this study, a new method was created to take tooth surface shape data with various properties of differing directions and cycles, and extract from it only those minute, specific waviness components radiating gear noise in order to quantitatively evaluate them. Therefore, in place of the traditional 2D measurements on the path of contact, the tooth surface was measured extensively in 3D. From the 3D data taken for the tooth surface, an image filtering technique was used to extract waviness and analyze their direction.

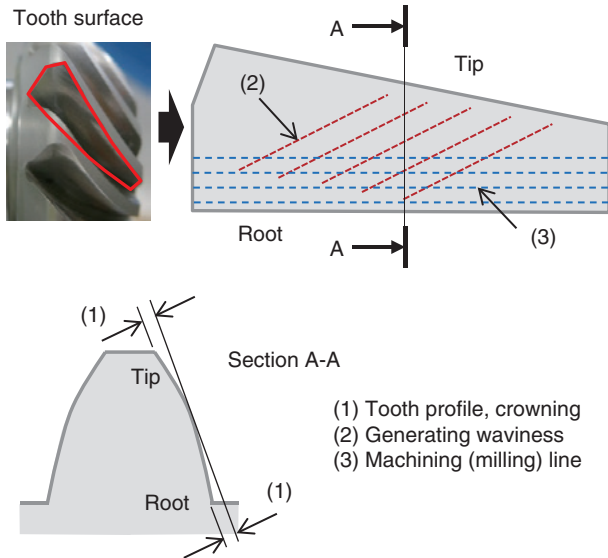


Fig. 8 Sketch of tooth surface and property

3. Creation of Methods to Measure and Evaluate Tooth Surface Waviness

3.1. Three-dimensional Measurement of Tooth Surface Shape

An Alicona optical non-contact 3D measurement device, capable of taking accurate 3D measurements of the tooth surface shape, was used for measurements. This

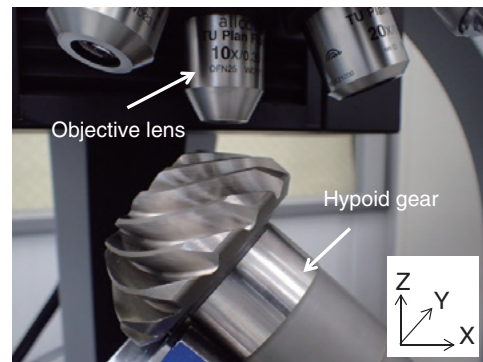
measurement device uses a moving focus measuring system. An image of measurement and basic specifications for the device are shown in Fig. 9. The imaging sensor captures images of the tooth surface via an objective lens that moves along the z-axis of the measurement device. While in motion, the objective lens shoots images at each height position in continuous shooting mode, then creates 3D property data by combining the pixel and height information for the parts in focus in the data for the multiple images taken. This method permits measurement of places with large height differences in a short period without damaging the measured part. Additionally, moving the measurement table in the X- and Y-directions and splicing multiple 3D images will produce extensive tooth surface data.

3.2. Tooth Profile Removal Using Polynomial Approximation

The captured image data contains long wavelength components due to the tooth profile, including crowning and bias. If measurement is greatly affected by these tooth profile components, it can be challenging to manifest the waviness component. To remove these shape deviations, polynomial approximation using the least squares method was applied. Defining the surface coordinates $Z = f(x, y)$, the n th polynomial is expressed by Eq. (1). Polynomial indices i and j were calculated using the least squares method.

$$Z_n(x, y) = \sum_{j=0}^n \sum_{i=0}^n a_{ij} x^{ij} \quad (1)$$

For this measurement, an n of 3 was used in calculations. Removing the shape components resulted in flat tooth surface data unimpacted by crowning (Fig. 10).



Horizontal resolution	1.76 μm
Vertical resolution	0.1 μm
Viewing range (X)	10 mm
Viewing range (Y)	5 mm

Fig. 9 Measuring equipment and basic specification

3.3. Detection of Waviness Components with Two-dimensional Fast Fourier Transform

With shape deviations removed, the 3D image data had evident marks thought to be waviness produced by cutting blade runout during cutting. Additionally, short wavelength marks running in a different direction were covering the waviness marks. To detect the frequency components of these two marks, a two-dimensional Fast Fourier Transform (2D FFT) was used. An overview of the 2D FFT is shown in Fig. 11. 2D images have two frequency components: horizontal and vertical. Thus, one-dimensional frequency analyses were run for each of the vertical and horizontal directions.

How to read the graph of the 2D FFT analysis results will now be explained while referring to the rough sketch in Fig. 12. The vertical and horizontal axes in the graph represent the vertical and horizontal directions, respectively, in the image data. Note that the top of the image data is

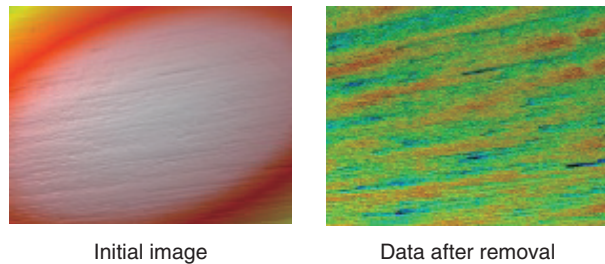


Fig. 10 Removal with polynomial approximation

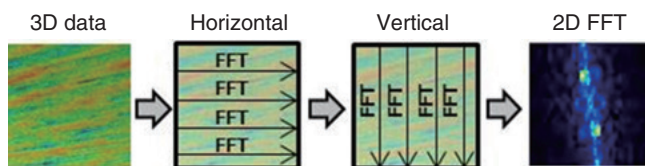


Fig. 11 Calculation procedure of 2D FFT

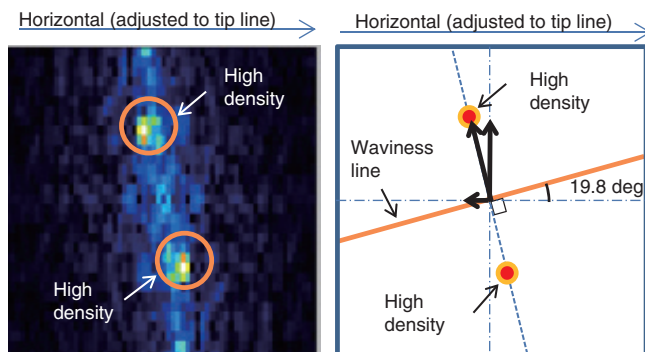


Fig. 12 Calculation result of 2D FFT

aligned with the face line for measurement. Distance from the center point represents the frequency of waviness; the further from the center, the higher the frequency component. Now, if the image data has an isocontour connecting the cyclical waviness peaks in a uniform direction, the isocontour will appear as a bright point in a position from the center corresponding to its wavelength, as in the graph on the right. Brightness here represents amplitude. The straight line vector connecting the center point and the bright point runs perpendicular to the waviness peak isocontour. In this data, the waviness is at a 19.8-degree slant with the face line.

3.4. Extraction of Three-dimensional Data with Inverse FFT

After first extracting to only isolate the waviness component of the bright point from the software-analyzed 2D FFT data, an inverse FFT is run using the same software to generate 3D data for the isolated waviness component. The 3D waviness component data, as extracted from the 3D tooth surface data with various properties of differing direction and cycles by the above filtering process, is shown in Fig. 13. The data captures an extensive area around the tooth surface waviness, including the mesh trajectory ("path of contact" in Fig. 13) corresponding to one meshing cycle. The top of the data is aligned with the face line ("tip" in Fig. 13), and the left and right sides give the measurement range in the perpendicular direction. Finally, the color tones in the contour diagram depict the waviness height in the tooth surface normal direction. This 3D data clearly captures waviness at the same angle and arranged at regular intervals across the path of contact, with approximately seven waviness in one meshing cycle. This method allows for extraction to isolate only the waviness components generating gear noise, achieving the objective of quantitatively evaluating waviness.

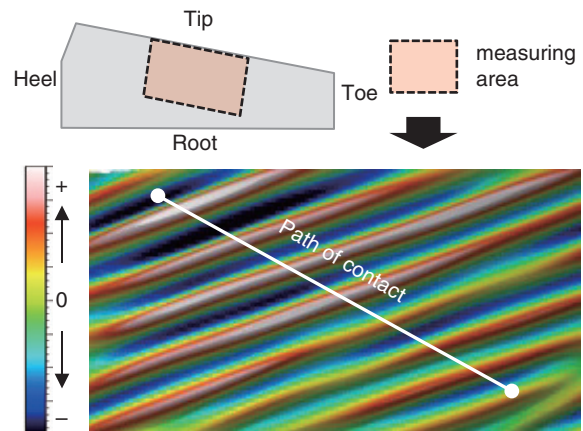


Fig. 13 3D data of tooth surface waviness

4. Quantitative Evaluation Results for Waviness and Actual Unit Validation

This method was used to measure and evaluate tooth surface waviness in an actual hypoid gear, and both the method and the gear noise generating mechanism were validated. A frequency analysis was run from the waviness on the path of contact in the extracted 3D tooth surface waviness data. The cyclical waviness order number and amplitude for one meshing cycle were calculated and evaluated. A section corresponding to one meshing cycle was identified in the 3D data based on the path of contact coordinates obtained from the Klingelnberg AG software and used to cut out the section curve.

Detailed conditions of the test hypoid gear set used in verification are shown in Table 1. In mass production, runout accuracy for the cutter blade mounted on the cutter head is normally controlled to within a few microns; the test gear, however, was intentionally cut such that significant tooth surface waviness would be reproduced. The test gear had the same specifications as the gear that the conventional method was unable to quantitatively evaluate, as described in Section 2.2.

4.1. Comparison of Cyclicities between Sound Radiation in Actual Gears and Tooth Surface Waviness

Analysis results for the shape data on the path of contact for the tooth surface waviness extracted with this method and their frequencies follow below. Figure 14 is a section curve graph with its x-axis showing the length direction of the path of contact for the 3D tooth surface data extracted to show waviness only and its y-axis showing shape in the tooth surface normal direction. Figure 15 contains the frequency analysis results for the data from Fig. 14. Here, the x-axis represents the cycle order number for one meshing cycle and the y-axis represents the amplitude in the tooth surface normal direction. Using this method, the tooth surface waviness occurred on the test gear was successfully evaluated to have an average pitch of 0.65 mm with a cycle peak near the 8th order for the one meshing cycle and maximum amplitude of 0.55 μm . Next are the test results with the same hypoid gear assembled in the actual unit, given in Fig. 16. This data is for sound radiation as measured by a microphone in a set location with the transmission and power transfer unit mounted to a test bench. The y-axis shows the average sound pressure levels, and the x-axis shows the order numbers for the frequency of one meshing cycle of hypoid gear. As denoted with an arrow on the data, significant sound radiation from gears was confirmed, peaking near a 7.6-order frequency. This result matches the property values obtained from the measurements and evaluations of the tooth surface waviness. Based on the above verification results, the generating mechanism for high-order sound radiation from gears due to tooth surface waviness has been clarified and

Table 1 Detailed conditions of hypoid gear for inspection

	Pinion	Gear
Number of teeth	21	29
Module	5 mm	3.724
Pinion offset		
Mean spiral angle	36.1 deg	30.8 deg
Sum of pressure angle	40 deg	40 deg
Cutter head type	Stick blade	Stick blade
Cutter head radius	2.25 inch	2.25 inch
Number of cutter blade pairs	11 pairs	11 pairs
Cutter rotation speed	211 rpm	239 rpm
Generating feed speed	4.54 deg/sec	5.71 deg/sec
Cutter runout on machine	0.03 mm	0.03 mm

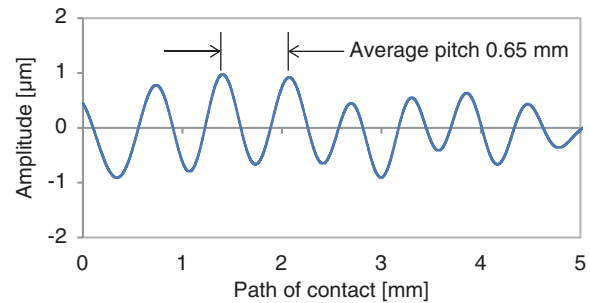


Fig. 14 Section curve data of waviness extracted

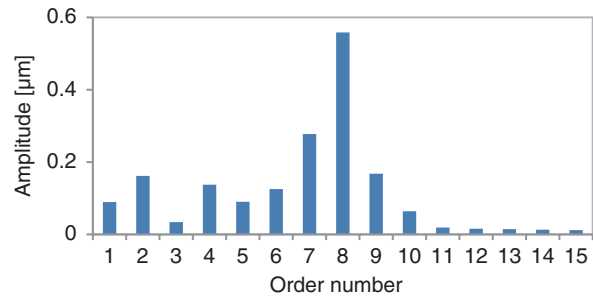


Fig. 15 Frequency analysis result of waviness extracted

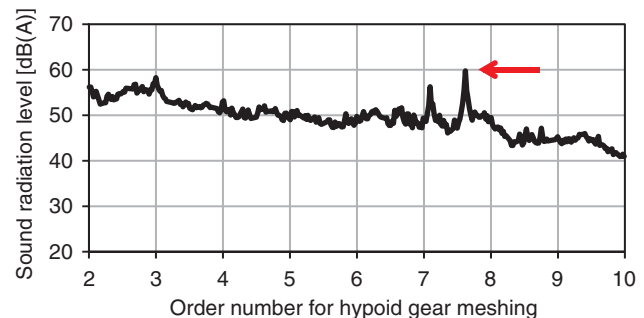


Fig. 16 Result of power transfer unit test

the detection accuracy of this evaluation method has been successfully verified.

4.2. Relationship between Pressure of Sound Radiation from Actual Gears and Amplitude of Tooth Surface Waviness

The same test measurements and assembled tests for tooth surface waviness were performed for multiple hypoid gears of the same specifications as the test gears in Section 4.1. These gears were selected at random, per mass production management criteria. From the results, a summary of the relationship between tooth surface waviness and high-order gear noise is given in Fig. 17. The x-axis of the graph shows the amplitude of tooth surface waviness, and the y-axis gives the pressure levels of peak sound radiation from gears in actual gearbox unit. As with the peak cycle order for waviness, the pressure levels of the sound radiation from gears plot to 7.6-order levels. This graph shows a positive correlation between sound radiation and amplitude of the waviness and quantifies the tendency of peak pressure levels of sound radiation from gears to increase as waviness amplitude increases.

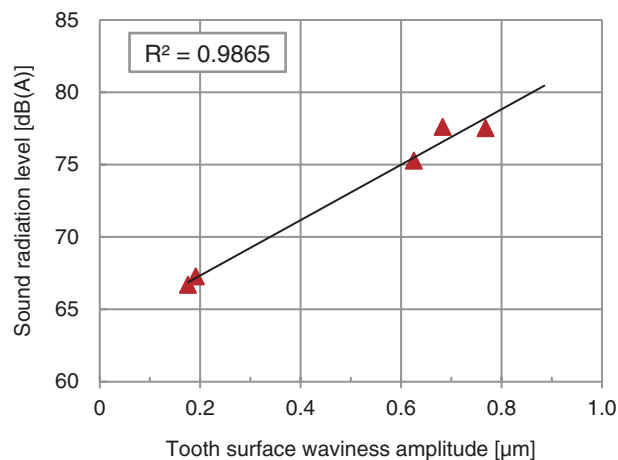


Fig. 17 Relationship between waviness amplitude and sound radiation level

5. Conclusion

In this study, through examination of an evaluation method for waviness, the following points have been achieved regarding the empirically known relationship between cyclical waviness as occurred by tooth surfaces in hypoid gears and high-order gear noises:

- (1) A new method for quantitative evaluation of tooth surface waviness has been created, using image filtering methods with 2D FFT and inverse FFT to extract the waviness component from 3D tooth surface

measurement data. Using measurement results from this method, it was revealed that tooth surface waviness is a generating mechanism for high-order gear noise.

- (2) The correlation between peak order waviness amplitude as evaluated with this method and pressure levels of sound radiation from actual gears was quantified, and indicator criteria for waviness amplitudes were obtained. The above has made quantitative evaluation of gears in isolation possible.

References

- (1) Kato, S.: Haipoidogiya Seido Sokuteigijutsu ni Kansuru Kenkyu, Kyoto University thesis for doctor, 205 p., (1996) (in Japanese)
- (2) Shibata, Y., Maki, H., Mori, K.: Influence to Transmission Error Such as Tooth Flank Modifications and Pitch Deviation in Meshing of Gears, TOYOTA Technical Review, Vol. 52, No. 2, p. 40-47, (2002)
- (3) Watanabe, T., Shiraki, K.: Fundamental Investigation of Hypoid Gear Noise and Vibration Using Gearbox Test Facility, Subaru Technical Review, No. 41, p. 138-144, (2014)
- (4) Egami, R., Kondo, T., Ikuma, M.: Mechanism of Occurrence of Ghost Noise in Helical Gears Resulting from Thread-type Tool Machining, Honda R&D Technical Review, Vol. 20, No. 2, p. 108-115

■ Author ■



Takeshi HORI



Naoto SHUKUSHIMA

Intelligent Power Unit for the 2018 Model Year Accord Hybrid

Jun MACHIDA* Naoyoshi MADOKA* Shogo HAYASE*

ABSTRACT

A compact intelligent power unit capable of being installed under rear seating was developed for the 2018 model year Accord Hybrid that is to be equipped with the SPORT HYBRID Intelligent Multi Mode Drive (i-MMD) system. The lithium-ion battery module was made more compact than in previous models, the structural parts were integrated, and the cooling structure was optimized. A 12 V DC-DC converter was also integrated with the power control unit in the engine compartment, making the intelligent power unit compact enough to install under the rear seat. As a result, the vehicle gained trunk room equal to gasoline-powered automobiles and realized a structure allowing continuous open space from the trunk through to the rear passenger compartment. The increased compactness also achieved a 10% reduction in weight.

1. Introduction

Global warming and air pollution are presently being treated as serious issues on a global scale, and these are major issues that should also be addressed by the automobile industry as a whole. The automobile manufacturers take the reduction of exhaust gas emissions as an urgent task, and in their search for effective solutions, they are pursuing the development of electric vehicles of all kinds, prominent examples being hybrid electric vehicles (HEVs), plug-in hybrid electric vehicles (PHEVs), battery electric vehicles (BEVs), and fuel cell vehicles (FCVs).

The two-motor hybrid system (SPORT HYBRID i-MMD) that Honda launched on the market in 2013 was initially installed in the Accord, which had both HEV and PHEV model lineups⁽¹⁾. In addition to an engine, this electric powertrain consists of an electric coupled continuously variable transmission (electric coupled CVT) with two built-in motors, a power control unit (PCU), and an intelligent power unit (IPU).

The IPU is made up of a high-voltage battery, the battery electronic control unit (ECU) that controls it, a high-voltage power distribution unit, and an air cooling

device. It functions to supply motor drive current, to discharge assist energy and store regeneration energy, to buffer surplus power when the generator is producing electricity, and so on. The IPU also manages the instantaneous input and output of energy and other such tasks in its comprehensive role of power management that supports hybrid systems.

The SPORT HYBRID i-MMD system that was announced in 2016 embodied a further evolution by means of technologies that increased the efficiency and reduced the size of the components^{(2),(3)}. In the Accord Hybrid, however, vehicle package constraints meant that the IPU had to be installed in the trunk, and as a result, the trunk room and usability were limited by comparison with gasoline-powered automobiles.

This paper introduces the development of the IPU for the new model SPORT HYBRID i-MMD system (Fig. 1) that is used in the 2018 model year Accord Hybrid. The development objective was a compact IPU capable of being installed under the rear seat of a hybrid sedan in order to provide trunk room equal to gasoline-powered automobiles, and realize a structure allowing continuous open space from the trunk through to the rear passenger compartment.

* Automobile R&D Center

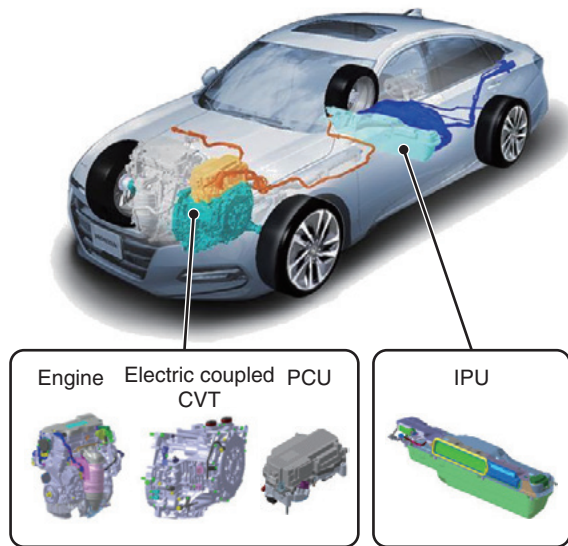


Fig. 1 Electric powertrain components in new SPORT HYBRID i-MMD system

2. Conceptual Approach

Figure 2 shows the configuration of the new model IPU. The IPU uses its case and cover to shape its space. The unit interior is occupied by a high-voltage Li-ion battery module, a high-voltage power distribution unit with an electric power shut-off function, battery cooling parts, and a battery ECU to monitor and control battery voltage and capacity.

In order to install the IPU under the rear seat, it has to be made a size that is compact enough to fit within the space constraints shown in Fig. 3. Table 1 shows the constraint

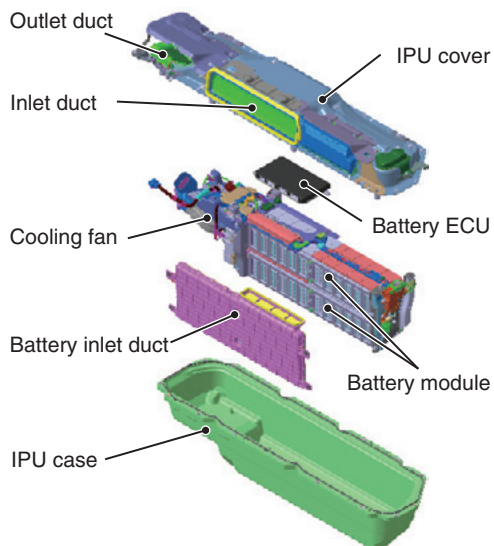


Fig. 2 Configuration of IPU

Table 1 Boundary constraints under rear seat

Direction	Constraint condition
Longitudinal	Passenger's leg space and fuel tank
Lateral	Space between side frames
Vertical	Rear seat height and cushion thickness Minimum ground clearance

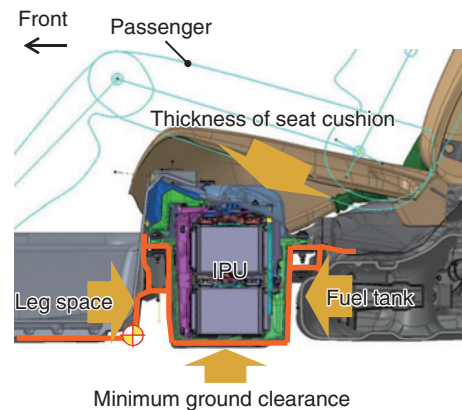
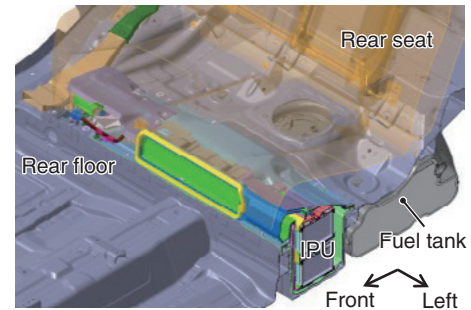


Fig. 3 Constraints on layout under rear seat

conditions in the directions that define that space.

Installation of the IPU under the rear seat was examined in the context of these layout constraint conditions.

3. Size Reduction Technology

3.1. Li-ion Battery Pack Size Reduction

As shown in Fig. 4, the Accord Hybrid IPU has 72 Li-ion battery cells installed. The previous model combined four modules, each containing 18 cells linked in a series. In the new model, this arrangement has been changed to have two modules, each containing 36 cells in a series, and the length in the cell stacking direction has been shortened. A center plate has also been added and the frame has been made stiffer so that the strength and stiffness of the modules are optimized to match the increase in the number of cells placed in a series.

The result was a 15.5% reduction in the ancillary

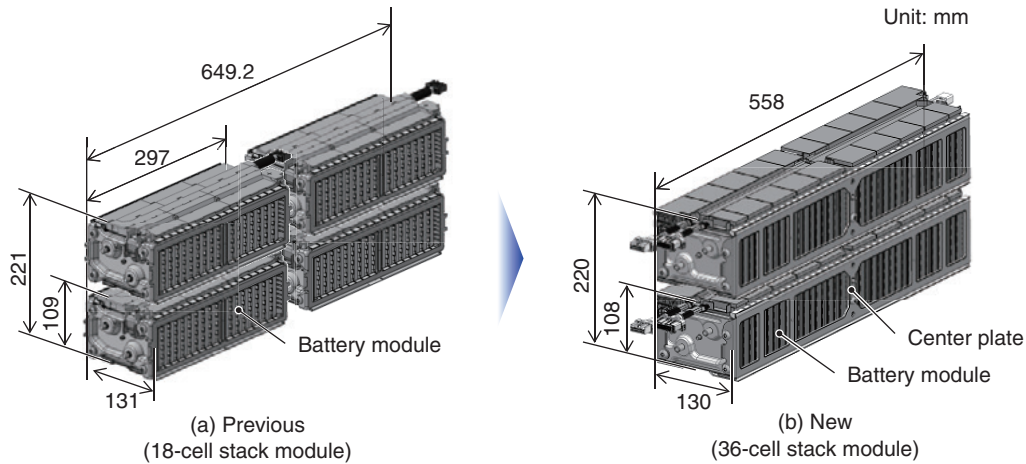


Fig. 4 18-cell stack battery module and modified 36-cell stack battery module

volume needed for module clamps and mountings, for high-voltage harness routing, and for harness connectors.

3.2. IPU Cover with Integrated Seat Stiffener Function

Most of the total height of the IPU is made up of the battery modules. In addition to that, however, there is the clearance between the cover, the case bottom, and the parts.

When the IPU is installed under the rear seat, there is a partial overlap of function between the IPU cover and the seat stiffener. By integrating the overlapping parts and functions, therefore, unnecessary clearance was eliminated.

The IPU cover also needs to be able to withstand the greatest load applied to it during a collision. The cover layout, materials, and thickness were therefore configured so that high-voltage parts, including the battery, would not come in contact with metal parts.

Figure 5 shows the CAE analysis results. These confirmed that the clearance between the cover and the high-voltage parts of the battery is maintained even if the cover on the upper surface of the IPU is displaced when the greatest load is applied.

3.3. Air Cooling Structure Integrated with IPU Case

The previous IPU had large ducts placed on both the inlet side and outlet side of the Li-ion battery modules to provide a route for cooling air to flow through. In the new model IPU, the case and cover connection surfaces were made into sealing structures so that the IPU interior would form an airtight structure. The use of a cooling fan to induce a cooling current then eliminated the need for an outlet duct.

Figure 6 shows the results for computational fluid dynamics (CFD) analysis of the cooling air flow. A flow channel is formed from the battery outlet to the air intake of the cooling fan. Figure 7 shows the distribution of the cooling air flow rate on the battery outlet side. This confirms that a cooling air flow is supplied to all the cells.

As a result, the new model IPU maintains cooling

performance equal to the previous model while reducing its dimension in the longitudinal direction of the vehicle by 26.3 mm (Fig. 8).

In a further step, where the previous IPU had the 12 V DC-DC converter unit built in, the new model SPORT

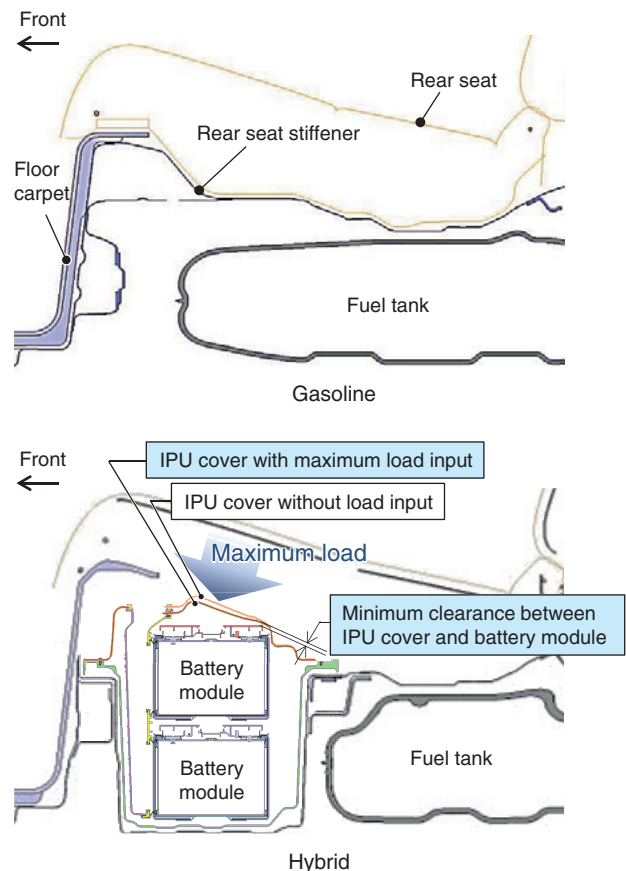


Fig. 5 Analysis results for IPU cover displacement under maximum load input

HYBRID i-MMD system functionally integrated this converter on the PCU side, in a single consolidated unit⁽⁴⁾.

By means of these techniques, the IPU volume was reduced by 32.1%, from the 70.8 L of the previous model to 48.1 L here. This enabled installation of the IPU under the rear seat.

As shown in Fig. 9, the 2018 model year Accord Hybrid successfully achieved the same VDA volume of 549 L as the gasoline-powered automobile. This additionally realized a continuous open space from the trunk through a large opening into the rear passenger compartment. This approach therefore achieved an affordable, highly efficient hybrid vehicle with trunk room equal to that of

gasoline-powered automobiles, together with a structure allowing continuous open space from the trunk into the rear passenger compartment⁽⁵⁾.

The size reductions were also accompanied by a reduction in IPU weight of 4.0 kg, making it 10% lighter than the previous model.

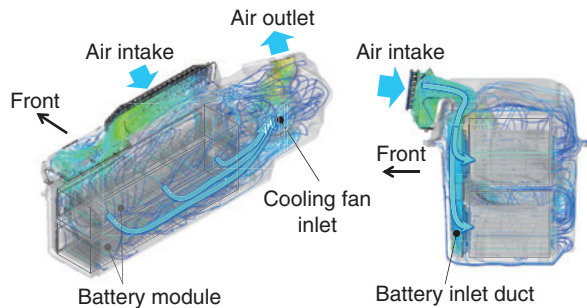


Fig. 6 Cooling air flow line inside IPU

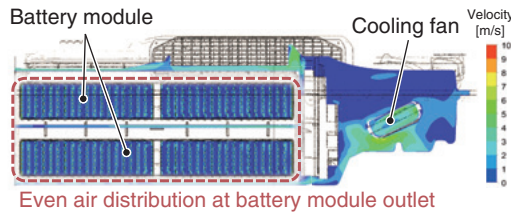


Fig. 7 Distribution of cooling air flow rate on battery outlet

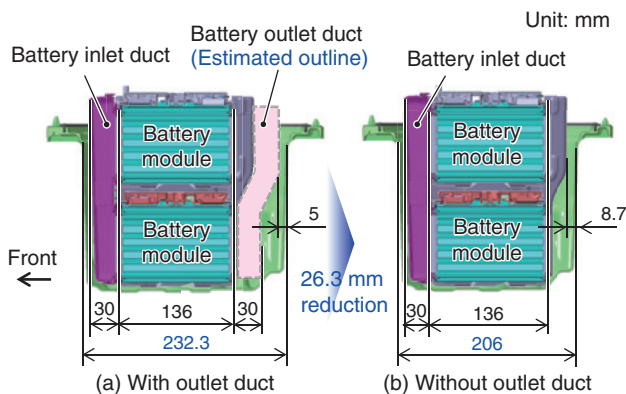


Fig. 8 Volume reduction of IPU due to elimination of battery outlet duct

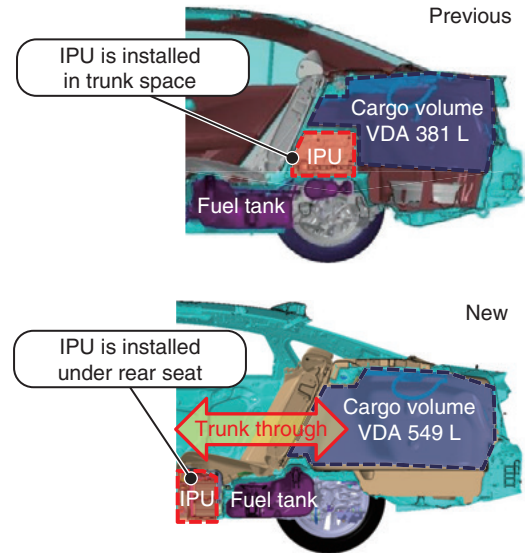


Fig. 9 Comparison of cargo volume of previous Accord Hybrid with that of new Accord Hybrid

4. Conclusion

An IPU capable of being installed under the rear seat was developed in order to achieve trunk room equal to gasoline-powered automobiles and realize a structure allowing continuous open space from the trunk through to the rear passenger compartment in the 2018 model year Accord Hybrid.

- (1) The Li-ion battery module was reduced in size, the IPU cover and car body seat stiffener were functionally integrated, and the air cooling structure was integrated with the case. These measures reduced the IPU volume by 32.1% from the 70.8 L of the previous model to 48.1 L, and enabled installation under the rear seat.
- (2) The reduction of the IPU size achieved a weight reduction of 4.0 kg over the previous model, making it 10% lighter.

References

- (1) Ide, H., Sunaga, Y., Higuchi, N.: Development of SPORT HYBRID i-MMD Control System for 2014 Model Year Accord, Honda R&D Technical Review,

Vol. 25, No. 2, p. 33-41

- (2) Takahashi, M., Aiba, T.: Development of a Transmission-Mounted PCU with High Power Density for 2nd Gen. 2-Motor Hybrid Systems, JSAE Annual Congress Proceedings (Autumn), p. 1916-1921, (2016) (in Japanese)
- (3) Ogami, M., Inoue, M., Takamatsu, H., Akai, K., Kamada, T.: Development of New-Configuration Motor for Hybrid System, JSAE Annual Congress Proceedings (Spring), p. 1100-1105, (2016) (in Japanese)
- (4) Sato, H., Nishioka, F., Ito, Y., Kobori, T., Machida, J., Yoshimoto, K.: Advanced Technology of Honda 2 Motor Hybrid System, Journal of Society of Automotive Engineers of Japan, Vol. 72, No. 9, p. 42-47, (2018) (in Japanese)
- (5) <http://hondanews.com/honda-automobiles/channels/accord-hybrid-press-releases/releases/all-new-2018-accord-hybrid-arrives-in-showrooms-as-the-new-benchmark-among-midsize-hybrids>, (2018)

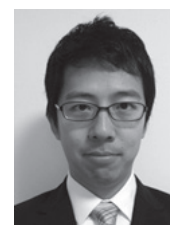
■ Author ■



Jun MACHIDA



Naoyoshi MADOKA



Shogo HAYASE

Initiative to Evaluate Effects of Atmospheric Impurities on PEM Fuel Cell Performance

Keiko YOSHIDA* Ryu OKANO*

ABSTRACT

The effect of atmospheric impurity concentrations on fuel cell performance when a fuel cell vehicle runs in a real-life environment was evaluated by bench tests using an actual automotive cell.

The effect of the concentrations of the gaseous atmospheric impurities SO_2 , H_2S , NO_2 , and NH_3 on fuel cell power over time was clarified. Comparison of nitrogen-based and sulfur-based impurities at the same concentrations revealed that sulfur-based impurities have a larger effect on power.

Regarding the non-gaseous impurity NaCl , as sea salt also scatters in aerosol form, these evaluation results were compared with the results for the conventionally used aqueous solution. The results showed that the effect on power differs according to the injection format. Therefore, when performing bench evaluation of NaCl -based impurities, it is necessary to select an appropriate NaCl form that conforms to the assumed environment.

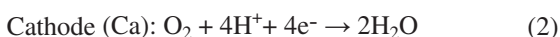
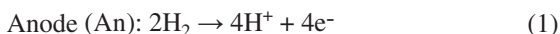
1. Introduction

Development of clean and fuel efficient vehicles is proceeding in order to address various issues such as suppression of global warming due to greenhouse gases (GHG), depletion of fossil resources, and reduction of environmental pollutants included in vehicle exhaust gas. Fuel cell vehicles (FCV) are gathering attention as a means of addressing the above issues as FCV do not emit CO_2 while running and have a driving range, refueling time, and cold start performance equal to that of gasoline vehicles⁽¹⁾.

Figure 1 shows a schematic image of a polymer electrolyte fuel cell (hereafter, “fuel cell”). The fuel cell (hereafter, “cell”) structure consists of a membrane electrode assembly (MEA) sandwiched between separators that form gas flow-fields, and a stack of multiple cells is called a fuel cell stack (hereafter, “stack”).

The MEA consists of a polymer electrolyte membrane (PEM), catalyst layers (CL), and gas diffusion layers (GDL).

The fuel cell reactions are the hydrogen oxidation reaction at the anode shown by Eq. (1) and the oxygen reduction reaction at the cathode shown by Eq. (2) below.



Fuel cells generally take in the oxygen to be supplied to the cathode from the atmosphere and react it with hydrogen inside the cells. Various atmospheric impurities may flow into the stack as FCVs become more widespread and run in increasingly diverse environments. Accordingly, it is thought that these impurity components and their concentrations will affect the stack power (hereafter, “power”). This calls for clarification of the effect of

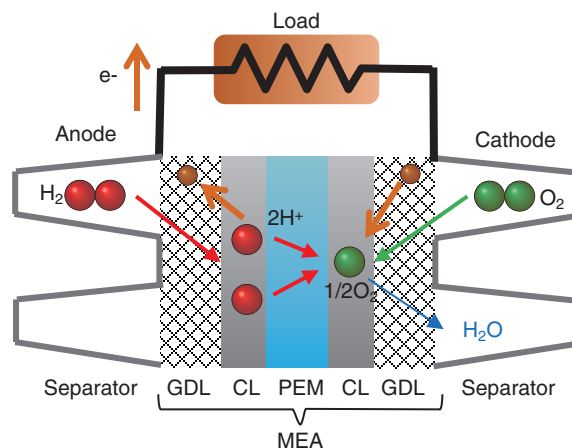


Fig. 1 Schematic image of PEM fuel cell

* Automobile R&D Center

atmospheric impurities in order to evaluate durability in real-life FCV operating environments. Under these circumstances, the effect of various atmospheric impurity types and concentrations on power is being studied in the fuel cell research field⁽²⁾. The main factor leading to reduced power is thought to be a drop in catalyst performance due to the adsorption of impurities to the surface of the Pt catalyst used in the cathode CL.

Fuel cell system costs remain high, and research is underway to reduce the amount of the precious metal Pt catalyst used as one method of reducing this cost⁽³⁾. However, simply reducing the catalyst amount reduces the Pt catalyst surface area, which is known to increase the effect to reduce power even at the same impurity concentrations⁽⁴⁾. The above demands to reduce catalyst amounts will continue to strengthen, so research to clarify the effect of impurity concentrations is important for stack development.

Most reports thus far on the effect of atmospheric impurities on power involve tests using commercial mini cells such as JARI cell⁽⁵⁾, and there are no known cases of using actual automotive cells.

Mini cells typically use a serpentine flow-field that differs from the flow-field of an actual automotive cell, so the diffusion behavior of air that contains impurities also differs. For this reason, the effect to reduce power may differ from that in an actual vehicle.

Table 1 shows typical atmospheric impurities that are thought to affect stack power⁽⁵⁾. These impurities include sulfur-based gases (H_2S , SO_2), nitrogen-based gases (NO_2 , NH_3), and the non-gaseous impurity NaCl.

Concentrations of atmospheric impurities are affected by various factors such as the geographical environment and weather conditions. For this reason it is not practical to quantitatively understand the effects of various impurities using only field trials. Therefore, this study conducted bench tests that generated power by supplying simulated atmospheric gases containing different impurity concentrations to the cathode side of an actual automotive cell.

Non-gaseous NaCl has conventionally been evaluated by the method of injecting an NaCl aqueous solution via the cell inlet⁽⁶⁾. However, in coastal areas NaCl also scatters in aerosol form⁽⁶⁾, so it is necessary to clarify the effect on fuel cell power of impurities entering in aerosol form. Therefore, this study performed bench tests using the method of aerosol injection in addition to the conventional method of injection by aqueous solution, and compared the results of both methods.

2. Bench Test Method

2.1. Gaseous Impurity Evaluation Method

2.1.1. Gas concentration settings

The assumed maximum concentration of each impurity was set as follows based on the values noted in Reference⁽⁵⁾.

Table 1 Atmospheric impurities to affect fuel cell

Impurity	Form	Concentration	Upper limit	Hot spot
H ₂ S	Gas	1700 ppb	1000 ppb	Hot spa Volcano
SO ₂	Gas	250 ppb	1000 ppb	
		19 ppb		
NO ₂	Gas	600-6000 ppb	10000 ppb	
NH ₃	Gas	21 ppb	100 ppb	Livestock Industry
		38 ppb		
NaCl	Solution Aerosol	100 µg/m ³		Coastal area

Table 1 lists these concentrations.

(1) H_2S

The main source of H_2S is volcanic gas. The measured H_2S concentration at Owakudani in Hakone is reported to have a maximum hourly average of 1700 ppb⁽⁵⁾. While higher concentrations may be observed depending on the volcanic area, the work environment management concentration for H_2S in the Industrial Safety and Health Act⁽⁷⁾ is 1000 ppb, so this study set an assumed maximum concentration of 1000 ppb.

(2) SO_2

SO_2 has the two sources of volcanic gas and fossil fuels. However, SO_2 originating from fossil fuels is tending to decrease due to advances in gasoline and factory exhaust gas desulphurization technology⁽⁸⁾. The concentration of SO_2 originating from volcanic gas was measured at 250 ppb at Owakudani in Hakone, but was 19 ppb in urban areas⁽⁵⁾. The environmental standard for SO_2 concentration established by the Ministry of the Environment⁽⁹⁾ is an hourly average of 100 ppb. Human body impact assessment results indicate that SO_2 becomes a disagreeable odor and unpleasantness increases at concentrations in excess of 1000 ppb⁽¹⁰⁾, so this study set an assumed maximum concentration of 1000 ppb.

(3) NO_2

NO_2 has various sources such as factories, thermal power plants, vehicles, and households⁽¹¹⁾. The measured value in urban areas is reported to be 600 ppb⁽⁵⁾. However, local concentrations of up to 6000 ppb at the exits of tunnels with heavy traffic in Tokyo have also been reported⁽¹²⁾. This study set an assumed maximum concentration of 10000 ppb.

(4) NH_3

Sources of NH_3 include vehicle exhaust gas and livestock manure and soil components in dairy regions. NH_3 originating from vehicle exhaust gas is reported to have a concentration of 21 ppb in areas of Tokyo with heavy traffic⁽⁵⁾. In addition, according to a report of atmospheric NH_3 concentrations originating from dairy⁽¹³⁾, a maximum

concentration of 38 ppb was observed in an area of Chiba Prefecture with a thriving livestock industry. Therefore, this study set an assumed maximum concentration of 100 ppb.

2.1.2. Experimental conditions

Evaluation used a stack consisting of ten actual automotive cells. Figure 2 shows a schematic image of the power generation evaluation apparatus. High purity hydrogen (purity 5N) was humidified and heated using a bubbling system to the same conditions as those in an actual vehicle and supplied to the stack as the fuel gas. Air with impurities removed (hereafter, “clean air”) was humidified and heated, then the specified impurity was mixed in via a T-shaped joint connected to the stack cathode gas inlet pipe and the mixture was supplied to the stack as the oxidant gas. The impurity flow rate from a gas cylinder adjusted to a constant concentration was controlled using a KOFLOC Model 3660 SERIES mass flow controller (MFC). Coolant was used to control the stack temperature to a constant temperature.

Figure 3 shows the operation mode used for evaluation. The horizontal axis shows the time, the 1st vertical axis shows the load, and the 2nd vertical axis shows the flow rate supplied to the cathode. Evaluation was performed using a mode (hereafter, “running mode”) that simulates the load variation produced by vehicle start, stop, acceleration, and deceleration. The cathode flow rate Q_{Air} changes according to the set load variation during the running mode as shown by Eq. (3).

$$Q_{Air} [\text{Nm}^3/\text{s}] = \frac{I}{4F} \times 0.0224 \times s \times \frac{1}{0.21} \quad (3)$$

I : Current [A]

F : Faraday constant [C/mol]

s : Stoichiometric number

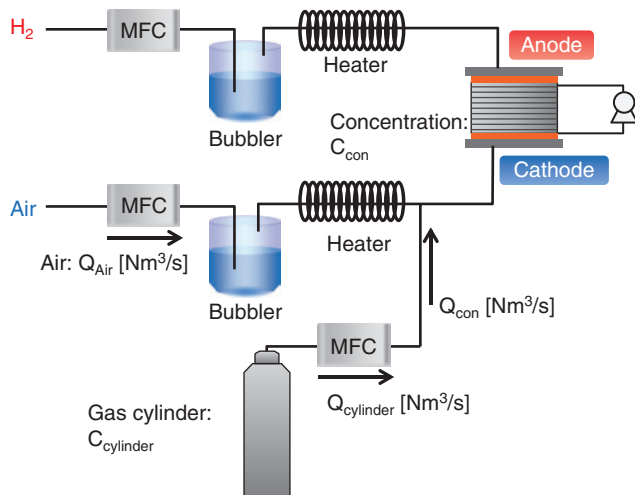


Fig. 2 Experimental scheme of gaseous impurity injection

The impurity flow rate Q_{con} entering the stack is obtained by Eq. (4) from the impurity concentration $C_{cylinder}$ of the gas cylinder and the gas flow rate $Q_{cylinder}$ [Nm³/s] supplied from the gas cylinder.

$$Q_{con} = C_{cylinder} \times Q_{cylinder} \quad (4)$$

The impurity concentration C_{con} in the air entering the stack was obtained by Eq. (5).

$$C_{con} = \frac{Q_{con}}{Q_{Air} + Q_{cylinder}} \quad (5)$$

The maximum power at the running mode during clean air supply was extracted and used as the reference for the decrease in power over time.

The difference between the power slope when various atmospheric impurities are supplied ($\Delta P_2/\Delta t$) and the power slope when clean air is supplied ($\Delta P_1/\Delta t$) was defined as the power decay rate as shown in Eq. (6) and used as the index for evaluating the effect on power when impurities are supplied (Fig. 4).

$$\text{Power decay rate [kW/h]} = \frac{\Delta P_2}{\Delta t} - \frac{\Delta P_1}{\Delta t} \quad (6)$$

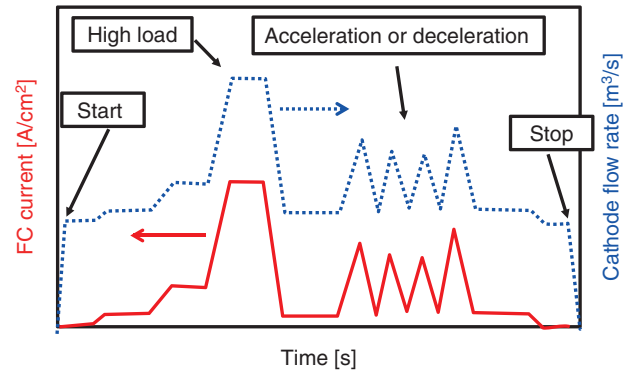


Fig. 3 Schematic drawing of evaluated operation mode

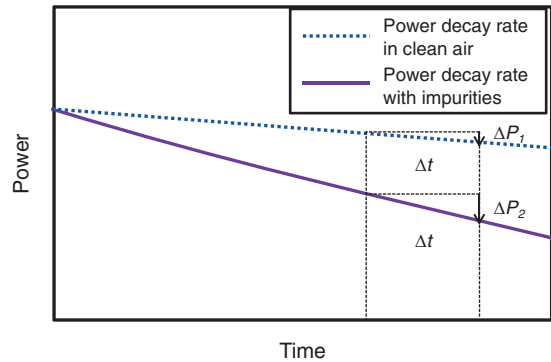


Fig. 4 Power decay rate by subtracting rate in clean air for correction

2.2. NaCl Evaluation Method

2.2.1. NaCl concentration setting

NaCl present in the atmosphere is called sea salt aerosol, and is an aerosol originating from the salinity of seawater. The sea salt aerosol concentration in the atmosphere is affected by various factors such as wind speed, wind direction, and weather, so long-term observation is needed to set the maximum value. Data released to the public could not be found, so this was measured by this study.

Preliminary investigations showed that the locations and times with the highest atmospheric sea salt aerosol concentrations in Japan are the southern islands such as Okinawa during the summer typhoon season⁽¹⁴⁾. Therefore, measurement was performed by the Japan Weathering Test Center's MIYAKOJIMA EXPOSURE TEST AND RESEARCH CENTER located in Miyako-jima, Okinawa Prefecture using a salinometer made by Electric Power Engineering Systems Co., Ltd.⁽¹⁵⁾ Figure 5 shows the frequency distribution of the NaCl concentration (4-hour average) for the one-month period from September 2 to October 2, 2016. This shows that the mode value was 20 $\mu\text{g}/\text{m}^3$, but there were also times when the concentration reached 100 $\mu\text{g}/\text{m}^3$.

Therefore, the NaCl concentration used in the tests was set to 100 $\mu\text{g}/\text{m}^3$, which was the maximum concentration measured in coastal areas.

2.2.2. NaCl test conditions

Evaluation was performed using the same stack specifications and conditions in order to clarify the difference when injecting NaCl by the aqueous solution of the conventional method and when injecting NaCl in aerosol form. Simulations were performed beforehand to investigate the vehicle power generation environment conditions that do not produce condensed water inside the flow-field, and the evaluation mode was set to the power generation mode with a fixed load of 0.1 A/cm². Evaluation used a stack consisting of ten actual automotive cells. High purity hydrogen (purity 5N) was humidified and heated using a bubbling system and supplied to the stack as the fuel gas. Clean air was humidified and heated, then NaCl

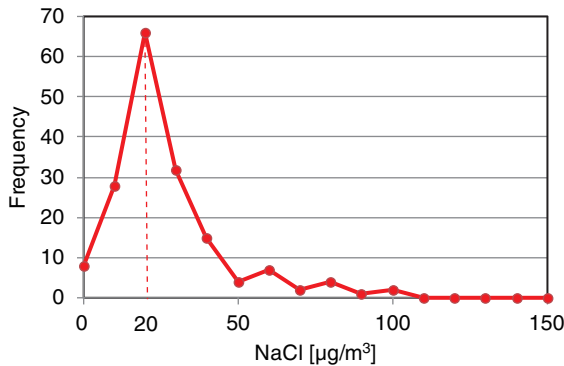


Fig. 5 Frequency distribution of sea salt aerosol

was mixed in via a T-shaped joint connected to the stack cathode gas inlet and the mixture was supplied to the stack as the oxidant gas. Coolant was used to control the stack temperature to a constant temperature.

When supplying NaCl by aqueous solution, the aqueous solution flow rate was controlled using an HPLC pump (Scientific Systems SERIES 1) to adjust the NaCl concentration in the clean air at the stack inlet to 100 $\mu\text{g}/\text{m}^3$. Figure 6 shows a schematic image of the power generation evaluation apparatus.

The NaCl flow rate Q_{NaCl} [kg/s] entering the stack was obtained by Eq. (7) from the NaCl aqueous solution concentration C_{solute} and the NaCl aqueous solution flow rate $Q_{\text{solution1}}$ [kg/s].

$$Q_{\text{NaCl}} = C_{\text{solute}} \times Q_{\text{solution1}} \quad (7)$$

The volumetric flow rate V_{Air} [m³/s] was obtained by Eq. (8) from the cathode flow rate Q_{Air} [Nm³/s], the stack inlet temperature T_{Air} [K], and the stack inlet pressure P_{Air} [kPa].

$$V_{\text{Air}} = Q_{\text{Air}} \frac{101.3}{P_{\text{Air}}} \times \frac{T_{\text{Air}}}{273} \quad (8)$$

The NaCl concentration C_{NaCl} [kg/m³] entering the stack was obtained by Eq. (9).

$$C_{\text{NaCl}} = \frac{Q_{\text{NaCl}}}{V_{\text{Air}}} \quad (9)$$

When supplying NaCl in aerosol form, the aerosol was generated using an aerosol atomizer (TSI Model 9306) and adjusted so that the NaCl concentration in the clean air at the stack inlet was 100 $\mu\text{g}/\text{m}^3$. The generated aerosol concentration was quantified according to the change in weight and the aqueous solution concentration.

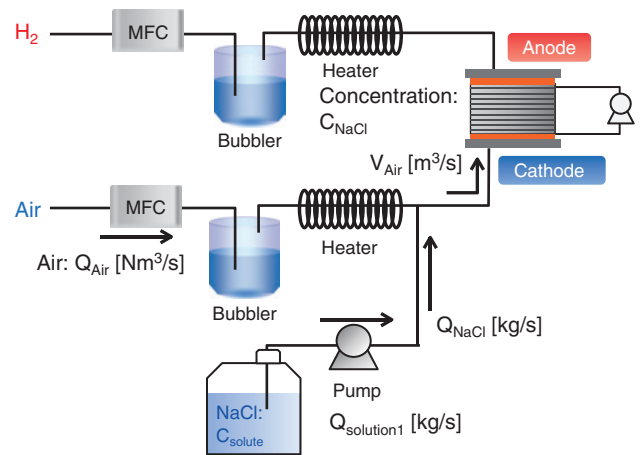


Fig. 6 Experimental scheme of salt water injection

Figure 7 shows a schematic image of the power generation evaluation apparatus.

The NaCl flow rate Q_{NaCl} [kg/s] entering the stack was obtained by Eq. (10) from the NaCl aqueous solution concentration C_{solute} and the change in NaCl aqueous solution weight $Q_{solution2}$ [kg/s].

$$Q_{NaCl} = C_{solute} \times Q_{solution2} \quad (10)$$

The total air quantity Q_t [Nm³/s] was obtained by Eq. (11) from the cathode flow rate Q_{Air} [Nm³/s] and the pressurized air flow rate Q_{pAir} [Nm³/s].

$$Q_t = Q_{Air} + Q_{pAir} \quad (11)$$

The volumetric flow rate V_t [m³/s] was obtained by Eq. (12) from the total air quantity Q_t [Nm³/s], the stack inlet temperature T_{Air} [K], and the stack inlet pressure P_{Air} [kPa].

$$V_t = Q_t \frac{101.3}{P_{Air}} \times \frac{T_{Air}}{273} \quad (12)$$

The NaCl concentration C_{NaCl} [kg/m³] entering the stack was obtained by Eq. (13).

$$C_{NaCl} = \frac{Q_{NaCl}}{V_t} \quad (13)$$

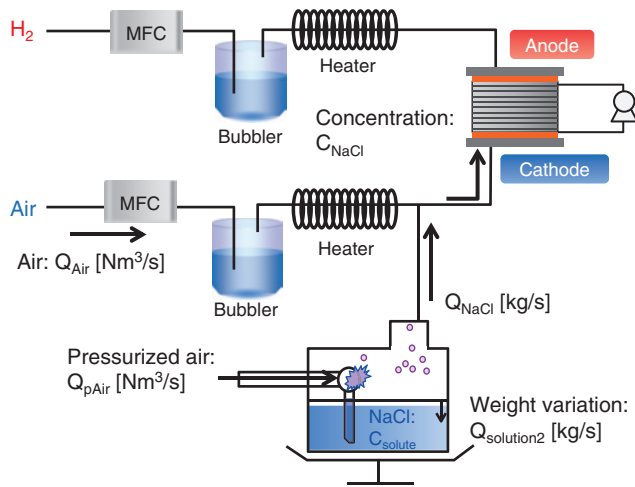


Fig. 7 Experimental scheme of salt aerosol test

3. Results and Discussion

3.1. Gaseous Impurity Results and Discussion

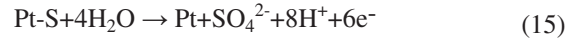
Figure 8 shows the relationship between the concentration of each impurity and the power decay rate.

The previously reported⁽⁵⁾ tendency where the drop

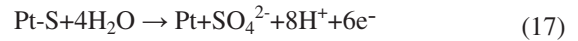
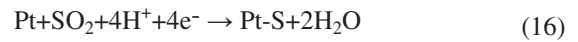
in power increases together with the concentrations of atmospheric impurities was also confirmed by this study.

Examination of the characteristics for each gas type shows that the power decay rate dependence on concentration exhibited similar behavior for the two types of sulfur-based impurities (H₂S, SO₂). The Pt catalyst surface reactions for H₂S and SO₂ proposed by Reference⁽¹⁶⁾ are as follows.

For H₂S:



For SO₂:



In both cases, the S adsorb to the Pt surface active points ultimately in the SO₄²⁻ form, so the power dependence on concentration is presumed to be similar for both H₂S and SO₂.

As described in Reference⁽⁵⁾, comparison at the same concentrations showed that NO₂ has a smaller effect on power reduction compared to sulfur-based impurities. However, Table 1 shows that whereas the maximum atmospheric concentration of sulfur-based impurities is 1000 ppb, there are some environments with a NO₂ concentration more than 1000 ppb, so it is thought that power may also be affected in regions with high NO₂ concentrations.

NH₃ did not exhibit an effect on power at the 100 ppb concentration anticipated in real-life environments. Like the previous report⁽⁵⁾, this study also confirmed that there is no effect on power as long as the concentration does not exceed 1000 ppb.

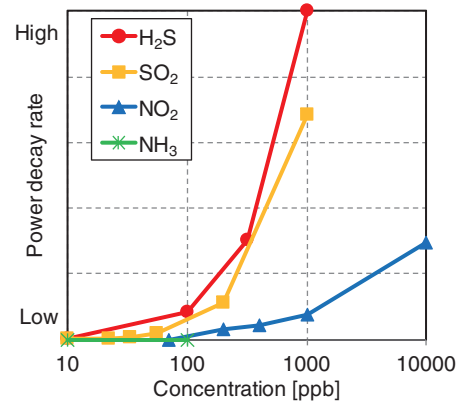


Fig. 8 Relationship between power decay rate and gaseous impurities concentration

3.2. NaCl Results and Discussion

Figure 9(a) shows the cell voltage after evaluation of NaCl injection. The difference from the beginning of life (BOL) performance indicates the degree of drop in power due to NaCl. The drop in power when NaCl is injected in aerosol form is less than that when injected by aqueous solution. Detailed analysis of the obtained values revealed a large rise in the ohmic overpotential component for cells where NaCl was injected by aqueous solution compared to cells where NaCl was injected in aerosol form, as shown in Fig. 9(b). The difference in cell voltage (ΔA) indicated by the shaded portion of the graph in Fig. 9(c) approximately matches the difference in ohmic overpotential (ΔB). Therefore, the difference in cell voltage is thought to be the difference due to this ohmic overpotential.

Reference⁽¹⁷⁾ states that the resistance increases when protons (H^+) on the PEM side chain are exchanged for Na^+ . The reason for this is thought to be that when NaCl is injected by aqueous solution, the Na^+ penetrates the electrolyte membrane and goes through a process of ion exchange with the protons (H^+) in the membrane, which increases the ohmic overpotential component.

In contrast, no increase in resistance was observed when NaCl was introduced in aerosol form. Therefore, the generation condition of condensed water inside the cathode flow-field was simulated using the Honda cell configuration and characteristics as the input to the polymer electrolyte fuel cell simulation software⁽¹⁸⁾. Figure 10 shows these results, and confirmed that condensed water is not generated in the cathode flow-field within the range of the test conditions used by this study. The reason why an increase in resistance was not observed under the test conditions used by this study is thought to be that NaCl

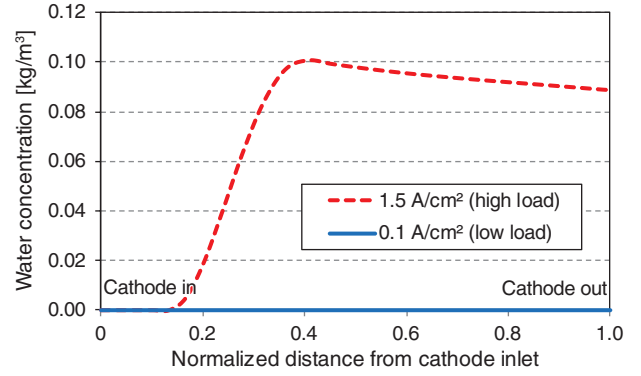


Fig. 10 Simulation of cathode channel

injected in aerosol form is not dissolved into condensed water and ionized, so Na^+ does not reach the membrane. However, consideration should also be given to cases where NaCl injected in aerosol form reaches the membrane under high load operating conditions where condensed water is generated in the cathode flow-field.

4. Conclusion

The effect of atmospheric impurity concentrations on fuel cell performance when a FCV runs in a real-life environment was evaluated by bench tests using a stack of an actual vehicle, and the following conclusions were reached.

- (1) Comparison of nitrogen-based impurities and sulfur-based impurities at the same concentrations showed that sulfur-based impurities have a larger effect on power.
- (2) Aqueous solution and aerosol were compared as the forms for injecting NaCl to the cells, and the results showed that each form affects power differently. When performing bench evaluation of NaCl-based impurities, it is considered important to select the appropriate NaCl form to be injected in accordance with the assumed environment.

References

- (1) Matsunaga, M., Fukushima, T., Ojima, K., Kimura, K., Ogawa, T.: Fuel Cell Powertrain for FCX Clarity, Honda R&D Technical Review, Vol. 21, No. 1, p. 7-15
- (2) Cheng, X., Shi, Z., Glass, N., Zhang, L., Zhang, J., Song, D., Liu, Z-S., Wang, H., Shen, J.: A review of PEM hydrogen fuel cell contamination: Impacts, mechanisms, and mitigation, Journal of Power Sources, Vol. 165, Issue 2, p. 739–756, (2007)
- (3) Wee, J-H., Lee, K-Y., Kim, S. H.: Fabrication methods for low-Pt-loading electrocatalysts in proton exchange membrane fuel cell systems, Journal of Power Sources, Vol. 165, Issue 2, p. 667-677, (2007)

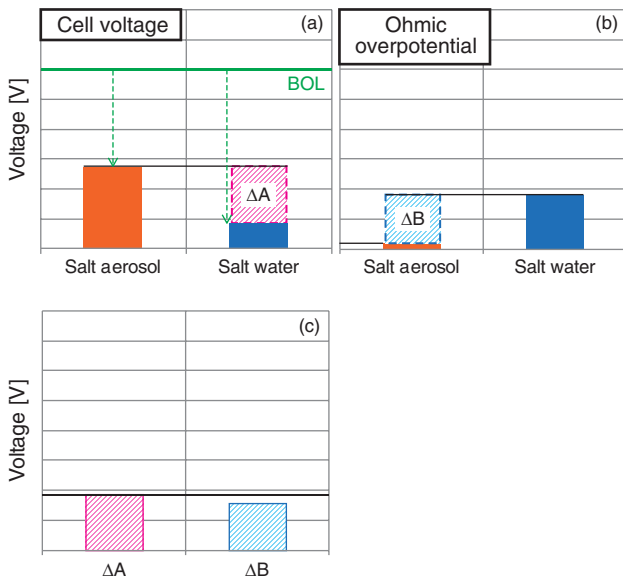


Fig. 9 Cell voltage and ohmic overpotential

- (4) Gasteiger, H. A., Panels, J. E., Yan, S. G.: Dependence of PEM fuel cell performance on catalyst loading, *Journal of Power Sources*, Vol. 127, Issues 1-2, p. 162-171, (2004)
- (5) New Energy and Industrial Technology Development Organization: Heisei 17-21 Nendo Seika Hokokusho Kotaikobunshigata Nenryodenchi Seru no Rekka Mekanizumu Kaiseki to Yojumyo Hyoka Shuho no Kaihatsu, (2010) (in Japanese)
- (6) Miura, K.: Production, and Chemical and Physical Properties of Sea Salt Particles, *Bulletin of the Society of Sea Water Science, Japan*, Vol. 61, Issue 2, p. 102-109, (2007)
- (7) Ministry of Health, Labour and Welfare: Sagyo Kankyo Sokutei Kijun no Ichibu wo Kaisei suru Kokujito no Tekiyoto ni tsuite, *Kihatsu 0207*, No. 3, (2012)
- (8) Ministry of the Environment: Heisei 17 Nendo Taikiosen Jokyo Hokokusho, (https://www.env.go.jp/air/osen/jokyo_h17/rep04.pdf)
- (9) <https://www.env.go.jp/kijun/taiki.html>
- (10) <http://www.city.yokohama.lg.jp/kankyo/mamoru/kanshi/worda/so2.html>
- (11) https://www.erca.go.jp/yobou/taiki/kids/aozora/haisyutu_04.html
- (12) Saito, S., Ishi, K., Ueno, H., Uchida, U., Yokota, H., Akiyama, K.: Characteristics of nitrogen oxides from vehicle emissions: A roadway tunnel study, *Annual report of the Tokyo Metropolitan Research Institute for Environmental Protection*, p. 37-42, (2012)
- (13) Chibaken Kankyoseikatsubu Kankyo Kenkyu Senta Suishitsu Kankyo Kenkyushitsu: Imbanuma wo Moderu toshita Tokuteiryuikiken niokeru Kankyokaizen to Saisei ni Kansuru Kenkyu Hokokusyo, p. 56, (2008) (in Japanese)
- (14) Okinawa Defense Bureau: Futemma Hikojo Daitaishisetsu Kensetsujigyo nikakaru Kankyoeikyo Hyokasho, Chapter 6, 6.11 Engai, p. 17 (in Japanese)
- (15) http://www.dentec.co.jp/business_contents/product_sale/salimeter.html
- (16) Franco, A. A.: *Polymer Electrolyte Fuel Cells: Science, Applications, and Challenges*, Pan Stanford, p. 428-485, (2013)
- (17) Bendert, J. C., Papadias, D. D., Myers, D. J.: The Effect of Na⁺ Impurities on the Conductivity and Water Uptake of Nafion 115 Polymer Electrolyte Fuel Cell Membranes, *Journal of The Electrochemical Society*, Vol. 157, Issue 10, (2010)
- (18) <https://www.psenterprise.com/sectors/fuelcell>

■ Author ■



Keiko YOSHIDA



Ryu OKANO

Method for Predicting Vibration Transmission of Car Air-conditioning Hose in Light of Prototype-less Development

Masahiro FUKAZAWA* Shingo UNIGAME*

ABSTRACT

A method for predicting the transmission of vibration by air-conditioning hose was devised for the purpose of examining the reduction of booming noise in the cabin and of engine noise caused by engine vibration during acceleration. For the stand-alone model of the hose, a multi-layered structure was recreated to take into account the anisotropy of the reinforcing layer. With regard to the internal pressure exerted by coolant gas sealed inside, influence factors that do not contribute to analytical precision were omitted, thereby shortening the time taken by analysis. A model of the vehicle state was created by joining a modal model of the body by the mode synthesis method, and vibration transfer function from the engine to the body was calculated. Calculation successfully yielded results with an error of 2% in the frequency and an error of 3 dB in the transfer function level. In addition, measures devised for the modeling method and calculation conditions enabled the total analysis time to be kept within 24 hours. With this prototype-less approach, the vibration transfer in air-conditioning hose can now be predicted.

1. Introduction

The demand for quietness in automobiles is increasing, and the reduction of booming noise composed of the main order components of engine vibration and engine noise composed of the high order components is an important issue⁽¹⁾ that largely determines the customers' impressions of a vehicle. At the frequency range of approximately 100 Hz to 1000 Hz where solid transfer of engine vibration is the main factor, there is a variety of transfer paths. One of these is the path shown in Fig. 1, whereby the engine vibration is transmitted from the air-conditioning hose (hereafter, the hose) and becomes cabin interior noise.

When the cause of engine noise and booming noise in the cabin interior was determined to be transmission by the hose at the stage when a prototype vehicle had been created, it would often be the case that the layout of parts in the engine compartment was already fixed. Consequently, making major changes to the hose routing could be a challenge, and taking noise countermeasures against hose transmission had to impose additional cost and mass.

Recently there has been a shift toward development with fewer returns so as to shorten the development schedule and reduce development cost. Therefore there is greater need for

a prototype-less development process that uses model-based systems engineering (MBSE). However, although there has been some research⁽²⁾ on vibration transmitted through the hose, there is no simulation technique for predicting the vibration transfer function (hereafter the transfer function) from the engine to the body. The reasons for that include the hose, which is made from multiple layers

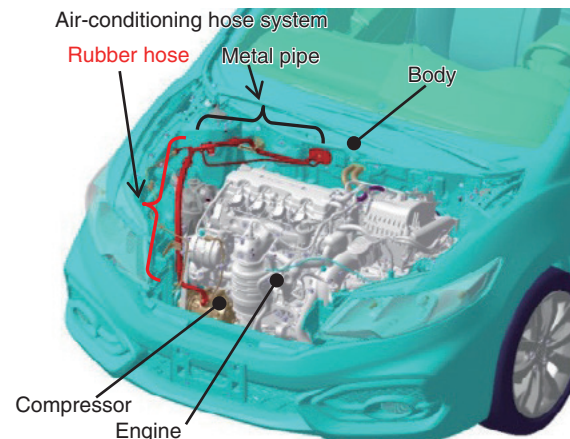


Fig. 1 Car air-conditioning hose system

* Automobile R&D Center

of different material, as shown in Fig. 2, the modeling of the influence of internal pressure from the coolant gas, the increased calculation time, and other such technical issues. The present research, therefore, has been oriented toward resolution of these various technical issues and so realization of simulation of the vibration transfer function of the hose in its installed state in a vehicle, with a view to prototype-less development using MBSE.

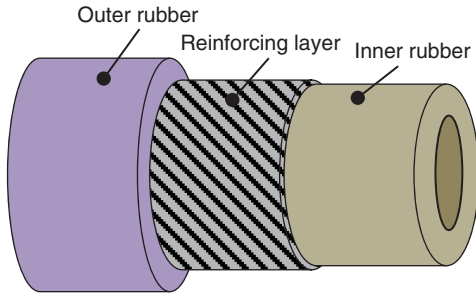


Fig. 2 Structure of air-conditioning hose

2. Development Goals

The goal of the present research has been to create a technique for predicting the transfer function of the hose in order to support prototype-less development.

It is conceivable that calculating precision can be heightened by more faithful modeling of the actual object. However, making the model more detailed than necessary leads directly to increases in modeling time and calculation time. The technique in this paper is intended for application to product development, and it is therefore required to achieve a balance between the two contradictory requirements of higher prediction precision and shorter analysis time.

First, a target for the minimum necessary transfer function prediction precision was set. As shown in Fig. 1, the hose is connected to the air-conditioning compressor. The compressor resonance frequency is generally from 100 Hz to 1000 Hz. Since this is close to the hose resonance frequency, it is necessary to examine the variance in the hose and compressor resonance frequencies. For that purpose, it was reasoned that prediction of the hose resonance frequency required precision of about $\pm 5\%$. In addition, in order to enable examination of specifications during development without obstructing the process, it is necessary to calculate the peak levels of the transfer function with a precision of ± 3 dB. Consequently, the target prediction precision was defined as a frequency error of $\pm 5\%$ for the vehicle state transfer function and ± 3 dB for the transfer function level.

The total time taken by analysis, including modeling time and calculation time, was also considered. On the assumption that the work time in a single day is eight hours,

then it was reasoned that design work could proceed without obstructing the development schedule if results could be obtained within three days from the start of analysis. A total time of 24 hours or less was set as the target.

3. Overview of the Created Technique

As shown in Fig. 3, the air-conditioning hose layout has two systems, the high-pressure pipe and the low-pressure hose. The low-pressure hose generally links the engine and the body, so structurally it is likely to transmit vibration. The low-pressure hose connects the air-conditioning compressor that is mounted on the engine with the evaporator that is attached to the body, and most of it is made up of metal pipe. In the middle between them, however, a rubber hose is ordinarily used in order to cut off the vibration. The object of the technique developed here is the low-pressure hose that, for the above reasons, is considered important in the transmission of vibration.

The applicable vehicle was defined as a medium-size sedan model. The state of the hose when predicting the transfer function fully reproduced the actual state as shown in Fig. 4.

The coolant gas that flows through the low-pressure air-conditioning hose in its installed state in a vehicle is at a pressure of approximately 1 MPa. In order to handle that

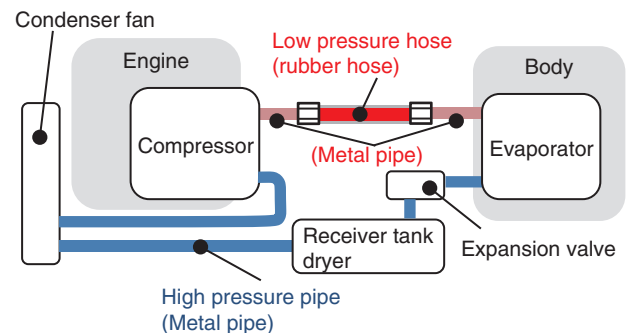


Fig. 3 Overview of car air-conditioning hose system

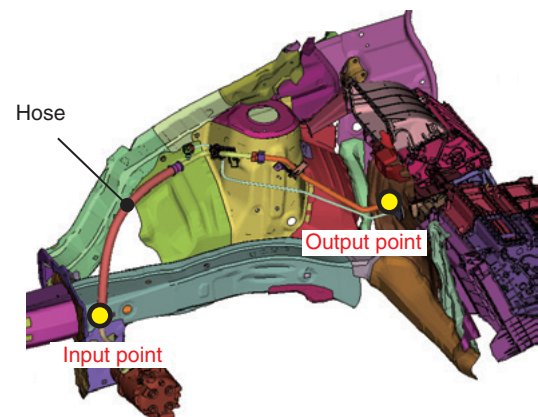


Fig. 4 Measuring points in sedan model

pressure, the hose has a multi-layered structure that includes a reinforcing layer of fiber. Transfer function measurements were made with and without internal pressure at the measuring points shown in Fig. 4. Comparison of the results, as shown in Fig. 5, indicates a change in level of 10 dB or more at the most, and it is also apparent that the transfer function varies across a wide range from approximately 150 Hz to 1000 Hz. This is thought to be due to the change in the Young's modulus and shape of the hose because of the varying magnitude of the internal pressure, and it showed that the internal pressure of the hose would need to be taken into consideration in order to determine the transfer function. For prediction of the transfer function, therefore, it was decided to reproduce a state that includes the internal pressure.

The following steps were therefore included in this technique:

- (1) Examine the physical property values and meshing conditions needed to reproduce the stand-alone hose mode in a state without internal pressure
- (2) Taking the hose created in step (1), create a model that adds the influence of internal pressure
- (3) In order to reproduce the state as installed in a vehicle, create a modeling technique that adds the pipe, body, compressor, and other surrounding components.

With regard to the body that is to be joined, particularly in step (3), use was made of existing FEM body model resources created for the purposes of other analyses. This made it unnecessary to create a large new model and reduced the time taken by analysis. After creating this technique using a medium-sized sedan model, the general applicability of the technique was verified on a compact SUV.

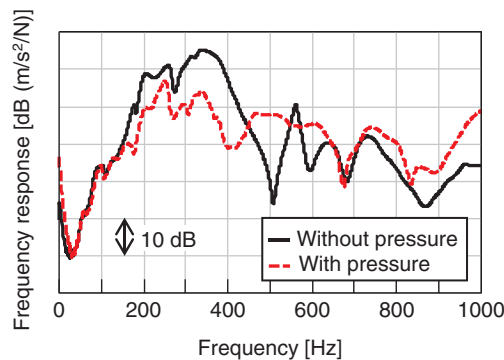


Fig. 5 Test results produced with and without pressure

4. Details of Created Technique

4.1. Creation of a Prediction Model for a Stand-alone Hose with a Multi-layered Structure

Since the hose has a multi-layered structure, as stated above, a CAE model that takes the physical properties of

each layer into consideration was created.

The hose has a diameter of approximately 20 mm, and the model as a whole was given a mesh size of 5 mm. For the meshing of each rubber portion in a radial direction, the size of each rubber layer was made finer so it could have three-layered meshes, as shown in Fig. 6. This made the model capable of precisely reproducing modes in which the hose is deformed in a radial direction.

With regard to the reinforcing layer, which is located between the other two layers, its influence on the Young's modulus of the hose is large. Therefore the reproducibility of the mode was enhanced by configuring the anisotropy in the tensile and compressive directions.

Next, starting with a hose of a straight shape that is not fixed in place, a model with a 90° bend was created and the transfer functions at the measuring points shown in Fig. 7 were calculated. Figure 8 shows the results. The reproduction precision with respect to the measurement was 2 dB of error in the transfer function level at the resonance peak. The model of the hose as a multi-layered structure that includes the hose fibers successfully reproduced the measurement with the bend of the hose also taken into consideration.

4.2. Modeling the Influence of Hose Internal Pressure

Two main factors that are influenced by the addition of internal pressure can be cited. The first is that the stress and strain are in a non-linear relationship because the hose is made from rubber that is a polymeric material, and the resulting changes in Young's modulus are an influencing

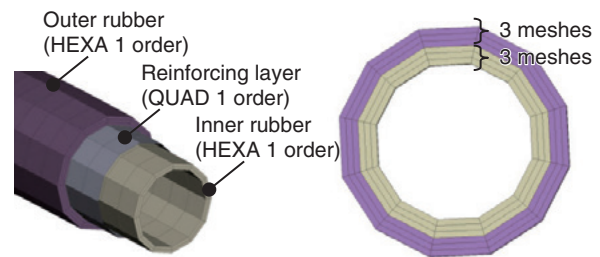


Fig. 6 Modeling of multi-layered hose

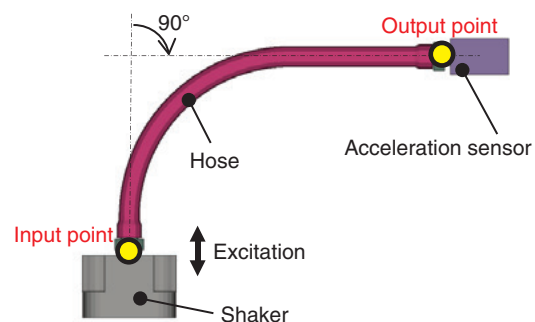


Fig. 7 Way to bend hose and measuring points

factor. The second is that the hose inflates due to pressure from the inside, and the resulting changes in shape due to varying diameter and total length is an influencing factor.

A model was first created that takes the changes in the Young's modulus and shape of the rubber hose into detailed consideration without omitting the two influence factors. The method followed was to specify the surface of the innermost side of the hose where it comes into contact with the coolant gas, and to apply pressure to that surface in order to reproduce the influence of the internal pressure. Non-linear analysis was then used to calculate the changes in the Young's modulus and shape of the hose, and the results from comparison with the model before internal pressure was added are shown in Fig. 9(a) and (b). The addition of internal pressure causes the Young's modulus and shape to change together, and it is apparent that the two influence factors have been successfully reproduced. This detailed model can be considered to fully reproduce the actual phenomenon, but even though this is a stand-alone model of the hose, the calculation took four hours or more.

The aim of this technique is to achieve a balance between calculating precision and reduction of the time taken. Therefore, as long as the calculating precision can be maintained, it is desirable to minimize as much as possible the influence factors that are taken into consideration for the purpose of shortening the time taken by analysis. Here the rate of dimensional change in diameter and total length of the hose due to the addition of internal pressure were calculated, with the results shown in Fig. 10. It is apparent that both are very small, at 1.9% and 0.3% respectively. Therefore verification was carried out to determine whether calculating precision can be maintained even in a state that ignores those deformations.

A simplified model was created that takes only the change in the Young's modulus of the rubber hose into consideration, ignoring the changes in shape. For this, the same method for applying internal pressure as described above for the detailed model was used. The changes in the

Young's modulus and shape due to internal pressure were then calculated using static analysis. The dynamic analysis that was used after that to calculate the transfer function, as shown in Fig. 9(c), incorporated only the changes in the Young's modulus. This analysis was performed without incorporating information about changes in the diameter

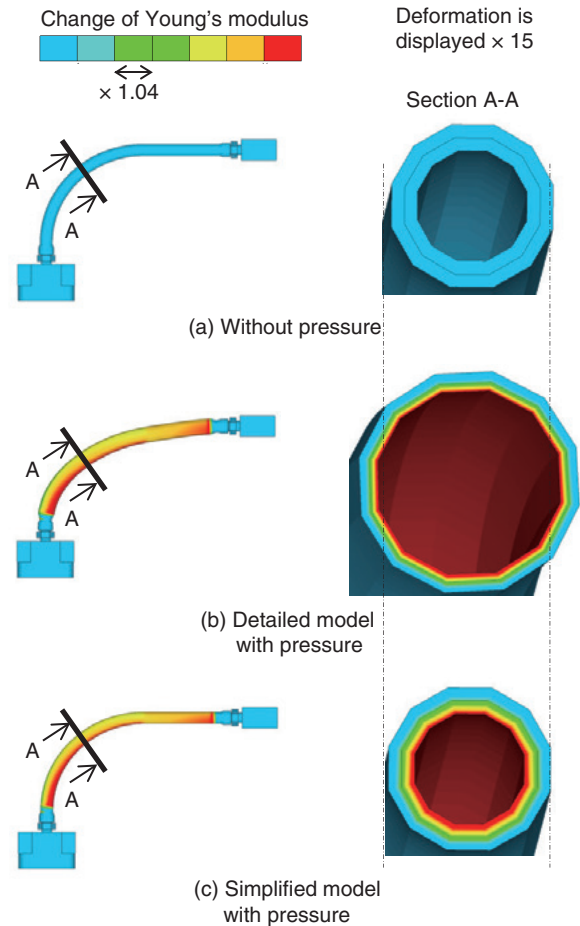


Fig. 9 Pressure effect result for each model

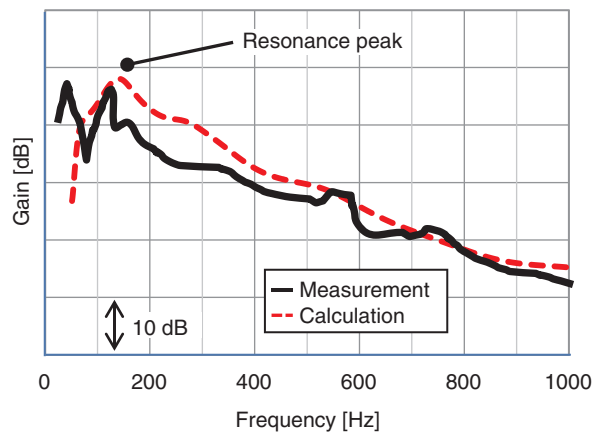


Fig. 8 Frequency response accuracy under conditions without pressure

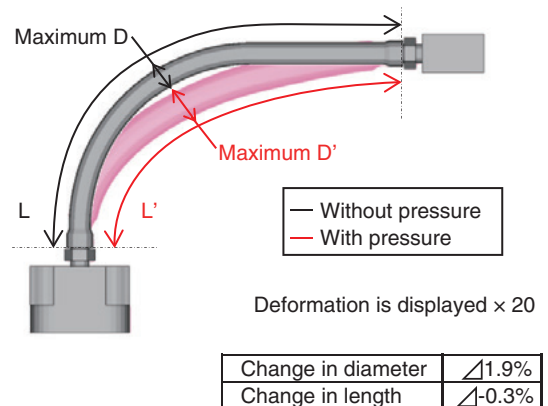


Fig. 10 Change in hose dimensions by pressure

and total length of the hose. This simplified model uses linear analysis and so was able to yield calculation results in one hour or less.

The difference between the calculation results from the detailed model and the simplified model of the transfer function are shown in Fig. 11. It is apparent that the differences in the resonance peak frequency was 1% and in the transfer function level it was 1 dB or less. Meanwhile, the calculation time of the simplified model was successfully shortened to approximately one-fourth that of the detailed model. Consequently, it was decided that for the present research, a simplified model would be adopted that takes into consideration only the change in the Young's modulus of the hose in order to maintain the minimum required calculating precision while shortening the calculation time. Figure 12 shows the precision with which the simplified model reproduces the actual measurements. At the resonance peak, the transfer function level error is 3 dB, indicating that a technique for predicting the hose transfer function with internal pressure applied has been successfully created.

4.3. Reproduction of the State as Installed in a Vehicle

The stand-alone hose model created as described up through the previous section was used to carry out prediction of the transfer function in the state as installed in a vehicle.

To reproduce the vehicle state, the pipes and clips that attach to the hose were added. The approach to the body, as described earlier, was to avoid the necessity for creating a large new model and instead to use an existing model. The present technique deals with the frequency range of vibration that is local, which means frequencies of approximately 100 Hz or higher and only in the vicinity of the hose. The rear end of the vehicle is distant from the engine compartment and so has little influence. Therefore a "cut body model" consisting only of the front end of the vehicle was used. Moreover, component mode synthesis method (CMS)^{(3), (4)} was used to join the modal models of

the body and engine to create the model. The time required for analysis was successfully shortened by these means.

Figure 13 shows the transfer function results. At the resonance peak, the reproduction precision with respect to measurement met the target by achieving reproduction precision of 2% for frequency error at the target $\pm 5\%$ value, and by achieving reproduction precision of 3 dB for transfer function level error at the target ± 3 dB value. The total time taken by analysis was 23.75 hours versus the target of 24 hours or less, so this also successfully achieved the target.

As indicated by the above, a technique was successfully created to predict the transfer function of a hose in the state as installed in a vehicle.

4.4. Verification of General Applicability

This technique was applied to a compact SUV and its general applicability was verified. Although this model vehicle differs from the medium-sized sedan model that has been covered up to the preceding section above in its engine, hose, pipe, and body, the same modeling method

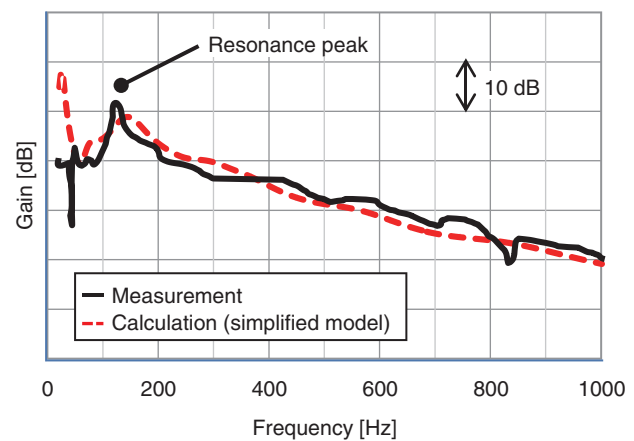


Fig. 12 Frequency response accuracy under conditions with pressure

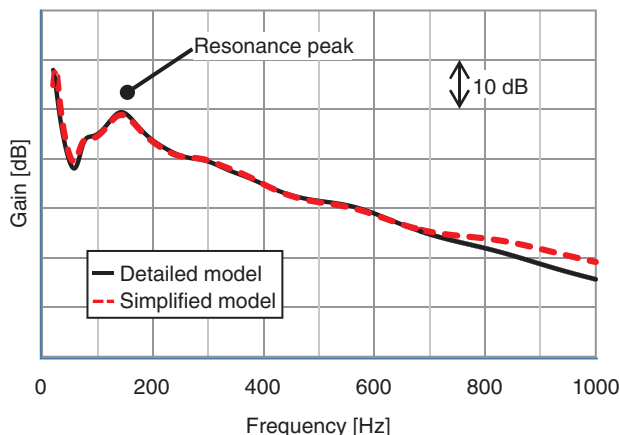


Fig. 11 Frequency response results for each model

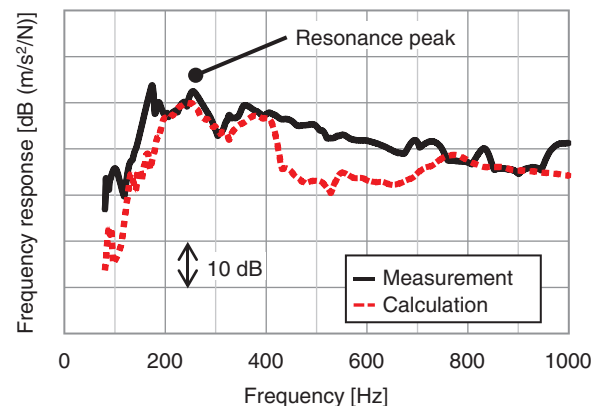


Fig. 13 Frequency response accuracy of hose model installed in sedan

was used to conduct an analysis. Figure 14 shows the measuring points and Fig. 15 shows the transfer function results.

At the resonance peak, the reproduction precision with respect to measurement met the targets at 2% for frequency error and 2 dB for transfer function level error. The total time taken by analysis was 23.25 hours, which successfully achieved the target.

This can therefore be considered to have sufficient general applicability as long as the hose layout is of the normal type used as samples. This verifies the usefulness of the created prediction technique.

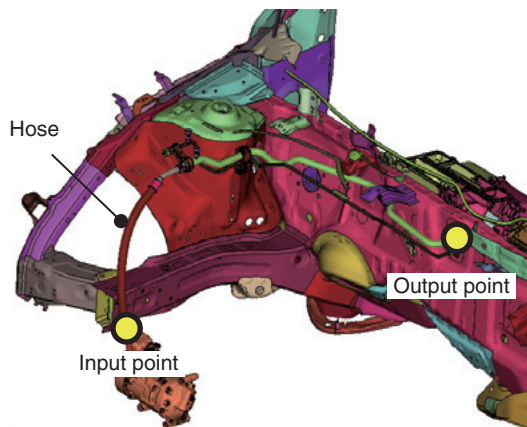


Fig. 14 Measuring points in compact SUV

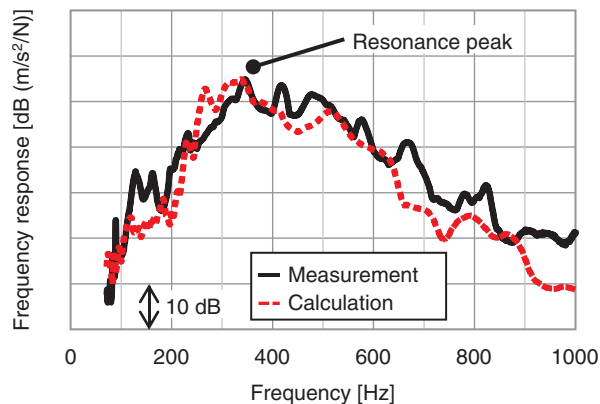


Fig. 15 Frequency response accuracy of hose model installed in compact SUV

5. Conclusion

In order to support prototype-less development in the initial development stages, a method for predicting the transmission of vibration was devised for use with low-pressure air-conditioning hose, which is one transfer path for the booming noise and engine noise that occur during

acceleration. The general applicability of this method was also verified.

In this process, the technique was also examined for its applicability to vehicle development. The following results were obtained.

- (1) In the vehicle state, the frequency error was 2% at the resonance peak and the transfer function level error was 3 dB, so vibration transmitted by the hose was successfully reproduced with a precision that presents no obstacle in vehicle development.
- (2) Measures devised for the method of analysis and simplification of the modeling kept the total time taken by analysis to 24 hours or less. The time taken by analysis was therefore successfully shortened while maintaining the calculating precision described in (1).
- (3) With regard to the body-sized model, the use of existing FEM model resources successfully shortened the time required.

As indicated by the above, a technique was successfully created for achieving sufficient precision in predicting the vibration transfer function of a hose in a state as installed in a vehicle with a view to prototype-less development using MBSE.

References

- (1) Matsumoto, S., Sumi, K., Yamamoto, K., Onishi, Y.: Analysis of Radiated Noise from Cylinder Block and Oilpan, JSAE Annual Congress Proceedings, No. 962, p. 229-232, (1996) (in Japanese)
- (2) Tsunoda, K.: Automotive Hose, The Society of Rubber Science and Technology, Japan, NIPPON GOMU KYOKAISHI, Vol. 80, No. 10, p. 375-379, (2007) (in Japanese)
- (3) Fukazawa, M., Murao, T., Unigame, S.: Prediction Method for Vibration Transmission of Hydraulic Engine Mount, Honda R&D Technical Review, Vol. 27, No. 1, p. 119-125
- (4) Nagamatsu, A., Okuma, M.: Bubun Kozo Goseiho, BAIFUKAN, p. 103-105, (1991) (in Japanese)

■ Author ■



Masahiro FUKAZAWA



Shingo UNIGAME

Analysis of Capacity Fading for High-power Lithium Ion Rechargeable Batteries

Pu QIAN*

Tomohiro KINOSHITA*

Hiroshi SAKAI*

Terumi FURUTA*

Mitsumoto KAWAI*

ABSTRACT

An analysis for elucidating the mechanism of capacity loss due to calendar aging in high-power lithium ion rechargeable batteries have been performed. The degradation of the batteries used in this research did not follow any particular trend with storage time, so that it could not be explained by the general, simple square root law. Electrochemical measurements revealed two factors that lead to degradation by a capacity balancing shift among the electrodes. The two factors are: the loss of active lithium which flows between anode and cathode, and cathode capacity fading. In order to identify the above two factors, we performed some material analyses of the battery. The results clearly showed that the loss of active lithium is caused by the formation of a solid electrolyte interphase due to reduction of solvent in electrolyte on the anode surface, and cathode capacity loss due to partial crystalline structure transition of the active material. A further detailed chemical composition analysis of the solid electrolyte interphase showed clearly that as the electrolyte components undergo electrochemical reductive degradation on the anode surface, an initial film largely dominated by organic components is first formed, then change in chemical composition and structure occurs whereby those organic components transform into inorganic components. However, the total amount of lithium in the film that was measured by material analysis did not match with the amount of cyclable lithium lost between the cathode and anode as obtained by electrochemical measurement techniques. We newly showed that there is the crucial need to determine the amount of the active lithium loss using electrochemical measurement techniques.

1. Introduction

Lithium ion rechargeable batteries came into practical use in 1991⁽¹⁾. Given their high capacity, high power⁽²⁾, and other such features, these batteries have entered notably widespread use in portable equipment since then. In recent years they have also been used as batteries to drive electric vehicles, and their use is expected to expand greatly in the future^{(3), (4)}. Although lithium ion rechargeable batteries can realize high capacity and high power, however, they face the issue of aging⁽⁵⁾ just as other chemical batteries do. Battery aging is known to occur from cycle aging, which depends on the number of charge-discharge cycles, and from storage aging, which depends on storage time (and is hereafter referred to as calendar aging in order to correspond to cycle aging). When batteries with features like these are used, a long-term performance prediction is generally made on the basis of test data gathered over a limited period of time. Therefore many evaluations are implemented under diverse test conditions, and maps of the relationships between each

condition and aging are created. Calendar aging in a battery is dependent on temperature and state of charge (SOC), but since they are not in a simple linear relationship, creating a map requires an enormous number of tests and times in accordance with the matrix of conditions. In order to reduce the development period this requires, the usual approach used to be to employ the prediction technique using the square root law⁽⁶⁾⁻⁽¹⁰⁾ on the hypothesis that the capacity fading by the battery was proportional to the square root of the storage time or the number of cycles. The foundation for use of the square root law is the rate of formation of the solid electrolyte interphase (SEI) film on the surface of the anode. In other words, the basic hypothesis is that the cyclable lithium ions that transfer back and forth between cathode and anode (hereafter referred to in short as active lithium) are lost to form the SEI so they can no longer transfer back and forth between the cathode and anode, and battery capacity declined as a result. However, battery aging is the result of a complex interaction of multiple factors^{(5), (11)-(18)}. As shown in Chapter 3 of this paper,

* Automobile R&D Center

behavior that does not agree with the simple square root law is also known to occur^{(11), (14)}. In light of reports that battery capacity can be evaluated from the single-electrode capacities of the cathode and anode severally and from the shift in the electrode balancing between them⁽¹¹⁾⁻⁽¹⁴⁾, it is anticipated that a new, highly accurate predictive equation will be derived to account for this kind of capacity fading behavior by incorporating the substance of these findings.

In order to derive a new predictive equation, the cause of the above balancing shift will need to be clarified, and the amount of the shift will need to be determined. There are reports that the amount of lithium incorporated into the SEI correlates with the active lithium loss^{(8), (10), (13)}. However, there have been no reports as yet that explain the cause and effect relationship between single-electrode capacity fading and active lithium loss and that cite the creation of predictive equations for capacity fading based on those mechanisms.

The aim of the present research, therefore, has been to explain the mechanism of battery aging, which is important for predicting battery aging behavior accurately, and for that purpose, to carry out an analysis that combines electrochemical measurements with an analysis of the substances that make up the battery. This report will discuss this process up to the results from an analysis of capacity fading in batteries. The derivation of a new predictive equation is scheduled to be covered in detail in the next report. Here, the below phenomenon of capacity fading due to battery aging will be referred to simply as aging.

2. Experimental Method

2.1. Specifications of Batteries Studied

Prismatic lithium ion batteries for high-power applications were used for this research. The external dimensions of the batteries were approximately 100 mm × 80 mm × 16 mm. The battery cathode was of NMC, a layered oxide composition of nickel, manganese, and cobalt. The anode was a carbon material. A microporous polyethylene film was used for the separator. The electrolyte was a carbonates solvent with LiPF₆ added as the supporting salt. The batteries used for testing had already been through a conditioning process when they were shipped, so they were in an immediately usable state.

The SOC of the batteries was determined from the state of charge-open circuit voltage (SOC-OCV) curve for the new batteries. Capacity retention was defined as battery capacity after aging as a percentage of the battery capacity before aging. For aging testing, the tested battery was charged to a specified SOC after which it was stored for a specified period of time in a thermostatic chamber at a specified temperature. Table 1 shows an example of changes in capacity fading over time obtained from charge-discharge test results for the batteries studied at each of various specified storage periods.

Table 1 Capacity retention during calendar degradation test

Temperature (°C)	SOC (%)	Aging time		Capacity retention (%)
		days	day ^{1/2}	
65	80	610	24.7	62.7
65	50	450	21.2	85.7
65	20	450	21.2	91.5
55	80	450	21.2	84.3
55	50	250	15.8	89.9
55	20	450	21.2	95.8
45	80	190	13.8	83.7
45	50	200	14.1	93.6

2.2. Electrochemical Measurements

The batteries used in testing were disassembled afterward and the electrodes, separator, and electrolyte were extracted separately. These components were used for the electrochemical measurements to be described later and for the analysis in section 2.3.

The disassembled electrodes were used to fabricate 2032 coin type cells configured as shown in Table 2^{(12), (13)}. These coin cells were then used to carry out remaining discharge capacity measurements and capacity measurements. Measurements were carried out by discharging the assembled cathode half-cells from OCV down to 2.5 V, and the anode half-cells from OCV up to 1.5 V, respectively. After that the cathode half-cells were cycled between 2.5 V and 4.2 V, and the anode half-cells between 1.5 V and 0.1 V, and these were taken as the capacity of the single electrodes, respectively. Three cells were prepared for each measurement and the average value was adopted.

The work of extracting electrodes from new batteries and aged batteries, and of incorporating the extracted electrodes in coin cells, was carried out entirely in a glove box filled with argon gas.

2.3. Analysis of Disassembled Batteries

2.3.1. Observation with scanning electron microscope and scanning transmission electron microscope

A cross section polisher (CP) was used to make cross sections of the cathodes and anodes extracted from disassembled batteries. An Ultra 55 model scanning electron microscope (SEM) made by Carl Zeiss AG was used to make observations of the cross sections. A specialized transfer vessel was then used to load the test specimens into the SEM apparatus without exposing it to the atmosphere. Separate samples were sliced off the extracted cathodes and the cross sections were observed using a JEM-ARM 200

Table 2 Coin cell configurations for electrochemical measurements

	Cathode half cell	Anode half cell	Full cell
Cathodes	Extracted cathode	Extracted anode	Extracted cathode
Anodes	Li metal	Li metal	Extracted anode
Separators	New	New	New
Electrolyte	New	New	New

model scanning transmission electron microscope (STEM) from JEOL Ltd. Samples for observation were made using a focused ion beam (FIB) apparatus.

2.3.2. Nuclear magnetic resonance spectroscopic analysis

A nuclear magnetic resonance (NMR) spectroscopic analysis of solid ^7Li was carried out for the purpose of confirming the state of lithium metal deposition on the anode. Working in a glove box, the composite layer was peeled off from a portion of the extracted anode, loaded into an NMR sample tube with an outside diameter of 4 mm, sealed, then subjected to measurement. Solid NMR measurement was performed using an AVANCE III 400 from the Bruker Corporation.

An electrolyte component analysis and anode SEI extract analysis was carried out with solution ^1H NMR. The electrolyte components extracted from the batteries were dissolved in deuterated acetonitrile (CD_3CN) and subjected to NMR measurement. For SEI component extraction, deuterium oxide (D_2O) was used to extract it from the anode composite peeled off inside a glove box. After separation of the extracted solid residue, the extract solution was subjected to NMR measurement. An ECA 400 from JEOL Ltd. was used for the solvent NMR measurement.

2.3.3. Measurement of separator air permeability

The separator extracted from disassembled batteries was washed using dimethyl carbonate inside a glove box. It was then removed from the glove box and its air permeability measured in the atmosphere. A G-B3C Gurley-type densometer from Toyo Seiki Seisaku-sho, Ltd., was used for measurement.

2.3.4. Analysis by inductively coupled plasma atomic emission spectroscopy

A quantitative analysis of lithium ion concentration in the electrolyte was carried out using inductively coupled plasma atomic emission spectroscopy (ICP-AES). A specified amount of electrolyte was measured, a mixed acid added to it, then it was heated and the electrolyte was decomposed to form a solution containing lithium

ions. The solution was diluted to a specified concentration using ultra pure water, and ICP-AES measurement was then performed. For an ICP-AES analysis of the anode composite, the composite layer peeled off inside the glove box was weighed to a specified amount, a mixed acid was added and the mixture was heated to decompose the carbon component and form a solution containing ions. The solution was diluted to a specified concentration using ultra pure water, and ICP-AES measurement was then performed. An SPS3000 from Seiko Instruments Inc., was used as the ICP-AES analysis apparatus.

2.3.5. Ion chromatography analysis

The anode composite that was peeled off in an inert atmosphere was extracted using ultra pure water, and the extract solution that remained after removal of solid residue was analyzed using ion chromatography (IC). The ions in the extract solution were qualitatively and quantitatively determined. A DX-320 ion chromatography system from Dionex Corporation was used. The column was a Dionex IonPac AS4A-SC, 4 mm in diameter and 250 mm in length. An aqueous solution of sodium bicarbonate was used as the eluent.

2.3.6. X-ray photoelectron spectroscopy analysis

The electrodes that were extracted inside a glove box were loaded using a specialized transfer vessel into an X-ray photoelectron spectroscopy (XPS) instrument and measurements were carried out. The apparatus used was a PHI Quantera SXM from ULVAC-PHI, Inc. Monochromatic $\text{Al K}_{\alpha 1,2}$ (1486.6 eV) was used as the X-ray excitation source. The X-ray beam diameter was 200 μm . The surface analysis carried out by XPS here reflects mainly information on components to a depth of approximately 2 nm from the surface.

2.3.7. Soft X-ray absorption fine structure technique

The electrodes that were extracted inside a glove box were loaded using a specialized transfer vessel into an apparatus for soft X-ray absorption fine structure measurement and measurements were made. The size of the incident light was 1 mm $\times$ 1 mm. The soft X-ray absorption fine structure technique obtains composition information on chemical compounds at different depths from the surface depending on the detection mode. The partial electron yield (PEY) mode obtains composition information on chemical compounds in a region to a depth of approximately 2 nm, the total electron yield (TEY) mode to a depth of approximately 20 nm, and the partial fluorescence yield (PFY) mode to a depth of approximately 200 nm.

2.3.8. In-depth analysis by Auger electron spectroscopy

The electrodes that were extracted inside a glove box were loaded using a specialized transfer vessel into a scanning Auger electron spectrometer (AES) and measurements were made. The apparatus used was a

SAM670 from ULVAC-PHI, Inc. The electron excitation beam had a diameter of 100 nm. Repeated measurements were made while etching the sample, yielding an in-depth analysis of changes in chemical composition.

3. Analysis Results and Discussion

3.1. Results of Capacity Fading Testing

Figure 1 shows an example of aging testing results from the batteries studied in the present research (Table 1). The graph plots the change of capacity retention against aging, with the horizontal axis showing the square root of the number of days in storage and the vertical axis showing capacity retention. The results (Table 3, Reference 1) indicate that capacity fading in storage testing does not accord with the simple square root law, or, in other words, does not fit on a single straight line. The aging rate was instead found to be divided into three different regions, shown in the figure as A, B, and C. Within each region, the capacity retention is proportional to the square root of the number of days in storage. Each region follows a square root law, but the regions are discontinuous and they could not be accounted for by a single simple square root law.

As this indicates, the aging behavior of a lithium ion battery does not follow a simple square root law. It became clear that the cause of capacity fading cannot readily be accounted for by SEI growth alone.

3.2. Results of Electrochemical Measurements

Figure 2 shows the results of electrochemical measurements^{(12), (13)}. The values shown in this figure were obtained by the methods shown in Fig. 3(a). That is, the remaining cathode discharge capacity $Q_{R,cathode}$ was determined by discharging down to 2.5 V from the terminal voltage of the assembled half-cell and the remaining anode

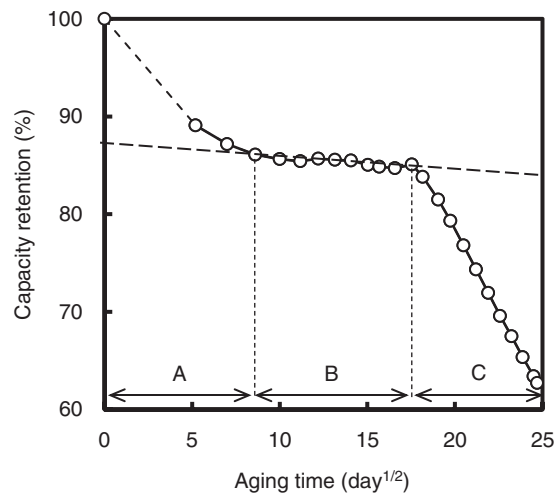


Fig. 1 Capacity retention of Li ion battery during aging (aging condition: SOC=80% at 65°C)

Table 3 Degradation analysis results

Aging phenomena	Analysis method	Results	Reference
Capacity loss	Electrochemical measurement	Observed	1
Active Li loss	ICP of electrodes and electrochemical measurement	Observed	2
Cracking	Cross section SEM observation	Not observed	3
Delamination	Cross section SEM observation	Not observed	4
Li plating	Solid state ⁷ LiNMR	Not observed	5
Loss of electrolyte	NMR, ICP of electrolyte	Not observed	6
Clogging of separator	Gurley densometer	Not observed	7

discharge capacity $Q_{R,anode}$ by discharging up to 1.5 V from the half-cell terminal voltage. These quantities were used to calculate the amount Q_{shift} of the capacity balancing shift among the electrodes by the following equation.

$$Q_{shift} = Q_{R,cathode} - Q_{R,anode} \quad (1)$$

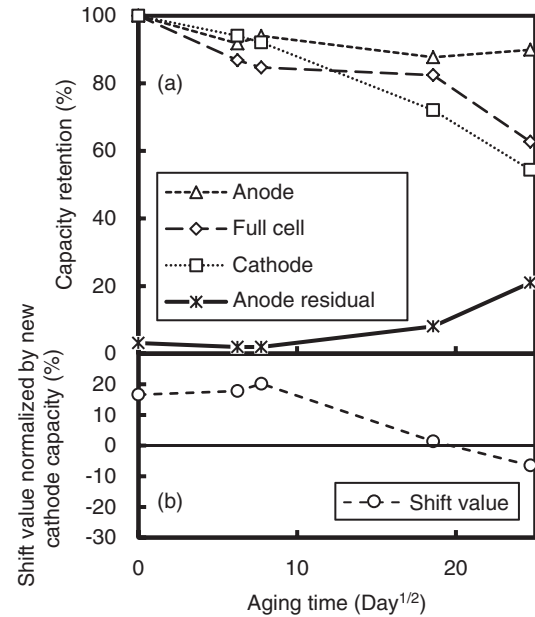
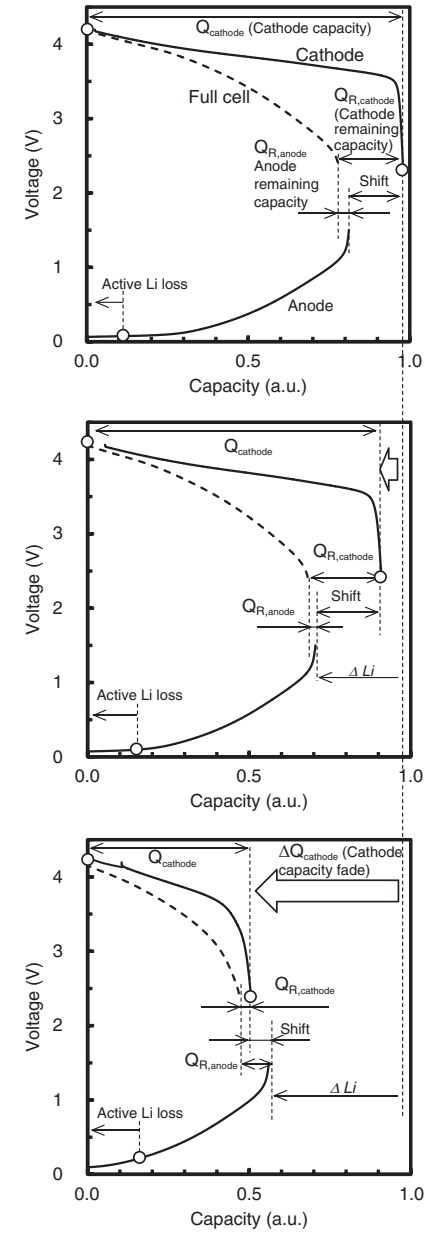
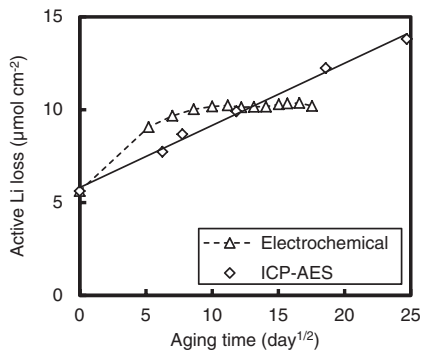


Fig. 2 Capacity retention of full cells and electrodes (a) Capacity retention (b) Shift value normalized by new cathode capacity. (aging condition: SOC=80% at 65°C) (Triangle: anode capacity retention; diamond: full cell capacity retention; square: cathode capacity retention; asterisk: residual anode capacity normalized by new anode capacity; circle: shift value normalized by new cathode capacity)



(a) Schematic drawings about active Li loss, cathode capacity fade, and capacity shift between anode and cathode.



(b) Li amount on the anode surface measured by ICP-AES (diamond) and active Li loss measured electrochemically (triangle)

Fig. 3 Active Li loss by SEI growing

The amount of the shift in the balance and the remaining anode discharge capacity were normalized by the cathode and anode capacities before aging, respectively.

As shown by the three schematic views in Fig. 3(a), the amount of active lithium loss ΔLi can be determined from the amount of the shift in the balance and the amount of cathode capacity fading using the below equation (Table 3, Reference 2).

$$\Delta Li = Q_{\text{shift}} + \Delta Q_{\text{cathode}} \quad (2)$$

Here,

$\Delta Q_{\text{cathode}}$: Amount of cathode capacity fading

Figure 3(b) shows the amount of active lithium loss derived by this method together with the ICP-AES measurement results.

3.3. Battery Materials Analysis Results

3.3.1. Observation with SEM and STEM

SEM observation of cross sections of electrodes from aged items found that neither cracking of electrode layers in both cathode and anode nor delamination from collectors had occurred (Table 3, Reference 3, Reference 4). On the anode, however, SEI formation was observed on the surface of the anode active material (white areas indicated by arrow symbols in Fig. 4).

Figure 5 shows the result from observation of cathode active material cross sections by STEM. The active material in the batteries used for the present research was a rhombohedral layered rock-salt type lithium transition metal oxide. The STEM images showed a marked contrast indicating the existence of two regions within a single crystal grain of this material. From fast Fourier transform (FFT) patterns in STEM images of both portions, spinel-like crystal structures can be observed in the transformed portions⁽¹⁷⁾⁻⁽¹⁹⁾, confirming that partial crystalline structural transformation had occurred in the active material.

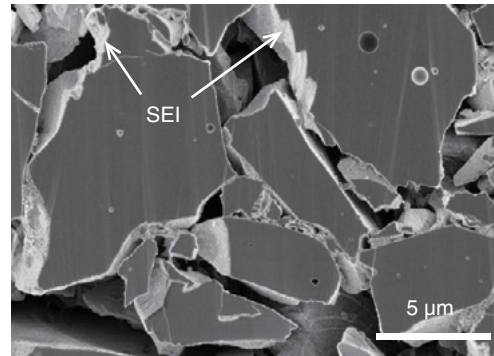


Fig. 4 Cross section SEM image of anode (arrows indicate SEI)

3.3.2. Nuclear magnetic resonance spectroscopic analysis

In the spectrum of the solid ^7Li NMR from the anode composite, only a peak in the vicinity of 0 ppm (LiCl standard) was observed. The peak in the vicinity of 0 ppm can be attributed to lithium salts. When deposition of lithium metal occurs, a peak should appear in the vicinity of 268 ppm. However, this peak portion was not detected in the batteries studied for the present research, and deposition was therefore not taken into consideration in this research (Table 3, Reference 5).

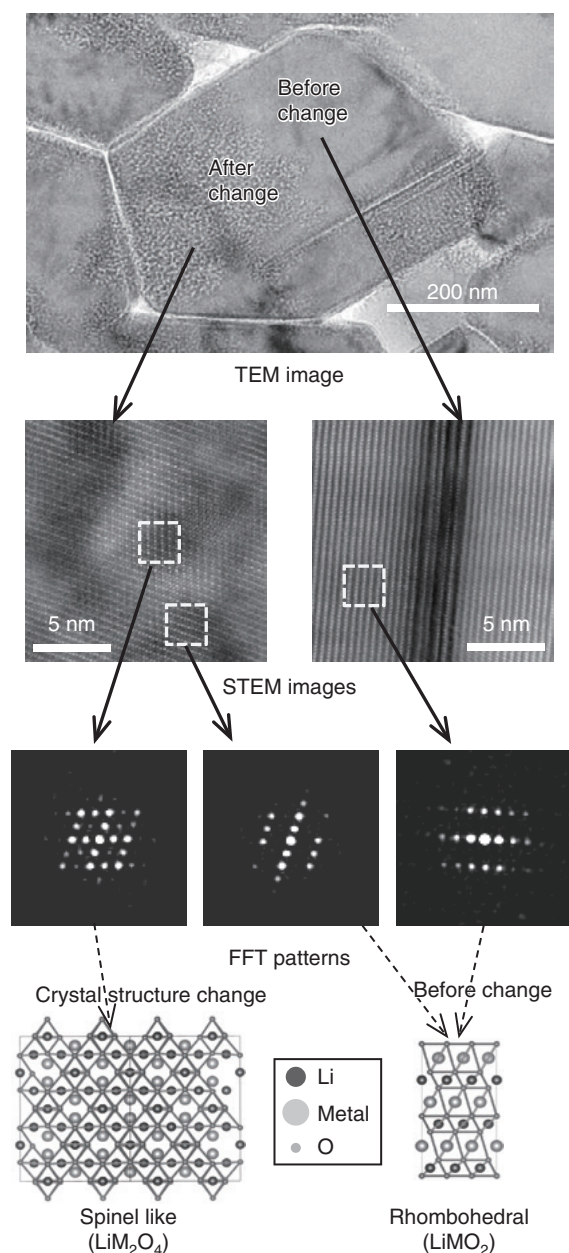


Fig. 5 Cross section TEM, STEM images and FFT patterns of cathode active material

The results from a solution ^1H NMR analysis of the extracted electrolyte solution showed that no transformation of the principal components of electrolyte occurred within the scope of the present research (Table 3, Reference 6).

The ^1H NMR results for anode deuterium oxide extracts show the detection of a small amount of trace organic components methoxy, ethoxy, glycoxy, formate ions, and acetate ions. Phosphorus compounds detected were phosphite and methyl phosphate derivatives. Alkoxy compounds are thought to be products of degradation of electrolyte solvent components, and phosphorus compounds the products of degradation of electrolyte or additives^{(15), (16)} (Table 3, Reference 2).

3.3.3. Measurement of separator air permeability

No noticeable difference was found in the air permeability of separator extracted from new batteries and from aged batteries. Therefore clogging of separator pores can be considered not to have occurred (Table 3, Reference 7).

3.3.4. ICP-AES analysis

In order to distinguish the lithium ions in the SEI from the lithium ions that exist in active material, the remaining discharge capacity of the anode was measured before carrying out ICP-AES measurement, and the amount of lithium ions remaining in the anode active material was quantified. The amount of lithium ions in the SEI was determined by subtracting the amount of lithium ions corresponding to the remaining discharge capacity [Fig. 2(a)] from the total amount of lithium ions in the anode (Table 4) obtained by ICP-AES measurement. Figure 3(b) and Table 4 show the relationship between the amount of lithium ions in the SEI and the square root of aging time. Although the amount of lithium ions agreed with the widely-known square root law^{(6), (7)} [Fig. 3(b)], it did not agree with the ΔLi value obtained by the method shown in Fig. 3(a) [Fig. 3(b)].

For ICP-AES measurement of the anode, quantifications of nickel, cobalt, and manganese were also made as shown in Table 4. The amount of the transition metals included in the SEI was less than 0.01 wt% in every case, and no increase due to storage time was observed. This therefore confirmed that deposition on the anode of transition metals originating in the cathode did not occur.

3.3.5. IC analysis

The results from an IC analysis of anode ultra pure water extracts showed that fluorine ions, oxalate ions, and phosphate ions were detected. These ions can be considered the products of degradation of electrolyte or of additives. Among them, the amount of fluorine ions increased as aging progressed. The existence of the fluorine ions can be considered to suggest the existence of LiF components in the SEI⁽²⁰⁾.

Table 4 ICP-AES results for anodes

Aging time (Day ^{1/2})	Results (wt%)			
	Li	Ni	Co	Mn
0.0	1.42	<0.01	<0.01	<0.01
6.2	1.73	<0.01	<0.01	<0.01
7.7	1.94	<0.01	<0.01	<0.01
18.6	3.01	<0.01	<0.01	<0.01
24.7	4.27	<0.01	<0.01	<0.01

Aging condition: SOC=80% at 65°C

3.3.6. X-ray photoelectron spectroscopy analysis

It was found from the XPS analysis results that, of the carbon components, carbonate and carboxylate made up from 3% to 7% of total components. Since XPS cannot readily distinguish between carbonate components and hydrocarbon components, carbonate components could not be quantified. However, it can be inferred that no major change accompanying the progressive aging of carbon components occurred down to approximately 2 nm from the surface.

3.3.7. Soft X-ray absorption fine structure analysis

Figure 6 shows the spectra of K-edge X-ray absorption near-edge structure (XANES) of carbon on the anode surface. Peaks at photon energy of 289.0 eV and 290.6 eV is attributed to the carbonyl group and carbonate, respectively⁽²²⁾. Compared to the peak of the carbonyl group at a depth down to approximately 2 nm, the carbonate peak was weaker [Fig. 6(a)]. When observing to a depth of 20 nm from the surface, however, the carbonate peak became stronger [Fig. 6(b)]. In the anode at 18.6 day^{1/2} and 24.7 day^{1/2} of storage, in particular, the peak of carbonate became greater than the peak of the carbonyl group. Information down to a depth of 200 nm from the surface mainly indicates carbon components originating primarily in anode active material [Fig. 6(c)].

Given these results, it is conceivable that more carbonate is included in regions that are close to active material than on the outermost SEI surface. Peaks at 293.0 eV and 295.5 eV can in both cases be attributed to binder C-F⁽²²⁾. Compared to new batteries, C-F peaks strength grew weaker as aging progressed. This happens because the SEI grows thicker as aging progresses, and the result here can be taken as indirect indication of that growth.

3.3.8. In-depth component analysis by Auger electron spectroscopy

Figure 7 shows the in-depth distribution of fluorine, carbon, lithium, phosphorus, and oxygen on the anode

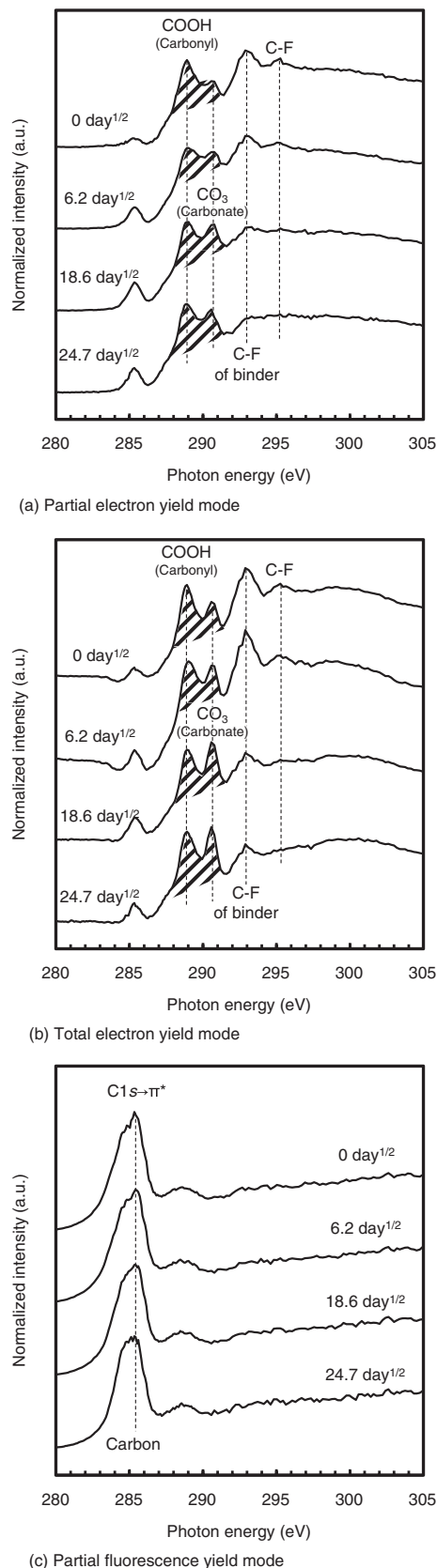


Fig. 6 Soft X-ray near edge absorption fine structure of carbon K-edge for anodes (aging condition: SOC=80% at 65°C)

surface. Since the measurement method used here involved measuring while etching the SEI surface, the accurate depth cannot readily be normalized, but a relative comparison could be made. Here, the half value of the oxygen component is defined as the SEI thickness, and the components in that portion were taken as SEI components. It is apparent from the component distribution in the interior of the active material shown in Fig. 7 that the main component of the active material was carbon, but a small amount of lithium was also included. As shown with the electrochemical measurements in Section 3.2, there was a minimal residual capacity in the extracted anode (Fig. 2), so the lithium in the active material can be considered residual lithium that was remaining in the anode when the battery was disassembled. In the SEI, however, the main

components in the layers close to the surface, as shown in Fig. 7(a), were lithium, carbon, and oxygen. Then as aging progresses, the growth of the SEI causes a reduction in the proportion of carbon in deep regions, while the amount of lithium instead increased. Considering the influence of additives, the proportion of fluorine was relatively small [Fig. 7(b)].

3.4. Discussion

3.4.1. Electrochemical measurements

As shown in Fig. 2, the capacity retention of the full cell diminishes with the passage of time in storage. The capacity retention of the anode, by contrast, diminished to about 90% and then stopped. The capacity retention of the cathode, however, diminished to 94.1% at 6.2 day^{1/2} in storage, to 92.1% at 7.7 day^{1/2}, to 72.0% at 18.6 day^{1/2}, and to 54.4% at 24.7 day^{1/2}. Meanwhile, although the amount of the shift as calculated according to Eq. (1) was 17% at 0 day^{1/2} in storage, then at 7.7 day^{1/2} it rose to 20%, at 24.7 day^{1/2} it turned back down and diminished to -6.5%. Here, when the amount of the shift in the balance is a positive value, it signifies that even in a fully discharged state, the cathode still retains some capacity to receive lithium. In other words, it indicates a state in which the amount of active lithium has diminished. When the value is negative, however, it signifies that even in a fully discharged state, there is still some lithium remaining in the anode. In other words, it indicates a state in which the cathode capacity has diminished. These results clearly showed that battery aging has two causes, the reduction in the amount of active lithium and the loss of cathode capacity.

3.4.2. Active lithium loss

As described in the previous clause, the amount of lithium ions incorporated into the SEI as determined by ICP-AES did not agree with the amount of active lithium loss determined by electrochemical measurements [Fig. 3(b)]. A portion of the SEI that is a product of reductive degradation of the electrolyte at the anode is known to be dissolved into the electrolyte and not incorporated into the SEI^{(15), (16), (20), (21)}. There have also been reports of instances in which a portion of the electrolyte components that are oxidized at the cathode is dispersed and travels to the anode, is reduced on the anode, and is incorporated into the SEI^{(20), (21)}. It is conceivable that this could be the reason why the amount of lithium included in the SEI did not agree with the amount of active lithium loss. In this case, it would appear that in order to achieve a highly accurate prediction, a value determined using Eq. (2) would need to be adopted for the amount of active lithium loss ΔLi .

Putting together the results of analyses from the previous section, the chemical composition of SEI includes a large proportion of organic lithium compounds on the surface, while the amount of inorganic components becomes greater in regions extending to a depth of more than 2 nm from the surface. As inorganic components, lithium

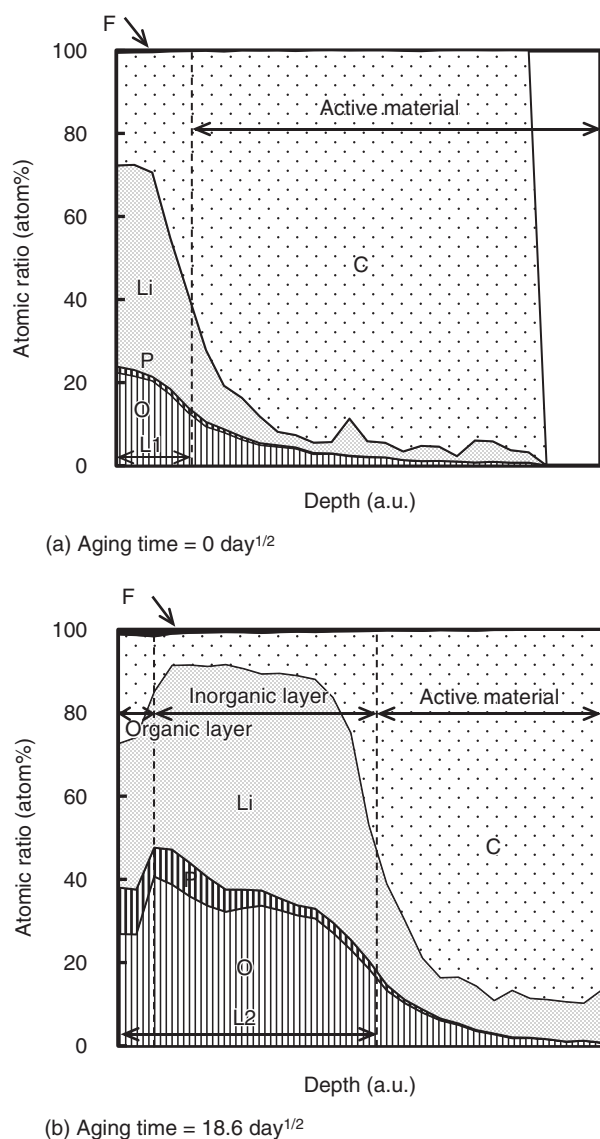


Fig. 7 AES depth profiles of anodes surfaces

carbonate, phosphate compounds, and lithium fluoride are thought to exist. The fact that the SEI takes on this kind of structure can be explained by the idea⁽²⁴⁾ that when additive components and electrolyte undergo reductive degradation at the anode, a layer containing large amounts of organic components is formed initially, and over time, the organic components transform to inorganic components.

It has also been reported⁽²²⁾ that the cause of active lithium loss is the dielectric breakdown of the SEI that occurs due to the deposition of transition metals on the anode surface. The experiment here, as related in Clause 3.3.4., confirmed that deposition did not occur. Consequently, it was decided not to take this into account in calculating the diminishment in lithium ions when creating the aging predictive equation to be covered in the next report.

3.4.3. Cathode capacity fading

It has been reported⁽¹⁷⁾⁻⁽¹⁹⁾ that in a charged state, the rhombohedral layered rock-salt type lithium transition metal oxide undergoes partial crystal structure transformation in its layered structure because of the inhomogeneous lithium distribution inside the crystal. In such cases, the parts that have transformed are thought to be a mixture of spinel-like structure expressed by the structural formula of LiM_2O_4 and rock-salt structure expressed by MO. Here, “M” represents transition metals nickel, cobalt, and manganese. As shown in Fig. 5, similar crystal structure transformation were also observed to take place in the present research in conjunction with calendar aging. Taking the case when M is manganese, for example, the amount of lithium that can be included per unit volume in a thermal equilibrium state is 42.8 mol/L in a layered rock-salt structure (LiMO_2) before aging, and after aging it is 22.2 mol/L in a spinel-like structure (LiM_2O_4)⁽²⁵⁾. The spinel-like structure contains less lithium. It is thought to be for this reason that cathode capacity diminished.

4. Conclusion

The calendar aging of lithium ion rechargeable batteries used to be predicted by a simple square root law that made aging proportional to the square root of the number of days in storage⁽⁶⁾⁻⁽⁹⁾. The foundation for this is explained to be that active lithium is lost due to the electrochemical degradation reaction of the electrolyte (SEI formation reaction) during the period of storage.

However, that aging did not follow the square root law in the prismatic high-power lithium ion rechargeable battery that was the object of the present research, and it could not be explained by a simple square root law alone. The present research, therefore, attempted to explain the mechanism of aging by a combination of electrochemical and materials analysis techniques. As a result, it was found that there are two main causes of aging in the battery, the partial crystal structure transformation in the cathode active material and the loss of active lithium due to the SEI

formation and growth reaction that occurs on the anode. Here it was also found that the quantitative results for the amount of lithium ions incorporated into the SEI as determined using an ICP-AES analysis, which is in general use, do not agree with the amount of active lithium loss as determined electrochemically. This made clear that the capacity balancing shift among the electrodes cannot be quantitatively determined from the amount of lithium ions in the SEI. Taking the findings obtained from the present research as a basis, it is expected that an equation can be created for the accurate and rapid prediction of aging in battery performance over the long term.

References

- (1) Sony History Chapter 13 Harete Kokusai Kikaku <3.5 inchi Maikuro Furoppi Disuku> Dai 3 Wa Takai Anzensei to Koenerugi, Chojumyo no Jitsugen, <https://www.sony.co.jp/SonyInfo/CorporateInfo/History/SonyHistory/2-13.html#block4>
- (2) Etacheri, V., Marom, R., Elazari, R., Salitra, G., Aurbach, D.: Challenges in the development of advanced Li-ion batteries: a review, *Energy Environ. Sci.*, Vol. 4, p. 3243–3262, (2011)
- (3) New Energy and Industrial Technology Development Organization (NEDO): Battery RM2013, (2013) (in Japanese)
- (4) Hannan, M. A., Lipu, M. S. H., Hussain, A., Mohamed, A.: A review of lithium-ion battery state of charge estimation and management system in electric vehicle applications: Challenges and recommendations, *Renewable and Sustainable Energy Reviews*, Vol. 78, p. 834-854, (2017)
- (5) Barré, A., Deguilhem, B., Grolleau, S., Gérard, M., Suard, F., Riu, D.: A review on lithium-ion battery aging mechanisms and estimations for automotive applications, *Journal of Power Sources*, Vol. 241, p. 680-689, (2013)
- (6) Peled, E.: The Electrochemical Behavior of Alkali and Alkaline Earth Metals in Nonaqueous Battery Systems-The Solid Electrolyte Interphase Model, *Journal of Electrochemical Society*, Vol. 126, p. 2047-2051, (1979)
- (7) Peled, E.: Film forming reaction at the lithium/electrolyte interface, *Journal of Power Sources*, Vol. 9, p. 253-266, (1983)
- (8) Tao, Y., Yamate, S., Ozaki, T., Inamasu, T., Yoshida, H., Okuyama, R.: Growth Process of SEI Film on Graphite Negative Electrode for Lithium-ion Secondary Battery, GS Yuasa Technical Report, Vol. 10, No. 2, p. 8-15, (2013)
- (9) Wang, D., Coignard, J., Zeng, T., Zhang, C., Saxena, S.: Quantifying electric vehicle battery degradation from driving vs. vehicle-to-grid services, *Journal of Power Sources*, Vol. 332, p. 193-203, (2016)
- (10) Saito, H., Saito, T., Murata, H., Miura, T., Sugita, Y.,

- Hirata, N., Nitta, Y.: Durability design and degradation mechanism analysis for EV battery, 2018 JSAE Annual Congress Proceedings (Spring), No. 96-18, p. 1-4, (2018) (in Japanese)
- (11) Lewerenz, M., Marongiu, A., Warnecke, A., Sauer, D. U.: Differential voltage analysis as a tool for analyzing inhomogeneous aging: A case study for $\text{LiFePO}_4/\text{Graphite}$ cylindrical cells, *Journal of Power Sources*, Vol. 368, p. 57-67, (2017)
- (12) Kobayashi, T., Shono, K., Kobayashi, Y., Miyashiro, H., Mita, Y.: Development of technique to analyze the degradation state of lithium-ion battery(I) - Discussion on cathode and battery capacity loss factor with reference electrode -, CRIEPI Research Report, Q10026, (2011) (in Japanese)
- (13) Shono, K., Ouchi, T., Kato, H., Kobayashi, T., Kobayashi, Y.: Degradation Analysis of Commercial Lithium-ion Batteries by Application of Pseudo Reference Electrode, CRIEPI Research Report, Q13006, (2014) (in Japanese)
- (14) Honkura, K.: Richiumuiondenchi no Rekkayosoku ni Muketa Hodenkyokusen no Surimoderu Kochiku, Thesis for the degree of doctor, Kyushu University, (2015) (in Japanese)
- (15) Ziv, B., Borgel, V., Aurbach, D., Kim, J., Xiao, X., Powell, B. R.: Investigation of the Reasons for Capacity Fading in Li-Ion Battery Cells, *Journal of The Electrochemical Society*, Vol. 161, p. A1672-A1680, (2014)
- (16) Aurbach, D.: Review of selected electrode-solution interactions which determine the performance of Li and Li ion batteries, *Journal of Power Sources*, Vol. 89, p. 206-218, (2000)
- (17) Nayak, P. K., Grinblat, J., Levi, M., Wu, Y., Powell, B., Aurbach, D.: TEM and Raman spectroscopy evidence of layered to spinel phase transformation in layered $\text{LiNi}_{1/3}\text{Mn}_{1/3}\text{Co}_{1/3}\text{O}_2$ upon cycling to higher Voltages, *Journal of Electroanalytical Chemistry*, Vol. 733, p. 6-19, (2014)
- (18) Hausbrand, R., Cherkashinin, G., Ehrenberg, H., Gröting, M., Albe, K., Hess, C., Jaegermann, W.: Fundamental degradation mechanisms of layered oxide Li-ion battery cathode materials: Methodology, insights and novel approaches, *Materials Science and Engineering B*, Vol. 192, p. 3-25, (2015)
- (19) Yan, P., Zheng, J., Chen, T., Luo, L., Jiang, Y., Wang, K., Sui, M., Zhang, J-G., Zhang, S., Wang, C.: Coupling of electrochemically triggered thermal and mechanical effects to aggravate failure in a layered cathode, *NATURE COMMUNICATIONS*, Vol. 9, p. 1-8 (2018)
- (20) Wang, A., Kadam, S., Li, H., Shi, S., Qi, Y.: Review on modeling of the anode solid electrolyte interphase (SEI) for lithium-ion batteries, *Computational Materials*, Vol. 15, p. 1-26, (2018)
- (21) Peled, E., Menkin, S.: Review-SEI: Past, Present and Future, *Journal of The Electrochemical Society*, Vol. 164, No. 7, p. A1703-A1719, (2017)
- (22) Nguyen, D-T., Kang, J., Nam, K-M., Paik, Y., Song, S-W.: Understanding interfacial chemistry and stability for performance improvement and fade of high-energy Li-ion battery of $\text{LiNi}_{0.5}\text{Co}_{0.2}\text{Mn}_{0.3}\text{O}_2/\text{silicon-graphite}$, *Journal of Power Sources*, Vol. 303, p. 150-158, (2016)
- (23) Cody, G. D., Ade, H., Wirick, S., Mitchell, G. D., Davis, A.: Determination of chemical-structural changes in vitrinite accompanying luminescence alteration using C-NEXAFS analysis, *Organic Geochemistry*, Vol. 28, Issues 7-8, p. 441-455, (1998)
- (24) Malmgren, S., Ciosek, K., Hahlin, M., Gustafsson, T., Gorgoi, M., Rensmo, H., Edström, K.: Comparing anode and cathode electrode/electrolyte interface composition and morphology using soft and hard X-ray photoelectron spectroscopy, *Electrochimica Acta*, Vol. 97, p. 23-32, (2013)
- (25) Nitta, N., Wu, F., Lee, J., Yushin, G.: Li-ion battery materials: present and future, *Materials Today*, Vol. 18(5), p. 252-264, (2015)

■ Author ■



Pu QIAN



Tomohiro KINOSHITA



Hiroshi SAKAI



Terumi FURUTA



Mitsumoto KAWAI

Durability Evaluation of Polymer Electrolyte Membrane for Fuel Cells by Molecular Weight Analysis

Atsushi HIRAIDE*

Akira NAKAHARA*

ABSTRACT

As molecular weight analysis of polymer electrolyte membrane for fuel cells has an advantage in sensitivity of membrane damage detection, it can detect slight changes in membrane damage even in a short-term durability test. Modelizing the relationship between the molecular weight of polymer and the probability that the polymer will react with a radical allows calculation of molecular weight reduction as an approximation to a curve. Hydrogen peroxide exposure testing was carried out on a membrane to experimentally confirm that the decrease in molecular weight due to degradation of the membrane follows the molecular weight reduction curve. It was also experimentally confirmed by a real fuel cell durability test that the reduction in molecular weight of membrane under the fuel cell operating conditions follows the molecular weight reduction curve. With the molecular weight reduction curve, it becomes possible to predict the degree of degradation in long term durability tests from degradation in short term tests, thereby shortening the test time necessary for evaluating the durability of the polymer electrolyte membrane to around one fifth or less.

1. Introduction

A fuel cell is a power source characterized by the use of electrochemical reactions to generate power without burning fuel. The advantages are that energy conversion efficiency is good because there are no waste heat losses, and that there are few reaction by-products because the reaction does not reach high temperatures. If hydrogen is used as fuel, power is generated by the reaction between hydrogen and the atmospheric oxygen, and water is produced as the reaction product (Fig. 1). Since fuel cells produce energy with a low environmental burden, fuel cells are increasingly being used as the power source for electric vehicles.

Reducing size and cost are requirements in the fuel cells developed for vehicles, and it is also important to optimize designs to balance those needs against durability, which is in a tradeoff relationship with them. However, it has been necessary to perform long-term testing that takes from several months to a year, in order to evaluate the durability of fuel cells. That has previously been a factor lengthening the development period for fuel cell vehicles.

Power generation in a fuel cell occurs in the membrane electrode assembly (MEA) (Fig. 1). It is generally known that the polymer electrolyte membrane (PEM), which is interposed between the anode electrode and the cathode

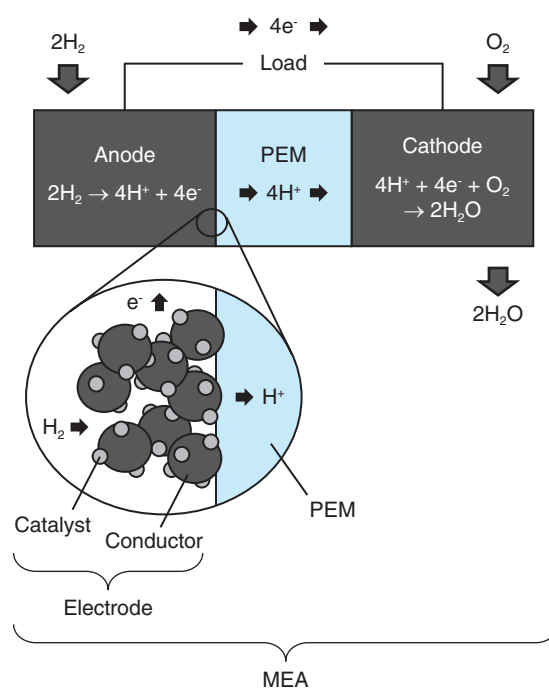


Fig. 1 Schematic cross section of typical fuel cell

* Automobile R&D Center

electrode, gradually degrades during the operation of the fuel cell. It is necessary to design fuel cells with sufficient durability, but the PEM is generally manufactured to be just a few tens of μm thick, in order to enhance fuel cell power generation performance. The main PEM material now is perfluorosulfonic acid (PFSA), a fluorine resin electrolyte which features a polytetrafluoroethylene backbone ($-\text{CF}_2-$), known as Teflon®. Figure 2 shows the chemical structure of Nafion®, a typical PFSA polymer molecule. The degradation mechanisms of PFSA, as shown in Fig. 3⁽¹⁾, are thought to involve two reactions: Polymer scission (indicated by the green scissors in Fig. 3), and the unzipping reaction (indicated by the blue arrows in Fig. 3), in which breakdown proceeds from the carboxyl groups ($-\text{COOH}$) at the ends of the polymer molecule. Hydroxyl radicals are thought to be the causative agent of both reaction mechanisms, and when scission of PFSA polymer molecules occurs, the new ends are carboxyl radicals, which become new starting points of the unzipping reaction⁽²⁾,⁽³⁾. The chemical reactions in the unzipping reaction are as stated below⁽²⁾.

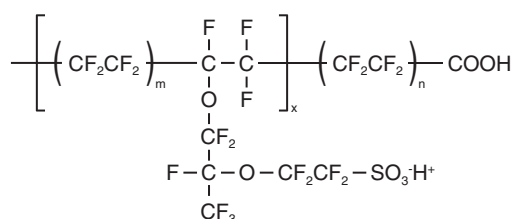
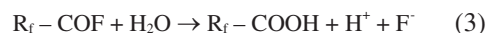
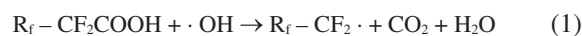


Fig. 2 Chemical structure of Nafion®

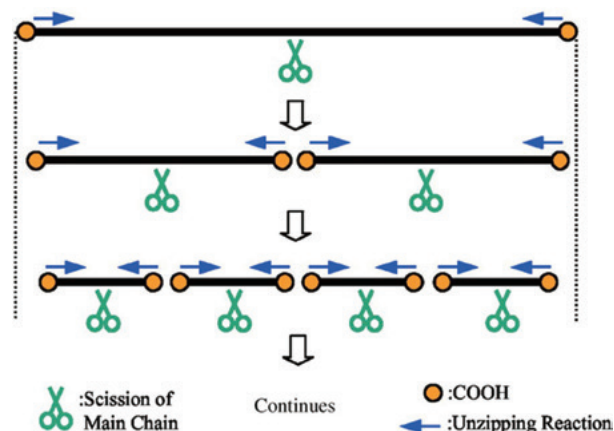


Fig. 3 Degradation mechanism of PFSA polymer

The PEM thickness should decrease during the unzipping reaction, because the carbon and fluorine, which are constituents of PFSA, are emitted from the PEM in the forms of CO_2 and fluoride ions⁽⁴⁾. The degradation rate of PFSA can be identified by measuring the quantity of emitted fluoride ions at that time, and that is termed “fluoride release.”

Before 2016, no method had been established for quantitative evaluation of the durability of the PEM made from PFSA. Therefore, to ascertain the PEM lifetime, it was necessary to perform long-term durability tests running until the end of the PEM lifetime. As reported previously⁽⁵⁾ on this subject, a method for PEM lifetime prediction has been developed that focuses on the behavior of increasing fluoride release. The starting points of unzipping reactions should increase exponentially, because two new ends are created when the PFSA polymer chain scission reaction occurs as shown in Fig. 3. A method that can calculate a fluoride release that increases exponentially was established by modeling the two reaction mechanisms of PFSA degradation, and mathematizing the amount of fluoride release as a function of durability time (Fig. 4). As a result, the degradation rate can be evaluated quantitatively, before the PEM reaches the end of its lifetime and even during the durability test, based on the behavior of increasing fluoride release. However, a long-term durability test of at least several months was still required to identify PEM degradation with adequate sensitivity, because the behavior of a fluoride release that increases exponentially shows little variation in the early stages of the durability test.

Therefore, a new evaluation method was developed as a substitute for fluoride release that was able to detect PEM degradation with high sensitivity even during a short durability test period, in order to evaluate PEM durability in a shorter period.

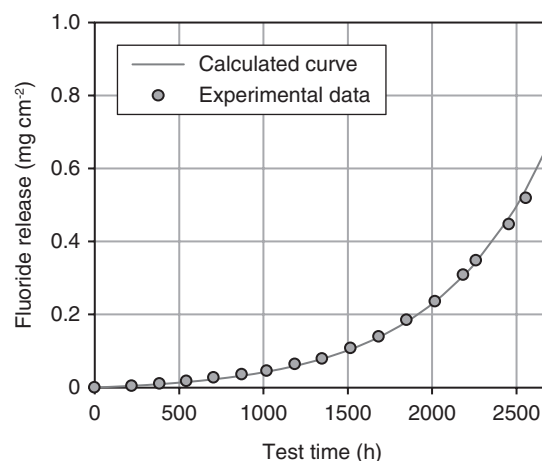


Fig. 4 Non-linear behavior of fluoride release in durability test

2. Modelization of Molecular Weight Reduction

In the degradation mechanism shown in Fig. 3, focusing on one polymer molecule, the number of polymer end groups double, and the molecular weight that represents the length of the polymer molecule is reduced to around half after one scission reaction occurs. Therefore, the amount of fluoride release increase from polymer end groups is small in the initial stage, but that growth increases exponentially with the passage of time. In contrast, the behavior of molecular weight decreasing could show exponential decay, similar to a half-life, which would mean a molecular weight decrease significantly at the initial stage, and the amount of the decrease becomes smaller with the passage of time. Focusing on this point, analyzing molecular weight should be better than analyzing fluoride release in order to detect PFSA degradation with high-sensitivity in a short term, because its variation is greater in the initial stages of PFSA degradation.

As a theoretical approach, the first step was modeling the reduction of molecular weight, in order to learn how molecular weight is reduced by polymer molecule degradation. The Monte Carlo simulation was used to modelize polymer molecular degradation due to a chain scission reaction caused by random reactions with hydroxyl radicals, in order to create an equation that can calculate the molecular weight reduction rate. The following assumptions were employed for modelization. For the two reaction mechanisms shown in Fig. 3, a single scission reaction between the polymer molecules causes a major reduction in molecular weight, while the molecular weight reduction due to the unzipping reaction, as shown in Eqs. (1), (2) and (3), is very slight. Therefore, only the scission reaction is considered in the model to simplify this study. The gel permeation chromatography method was used for molecular weight measurement. Regarding the principle that separating polymer molecules based on the size of polymer molecules in solvent, the measurement results mainly reflect the main chain length of the polymer molecule, and are affected little by the presence of side chains. Therefore, the model assumes that polymer molecules are linear, with no branching, and that the molecular weight is proportional to the length of the polymer molecule.

Next, a scission reaction that occurs through a reaction with hydroxyl radicals was modeled. As the reactivity of hydroxyl radicals is extraordinarily high, the model assumes that scission occurs at the point where a hydroxyl radical comes into contact with a polymer molecule. According to the lattice model by Flory⁽⁶⁾ and Huggis⁽⁷⁾, the volume that is occupied by a polymer molecule is proportional to its molecular weight, so the probability of contact between hydroxyl radicals diffusing randomly in space and any given polymer molecule is thought to be proportional to the molecular weight. Therefore, the probability of the scission reaction is assumed to be proportional to

its molecular weight, which is to say, to its length in the simplified model. Figure 5 shows the algorithm for computer implementation. Firstly, as shown in Fig. 5(a), the initial lengths and number of polymer molecules to be handled by the model was determined from previously measured PEM molecular weight distribution information. To simplify representation, the distribution of molecular weight was plotted as a histogram with the molecular weight range 10^3 g/mol to 10^7 g/mol divided into 16 equal intervals on the logarithmic axis, then with the number of polymer molecules in each interval determined. Next, as shown in Fig. 5(b), the aggregate of all polymer molecules with lengths and numbers determined was joined into a single chain. Then the scission points were determined by generating uniform random numbers against that total length. The scission points determined by random numbers were indicated by green scissor symbols in Fig. 5(c). This method expressed the probability of contact and reaction between randomly diffusing hydroxyl radicals and each polymer molecule, which is to say, it expresses the fact that probability of scission of a polymer molecule is proportional to its length. Finally, as shown in Fig. 5(d), the lengths and numbers of polymer molecules after scission are plotted in a histogram, to obtain the molecular weight distribution after degradation. Figure 6 presents calculation results produced by the created molecular weight reduction model. The scission count N is indicated as the number of scissions per molecule when averaged by the original number of polymer molecules.

$$N = (\text{scission count}) / (\text{original number of polymer molecules}) \quad (4)$$

Figure 7 presents the relationship between scission count N and molecular weight reduction. The vertical axis expresses the molecular weight retention rate, relative to the initial molecular weight as 100%. Calculating with $N=1$

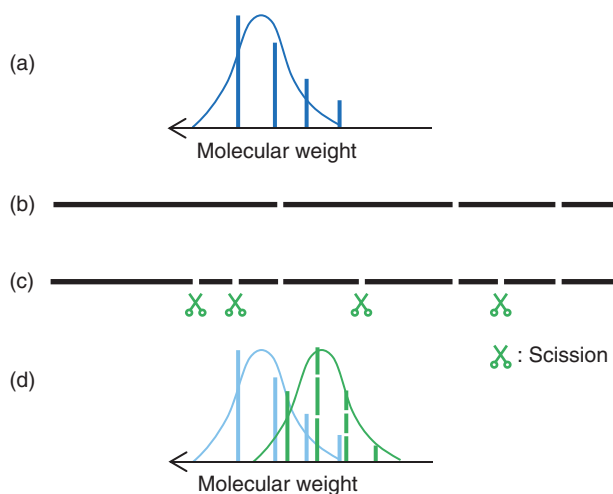


Fig. 5 Modelization of random polymer scission and molecular weight reduction

means that all polymer molecules have one scission, on average, so the molecular weight retention rate is around 50%, approximately half.

From the above results, it was anticipated from the created molecular weight reduction model that when a polymer material is degraded at a constant rate, the average molecular weight would decline along the curve shown in Fig. 7. Therefore, this curve was named the molecular weight reduction curve. The horizontal axis shows the polymer molecule scission count, so in a durability test in which the generation rate of hydroxyl radicals is constant, this curve can be used by replacing the horizontal axis with test time.

In Fig. 4, the amount of fluoride release followed exponential growth, so the initial change was the smallest, whereas the molecular weight reduction curve shown in Fig. 7 follows exponential decay, so the initial change is the largest. Therefore, it can be stated that measuring molecular weight rather than fluoride release is a more suitable method

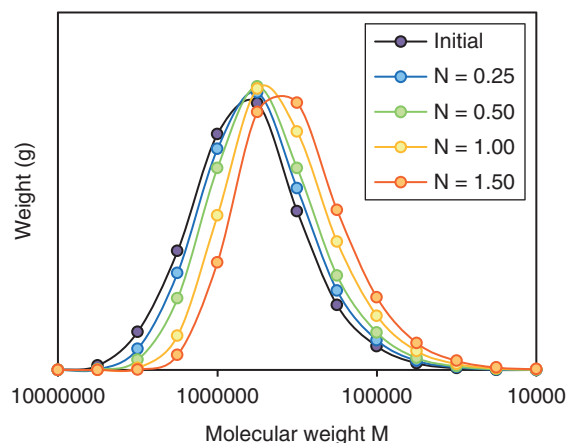


Fig. 6 Calculated shift of molecular weight distribution using random polymer scission model

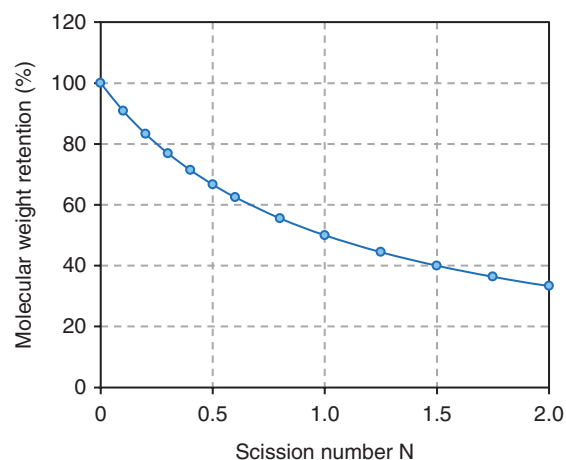


Fig. 7 Calculated molecular weight reduction curve using random polymer scission model

for high-sensitivity detection of material degradation in an early diagnosis.

3. Experiments and Observations

In Chapter 2, the molecular weight reduction when polymer material is degraded at a fixed speed was modeled and the molecular weight reduction curve was obtained. In the process of creating the model, some assumptions were set for the purpose of simplification, and the accuracy of the created model is validated experimentally in this Chapter 3.

3.1. Validation of the Model in Single-material Testing

This model was created by simplifying the structures and reactions of polymer molecules, but the PFSA polymer molecules used in the PEM for actual fuel cells differ from the simplified model in the following three points. (1) The model assumes that the polymer molecules are linear and the scission reactions occur at random positions, but actually they have a main chain structure as shown in Fig. 2, so the positions where the scissions occur could be limited to certain positions⁽³⁾. (2) The model assumes that the polymer molecules are linear with no branching, but actually they have side chains as shown in Fig. 2. (3) The model only considered polymer molecule scission, but actually polymer molecules are degraded by two reaction mechanisms, scission and unzipping. Therefore, to confirm the accuracy of the model, the question of whether molecular weight reduction of PFSA polymer molecules follows the molecular weight reduction curve was validated experimentally in a test that causes PEM degradation in isolation. If the experimental molecular weight reduction follows the molecular weight reduction curve, even a model that has been simplified in the three points stated above can be said to be practically applicable to the prediction of molecular weight reduction in PFSA polymer molecules.

The test method was a hydrogen peroxide exposure test⁽⁸⁾. The hydrogen peroxide exposure test is a test that simulates the environment in which PEM is degraded during fuel cell operation, in order to evaluate PEM durability. The PEM is exposed to steam containing hydrogen peroxide at a constant concentration, so that it is exposed to hydroxyl radicals supplied at a constant rate. Provided the generation rate of hydroxyl radical is constant, the number of reactions degrading the PEM is thought to be proportional to the exposure time. In order to confirm that the relationship between exposure time and molecular weight reduction in the hydrogen peroxide exposure test follows the molecular weight reduction curve in Fig. 7, samples were prepared at five levels of varied exposure time, and their molecular weights were measured. The conditions for the exposure test were set at a temperature of 90°C and with 30% humidity, based on operation conditions expected for fuel cells.

First, the water used for washing the PEM after the test was analyzed with ion chromatography, confirming the generation of fluorine ions, which confirmed that the

unzipping reaction occurred in the hydrogen peroxide exposure test. Figures 8 and 9 present the molecular weight measurement results. Figure 8 shows the molecular weight distribution after testing for each exposure period. The vertical axis shows the refractive index detector output of the molecular weight measurement instrument. The refractive index of the polymer molecule solution is known to be proportional to the mass concentration of the solute. Similar to the calculation results from the model shown in Fig. 6, it has been confirmed that the peak of the molecular weight distribution maintains its shape as it shifts to the low molecular weight side. Also, Fig. 9 plots the average molecular weight after the test for each exposure time, with the horizontal axis adjusted, superimposed on the molecular weight reduction curve from Fig. 7. The relationship between exposure time in the hydrogen peroxide exposure

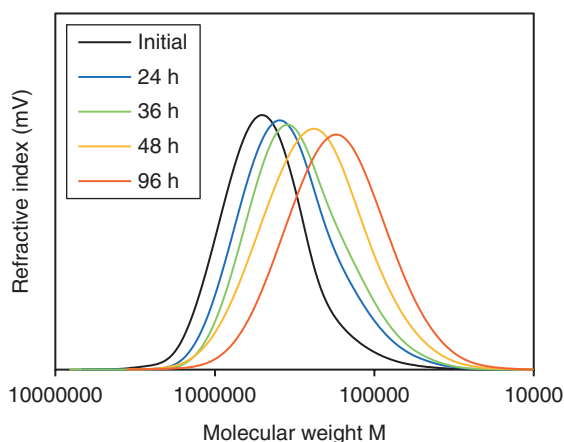


Fig. 8 Membrane degradation detected by shift of molecular weight distribution in hydrogen peroxide exposure test

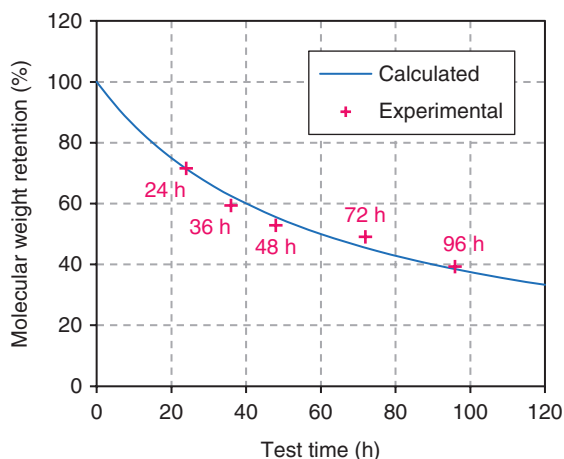


Fig. 9 Relationship between exposure time and molecular weight reduction follows model-calculated curve

test and molecular weight reduction matched the molecular weight reduction curve, underlining the correctness of the three assumptions stated above.

- (1) Even in a PFSA polymer molecule with a main chain structure as shown in Fig. 2, the positions at which scission of the polymer molecule occurs are distributed with a level of evenness such that positions may reasonably be assumed to be random.
- (2) Even in a PFSA polymer molecule having side chains as shown in Fig. 2, the probability of the scission reaction of a polymer molecule is proportional to its length, and by extension, to its molecular weight.
- (3) Even for PFSA polymer molecules degraded by the two reaction mechanisms of scission and unzipping reaction, the reduction of molecular weight follows the molecular weight reduction curve, which only consider scission reactions because the influence of unzipping on molecular weight reduction is smaller than that of a scission reaction.

From the above results and observations, it was confirmed that when a PFSA polymer molecule is degraded at a constant rate, the reduction of its molecular weight follows the molecular weight reduction curve.

3.2. Validation of the Model in a Fuel Cell Operation Environment

In the preceding section, it was confirmed in the hydrogen peroxide exposure test on the PEM alone that molecular weight reduction due to degradation of PFSA follows the molecular weight reduction curve. However, in a real fuel cell operation environment, the following three points differ from the hydrogen peroxide exposure test on the PEM alone: (1) In an MEA which electrodes including catalysts are placed on both sides of the PEM, there is the possibility that multiple side reactions, other than the main reaction shown in Fig. 1, could contribute to membrane degradation⁽³⁾. (2) Changing the electrical potential during power generation could influence the reaction rate and reaction pathway, so that the two reaction mechanisms of polymer molecule scission and unzipping reaction may occur in different ratios from the hydrogen peroxide exposure test performed on PEM alone. (3) There is the possibility that the generation rate of hydroxyl radicals may not be constant, due to factors such as degradation of the constituent materials of the MEA.

Therefore, the question of whether molecular weight reduction of PFSA polymer molecules follows the molecular weight reduction curve, even in PEM degradation during real fuel cell operation, was validated experimentally in a fuel cell durability test. If molecular weight reduction during a power generation test follows the molecular weight reduction curve, the modeled molecular weight reduction curve can be said to be practically applicable to the prediction of molecular weight reduction in fuel cell operation. The durability test has a further objective, to experimentally associate the number of scission reactions

of the polymer molecule on the horizontal axis of the molecular weight reduction curve shown in Fig. 7 with the operation time of the fuel cell. As durability testing of fuel cell materials requires a long period, durability testing is generally performed under accelerated stress⁽³⁾. In this case, the accelerated test was performed at a temperature of 90°C and relative humidity of 30%. These are severe conditions for chemical degradation within the range of operational environment conditions expected for fuel cells^{(3), (4)}. In order to confirm that the relationship between fuel cell operation time and PEM molecular weight reduction follows the molecular weight reduction curve in Fig. 7, durability tests were performed at five levels of varied operation time, and the molecular weight of PEM was measured after the end of the durability test.

The results are as shown in Figs. 10 and 11. Figure

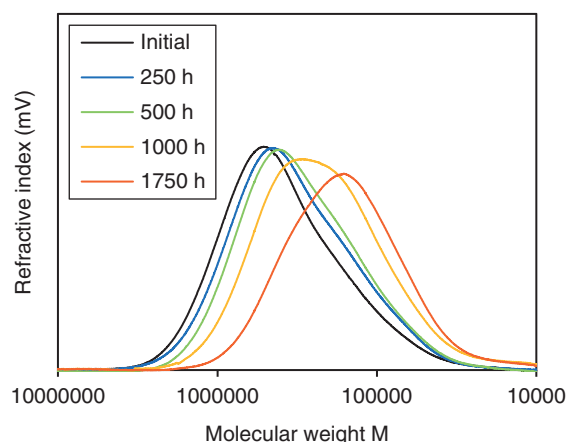


Fig. 10 Membrane degradation detected by shift of molecular weight distribution in real fuel cell test

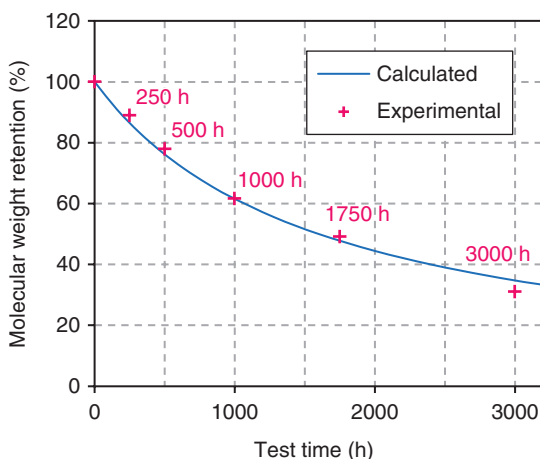


Fig. 11 Relationship between accelerated test time and molecular weight reduction follows model-calculated curve

10 shows the molecular weight distribution after testing for each durability period. Similar to the calculation results from the model shown in Fig. 6, it has been confirmed that the peak of the molecular weight distribution largely maintains its shape as it shifts to the low molecular weight side. Also, Fig. 11 plots the average molecular weight after the test for each durability time, with the horizontal axis adjusted, superimposed on the molecular weight reduction curve from Fig. 7. As the relationship between fuel cell operation time and molecular weight reduction matches the molecular weight reduction curve, it is confirmed that molecular weight reduction in PFSA follows the molecular weight reduction curve in PEM degradation occurring in the real fuel cell operation environment. This result also supports the constant-rate generation of hydroxyl radicals during fuel cell operation under constant conditions. Furthermore, based on the modeled molecular weight reduction curve, it was possible to predict the amount of PEM degradation in a long-term durability test by analyzing the molecular weight reduction of the PEM in a short-term durability test.

Figure 12 is a graphical representation comparing the molecular weight reduction curve with membrane thickness reduction and fluoride release behavior. In conventional methods for measuring membrane thickness and fluoride release, the amount of degradation exhibits exponential growth, so a long-term durability test was necessary before degradation could be detected. With the new focus on molecular weight reduction in this research, it is now possible to detect PEM degradation with high sensitivity, even in a short-term durability test in which almost no change appears in membrane thickness or fluoride release. Specifically, if an approximately 10% change is needed to detect PEM degradation accurately in both membrane

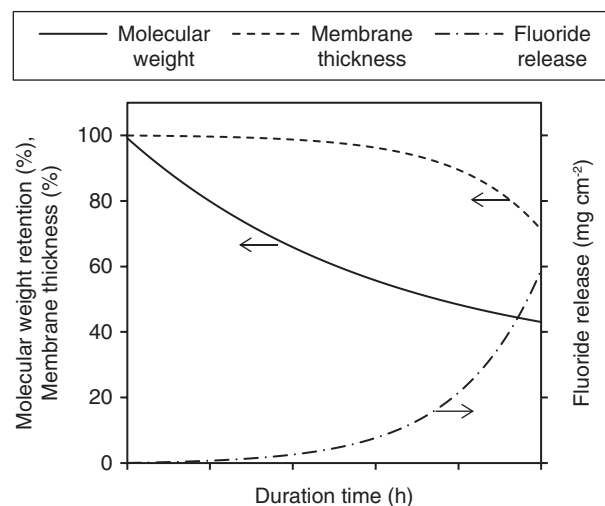


Fig. 12 Non-linear behavior of molecular weight reduction, membrane thickness loss, and fluoride release in durability test

thickness and molecular weight in Fig. 12, the required durability time can be shortened to one fifth or less using a modeled molecular weight reduction curve. This has made it possible to shorten the time of durability testing required to detect PEM degradation accurately to a period of several weeks, rather than the several months or more previously.

4. Conclusion

Until now, the development of fuel cells has faced the challenge that evaluation of PEM durability required long-term durability testing for at least several months. By focusing on molecular weight reduction in PFSA polymer molecules, this research has found that PEM degradation can be detected with high sensitivity, even in short-term durability testing. By modeling the relationship between molecular weight and the probability of a polymer molecule reacting with a radical in a polymer degradation reaction, this research has predicted the reduction of molecular weight through calculation, and created a molecular weight reduction curve for polymer materials. The accuracy of the created model has been validated by experimental confirmation that molecular weight reduction in PFSA polymer molecules used in the PEM follows the molecular weight reduction curve, and molecular weight reduction of the PEM occurring in a fuel cell operation environment also follows the molecular weight reduction curve. Creating and using the molecular weight reduction curve has made it possible to predict the degradation amount in long-term durability tests based on the molecular weight analysis result for the PEM after short-term durability tests, achieving as a result a reduction of the test period required for PEM durability assessment to around one fifth or less.

References

- (1) Hommura, S., Kawahara, K., Shimohira, T., Teraoka, Y.: Development of a Method for Clarifying the Perfluorosulfonated Membrane Degradation Mechanism in a Fuel Cell Environment, *J. Electrochem. Soc.*, 155(1), p. A29-A33, (2008)
- (2) Curtin, D. E., Lousenberg, R. D., Henry, T. J., Tangeman, P. C., Tisack, M. E.: Advanced Materials for Improved PEMFC Performance and Life, *J. Power Sources*, 131, p. 41-48, (2004)
- (3) Coms, F. D.: The Chemistry of Fuel Cell Membrane Chemical Degradation, *ECS Transactions*, 16 (2), p. 235-255, (2008)
- (4) Li, J., Wang, K., Yang, Y.: Membrane Durability and Degradation under Dry Fuel Cell Operation Conditions, *ECS Transactions*, 25 (1), p. 385-394, (2009)
- (5) Hiraide, A: Life Prediction Model Estimating Chemical Degradation of Polymer Electrolyte Membrane for Fuel Cells, *Honda R&D Technical Review*, Vol. 28, No. 1, p. 50-58
- (6) Flory, P. J.: *Thermodynamics of High Polymer Solutions*, *J. Chem. Phys.*, 10, p. 51-61, (1942)
- (7) Huggins, M. L.: Some Properties of Solutions of Long-Chain Compounds, *J. Phys. Chem.*, 46(1), p. 151-158, (1942)
- (8) Hommura, S., Kawahara, K., Shimohira, T.: Degradation Mechanism of Perfluorinated Membrane in Polymer Electrolyte Membrane Fuel Cell Environment, *Polymer Preprints, Japan* 54(2), p. 4517-4518, (2005)

■ Author ■



Atsushi HIRAIDE



Akira NAKAHARA

Technique for Simulating Vehicle Handling in Very Low Speed Range for Model Based Development

Takayuki TOYOSHIMA*

Yoshitomo AZEKATSU*

Yoshiaki KOBAYASHI*

Naofumi IKEDA*

Hiroki TAKANO*

Isao OBATA*

ABSTRACT

A conventional performance design simulation model, used in the development of mass production vehicles, was taken as the base to which a steering system model was newly added on to create a technique capable of simulating vehicle handling at very low speeds by performing a bulk calculation of those indices together during the initial stages of development. As a result, there is no longer a need to create separate simulation models for each index, and the efficiency of performance design has been enhanced. Formerly, the performance index of steering return torque could not readily be examined by means of simulation during the initial stages of development, but this can now be estimated seamlessly from high speeds to very low speeds by changing the vehicle speed condition. This technique is effective for performance design during the initial stages of development, including for investigation of specifications to achieve a balance between vehicle handling at the very low speed range and handling stability at the high speed range, for analysis of factors in torque around the king pin axis, and other such investigation. It has been applied to the development of chassis platforms for compact vehicles that were adopted, starting with the 2014 model year FIT.

1. Introduction

In recent years, model based development (MBD) has become an effective means of raising the efficiency of automobile development, which is growing increasingly complex^{(1), (2)}. Figure 1 shows the development process for mass-production vehicles following the V-model. MBD is a development method that uses simulation to advance the examination of specifications at the stages of function design, performance design, and so on, in the area at the bottom left of Fig. 1 shown enclosed by a red border.

In MBD for mass-production vehicles, it is important to calculate a variety of characteristic values within a limited time and to create an environment that allows information to be shared promptly⁽³⁾. This approach is also called on to respond to requests for investigation of performance design items that formerly were not readily subject to calculation or measurement.

One dynamic performance design item in the high speed range that has had to be created along the way is the model for prediction of lap time⁽⁴⁾ and another is the model that can compute predicted crosswind stability⁽⁵⁾, which is an item of performance robustness with regard to external disturbance. These and other such items were part of the measures taken

to expand the MBD domain. Recent years have also brought increasing request to investigate vehicle handling in the very low speed range that supports automatic parking functions and autonomous driving functions (hereafter very low speed range vehicle handling). Figure 2 shows the vehicle speed ranges covered in vehicle dynamics simulations created to date for the initial stages of development. Conventional

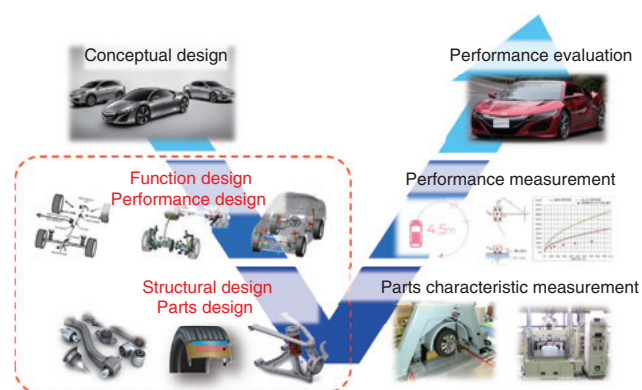


Fig. 1 Positioning of MBD in V-Model

* Automobile R&D Center

simulation covered vehicle speeds ranging from tens of kilometers per hour to hundreds of kilometers per hour, but did not support study of the very low speed range. The reason for this is that conventional simulation for use in MBD linearized the characteristics of vehicles in a driving state and approximated them with a laterally symmetrical model in order to make calculation more efficient.

In scenarios that have a vehicle driven with its steering wheel turned sharply to one side, however, there is an issue with conventional models because they could not increase the accuracy of their calculation due to the influence of the non-linear characteristics and the lateral asymmetry of the steering system. Consequently, vehicle dynamics characteristics that take into account non-linearity and asymmetry when driving in the very low speed range were investigated using full vehicle simulation. Figure 3 shows a conceptual image of a full vehicle simulation model. This approach has to model systems at the level of individual

components, and such a model could not readily be created with the limited information available during the initial stages of development. This was not suited as a predictive model for use in MBD of mass-production vehicles in the initial stages of development.

There have been instances of research in predictive models that are limited to very low speed ranges⁽⁶⁾, but there have not been many in simulation models for use in MBD. Therefore a new MBD simulation model was created that can perform bulk calculation of dynamic performance across a wide range of vehicle speeds from very low speed to high speed using only the information available during initial development.

2. Aims

With vehicle speeds of 10 km/h or less defined as the very low speed range, the following were the objectives set for a simulation model that would enable investigation of dynamic performance from very low speed to high speed under the same environment.

- (1) It is a simulation model capable of prediction using only information available in the initial stages of development.
- (2) It is a model capable of simultaneously predicting characteristic values for vehicle handling in a very low speed range and characteristic values for handling stability when driving at high speed.
- (3) It is a model capable of outputting prediction results with accuracy and efficiency that are usable in the development of mass-production vehicles.

3. Vehicle Dynamics Characteristics at the Very Low Speed Range

Research on vehicle handling at the very low speed range has been conducted from a variety of angles⁽⁷⁾⁻⁽¹⁰⁾. Vehicle dynamics characteristics at the very low speed range are closely related to vehicle handling in such scenarios as left and right turns at intersections, U-turns, and turning one direction and then another when parking. Indices of vehicle handling at the very low speed range can be classified under vehicle maneuverability and torque characteristics generated in the steering wheel. A prime example of the former type of index is minimum turning radius, and of the latter type of index is steering return torque. These two indices were taken as the objects of evaluation for the present research.

3.1. Minimum Turning Radius

Minimum turning radius is defined as the radius of the trajectory drawn by the tire ground contact center when the vehicle is moving forward slowly with the steering wheel turned all the way in one direction^{(11), (12)}. Figure 4 shows an overview. The four wheels all describe a different radius on the turn trajectory, but for this paper, the radius B of

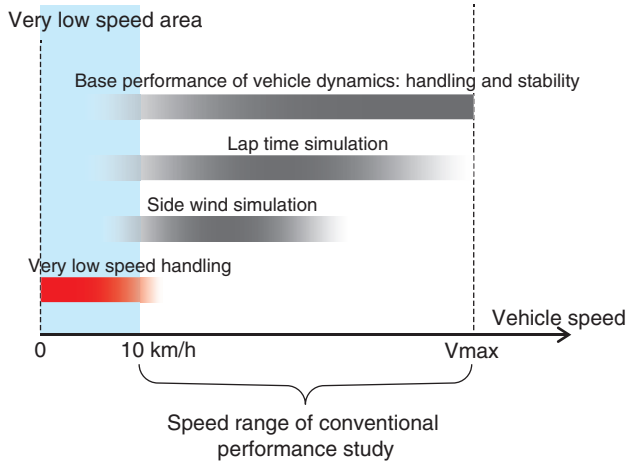


Fig. 2 Domain of performance design

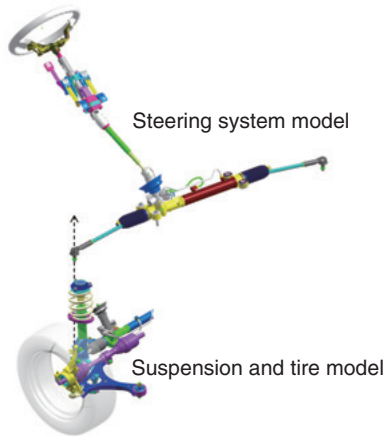


Fig. 3 Image of full vehicle simulation model

the trajectory drawn by the tire ground contact center on the outside end of the front axis is taken as the minimum turning radius. When a vehicle that is making a constant turn with a fixed steering angle gradually reduces its speed, ordinarily the turning radius also grows smaller, as shown in Fig. 5. The turning radius becomes smallest when the vehicle speed approaches zero while turning with the sharpest steering angle, and in this paper, that is taken as the minimum turning radius R_{min} .

3.2. Steering Return Torque

When the steering wheel is turned, the torque that works to return the wheel to the straight ahead state is called the steering return torque (M_{Steer}). The solid line in Fig. 6 represents M_{Steer} with respect to the steering wheel angle (A_{Steer}). Depending upon whether the steering wheel is turned farther toward the turn or turned back toward the center, the friction (M^*) is added to M_{Steer} in the opposite direction, respectively. Consequently, M_{Steer} with respect to A_{Steer} forms the Lissajous figure of an S-curve with a width shown by the dotted lines on the graph.

Figure 7 shows a simplified image of a typical

steering system. When M^* elements are excluded, M_{Steer} is proportional to the input of the resultant force (F_{Rack}) from the left and right of the rack, while A_{Steer} is proportional to the rack stroke (S_{Rack}), and their relationship is described by Eq. (1).

$$M_{Steer} / A_{Steer} = F_{Rack} / S_{Rack} \quad (1)$$

According to Eq. (1), the A_{Steer} - M_{Steer} characteristic and S_{Rack} - F_{Rack} characteristic will form similar figures. For the present simulation, therefore, the relationship between A_{Steer} and M_{Steer} is replaced with the relationship between S_{Rack} and F_{Rack} , based on this principle, and the aim is made to calculate the S_{Rack} - F_{Rack} characteristic.

With regard to steering return characteristics, it intuitively makes sense for F_{Rack} to increase uniformly with respect to S_{Rack} . Even when assist control from a motor or other such source is taken into consideration, it is more advantageous for it to change uniformly than to have it change by reversing its polarity at unspecified points⁽¹³⁾⁻⁽¹⁵⁾. In terms of ease in returning the steering wheel to the straight forward state, it is also desirable for the steering

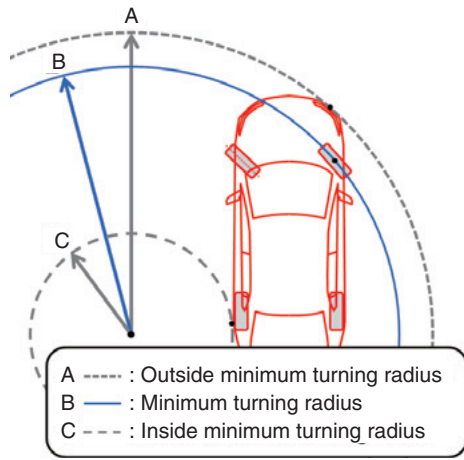


Fig. 4 Scheme of minimum turning radiuses

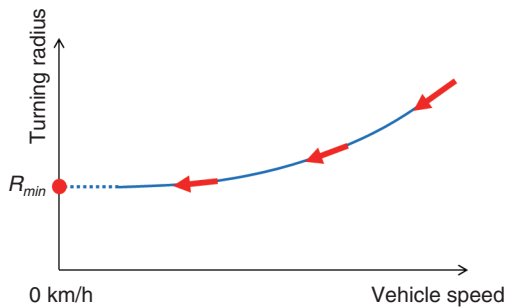


Fig. 5 Turning radius with respect to vehicle speed

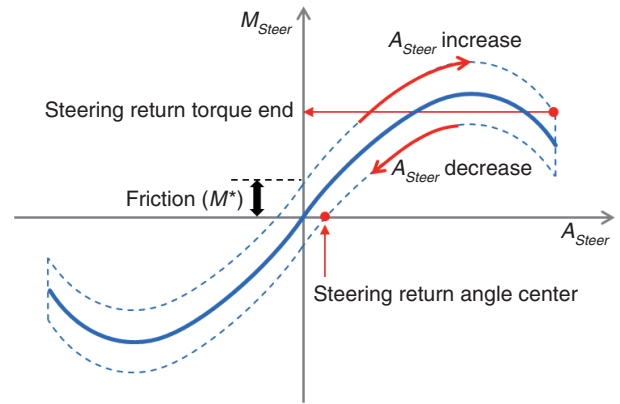


Fig. 6 Steering return torque (M_{Steer})

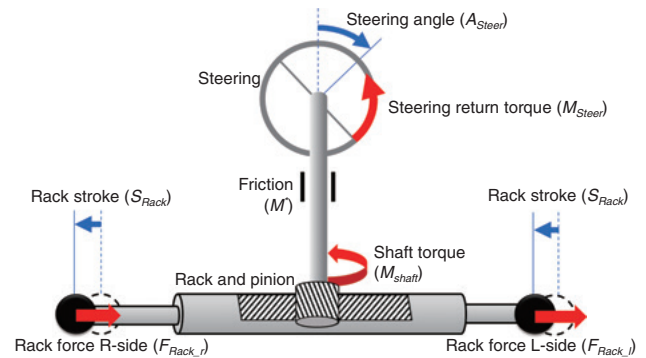


Fig. 7 Steering system

return angle center to be smaller. In other words, it has also been observed that F_{Rack} increases uniformly with respect to S_{Rack} , and at the point where $F_{Rack} = 0$, it is desirable for S_{Rack} to be as small as possible.

4. Means of Accomplishment

4.1. Overview of Predictive Model

Figure 8 shows the elements added to the conventional simulation model. In order to obtain R_{min} with sufficient accuracy, the basic elements of the suspension and steering system that make up the Ackermann geometry were modeled and incorporated. Incorporating the concept of Ackermann geometry produces a difference in the changing left and right wheel angle (δ) and the changing tire slip angle (θ). Taking the difference in the left and right wheel angle (δ) change and tire slip angle (θ) change also into consideration, the relational expressions for the tire lateral force (F_y) and drag (F_d) are determined for each wheel. The steered wheels F_y are converted into the torque around the king pin axis (M_{KIP}) on the left and right, respectively. M_{KIP} is transmitted separately to left and right through interposed basic elements making up the Ackermann geometry and become F_{Rack} . The ratio of F_{Rack} to M_{KIP} is referred to here as link efficiency.

When link efficiency can be defined, then F_{Rack} can be derived from M_{KIP} and S_{Rack} can also be derived from δ . In this way, the R_{min} and S_{Rack} - F_{Rack} characteristic can now be predicted simultaneously with the prediction of performance indices in other vehicle speed ranges.

Given the above, the basic concepts added to the conventional model to form the means of accomplishment can be classified under the following three headings.

- (1) Prediction of F_y for each of the four wheels taking the Ackermann ratio into consideration
- (2) Prediction of M_{KIP} for left and right independently
- (3) Prediction of S_{Rack} and F_{Rack} that takes link efficiency into consideration

4.2. Force Exerted on Contact Patches of the Four Wheels

When the wheel angle becomes larger while turning in the very low speed range, F_y with differing directions is generated in the left and right wheels. That is accompanied by the generation of F_d and in order to maintain the vehicle at a constant speed, a drive force commensurate with F_d will also be needed. These forces cannot be ignored when driving in the very low speed range. The forces generated in the tire contact patch were therefore classified under the following three headings and incorporated into the model.

- (A) Lateral force generated in the direction of the center of the turn
- (B) Resistance force that causes the vehicle to decelerate
- (C) Drive force that causes the vehicle to accelerate

4.3. Force Exerted on Contact Patches of the Four Wheels

When the wheel angle becomes larger while turning in the very low speed range, the direction of the tire slip angles on the left and right wheels will differ according to how the Ackermann geometry is incorporated. For that reason, F_y is generated in an opposing direction, in conjunction with which F_d is also generated in the respective tires. In order to maintain the vehicle at a constant speed, therefore, a drive force commensurate with F_d will also be needed. These forces cannot be ignored when driving in the very low speed

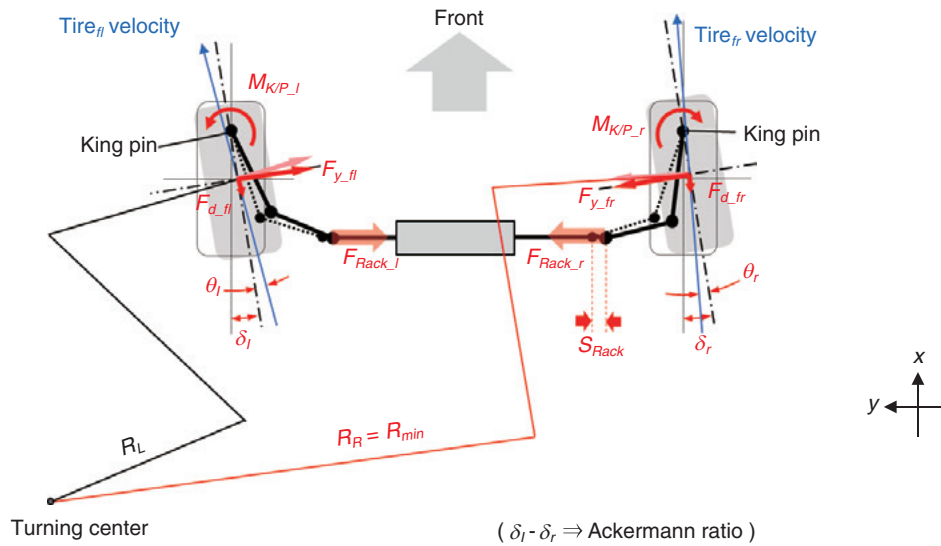


Fig. 8 Outline of model for calculating very low speed driving

range. All the forces generated in the tire contact patch were therefore classified under the following three headings and incorporated into the model.

(A) Lateral force generated in the direction of the center of the turn

(B) Resistance force that causes the vehicle to decelerate

(C) Drive force that causes the vehicle to accelerate

If the low speed state is so extreme that lateral acceleration can be ignored, the resultant force (A) that is generated in the vehicle as a whole can be treated as 0 N. When turning at a speed that does not allow lateral acceleration to be ignored, however, the force (B) will also be generated in each wheel according to the slip angle simultaneously with force (A). In order to continue turning at a constant speed at this time, a force (C) that is commensurate with force (B) will be needed.

Regarding force (B), when W is as the work rate for the vehicle as a whole, W_i is the work rate of each wheel, F_{d_i} is the resistance force defined by the coordinates for each wheel, R_i is the turning radius of each wheel, and r is the angular velocity at the vehicle center of gravity, then, as shown in Eq. (2), then W can be found from the summation of W_i .

$$\begin{aligned} W &= W_{fl} + W_{fr} + W_{rl} + W_{rr} \\ &= F_{d_{fl}} R_{fl} r + F_{d_{fr}} R_{fr} r + F_{d_{rl}} R_{rl} r + F_{d_{rr}} R_{rr} r \quad (2) \\ &= (F_{d_{fl}} R_{fl} + F_{d_{fr}} R_{fr} + F_{d_{rl}} R_{rl} + F_{d_{rr}} R_{rr}) r \end{aligned}$$

F_{d_i} will be in the following relationship, according to θ .

$$F_{d_i} = F_{y_i} \sin(\delta_i - \theta_g + \theta_i) + F_{x_i} \cos(\delta_i - \theta_g + \theta_i) \quad (3)$$

When F_{x_i} is the longitudinal force on each tire contact patch, F_{y_i} is the lateral force on each tire contact patch, δ_i is the wheel angle of each wheel, and θ_g is the slip angle of the vehicle center of gravity, then the slip angle θ of each tire is expressed by the relationship below.

$$\theta_i = \theta_g + l_i r + d_i r \quad (4)$$

The position vector for each wheel with respect to the center of gravity is (d_i , l_i). If the vehicle is turning to the left, then R_{min} corresponds to the lowest value of R_{fl} , and if the vehicle is turning to the right, then it corresponds to the lowest value of R_{fr} . The relational expressions Eqs. (2)-(4) will later become the conditions for determining R_{min} .

In cases where the influence of lateral acceleration cannot be ignored, a roll motion is generated in the vehicle and the state of the tire contact patch changes. The shift of load from the inside wheel to the outside wheel also affects the state of tire contact, so that the influence on those forces generated in the contact patch also needs to be taken into consideration. The influence of the load shifting from the inside wheel to the outside changes the force generated in the wheel contact patch.

The changes in F_x and F_y relative to the change in the moved load (F_z) are incorporated into Eq. (3) and the calculation is carried out.

4.4. Torque Around the King Pin Axis

Working from a conceptual approach like the above, the force generated in the contact patch is sought and on the basis of that force is converted into the torque around the king pin axis (M_{KIP}). F_{Rack} is generated by the M_{KIP} of the front left and right wheels. The M_{KIP} factors are classified under the following four headings.

- The factor of self-aligning torque that is generated in the tires and is converted to torque around the king pin axis
- The factor of lateral force that is generated in the tires and is converted to torque around the king pin axis
- The factor of vertical force that works on the tire-ground contact point and is converted to torque around the king pin axis
- The factor of force that is generated as spring reaction force by stabilizer or other such spring elements during steering and is converted to torque around the king pin axis

Items (a) and (b) can be converted into torque by the lateral force generated in the tire contact patch and by the moment arm length of the pneumatic trail portion and caster trail portion. When the pneumatic trail portion and caster trail portion of torque are designated M_{KIP_1} and M_{KIP_2} , respectively, then

$$\begin{aligned} M_{KIP_1} &= \Delta F_y \zeta_{pn} \cos \theta_{ch} \\ M_{KIP_2} &= \Delta F_y \zeta_{ct} \cos \theta_{ch} \end{aligned} \quad (5)$$

Here ζ_{pn} is the pneumatic trail, ζ_{ct} is the caster trail, ΔF_y is the lateral force generated while turning, and θ_{ch} is the caster angle. The values of ζ_{pn} and ΔF_y are mainly determined by the stand-alone tire characteristics, and the values of ζ_{ct} and θ_{ch} are mainly determined by the link geometry of the suspension and steering system. Figure 9 shows an overview of M_{KIP_1} and M_{KIP_2} .

With regard to item (c), the torque around the king pin axis resulting from the vertical force from the ground surface (M_{KIP_0}) can be found using Eq. (6).

$$M_{KIP_0} = F_z \zeta_z \sin \theta_z \quad (6)$$

F_z is the ground contact load, ζ_z is the distance between a normal line through the ground contact center and the king pin axis, and θ_z is the angle formed by a normal line through the ground contact center and the king pin axis. Therefore when the change in the ground contact load during a turn is ΔF_z , the torque around the king pin axis (M_{KIP_3}) according to item (c) is given by Eq. (7). Figure 10 shows an overview of M_{KIP_3} .

$$M_{KIP_3} = (F_z + \Delta F_z) \zeta_z \sin \theta_z \quad (7)$$

With regard to item (d), when elements that rotate around the king pin axis and spring elements are joined and when the vector of the deflection reaction force generated at the joint by spring elements does not intersect with the king pin, then the torque is that generated when the moment arm length is the distance between the vector of the deflection reaction force from the spring elements and the king pin axis. When this torque is designated M_{KIP_4} , then

$$M_{KIP_4} = \Delta F_s * \zeta_s * \sin \theta_s \quad (8)$$

Here ΔF_s is the reaction force of spring elements generated by suspension stroke changes or other such causes that occur while turning, ζ_s is the distance between the vector of the spring reaction force and the king pin axis, and θ_s is the angle formed by the vector of the spring reaction force and the king pin axis. Figure 11 shows an overview of M_{KIP_4} .

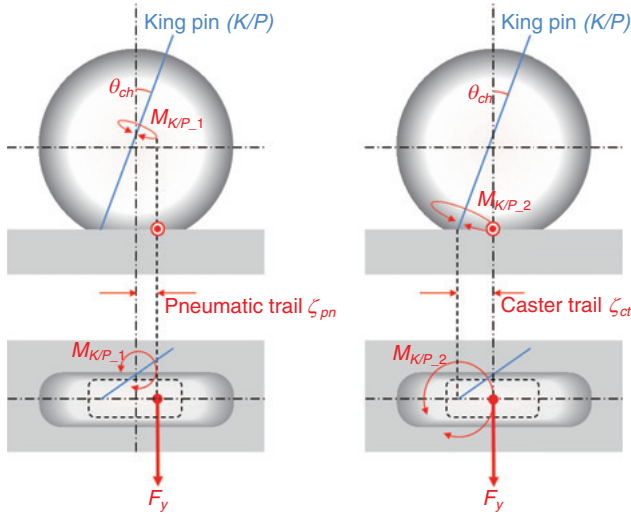


Fig. 9 Torque around king pin due to F_y

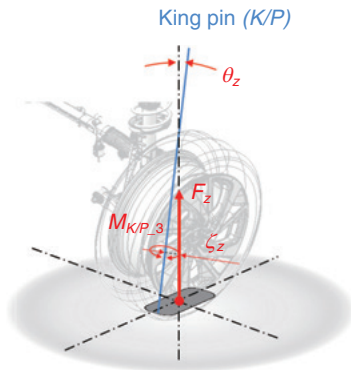


Fig. 10 Torque around king pin due to F_z

Figure 12 shows M_{KIP} changes with respect to S_{Rack} . The total of the torque around the king pin axis is M_{KIP} .

$$M_{KIP} = M_{KIP_1} + M_{KIP_2} + M_{KIP_3} + M_{KIP_4} \quad (9)$$

Through the link geometry, M_{KIP} becomes F_{Rack} .

4.5. Link Efficiency

Figure 13 shows a schematic representation of a steering system model. M_{KIP} reaches the rack through the link geometry of the steering system. The proportion of force transmitted changes according to changes in the angle of join in the tie rod link. When this happens, the relationship between the axial force on the link (F_{TR}), F_{Rack} , and M_{KIP} is described by Eq. (10).

$$\begin{aligned} F_{TR} &= M_{KIP} / (L_1 \cos \theta_a) \\ F_{Rack} &= F_{TR} \cos \theta_b \\ &= M_{KIP} \cos \theta_b / (L_1 \cos \theta_a) \end{aligned} \quad (10)$$

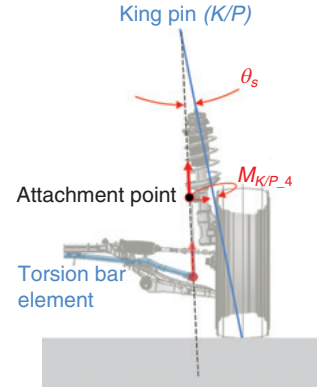


Fig. 11 Torque around king pin due to ΔF_s

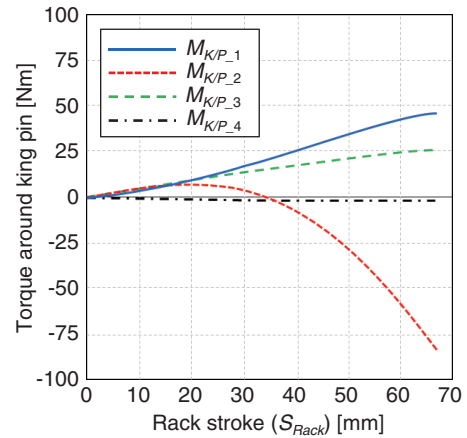


Fig. 12 Torque around king pin

In Fig. 13, θ_a and θ_b are the angles of the join portion of the link, L_1 is the length of the knuckle arm, and $\cos\theta_b / (L_1 \cos\theta_a)$ is the link efficiency, a coefficient that determines the relationship of $M_{K/P}$ and F_{Rack} . The link efficiency of the inside and outside wheels displays a different change with respect to S_{Rack} .

As described above, the concepts of calculation of the force generated in the contact patch of each tire, calculation of $M_{K/P}$ by each factor, and calculation of the link efficiency were successfully incorporated into the model. Incorporating these concepts into the model enables description of the S_{Rack} - F_{Rack} characteristic from the results in Fig. 12 and the relationship in Eq. (10). As a result, it became possible to investigate the S_{Rack} - F_{Rack} characteristic from the high speed range to the extreme low speed range working only from the information available at the initial stages of development.

4.6. Prediction of Minimum Turning Radius

The created steering system model is capable of deriving the left and right wheel angle changes individually in order to predict the link efficiency, and as a result it can also simultaneously predict the Ackermann ratio. The turning radius of each wheel was derived by convergence calculation from the Ackermann ratio and S_{Rack} highest value as well as from the relationships in Eqs. (2)-(4). Then the turning radius of the outside front wheel was defined as R_{min} .

4.7. Prediction Results

Figure 14 shows a comparison of prediction results for turning radius and the measurement results for minimum turning radius. The measurement results have a margin because they include the measurement error. The prediction results have sufficient accuracy, and they confirmed that changes in the turning radius could simultaneously be predicted across vehicle speeds from 2 km/h to the vicinity of 10 km/h.

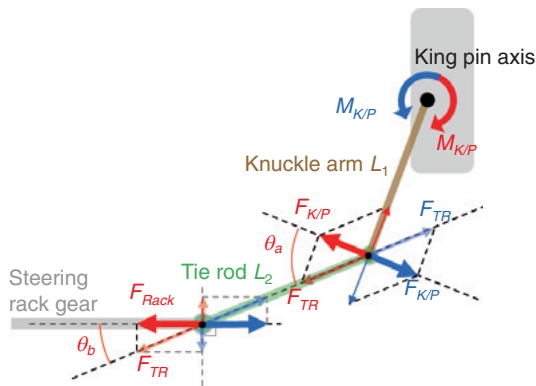


Fig. 13 Diagram of steering system model

Figure 15 shows changes in link efficiency with respect to S_{Rack} . Comparison was made with prediction results from a full vehicle model, and it was confirmed that the differing characteristics of the inside and outside wheels were reproduced satisfactorily.

Figure 16 shows the prediction results for the S_{Rack} - F_{Rack} characteristic. Comparison was made with prediction results from a full vehicle model, and it was confirmed that the F_{Rack} peak value, S_{Rack} value at peak F_{Rack} , changes in F_{Rack} value in the vicinity of the greatest S_{Rack} , and other such characteristics that are needed for investigation were largely successfully reproduced.

Figure 17(a) shows prediction results for very low speed range vehicle handling indices and Fig. 17(b) shows indices for vehicle dynamics characteristics predicted simultaneously while driving at a speed of 120 km/h. The legend indicates the different specifications for which

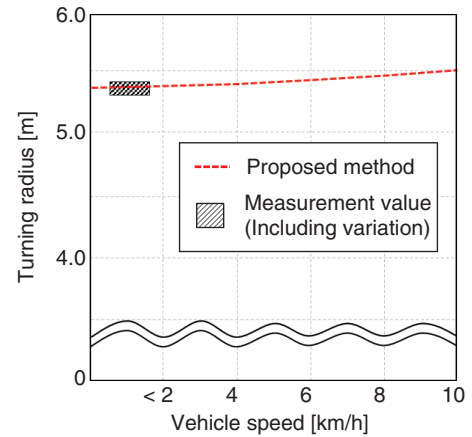


Fig. 14 Comparison of minimum turning radius

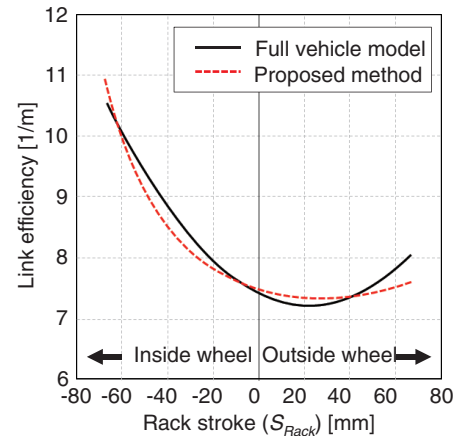


Fig. 15 Comparison of link efficiency

prediction results were obtained. It was confirmed from Fig. 17 that the investigation of specifications could proceed while comparing performance driving at the very low speed range and at high speed.

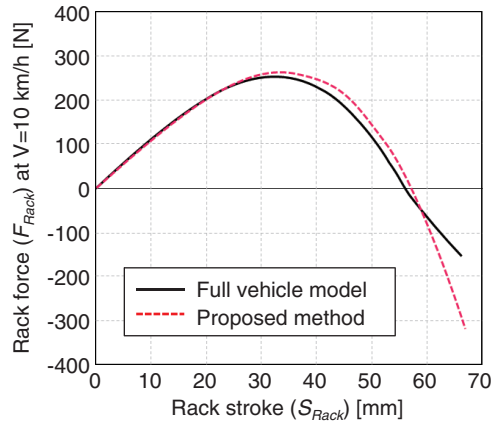


Fig. 16 Comparison of axial force of rack

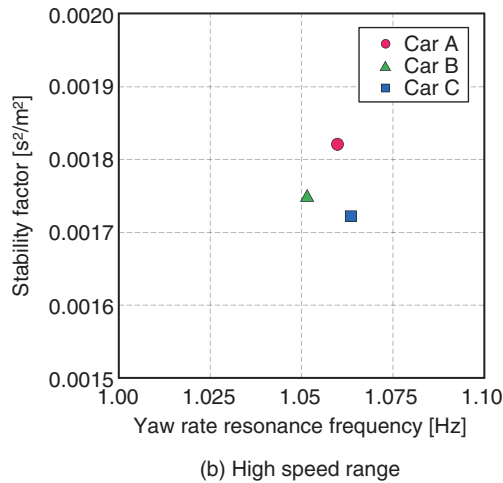
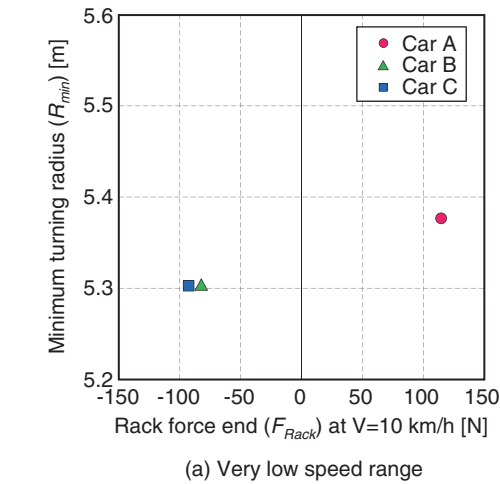


Fig. 17 Performance design including very low speed area

5. Cases of Application

Figure 18 shows examples of proposed countermeasures for chassis elements investigated on the desk after analyzing actual factors with regard to M_{Steer} . These examples show a breakdown of M_{KIP} , and the extent of influence that the specification changes for each chassis element part have on M_{Steer} is apparent. Given these analysis results, the investigation of specifications for chassis elements can be carried out efficiently. It is anticipated that the prediction results for F_{Rack} characteristic can be used not only in investigating the specifications for chassis elements, but also for examining the specifications for the control motors used to assist steering torque. This kind of investigation now provides the means to obtain a view of prospective chassis specifications early in the process. This technique was applied in the performance design of the chassis platform for compact vehicles that has been adopted from the 2014 model year FIT.

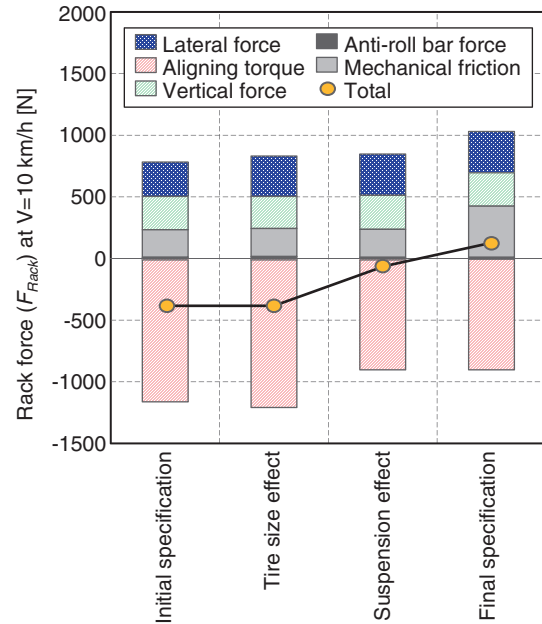


Fig. 18 Factor analysis calculation result

6. Conclusion

The simulation model used for performance design in the development of mass-production vehicles was taken as a base for creating a simulation technique that is capable of predicting indices for vehicle handling at the very low speed range together in bulk, where these were previously calculated separately by creating multiple models. The substance of this created technique is summarized in the following.

- (1) The elements that provide for investigation of very low speed range vehicle handling were incorporated into the conventional model and a model that is capable of seamless prediction from high speeds to the very low speed range was created.
- (2) It was confirmed that analysis of the contribution to very low speed vehicle handling by chassis elements could proceed simultaneously with investigation of the specifications for performance at other vehicle speed ranges.
- (3) This simulation technique was applied in development of the 2014 model year FIT, it was used in actual mass-production vehicle development, and the technique was confirmed to be useful.

References

- (1) Fujikawa, S.: Model Based Development in Mazda, MAZDA TECHNICAL REVIEW, No. 31, p. 45-47, (2013) (in Japanese)
- (2) Ohtomi, K., Hato, T.: Design Innovation Applying IDCAE, TOSHIBA REVIEW, Vol. 67, No. 7, p. 7-10, (2012) (in Japanese)
- (3) Toyoshima, T., Matsuzawa, T., Endo, Y., Ito, K., Takano, H., Ushimura, T.: Development of Chassis Performance Design Technologies for MBD of Mass-production Vehicles, Honda R&D Technical Review, Vol. 29, No. 1, p. 52-62
- (4) Toyoshima, T., Matsuzawa, T., Ushimura, T.: Lap Time Simulation Technology for Performance Design during Production Car Development, Honda R&D Technical Review, Vol. 29, No. 2, p. 124-133
- (5) Toyoshima, T., Azekatsu, Y., Ikeda, N.: Crosswind Stability Simulation Technology for Performance Design in Model Base Development of Mass-production Vehicles, Honda R&D Technical Review, Vol. 30, No. 1, p. 66-73
- (6) CHO Y.G.: VEHICLE STEERING RETURNABILITY WITH MAXIMUM STEERING WHEEL ANGLE AT LOW SPEEDS, Int J Automot Technol, Vol. 10, No. 4, p. 431-439, (2009)
- (7) Matschinsky, W.: Road Vehicle Suspensions, Professional Engineering, p. 195-242, (2000)
- (8) Shinoto, S., Miyanaga, S., Shinozuka, G., Hanamura, Y.: Evaluation Technology. Evaluation Prediction of Some Performances Affected by Front Wheel Geometry at the Initial Stage of Designing., Isuzu technical journal, No. 101, p. 56-61, (1999) (in Japanese)
- (9) Fujioka, T., Kamata, M., Nakamura, S., Tsugawa, S., Yamamoto, M., Takeda, N.: Jidosha Project Kaihatsu Kogaku -Kankyo, ITS, Undoseino, Butsuryu-, Gihodo Shuppan, p. 226, (2001) (in Japanese)
- (10) Tanaka, T., Uki, H., Osaki, M.: A Study of Steering Characteristics, Journal of Society of Automotive Engineers of Japan, Vol. 47, No. 12, p. 44-47, (1993) (in Japanese)
- (11) Japanese-English Dictionary for Automobiles, p. 394, (1997) (in Japanese)
- (12) Automotive Technology Handbook Chassis design, Chapter 8, p. 527-535, (in Japanese)
- (13) Desaki, Y., Kubota, M., Washizu, S., Oshita, S.: Self-centering Torque of Steering System at High Angles and Very Low Manoeuvring Speed, Journal of Society of Automotive Engineers of Japan, Vol. 36, No. 3, p. 310-312, (1982) (in Japanese)
- (14) Nakamura, I., Ikawa, K.: Analysis of Steering Effort at Low Speeds, Journal of Society of Automotive Engineers of Japan, Vol. 19, p. 53-58, (1979) (in Japanese)
- (15) Takei, Y., Nishimura, H., Tajima, H.: Three-dimensional Motion Analysis of A Four-wheel Vehicle Taking Account of Suspension and Steering Geometry: Dyn Des Conf, Vol. 2006, ROMBUN No. 207, (2006) (in Japanese)

Author



Takayuki TOYOSHIMA



Yoshitomo AZEKATSU



Yoshiaki KOBAYASHI



Naofumi IKEDA



Hiroki TAKANO



Isao OBATA

CFD Calculation of Cavitation in Outboard Motors

Ayaka NANGO*

Yuka IGA**

ABSTRACT

CFD calculations accounting for cavitation in outboard motor propellers were performed to save calculation cost and calculation accuracy. The impact of the boundary layer mesh resolution on the main propeller performance values of thrust coefficient, torque coefficient, and propeller efficiency was clarified. The obtained results of CFD calculation were qualitatively consistent with the results of model tests, and its accuracy was within $\pm 7\%$. With the above, cavitation effects can be accounted for from the early stages of outboard motor development using CFD calculation.

1. Introduction

Cavitation is known to occur in the propellers and gear case of outboard motors. As depicted in Fig. 1, the phenomenon of cavitation is when drops in pressure cause liquid to phase change into gas.

With some exceptions⁽¹⁾, cavitation is known to decrease performance⁽²⁾. Propeller cavitation degrades thrust and gear case cavitation increases resistance values, in turn decreasing boat speed and increasing fuel consumption of outboard motors. Furthermore, cavitation can also cause erosion, leading to degradation of appearance quality. Thus, it would be preferable to account for cavitation effects from the early stages of development.

In outboard motor fuel economy and power performance projections for Honda outboard motor development to date⁽³⁾, performance has been calculated using calculations ignoring cavitation. However, as outboard motors have increased in output in recent years, they are approaching

speed ranges high enough to generate cavitation. This increases the need for calculations which account for cavitation.

While the cavitation phenomenon has become computable using commercially-available Computational Fluid Dynamics (CFD) software^{(4), (5)}, no method of modeling cavitation has yet been established⁽⁶⁾. Generally speaking, increasing mesh resolution in CFD calculations will allow for cavitation to be predicted more accurately at the expense of increased computational time, and thus computation cost. Due to this factor, a balance should be struck between calculational accuracy and cost if CFD is to be used in the initial development phase.

There have been several studies on predicting cavitation in outboard motors, including one study comparing two-dimensional blade tests and CFD calculations⁽⁷⁾, and another on predicting erosion based on CFD calculation results for cavitation⁽⁸⁾. However, in terms of three-dimensional cavitation prediction accuracy, neither of the above studies address the practical applications in outboard motors at any great length.

Thus, in this paper, the authors will clarify the connections between computational cost and accuracy in order to predict performance, accounting for the cavitation that occurs in outboard motor propellers.

2. Cavitation Prediction Calculation

2.1. Calculation Subject

The purpose of this paper was to reproduce the pressure around objects in a liquid and predict the areas where cavitation will be generated.

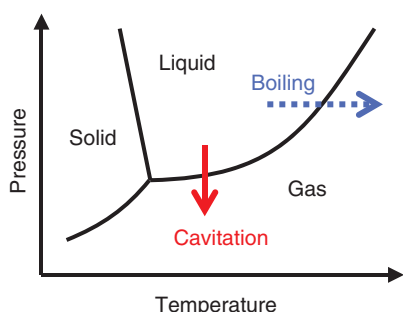


Fig. 1 Phase diagram of water

* Power Products R&D Center

** Institute of Fluid Science, Tohoku University

Given the intended application in outboard motors, as well as the multiple studies on external flow and in general, calculations were performed for a propeller shape in water.

2.2. Calculation Method

2.2.1. CFD solver

Commercial CFD software was used for calculations. The calculation settings are given in Table 1. Whereas actual cavitation is an unsteady phenomenon, it was treated as a steady flow using a Reynolds-averaged Navier-Stokes (RANS) equation in light of calculational costs. Assuming flow to be roughly equivalent in liquid and gas phases, an Euler mixture model with no speed difference between liquid and gas was used to model the gas phase. For the cavitation model, a Zwart-Gerber-Belamri bubble dynamics model⁽⁹⁾ was used.

2.2.2. Stabilization of the calculation procedure

When cavitation occurs, the density of a liquid will change suddenly, converting it to gas. Calculations are also destabilized by the sudden change in physical quantities; the calculation diverges at the point when the cavitation occurs, making the equation unsolvable in some cases.

As seen in Table 2, the calculation was stabilized by starting with the pressure environmental conditions set high, calculating for single-phase and multi-phase flow in separate steps, and then reducing the pressure gradually.

2.2.3. Calculational domain

The calculational domain is given in Fig. 2. The domain dimensions were aligned with conditions for the cavitation tunnel in the test to be described below, against which results were compared to verify computation accuracy for this study. The calculational domain used was a cylindrical shape of equivalent area to the flow path cross section of the tunnel test device. Propeller rotation was simulated by rotating the entire calculational domain with the outer periphery wall of the cylinder fixed to the absolute coordinate system. To reduce calculational costs, the calculational domain was reduced to a three-fold symmetrical domain centered on the propeller axis for just one propeller blade, using a periodic boundary surface.

2.2.4. Calculational mesh

As mentioned above, reproducing the flow near the

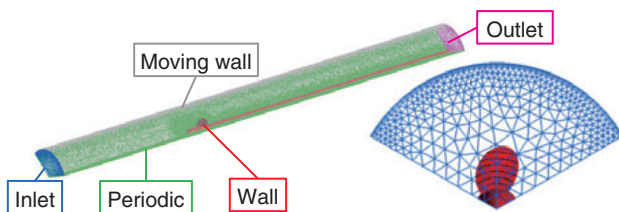


Fig. 2 Calculational domain

Table 1 Calculation settings

CFD software	ANSYS Fluent 16.0
Time	Steady
Viscous model	SST k- ω
Multiphase flow model	Mixture model
Cavitation model	Zwart-Gerber-Belamri

Table 2 Calculation procedures

Step 1	Single phase flow calculation with high pressure
Step 2	Multiphase flow calculation with high pressure
Step 3	Cavitation calculation with high pressure
Step 4	Cavitation calculation while reducing pressure

Table 3 Mesh generation parameter for boundary layer

	Mesh i	Mesh ii	Mesh iii
Number of boundary layers	25	15	5
First layer height [mm]	0.025	0.05	0.2
Growth rate	1.05		

object surface is important to simulating cavitation. As this study uses a low-Reynolds-number SST k- ω turbulence model, calculational accuracy should increase as the mesh resolution near the object surface is increased⁽¹⁰⁾. Conversely, increased mesh resolution will also increase the calculational cost. Thus, in order to verify the trade-off between calculation results and calculational cost due to mesh resolution, the mesh near the object surface was changed for the boundary layer only.

The mesh generation parameters for the calculated boundary layers are shown in Table 3. Note here that the mesh domain thickness was set to be about the same for each calculated boundary layer mesh.

3. Results and Discussion

3.1. Verification Tests for Calculational Accuracy

In order to verify their calculational accuracy, the CFD calculation results were compared to results of a model propeller test conducted in a cavitation tunnel. The cavitation tunnel test was conducted in the Marine Propeller Cavitation Tunnel at the University of Tokyo⁽¹¹⁾. A cavitation tunnel is a circulating water tank. Cavitation can be generated simulating actual usage by changing pressure and flow velocity in the tunnel. In the test, a propeller was rotated within a quadratic prismatic flow path, then a dynamometer was used to measure thrust T and torque Q while observing the cavitation patterns from the observation window.

The key propeller characteristics are given in Table 4. Three types of propeller were used in this paper. D in the table refers to the propeller diameter.

Here, the equations given below use the common, non-dimensional advance coefficient J , thrust coefficient K_T , torque coefficient K_Q , propeller efficiency η , and cavitation number σ_n as used in the field of ship propulsion.

$$J = \frac{V_A}{nD}$$

$$K_T = \frac{T}{\rho n^2 D^4}$$

$$K_Q = \frac{Q}{\rho n^2 D^5}$$

$$\eta = \frac{J \cdot K_T}{2\pi \cdot K_Q}$$

$$\sigma_n = \frac{p_0 - p_v}{\frac{1}{2} \rho n^2 D^2}$$

V_A represents propeller inflow velocity, n the propeller speed, ρ the fluid density, p_0 the fluid standard pressure, and p_v the fluid saturated vapor pressure.

Advance coefficient J is a non-dimensional number for the angle of flow into the propeller. At the same propeller speed, J increases with boat speed. Cavitation number σ_n is a non-dimensional number for how easily propeller cavitation occurs. Cavitation is increasingly likely the smaller σ_n becomes.

In a cavitation tunnel test, advance coefficient J and cavitation number σ_n can be matched to actual usage conditions to observe similar cavitation.

In our test, an advance coefficient J of 0.99 and a cavitation number σ_n of 0.53 were set as the standard test conditions. These conditions were set to correspond to a flow speed of 70 km/h and propeller speed of 3000 rpm: the maximum speed range for a planing boat equipped with a 250-horsepower outboard motor under actual usage conditions. During the test, propeller speed was kept constant and the advance coefficient J was changed by controlling the flow velocity in the tunnel.

3.2. Calculation Mesh Verification Results

As mentioned in Clause 2.2.4., three calculation meshes with different boundary layers were used in CFD calculation on Propeller A from Table 4. Taking the calculation time for Mesh i as the baseline, the calculation times for one iteration under each calculation mesh condition are given in Table 5.

The cavitation patterns on the propeller blade suction side are given in Fig. 3, and a diagram of propeller performance is given in Fig. 4. The domain covered in cavitation from the propeller blade suction side grows as mesh resolution is increased. Given that thrust will increase the wider the domain covered by blade suction side

Table 4 Key propeller characteristics for cavitation tunnel test

Propeller name	A	B	C
Diameter [m]	0.220		
Pitch ratio (at 0.7D)	1.26	1.06	1.12
Expanded area ratio	0.481	0.481	0.460
Number of blades	3		
Rotational direction	Counter-clockwise		
Material	Corrosion resistant aluminum alloy		

Table 5 Relative calculation time

	Mesh i	Mesh ii	Mesh iii
Relative calculation time (Mesh i as 1.00)	1.00	0.83	0.58

cavitation, this is thought to account for the differences in thrust coefficient K_T in Fig. 4.

In the verification results, Mesh i and Mesh ii exhibited very similar cavitation patterns and propeller performance with a difference within 1%. Given this, Mesh ii was selected for subsequent calculations due to balance between calculational accuracy and cost.

3.3. Summary of Test Verification Results

The test verification results are separated into cavitation patterns for qualitative comparison and propeller performance for quantitative comparison. Note here that calculations took approximately 9 hours using a dual-CPU, 48-core parallel processor for each calculation condition.

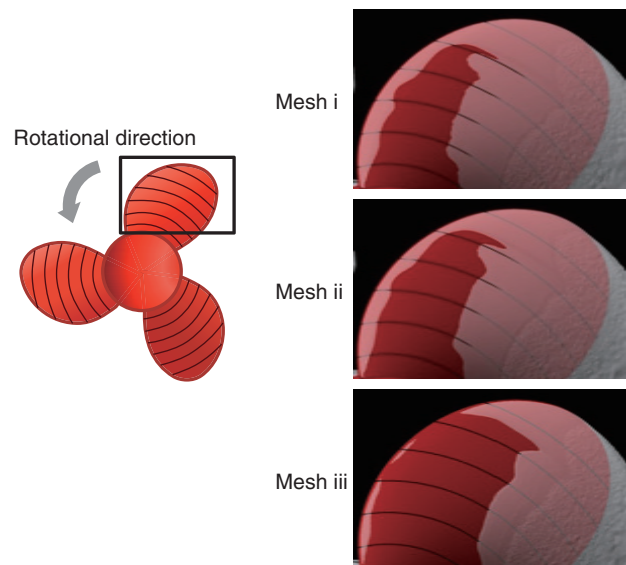


Fig. 3 Influence of boundary layer mesh resolution on cavitation on blade suction side

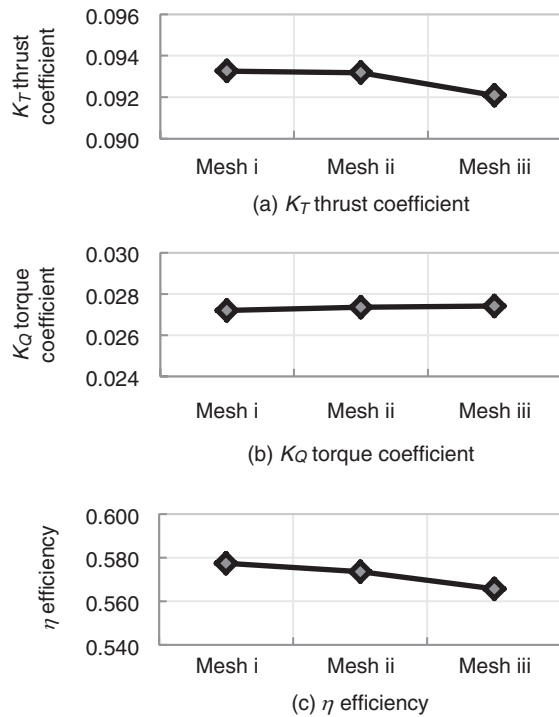


Fig. 4 Influence of boundary layer mesh on propeller performance

3.3.1. Cavitation patterns

The resulting cavitation patterns from the model propeller test and CFD calculations on the blade suction side of Propeller A are compared in Fig. 5. From left to right, the results are for an advance coefficient J of 0.80, 0.99, and 1.10, respectively.

- At J values of 0.80 and 0.99, the entire blade leading edge generates sheet cavitation; at a J value of 1.10, cavitation is only generated near the blade tip from positions of D values of 0.8 or higher.
- Cavitation generated by the preceding blade gets elongated when J is smaller.

These patterns were very similar.

While calculations did not replicate the bubble cavitation observed in the blade center at a J value of 1.10, bubble cavitation is highly unsteady and thus likely challenging to reproduce with the steady-state calculations done in this study. Furthermore, a J value of 1.10 exceeds peak propeller efficiency and thus is outside of actual usage ranges. Given that calculations successfully reproduced the sheet cavitation patterns generated at J values of 0.80 and 0.99 within the actual usage range, the test was deemed to be able to reproduce steady cavitation patterns.

3.3.2. Propeller performance

A comparison of propeller performance from the model

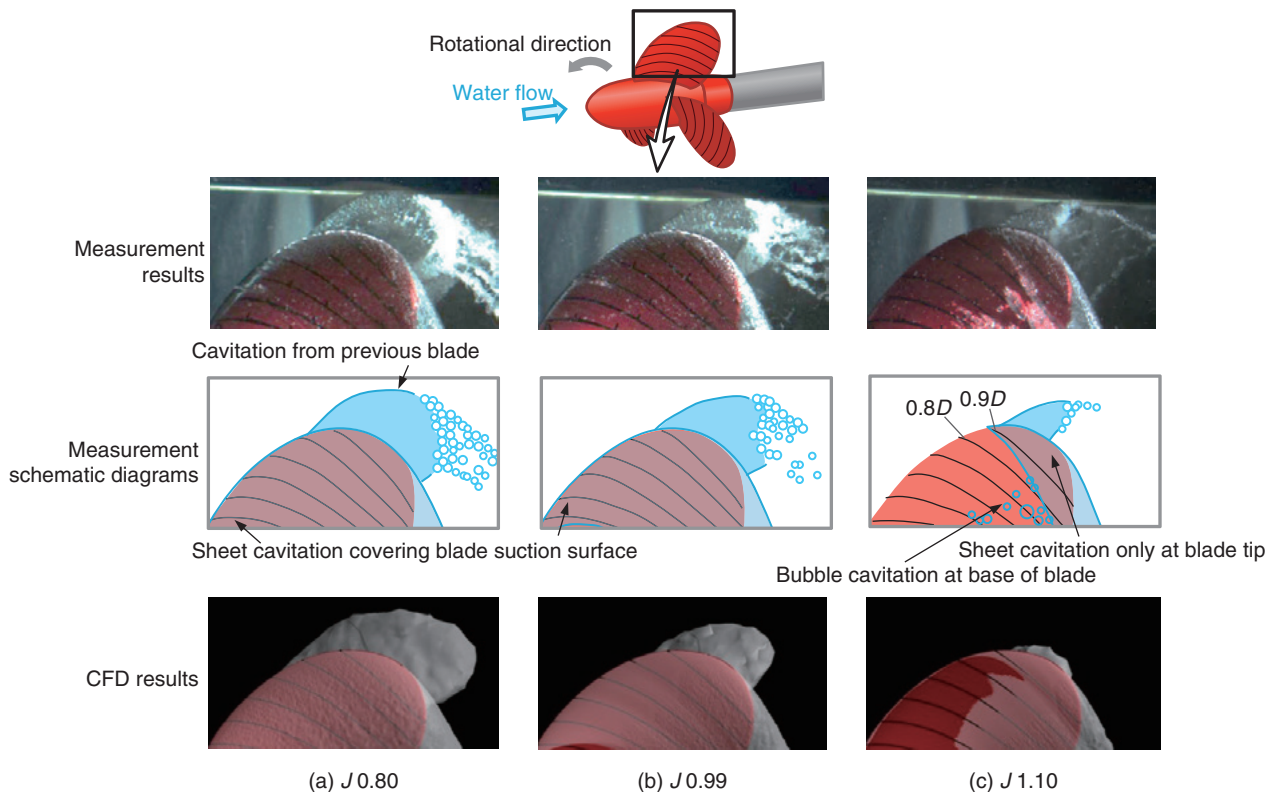


Fig. 5 Propeller blade cavitations, compared between test and calculation

propeller test and CFD calculations is given in Fig. 6. The high-to-low sequence of thrust coefficient K_T between the propellers and peak positions for propeller efficiency η

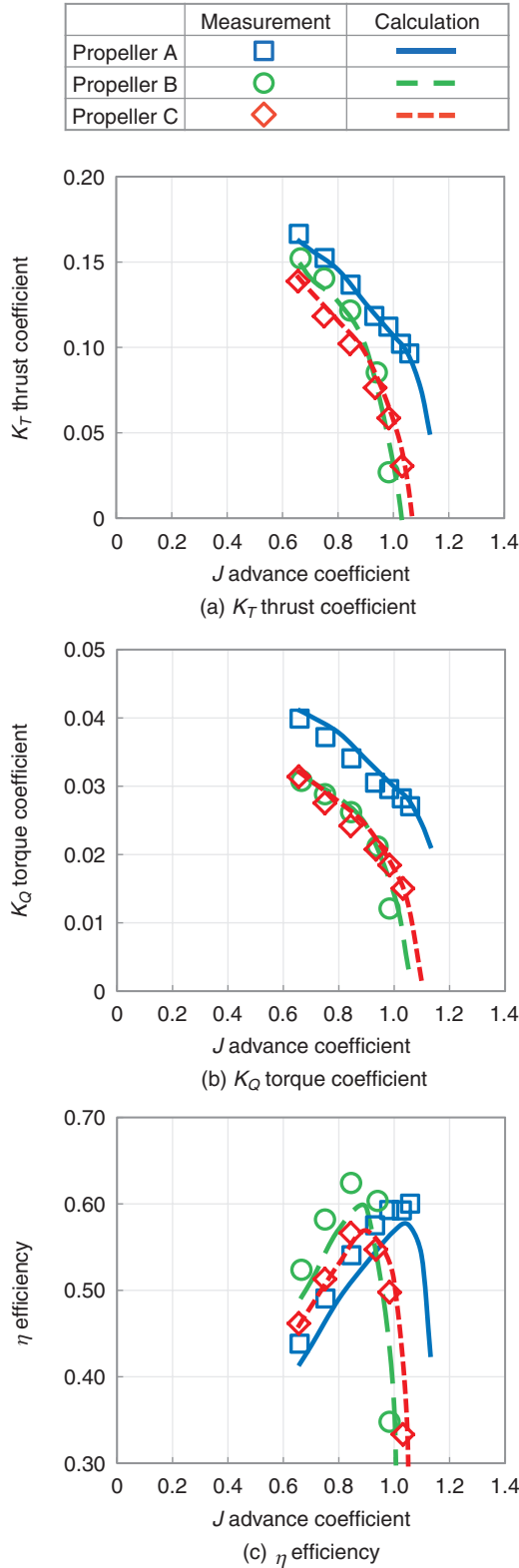


Fig. 6 Propeller performance curve compared with model test results

were consistent between the model propeller test results and CFD calculation. Thus, the CFD calculations are capable of reproducing the differences in performance from different propeller specifications.

Furthermore, within the actual usage range up to the efficiency peak for advance coefficient J , the model test and CFD calculation results were quantitatively consistent to within an error of $\pm 7\%$ for the performance values (thrust coefficient K_T , torque coefficient K_Q , and propeller efficiency η).

Thus, the results of the study illustrate that propeller performance in cavitation-generating conditions for actual outboard motor propeller usage can be quantitatively reproduced using CFD calculations.

4. Conclusion

In order to balance computational costs and accuracy in predicting the phenomenon of cavitation for the shape of outboard motor propellers, the following results were obtained:

- (1) The computational costs of changing the boundary layer mesh near the object surface was clarified, along with the impact of this change on the main propeller performance values of thrust coefficient K_T , torque coefficient K_Q , and propeller efficiency η .
- (2) The results of model tests and CFD calculation were compared, confirming that cavitation patterns were qualitatively consistent and that quantitatively, main propeller performance value accuracy is consistent to within $\pm 7\%$.

From the above results, cavitation effects can be accounted for from the early stages of outboard motor development as output and speed increases.

References

- (1) Ukon, Y., Fujisawa, J., Kudo T., Kurobe Y.: Application of a Theoretical Design Method to Full Scale Supercavitating Propellers for High-Speed Boats, Transactions of the West-Japan Society of Naval Architects, Vol. 100, p. 133-143, (2000) (in Japanese)
- (2) Suzuki, K., Sasaki, N., Kawamura, T.: Sentaiteiko to Suishin (Sempaku Kaiyo Kogaku Sirizu 2), SEIZANDO-SHOTEN PUBLISHING CO., LTD., p. 143-162, (2012) (in Japanese)
- (3) Kurioka, S.: Fuel Consumption and Performance Prediction in Outboard Motors for High-speed Planing Boats using CFD Simulation, Honda R&D Technical Review, Vol. 25, No. 1, p. 176-183
- (4) Fujiyama, K.: Numerical Simulation of Ship Hull Pressure Fluctuation Induced by Cavitation on Propeller with Capturing the Tip Vortex, International Symposium on Marine Propulsors smp'15, June, (2015)
- (5) Tomimatsu, S., Kado, K., Otani, R.: Cavitation CFD in Pump Development, Turbomachinery, Vol. 42, No. 11,

- p. 707-712, (2014) (in Japanese)
- (6) Nohmi, M.: Remaining Problems on Cavitation CFD, Turbomachinery, Vol. 42, No. 11, p. 694-700, (2014) (in Japanese)
 - (7) Kamiya, A.: Prediction of the Cavitation Effect on the Flow Around the Outboard Motor Propeller Blade Hydrofoil Section Using CFD, SAE Technical Paper, 2013-32-9157, (2013)
 - (8) Watanabe, T., Sakamoto, H.: Method for Predicting Erosion Due to Cavitation of Outboard-Motor, SAE Technical Paper, 2014-32-0054, (2014)
 - (9) Zwart, P. J., Gerber, A. G., Belamri T.: A Two-Phase Flow Model for Predicting Cavitation Dynamics, ICMF 2004 International Conference on Multiphase Flow, Japan, Paper No. 152, May 30-June 3, (2004)
 - (10) ANSYS, Inc. Fluent 16.0 Theory Guide, Chapter 4: Turbulence, (2014)
 - (11) Kawamura, T., Maeda, M.: Marine Propeller Cavitation Tunnel at The University of Tokyo, Journal of the Japan Institution of Marine Engineering, Vol. 43, No. 5, p. 720-725, (2008) (in Japanese)

■ Author ■



Ayaka NANGO



Yuka IGA

Honda R&D Co., Ltd.

Honda R&D Technical Review Editorial
Committee

Motorcycle R&D Center

Automobile R&D Center

Power Products R&D Center

Aircraft Engine R&D Center

R&D Center X

Editorial Afterword

The world is experiencing an AI boom. It is claimed that AI is capable of recognition and judgment, which have previously been considered unique human abilities. AI will make autonomous driving and more intelligent vehicles possible, and will affect not only the automotive industry but also work and the social structure as a whole.

We are in the midst of the type of enormous transformation that happens once a century. Proofreading, reviewing, and editing is work which requires very subtle consideration – will even this be replaced by AI in some years? Nevertheless, I wonder how an author would feel if a computer was to say, “This manuscript fails to reach the standard for acceptance in 23 points. Re-submit after increasing novelty and logical presentation by 15 points each”. The author would probably be irritated. Or what if a computer was to automatically convert an author’s manuscript into a flowing prose that could have been written by someone else entirely, and state “Manuscript completed.” This would also be unacceptable. Naturally, I believe that the acts of writing and reading about technology are eminently human activities, because these activities are performed for humans.

In this issue of Technical Review we have gathered together and presented the essence of Honda’s latest technologies. It includes a mini special feature on Honda’s flagship motorcycle, the GL1800, which has realized an exhilarating ride feel and the joy of agile handling. I would like to take this opportunity to thank all of the authors who play the leading role in the TR, in addition to the many people who have supported us on the journey to publication. To the readers who open these pages, I will be delighted if they offer a sense of the living breath of our engineers, who bring these technologies into existence. We remain committed to the idea that “Technology is a means to serve people,” as our founder, Soichiro Honda, never tired of repeating.

Watanabe

©2018 Honda Motor Co., Ltd. (All rights reserved)

ISSN 1881-6223

Published by:

Honda Motor Co., Ltd.

2-1-1, Minami-Aoyama, Minato-ku,

Tokyo 107-8556, Japan

Edited by:

Honda R&D Co., Ltd.

Honda R&D Technical Review Editorial Committee

Printed:

Sep. 1, 2018

Honda R&D Co., Ltd.