

Honda R&D

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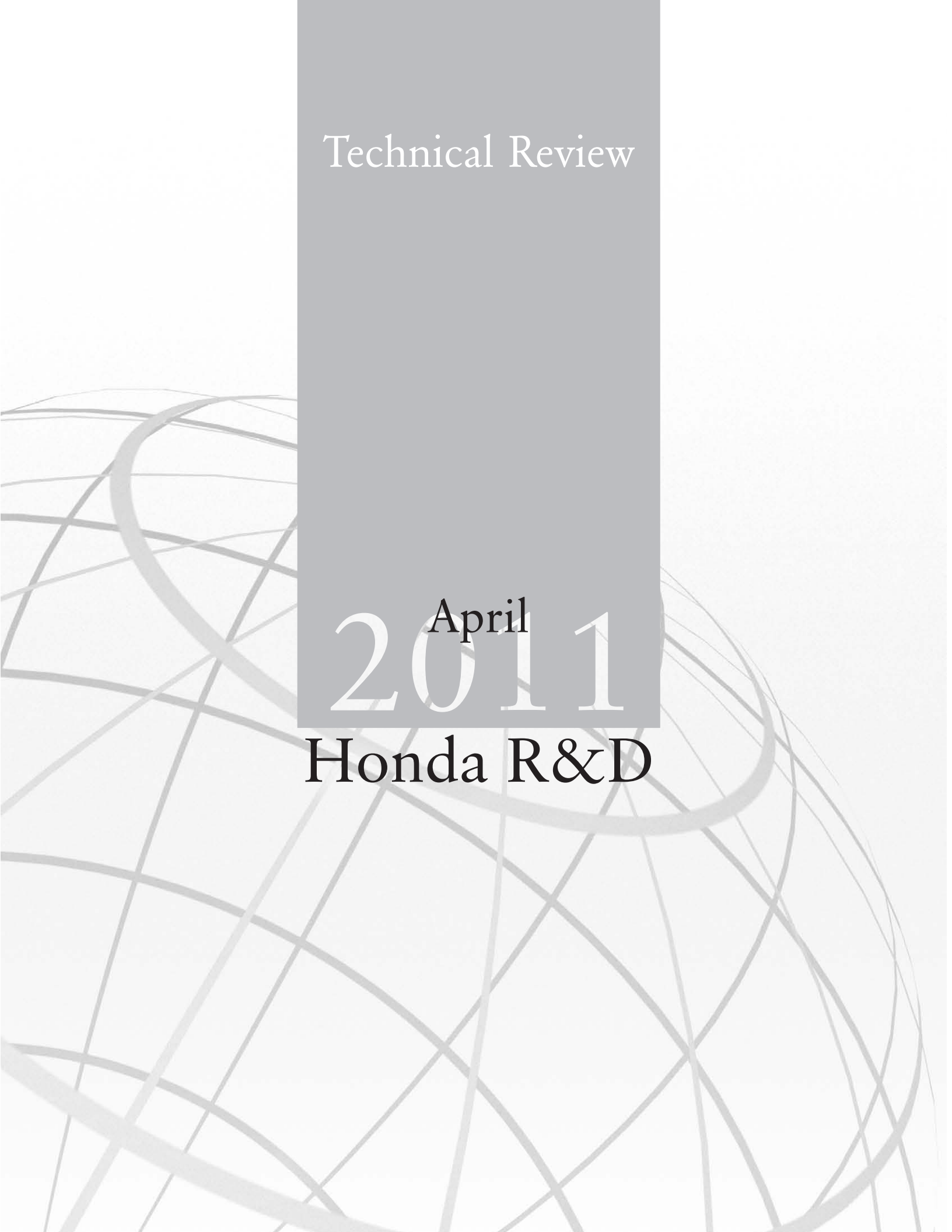
Comment on the Cover

The finished form of Honda R&D's efforts have great attraction; an appeal more than just the appearance of the outer surface of the body or the physical components. It is the summation of the efforts of so many Honda associates. The finished project represents a bit of the soul of each of them. In this image, the external excitement and beauty of the CR-Z SEMA show car holds the reflections of the sketches, the welding and metal work, and fabrication activities of several associates from HRA and HPD involved with its creation.

Each Honda associate is reflected in the final form of each Honda product.



Honda R&D Americas, Inc.
Fred Soares



Technical Review

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Development of an Auto Throttle System for High-Pressure Washer Engines

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ABSTRACT

An Auto Throttle system for residential high-pressure washer (HPW) engines has been developed. During periods of non-loaded use, engine speed is reduced from 3850 rpm to 2500 rpm.

A residential high-pressure washer has two modes, loaded (the operator is using the spray wand) and unloaded (the operator is not using the spray wand). Compared to the base model, in an unloaded condition, the Auto Throttle control system achieves a noise level reduction of 9 dB(A). The fuel economy is increased by 43%, overall carbon dioxide emissions are reduced, and engine and pump life are enhanced. User fatigue is reduced through a reduction of initial spray wand reaction force.

1. Introduction

One of the Honda philosophies is the continuous enhancement of products through economic, ecological, and ergonomic characteristics. Launched in 1997, the Honda GC series (GC Horizontal crankshaft, GCV Vertical crankshaft) engines are the highest sales volume Honda general purpose engine for consumer usage. In the late 1990s, residential high-pressure washers became very popular. Consequently, sales of the GCV engine increased and today the majority of GCV engines are used in the residential high-pressure

washer (HPW) market. The Auto Throttle system for the GCV engine was launched to U.S. customers in March 2010. Figure 1 shows the first application for the Auto Throttle engine on a residential HPW with an illustration of the Auto Throttle control system and its location.

2. Throttle Control for the Base High-Pressure Washer Engine

Table 1 shows the specifications for the Honda GCV190 engine, which is the base engine for the Auto

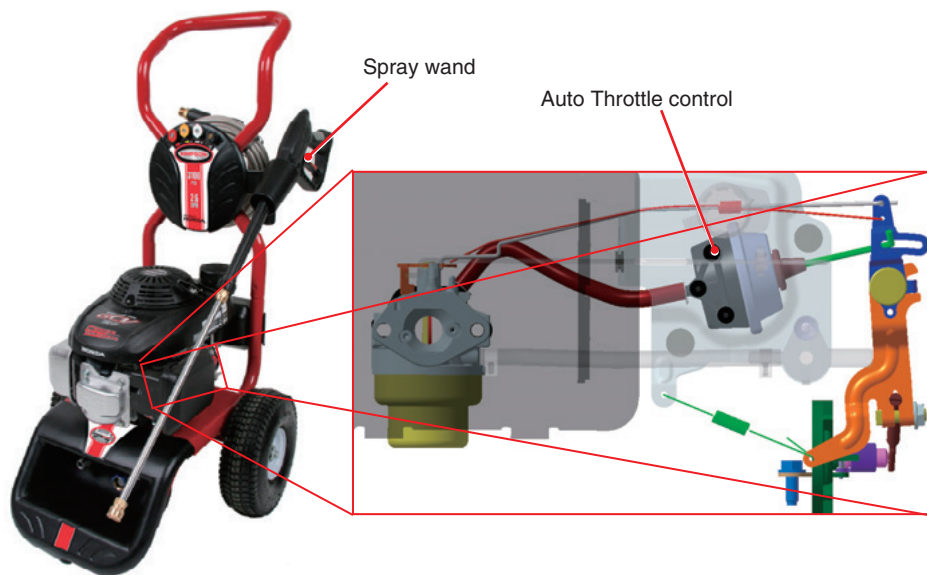


Fig. 1 High-pressure washer unit with Auto Throttle system

* HONDA R&D AMERICAS, INC.

Table 1 Honda GCV190 engine specifications

Engine type	Air-cooled 4-stroke OHC single cylinder
Bore x stroke	69 x 50 mm (2.7 x 2.0 in)
Displacement	187 cm ³ (11.4 cu in)
Compression ratio	8.5 : 1
Net horse power output	3.8 kW (5.1 HP) at 3,600 rpm
Net torque	11.3 Nm (8.3 lbs ft) at 2,500 rpm
Ignition system	Transistorized magneto
Dimensions (L x W x H)	367 mm (14.4 in) x 331 mm (13.0 in) x 349 mm (13.8 in)
Starting system	Recoil or electric starter
Carburetor	Horizontal type butterfly valve
Lubrication system	Forced splash
Oil capacity	0.55 l (0.58 US qt, 0.48 Imp qt)
Fuel tank capacity (liter)	0.93 l (0.98 US qt)
Dry weight	12.5 kg (27.6 lbs)

Throttle system. The engine has two modes of operation, loaded and unloaded. Loaded mode is when the user has activated the spray wand and the HPW pump is in its full-flow, high-pressure setting. The unloaded mode is when the spray wand is released. This causes the pump to re-circulate in a bypass function and the engine load decreases 90%. The base engine control, shown in Fig. 2, uses a mechanical centrifugal governor system with a balancing spring to regulate the throttle's position. Engine speed is fixed and the typical HPW setting is 3600 rpm. The inherent characteristic of this type of control is to increase the engine speed when the engine load is removed. On average, this increase is 250 rpm above the load set point. When the engine is idling at 3850 rpm, it is noisy, unnecessary fuel is being consumed, and there is more engine wear. The Auto Throttle control system decreases these characteristics.

3. Auto Throttle Control for GCV High-Pressure Washer Engine

The fundamental purpose of the Auto Throttle control system is to sense an engine load demand. In the beginning of the development, five goals were set: reduce noise, reduce CO₂, minimize cost impact, increase component life, and create an automatic, self-contained system.

Figure 3 shows the components of the Auto Throttle system. The exclusive components developed for this system include a vacuum diaphragm and pivoting governor arm. Their function is to provide load-sensing feedback. Other contributing components include a centrifugal governor and supporting governor spring, a throttle return spring, and a ported carburetor insulator.

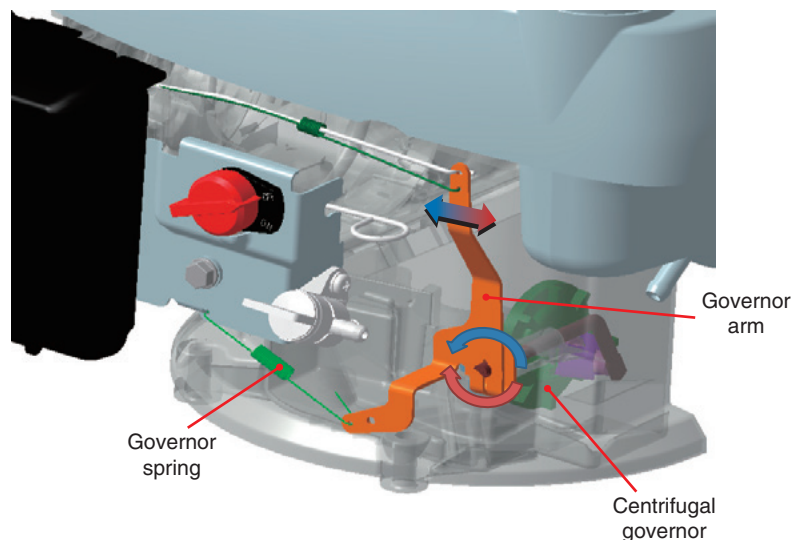


Fig. 2 Fixed Throttle control of GCV engine

3.1. Operation Cycle

Figure 4 shows the main operation cycle and noise level of a HPW using both the base fixed throttle and Auto Throttle systems. When the engines are loaded, the performance is the same. But when the spray wand is released and there is no pump demand, the base engine

increases engine speed while the Auto Throttle system reduces it.

3.2. Parts Function in the Auto Throttle Control System

3.2.1. Vacuum Diaphragm

Figure 5 shows a cross-sectional view of the diaphragm

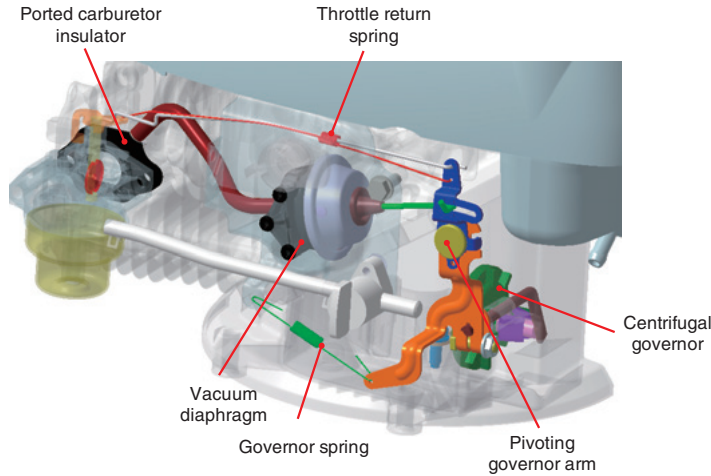


Fig. 3 Auto Throttle system components

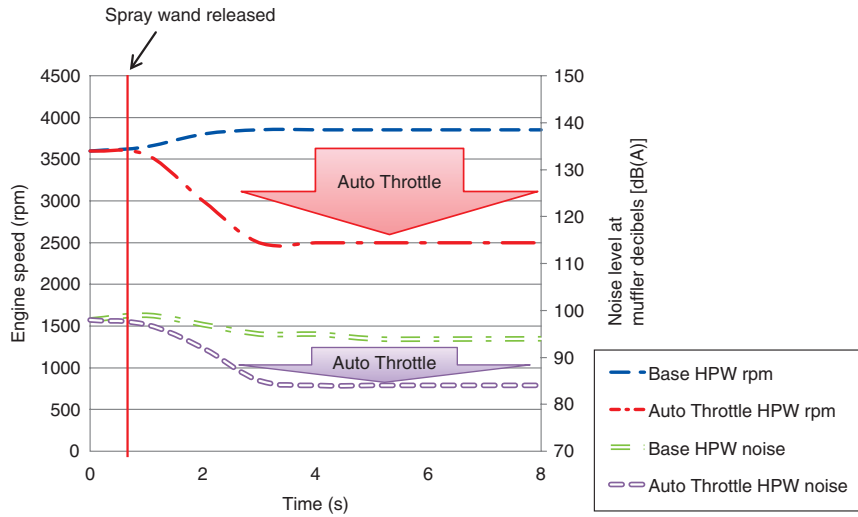


Fig. 4 Comparison of base engine and Auto Throttle engine noise

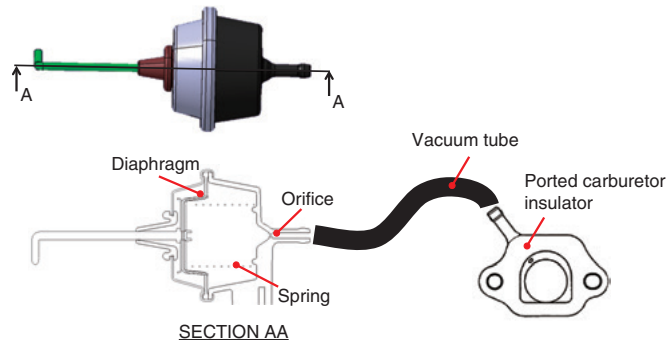


Fig. 5 Vacuum diaphragm construction

assembly. The function of the diaphragm is to convert engine vacuum into a mechanical force to close the throttle. Manifold vacuum pressure is supplied to the diaphragm through the ported carburetor insulator. As the engine load decreases, the internal manifold vacuum increases. When manifold pressure reaches approximately -6.6 kPa, the diaphragm begins to close the throttle. Diaphragm force is affected both by its stroke position and by the vacuum pressure signal of the engine intake. Since the application is a single cylinder engine, the vacuum pressure is always pulsating, which is undesirable for system stability and durability. Therefore, an orifice was added in the diaphragm inlet to reduce the pressure fluctuations inside the diaphragm body. The orifice diameter and diaphragm body volume have inverse effects on the pressure fluctuation and response time. By reducing the orifice diameter or

increasing the diaphragm body volume, it is possible to reduce the pressure fluctuation within the diaphragm. However, these changes also influence the response time of the engine. Once body volume was determined based on force requirements and packaging space, the orifice diameter was optimized through testing. Figure 6 shows a portion of the test results.

3.2.2. Centrifugal Governor

The centrifugal governor's primary purpose is to regulate engine speed at the rated load of the engine. As shown in Fig. 7, as engine speed increases, the centrifugal governor closes the throttle. The rotational movement is provided by the engine crankshaft. As the engine speed increases, the flyweights swing out and push the slider. The slider then generates a moment on the governor shaft

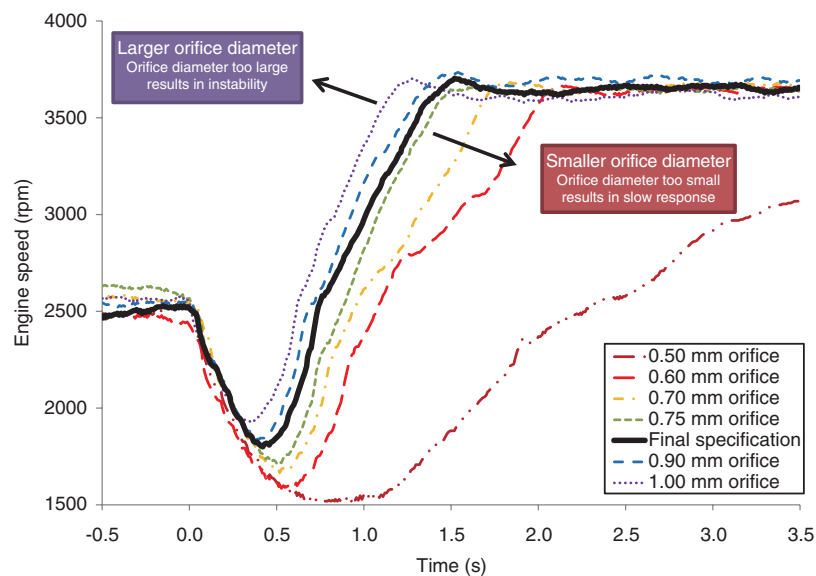


Fig. 6 Orifice effect on system response time

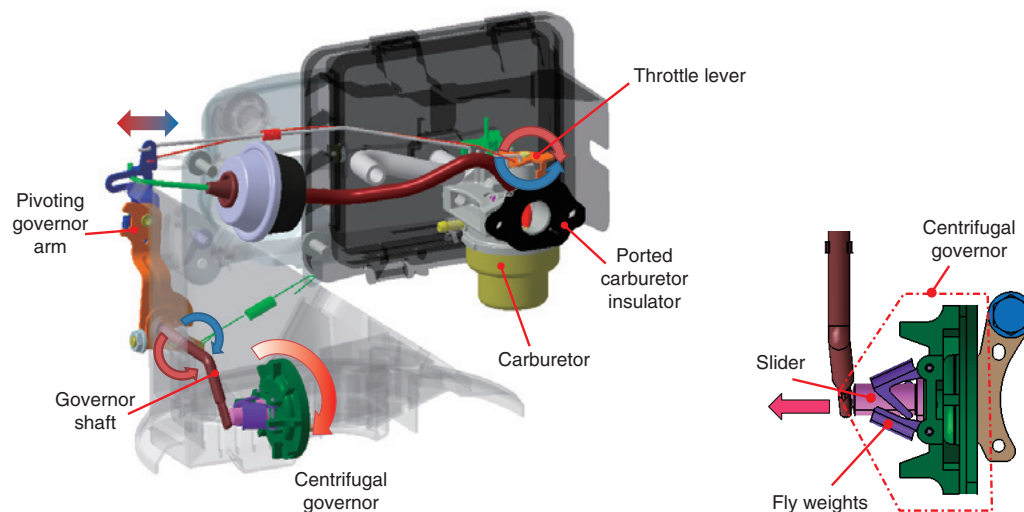


Fig. 7 Centrifugal governor configuration

that closes the throttle. The speed at which the centrifugal governor maintains steady state is balanced by the governor spring. Typical HPW setting is 3600 rpm while loaded. The centrifugal governor is unchanged from the base engine.

3.2.3. Pivoting Governor Arm

The governor arm's role in the system is to function as connection between input forces and regulation of throttle position. As shown in Fig. 8, the arm receives input from the centrifugal governor (F_G), diaphragm (F_D), governor spring (F_{GS}), and throttle return spring (F_{TS}).

A governor arm pivot was created to alter the moment induced by the governor springs when the engine is idling. The force of the governor spring is relatively high, and adding a pivot joint to the governor arm allowed the force required of the diaphragm to be smaller in comparison, thus reducing its overall size. The pivot allows a change in throttle position to be isolated from the governor shaft. As shown in Fig. 9, the optimum pivot travel was determined through testing. If the pivot angle is too great, the engine will not return to high speed from idle; if the pivot angle is too small, the engine is not stable in the idling condition.

3.2.4. Governor and Throttle Return Springs

Figure 3 shows the governor spring and throttle return spring. The governor spring is critical for maintaining the proper engine speed when the engine is loaded; therefore, it was important to maintain the basic spring design from

the base engine. Early in development, testing showed that this spring generated a moment that was too strong when the engine was idling, so it was not possible to balance the system to operate correctly in both idle and high-speed conditions without changing the base engine governor system. The governor arm pivot joint was added to keep the base engine governor spring and maintain a small diaphragm package. Consequently, a stronger throttle return

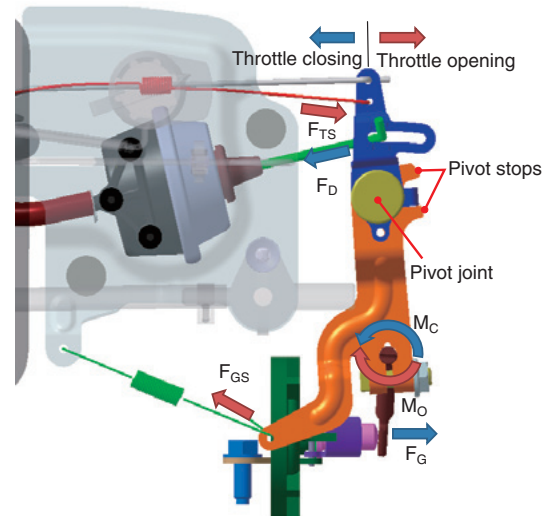


Fig. 8 Pivoting governor arm force summary

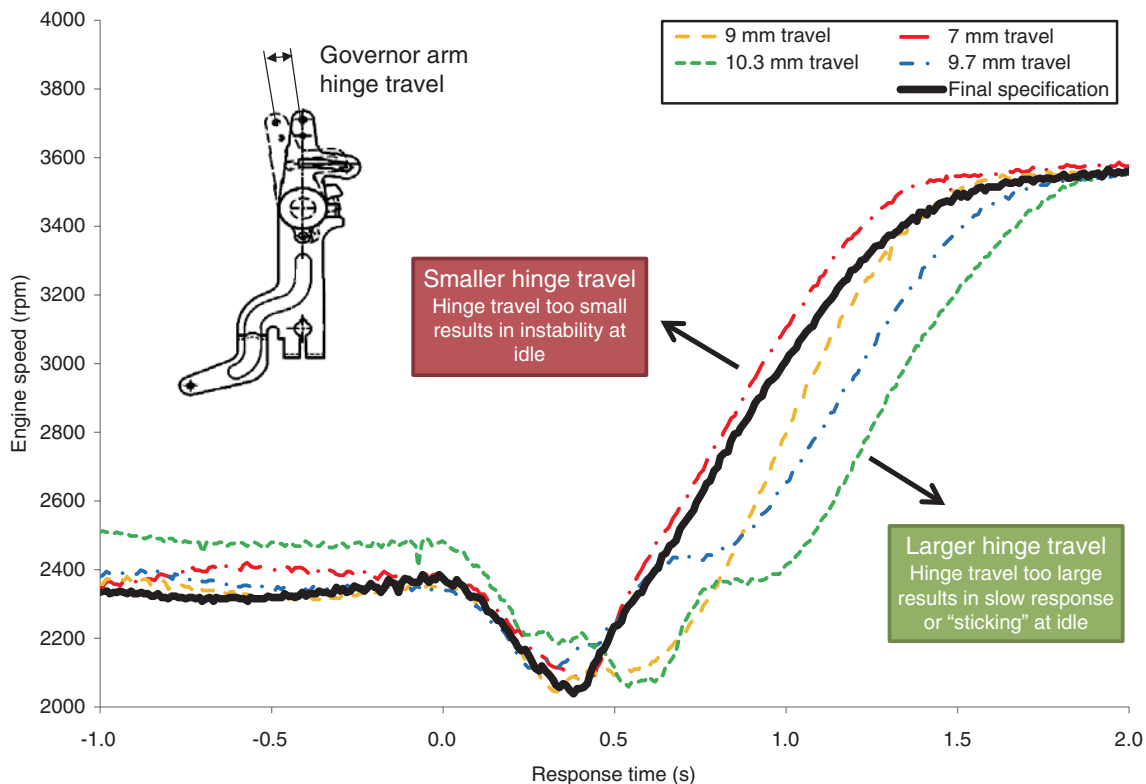


Fig. 9 Governor arm performance

spring was required to overcome the friction of the governor arm pivot joint.

4. Basic System Configuration

There are four forces that work to regulate the throttle position of the engine. As shown in Fig. 8, the diaphragm (F_D) and centrifugal governor (F_G) create a moment about the governor shaft that controls the closing of the throttle. The governor spring (F_{GS}) and throttle return spring (F_{TS}) create a moment about the governor shaft that controls the opening of the throttle. If the opening moment is too long, the engine cannot idle properly. If the moment is too short, the engine cannot return to high speed when idling. Figure 10 depicts graphically the Auto Throttle system controlling the engine speed. From $t = -1$ to 0, the throttle close moments ($F_G + F_D$) are longer than the open moments ($F_{GS} + F_{TS}$). At $t = 0$, the spray wand is activated, the engine load increases, and the vacuum signal weakens. During this transition from idle to high speed, the open moments overcome the close moments and the engine speed is then solely regulated by F_G and $F_{GS}+F_{TS}$. From $t = 2$ to 7, the engine operates at rated load. Rated load for the GC series engine varies from 50% to 80% throttle opening depending on the application. At $t = 7$, the spray wand is again

released. As the engine load is removed, the engine speed immediately increases, the throttle plate closes, the vacuum signal builds again, and F_D overcomes F_{TS} initially. As the engine speed is decreasing, F_{GS} and F_G also decrease, F_D pulls the throttle plate closed, the engine vacuum stabilizes, and the system is balanced at idle.

5. Performance Evaluation

5.1. System Design Verification

In order to determine an appropriate system response time, a survey was conducted using several Auto Throttle engines that were modified to exhibit differing response characteristics when the spray gun was engaged and released. Actual HPW customers participated in the survey and provided feedback on the response characteristics. Based on the results, a maximum response time criteria was determined for transient conditions when spraying water and not spraying water.

In the final product, any factor that changes the engine output power, which includes engine temperature, engine friction, air-fuel ratio, and barometric pressure, also affects vacuum pressure. Other factors that affect system operation include pressure washer pump load part tolerance, manufacturing variability, and assembly. To guarantee the

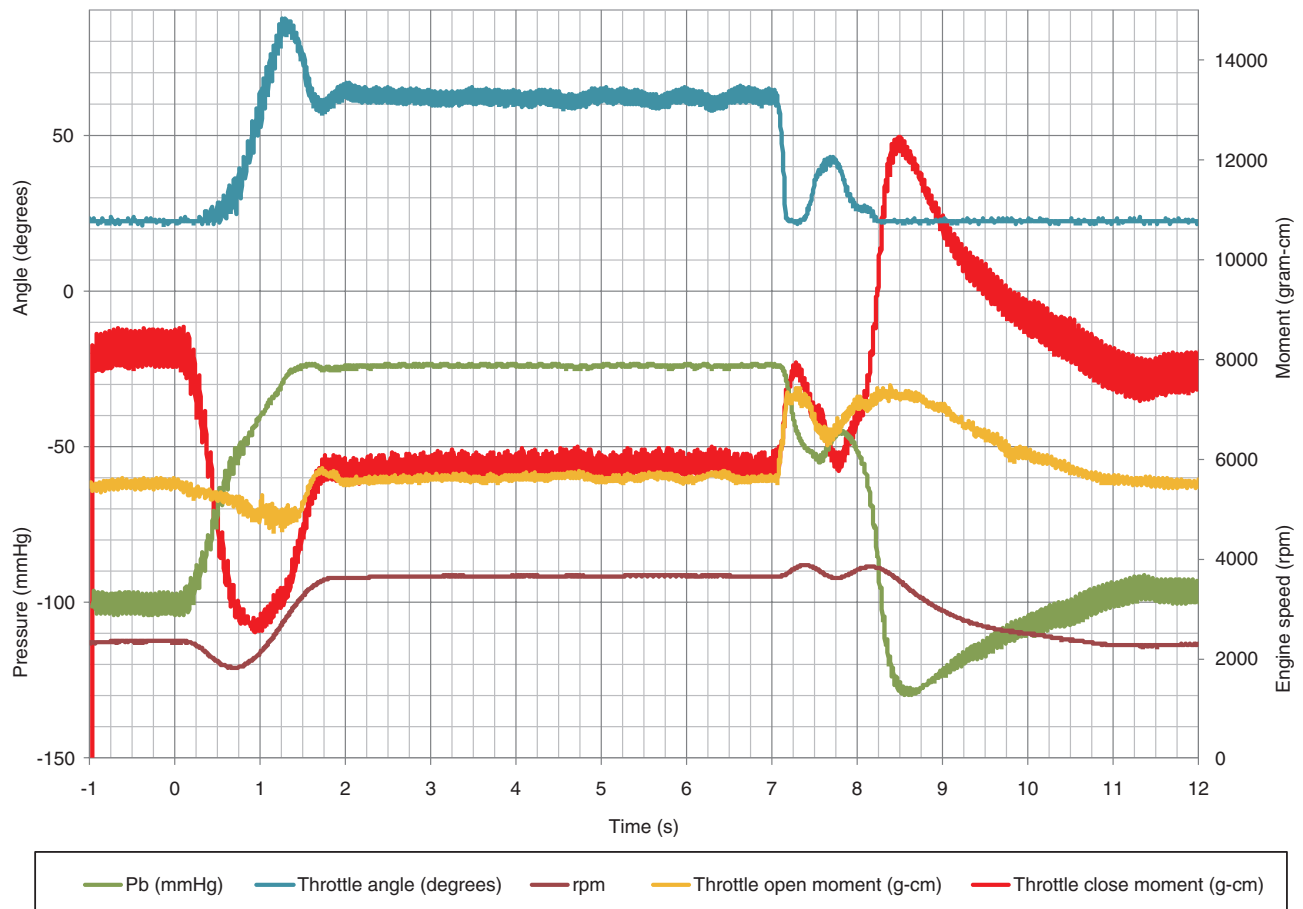


Fig. 10 Overall system performance

final performance of the Auto Throttle system, machines were built in “worst-case” configurations and verified to meet the design criteria. Figure 11 shows the testing results.

5.2. Noise and Fuel Consumption Enhancement

Two goals in the development were reducing noise and fuel consumption. The achievement level was determined by running the engine in high and low idle conditions

and measuring the differences. The sound pressure level was measured in a hemi-anechoic chamber following the guidelines described in ISO11094⁽¹⁾. Fuel consumption was measured by applying a steady load with a dynamometer and using a gravimetric flow meter. The loading conditions during these tests were set based on the typical loading applied by a HPW pump. Figure 12 shows the testing results.

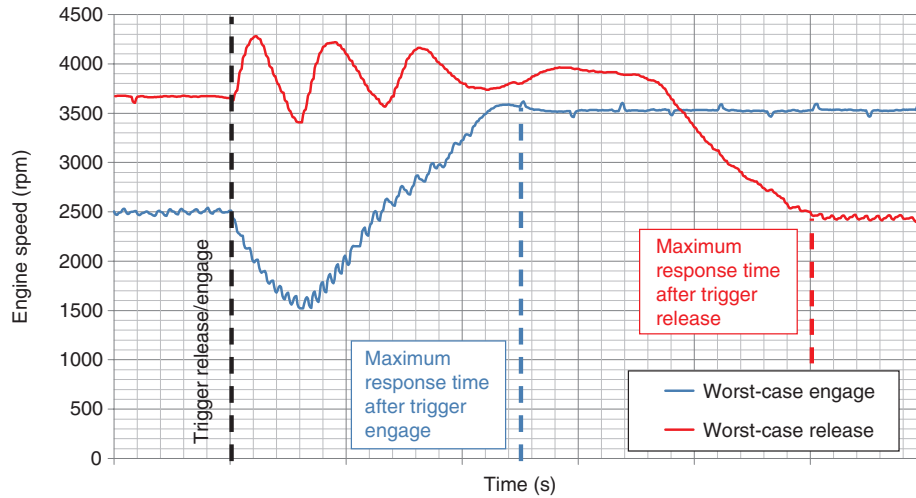
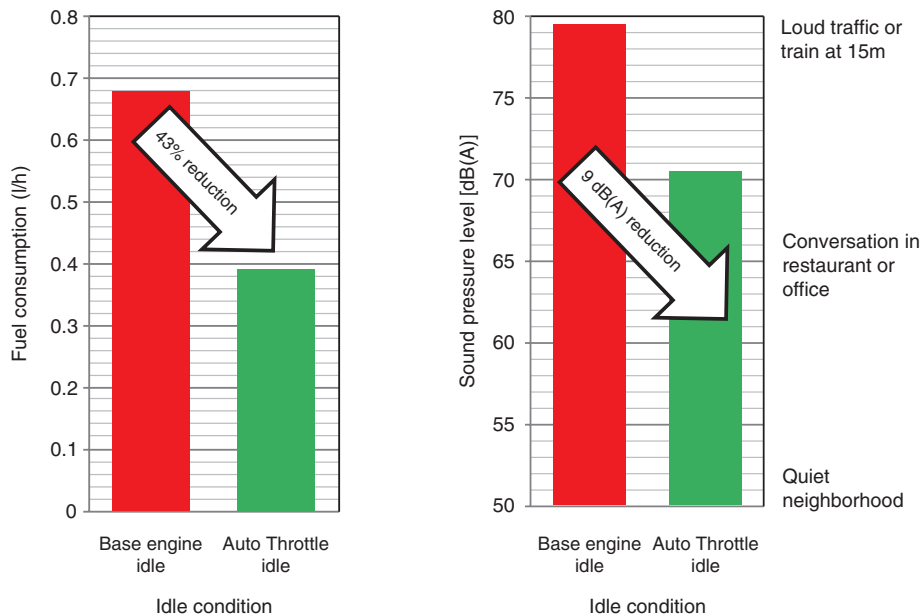


Fig. 11 Load on and off response time limits



Test condition		
	Engine speed (rpm)	Load (Nm)
Auto Throttle idle	2500	1.15
Base engine idle	3850	0.95

Fig. 12 Noise and fuel consumption achievements

5.3. Spray Gun Force Reduction

Another benefit of the Auto Throttle system controlling a HPW engine is the reduced level of reaction force (or recoil) that occurs when the operator activates the spray wand. The initial engine speed and water flow are less than the base model HPW; therefore, the operator does not have to apply as much force to counteract the recoil caused by water exiting the wand. Instead, the reaction force builds gradually as the engine speed increases. This force transition was measured by supporting the HPW spray wand with a load cell and recording the force generated when the wand is activated on a base model engine and an Auto Throttle engine. Figure 13 shows the test results.

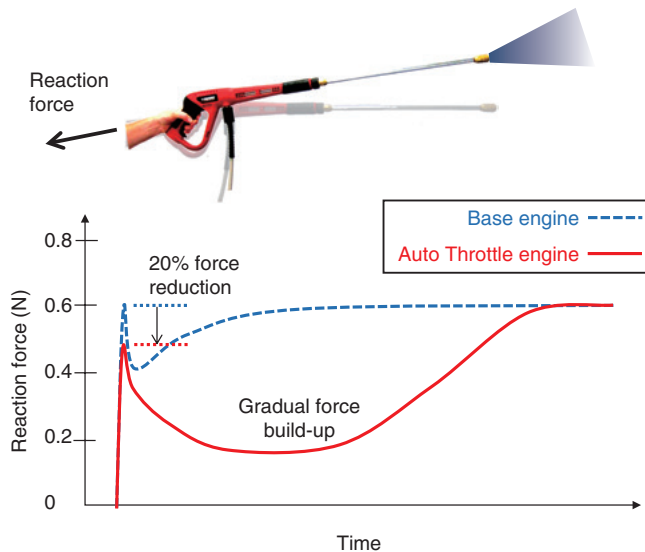


Fig. 13 Operator initial force

6. Conclusion

The Auto Throttle system accomplished the five development goals. The noise level of the engine in its idle state was reduced by 9 dB(A). This achievement is a result of reducing base engine speed from 3850 rpm to 2500 rpm using the Auto Throttle system. CO₂ emissions from the engine in its idle state were reduced. The emission reduction was confirmed by measuring the change in fuel consumption while the engine is idling. Idle fuel consumption was reduced by 43%. The system surpassed the initial cost reduction target by using common components between the base and Auto Throttle models. By reducing the engine idle speed, the Auto Throttle system achieves a 35% reduction in pump and engine revolutions in the idle state. Using engine vacuum as the actuation mechanism, the Auto Throttle system can automatically control the idle speed; no activation or adjustments by the operator are necessary.

In addition to the initial goals, the Auto Throttle system also achieved a reduction in user fatigue. This was

accomplished by a 20% reduction in the initial reaction force that the operator must support when the spray wand is activated.

The Auto Throttle system is an industry first, fully autonomous throttle control for a residential HPW. A patent application has been submitted for the achievement method of the Auto Throttle system. Currently, this type of control is being considered for other applications.

Reference

- (1) Acoustics — Test code for the measurement of airborne noise emitted by power lawn mowers, lawn tractors, lawn and garden tractors, professional mowers, and lawn and garden tractors with mowing attachments. ISO 11094:1991(E). Geneva, Switzerland: ISO, 1991.

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Development of Dual Clutch Transmission for VFR1200F

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ABSTRACT

The dual clutch transmission (DCT) for motorcycles that provides direct response to throttle operations without interruption of traction at shift was developed and installed on the VFR1200F.

This system has two separate hydraulic circuits for two clutches to provide individual controls. Weak creep control at departure, deceleration adaptive shift-down control, predictive cornering shift control, and other controls were developed to achieve the advanced performances in the vehicle departure and the shift change characteristic that are demanded for motorcycles. With those performances, riders can enjoy the pleasures of riding without depending on rider's skills. Compact design of this DCT system has made it possible to share the same crankcase raw casting materials with manual transmission (MT) type which is developed in parallel, and this design has enabled the installation on the common frame with MT system.

1. Introduction

There are two types of mechanism commonly used for the automatic transmission for large motorcycles. One is the belt converter type for scooters, and the other is HFT (Human Friendly Transmission)⁽¹⁾ which is applied to Honda DN-01. These are both categorized in Continuously Variable Transmission (CVT), and they provide smooth gear shifting for enjoyable driving performance. On the other hand, there are people who like conventional manually operated multi-step transmissions seeking the sporty feeling with linear and direct responses of the driving force to the throttle operation, which is one of the requirements for sports motorcycles. Having these two opposing requirements in consideration and pursuing the value of this new automatic transmission motorcycle which is not dependent on rider's skills, the world's first DCT⁽²⁾ system for motorcycles has been developed and installed on the VFR1200F. The present paper will describe the main technical points of this development.

2. Overview of VFR1200F

VFR1200F (Fig. 1) was developed with the aim of creating a large sports-tourer motorcycle that achieves high-level integration of advanced technologies and its original styling.

The engine (Fig. 2) adopts "UNICAM"⁽³⁾ that actuates both intake and exhaust valves with a single camshaft, making its cylinder head compact to concentrate the engine



Fig. 1 VFR1200F with dual clutch transmission



Fig. 2 External view of V4 engine with newly developed dual clutch transmission

* Motorcycle R&D Center

Table 1 VFR1200F specifications

Dimensions (L × W × H) (m)	2.250 × 0.755 × 1.220
Wheel base (m)	1.545
Weight (kg)	267
Engine type	Liquid-cooled 4-stroke OHC 4 valve 4-cylinder V-type
Engine displacement (cm ³)	1,236
Bore and stroke (mm)	81.0 × 60.0
Compression ratio	12.0
Max, power (kW/rpm)	127/10000
Max, torque (N-m/rpm)	129/8750
Fuel system type	Electronic fuel injected (PGM-FI)
Front tire size	120/70ZR17M/C(58W)
Rear tire size	190/55ZR17M/C(75M)

mass and realizing a lower center of gravity. The engine is V-type and has a cylinder bank angle of 76 degrees combined with a 28-degree phase pin crankshaft so that primary vibration is counterbalanced without any balance shaft. It provides powerful and flat torque in low-speed range while realizing smooth throttle response in high-speed range, together with the unique pulse feeling and exhaust sound of V4-engines.

The throttle-by-wire (hereafter TBW) system, which controls the throttle valve electrically, and the electronically controlled fuel injector (hereafter PGM-FI) enable good engine responses that reflect the rider's intentions.

Pro-Arm type is applied to the swing arm. The pivot and the propeller shaft are placed in an offset position. Variations in shaft length that arise from vertical movements of the suspension can be compensated by a constant-velocity joint located on the propeller shaft. This structure achieves both ride comfort and the driving stability as a sports-tourer.

Adoption of the DCT makes riders free from clutch operations that may be annoying when driving on city streets while allowing them to enjoy sport riding easily on winding roads.

Table 1 shows the main specifications of VFR1200F.

3. Development Aim of DCT

DCT has been available on the market for automobile applications. However, installing the DCT on a motorcycle comes along with challenges such as in making its size fit into the engine mounting space, while accommodating changes in attitudes of a vehicle (assuring the banking angle). It is also important to keep a good balance in longitudinal and lateral weight distributions with the arrangements of the parts. In addition, it is necessary to give good consideration to parts commonality with the basic engine structure because a motorcycle with manual transmission (hereafter MT)⁽³⁾ has been used as the base

vehicle. In order to resolve these issues and design the mechanisms optimized for motorcycles, the following development objectives have been defined:

- (1) To create the mechanisms that satisfy the smooth departing characteristics considering shift shocks which disturb the vehicle behavior.
- (2) To provide the shift management that matches the rider's feelings without discrepancy.
- (3) To use the same crankcase raw casting materials and to share the same frame with the MT type by keeping the same basic shafts arrangement and the spacing between shafts, as well as keeping the same basic configuration of the clutch and the transmission including the gearshift mechanism.

4. DCT System Configuration

4.1. Overview of the System

Figure 3 shows a cross-section of the engine developed here, and Table 2 shows a comparison of the specifications of MT and DCT. When you look at the differences between DCT and MT, it is apparent that DCT has two clutches and coaxial two main shafts together with four shift forks (one added to three). The width of transmission and shafts arrangement were made the same as MT type.

So far, Honda had applied a DCT system to TRX420FA (Fig. 4), an all terrain vehicle (ATV)⁽⁴⁾, which was put on the market in 2008. The DCT system applied to ATV is using a centrifugal clutch which is dedicated to the departure motion of a vehicle. However, for motorcycle applications, #1 clutch, which connects odd-numbered gear set, is used both for departing and for gear shifting to make the system light weighted and compact. Moreover, the system for ATV has only one hydraulic control circuit and, by switching valves, controls two clutches. However, in the application to motorcycles where precise control is demanded for both departing and shifting, two individual hydraulic control circuits are implemented for each of the clutches.

Table 2 Comparison of specifications of manual transmission and dual clutch transmission

		Dual clutch transmission	Manual transmission
Clutch type		Hydraulic dual	Mechanical single
Trans-mission gear ratio	Low	2.466	2.600
	2 nd	1.789	1.736
	3 rd	1.409	1.363
	4 th	1.160	1.160
	5 th	1.032	1.032
	Top	0.939	0.939
Number of main shafts		2 (Inner + outer)	1
Center distance (mm)		75	75
Engine oil capacity (lit.)		4.9	4.0
Clutch control pressure (kPa)		900	-

4.2. Features of the Clutch Unit

The clutch systems employ hydraulic control and are kept disengaged by the action of return springs when the oil pressure is zero. The oil pressure is introduced from the end of the inner main shaft and conveyed through feeding pipe to make the clutch get engaged.

The hydraulic pistons that generate the thrust for the clutches are situated in pairs with the pressure canceling volume chambers to counterbalance the pressure derived from the centrifugal force. The main parts of the unit are made by aluminum die casting for heat dissipation and

weight reduction (Fig. 5).

The structure of #1 and #2 clutches are basically the same but thicker clutch plates are implemented to cope with larger heat generation on clutch #1, because this clutch is used when vehicle is departing.

4.3. Features of the Transmission Unit

The basic configuration of DCT system is designed by modeling after the system of MT. The spacing between shafts and the width of the transmission system were kept the same between those two systems, allowing them to be

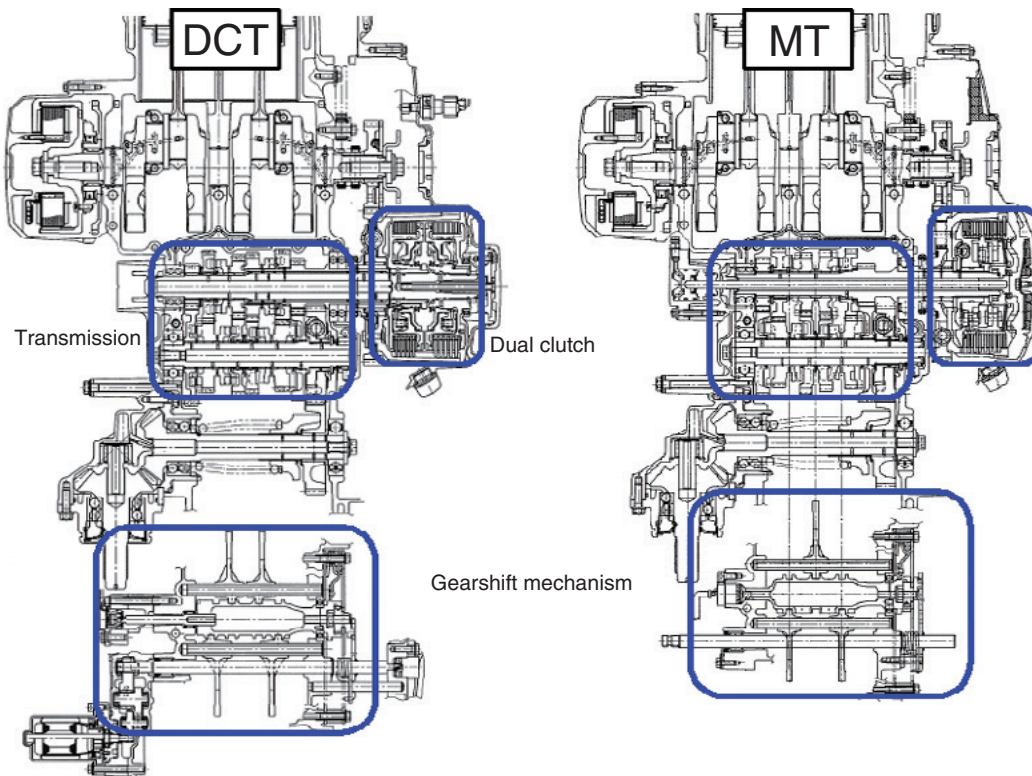


Fig. 3 Comparison of engine cross-sectional drawing

manufactured from the same raw casting materials. Inner main shaft and odd-numbered gears, such as first gear, third gear and fifth gear, are connected to #1 clutch while outer main shaft and even-numbered gears, such as second gear, fourth gear and sixth gear, are connected to #2 clutch (Fig. 6). It also adopts the same scheme of dog shifting as MT system in which gears themselves are moved along the shaft to establish direct coupling.

In the DCT system, so called pre-shifting is necessary where the next expected gear is connected in advance to get ready for the following shifting by switching of oil pressure to create the actual transitions of driving force. However, in the dog clutch type transmission systems, not like in the



Fig. 4 TRX420FA

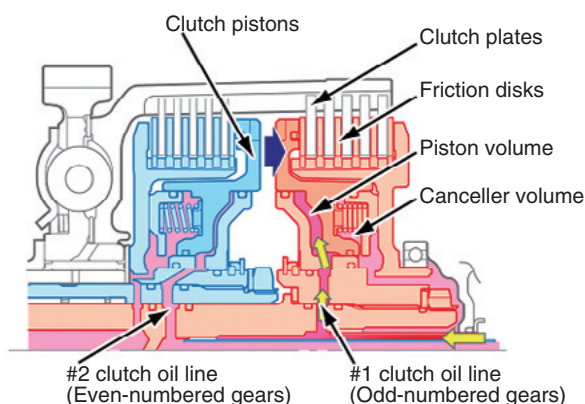


Fig. 5 Structure of dual clutch

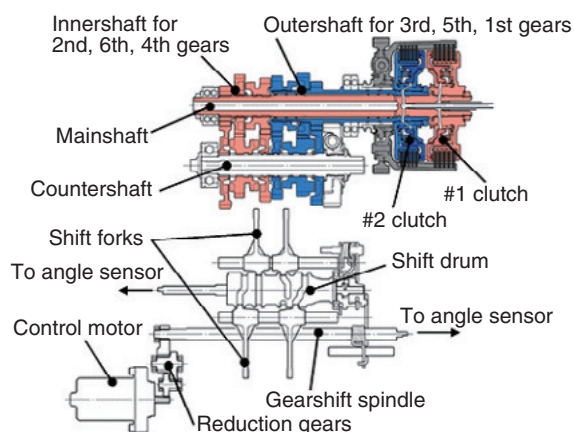


Fig. 6 Transmission and gearshift mechanism

Table 3 Shift drum position and gearshift conditions

Shift drum position #	Gearshift conditions	Engaged gear	
		Odd-numbered gears	Even-numbered gears
1	Neutral	N	N
2	Start, 1st	1	N
3	1st⇔2nd shift	1	2
4	2nd	N	2
5	2nd⇔3rd shift	3	2
6	3rd	3	N
7	3rd⇔4th shift	3	4
8	4th	N	4
9	4th⇔5th shift	5	4
10	5th	5	N
11	5th⇔6th shift	5	6
12	6th	N	6

case of the systems for four-wheeled vehicles equipped with synchronizing mechanisms, the impact noises are generated associated with the abrupt changes in revolutions of gears and shafts when they are engaged for pre-shifting. Even though the same impacting noise is observed in MT systems as well, it is important for DCT systems to take care of the reduction of the noise because this noise, when it's compared to MT systems, is more likely to be annoying as this noise generation, in this particular case, has no correspondences to the rider's operations. Accordingly, gear ratios, from first speed to third speed, have been changed from those of the MT system (Table 2) and the drive train damper dedicated to the DCT system is implemented and the noise transfer paths are optimized as the noise reduction measures. These measures contribute to the reduction of the shock as well.

4.4. Features of the Shift Mechanism

Based on the design for MT type, the shift mechanism is structured by changing means of operation for the shift spindles from a foot-manipulated shift pedal to the controlled electric motor with reduction gear train. The shift drum has twelve positions suited for neutral position, first speed position through sixth speed position and the positions in between that are to make both gears engaged for the adjacent positions to prepare for the shift transitions (Table 3).

In this system, gears out of driving force stands still in neutral position in order to keep the same response time for both shift-up and shift-down and to get good fuel economy by reducing the frictional losses in the non-driving clutch.

4.5. Features of the Hydraulic Circuits

The oil pressure for controlling of clutch piston thrust, which is generated by the oil pump specifically designed for DCT system, is branched into two paths and controlled by linear solenoid valves in the individual hydraulic circuits of the two clutches. Each circuit has its own oil pressure sensor to cope with the variations of engine oil, which works as operational fluid for hydraulic circuits. Those

oil variations are the differences in kinds of oil and the changes in oil viscosity that depends on its temperature. By arranging those hydraulic circuits all within the right-hand engine covering, it has been made possible to share the same crankcase with MT type (Figs. 7 and 8).

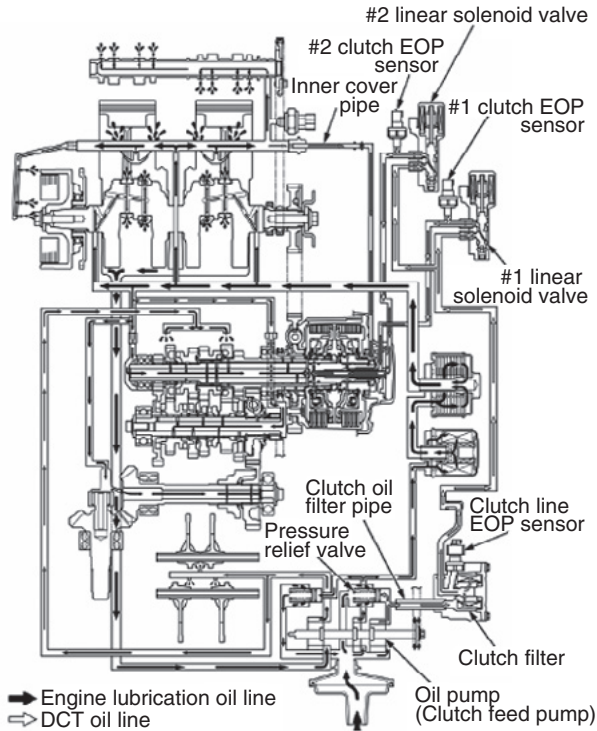


Fig. 7 Hydraulic circuits

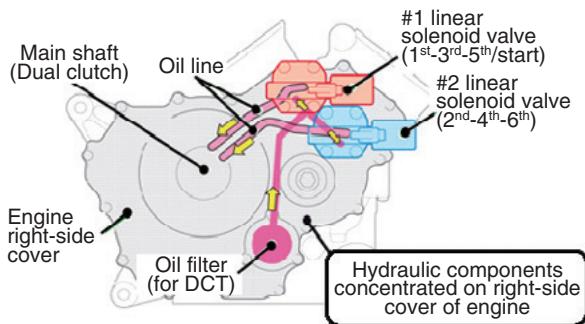


Fig. 8 Engine right-side cover layout

5. DCT Control

5.1. Overview of Control System

Figure 9 illustrates the configuration of input signals to the electronic control unit (hereafter ECU), such as signals from switches and sensors, and output signals from ECU. The rider's intentions, detected through throttle opening and operations of switches, the status of the engine speed, the status of the vehicle traveling, through vehicle speed, and the state of operation of DCT system, through revolution

speeds of each shafts, angel of shifting mechanism, oil temperature, clutch oil pressure, and so on, are monitored to calculate the targeting position referring to the shifting map followed by outputting signals that control the clutch oil pressure and the shift mechanism.

5.2. Measures Taken for Engine Oil Variations

Motorcycles ordinarily share the same oil for transmission lubrication and for the engine oil. This feature has also been applied to this system and the engine oil is used for hydraulic fluid and for lubricant as well. However, engine oil generally experiences greater changes in viscosity due to temperature variations when it's compared to automatic transmission fluid (hereafter ATF), making it necessary to have measures to deal with the changes in viscosity in flexible manner.

Therefore, this system adopts hydraulic feedback control system using oil pressure sensor, which continuously monitors oil pressure to clutches, and the function that estimates viscosity of the oil through characteristics of the oil pressure responses has been incorporated as well. The system is configured to provide a flexible response to oil temperature and viscosity changes by switching feedback gain based on the estimated oil viscosity. This minimized the influence of viscosity changes on the feeling of departing operations and the gear shifting operations, enhancing the responses of the hydraulic control system as well.

5.3. Control of Departure

Since the motorcycle rider must hold the vehicle steady using his or her foots when stopping, the system has to restrain the generation of creep-torque by disengaging the clutch when the motorcycle is in gear and stopped with throttle closed. However, it becomes challenging to make precise control of the torque transfer on the clutch if the hydraulically controlled departure clutch is under the control of without creep torque, because it is necessary to move the clutch piston with a large amount within a short period of time just after the departing operation is applied.

In order to resolve this issue, this system employs the weak creep control. Making use of the time lag in engine responses, this control method is devised to make the transition of the clutch piston complete within the time between the beginning of throttle opening to the point that the engine revolution goes up. First, when opening of the throttle is detected, the target pressure for the hydraulic clutch control is raised to the state where weak creep occurs. Since the throttle is already open at this time, there is no issue in terms of usability, even if the vehicle moves forward in a creep state. When there is the response of engine output followed by the increased engine speed, the target transfer torque is applied to the clutch in accordance with the throttle opening and the engine speed (Fig. 10). Applying this control sequence, smooth departure performance has been realized by providing the clutch

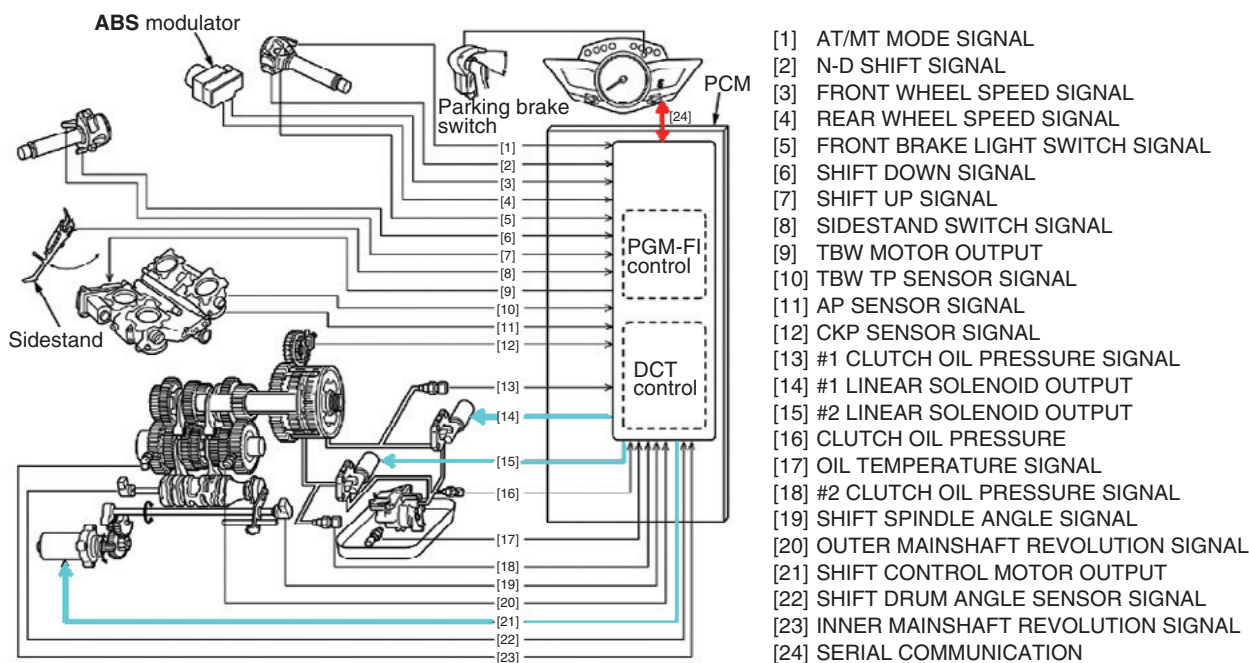


Fig. 9 Diagram of control system

status just ready to transfer the torque before engine speed goes up while ensuring disengagement of the clutch before departure.

5.4. Shift Operation

DCT shift operation occurs in the following order:

- (1) When a shift command is applied, the system makes the gear, which is to be activated in the following shift stage, engaged, establishing the status with both gears

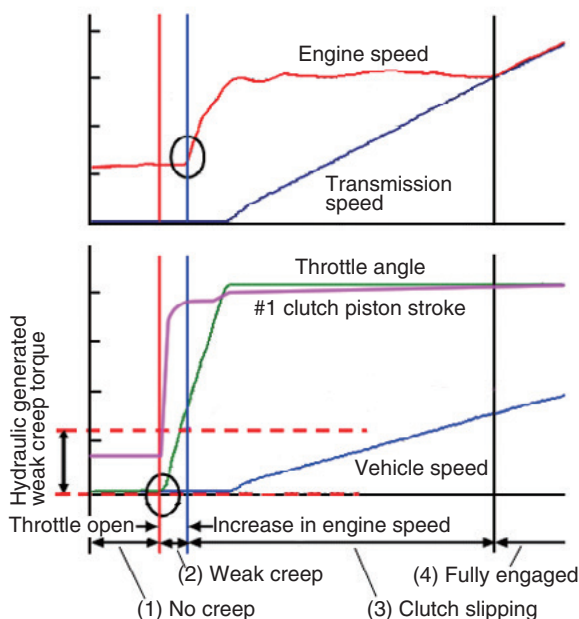


Fig. 10 Time chart during starting

engaged for one odd-numbered gear and one even-numbered gear (pre-shift).

- (2) By switching the clutch hydraulic pressure the engaging clutch is to be disengaged, and vice versa.
- (3) The drop in oil pressure in the clutch of non-driving side is detected by the sensor, and by disengaging the dogs on the non-driving side it is placed in neutral standing-by state. This completes the gear shifting.

The gearshift mechanism applied to this motorcycle is of the dog clutch type and, using this mechanism for the DCT shifting means, it gets the advantage of short pre-shift time compared to the synchromesh type mechanism generally used in four-wheel vehicles. Accordingly, in this system, the time lag before completion of pre-shift is about 70 ms which achieves the good performance in gearshift responses.

However, because of the structural nature of the mechanism, it is impossible to perform a jump-shift, where the system shifts the gears between two or more speeds at a time. Therefore, the time needed for shifting has been made short by constantly monitoring the outputs from the sensors and by coupling the gear shifting control with the hydraulic pressure control.

By making the operation time as short as possible through coordination of mechanical functions, it has achieved the interval of 500 ms or less between successive receptions of requests for gearshift (Fig. 11).

In MT systems, the interruption of the driving force is needed for gear-shifting by operations of clutch and throttle. On the other hand, in this DCT system, it has been made possible to make gear-shift without interruption of the

traction by switching two clutches in continuous manner. Figure 12 shows a conceptual view of this scheme. Maximizing the advantage of this feature, the system is controlled to get the smooth torque transition to the rear wheel by determining the optimum clutch transfer torque through the estimated values of engine torque and the differences between gear ratios before and after shifting. This enables to maintain the uninterrupted acceleration and to reduce the vehicle pitching at gearshifts.

5.5. Features of Shift Timing Control

Shift timing is basically determined by base shift-tables for individual driving modes referring vehicle speeds and throttle opening signals. As the driving modes, it has three modes, D mode, S mode and MT mode. D mode is designed by assuming city driving with consciousness of fuel economy. S mode is designed for driving with relatively high engine speed aiming at sport riding on winding roads. MT mode provides the rider's choice of gear positions that are changed by UP/DOWN switch on the left steering grip (Fig. 13).

In order to realize the performances of sport riding expected for motorcycles with an AT system as well, it is important for the system to select appropriate gears during the entire stretch of corner driving from the approaching at the entrance of a corner to the exit of the corner. Keeping that in mind, several controls have been added based on the analysis of gear shifting patterns of expert riders. These controls are described below.

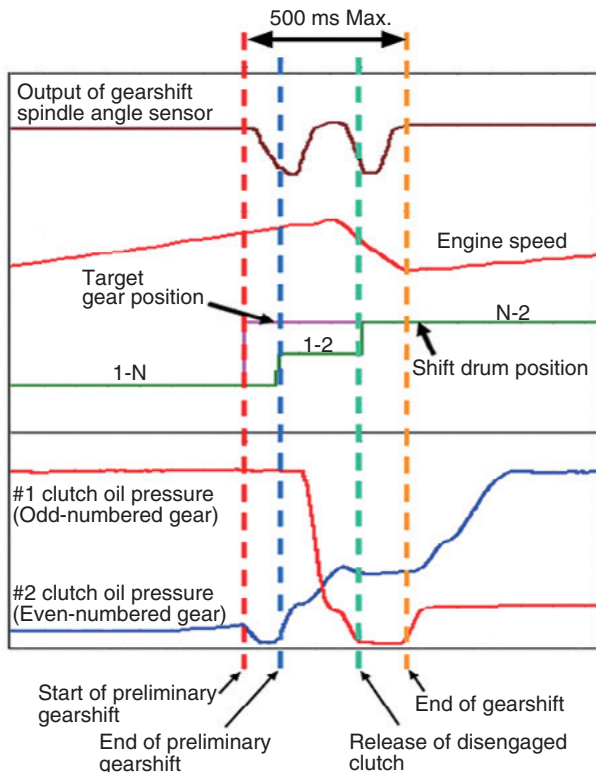


Fig. 11 Time chart of gearshift sequence

5.5.1. Deceleration adaptive shift-down control

It is important for sport riding to keep the stability of the vehicle with engine braking when it is heading into a corner by applying shift downs, and it also contributes to make a good preparation to get sufficient acceleration when getting out of a corner. To cope with this requirement, the system employs the control logic which makes early shift downs in accordance with the deceleration of the vehicle to realize a gearshift pattern similar to that of manual operations by real riders (Fig. 14).

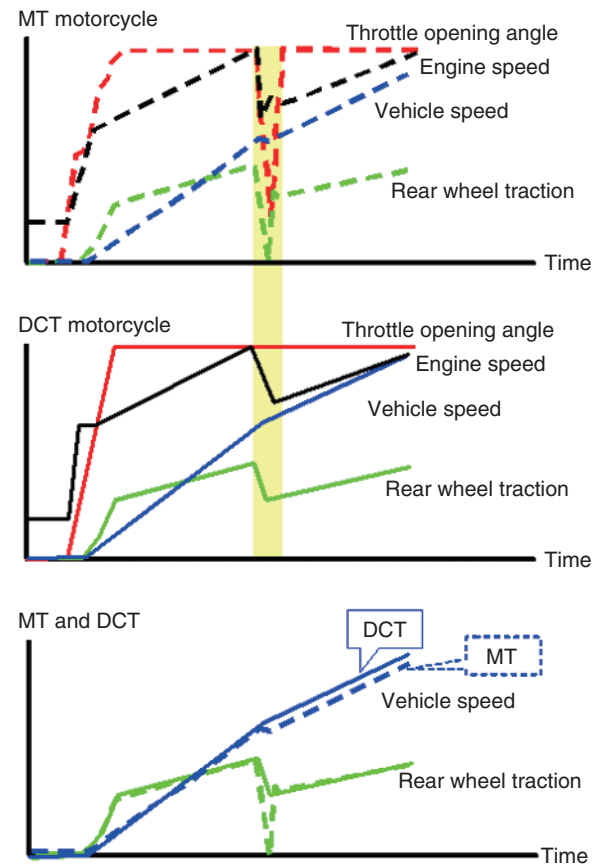


Fig. 12 Comparison between MT motorcycle and DCT motorcycle using time chart of gearshift

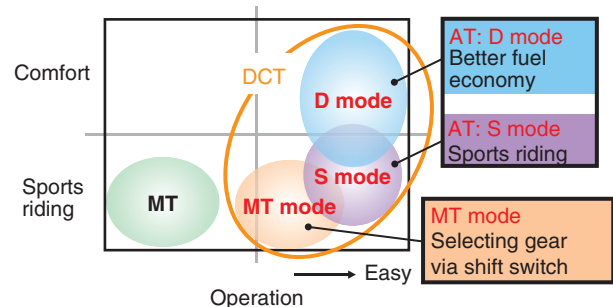


Fig. 13 Gearshift modes

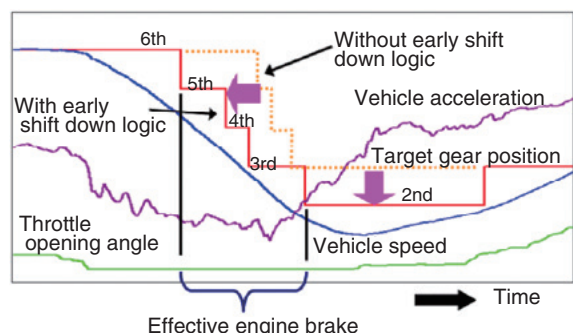


Fig. 14 Early shift down logic during deceleration

5.5.2. Predictive Cornering Shift Control

It is one of the features of a motorcycle which is leaned when driven in a corner. Consequently, frequent occurrence of abrupt gearshifts is highly unfavorable for motorcycles. However, it is challenging in motorcycle systems to get the information whether the vehicle is in cornering state while driving compared with four-wheel vehicles where the detection of cornering status is relatively easy by using, for examples, the steering angle and/or the rotational differences between left and right wheels. To overcome this issue, the quasi cruising status which appears while cornering has been noticed and the driving status has been divided into three stages, acceleration stage, cruising stage and deceleration stage. The logic has been introduced to suppress the shift-ups while cornering, by dynamically changing the shifting schedule when a transition occurs from acceleration stage to cruising stage or from deceleration stage to cruising stage. The adoption of this cornering detection logic enabled effective suppression of shifting operation during cornering without adding sensors or other such devices for bank angle detection (Fig. 15).

5.6. Blip Control

As rider's intention involves when the system initiates

gearshifts in MT mode, the control design can be put more emphasis on shorter shifting time. So called blip control has been implemented in this system in which, while downshifting, both clutches of odd-numbered and even-numbered are made disengaged during the shift operations and, having the coordination with TBW system, the engine speed is raised to meet the rotation speed of the next gear to be engaged (Fig. 16). With this blip control, the time needed for the completion of a gear shift has been made short from 500 ms to 170 ms compared with AT mode.

This control can provide sport feelings for inexperienced riders as well because riders can enjoy the blipping while shift-down that is usually performed only by expert riders.

5.7. Initial Learning of Clutch

The performance of the clutch engagement in each individual system varies because of the variation in the clutch engagement points caused by deviations of the

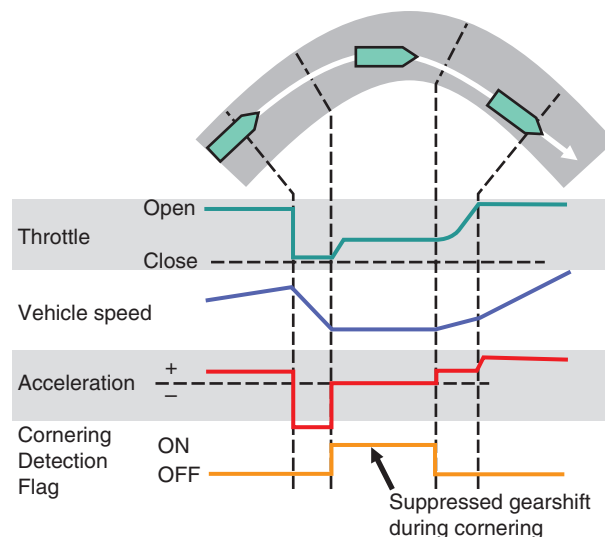


Fig. 15 Cornering detection logic

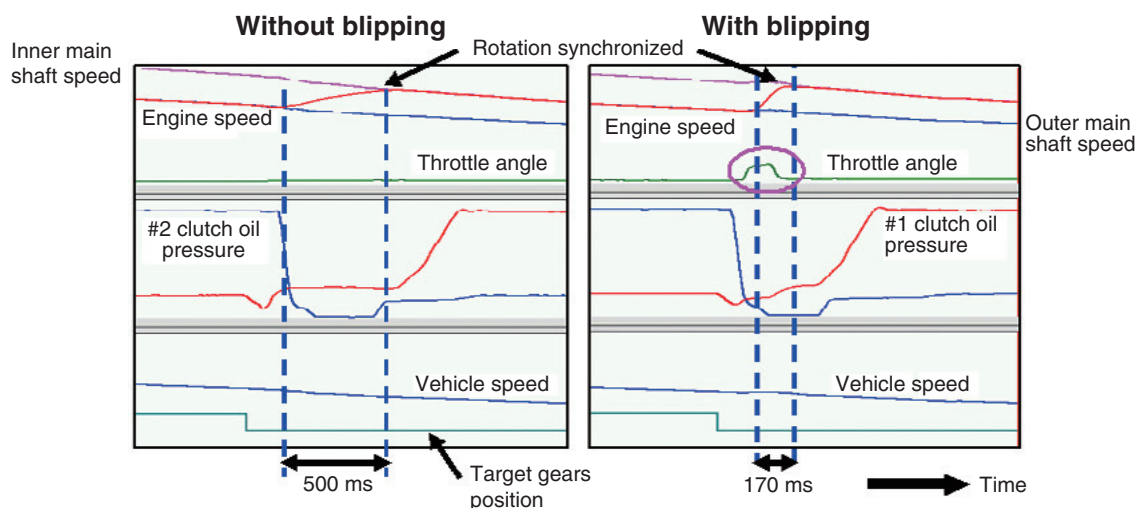


Fig. 16 Comparison of blipping effect

clutch disk clearance and the clutch return spring load. It is important to calibrate those deviations of individual systems because, in this DCT system for motorcycle, all the controls of clutch engagements are done based on the reference oil pressure level which corresponds to the clutch engagement point, and because highly precise clutch controls with good accuracy is needed in this system. In addition to that, the calibration of the system needs to be done easily in the service field when the clutch system is replaced for maintenance, as well as in the factory for pre-shipment adjustments. Taking those requirements in consideration, the initial learning method for clutches in DCT system has been developed.

The initial learning of clutch involves raising the target hydraulic pressure at the set speed. The clutch engagement point is then estimated from varying behavior of the actual hydraulic pressure and the value of that point is recorded in the ECU. The introduction of this calibration has enabled the uniform engagement performance across the entire tolerance range and contributes to the stable, precise clutch engagement control.

6. Conclusions

A DCT system for motorcycles, which provides direct responses to throttle operations without causing traction interruptions at shifts, was developed and installed on VFR1200F. This realized a motorcycle with new value. The main points of technologies involved are as follows:

- (1) Realization of smoothness both in vehicle departure characteristics and shifting operations by the optimized transmission ratios and damper characteristics, the weak creep control, and the control using initial learning of the clutch conditions.
- (2) Realization of the shift management without providing any feeling of discrepancy to the riders by predictive cornering shift control, the deceleration adaptive shift-down control and blip control.
- (3) Realization of common use of raw casting materials for the crankcase with the MT system and the installation on the same frame as MT's, by arranging all the DCT hydraulic control circuits within the right-hand engine cover, and by adopting the arrangement of the two clutches in series and the coaxial dual main shafts structure.

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Full-scale Wind Tunnel Equipped with Single-belt Rolling Road System

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ABSTRACT

Honda has begun operating a full-scale wind tunnel with a single-belt system. In order to simulate flow around a vehicle in actual motion, the wind tunnel is equipped with (1) a large-scale single-belt rolling road system, (2) a test section that can convert between open jet and adaptive wall systems, and (3) a model motion system that controls the attitude of a scale model. This paper gives an overview of the wind tunnel facility and gives an example of wind tunnel testing with and without an actual vehicle in place.

1. Introduction

With the growing demand for fuel economy felt in recent years, the role of aerodynamics has also increased. The demand for high-speed stability for the purpose of enhancing product market appeal and safety has grown as well, and the necessity for full-scale wind tunnels equipped with a rolling road system (RRS) has increased accordingly. Honda therefore formed plans for new construction of a full-scale wind tunnel with RRS, and began trial operation of the facility in April 2009.

The main features of this wind tunnel are the adoption of a single-belt RRS⁽¹⁾⁻⁽³⁾ and an adaptive wall system (AWS)^{(4), (5)}. A single-belt RRS can recreate the flow under the floor of a vehicle in actual motion, thus enabling measurement of drag and lift that are closer to a vehicle in actual motion than in a conventional wind tunnel with a fixed ground plate. Wall interference is caused by the phenomenon of acceleration (blockage) in the flow around the vehicle that is due to the ratio between the area of the test section and the area of the vehicle. The AWS can minimize this wall interference to the equivalent of an open jet, and this enables testing at 288 kph⁽⁶⁾.

The present article presents an overview of this wind tunnel and an example of the testing performed.

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2. Overview of the Wind Tunnel

2.1. Wind Path Design

Figure 1 shows an outline drawing of the wind tunnel. The wind path is constructed of steel-reinforced concrete and allows conversion between an open jet test section that has a nozzle size of 24.1 m² (maximum wind speed of 200 kph) for aerodynamic and acoustic testing of actual vehicles and an AWS that has a nozzle

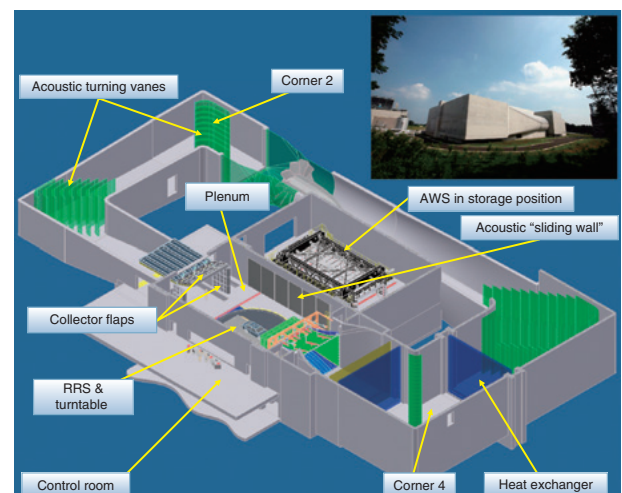


Fig. 1 Wind tunnel layout

size of 17.1 m^2 (maximum wind speed of 288 kph) for aerodynamic testing of actual vehicles and scale models. This test section conversion is performed by placing the AWS assembly inside an open jet plenum and closing the movable plenum wall.

2.2. Open Jet Test Section and Plenum

Figure 2 shows a photograph of an actual vehicle emplaced in the open jet test section. The open jet test section is 14.5 m long. Collector flaps consisting of three flaps (a top flap and two side flaps) are installed at the end of the test section.

The primary boundary layer control system (PBLRS) is located 2 m upstream of the open jet nozzle outlet (1 m downstream of the AWS nozzle). It has a scoop height of 60 mm.

Acoustic tests with an open jet configuration are conducted by covering the scoop with a fairing with the PBLRS either in operation or stopped.

Distinctive features of the test section are the RRS manufactured by MTS Systems Corporation of the United States and the scales of various types installed in the RRS. Details will be provided below.

The test section is enclosed by a plenum that is 24 m long, 16 m wide, and 12.4 m high. The plenum walls are enclosed in acoustic material manufactured by the German firm FAIST GmbH. The wall opposite to the control room is movable and consists of a wall of acoustic material panels suspended from above. These panels are opened and closed when the AWS is moved between storage and the plenum.

2.3. AWS Test Section

Figure 3 shows a photograph of an actual vehicle emplaced in the AWS test section. The AWS provides for a maximum movement of 200 mm in the side walls and ceiling wall so as to reproduce in the wall geometry the flow lines around the vehicle when it is in motion. This is a device that can minimize the influence of blockage between the emplaced vehicle and the walls. The AWS test section is 17.5 m long and the cross-section area of the

inlet is 17.1 m^2 . The initial position of the walls provides an enlarged shape, in consideration of the developed portion of the boundary layer, so that the cross-section area of the test section outlet is 17.7 m^2 . The AWS is made up of the steel support pillars on the outside of the test section and clear acrylic panels in the side and ceiling walls. The side walls have 30 acrylic panels (15 on each side) that are 3 m high, 1 m wide, and 25.4 mm thick. The ceiling wall is made up of 15 acrylic panels that are 1 m long, 3.5 m wide, and 25.4 mm thick. In order to make the walls movable, the support pillars are mounted with 30 electromechanical actuators for the side walls and 30 for the ceiling.

Figure 4 shows an example of the effect of the AWS. The flat wall in the figure indicates the initial configuration of the AWS for comparison with the static pressure distribution along the central line of the vehicle when an actual vehicle is emplaced in the open jet and AWS. It is apparent that the static pressure distribution after operation of the AWS coincides with the distribution of the open jet. The static pressure distribution in the initial configuration before operation of the AWS is lower than the results after AWS operation and the open jet because the influence of blockage causes acceleration of the flow around the vehicle.



Fig. 3 AWS test section view from upstream



Fig. 2 Open jet test section view from upstream with collector in background

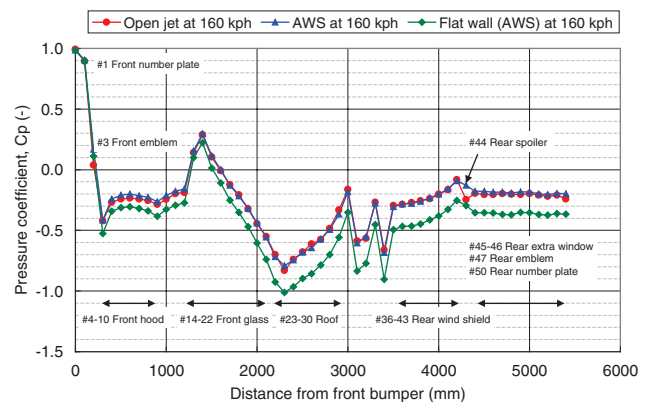


Fig. 4 Static pressure distributions along vehicle surface

2.4. Fan

The fan shown in Fig. 5 is manufactured by the United States firm FläktWoods. It is driven by a motor that provides maximum power of 3.6 MW (manufactured by TMEIC), and it generates a maximum flow volume of 1400 m³/s. The fan is 8 m in diameter and is made up of 24 rotor vanes made of carbon fiber-reinforced plastic and 35 stator vanes made of steel.

2.5. Heat Exchanger

The heat exchanger is located 9.8 m downstream from the outlet surface at corner 3. This heat exchanger consists of a combination of two rows of circular pipe and a horizontal fin. A mixture of 30% glycol in water is used as a cooling solvent.

2.6. Boundary Layer Control System and Single-belt Rolling Road

The boundary layer control system in the test section is made up of the scoop-shaped PBLRS described above and the secondary boundary layer control system (SBLRS), which uses a suction and tangential blowing system. The SBLRS has slots in the upstream roller portion of the RRS, and its width is the same as that of the belt at 3.2 m. The PBLRS uses a high-speed diffuser (HSD) for reinjection, and the SBLRS operates by tangential blowing from the upstream end of the belt immediately after suction.

Figure 6 shows the single-belt RRS. The present wind tunnel has a single belt that has horizontal force



Fig. 5 Main fan with 9 inlet guide vanes, 24 rotor blades and 35 stators

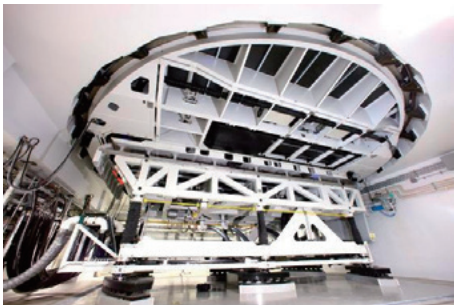


Fig. 6 Rolling Road System and turntable

measurement (HFM) mounted. This HFM is a system that is installed on structural members of the RRS; that conducts highly accurate measurement of the horizontal aerodynamic force exerted on the RRS, which floats using oil bearings; and that does so without measuring the resistance internal to the vehicle.

The stainless steel belt of the RRS is 3.2 m wide, 9 m long (the distance between rollers), and 1 mm thick. It operates at a maximum speed of 300 kph, and is operated so that the belt speed is synchronized with the wind speed in the wind tunnel. The RRS is synchronized with the turntable, which is 12.3 m in diameter, so as to enable testing with $\pm 10^\circ$ yaw angles in order to simulate crosswinds.

The lift and downforce exerted on the vehicle can be measured independently in the four wheels by load cells installed under the stainless steel belt. There are load cells for use with actual vehicles and load cells for use with scale models. The load cells are placed in an eccentric system so that they can handle vehicles with different wheelbase and treads.

The following is one example of testing that involves the boundary layer control system and RRS.

Figure 7 shows a comparison of results from the main manufacturers for static pressure distribution in the X-axis direction along the central axis of the wind path. The graph was created on the basis of reference sources^{(2), (7), (8)}. In the present wind tunnel, the static pressure gradient is held at even levels at the vehicle emplacement location.

Figure 8 shows a comparison of boundary layer distribution at the center of the belt for different boundary layer control conditions. It is apparent that the use of three stages of boundary layer control (PBLRS, SBLRS, and RRS) yields an even distribution.

Figures 9 and 10 show the total pressure distribution

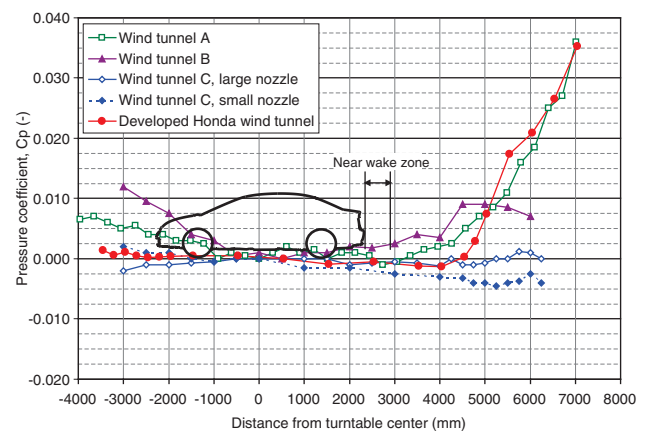


Fig. 7 Axial static pressure distributions along centerline of empty test section in selected wind tunnels (Outline of 4.5 m long test vehicle centered at $x = 0$ is shown on open jet mode)

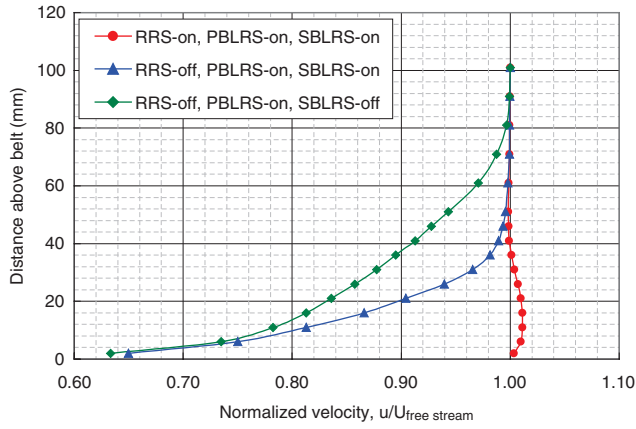


Fig. 8 Boundary layer velocity profiles at $x = -2000$ mm at 200 kph for three boundary layer control conditions on open jet mode

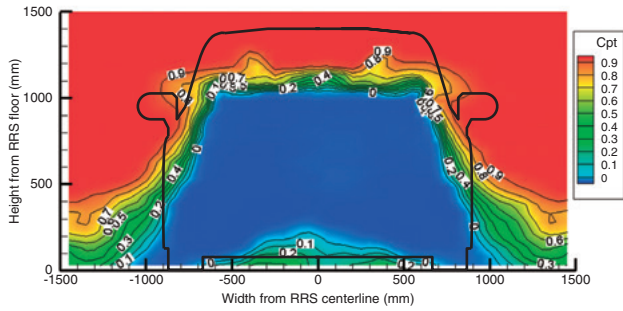


Fig. 9 Total pressure coefficients in cross-section at 10 mm behind vehicle with RRS-off at 100 kph on open jet mode

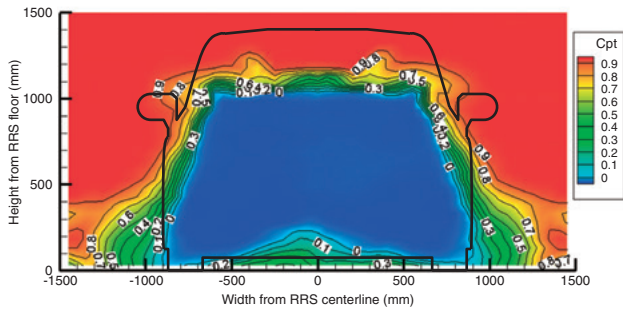


Fig. 10 Total pressure coefficients in cross-section at 10 mm behind vehicle with RRS-on at 100 kph on open jet mode

measurement results at the rear of an actual vehicle when the RRS is off and when it is on. It is apparent that the wake spreading out from the tires varies according to whether the RRS is operating or not.

2.7. Model Motion System

The model motion system called the hexapod, which is equipped with six electromechanical actuators, was also manufactured by MTS Systems Corporation of the United States. This system can operate scale models through

Table 1 Wind tunnel aerodynamic performance summary

Test parameter		Open jet results	AWS results
Total pressure uniformity		$\Delta Pt/q = 0.08\%$ (1σ)	$\Delta Pt/q = 0.10\%$ (1σ)
Static pressure uniformity		$\Delta Ps/q = 0.33\%$ (1σ)	$\Delta Ps/q = 0.19\%$ (1σ)
Longitudinal turbulence intensity, u'/U		0.20% (at 200 kph)	0.199% (at 288 kph)
Flow angularity		$\Delta\alpha = 0.09^\circ$	$\Delta\alpha = 0.10^\circ$
		$\Delta\beta = 0.09^\circ$	$\Delta\beta = 0.10^\circ$
Boundary layer profile	$z = 2$ mm	$ u-U /U = 0.82\%$	$ u-U /U = 0.15\%$
	$z = 50$ mm	$ u-U /U = -0.42\%$	$ u-U /U = 0.18\%$
Sound pressure level	100 kph	56.2 dBA	
	180 kph	71.8 dBA	

movement in multiple degrees of freedom, including pitch, roll, and heave. The yaw motion system is placed inside the struts and its operation is synchronized with the turntable movement.

2.8. Airflow Measurement Results

The airflow measurement results are summarized in Table 1. The open jet and the AWS both show figures that are on a par with or that exceed performance at satisfactory levels for actual use.

3. Conclusion

The distinctive features of this wind tunnel were introduced and an example was given of aerodynamic testing with and without a vehicle.

As a device capable of simulating flow around a vehicle that is in actual motion, this wind tunnel can be considered a test device that is effective in reducing vehicle air resistance and enhancing driving stability.

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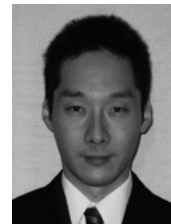
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New MBD Software Development Process for Electronic Control AWD System

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ABSTRACT

A new model-based design (MBD) software development process was constructed for the development of electronic control all-wheel-drive (AWD) systems. This new process enhances system control design with the aim of increasing specification quality compared to the previous software development process that applies executable model-based control design specifications. The new process changes the verification process and adjusts the division of development duties between system control design and software detail design. In addition, a method for generating the related specifications was constructed, and plug-in structuring was performed for applications to increase the specification reuse rate.

The new process was applied to the development of software for a new mass-produced electronic control AWD system, and resulted in a 50% reduction in the volume of specification changes compared to the previous software development process.

1. Introduction

Recently, the complexity and scale of electronic control systems and control software mounted in vehicles is increasing. However, demands to increase the reliability of control, reduce development times, and increase development efficiency are also increasing. Therefore, the introduction of MBD specifications based on executable models is proceeding as a means of developing software and describing specifications, and model description guidelines are also being established⁽¹⁾.

A general-purpose automatic code generation tool⁽³⁾ and a data setting tool developed internally by Honda⁽⁴⁾ have been integrated with MATLAB/Simulink and Stateflow,⁽²⁾ and MBD specifications based on fixed-point data model expression have been applied to develop electronic control AWD systems from 2007 mass-production models. These MBD specifications have been applied to increase system developers' understanding of specifications, reduce inconsistency between MBD specifications and software, and facilitate software design from the specifications created by system control designers to the actual code

generation. To reduce the verification load for model and code generation, an MBD software development process has been constructed that limits the types of blocks that can be used as MBD specifications, restricts the data setting values for the parameters and I/O applied to the blocks, and specifies the data names and other description methods.

This process is expressed by a V-process as shown in Fig. 1. When creating an MBD specification model, the system control designers set fixed-point data defined within the model. However, these data settings require software design skill to perform the appropriate scaling. In addition, the process to set the data in the model involves not only making the data settings but also assembling the data descriptions and other incidental information, so generating MBD specifications required enormous amounts of time. However, these design tasks also fall under the scope of software detail design, and the division of duties between system control design and software detail design was unclear.

In addition, this process enabled system control designers to generate MBD specifications while checking the consistency of the requirement specification and the

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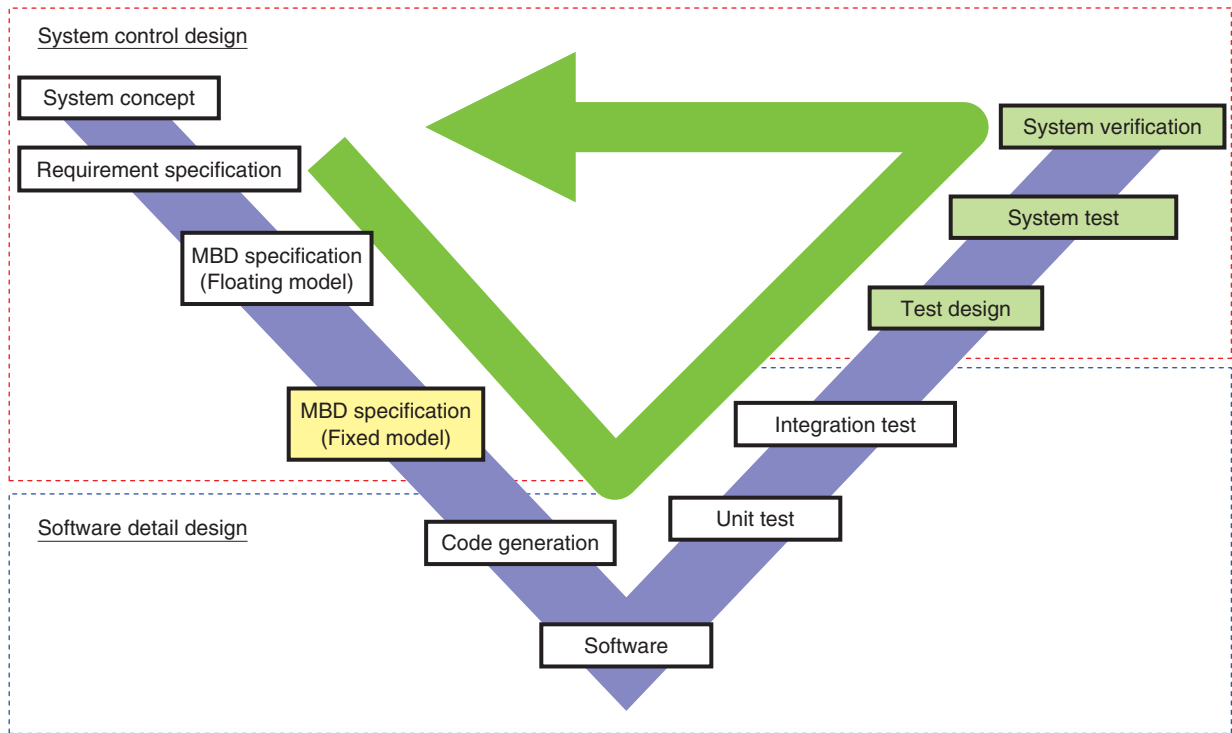


Fig. 1 V-process of previous MBD process

model, but means of confirming the correctness of the specifications were not defined, and there was also no system for managing the operation check results in the software design process of the V-process. This resulted in waste as the same tests to check operation when generating the MBD specification were performed again during system verification, which is the final software development process, and there were also cases where specification imperfections that could not be detected by the operation check when generating the MBD specification were discovered during system verification. The specification correction loop in these cases was quite large, which increased the ratio of software development due to specification changes in the latter stages of system development, and raised the issue of whether executable MBD specifications had sufficient merit for use.

2. Aims of New MBD Software Development Process

The following methods were considered for the new MBD software development process to address the issues with the previous MBD software development process and enhance specification quality.

- (1) Introduction of front loading verification using a floating-point data model
- (2) Creation of a specification format for enhanced system control design
- (3) Development of a tool for extracting MBD specification data

- (4) Plug-in structuring of applications to increase the specification reuse rate

These methods aimed to increase software development efficiency, reduce specification corrections, and reduce the ratio of the software development load in the latter stages of system development.

3. Front Loading Verification

3.1. V-process Type New MBD Software Development Process

In contrast to the previous V-process shown in Fig. 1, the new MBD software development process uses the V-process shown in Fig. 2. The changes from the previous process are the following two points. First is that front loading verification is performed before shifting from system control design to software detail design. Second is that the MBD specification generated by system control designers is expressed by a floating point data model, and the fixed-point data model and data scaling have been transferred to the software detail design area.

The aim of performing front loading verification before shifting to software detail design is to prevent outflow of imperfections with the requirement specification and MBD specification and inconsistency between these two specifications to downstream processes, and to increase development efficiency by reducing the size of the correction loop. In addition, the verification process is divided, enabling reduction of the system verification load, which is the final system development process.

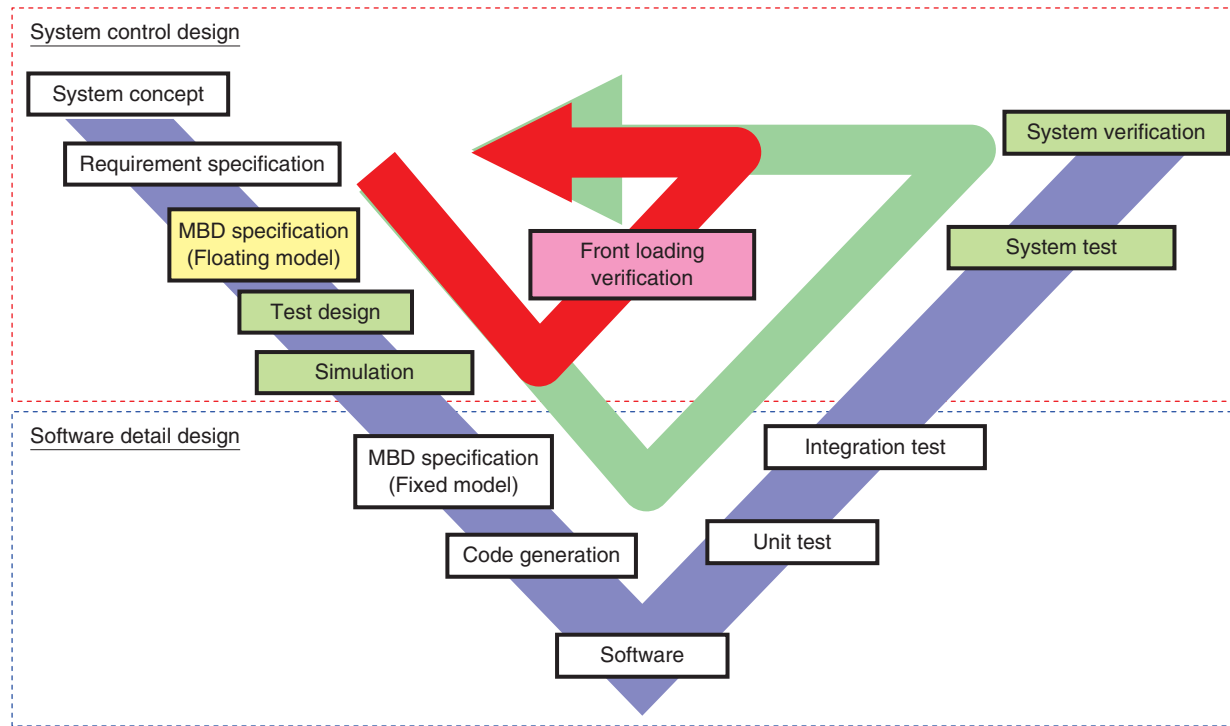


Fig. 2 V-process of new MBD process

Keeping the MBD specification as a floating-point data model when performing front loading verification reduced the data setting processes and secured efficient flow in the loop from simulation to front loading verification. This resulted in a verification process focused on system control design.

3.2. Dispersion of Verification Process

The new MBD software development process disperses the verification load by dividing the verification process into the two parts of front loading verification and system verification using an actual vehicle as shown by the V-process in Fig. 2.

Front loading verification performs simulations and theoretical verification to check the steady state control operation, make sure the requirement specification and the MBD specification are consistent, and verify the validity of the parameter design values. System verification using an actual vehicle evaluates the vehicle dynamics performance, verifies the conformance with the vehicle environment, and verifies the transient states (Fig. 3).

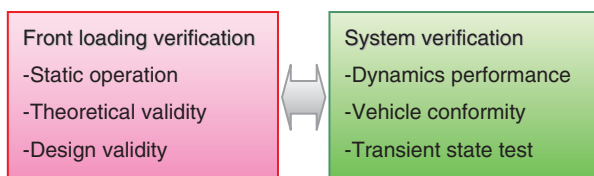


Fig. 3 Division of verification items

The new MBD software development process generates a test design specification for the front loading verification items and system verification items at the same time as the MBD specification is generated. In the front loading verification event, the various verification methods and the front loading verification results are reviewed. By reviewing front loading verification in this manner, including system verification methods using an actual vehicle, system control designers and verification approvers can deepen their understanding of system verification process test contents and check the validity of the test contents. This approach aimed to reduce the discovery of imperfect verification contents as well as imperfect specification requirements and inconsistency with the software during system verification, which is the final system development process.

In addition, in the previous process, the data introduced to the software in the initial development stages was not precisely set data, and the parameters were often adjusted while conducting tests using an actual vehicle. This meant that parameter changes sometimes affected other specification areas within the system or other systems. In the new process, specifications are verified and logical data is set in the software even in the initial development stages, which reduces parameter changes and repetitive work such as checking the effects of that software.

4. Specification Format Configuration

The new MBD software development process provides the following four formats in addition to the MBD

specification to promote consistent verification from system concept design through system verification (Fig. 4).

- (1) System concept sheet
- (2) Requirement specification
- (3) Test design specification
- (4) Test report

The features of each format are described below.

- (1) System concept sheet

This format is described together with the requirement specification of the functions to be introduced. It describes the purpose, the background and concept of the requirements, the events leading to the requirements, and other know-how.

- (2) Requirement specification

This specification clearly describes the specification requirements to the software detail design side. It excludes the background and system concept information from the system concept sheet, and describes only the purpose, requirements, and conditions of the functions to be introduced. The detailed version of this requirement specification becomes the MBD specification.

- (3) Test design specification

The verification review describes the verification methods and parameter setting methods beforehand, and

presents the verification results obtained in accordance with those methods. To facilitate checking of inconsistency between the requirements and the actual contents, the format also describes the requirement items, verification methods, viewpoint from which verification is performed, and parameter setting methods. At this time, the verification methods and parameter setting methods are divided into front loading verification items and system verification items, and are described so that the respective contents can be linked.

- (4) Test report

This format describes the front loading verification results and system verification results in a single sheet, and also describes front loading verification results that carry over to system verification, and other items. This helps prevent verification omissions between front loading verification and system verification.

5. Automatic Extraction of Data Information within Models

During system control design, the MBD specification is expressed by a floating-point data model. However, a data scaling standard is needed for software detail designers to create the software. In addition, system control designers also require scaling and other settings for certain data in order to perform appropriate control. Therefore, the following data settings are requested of software detail designers.

- a. Accuracy requirements for the final output of each application
- b. Parameter setting scaling requirements

This is accomplished by generating a specification (hereafter, “data setting requirement sheet”) that prescribes the data setting requirements for the above two items separately from the MBD specification, and providing it to software detail design as a set together with the MBD specification. Software detail design then uses this data setting requirement sheet as a reference to perform scaling for all data including data other than the required data, and creates the software.

However, if the data is described by hand entry when generating the data setting requirement sheet, data entry errors and other errors may result in inconsistent data names and data descriptions between the MBD specification and data setting requirement sheet. Therefore, specification expression errors were reduced by developing a tool that extracts data requiring data settings in the MATLAB/Simulink model, and reflects it to the data setting requirement sheet.

The automatic extraction tool used to generate the data setting requirement sheet consists of the following four elements.

- (1) MBD specification (floating-point data model)
- (2) Data information extraction program
- (3) Data request and insertion program
- (4) Data setting requirement sheet

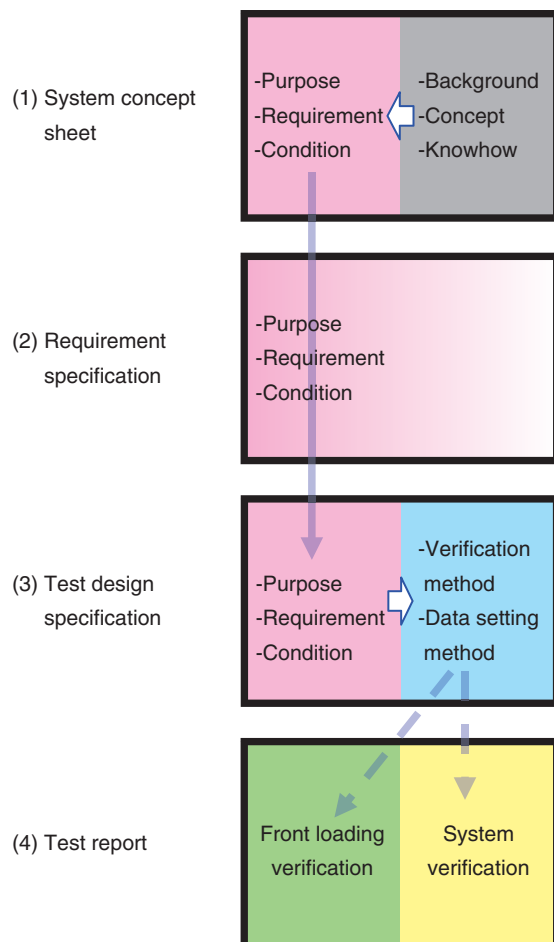


Fig. 4 Format of new process documents

The data setting requirement sheet is generated according to the flow shown in Fig. 5, and the features of this flow are described below.

(1) MBD specification (floating-point data model)

The variables and parameters to be defined in the data setting requirement sheet are expressed in model blocks within the MBD specification. In addition, the MBD specification prescribes the block applications and data names within the model in order to facilitate recognition of data attributes, enhance the legibility of the MBD specification, and increase verification efficiency. The data to be extracted for the data setting requirement sheet is also described by the MBD specification to add prefixes and set properties in order to define the data to be extracted.

(2) Data information extraction program

The data information extraction program analyzes the hierarchical structure of the model and searches using the information set in the model in (1) above as keywords. This program extracts only the data names, the data description information embedded in the blocks, and the data hierarchy from among the various information described within the model.

In addition, regarding parameter extraction, some data requires that multiple parameters be set for a single block, so the program can request the optimum number of parameters according to the applicable block.

(3) Data request and insertion program

This program is started up by the data setting requirement sheet, and the data information extraction program is executed by a request from this program. This program also inserts the data names, descriptions and data hierarchy in the list generated by the data information extraction program, into the data setting requirement sheet. In addition, this program has a function that performs data merge processing when creating a revision using existing data in the data setting requirement sheet.

This program also has a data check function that checks for duplicate data names and other errors, and a function that can extract changes in the information reflected when revising a data setting requirement sheet. These functions reduced imperfections in the data described in the MBD specification.

(4) Data setting requirement sheet

The data extracted from the model by the above program is arranged in the data setting requirement sheet, and the scaling to be performed for each piece of data and the data units are also described in order to request data settings of the software detail design.

The data is categorized beforehand and the format enables only selection of applicable data setting requirement values from lists corresponding to each category in order to reduce data entry errors and variance in data setting requirements due to multiple specification creators when describing the specification requirements.

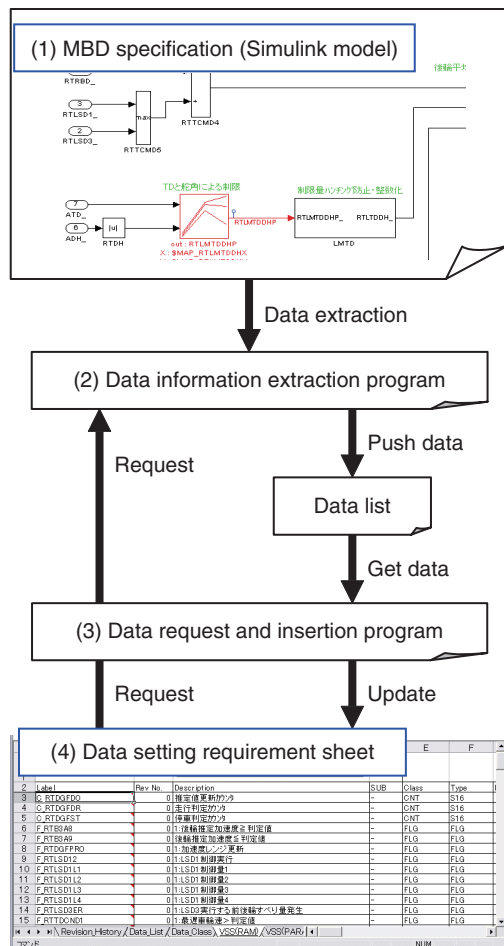


Fig. 5 Generation of data setting requirement sheet

6. Application Plug-in Structure

In the application configuration of the previous MBD software development process, the data I/O between applications exchange variables directly. As a result, when there is a change in the external output of an application, the specifications of all applications that reference those variables must be changed in consideration of the effects of the change in the reference source, even when the applications themselves do not change.

In the new MBD software development process, an application configuration was devised that reduces the need to remake specifications by reducing the effects of changes to application specifications on other applications and by increasing the specification reuse rate. This application configuration exchanges variables using a plug-in structure that enables easy replacement of applications while maintaining a software processing configuration that performs constant-time processing.

This plug-in structure features an Application Programming Interface (API) that performs physical value conversion in order to use OS data for calculations by each application, and an API for each application as shown in

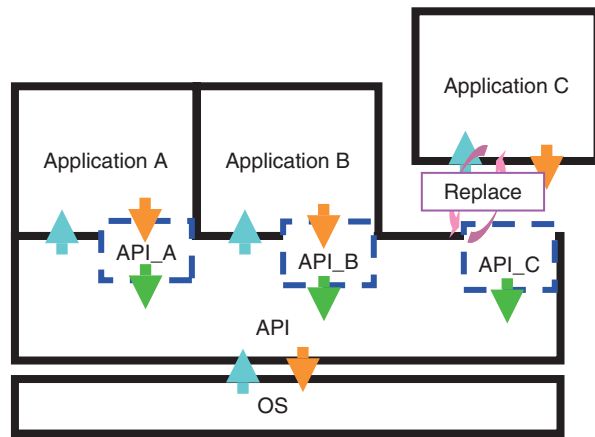


Fig. 6 Plug-in structure of applications

Fig. 6. When the output from an application is used as the input for other applications, the API of that application converts the output to a general-purpose variable to enable use as the input from the API. This enabled the API of each application to absorb changes in application outputs, which enabled the generation of software even when the referring application does not change.

For example, in addition to normal vehicle control applications, there are also applications that perform substitute value control such as control by a diagnostics unit, and it may be desired to change the substitute control values into the normal control values. In this case, instead of forcibly inserting the substitute values into the vehicle control applications, giving the vehicle control application API a function that performs processing to switch the control values enables each application to express specifications that require only the intended control. This reduced the specification changes that had to be made for various areas when specifications were added in previous development, which increased the specification reuse rate and development efficiency.

7. Evaluation of the Process

7.1. Evaluation of Specification Quality

The new MBD software development process was applied to the development of software for a new mass-production electronic control AWD system. The volume and contents of the specification adjustment sheets issued by software detail designers with respect to the specifications provided by system control designers were analyzed and compared with those when the previous MBD software development process was initially applied to new system development.

First, the ratio of the number of specification adjustment sheets issued to the total specification volume was reduced by approximately 20% compared to development that applied the previous MBD software development process.

In addition, the contents of the specification adjustment sheets were analyzed and categorized into the “Description error,” “Inconsistency between setting data” and “Imperfection with specification” items related to specification expression errors and inconsistency between the requirement specification and the MBD specification, and other items. Figure 7 shows these results. In the previous MBD software development process, specification expression errors and inconsistency between the requirement specification and the MBD specification accounted for over 80% of the total. However, this ratio dropped to 40% or less in the new MBD software development process that performs front loading verification. In addition, the new MBD software development process reduced the ratio of specification changes due to specification expression errors and inconsistency between the requirement specification and the MBD specification, as a percentage of the total specification volume, by approximately 50% compared to when applying the previous MBD software development process.

7.2. Evaluation of Software Development Load

Figure 8 shows the results of comparing the volume of issued software specifications over time during the system development schedule. In the previous MBD software development process, specification issue peaks in the latter stages of system development. In contrast, the new MBD software development process issues fewer specifications towards the end of development.

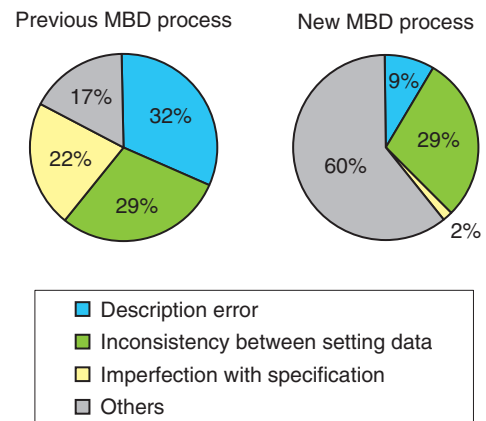


Fig. 7 Comparison of subject in requirement confirmation sheet

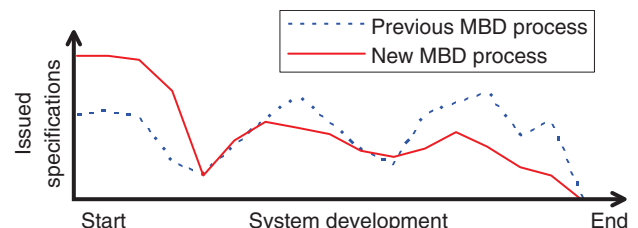


Fig. 8 Volume of issued specifications

8. Conclusion

The system control design and software detail design areas of the software development process for electronic control AWD systems were separated by expressing the MBD specification as a floating-point data model. This enabled efficient performance of front loading verification, which increased specification quality.

The new process was applied to the development of software for a new mass-produced electronic control AWD system, and reduced the volume of specification corrections due to inconsistency between the requirement specification and the MBD specification, and specification expression errors. This reduced both the specification creation process time and the verification work, which enabled reducing the ratio of the software development load in the latter stages of system development.

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Full Vehicle HILS for Enhancing Development Efficiency of Electronic Control Systems

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ABSTRACT

Electronic control systems mounted in vehicles are increasingly functioning cooperatively by means of multiple communication that uses multiple ECUs (electric control systems), and verification of actual ECUs as a full vehicle integrated electric control system that links these ECUs is more efficient in securing quality. To increase the verification efficiency, Honda has developed the Honda Electrical Revolution for Control Unit by using the Large Examination System (HERCULES) as an environment that can perform integrated verification of all ECUs that comprise the on-board network, in advance of tests using actual vehicles. This enables verification of ECUs approximately three months earlier than with the conventional method, and automated test operation functions constructed at the same time also reduced the testing period by one-half.

1. Introduction

Recently, large numbers of electronic control units (ECU) are mounted in automobiles to satisfy high-level demands in diverse fields such as safety, environment, and comfort. These ECUs boost their cooperative functions by sharing sensor signals and influencing individual actuator control via multiplex communication using multiple on-board networks such as the controller area networks (CAN) shown in Fig. 1. This realizes advanced functions and high performance throughout the vehicle⁽¹⁾.

As a result, the incorporation of ECU verification (hereafter, “ECU integrated verification”) is increasing as a key to securing quality as a full vehicle integrated electronic control system that links the entire on-board network. However, verification centering on prototype vehicles had the following efficiency issues.

(1) Delay in the testing

It is a challenge to secure prototype vehicles that correspond to all vehicle-type variations (model, destination, grade, options) in the initial development stages. This delayed the timing at which ECU integrated verification was performed, with the result that testing processes were concentrated in the later development stages.

(2) Prolonged testing time

It is a challenge to accurately simulate actual conditions such as running, operations, degradation elements, and other factors under special environments, to measure the electrical characteristics of ECUs in the condition assembled in the vehicle, and to reproduce identical conditions for analysis.

The testing procedure including preparation and analysis therefore takes a long time.

In response to these issues, European and American automobile manufacturers are leading the introduction of verification environments for actual ECUs without using actual vehicles. These include the creation of static verification environments that connect multiple ECUs and their control subjects on a bench, and dynamic verification environments that replace multiple control subjects with Hardware In the Loop Simulation (HILS)^{(2), (3)}.

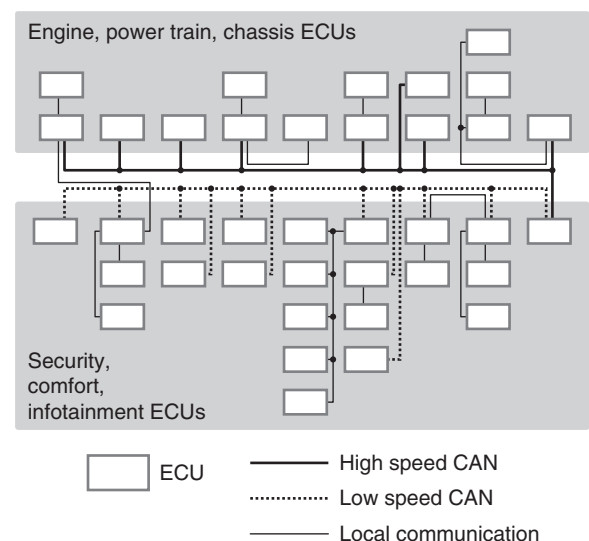


Fig. 1 Example of ECU network

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To address the issues noted above, Honda has developed HERCULES, which replaces the control subjects of all the ECUs that comprise the on-board network with HILS, and applied it to the mass production vehicle development process on a trial basis. This paper describes the system composition and reports the effects obtained by trial application.

2. Development Goals

The HERCULES development goals were set as follows.

- (1) Construction of an environment that enables dynamic ECU integrated verification at a whole vehicle level, in advance of tests using actual vehicles. This is achieved by modeling all control subjects of all the ECUs that comprise the on-board network.
- (2) Working to shorten the period required for testing and analysis, by creating data of standard manned test conditions to secure reproducibility of identical conditions and employing a structure that enables automatic test operation.

3. System Composition

As shown in Fig. 2, HERCULES consists of a vehicle model that includes all ECU control subjects comprising the onboard network, environment and driver models that simulate running tests, processor boards that perform the calculations for these models in real time and feed the results back to the ECUs as electrical signals, Input/Output (I/O), and a host PC that manages automatic operation of the system.

3.1. Hardware

3.1.1. I/O, processor boards, racks

The following I/O were incorporated to support

approximately 40 ECUs.

- Digital I/O = 828 ch
- Analog I/O = 454 ch
- Other I/O = 484 ch

Relay circuits were integrated in the wiring between these I/O and the ECUs to generate intentional connection breakdowns (broken wires, short circuits). In addition, a total of seven full-size racks were used as frames to house all of the processor boards, power supplies and other hardware.

3.1.2. ECU power supply

Batteries mounted in vehicles produce voltage fluctuations as a result of current loads due to factors such as degradation and the state of charge. Therefore, as shown in Fig. 3, a programmable power supply was mounted that generates voltage fluctuations equivalent to those of an actual battery. In addition, separate relay circuits were mounted for each ECU to vary the ECU power supply connection structure that differs for each model, and reproduce various blown-fuse conditions. This enabled simulation of various power supply states equal to those of actual vehicles. Furthermore, separate ammeters were mounted in the wiring upstream from the ECU power supplies, enabling measurement of the current consumption of each ECU.

3.1.3. Other characteristic functions

Additional I/O functions were developed exclusively for ECUs that have I/O channels other than those described as standard in Sub-section 3.1.1.

(1) Power level simulator

The electric power steering (EPS) ECU incorporates a power circuit to drive the motors, so the actual motor current that corresponds to the load of the steering system that is the control subject must be fed back to the ECU.

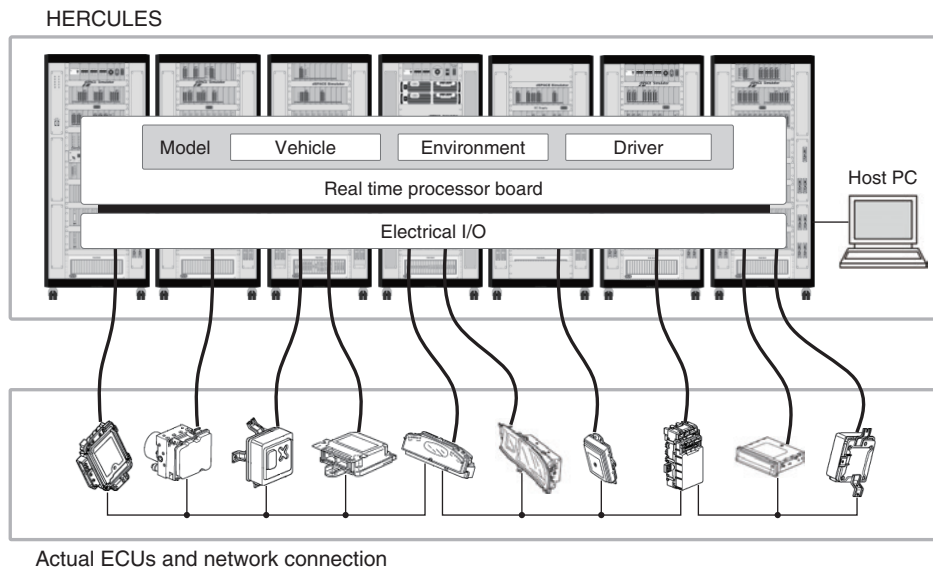


Fig. 2 HERCULES structure

Therefore, a power level simulator was constructed as shown in Fig. 4, using a field-programmable gate array (FPGA) board that calculates the electrical state of the brushless motor in real time, and an electronic load unit that generates the calculated motor current in real time.

(2) Wireless simulator

The tire pressure monitoring system (TPMS) ECU receives the pressure values detected by sensors built into each wheel as radio frequency (RF) band wireless signals. Therefore, a wireless simulator was constructed that can generate simulations of these signals in accordance with the conditions of each vehicle model.

3.2. Models

Figure 5 shows the overall model structure. The overall

model is divided into vehicle, environment and driver models, and these are linked in a loop.

A model package made by dSPACE GmbH was used as the base, and functions related to original mechanisms, locations in the base model lacking body electronics and electrical components, and I/O control were added.

3.2.1. Vehicle model

The vehicle model includes all the ECU control subjects connected to the network. The accuracy requirement for model creation was that all control subjects satisfy the behavior within the normal range specified by the ECUs. The features of this model are described below.

(1) Engine auxiliary units

Models that simulate exhaust gas recirculation (EGR)

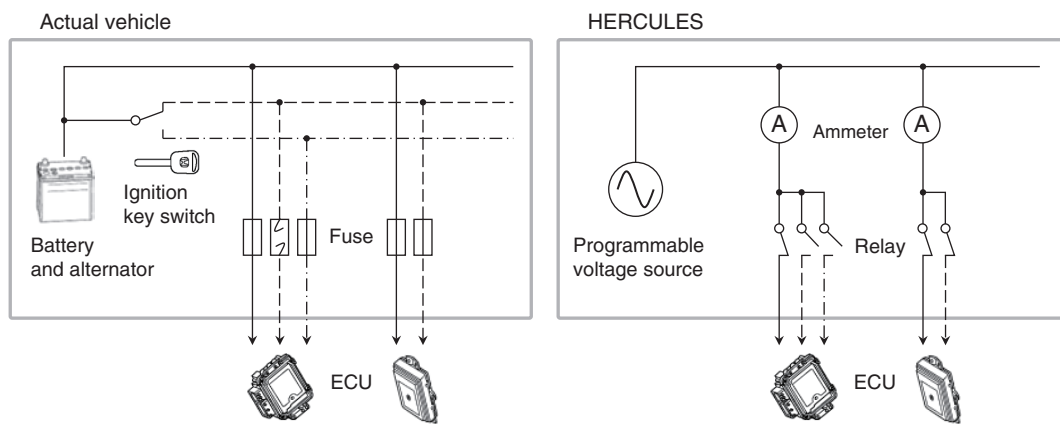


Fig. 3 ECU power supply structure

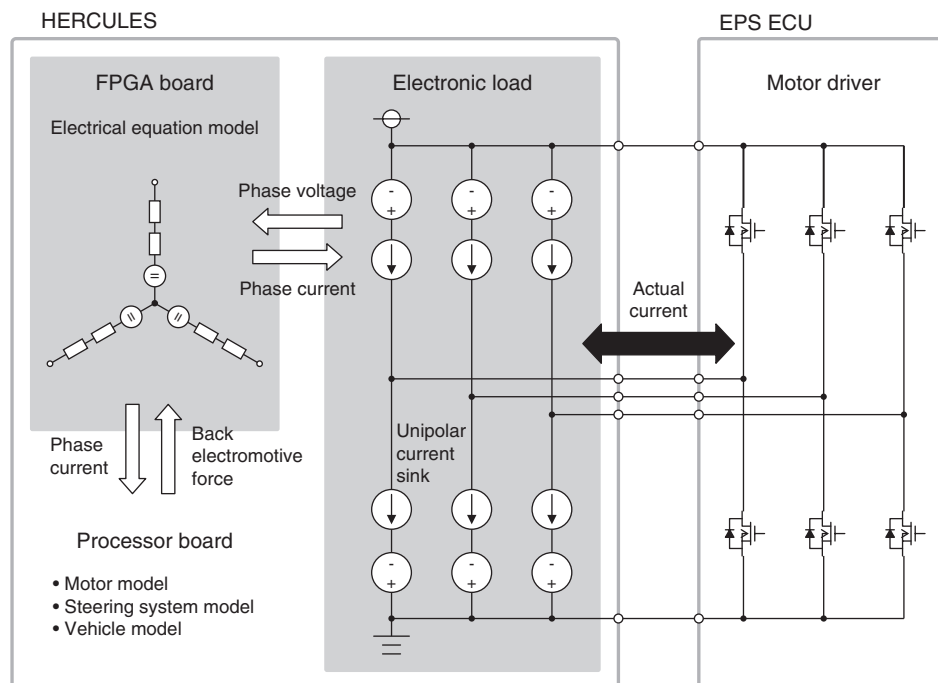


Fig. 4 Power level simulator for EPS ECU

and the evaporative emission control system (EVAP), and models of the alternator and starter motor, which influence the battery state, were added.

(2) Power steering

A brushless motor model and a steering gear box model were added for the EPS power level simulator.

(3) Battery

A model was added to simulate voltage fluctuations corresponding to electrical loads due to degradation and the state of charge.

(4) Body electronics and electrical components

Models were added to simulate the motorized actuators used for power windows and various comfort equipment.

3.2.2. Environment model

The model structure enables arbitrary definition of road conditions such as course layout and road surface friction to simulate various running conditions.

3.2.3. Driver model

This model enabled autonomous lane-following running (autonomous accelerator pedal, brake, and steering operations), and vehicle equipment operations (ignition key, various switch operations) at arbitrary timings synchronized with the running conditions, to simulate manned test conditions. In addition, the structure also enabled the use of external diagnosis tools that presume vehicle inspection processes.

3.2.4. Model calculations

The basic model calculation cycle was set at 1 ms, and partial iterative operations are used in locations that require high speed. Calculations are performed in real-time by parallel processing using seven processor boards.

3.3. Automated Tests

A function that continuously and automatically performs multiple test items was constructed to reduce the process time for periodically repeated tests and the operation time for manipulations and measurements. Users define arbitrary test conditions (test patterns) and designate the execution sequence and judgment conditions, and the series of tasks through generation of the results report are performed automatically.

3.3.1. Test patterns

Figure 6 shows an example of a basic test pattern definition. Here, vehicle speed and equipment operations at different running positions or over time are converted to data according to the manned vehicle running and operation test mode definitions.

3.3.2. Judgment conditions

The vehicle status (model variables) and the status output values of each ECU can be judged according to arbitrarily defined threshold values and logical equations.

3.3.3. Results report

Figure 7 shows an example of a results report generated by an automated test. In this example, the judgment conditions define the failure warning lamp indication, standby current after shutdown, and diagnostic trouble code (DTC) storage for each ECU, and the results for each item are displayed in ECU units.

3.4. System Overview

Figure 8 shows the general appearance of the completed system. The ECUs subject to verification and their network connections are arranged in the two ECU racks in the center

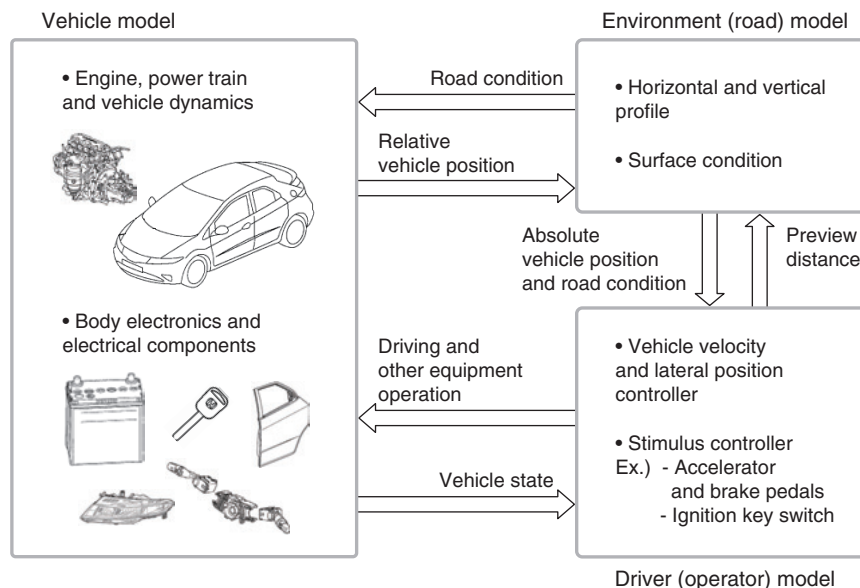


Fig. 5 Model structure

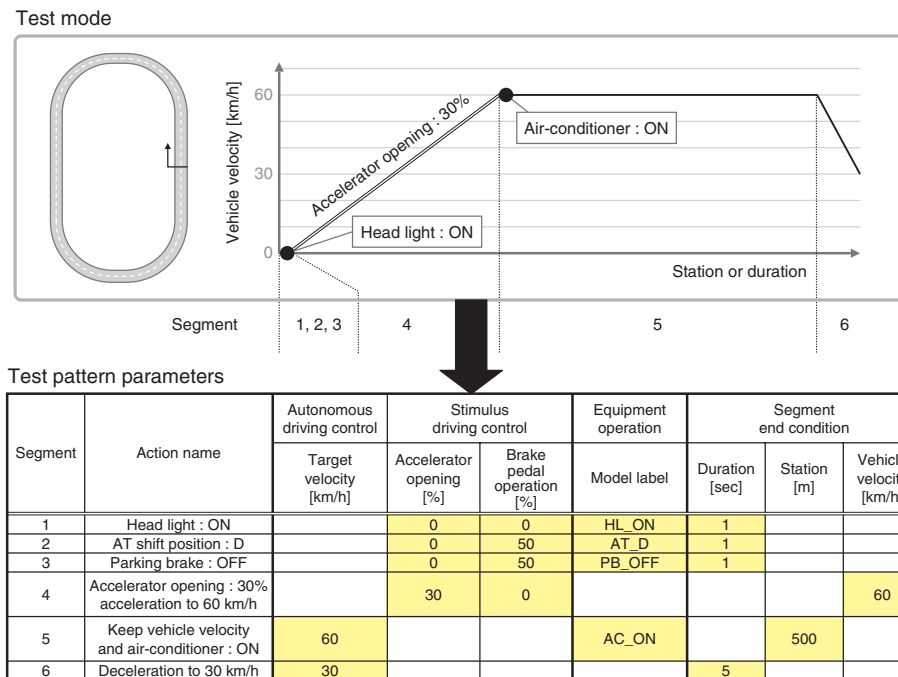


Fig. 6 Example of test pattern parameters

Test pattern No.1

ECU name	Warning lamp	Standby current	DTC
ECU_A	Failed	Passed	Failed
ECU_B	Passed	Passed	Passed
ECU_C	Passed	Passed	Passed
ECU_V	Passed	Passed	Passed
ECU_W	Passed	Failed	Passed
ECU_X	Passed	Passed	Passed
ECU_Y	Passed	Failed	Passed
ECU_Z	Passed	Passed	Passed

Fig. 7 Example of report file

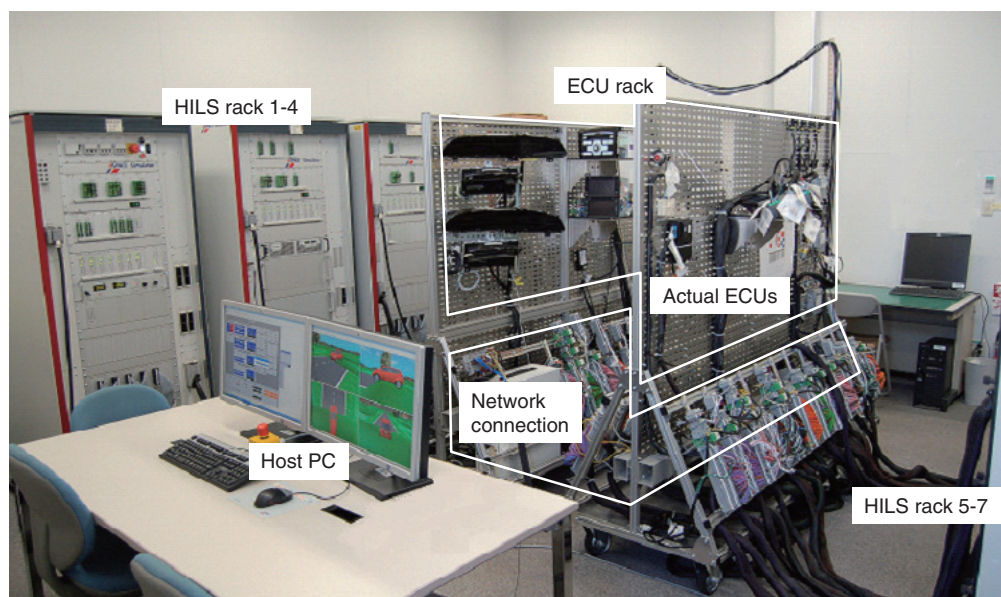


Fig. 8 HERCULES overview

of the photograph. The vehicle model for verification is switched by replacing these ECU racks.

4. Verification of System Accuracy

All control subjects replaced with HILS were verified to satisfy the behavior within the normal range specified by the ECUs.

The verification example for the vehicle model, in which all the ECUs that comprise the on-board network were connected to each other and activated, is presented below. Figure 9 shows the model calculation results and actual vehicle measurement results for the vehicle trajectory, lateral acceleration, and yaw rate while turning.

In this example, the correlation of the model calculation results for lateral acceleration and yaw rate was confirmed to be within the normal ranges specified by the ECU.

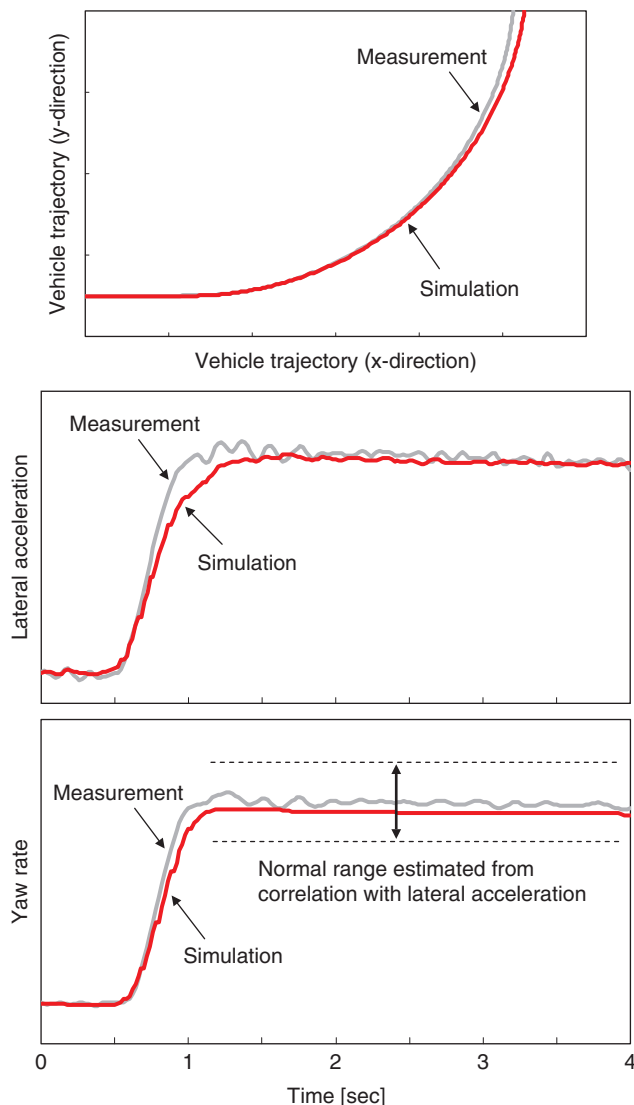


Fig. 9 Example of HILS verification

5. Effects of Trial Application

Trial application was made of HERCULES to the 2011 model year mass-production vehicle development process, and the following effects were confirmed.

- (1) Approximately 1200 patterns of test items could be performed approximately three months earlier than with conventional tests using actual vehicles. It was confirmed that HERCULES was able to reproduce and extract trouble phenomena in the same manner as those extracted by tests using actual vehicles.
- (2) Day-and-night continuous operation using the automated test operation functions, and the high reproducibility of various conditions and ease of analysis provided by HILS enabled reduction of the time needed for testing and analysis to half or less of the previous period.

6. Conclusion

The development of HERCULES enabled ECU integrated verification in advance of tests using actual vehicles, without using actual vehicles, and also enabled reduction of the testing period. The newly constructed system and test patterns have expandability and compatibility, and further effects can be expected by expanding the system to cover more models in the future.

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Development of Fixture Jig for High-precision Turbine Blade Machining

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ABSTRACT

A machining jig optimized for high-precision machining of turbine blades, which are principal parts in aviation gas turbine engines, was developed. The objective was methods of manufacturing that are high-precision, low-cost, environmentally considerate, and capable of application to mass production.

Turbine blades are airfoil parts with three-dimensional geometry, and they are the most important parts in determining the performance of high-speed engines. They are manufactured by precision casting because they employ large amounts of heat-resistant alloys that are difficult-to-machine materials. The blade attachment part has a special shape and requires high precision, and is therefore finished by precision machining. It used to be that low-melting-point alloys would be poured into a fixture jig, using the airfoil surface geometry as a reference, and the piece would then be machined. However, fixing the blade in place and removing it involved a high-temperature dissolution process, and the labor required for the manual steps made this method unsuitable for mass production.

An alternative material was sought in the recyclable, low-cost resins used in motor vehicles. The resin jigs were in a compact form machined to standards that would allow multiple blocks to be placed on the machining equipment; the material could be formed simply using the insert molding manufacturing process; it possesses the rigidity to allow high-precision machining; removal of the jig after machining would not result in blade deformation; and the resin should be recyclable. Based on experimentation, PPA was selected as the material and adopted for the fixture jig.

1. Introduction

A gas turbine engine is made up of a compressor, a combustor, and a turbine. These engines are used to propel aircraft as well as to drive propellers, generators, vehicles, and so on. Air being taken in continuously is pressurized in the compressor after which fuel is added and then ignited in the combustor, yielding combustion gas that contains high energy from combustion. This combustion gas is then directed into the turbine, where part of the energy is extracted in the form of rotary movement. This rotary output is the source of power that drives the compressor. The turbine blades (hereafter referred to as blades) are among the parts that make up the turbine, and they consist of an airfoil portion on which the combustion gas acts and a tree portion that is mounted on a disk. This tree portion is designed and formed to hold the blade to the disk and to withstand centrifugal force when the turbine is rotating. That portion is called the tree because it resembles a Christmas tree in shape, and since it is a critical part, it

requires surface roughness and dimensional accuracy on the micron order. There are also blades that have a structure called a shroud at the outside end of the airfoil such that the shrouds form a ring around the periphery when the blades are mounted on the disk. This structure helps keep gas from escaping and helps prevent resonant vibration in the blades (Fig. 1).

A heat-resistant Ni-base alloy is used for the blades. Creep strength against centrifugal force is important. Therefore, the rate of solidification is regulated during the precision casting process so that there will be no crystal grain boundaries transverse to the tensile stress, thus yielding a single crystal or a directionally solidified material. Heat-resistant alloys have low thermal conductivity and are rigid, tough, difficult-to-machine materials. Consequently, cutting tools experience large amounts of wear, and machining costs are higher. Blades are also difficult parts to position and immobilize for machining, making them a factor in increased manufacturing cost.

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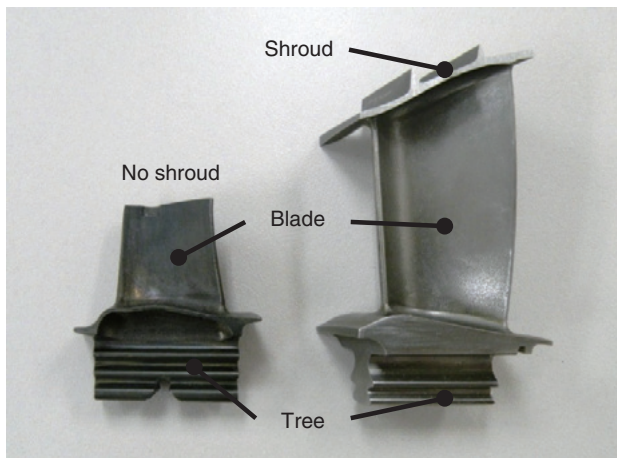


Fig. 1 Image of turbine blade

2. Blade Machining Process

2.1. Overview of the Process

First, an ingot of heat-resistant alloy is heated and melted in a vacuum and precision cast to create blade material. The blade material is then positioned using a fixture jig, and the tree and shroud portions are machined. Dimensional inspections are carried out during this process. After machining, the fixture jig used to immobilize the blade is removed, burr is removed, the surface is polished, and the piece undergoes a washing process. A completion inspection is then conducted. The airfoil portion satisfies the design requirements for dimensional accuracy and surface roughness in the precision casting state, so no machining of it is done and the material is used as it is. The tree and shroud portions cannot satisfy the design requirements the way they are as they come out of precision casting, so they go through the finishing process by machining.

2.2. Former Method of Immobilizing Blades and Issues Involved

The only place that can be used to immobilize the tree portion and the shroud portion for machining is the airfoil portion, which does not undergo machining. If the piece is held in place by means of the airfoil directly, however, the airfoil may be deformed, so machining cannot be performed in that way. Consequently, a fixture jig that can hold the airfoil and not deform it is needed. The fixture jig is attached using the airfoil as reference, and it serves as a retainer during machining.

There are different methods of using the fixture jig. In one, the piece is held by clamping the airfoil part with a form jig, and in the other, the airfoil part is enfolded in some kind of immobilizing material. The method of clamping it with a form jig necessitates preparation of the same number of jigs as blades that will be machined simultaneously, for each type of blade. This means that the initial jig fabrication cost is high. This method is also unable to deal with variations

in the dimensions of the airfoil portion created by precision casting, and it can cause damage to the airfoil surface when clamping it. It can also happen that the position slips during machining due to inadequate immobilization, so that dimensional errors occur. Consequently, the method of enfolding the airfoil portion in an immobilizing material was adopted, and a low-melting-point alloy that dissolves at approximately 200°C was used as the immobilizing material. However, when the blades were being immobilized, and when the low-melting-point alloy was being removed after machining, it was necessary to heat and then cool the blades and the low-melting-point alloy, and the long lead time was an issue. When removing the material, the alloy foreign matter that remained on the blades had to be removed by hand, further lengthening the lead time. Other issues included the fact that the low-melting-point alloys are heavy and not easy to handle, and that the materials cost was high.

3. Immobilizing the Blades with Resin

3.1. Immobilization by Fiber-Reinforced Thermoplastic (FRTP)

Consideration of a fixture jig that is more inexpensive, lighter, and allows lead time to be shortened resulted in the decision to replace low-melting-point alloys with fiber reinforced thermoplastic (FRTP) as the immobilizing material. Thermoplastic cools and solidifies rapidly, and it can immobilize a blade with insert molding within several minutes. Another company has applied for a patent on a method of immobilizing blades using resin that is not fiber-reinforced, but the use of FRTP (hereafter resin) can enhance the rigidity of the fixture jig. The fixture jig does not deform even under a greater machining load, which means that the blade machining conditions can be made better and enhanced machining efficiency and quality can be expected.

In order to select a resin, it was necessary to perform a comprehensive evaluation and judgment of formability, blade machining efficiency, ease in removal of the resin, and resin material costs. The suitability of resins cannot be judged solely by their characteristic values alone. Therefore, a number of resin types were selected, subjected to tests of forming and removal, and judged accordingly. This article will focus on forming of the blades before machining and on removal after machining, which are characteristics of this process. Tests of forming were conducted first. The characteristic values of resins vary greatly with the state of forming. Formed items were therefore observed, the dimensions were measured, and the molding conditions were set to optimal levels for demonstrating resin performance. Tests of removal were then conducted and resin removal methods were decided. Figure 2 shows the blade materials and the external appearance of the resin fixture.

3.2. Forming Mold

The blade material is made by precision casting with a dimensional tolerance greater than 0.05 mm. Consequently,

it was necessary to set the gap between the blade material and the mold to be greater than the 0.05 mm thought to enable insert molding. This gap became a cause of resin leakage, and it caused a deterioration in resin dimensions and mechanical properties. It was necessary to address these issues by means of the molding conditions (Fig. 3).

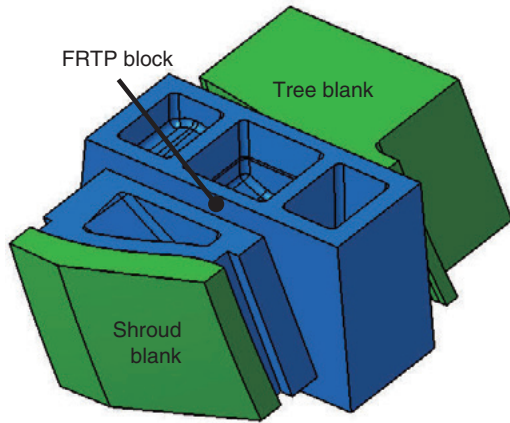


Fig. 2 Overview of turbine blade with F RTP block

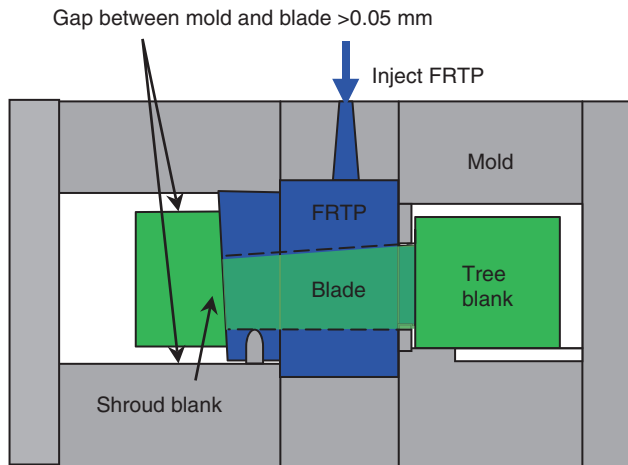


Fig. 3 Section view of insert mold

4. Immobilization of the Blades

4.1. Selection of the Resin

Consideration of the below items in connection with selection of the resin resulted in identification of three candidates for immobilizing material. These are polyphthalamide (PPA) reinforced with glass fibers, polylactic acid (PLA) reinforced with bamboo fibers, and polypropylene (PP) reinforced with glass fibers (Table 1).

(1) Chemical resistance

This is the resistance of a resin to chemicals. Resins that are not attacked by and do not have their properties changed by the cutting fluids, cleaning fluids, oils, and other such substances employed are used.

Table 1 Material data sheet of selected F RTPs

	PPA	PLA	PP
Resin content	Polyphthalamide	Polylactic acid	Polypropylene
Fiber content	Glass	Bamboo	Glass
Fiber weight ratio (%)	45	60	40
Density (kg/m ³)	1560	1390	1220
Young's modulus (GPa)	13.8	4.4	9.5
Charpy absorbed energy (kJ/m ²)	10.3	3.3	37
Mold shrinkage (%)	0.6	0.3	0.7

(2) Coefficient of elasticity

This is a mechanical property that serves as an indicator of the rigidity of the fixture jig. The larger the value, the smaller the amount of jig deformation when a load is applied.

(3) Cost

Commercially available resins that are reasonable in cost, readily obtainable, and distributed in large volumes are used.

(4) Mold shrinkage

This value affects the dimensional accuracy and deformation of the formed item. The smaller the value, the less the amount of shrinkage during solidification. Lower anisotropy is preferable.

(5) Impact strength

This value is an indicator used to judge susceptibility to cracking when the resin is removed. It indicates the amount of energy required to destroy the resin when an impact load is applied to it. Resins with smaller values are considered to be more susceptible to cracking.

(6) Environmental suitability

Resins that impose a lower environmental burden are used. Resins are judged on whether or not they can be recycled, whether or not they are biodegradable, and other such properties of materials that are readily disposable.

4.2. Molding Conditions

Resin for forming should ideally have positive flowability, should fill the mold from corner to corner under low pressure and in a short time, and should cool and solidify evenly. In the blade insert molding that is the subject here, however, there was a large gap between the blade material and the mold, as discussed above, and resin leakage occurred under the molding conditions recommended by the resin manufacturer. An effort was therefore made to lower the resin flowability in the vicinity of the gap in order

to help prevent resin leakage. Conceivable methods for lowering the resin flowability were lowering the melting temperature and lowering the temperature of the mold. The method of lowering the melting temperature reduced resin flowability so that it was necessary to increase the injection pressure, which was at cross-purposes with prevention of resin leakage. Lowering the melting temperature also increased the likelihood of embrittlement due to insufficient melting. The method of lowering the melting temperature was therefore judged to be inappropriate. The method of lowering the mold temperature does not have drawbacks like those noted above. Ordinarily, no change is made to the mold temperature, but this represented a new attempt at localized control of the flowability of resin in contact with the mold without increasing the injection pressure. It was therefore determined possible that this could broaden the range of applicability of injection molding.

The following four molding conditions are parameters that should be adjusted:

(1) Resin temperature and resin injection pressure

Higher figures for both resin temperature and injection pressure yield more favorable resin flowability. This makes forming and filling easier to perform, but the resin becomes more susceptible to leaking. The resin temperature is set relatively high to where there is no resin leakage and in the range within which the resin does not undergo thermal decomposition. The resin injection pressure is to be set starting at a low figure, and is to be adjusted according to filling conditions.

(2) Mold temperature

When the mold temperature is higher, the resin inside the mold is less likely to cool and the resin flowability is more favorable, so that it is easier for the resin to fill the mold. In general, the mold temperature is set somewhat higher than the glass transition temperature of the resin, but the resin is then more susceptible to leaking. This also results in greater shrinkage of the resin during cooling, which can cause sinking to occur. The mold temperature is therefore to be set at a level as low as possible while still maintaining sufficient resin flowability for filling.

(3) Injection speed

Higher injection speeds mean that the resin is fed into the interior of the mold more quickly. The resin retains positive flowability while filling the mold, so that the filling density is greater, as is the mold shape repeatability. At higher injection speeds, however, the resin is more susceptible to leaking. The injection speed is to be set to a high level in the range within which resin leakage will not occur.

(4) Holding pressure

Higher holding pressures allow the resin to be fed in without interruption while it is solidifying, so that the filling density is greater. However, the resin becomes more susceptible to leaking. Furthermore, the application of excessive holding pressure will cause the residual stress in the resin interior to increase, resulting in deformation when separated from the mold. The holding pressure is to be set

high at a level that does not cause resin leaking and where deformation does not occur.

In order to help prevent the occurrence of resin leakage, the mold temperature this time was set somewhat low, the injection speed was adjusted to be as high as possible in the range within which mold leakage would not occur, and tests were conducted to determine the optimal conditions. Raising the injection speed results in the resin filling the interior of the mold before resin flowability decreases. This results in positive resin uniformity and reproducibility of the product shape. The relationship between mold temperature and injection speed will be described using the example of PPA (Fig. 4).

The glass transition temperature of the PPA used was 123°C, and the manufacturer's recommended mold temperature was 135°C or higher. The present testing, however, was conducted with the mold temperature set at 100°C and 120°C. The testing with a mold temperature of 100°C resulted in insufficient filling and a defect in appearance called a weld line, which is a remnant of the boundary line where two bodies of cooled resin come into contact with each other. The injection speed was raised higher in an attempt to ameliorate the defect, but resin leaking occurred at a point before the defect was eliminated. The resin flowability therefore was not enhanced and it was not possible to fill the mold sufficiently. When insufficient filling occurs, the mechanical properties of the resin cannot be fully brought out. It was judged, therefore, that appropriate conditions could not be set at a mold temperature of 100°C.

At a mold temperature of 120°C, the increased temperature of the mold enhanced resin flowability and also enabled its control. Raising the injection speed eliminated the resin filling insufficiency and the weld line at a point before resin leaking occurred. The results of the testing clearly showed the injection speeds that adversely impacted resin leaking and process yield rate, and made it possible to discern the optimal injection speed.

Next, the relationship between the holding pressure and resin injection pressure will be described using the

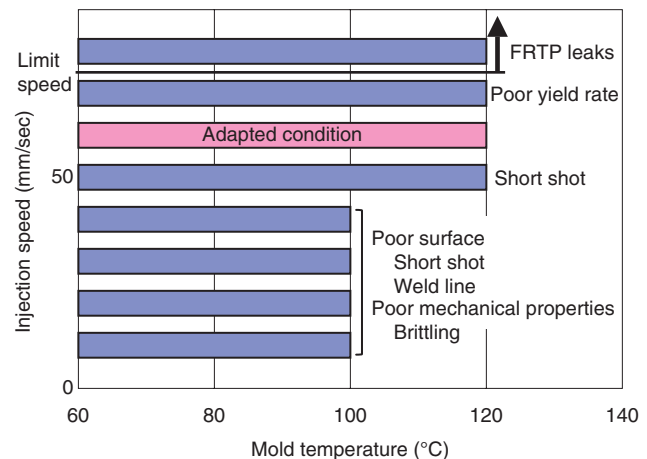


Fig. 4 Injection speed - mold temperature diagram

examples of PP and PPA. Injection molding can be divided into the injection process of filling the mold with resin and the cooling process of solidifying the resin. The pressure applied in order to extrude the resin in the injection process is termed the injection pressure, and the pressure applied during the solidification process is the holding pressure.

The resin shrinks during cooling and solidification. When this shrinking causes a gap to form between the mold and the resin filling the mold, the rate at which the resin not in contact with the mold cools becomes slower than that rate of the resin that is in contact with the mold. This difference in the cooling rate causes residual stress in the interior of the resin, and this can cause deformation of the formed product. Feeding in additional resin can be an effective countermeasure for this, if the resin is maintained in a state of flowability.

The injection pressure is decided by a comprehensive consideration of the type of resin, the melting temperature, the mold temperature, the mold geometry, and other such factors. Here, the forming is being controlled by means of the injection speed, and the injection pressure was set to as low a level as possible. The holding pressure enhances the resin filling density and contributes to retention of contact with the mold. This factor is important in the insert molding of blades, therefore, in order to keep the resin and blade in close contact and immobilize the blade securely. Consequently, the holding pressure was set as high as possible within the range that would not cause leaking of resin but would allow a favorable yield rate.

Tests were conducted on the relationship between injection pressure and holding pressure in PP and PPA (Fig. 5). The test results for PP will be explained first. The glass transition temperature of PP is lower than the ambient temperature, and adequate flowability is shown even when the mold is at room temperature. Testing to confirm the status of resin filling was therefore carried out with the injection pressure set to a somewhat low 50 MPa, and the holding pressure raised from 20 MPa, which was below that injection pressure. Even with the holding pressure set to the same 50 MPa as the injection pressure, shrinkage still resulted in major sinking, and insufficient filling and deformation

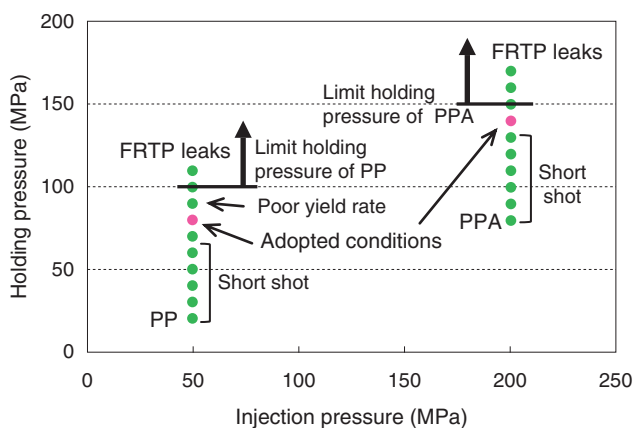


Fig. 5 Holding pressure - injection pressure diagram

occurred. It was determined that when the holding pressure was raised higher than the injection pressure, however, the sinking was ameliorated and resin leaking did not occur until the holding pressure reached 100 MPa. In light of the yield rate, a holding pressure of 80 MPa was finally adopted.

Next, the results of PPA testing will be described. With PPA, the mold temperature is lower than the glass transition temperature, and the injection speed is also set somewhat lower. Consequently, the flowability is conspicuously reduced. The injection pressure was then set to 200 MPa, which is somewhat higher than the manufacturer's recommended value, and the holding pressure was raised from 80 MPa, which was lower than the injection pressure, as in the case of the PP, to check the filling status. With the holding pressure at 80 MPa, the filling was somewhat insufficient and deformation was observed, but these were ameliorated as the holding pressure was raised. Resin leaking was confirmed to occur when the holding pressure exceeded 150 MPa. In light of the yield rate, therefore, it was judged that 140 MPa would be the optimal value. This value is lower than the injection pressure.

The forming tests conducted in this way made it possible to set molding conditions that would not result in resin leaking with either PPA, PP, or PLA. Figure 6 shows the example of insert molding of a blade with a PPA fixture.

The selection of a resin was made according to the judgment regarding removal of the resin.

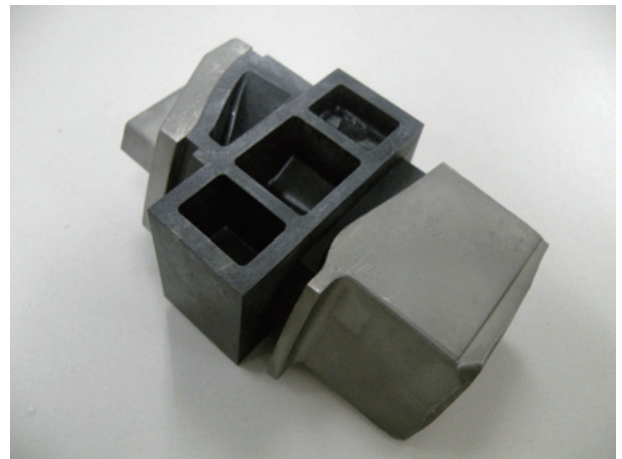


Fig. 6 Turbine blade with F RTP block

5. Resin Removal

5.1. Method of Removal

The resin that has immobilized a blade must be removed safely, simply, and in a short time after machining. The following five methods of removing the resin were compared and considered. The breaking method was judged to provide the greatest work efficiency because it did not cause any temperature change in the blade or resin and it allowed easier handling.

(1) Burning

This is a method of igniting the resin and burning it. The blades are heated to a high temperature, so there are concerns about oxidation, pollution, deformation, and so on. There are safety-related issues, as well, and it becomes impossible to recycle the resin.

(2) Heating

In this method, the resin is heated by placing the blade in a furnace or by some other such means, where it is melted and removed. This requires removal of resin that adheres to the blade surface. It is possible to recycle the resin.

(3) Chemical treatment

This method employs chemicals, ultraviolet radiation, or other such means to embrittle the resin and remove it. The process takes a long time and it requires extensive equipment to assure worker safety, making this method more costly. It is not possible to recycle the resin.

(4) Cooling

This method uses liquid nitrogen or other such means to cool the resin and cause embrittlement so it can be removed. It requires a refrigerant and necessitates measures to assure worker safety. It is possible to recycle the resin.

(5) Breaking

In this method, a load is applied to the resin to break it so it can be removed. There is no temperature change and the method is easy to handle. It is possible to recycle the resin. However, the blade must not be deformed by the load applied to the resin during removal.

5.2. Resin Removal by Breaking

The method of removing resin by breaking presupposes that the blade will not be deformed. Consequently, it is necessary to reduce the load applied to the resin.

Testing was carried out on blades without shrouds or undercuts. A hydraulic hand press was used to apply a load directly to the resin, and cracks were made to spread from wedge-shaped notches made in the bottom surface of the resin by the tensile stress of bending (Fig. 7).

The load this required for breaking in the case of PPA was 1800 N, for PLA 500 N, and for PP 1500 N. No adhesion of resin to blades nor deformation of blades was observed

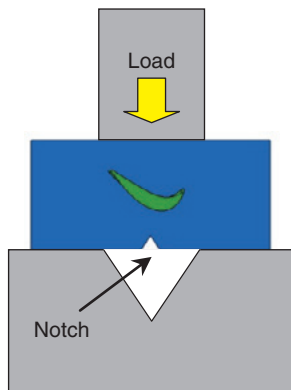


Fig. 7 Breaking method with notch

with either PPA, PLA, or PP. The cracks that formed in the PPA and PLA spread instantaneously from the front edge of the airfoil cross-section to the opposite side so that the resin broke into two pieces that could be removed from the blade as they were. With the PPA, the application of a load caused the hand press cylinder to press several millimeters into the resin block. After the PPA deformed, cracks formed in the notches and the PPA broke apart. The PLA showed particular fragility, such that the resin block broke apart even before it had been pushed in to any extent at all, allowing removal from the blade. With the PP, the progress of cracking stopped partway through. Even if the wedges were pressed in to apply further load, the resin only deformed and it could not be removed from the blade. The blade could not be removed solely with the process employing a hydraulic hand press, so the resin was broken manually and removed. The PP was judged to be a disadvantageous resin for purposes of removal and was therefore removed as a candidate material (Fig. 8).

No deformation of the blades occurred in this testing. However, the breaking method uses deformation of the resin block to cause cracks to form at the notches. In the case of blades with low rigidity, therefore, there is concern about possible deformation of the blade, so this method was judged to be impractical. Use of resin breaking methods that would not be affected by blade rigidity was studied, and the method of breaking the resin by applying a load locally was devised. The following test was performed using PPA and PLA. Wedges were pressed into the notches made in the top and bottom surfaces of the resin and load was applied locally. When this is done, the notches made in the resin serve as guides for the wedges (Fig. 9).



Fig. 8 Results of breaking method with notch

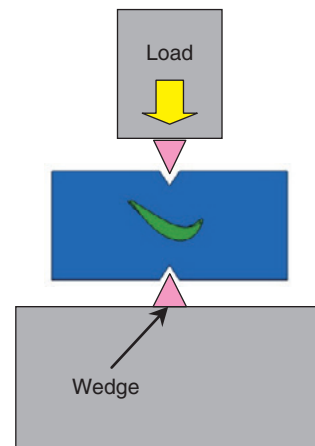


Fig. 9 Breaking method with 2 wedges

Table 2 Comparison of breaking load

	Breaking load	
	Notch	2 wedges
PPA	1800 N	100 N
PLA	500 N	30 N
PP	1500 N	N/A

Results of the testing showed that the load necessary for breaking the PPA was 100 N or less, for the PLA it was approximately 30 N, and there was no deformation in the blades after removal (Table 2). This was a significantly smaller load than was applied for breaking in the preceding tests, and the deformation in the resin when cracks formed was localized and smaller. It was found that the method of using wedges to apply a load caused cracks to form in the PPA and PLA, enabling breaking.

5.3. Examination of the Number of Breaks

The discussion to this point has covered blades that do not have shrouds. However, the actual blades have shrouds that extend around the outer periphery, and some undercutting is therefore involved. In the case of blades with shrouds, the inside part of the shroud and the airfoil surface side of the blade have an undercut shape. Because of this undercutting, the resin cannot be removed with two breaks. The approach of breaking the resin apart into smaller pieces in order to remove the resin from the undercut portions was examined.

Breaking the resin into three pieces was considered first. The PPA and PLA had wedges placed at one location on the top surface of the resin and at two places on the bottom surface. Figure 10 shows a simplified view of the jig when breaking takes place.

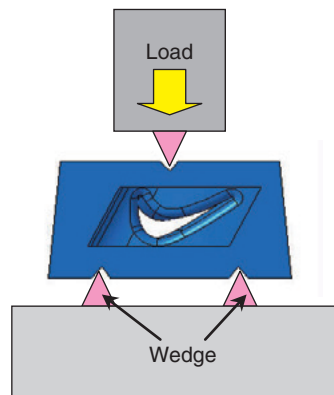


Fig. 10 Breaking method with 3 wedges from 2 sides

Table 3 Results of breaking method with 3 wedges from 2 sides

	Breaking load	Results	Yield rate (%)
PPA	500 N	Crack was unstable. Half of blocks were left.	50
PLA	50 N	Crack was unstable. 90% of blocks were left.	10

The breaking was unstable with both the PPA and PLA, and there were cases in which the resin could not be removed. The yield rate was approximately 50% with the PPA and approximately 10% with the PLA (Table 3). Wedges were placed at two locations on the bottom side, but cracks formed at only one or the other of the wedges on the left and right, while no cracks at all formed at the other of the two wedges. In the testing, symmetrical formation of cracks at both wedges at left and right on the bottom surface of the resin occurred only rarely. Cracking proceeded asymmetrically left and right (Fig. 11).

With the PLA, discoloration and adhesion of resin to the blade airfoil surface was observed. This phenomenon was not found on blades without shrouds, and it is considered to have occurred due to differences in the mold and the molding conditions in the case of blades with shrouds. As a result, it was judged the PLA was unsuitable for blades with shrouds, and PLA was removed from the candidate materials.

Based on the results from the method of breaking the resin into three parts, it was determined that there were issues with the method of placing wedges at two locations on one surface. The placement of the wedges was therefore reexamined with a view to causing stable cracking. The method of using four surfaces of the resin with a wedge placed at one location on each surface was therefore adopted. Calculation of the stress distribution by means of the finite element method (FEM) showed that increases in stress in the vicinity of the wedges occurred locally with this method, and it was anticipated that deformation of the block would be limited and there

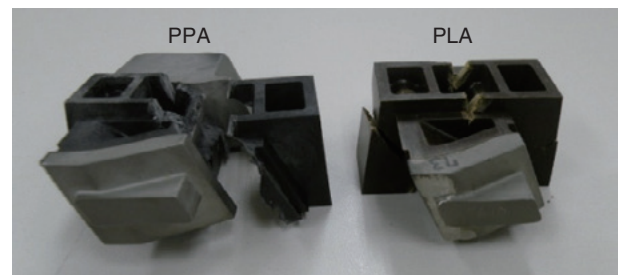


Fig. 11 Result of breaking method with 3 wedges from 2 sides

would be no impact on the blades (Fig. 12).

A special jig was fabricated to enable application of a load to four surfaces in a single process using a hand press. This is a mechanism that uses the descending motion of a cylinder to sandwich the resin from top and bottom and from left and right (Fig. 13).

The test with wedges placed at one location each on four surfaces was conducted only with the PPA. The load of 200 N applied in the test resulted in uniform cracks forming on all four sides, and it was able to produce stable breaking. The resin at the problematic undercut portions was also successfully removed. Repetition of the testing confirmed that stable cracking would occur (Fig. 14). Moreover, the

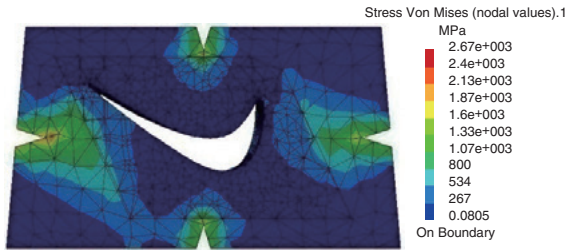


Fig. 12 Stress contour of F RTP block

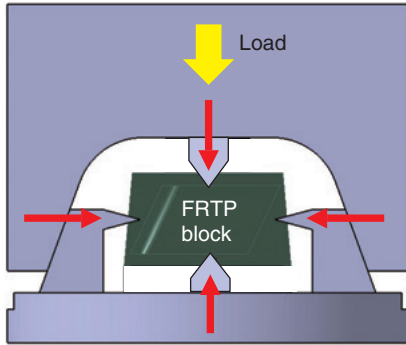


Fig. 13 Breaking method with 4 wedges from 4 sides

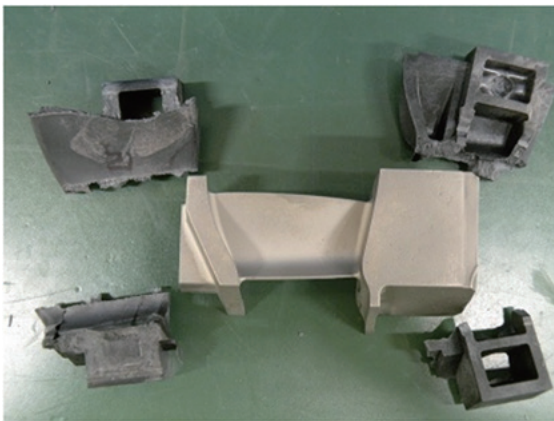


Fig. 14 Result of breaking method with 4 wedges from 4 sides

time required for the process of removing the resin was approximately one minute, and it was confirmed that the resin could be removed from a blade safely and in a short time. This breaking method was therefore adopted, and PPA, which had been demonstrated in resin removal testing as readily developing progressive cracks and not adhering to the blade, was selected as the blade immobilizing material.

6. Conclusion

- (1) The optimal conditions were found for the process of immobilizing blades using resin, for locally controlling the flowability of the resin that is in contact with the mold surface during insert molding, and for forming the piece without leaking of resin from the gap between it and the mold.
- (2) A method was found for breaking the resin apart to safely and efficiently remove it from the blade using a hand press and a special jig. This method also facilitates recycling of the resin.
- (3) PPA, PP, and PLA were selected as materials for use in immobilizing the blade. Testing performed to compare them by forming, machining, and resin removal resulted in selection of PPA as the optimal material.

The above made it possible to replace low-melting-point alloy with PPA as the material for the blade fixture jig. The process was thereby simplified and shortened, and it became possible to reduce the cost for fixing the blade. The weight reduction also enhanced the handling of the item during work.

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Visualization of Liquid Water Distribution in Fuel Cell by Using Neutron Radiography

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ABSTRACT

The accumulation of liquid water in the cell is given as a major factor affecting the electric generation performance of fuel cells, and the visualization and quantification of this liquid water is a key technology for fuel cell water management.

This study used neutron radiography to visualize the liquid water distribution inside a fuel cell during electric generation, and quantified the liquid water.

The results showed that resistance overpotential, which is one factor that reduces electric generation performance, tends to drop as the quantity of liquid water in the CCM increases. However, the resistance overpotential becomes constant at a certain quantity of liquid water or more. In addition, it was also shown that the amount of voltage drop during constant-load electric generation is strongly correlated with the quantity of liquid water in the GDL, and that the amount of voltage drop increases as the quantity of liquid water in the GDL increases.

1. Introduction

Fuel cell electric vehicles are being developed as a vital technology capable of addressing current issues facing mobility, including environmental issues such as global warming and air pollution, and energy issues due to depleting petroleum resources.

Fuel cells generate electric power by reacting the hydrogen supplied as fuel from the anode side with the oxygen in the air supplied from the cathode side. The reaction formulae for each side are as follows:

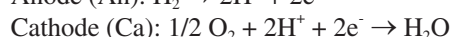


Figure 1 shows the sectional structure of a fuel cell. The fuel cell reaction is performed by a membrane electrode assembly (MEA) obtained by joining a catalyst-coated membrane (CCM) created by attaching catalyst layers to both sides of a polymer electrolyte membrane (PEM), with gas diffusion layers (GDL). Inside the MEA, the PEM is a proton conductor that manifests proton conduction functions in the water retaining state, which is thought to lower the resistance overpotential that is one factor reducing electric generation performance. Conversely, liquid water accumulation in the GDL would lower the gas diffusivity

of the GDL and reduce the oxygen level around the catalyst layers, which is thought to be one factor reducing electric generation performance.

However, the exact impact on cell performance of liquid water in the PEM, GDL and other parts of the MEA is not yet well known.

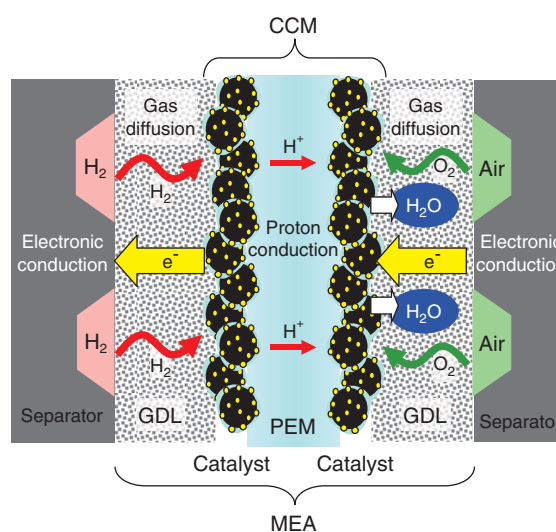


Fig. 1 Schematic of fuel cell

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2. Research Goals

The goal of this study was to use neutron radiography to visualize the liquid water in the MEA section during electric generation, and to understand the relationship between the quantity of liquid water in each part of the MEA obtained from these visualization results, and the resistance overpotential and voltage drop during constant-load electric generation.

3. What is Neutron Radiography?

Neutron radiography is a method enabling the quantification of materials using the neutron attenuation characteristics of the constituent atoms of the subject. Al, C, F and other elements often present in the component materials of fuel cells have small attenuation characteristics, but substances containing H atoms have large attenuation characteristics. These characteristics were used to measure the neutron attenuation of liquid water, which has a high concentration of H atoms, and to visualize and quantify the liquid water in the cell section.

Figure 2 shows the mass attenuation coefficients of typical atoms and molecules with respect to thermal neutrons⁽¹⁾. This shows that H₂O molecules, which include H atoms, have a larger mass attenuation coefficient compared to other atoms. (Note that H₂O is plotted according to the atomic number of H.) The Lambert-Beer law indicated by Eq. (1) is used to deduce the thickness t [cm] of the test piece from the neutron attenuation.

$$I_{out} = I_{in} \exp(-\alpha \cdot t) \quad (1)$$

Here, I_{in} is the incident neutron intensity, I_{out} is the transmitted neutron intensity, and α indicates the neutron mass attenuation coefficient (cm²/g) of the test piece. The thickness of the liquid water in the MEA was measured as described above under various electric generation conditions. The liquid water in the MEA was then quantified by dividing the liquid water thickness by

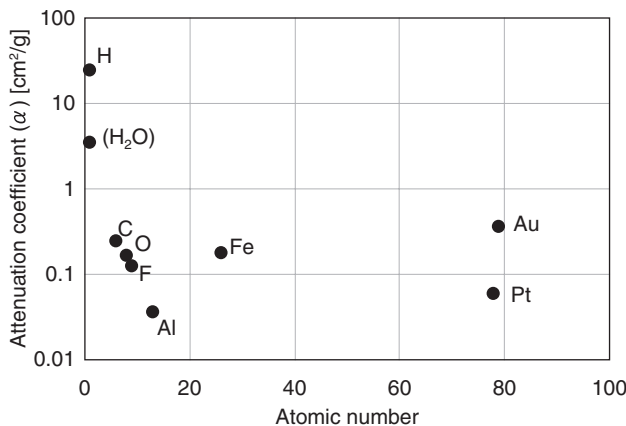


Fig. 2 Attenuation coefficient of elements

the thickness (5 mm) in the neutron ray direction of the effective electrode (described hereafter), and expressing the result as the liquid water occupancy (vol%) per unit volume.

Neutron radiography measurements were performed at the ICON beam line of the Paul Scherrer Institute (PSI) in Switzerland, in collaboration with the PSI Electrochemistry Laboratory (ECL) and Neutron Imaging and Activation Group (NIAG). These measurements were based on the recent advances in high resolution imaging of fuel cells realized at PSI^{(2), (3)}.

4. Test Method

Figure 3 shows the cell configuration. The cell for neutron radiography was created by clamping a MEA consisting of a CCM and GDL, together with gaskets to prevent gas leakage, between two separators.

The neutron transmission image was measured from the cell section direction as shown in Fig. 3. To increase the spatial resolution in the section direction in the observation region where the total of the flow field depths (An, Ca) and the MEA thickness is between 1 mm and 2 mm, measurement was performed by inclining the detector (CCD) relative to the section direction as shown in Fig. 4⁽²⁾.

By measuring with only one direction (in this study, the MEA thickness direction) of the neutron transmission image stretched in this manner, the resolution of the obtained image was increased from 13.5 μm/pixel to 2.3 μm/pixel.

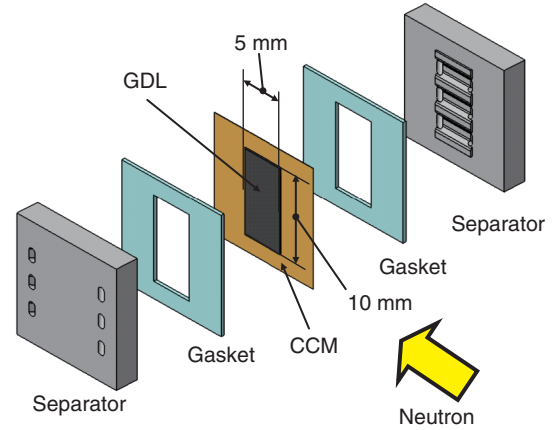


Fig. 3 Configuration of cell for neutron radiography

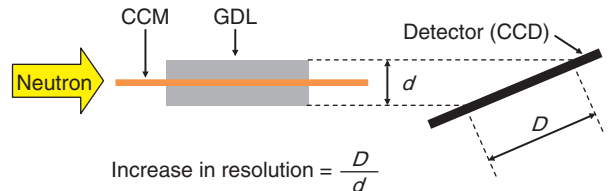


Fig. 4 Schematic of increase in resolution

Table 1 Cell configuration and test conditions

Cell configuration	Electrode area	0.5 cm ²
	Separator	3 channels
Test conditions	Current density	1.0 A/cm ²
	Cell temperature	50, 70, 85 °C
	An gas	H ₂
	Ca gas	Air
	An gas humidity	50, 100 %RH
	Ca gas humidity	40, 70, 100 %RH

However, due to the limited collimation of the neutron beam and the inherent blurring of the detector, the actual spatial resolution was approximately 20 μm .

Table 1 lists the cell specifications and the test conditions. Electric generation was performed while varying the three parameters of the cell temperature, Ca gas humidity, and An gas humidity. The terminal voltage and resistance overpotential during electric generation were measured, and the liquid water distribution in the MEA at that time was also visualized simultaneously.

5. Effects of Liquid Water Distribution on Electric Generation Performance

Figure 5 shows examples of images obtained by neutron radiography. The blue-colored portions indicate liquid water. Electric generation was performed with a fixed current density of 1.0 A/cm², so the quantity of water generated by the fuel cell reaction was the same for any

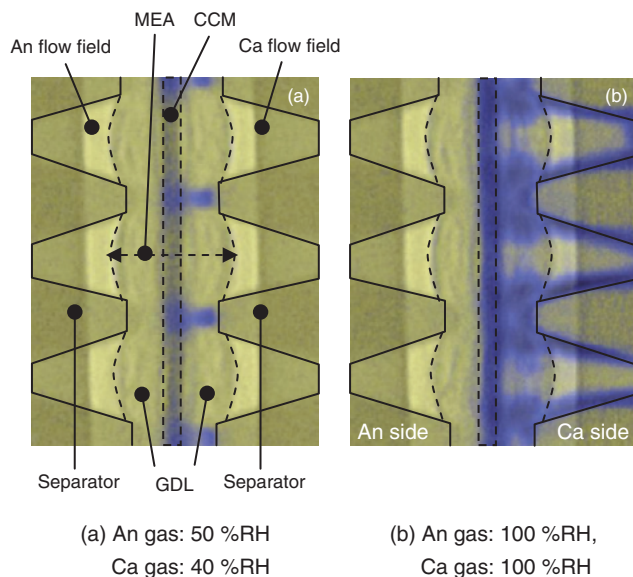


Fig. 5 Images of liquid water distribution during electric generation and regions of quantitative analysis

electric generation condition. However, when low-humidity gas was flowed to both electrodes as shown in Fig. 5(a), the inside of the MEA was drier and there was less liquid water compared to when high-humidity gas is flowed as shown in Fig. 5(b). (The cell temperature was 70°C in either case.)

Next, Fig. 6 shows the electric generation performance (terminal voltage) under each test condition listed in Table 1.

Under cell temperature conditions of 50°C at which the water saturation pressure is low and the MEA does not dry easily, the terminal voltage dropped as the Ca gas humidity increased. This is thought to be because the liquid water accumulated in the catalyst layers and GDL impeded the movement of the gas to the catalyst, which reduced the effective electrode area. On the other hand, at a cell temperature of 85°C that results in a high water saturation pressure, the terminal voltage was confirmed to drop as the relative humidity decreased. This is thought to result from a drop in PEM proton conduction functions (increased resistance overpotential) due to drying. At a cell temperature of 70°C, high performance was obtained under Ca gas humidity conditions of 70%RH, and the test conditions that yielded the highest terminal voltage were an An gas humidity of 100%RH and a Ca gas humidity of 70%RH.

To clarify the voltage behavior relative to the cell temperature and gas humidity, the liquid water in the CCM and GDL regions shown in Fig. 5 were quantified using neutron radiography. The correlation with the terminal voltage and resistance overpotential also obtained at the same time was then analyzed, and the effects of the liquid water distribution on electric generation performance were clarified.

5.1. Relationship between Quantity of Liquid Water in the CCM and Resistance Overpotential

Figure 7 shows the resistance overpotential measured under each electric generation condition listed in Table 1.

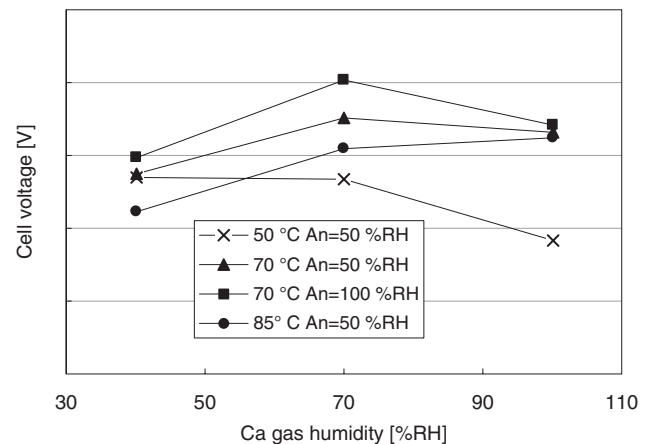


Fig. 6 Cell voltage of each test condition

At all cell temperatures, lowering the Ca gas humidity increased the resistance overpotential. In particular, at a cell temperature of 85°C that results in a high water saturation pressure, the increase in resistance overpotential as the Ca gas humidity decreased was larger than that at other cell temperatures. The behavior of the terminal voltage at 85°C shown in Fig. 6 can be explained by this resistance overpotential behavior. The quantity of liquid water measured using neutron radiography was compared to verify the relationship between the resistance overpotential and the quantity of liquid water in the MEA at this time. As described above, the quantity of water in the PEM greatly influences the change in resistance overpotential, so the ideal approach is to quantify only the liquid water in the PEM. However, the PEM thickness is relatively small given the neutron radiography resolution, and it is a challenge to clearly separate the quantities of liquid water in the PEM and the catalyst layers. Therefore, this study used the quantity of water in the CCM. Figure 8 shows the relationship between the resistance overpotential and the quantity of liquid water in the CCM.

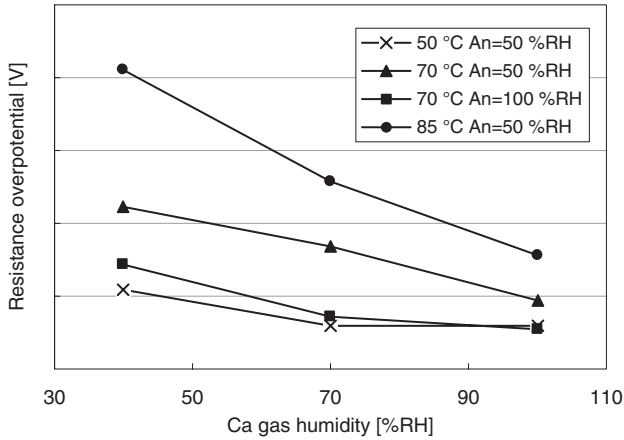


Fig. 7 Resistance overpotential of each test condition

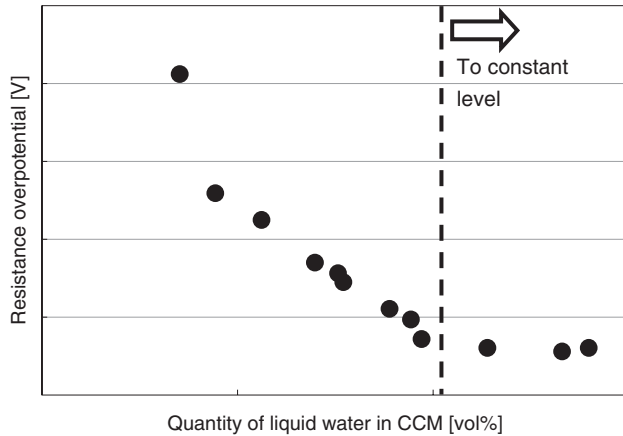


Fig. 8 Relation between resistance overpotential and liquid water in CCM

The resistance overpotential dropped as the quantity of liquid water in the CCM increased, but then became constant at a certain quantity of liquid water or more. This indicates that in order to increase electric generation performance, a certain quantity of liquid water or more should be maintained for the PEM to exhibit sufficient proton conduction functions.

5.2. Relationship between Quantity of Liquid Water in the GDL and Voltage Drop

There are two possible factors affecting the voltage drop during constant load electric generation due to accumulation of liquid water in the MEA. One is a drop in gas diffusivity within the catalyst layer due to liquid water accumulating in the catalyst layer, and the other is a drop in the amount of gas movement to the catalyst layer due to liquid water accumulating in the GDL. Accurate measurement of the quantity of liquid water in the catalyst layer is a challenge because the catalyst layer thickness is relatively small given the neutron radiography resolution. Therefore, the relationship between the liquid water accumulated in the GDL and the voltage drop was verified. The voltage drop behavior was quantified by measuring the amount of voltage drop from the start of constant-load electric generation at 1.0 A/cm². The terminal voltage includes the aforementioned change in resistance overpotential, so the drop in IR free (sum of the terminal voltage and the resistance overpotential) that cancels out this change was used as the amount of voltage drop. Figure 9 shows an example of the IR free behavior when electric generation is performed with a cell temperature of 50°C, An gas humidity of 50%RH and Ca gas humidity of 70%RH. Figure 10 shows the neutron visualization results measured at the same time (0 min and 10 min after the start of electric generation).

Figure 9 shows a high IR free level immediately after the start of electric generation, and Fig. 10(a) shows that almost no liquid water has accumulated in the MEA at that time. However, as constant-load electric generation

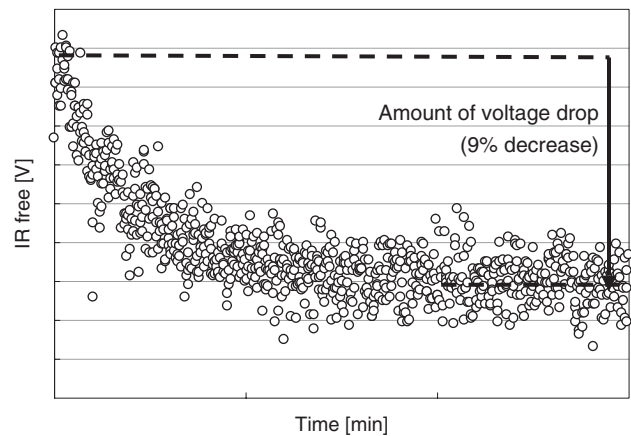


Fig. 9 Time evolution of IR free and voltage drop

continues, the IR free level drops to a constant value. Figure 10(b) shows that liquid water accumulates in the MEA by the time this IR free constant region is reached, and that most of this liquid water accumulates in the GDL near the separator ribs.

Next, Fig. 11 shows the amount of voltage drop during constant-load (1.0 A/cm^2) electric generation under each electric generation condition listed in Table 1.

The increase in the voltage drop amount due to changes in the Ca gas humidity is larger for a cell temperature of 50°C , at which there is low water saturation pressure, than at other cell temperatures. The terminal voltage behavior at 50°C shown in Fig. 6 can be explained by this voltage drop behavior. The quantities of liquid water in the GDL measured by neutron radiography were compared to verify the relationship between the amount of voltage drop and the quantity of liquid water in the GDL. As shown in Fig. 10(b), the quantity of liquid water in the GDL differs greatly near the gas flow fields and near the ribs (the convex portions of the separator that contact the GDL for

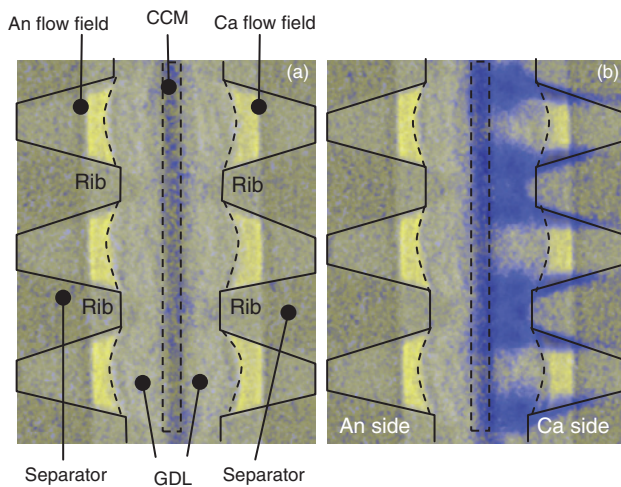


Fig. 10 Change in liquid water distribution during electric generation (a) $t = 0 \text{ min}$, (b) $t = 10 \text{ min}$

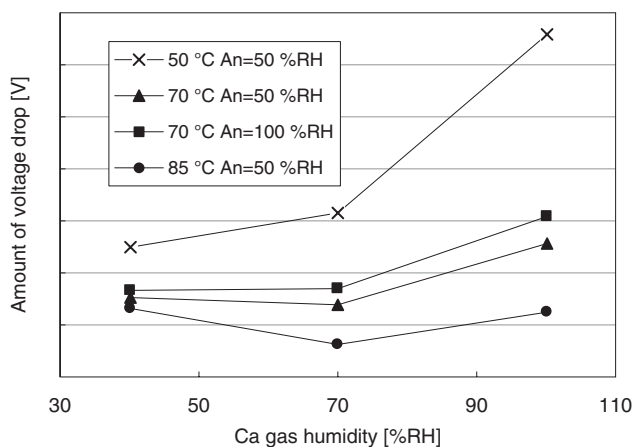


Fig. 11 Voltage drop of each test condition

electric conduction), with more liquid water tending to be distributed near the ribs. This is thought to be because gas did not easily diffuse in the GDL near the ribs, so there was less drying of the generated water compared to the flow fields. Therefore, the quantity of liquid water in the GDL was measured for the three patterns of near the ribs, near the flow fields, and the average of the two (GDL average), and the correlation between each pattern and the amount of voltage drop was verified. The results for all patterns showed that when the quantity of liquid water in the GDL increases, the amount of voltage drop also tends to increase. This confirmed that in order to suppress the voltage drop during constant-load electric generation, it is necessary to reduce the quantity of liquid water in the GDL. Of the three patterns used to measure the quantity of liquid water, the GDL average was most strongly correlated with the amount of voltage drop. Figure 12 shows the relationship between the average quantity of liquid water in the GDL and the amount of voltage drop.

The reason for the strong correlation between the average quantity of liquid water in the GDL and the amount of voltage drop is thought to be that the accumulation of liquid water reduces the GDL diffusivity. This behavior, where the movement of gas to the catalyst layers is impeded, is most closely represented by the average quantity of liquid water in the GDL.

5.3. Observations Regarding the Ideal Water Distribution

Figure 13 shows the results of combining the relationship between the quantity of liquid water in the CCM and the resistance overpotential shown in Fig. 8, and the relationship between the quantity of liquid water in the GDL and the voltage drop shown in Fig. 12.

Of the various test conditions used in this study, the maximum terminal voltage was obtained at a cell temperature of 70°C , An gas humidity of 100%RH, and Ca gas humidity of 70%RH as shown in Fig. 6.

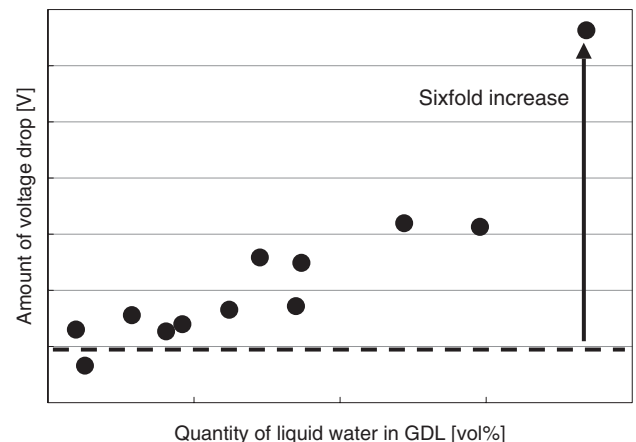


Fig. 12 Relation between voltage drop and liquid water in GDL

Figure 14 shows the neutron visualization image under these conditions, and indicates that these conditions result in the minimum quantity of liquid water in the CCM needed to reduce the resistance overpotential to a constant value as shown in Fig. 13. When drier conditions are used, the PEM cannot retain sufficient liquid water, which increases the resistance overpotential. When wetter conditions are used, the quantity of liquid water in the GDL increases, which reduces the effective electric generation area. Both of these cases are thought to result in lower performance.

When considering the ideal water distribution in the MEA, it is thought important to retain a sufficient quantity of liquid water in the CCM (PEM) to manifest proton conduction functions, while at the same time minimizing the quantity of liquid water in the GDL that reduces the effective electric generation area. However, when the cell temperature, gas humidity or other electric generation

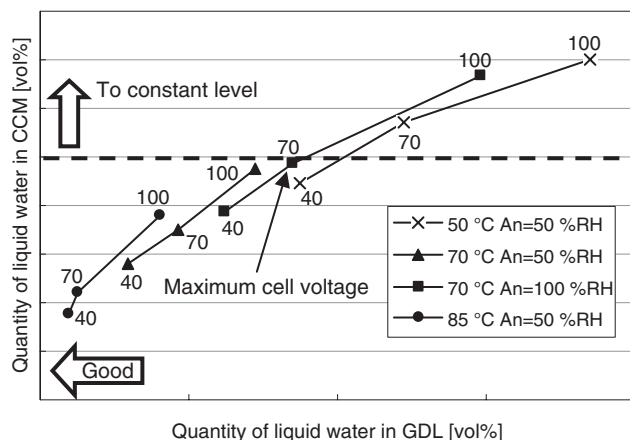


Fig. 13 Relation between liquid water in CCM and GDL (represented by cathode gas humidity)

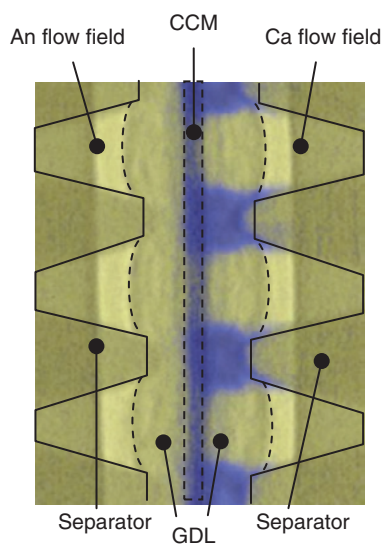


Fig. 14 Image of liquid water distribution under the best condition (An gas: 100 %RH, Ca gas: 70 %RH)

conditions are simply changed, the quantity of liquid water in the GDL increases together with that in the CCM. Therefore, to enhance electric generation performance, it is considered necessary to develop materials, optimize layer structures, and investigate flow field structures in order to approach the ideal water distribution in the MEA.

6. Conclusion

The water distribution during electric generation was visualized and quantified using neutron radiography to verify the relationship between the resistance overpotential and amount of voltage drop during constant-load electric generation, which affect the electric generation performance of fuel cells, and the quantity of liquid water in each part of the MEA. The results were as follows.

- (1) The resistance overpotential tends to drop as the quantity of liquid water in the CCM increases, but becomes constant at a certain quantity of liquid water or more.
- (2) The amount of voltage drop increases after the start of constant-load electric generation as the quantity of liquid water in the GDL increases.
- (3) The electric generation environment that can provide high electric generation performance is the environment in which the quantity of liquid water in the CCM (PEM) is the minimum required to manifest proton conductivity. However, further material development and flow field design are needed to realize the ideal water distribution.

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Development of in-situ X-ray Imaging Technologies for Measuring Water Distribution in Generating PEFC

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ABSTRACT

Water management in fuel cells is a key technology for enhancing the performance and durability of polymer electrolyte fuel cells (PEFC), and water transport phenomena in PEFC have to be clarified.

In-situ measurement techniques in generating PEFC using X-ray computed tomography (CT) and X-ray radiography were developed, and water transport phenomena in the Gas Diffusion Layer (GDL) and gas channels were analyzed.

The water inside the GDL was observed to increase in accordance with the current density, and it was also observed that water is discharged from the GDL to the channels along the hydrophilic channel wall surfaces. In addition, image processing was performed, and the water distribution inside the GDL during generation was quantitatively evaluated. The results confirmed that a large amount of water easily accumulates in the GDL under the rib.

1. Introduction

In recent years, fuel cells have been the focus of attention as a high-efficiency, zero-emission energy conversion system. Polymer electrolyte fuel cells (PEFCs), in particular, are the target of research and development as a next-generation automobile power source due to its low operating temperature and high output density. One advantage of PEFC is that only water is emitted. On the other hands, water management is an important issue. To further enhance PEFC generation performance and durability, it is necessary to prevent the condensed water in the catalyst layers, Gas Diffusion Layers (GDLs) and gas channels from impeding the diffusion and flow of oxygen, while at the same time maintaining an appropriate water content to prevent the proton exchange membrane from drying out. The water transport mechanism must be clarified to solve this type of water management issue, so active efforts are underway to develop measurement technologies that uses neutron imaging⁽¹⁾ and Magnetic Resonance Imaging (MRI)⁽²⁾, and prediction technology based on numeric simulations⁽³⁾. Though X-ray computed tomography (X-ray CT) is also used to measure water in PEFCs, most of the measurements are made after generation. However, since water distribution after generation is different from that during generation, measurement after generation is insufficient to clarify the water transport mechanism during

generation. In addition, the fibrous structure of the GDL and the water are not separated and water distribution in PEFC was not evaluated quantitatively^{(4), (5)}. Therefore, in this study, measuring technologies are developed for quantitative evaluation of the water distribution in the GDL and the gas channels. These include X-ray CT to measure the three-dimensional water distribution inside a generating fuel cell with high spatial resolution, X-ray radiography to measure the dynamic behavior of water with high temporal resolution, and image processing to evaluate the water distribution quantitatively.

2. Analysis of Water Transport Using X-ray CT

2.1. Overview of X-ray CT Imaging

X-ray CT is a technique for generating a three dimensional image by collecting attenuated transmitted X-ray signal data while rotating the X-ray source and detector or the observation sample by 180° or 360°⁽⁶⁾. Compared to MRI and neutron imaging, X-ray CT has the advantages of a high spatial resolution (in submicron-order), the ability to measure both the 3D object structure and the water distribution, low cost, and high safety. On the other hand, the X-ray absorption coefficient is proportional to the molecular mass, so X-ray CT has lower sensitivity for water molecules, which include hydrogen atoms that have a small molecular mass, compared to neutron imaging which

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has better sensitivity to substances with a small molecular mass. This issue, however, can be solved by reducing the X-ray energy and enhancing the S/N ratio of the carbon and the water and pores.

Figure 1 shows an overview of the X-ray CT measuring system. The X-ray tube used a reflective-type sealed tube with tungsten as both the filament material and the target material, and the detector used a 1.45 M-pixel CCD with an image intensifier. Measurement was performed at a tube voltage of 110 kV and a tube current of 80 μ A.

The spatial resolution of the image is determined by the relative positions of the X-ray source, observation sample and detector, and the detection element matrix as shown in Eq. (1).

$$S = D/(L/I)/A \quad (1)$$

Here, S is the spatial resolution, D is the effective photoreceptor size, L is the distance from the X-ray source to the detector, I is the distance from the X-ray source to the observation sample, and A is the detection element matrix. In addition, L/I is the projection magnification ratio, and the spatial resolution increases as the sample approaches the X-ray source. In this test, the cell was located at a distance of 2.5 mm from the X-ray source. In consideration of the balance between the degree of geometric shape detail, the contrast and the measurement time, the number of views was set at 1000, the average number of frames at 20, and the time per CT scan at 30 minutes, and measurement was performed with a spatial resolution of 2.5 μ m.

Figure 2 shows an exploded view of the fuel cell used for X-ray CT measurement. The electrode area was 10 mm x 10 mm, and consisted of a 15 mm x 15 mm carbon separator with 11 straight channels each having a groove width of 0.5 mm, peak width of 0.5 mm, and depth of 0.5 mm. The temperature was measured by a thermocouple and adjusted with a ceramic heater. To obtain high contrast CT images of the GDL and the channels, the cell was positioned horizontally relative to the X-ray source, so that only a small portion of the X-ray beam transmits through the catalyst layer containing platinum with a large X-ray attenuation coefficient (Fig. 1). In addition, a boltless surface pressure rotating mechanism was developed to

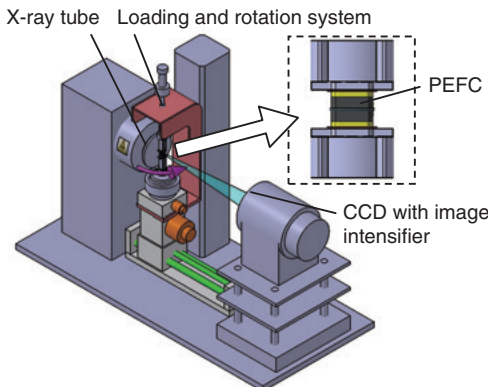


Fig. 1 X-ray CT imaging system

avoid X-ray attenuation due to the metal parts used in the cell surface pressure supporting bolts. The fuel cell was rotated while performing control to maintain a constant load using a surface pressure cylinder. When generation was performed at a constant current, there was no cell position offset or gas leaks, and measurement could be performed at a constant cell voltage.

2.2. Experimental Procedure

The proton exchange membrane used a DuPont NR212, and the catalyst layer was composed of a platinum content of 0.5 mg/cm², carbon content of 0.5 mg/cm², and an electrolyte content of 0.38 mg/cm². In addition, the GDL used Toray TGP-H-060 carbon paper, and constant current aging tests were performed at 0.6 A/cm² with the GDL assembled into the fuel cell. Following the aging tests, the Membrane Electrode Assembly (MEA) and GDL were removed from the cell and vacuum degassing was performed to completely saturate the insides of the MEA and GDL with water. The MEA and GDL were then reassembled into the fuel cell in the completely water-saturated condition and X-ray CT measurement was performed to obtain reference images in the completely water-saturated condition. After that, N₂ purge was performed and X-ray radiography was used to confirm the internal drying process to the dry condition, and then the interior condition of the fuel cell was measured using X-ray CT to obtain reference images in the dry condition.

After that, generation was performed at a constant current density, and X-ray radiography and CT measurement were performed while maintaining a stable voltage. Image processing was performed on the results to visualize the water distribution and quantify the saturation distribution inside the GDL.

2.3. X-ray CT Measurement Results

Hydrogen moisturized at a dew point temperature of 300 K was supplied to the anode side at a rate of 50 ml/min, oxygen moisturized at a dew point temperature of 300 K was supplied to the cathode side at a rate of 150 ml/min, and the test was performed at a cell temperature of 300 K.

Figure 3 shows a fuel cell image reconstructed in three dimensions by X-ray CT. This image is the result of observing the 2.5 mm diameter area around the center axis of the fuel cell shown in Fig. 2. The top side of Fig. 3 is the

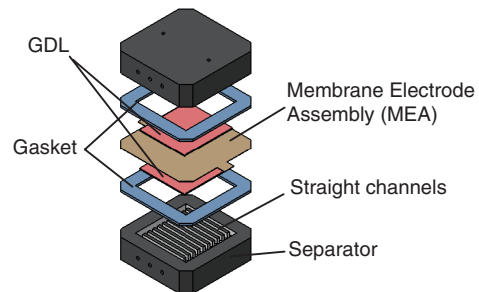


Fig. 2 Fuel cell for X-ray CT imaging

anode, the bottom side is the cathode, the green rectangular areas indicate the gas channels, and the orange rectangular areas indicate the separator ribs. In addition, the red areas indicate the GDLs, and the blue area indicates the MEA (catalyst layers and proton exchange membrane). Figure 4

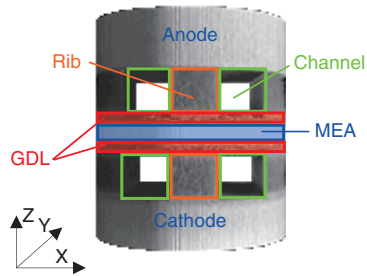


Fig. 3 Three dimensional X-ray CT image of PEFC

shows the I-V performance curve of this fuel cell, and Fig. 5 shows CT slice images obtained under some generating conditions. The respective images in Fig. 5 show (1) the reference images (dry condition), (2) the images at a current density of 0.30 A/cm², (3) the images at a current density

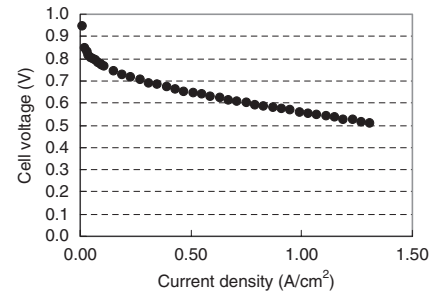


Fig. 4 I-V performance curve of PEFC

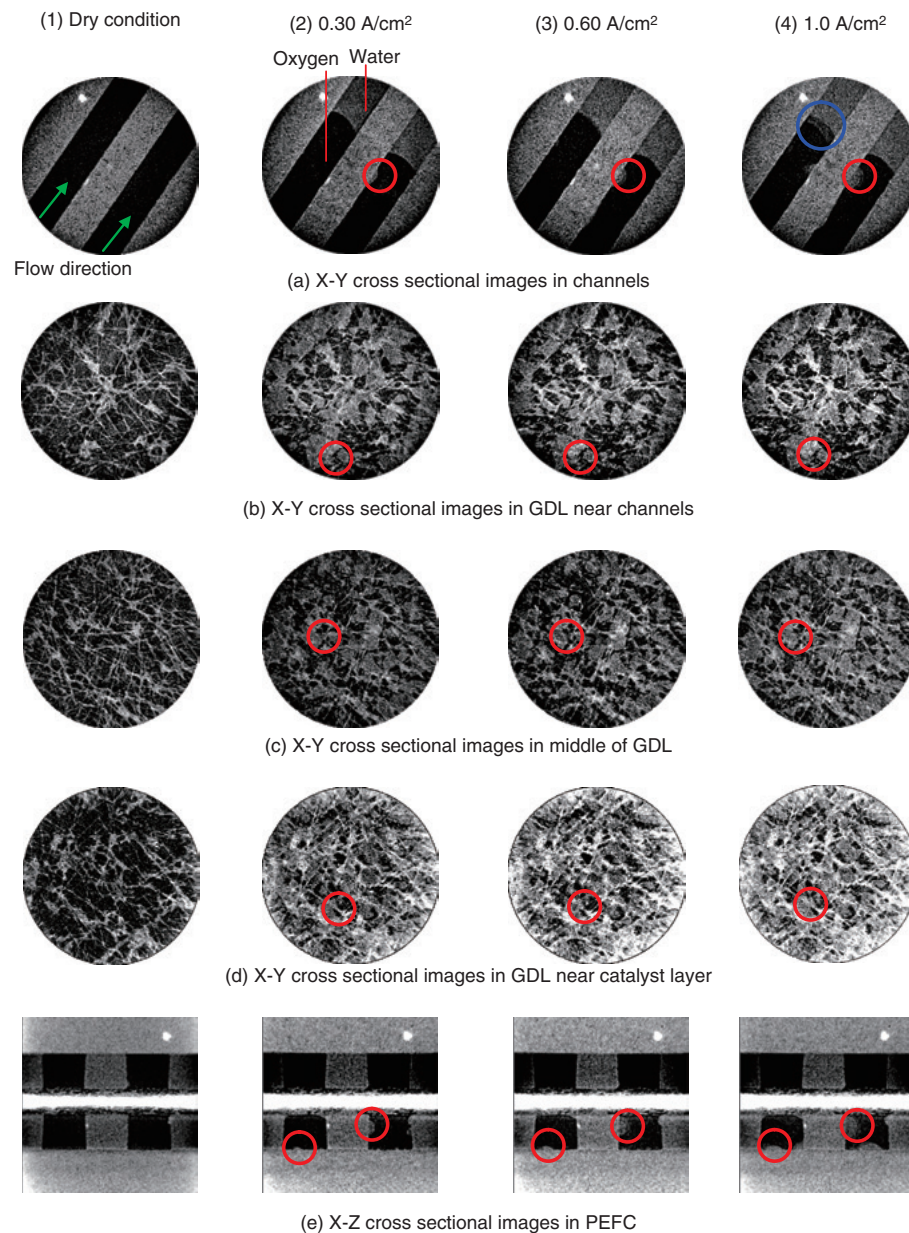


Fig. 5 Cross sectional images in generating PEFC

of 0.60 A/cm^2 , and (4) the images at a current density of 1.0 A/cm^2 . In addition, (a) to (e) show the images at the cathode, and are respectively (a) the (X-Y) cross-sections in the center of the channels, (b) the (X-Y) cross-sections of the GDL near the channels, (c) the (X-Y) cross-sections in the center of the GDL, (d) the (X-Y) cross-sections of the GDL near the catalyst layer, and (e) the (X-Z) cross-sections of the cell. The areas enclosed by the red circles show examples of the generated water.

Figure 5(a) shows the plugging phenomenon whereby liquid water is discharged to the channels at a current density of 0.30 A/cm^2 or more, with the result that the water blocks the channels. In addition, this water in the channels does not move much, so the contact boundary between the separator and the water can be observed, and the change in wettability can be measured. Figure 5(b) through Fig. 5(d) show that the water inside the GDL increases in both quantity and droplet size as the current density increases, and that the quantity of water tends to increase from the gas channel side towards the catalyst layer side. In addition, Fig. 5(e) shows that size of the water droplets in the channels increase as the current density increases. Traces of the water boundary inside a channel having moved during one rotation (30 min) of the cell can be confirmed in the area enclosed by the blue circle in Fig. 5(a)-(4). In this manner, the water distribution inside a generating fuel cell could be measured by using X-ray CT. Next, image processing technique was applied to quantify the water.

2.4. Image Processing Procedure

Image processing was performed on the data obtained by X-ray CT to identify and quantify the water distribution inside a generating fuel cell. Water stays statically under

the test conditions used in this study, so images with good contrast could be obtained. Therefore, the water distribution was extracted by the procedure shown in Fig. 6.

- (1) The dry condition images were converted to binary images and divided into fiber parts and pore (oxygen mixed with water vapor) parts, while taking into account the histogram of each cross-section and the diameter of the GDL carbon fibers.
- (2) The generating condition images were converted to binary images and divided into pore parts and fiber with water parts in the same manner as in step (1).
- (3) 3D position alignment was performed for the images obtained in steps (1) and (2) to eliminate the effects of expansion of the proton exchange membrane and thermal expansion of the materials due to generation.
- (4) The difference between the images aligned in step (3) was obtained, and the distribution of the water that increased during generation was extracted.
- (5) 3D ternary voxel data of the pores, fibers and water was created using the water data separated in step (4) and the data in step (1).

2.5. Image Processing Results

Figure 7 shows the water distribution extraction results. The respective images in Fig. 7 show (1) the images in the dry condition after N_2 purging, (2) the images at a current density of 0.30 A/cm^2 , (3) the images at a current density of 0.60 A/cm^2 , and (4) the images at a current density of 1.0 A/cm^2 . In addition, (a) to (d) show the images at the cathode, and are respectively (a) the (X-Y) cross-sections in the center of the channels, (b) the (X-Y) cross-sections of the GDL near the channels, (c) the (X-Y) cross-sections of the GDL near the catalyst layer, and (d) the (X-Z) cross-

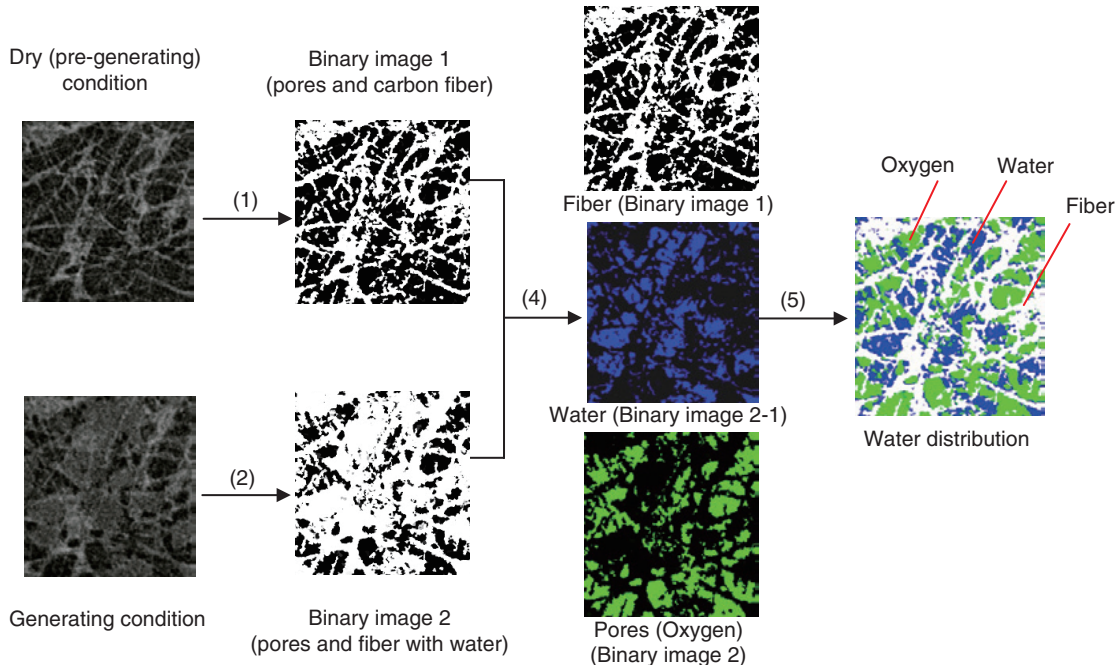


Fig. 6 Image processing procedure

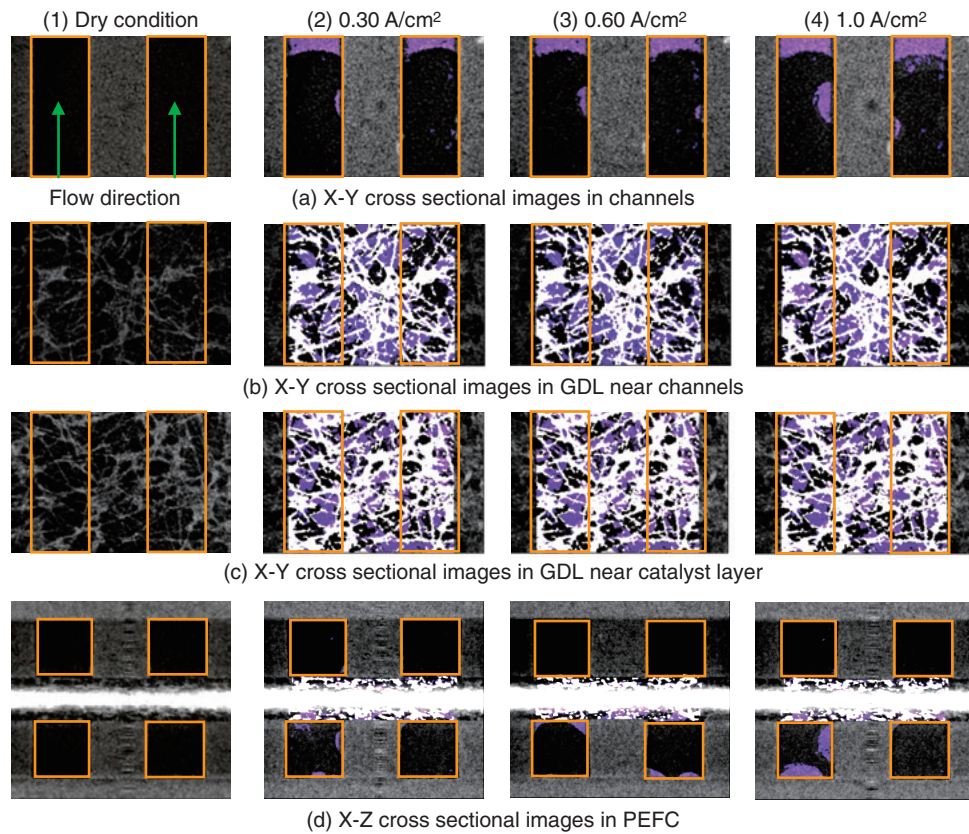


Fig. 7 Cross sectional images in generating PEFC

sections of the cell. As shown by the purple color in Fig. 7, the water distribution inside the generating fuel cell could be extracted by the image processing procedure described above.

Figure 8 shows a 3D representation of the water distribution inside the cathode gas channel and GDL at 1 A/

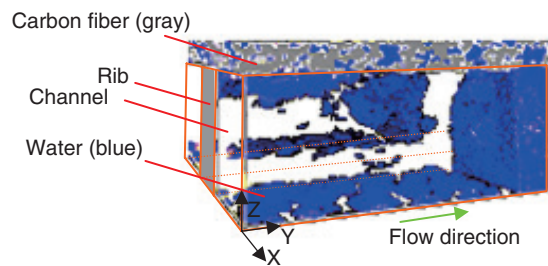


Fig. 8 Three dimensional water distribution in GDL and gas channels obtained by X-ray CT imaging

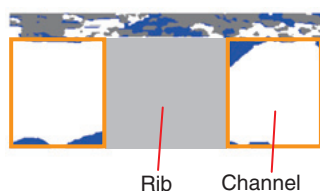


Fig. 9 X-Z cross sectional image

cm^2 . The condensed water adhering to the gas channel wall surface opposite the GDL, water being discharged along the channel wall surface, and the plugging phenomenon can all be confirmed from Fig. 8.

In addition, Fig. 9 shows the X-Z cross-sectional image and Fig. 10 shows the X-Y cross-sectional image of the water inside the GDL directly under the channels and rib. Figures 9 and 10 confirm that a large quantity of water accumulates in the GDL under the rib, and that there is less water in the GDL directly under the channels than directly under the rib.

Next, Figs. 11(a) and 11(b) show the results of analyzing the water saturation distribution at positions from the catalyst layer side to the channel side of the cathode GDL, based on the water distribution extracted by the 3D

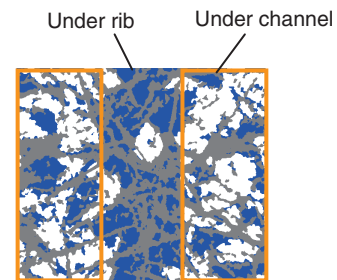


Fig. 10 X-Y cross sectional image directly under rib and channels

processing described above. Here, the water saturation is defined as the percentage of the pore area inside the GDL occupied by water. Figure 11(a) shows the water saturation distribution in the area under the channel, and Fig. 11(b) shows the water saturation distribution in the area under the rib. Overall, the area under the rib exhibits a higher water saturation than the area under the channel, and the water saturation in the area under the channel tends to decrease from the catalyst layer side towards the channel. Conversely, the water saturation in the area under the rib tends to be higher directly under the rib compared to the catalyst layer side.

The factors for water accumulating under the rib are thought to be as follows.

- (1) The separator that forms the rib is hydrophilic; that is to say, contact angle of the water becomes smaller so the water moves less easily.
- (2) The discharge distance under the rib to the channels is long, and the GDL pores also become crushed, so gas does not flow easily and the gas flow cannot easily remove the water.
- (3) The rib is a part of the separator that is cooled by the outside air, so water condenses easily.

A large quantity of water accumulates under the rib even in tests at room temperature such as the tests in this study. This indicates that wettability and pore crushing under the rib have a large effect on the water discharge behavior from the GDL to the gas channels.

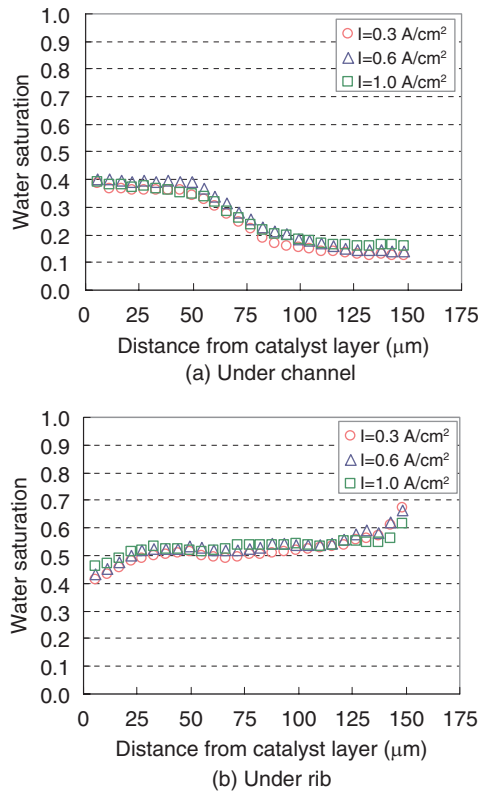


Fig. 11 Water saturation in GDL

3. Analysis of Water Transport Using X-ray Radiography

3.1. Overview of X-ray Radiography Imaging

The X-ray radiography method acquires X-ray transmission images without rotating the observation subject in the manner of X-ray CT. This means that the structure and water distribution cannot be measured in three dimensions, but real-time observation is possible. In the X-ray radiography tests, the X-ray tube used lanthanum hexaboride (LaB6) as the filament material, and a transmission-type open tube with tungsten as the target material. The detector used a 5 M-pixel CCD with an image intensifier, and measurement was performed at a tube voltage of 70 kV and a tube current of 110 μA. The exposure time was 3 s, and measurement was performed at a spatial resolution of 4.3 μm. Figure 12 shows the fuel cell used in the X-ray radiography tests. The electrode area was 10 mm x 9.3 mm, and used a 15 mm diameter carbon separator with three trapezoidal straight channels having a lower side length of 2.1 mm, an upper side length of 0.58 mm, and a height of 0.58 mm. The test procedure and image processing procedure conformed to the X-ray CT tests. Hydrogen moisturized at a dew point temperature of 310 K was supplied to the anode side at a rate of 50 ml/min, air moisturized at a dew point temperature of 317 K was supplied to the cathode side at a rate of 200 ml/min, and the test was performed at the three cell temperature conditions of 323 K, 328 K and 333 K.

3.2. X-ray Radiography Measurement Results

The effects of cell temperatures on the internal water distribution were analyzed. Figure 13 shows an image transmitted in the X direction through the fuel cell shown in Fig. 12, with each cell component labeled. The top part is the anode, and the bottom part is the cathode. In addition, the green areas indicate the gas channels, the orange areas

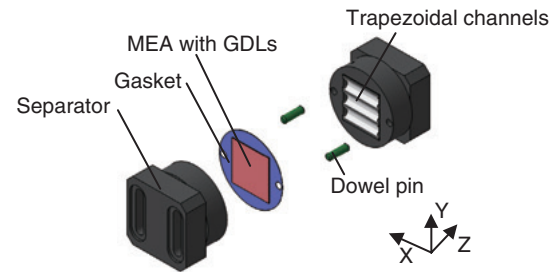


Fig. 12 Fuel cell for X-ray radiography imaging

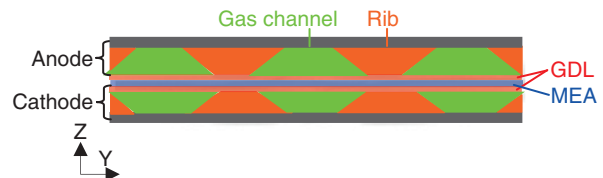


Fig. 13 X-ray radiography image of PEFC

indicate the separator ribs, the red areas indicate the GDL, and the blue area indicates the MEA. Figure 14 shows the change over time in the water content distribution inside the PEFC during the 1 A/cm² constant current test at a cell temperature of 328 K. The water content was normalized and calculated using the dry condition image and the image of the completely water-saturated condition after degassing. These images confirm that starting from the dry condition in Fig. 14(a), water produced by generation accumulates in the GDL under the ribs [Fig. 14(b), (c)], and this water is then discharged from under the ribs along the channel wall surfaces to the gas channels [Fig. 14(d), (e)]. Figure 15 shows the change over time in the water content inside the cathode GDL and gas channels at the cell temperatures of 323 K, 328 K and 333 K. At 328 K and 333 K the internal water content reaches a steady state, but at 323 K the water did not reach a steady state and continued to increase. This is thought to occur because, at a cell temperature of 323 K, the water vapor contained in the supplied gas and the water vapor produced by generation cannot be completely eliminated, and condense in the GDL and channels. Figure

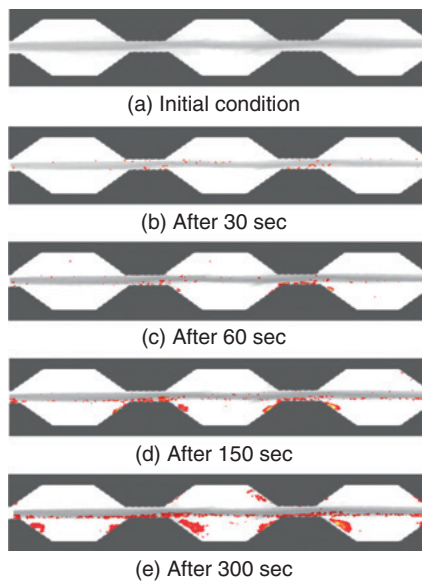


Fig. 14 X-ray radiography images

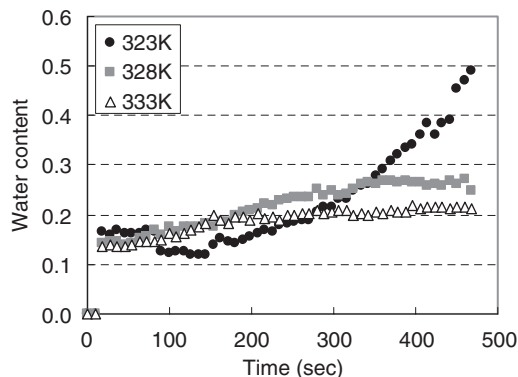


Fig. 15 Effect of operating temperature on water content in GDL and gas channels

16 shows a graph of the water content inside the cathode GDL and gas channels at 450 s after the start of generation. Figure 16 shows that when the cell temperature increased from 323 K to 328 K, the water content in the gas channels decreased by more than the water content in the GDL, and when the cell temperature increased from 328 K to 333 K, this difference became smaller. This indicates that the water inside the channels dried faster than that inside the GDL. In this manner, the change in water distribution over time could be measured using X-ray radiography.

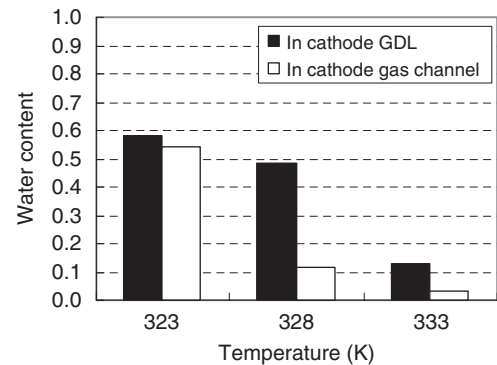


Fig. 16 Water content in GDL and gas channels

4. Conclusion

An X-ray CT measuring system that can measure the water distribution in a generating fuel cell in three dimensions with high spatial resolution and contrast, and an X-ray radiography system that can analyze the dynamic behavior of water with high temporal resolution were developed. Image processing technology was also developed, and the water distribution was quantitatively analyzed. The results showed that water produced by generation easily accumulates inside the GDL near the catalyst layer and under the ribs and water is discharged from the GDL to the channels along the separator. In addition, the plugging phenomenon, water condensation on the channel wall surfaces, and the growth process of the condensed water were also observed.

In the future, X-ray CT, which has high 3D spatial resolution, and X-ray radiography, which has high temporal resolution, will be combined to analyze the effects of operating temperature and the supply gas velocity and humidity on water discharge behavior. In addition, the optimal materials and structure will also be investigated.

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Research of Simulation Method of Hydrogen Diffusion for Fuel Cell Electric Vehicle Development

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ABSTRACT

Research was conducted on a method that can simulate hydrogen diffusion behavior in the event that hydrogen leaks into the complexly shaped space around the hydrogen tank of a fuel cell electric vehicle.

Calculations cannot be performed in a permissible time period for the vehicle development with the conventional analysis region partitioning method, so a three-part partitioning calculation method was newly devised to shorten the calculation time. In addition, in order to enhance the calculation accuracy, the turbulence models that can appropriately calculate turbulent diffusion and mixing were selected.

The results confirmed that the peak concentration when hydrogen spouts and the residual concentration after spouting can be calculated at an accuracy of within 5%, while keeping the calculation time to just four days. In addition, it was presumed that the new method can be applied to differing hydrogen spouting conditions. These results indicated that the new method is useful for the development of fuel cell electric vehicles.

1. Introduction

Honda is conducting research and development of hydrogen application technology, such as fuel cell electric vehicles that are the ultimate form of carbon reduction, with the aim of realizing a hydrogen society.

Hydrogen is colorless, odorless, has the lowest specific gravity of all the elements, and diffuses rapidly, characteristics not present in conventional fuels. Therefore, it is important to perform design in consideration of these characteristics.

Hydrogen sensors are installed in fuel cell electric vehicles to detect possible hydrogen leaks (Fig. 1). Conventionally, the installation positions of these sensors were examined using actual vehicles, and it took a few months to determine the positions. If the behavior of hydrogen that flows while diffusing can be analyzed using simulations within a short period of time, the proper

hydrogen sensor positions can be investigated from the design stage.

However, although there have been reports thus far of simulating hydrogen diffusion and verifying accuracy for simple shapes^{(1), (2)}, a survey of research reports in the field of hydrogen application technology finds no examples of proven methods with verified accuracy that can be applied to the design investigation as a simulation method for complex shapes of fuel cell electric vehicles.

Simulations of hydrogen diffusion using complex vehicle shapes have a large calculation scale and it has been challenging to perform the calculations in a practical time so far. Therefore, the method of partitioning the analysis region was modified to shorten the calculation time. In addition, turbulence models that can accurately calculate the turbulent diffusion and mixing of hydrogen were selected to secure an accuracy that enables application to design investigations.

Hydrogen diffusion around the hydrogen tank of a vehicle was simulated using these methods, and accuracy was verified by comparison with measured values obtained by high-speed response concentration measurements.

2. Development Goals

The goal was to construct a simulation method of hydrogen diffusion that can be used to investigate the

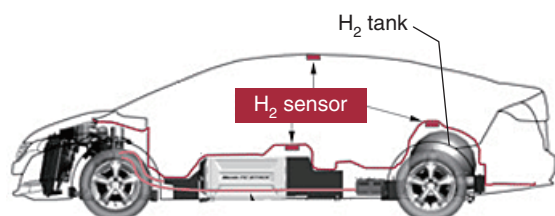


Fig. 1 Position of hydrogen sensors

* Automobile R&D Center

design of fuel cell electric vehicles. This entails analysis of changes in the hydrogen concentration over time for a complex vehicle shape, so methods for unsteady state calculations using a 3D space shape that reflects a CATIA model were considered.

The targets for calculation time and accuracy were set as follows:

- (1) Calculation time: One week or less
- (2) Calculation accuracy: Simulated hydrogen concentrations quantitatively match with measured values

The target value for calculation time was set to be advantageous when compared to the time consumed in the investigation of installation positions using conventional tests.

3. Construction of Simulation Method

3.1. Characteristics of Hydrogen Diffusion Phenomena and Issues for Calculation

In the event that hydrogen leaks inside a vehicle, the hydrogen is thought to flow and diffuse through the phenomena shown in Fig. 2. First, jet turbulence that involves compression is formed at the spouting point, and the hydrogen diffuses while rolling up the surrounding air. After that, the flow direction is changed by the structures inside the vehicle, and the behavior shifts to flow and diffusion resulting from wall turbulence due to the effects of the wall surface viscosity. As the flow progresses, the inertia force of the spouting weakens, and the effects of buoyancy become relatively dominant, so the hydrogen rises and remains in the upper areas.

When analyzing these phenomena, the following two issues arise:

- (1) Compressible flow calculations with a high calculation load must be performed using unsteady state calculations with a large number of space meshes and small time steps, which requires longer calculation time; and
- (2) The turbulence models that can accurately analyze the effects of flow and diffusion due to turbulence are unknown.

The first of these issues is described in detail in Section 3.2, and the second in Section 3.3 below.

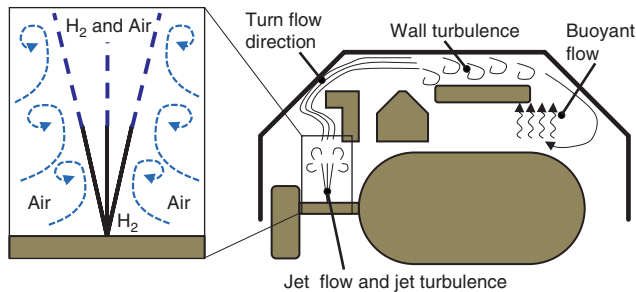


Fig. 2 Flow phenomena in case of hydrogen leak inside vehicle

3.2. Reduction of Calculation Time

3.2.1. Analysis of factors

As described in Section 3.1 above, the reasons for the longer calculation time are the following three points:

- (1) Compressible flow calculations are required
- (2) A larger number of space meshes
- (3) A larger number of time steps

Compressibility affects only the area near the spout, but compressible flow calculations must be performed for the entire analysis area, which increases the calculation load.

The reasons for a larger number of space meshes are that a large number of meshes are required to accurately reproduce the complex shapes inside the entire vehicle, and that even finer meshes are required around the spout to obtain the needed resolution for the jet flow phenomenon.

The reason for a larger number of time steps is that it is necessary to satisfy the courant number restriction in unsteady state calculations. The courant number C is expressed by Eq. (1).

$$C = \frac{v\Delta t}{\Delta x} \quad (1)$$

v : Velocity

Δt : Time step

Δx : Mesh size

The courant number expresses the number of movement meshes of the fluid per time step, and when this number is excessively large, it lowers the accuracy due to value divergence during iterative calculations, or loss of high-frequency components in changes over time. The upper limit value for performing reliable calculations is said to be approximately 40, even when using the implicit method that enables unsteady state calculations with a courant number of 1 or more⁽³⁾. In unsteady state calculations that include the area near the spout where the flow velocity v is high and the mesh size Δx is small, the time step Δt must be reduced to satisfy the courant number restriction. This results in a larger number of time steps.

3.2.2. Conventional technique

The method of imaginary source and the analysis region partitioning method are introduced in “Guidebook on Effective Utilization of H₂” published by the New Energy and Industrial Technology Development Organization (NEDO), as techniques for shortening the calculation time for simulation of hydrogen diffusion⁽⁴⁾.

The method of imaginary source does not calculate the region near the spouting point that requires compressible flow calculations. Instead, it sets the boundary condition using a logical formula for the jet flow from a circular spout, and calculates the remaining regions. It is a challenge to apply this method when there are obstacles near the spouting point, so it is not suitable for the analysis of complex shapes.

The analysis region partitioning method partitions and calculates the region near the spouting point that has a Mach number larger than 0.5 as a compressible flow region, and uses those results as the boundary condition to calculate the remaining incompressible flow region. Compressible flow calculations are limited to only the necessary region, enabling a shorter calculation time. Figure 3 shows the concept of the analysis region partitioning method. Unlike the method of imaginary source, complex space shapes can be reproduced and calculated.

The calculation time was investigated when using the analysis region partitioning method to partition at the Mach number 0.5 position, and performing unsteady state calculations for the second part. In consideration of the sound velocity of hydrogen of 1200 m/s, when the flow velocity v is 600 m/s, the courant number is 40, and the minimum mesh size required to reproduce a complex shape is 5 mm, the time step obtained by Eq. (1) is 3.3×10^{-4} s. Given current computer performance, the calculations for a time of 1 s would require approximately three weeks, even if parallel computing were performed. The aim is to analyze hydrogen diffusion behavior on the order of several seconds to several tens of seconds, so this means that the calculations cannot be performed within the goal of one week.

In order to shorten the calculation time for the second part, the partitioning position was set farther

away from the spout and the flow velocity handled by the unsteady state calculations was reduced. However, the compressible flow calculation area of the first part expanded in this case, so calculations for the first part took one week or more.

3.2.3. Three-part partitioning calculation method

The conventional analysis region partitioning method is unable to avoid the vast amount of the compressible flow calculations in the first part or the unsteady state calculations in the second part. Therefore, a three-part partitioning calculation method was devised that further increases the number of partitions. Figure 4 shows an outline of the three-part partitioning calculation method. In the first part, this method calculates only the area near the spout that requires compressible flow calculations, in the same manner as the conventional analysis region partitioning method. In the second part, incompressible flow steady state calculations are performed for the region up to the position where the courant number in the unsteady state calculations for the third part is 40 or less. The flow velocity at the partition position between the second and third parts should be approximately 10 m/s or less. In the third part, incompressible flow unsteady state calculations are performed for the entire analysis region.

This method was considered to enable the calculations for all analysis regions to be performed in a practical time.

3.3. Enhancement of Turbulence Calculation Accuracy

3.3.1. Turbulence calculation methods

In the diffusion phenomena within a flow field, the effects of turbulent diffusion and mixing whereby mixing is induced by turbulence are more dominant than the effects of molecular diffusion produced by differences in concentration. Methods for calculating turbulence include Direct Numerical Simulation (DNS), Large Eddy Simulation (LES), and the Reynolds Averaged Navier-Stokes (RANS) model. DNS and LES precisely calculate all or part of the vortex, so the calculation load is too high, which means that in practical terms it is appropriate to select RANS. Numerous RANS turbulence models have been developed, but there is no all-purpose model yet that can be applied to any calculation subject. This means that to achieve the target simulation accuracy, it is necessary to select models that can accurately calculate each partition calculation region.

Five different turbulence models that have been applied to engineering calculations were taken as candidates, and turbulence models were selected for jet turbulence produced at the spouting point and wall turbulence produced by the flow along a wall surface. Table 1 lists the candidate turbulence models.

3.3.2. Selection of jet turbulence calculation model

The case with a spout diameter of $\phi 1.4$ mm, a flow

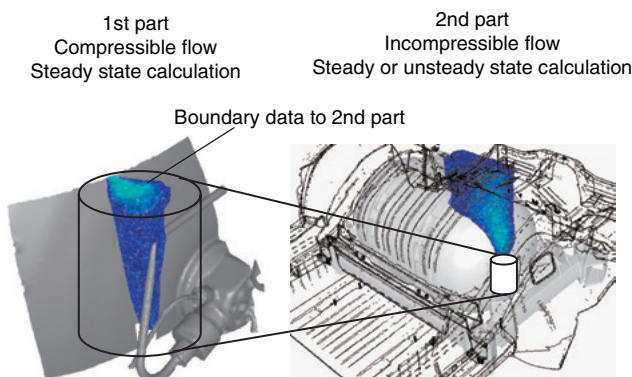


Fig. 3 Analysis region partitioning method

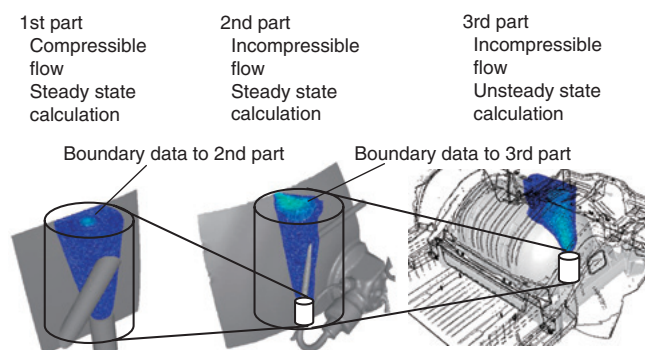


Fig. 4 Three parts partitioning method

Table 1 Turbulence models

Model	Number of equations	Characteristics
k- ω standard	2	Possible to calculate near wall region without wall function
k- ω SST	2	Hybrid model of k- ω standard and k- ϵ standard
k- ϵ standard	2	Basic 2 equations model and popular in engineering field
k- ϵ realizable	2	k- ϵ standard model with enhanced wall function
RSM	7	Possible to calculate turbulent anisotropy

SST : Shear Stress Transport
RSM : Reynolds Stress Model

rate of 120 NL/min, and spouting in the vertically upward direction was verified as a test case. Steady state calculations were performed for the region corresponding to the first and second parts of the three-part partitioning calculation method, and the hydrogen concentration calculated by each turbulence model was compared with the values measured on the spouting axis up to the position 0.2 m from the spout. Figure 5 shows these results.

This shows that the k- ω SST model is the most accurate. In addition, it was verified that hydrogen concentrations can be accurately calculated using the k- ω SST model even when the spouting conditions (spout diameter and flow rate) change. Figure 6 shows these results. The hydrogen concentrations were calculated accurately for any spouting conditions, indicating that the k- ω SST model is valid regardless of the spouting conditions. The effects of jet turbulence are thought to be dominant in the first and second parts of the partitioning calculations, so the k- ω SST model was used for these parts.

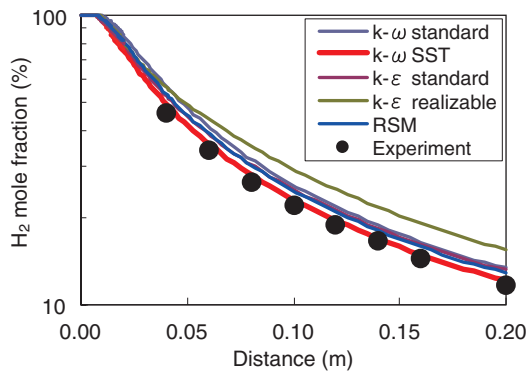


Fig. 5 Comparison of calculations with experiment to select jet turbulence model

3.3.3. Selection of wall turbulence calculation model

Verification was performed for the case with a spout diameter of $\phi 7$ mm, a flow rate of 212 NL/min, and spouting parallel to a horizontal wall surface. The range up to the 0.14 m position on the spouting axis was calculated as the first and second parts of the three-part partitioning calculation method, and the range from 0.14 m to 1 m was calculated using the different turbulence models as the calculations for the third part. Figure 7 shows the results of comparison with the measured values.

The hydrogen concentration values calculated by all of the two equation models for wall turbulence were higher than the measured values, and were unable to adequately express diffusion. On the other hand, the seven equation RSM model matched closely with the measured values. This is thought to be because the RSM model can calculate the anisotropy of the turbulent vortex, so it can express the effects of the longitudinal vortex with strong anisotropy produced by the difference in velocities near the wall surface. Therefore, the RSM model was used for the third part of the partitioning calculations that mainly calculate the flow along the vehicle floor panels.

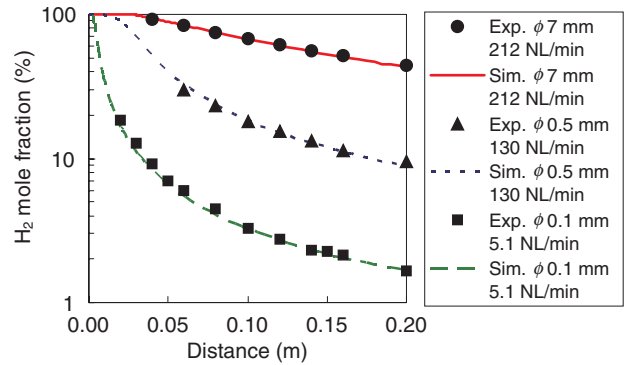


Fig. 6 Comparison of various jet flow calculations with experiments

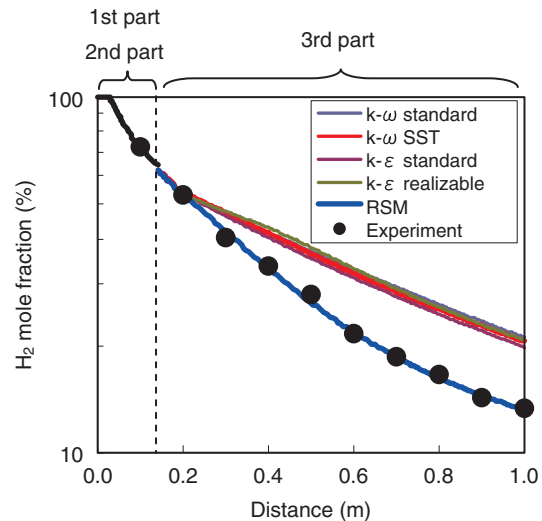


Fig. 7 Comparison of calculations with experiment to select wall turbulence model

4. Method of Verifying the Simulation Method

4.1. Verification Object and Conditions

To confirm the effects of the calculation methods, the calculation time and calculation accuracy were verified by testing and simulating the vehicle underfloor space around the hydrogen tank in a fuel cell electric vehicle in the initial development stage. Typical spouting conditions of a spout diameter of $\phi 7$ mm and a flow rate of 212 NL/min were used. Spouting position was located near the piping joint on the side of the hydrogen tank, and hydrogen was spouted in the vertically upward direction. The spouting time was set at 4 s. Figure 8 shows the hydrogen spouting position and hydrogen concentration evaluation points. Hydrogen concentration evaluation points were set in the three locations of the uppermost point of the floor panel and to the front and rear of that uppermost point.

4.2. Solver

FLUENT6.3 was used as the solver, and the calculations were performed using the implicit method and the SIMPLE (Semi-implicit Method for Pressure-linked Equations)

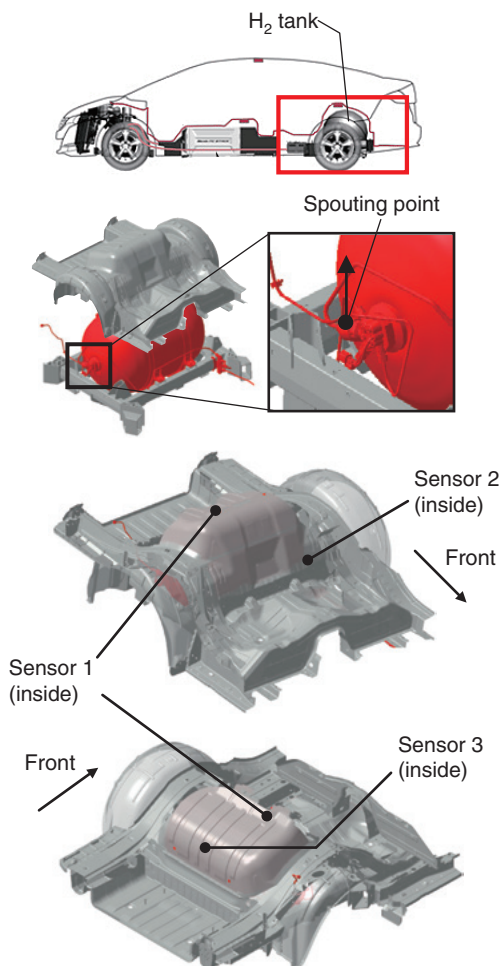


Fig. 8 Model used for experiment

algorithm. A gravity formula was added to the body force item of the Navier-Stokes equation of the model used, to take into account the buoyancy of hydrogen. A Species model based on Fick's diffusion formula that also takes into account turbulent diffusion was used to calculate the hydrogen and air mixing and diffusion.

4.3. Analysis Mesh

The analysis mesh was based on a CATIA model, and was created by first creating surface meshes for each part, then joining these surface meshes, and finally generating the entire space mesh. In order to reproduce the space shapes, the maximum mesh size around each part was 5 mm, and the mesh shape was a tetra mesh.

4.4. Experimental Method

Measurements used hydrogen and vehicle parts corresponding to the CATIA model used for the calculations, and were performed with sufficient care for safety.

Comparative verification of unsteady state calculations for the vehicle underfloor area requires high-speed response and accurate measured values over a wide concentration range. The gas thermal conductivity-type sensors often used to measure hydrogen concentrations over a wide concentration range have a 90% response time of approximately 5 s, which is insufficient for the measurements in this study. Catalytic combustion-type sensors are used to measure concentrations of 4% or less, so these are also not suited for measuring high-concentration hydrogen. Therefore, measurement used the HX-07i ionization detector (made by NOHMI BOSAI LTD.)⁽⁵⁾, which can measure a wide range of hydrogen concentrations and is designed specially for high-speed response measurement.

5. Verification Results

Figure 9 shows the calculation results for each part of the three-part partitioning calculations as hydrogen concentration contour diagrams. The calculation results for the third part enabled visualization of the flow behavior as the hydrogen diffused. The time required for the calculations was one day for the first and second parts, and three days for the third part, for a total of four days, so the calculations could be performed within the target time of one week.

Figure 10 shows the results of comparison with the measured values at each hydrogen concentration evaluation point. The calculated values matched closely with the measured values at each evaluation point. In particular, the concentration at the uppermost point of the floor panel (Sensor 1) could be calculated at an accuracy of within 5% up to 30 s after hydrogen spouting. In addition, the peak concentration and the concentration after 30 s could be calculated at an accuracy of within 5% at the two evaluation points to the front and rear of the floor (Sensors 2 and 3).

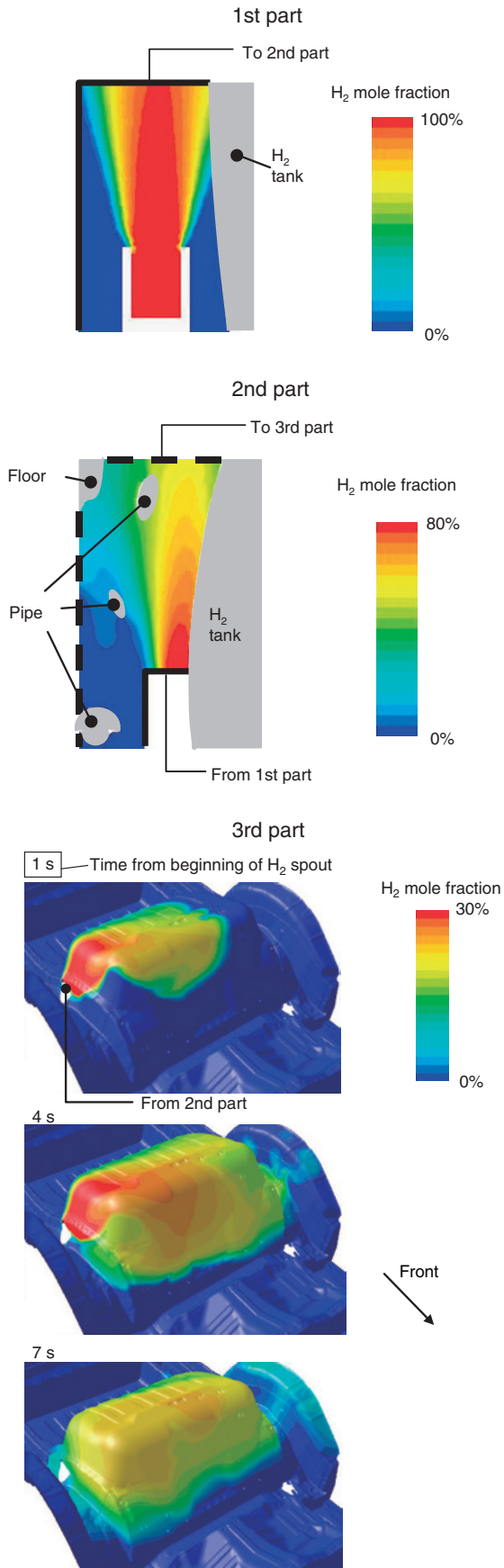
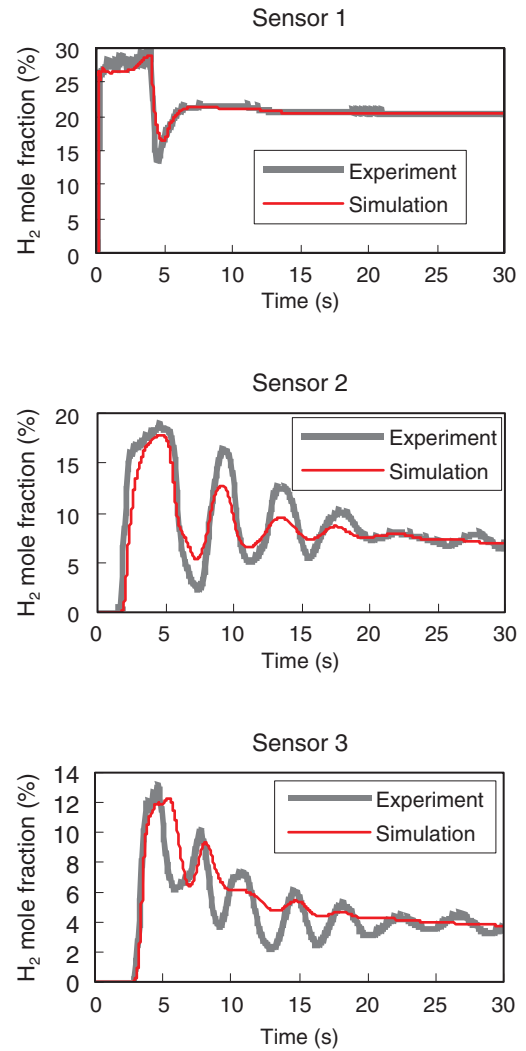


Fig. 9 Hydrogen mole fraction contours


 Fig. 10 H₂ mole fractions at sensor positions

The hydrogen concentration rise time and concentration fluctuation behavior after the end of hydrogen spouting were also closely reproduced.

6. Conclusion

A three-part partitioning calculation method was newly devised as a simulation method of hydrogen diffusion that is useful for investigating the design of fuel cell electric vehicles, and appropriate turbulence models were selected. The effects were verified, and the results confirmed the following two points:

- (1) The calculation time for all regions was four days, which achieved the target of one week or less; and
- (2) The calculated hydrogen peak and residual concentrations matched the measured values within 5%.

It was confirmed that the hydrogen behavior near the spout can be accurately calculated even under differing spouting conditions. The new method showed a wide application prospect for the analysis of various leakage

phenomena inside a vehicle.

High expectations are placed on hydrogen energy for the future of the global environment, and it is desirable to be able to safely handle 'invisible' hydrogen. There are hopes that this study and similar research can help reliably promote technology to make hydrogen more 'visible,' and by using this technology, the authors wish to accelerate the development of vehicles and systems for the control and management of hydrogen.

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Study of Oil Consumption in Small-displacement 4-stroke Air-cooled Engines

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ABSTRACT

A causal analysis of oil consumption was conducted for 4-stroke small-displacement air-cooled engines, which experience high temperature operations. The majority of oil consumption occurs as a result of oil loss that goes up into the combustion chamber, and the temperature of the cylinder walls affects the oil consumption rate. Results indicated that it was possible to control the oil consumption rate by reducing the cylinder wall temperature. It was also shown that the reduction of the width of the oil ring assembly and the increase of its coefficient of flexibility, in addition to the alleviation of the piston slap, were effective means to reduce the oil consumption rate. It was further demonstrated that the NOACK volatility of the oil has a significant effect on oil consumption in an air-cooled engine, in every part of which the temperature gets high.

1. Introduction

In 4-stroke engines, as in other types of engines, oil consumption is uneconomical and may result in a cause of functional failures due to the carbon deposits accumulated in the combustion chamber. A great deal of researches have attempted to analyze oil consumption in order to resolve these issues⁽¹⁾⁻⁽⁹⁾.

The air-cooled engines extensively used in motorcycles are characterized by higher operating temperatures than water-cooled engines. Because small-displacement engines in particular are often used in the high-load and in high-speed range, they are usually put under high temperature operating conditions. In addition, small-displacement engines have smaller cylinder-bore diameters but the tolerances for manufacturing of their pistons, piston rings and other parts cannot be proportionally reduced to their smaller bores, resulting in a comparative disadvantage to larger engines, in terms of accuracy assurance. Because few studies of oil consumption have focused on these characteristics, in this paper, a causal analysis on oil consumption is conducted using an air-cooled 49 cm³ scooter engine for Japanese domestic market, aiming at the reduction of oil consumption. This paper will present the results of this research.

2. Test Method

2.1. Oil Consumption Measurement Method

As a method of measuring the oil consumption rate, a volumetric method, in which the rate of oil consumption is measured directly from the difference between the volumes of oil before and after engine operation, is available. This is a simple and highly precise method of measuring oil consumption, but it necessitates a prolonged period of testing, and can be used only when specific operating conditions are maintained. Furthermore, by this method, it is impossible to measure the instantaneous oil consumption rate during transient operation. On the other hand, the SO₂ tracer method is also available for the measurement of oil consumption. In this method, sulfur is added to the oil and the oil consumption rate is measured from the concentration of sulfur in the exhaust gas⁽¹⁰⁾. This method makes it possible to measure the instantaneous oil consumption rate under transient operating conditions in a short period.

Tests described in this paper were conducted using either of these measurement methods. The measurement conditions and methods will be described in detail in the relevant sections below. SAE viscosity 10W-30 oil was used in the tests, and the MEXA-1170SX exhaust SO₂ analyzer manufactured by HORIBA, Ltd. was employed as the measurement device for the SO₂ tracer method.

* Motorcycle R&D Center

Table 1 Specifications of test engine

Engine type	4-stroke, SI engine
Cooling system	Forced air-cooling
Bore x Stroke	37.8 x 44.0 mm
Number of cylinders	1
Displacement	49 cm ³
Compression ratio	10.1
Valve drive system	OHC
Maximum output	3.0 kW / 8250 r/min
Cylinder	Cast-iron sleeve
Oil ring	Multi-piece steel rail ring (2.0 mm width)

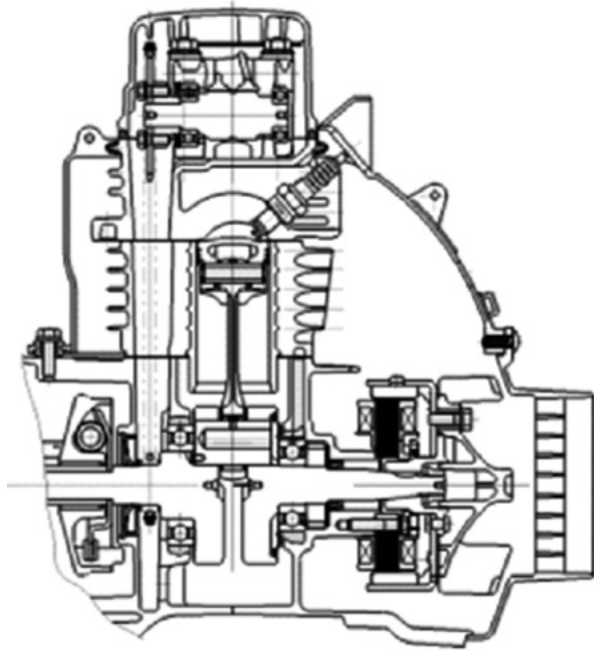


Fig. 1 Layout of test engine

2.2. Test Engine

Table 1 shows the specifications, and Fig. 1 shows the layout, of the small-displacement, forced air-cooled scooter engine used in the tests.

3. Factor Analysis of Oil Consumption

3.1. Oil Consumption in Small-displacement Air-cooled Engines

In order to obtain a macroscopic understanding of oil consumption, the oil consumption rate was measured using the SO₂ tracer method in relation to the temperature conditions and/or the operating conditions in small-displacement air-cooled engines. The oil consumption rate was measured while increasing oil temperature from cold engine temperature to the maximum temperature under actual driving condition of the fixed vehicle speed of 50 km/h (engine speed: 8000 rpm; throttle opening: 52%). The results are shown in Fig. 2. This graph shows that the oil consumption rate increases with increasing oil

temperature, and that the measured value is not stable in the higher oil temperature conditions.

Figure 3 shows the results of measurements of the oil consumption rate in relation to the engine speed and the load conditions. The measurements were carried out when the oil temperature became stable in steady driving mode. This graph indicates that the oil consumption rate tends to increase as load and engine speed increase. In Fig. 4, the vertical axis is altered to the oil temperature for the same tests as shown in Fig. 3. This graph indicates that the oil temperature has the same tendency of increase with the rise of the engine load and the engine speed just as the oil consumption rate mentioned above. Figure 5 shows the results of a survey of oil temperature in Honda motorcycle engines using different cooling systems. This figure shows maximum oil temperatures under actual driving conditions, and indicates that oil temperature tends to become higher in air-cooled engines than in water-cooled engines. Thus, in terms of oil consumption, small-displacement engines are in adverse conditions such as frequent use with heavy loads, high engine speed, and high oil temperature with air-cooled systems.

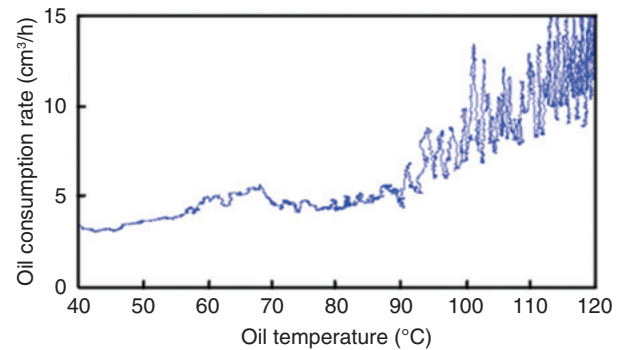


Fig. 2 Relation between consumption rate and temperature of engine oil

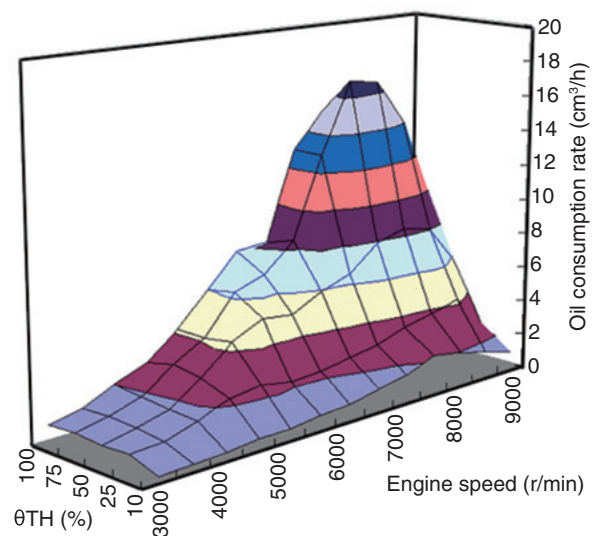


Fig. 3 3D-map of engine oil consumption rate

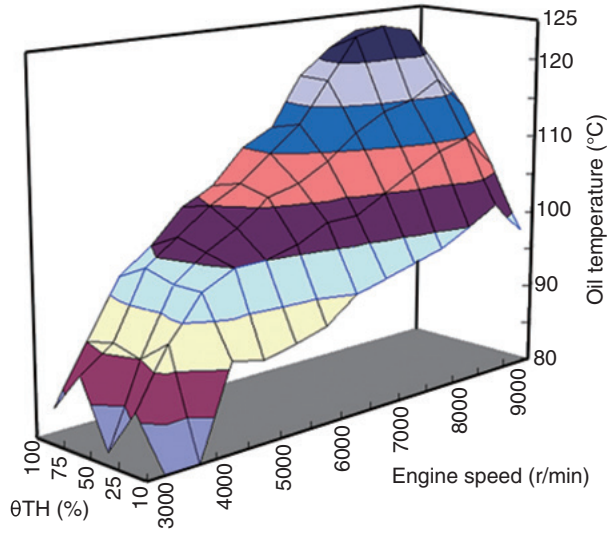


Fig. 4 3D-map of engine oil temperature

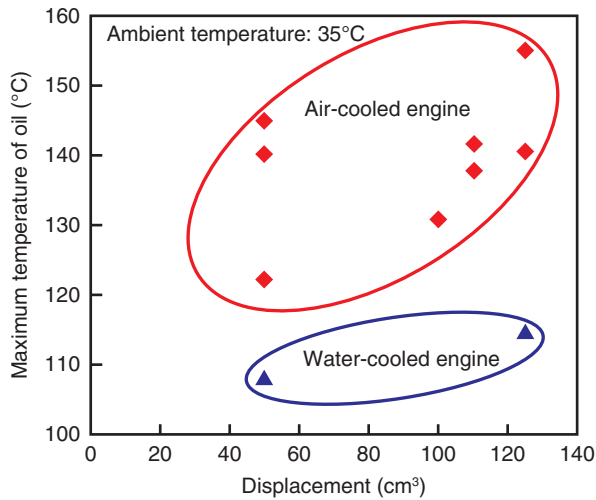


Fig. 5 Difference of oil temperature by cooling system and engine displacement

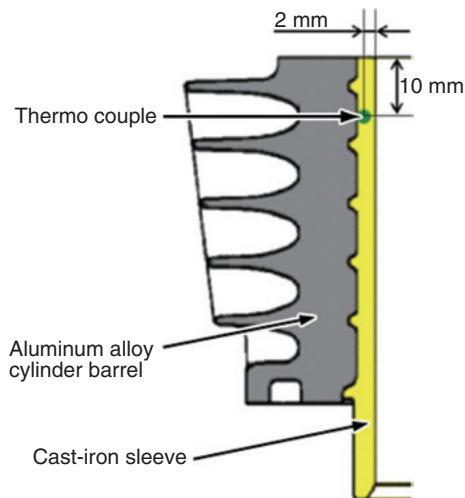


Fig. 6 Measurement point of cylinder wall temperature

3.2. Temperature Factors which Affect Oil Consumption

The results discussed in Section 3.1 indicate that the oil consumption rate is correlated to temperature, but the dominant temperature factors were not known yet. Therefore, the oil temperature (in the oil pan), the temperature at the spark plug seat, and the cylinder wall temperature were measured, as they are considered to be the potential factors to give a large amount of effects on oil consumption. Figure 6 shows the measurement point for cylinder wall temperature. In this research, the cylinder wall temperature on the inlet port is selected as the representing value. In order to investigate the effect of the temperature factors on oil consumption which is outlined above, the oil consumption rate was measured using the SO_2 tracer method while varying temperatures in those three measurement points by changing flow rate of cooling air against the engine in steady driving mode. The results of these measurements are shown in Fig. 7. These graphs indicate that the oil consumption rate has the highest correlation with cylinder wall temperature. Therefore, the cylinder wall temperature was chosen as the reference point in this research.

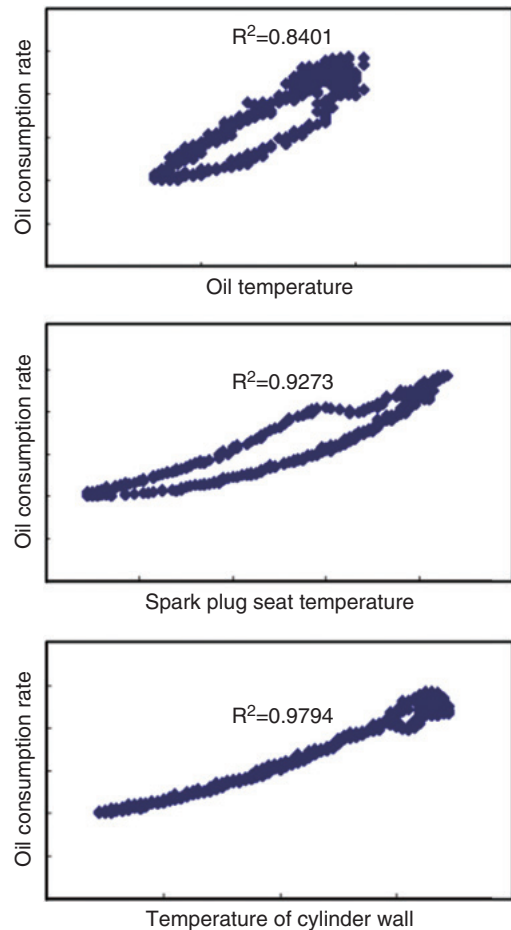


Fig. 7 Relation between oil consumption rate and temperature in each part of engine

3.3. Oil Dissipation Paths

Oil dissipation processes can be divided broadly into the following categories:

- (1) Discharge from the crankcase together with blow-by gas
- (2) Oil that goes up into the combustion chamber via gaps among the piston, the piston ring and the cylinder
- (3) Oil that goes down into the port via gaps among the valve, the valve stem seal and the valve guide

With regard to (1), blow-by gas normally flows back into the intake system, and no difference is observed in the oil consumption rate even when it is discharged to the atmosphere without introduction to the intake system. Given this, it was understood that (1) represents an extremely small amount. Regarding (2) and (3), the engine with special configuration shown in Fig. 8 was employed to measure the ratio between (2) and (3). In this configuration, the internal space within the engine casing where the lubrication oil circulates is divided into two areas: the area where the oil might go up into the combustion chamber (hereafter “Area *a*”); and the area where the oil might go down into the port (hereafter “Area *b*”). The SO₂ tracer method was employed as the measurement method, and oils with sulfur and without sulfur were used.

Vehicle speed was fixed at 50 km/h (engine speed: 8000 rpm; throttle opening: 52%) for the tests, and the oil consumption rate was measured as the cylinder wall temperature increased from cold engine temperature. In the first round of measurements, oil with added sulfur was used

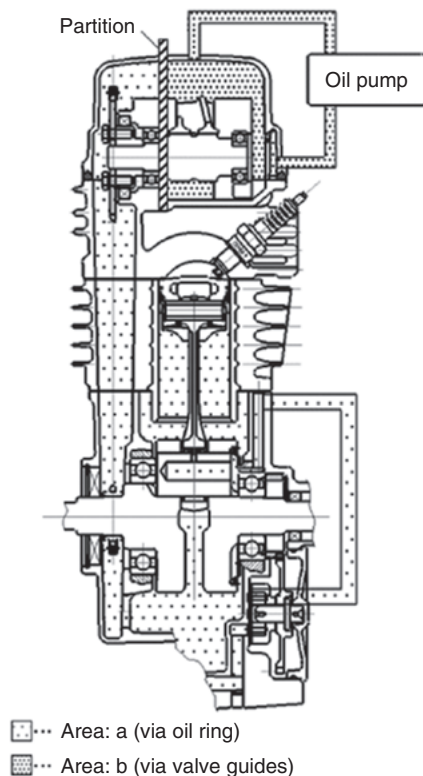


Fig. 8 Isolated structure of lubrication area for analyzing inflow path to combustion chamber

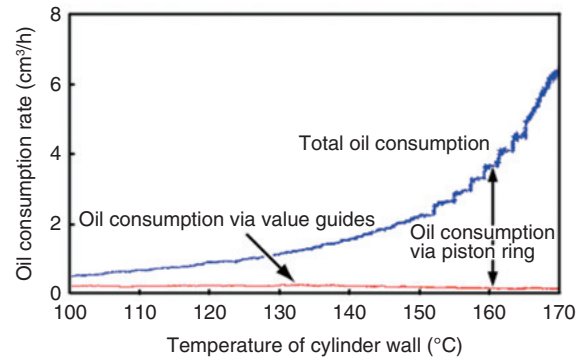


Fig. 9 Analysis of inflow path to combustion chamber

in both Area *a* and Area *b*, and the total oil consumption rate was measured. Next, oil without added sulfur was used in Area *a* and oil with added sulfur was used in Area *b*, and only the oil consumption rate due to oil loss via the valve guides was measured. The results of these measurements are shown in Fig. 9. This graph shows the total oil consumption rate and the oil consumption rate due to oil loss via the valve guides. The difference between the two rates is the oil consumption rate due to oil loss via the piston rings. These results indicate that as the temperature of the cylinder wall increases, the oil consumption rate due to oil loss via the piston ring goes up extensively, making up large portion of the total. Temperature has almost no effect on the rate of oil consumption which arises from the oil loss via the valve guides.

4. Methods of Reduction of Oil Consumption Rate

4.1. Method of Reduction of Oil Consumption Rate by Modifying Oil Rings

The fact that the oil consumption rate is not stable at high temperatures, as indicated by the results shown in Fig. 2 (Section 3.1.) was considered, focusing on the effect of cylinder deformation on this phenomenon. The test engine used in this research employed an aluminum cylinder with a cast-iron sleeve which is shown in Fig. 6. Figure 10 shows the deformation that appears in the cylinder bore and

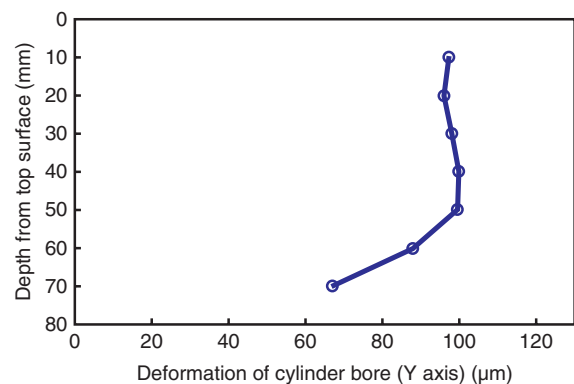


Fig. 10 Deformation of cast-iron sleeve type cylinder bore at 160°C

Fig. 11 shows deformation in circularity of the cylinder on the condition of the cylinder block fixed with bolts tightened at the specific torque and heated up to 160°C. The potential of reduction of oil consumption by enhancing the performance of the oil ring in relation to this deformation of the cylinder was investigated. The test engine used in this research employed a three-piece oil ring type shown in Fig. 12. It has been reported for the systems using this type of oil rings that, when piston slap is occurring, oil consumption rate gets lower if the post-assembly width h_1 of the oil ring is small, because the smaller the width h_1 is, the smaller the separation distance away from the cylinder wall is (shown in Fig. 12)⁽¹¹⁾. The coefficient of flexibility is available as an index of the flexibility of the oil ring in relation to deformation of the cylinder in the radial direction⁽¹¹⁾. The coefficient of flexibility, k , is defined by Equation (1). Higher values for k indicate higher flexibility.

$$k = \frac{3 \cdot Ft \cdot d_1^2}{E \cdot h_2 \cdot a_1^3} \quad (1)$$

k : Coefficient of flexibility

Ft : Tension (N)

d_1 : Cylinder diameter (mm)

E : Young's modulus (MPa)

h_2 : Width of side rail (mm)

a_1 : Thickness of side rail (mm)

Based on these facts, an experimental oil ring (Oil ring b), in which h_1 , h_2 , and a_1 were reduced in relation to the

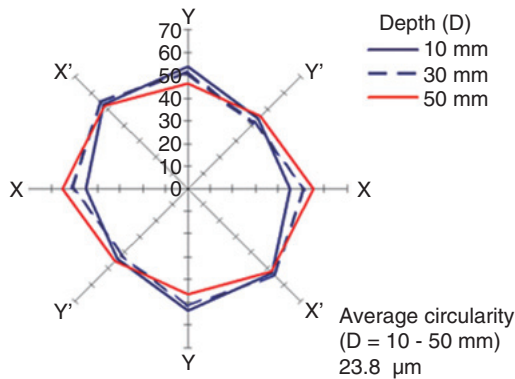


Fig. 11 Circularity of cast-iron sleeve cylinder bore at 160°C

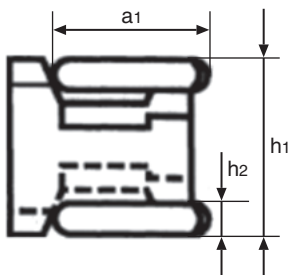


Fig. 12 Structure of oil ring

Table 2 Specifications of oil ring

	a(Standard)	b(Modified)
h_1 (mm)	2.0	1.5
h_2 (mm)	0.35	0.3
a_1 (mm)	1.7	1.1
Piston ring closed gap (mm)	0.45	0.35
Tension(N)	11.7	13.2
Coefficient of flexibility: k	0.15	0.72

standard ring (Oil ring a), was produced. Table 2 shows the specifications of the oil rings. Using Oil ring b , the oil consumption rate was measured using the SO_2 tracer method as the temperature of the cylinder walls increased from cold engine temperature, at a fixed vehicle speed of 50 km/h (engine speed: 8000 rpm; throttle opening: 52%). The results are shown in Fig. 13 (as “B (Oil ring b)”). The result of oil consumption rate shown as A in Fig. 13 is plotted against the temperature of the cylinder wall, which is altered from the oil temperature shown in Fig. 12, where both temperatures are detected simultaneously in the test. The use of oil ring b in place of oil ring a reduced the oil consumption rate at a cylinder wall temperature of 170°C from approximately 15 cm³/h to 8 cm³/h, a reduction of approximately 45%. At the same time, the fluctuations in the high-temperature range disappeared. This is believed to be because of an increase in the flexibility of the oil ring in relation to the piston-slap phenomenon and the deformation of the cylinder. This demonstrates that the reduction of the abovementioned dimensions is an effective method of reducing the oil consumption rate even in small-displacement engines with small cylinder bores.

4.2. Reduction of Cylinder Wall Temperature

Modification of cooling fins and the cooling fan in order to enhance cooling performance may be considered among methods of reducing the temperature of the cylinder walls, but the present research studied the effect of the use of a solid aluminum cylinder in reducing the

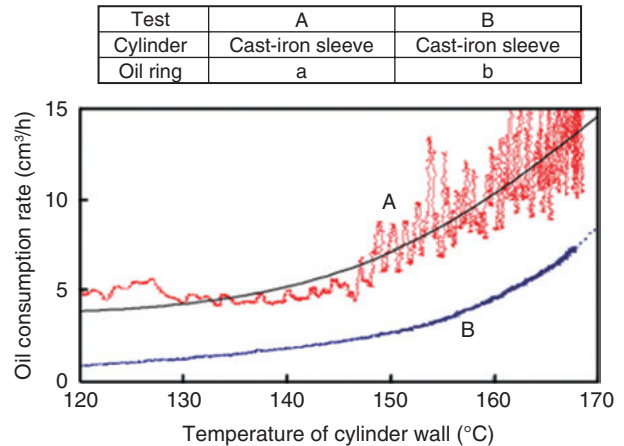


Fig. 13 Effect of modified oil ring for oil consumption rate

oil consumption rate. Cylinders of this type are generally indicated as displaying good heat radiation and as offering advantages in relation to thermal deformation. The cylinder is equivalent to the aluminum cylinder with a cast-iron sleeve in its external shape and the roughness of cylinder wall surfaces, and in the tolerance of the internal diameter in its preassembled state. Oil ring *b*, shown in Section 4.1 to be effective in reducing the oil consumption rate, was used as the piston ring. Figure 14 shows the maximum cylinder wall temperature by engine speed. The temperature in the case of the solid aluminum cylinder is lower than for the aluminum cylinder with a cast-iron sleeve, with a difference of approximately 15°C in maximum temperatures between the two.

Figures 15 and 16 respectively show the deformation that appears in the cylinder bore and the circularity of the solid aluminum cylinder when it was tightened at the specific torque and heated to 160°C. Compared to the degree of deformation of the aluminum cylinder with a cast-iron sleeve, the circularity of the solid aluminum cylinder was lower, but the deformation of the cylinder bore due to thermal expansion was greater.

The oil consumption rate, while the temperature of the cylinder wall was increased from the temperature of cold

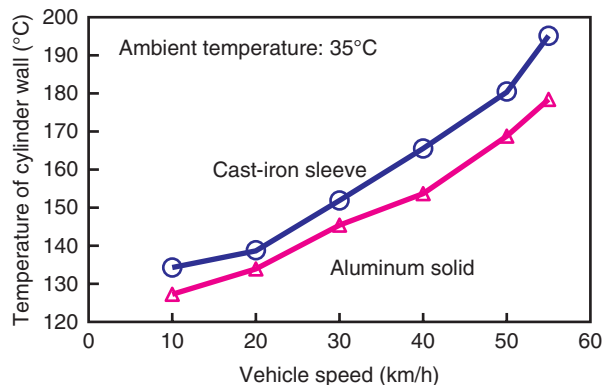


Fig. 14 Comparison of cylinder wall temperatures by difference of material

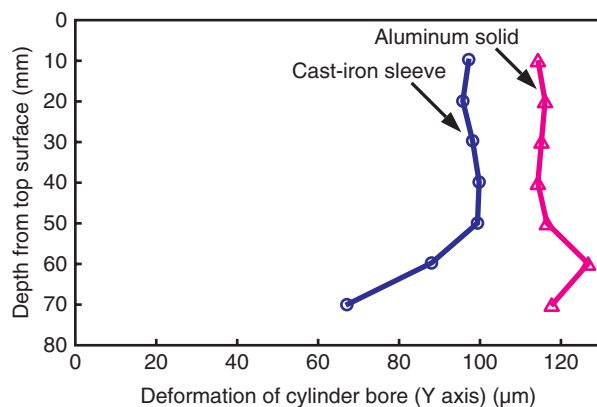


Fig. 15 Deformation of aluminum solid cylinder bore at 160°C

engine at a fixed vehicle speed of 50 km/h (engine speed: 8000 rpm; throttle opening: 52%), was measured by SO₂ tracer method and compared with that in the case of the aluminum cylinder with a cast-iron sleeve. The results are shown in Fig. 17 (as "C"). The oil consumption rate for the solid aluminum cylinder is slightly higher at identical cylinder wall temperatures, and the level of fluctuation in the oil consumption rate was also higher in the high-temperature range. This is thought to be because, despite the fact that results for circularity were better in the solid aluminum cylinder, piston slap increased due to an increase in the deformation of the cylinder bore.

In order to alleviate piston slap, tests were therefore conducted under the conditions described above using a piston in which the top land diameter and the second land diameter were increased by 50 μm. The results are shown (as "D") in Fig. 17. The results show that increasing the land diameters reduced the oil consumption rate. This effect could also be verified by the fact that the figures were lower than those for "B (Oil ring: b)" in Fig. 13, and that the fluctuation in the high-temperature range was also not present.

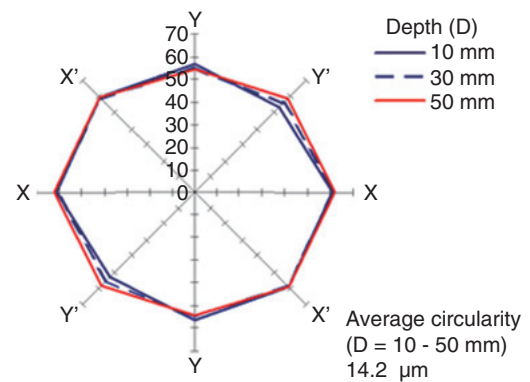


Fig. 16 Circularity of aluminum solid cylinder bore at 160°C

Test	B	C	D
Cylinder	Cast-iron sleeve	Aluminum solid	Aluminum solid
Oil ring	b	b	b
Piston	Standard	Standard	Top/2nd land+ 50 μm

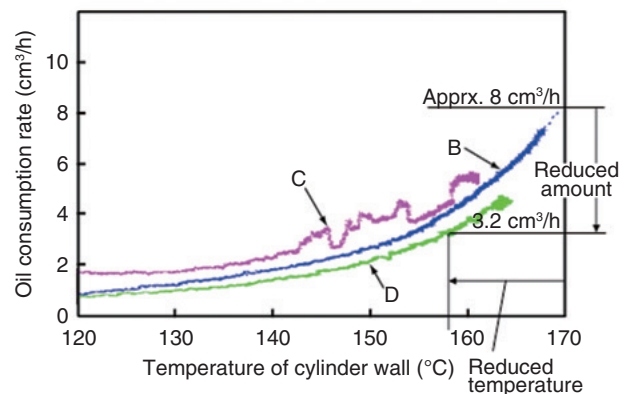


Fig. 17 Effect of aluminum solid cylinder with modified piston for oil consumption rate

Given the above, a significant reduction in the oil consumption rate can be expected from this modification when combined with the reduction in the maximum temperature of the cylinder walls, as shown in Fig. 14. Assuming a cylinder wall temperature of 170°C in the aluminum cylinder with a cast-iron sleeve and projecting a temperature of 158°C in the solid aluminum cylinder (from the temperature difference at a vehicle speed of 50 km/h in Fig. 14) under identical conditions, the oil consumption rate for a combination of the solid aluminum cylinder and the piston modified to counter piston slap would be reduced from approximately $8\text{ cm}^3/\text{h}$ to $3.2\text{ cm}^3/\text{h}$, a reduction of approximately 60%. Compared to the unmodified base engine, the oil consumption rate would be reduced from approximately $15\text{ cm}^3/\text{h}$ to $3.2\text{ cm}^3/\text{h}$, a reduction of approximately 80%.

4.3 Effect of NOACK Volatility on Oil Consumption

Because the cylinder walls of small-displacement air-cooled engines reach high temperatures, it is thought that oil consumption due to oil loss via the piston ring also includes a significant proportion of oil consumption due to oil evaporation. NOACK volatility (amount of evaporation over one hour at 250°C in a vacuum) is generally used as an index of the amount of oil evaporation. In the tests described above in this paper the oil with NOACK volatility of 14.5% was used. Oils with NOACK volatility figures of 10.5% and 3.2% were created and compared. Because it is considered that fixed operating conditions must be maintained in order to measure the oil consumption rate due to evaporation, the maximum oil temperature under actual driving conditions was maintained at a fixed vehicle speed of 55 km/h (engine speed: 8800 rpm; throttle opening: 100%), and measurements were conducted following 20 hours of continuous operation using the volumetric method. In addition to the test engine, the same measurements were conducted in a water-cooled engine of identical displacement. The results of these measurements are shown in Fig. 18. The reduction of NOACK volatility reduced the

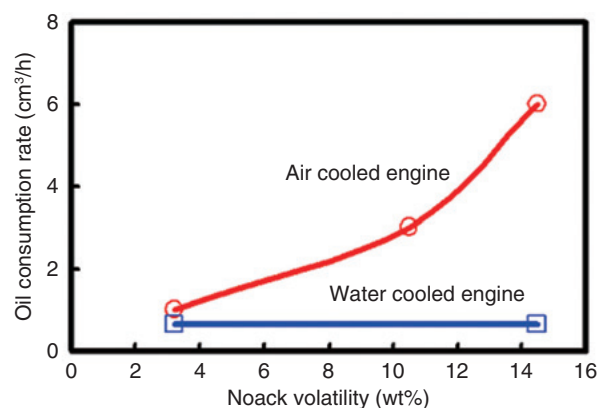


Fig. 18 Influence of NOACK volatility for oil consumption rate

oil consumption rate in the air-cooled engine. By contrast, there was virtually no change in the oil consumption rate in the case of the water-cooled engine. The reduction of NOACK volatility can therefore be indicated as an effective method of reducing the oil consumption rate in high-temperature, small-displacement air-cooled engines.

5. Conclusion

The dominant factors in oil consumptions in 4-stroke small-displacement air-cooled engines, and the methods of reducing those oil consumptions were studied. The following results were obtained.

- (1) It was demonstrated that the oil consumption rate tended to increase with increasing temperature. In addition, the temperature of the cylinder walls was the temperature factor which displayed the highest correlation with the oil consumption rate.
- (2) With regard to oil flow paths, a high proportion of oil consumption in the high temperature range occurs due to oil loss into the combustion chamber via gaps between the pistons, piston rings, and cylinders.
- (3) The following methods are effective in reducing the oil consumption rate in small-displacement air-cooled engines.
 - The oil consumption rate can be reduced by reducing the temperature of the cylinder walls.
 - Enhancing the flexibility of the oil rings in relation to cylinder deformation and piston slap, and alleviating piston slap, are effective methods of reducing the oil consumption rate even in small-displacement air-cooled engines, which has comparative disadvantages to larger engines in terms of accuracy assurance for the pistons, piston rings and other parts.
 - The use of the two measures described above can be expected to reduce the oil consumption rate of the test engine by approximately 80% against the unmodified base engine (cylinder wall temperature: 170°C).
 - The oil consumption rate can be reduced through the use of an oil with a reduced NOACK volatility. The effect is significant in small-displacement air-cooled engines, which operate at higher temperatures than water-cooled engines.

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Study on Controlled Auto-ignition Using Sequence of Low-temperature Combustion States

- Macroscopic Understanding of Controlled Auto-ignition and Approach from Gas Exchanging States -

Hideaki MORIKAWA*

ABSTRACT

A possibility to control the auto-ignition timing and the heat release process was experimentally examined. Through a macroscopic analysis of the controlled auto-ignition (CAI), two physical parameters were introduced to explain the CAI condition. These are T_{AI} , which is the in-cylinder gas temperature at the end of compression stroke, and ΔT , which indicates the heat capacity of the working gas. The experimental analysis revealed that T_{AI} dominates the onset of auto-ignition and ΔT dominates its heat rising process. If these parameters were maintained in certain ranges, desirable CAI should be generated. The both parameters are thermo-dynamically calculated with the exhaust gas recirculation (EGR) ratio and the equivalence ratio only.

In order to examine the gas exchange state that allows of the above-mentioned condition, a typical gas exchanging chart that represents all the possible gas-exchange states in a naturally aspirated four-stroke engine was introduced. On this map, the CAI area and the surroundings were carefully plotted. As the result, it was revealed that there was the approach to the CAI from the SI combustion with a sequence of low-temperature combustion due to the dilution on the map.

The experimental results finally suggest that BSFC and BSNO_x of a 1300 cm³ motorcycle could be decreased by 20% and 90% respectively in the LA-4 mode, by utilizing the sequence of the low-temperature combustion including the CAI.

1. Introduction

Controlled auto-ignition (CAI)⁽¹⁾ has attracted attention as a method of combustion that simultaneously reduces both fuel consumption and NO_x emissions. The issues to be resolved for practical use of CAI are (1) the realization of methods of controlling ignition timing and heat release following ignition, (2) expansion of the zone of operation of CAI, and (3) appropriate transition between auto-ignition (AI) and spark ignition (SI).

In-cylinder working gas temperature is the key to ignition timing. 1000 K + α is known to be the auto-ignition temperature (T_{AI}) in the case of gasoline engines⁽²⁾. With regard to a method of controlling heat release, the concept of ΔT as a parameter for the evaluation of the speed of heat release has been introduced⁽³⁾. ΔT can be theoretically derived by treating in-cylinder combustion phenomena as zero-dimension phenomena. Amid a great deal of detailed research concerning CAI⁽⁴⁾ from the perspective of chemical reactions, this paper will attempt an experimental

investigation of the potential for the realization of adequate CAI by achieving appropriate values for T_{AI} and ΔT . It is believed that adopting a macroscopic understanding on CAI and taking a simpler approach to combustion control in this way will be effective when CAI is applied practically.

The major characteristic of AI combustion is the potential for high thermal efficiency and low NO_x emissions offered by low-temperature combustion. Because of this, the expansion of the zone of operation of CAI is one area of focus⁽⁵⁾. Supercharging may be considered as an effective technological method of expanding the upper load limit for CAI⁽⁶⁾. However, because vehicle engines have a wide variety of load ranges, even if the zone of operation of CAI is expanded, the employment of AI-SI hybrid combustion may be considered as a practical solution^{(7), (8)}.

This paper will propose a method for the realization of a smooth transition between AI and SI from the perspectives of in-cylinder gas exchange and combustion control. In order to do so, the paper will first attempt to clarify all the possible gas exchange characteristics of a naturally

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aspirated 4-stroke engine. The paper will then propose a control method making use of the benefits of a sequence of LTC states achieved via continuous control of gas exchange, and will further verify this method.

2. Introduction of ΔT Concept

ΔT has been introduced as a parameter that expresses heat generation per unit of heat capacity of the working gas. This functions as a parameter for the evaluation of the speed of heat release. Assuming isochoric combustion with no heat loss, dissociation, or non-uniformity of the mixture, from the first law of thermodynamics, heat capacity Q is given by

$$Q = \Delta T (C_{vf} \cdot G_f + C_{vr} \cdot G_r) \quad (1)$$

where C_{vf} is the constant volume specific heat of the mixture, C_{vr} is the constant volume specific heat of the residual gases, G_f is the mass of the mixture, and G_r is the mass of the residual gases.

When V =displacement, η_c =charging efficiency, ρ =density of air, ϕ =equivalence ratio, α =stoichiometric A/F ratio, ξ =EGR rate, and H_u =lower heating value of fuel,

Mass of intake air is: $G_a = V \cdot \eta_c \cdot \rho$

Mass of supplied fuel is: $M_f = G_a \cdot \phi / \alpha$

Mass of mixture is: $G_f = G_a + M_f = G_a (1 + \phi / \alpha)$

Mass of residual gas is: $G_r = G_f \{ \xi / (1 - \xi) \}$
 $= G_a (1 + \phi / \alpha) \{ \xi / (1 - \xi) \}$

Calorific value of fuel is: $Q = H_u \cdot M_f = H_u \cdot G_a \cdot \phi / \alpha$

Dividing Eq. (1) by G_a gives

$$\Delta T = H_u (\phi / \alpha) / [C_{vf} \cdot \{1 + (\phi / \alpha)\} + C_{vr} \cdot \{1 + (\phi / \alpha)\} \{ \xi / (1 - \xi) \}] \quad (2)$$

Here, ΔT is given as an equation between ϕ and ξ , irrespective of displacement or the load conditions of the engine. In physical terms, ΔT refers to an increase in the temperature of the in-cylinder working gas following auto-ignition.

3. Test Apparatus and Test Method

A 4-stroke single-cylinder water-cooled gasoline engine with a bore of 100 mm by a stroke of 82.6 mm and a displacement of 650 cm³ was employed as the test engine. Table 1 shows the specifications of the test engine. An electrohydraulic valve actuator system (Lotus AVT)⁽⁹⁾ was employed as a variable valve mechanism, and the EGR rate was varied with negative overlap⁽¹⁰⁾. A compression ratio of 10 was used. Port injection fuel supply was employed to form a uniform pre-mixture in the cylinder. Regular gasoline (RON91) was used as the test fuel.

Figure 1 shows an outline of the test apparatus. Because high-temperature residual gases are essential to auto-ignition operation, the transition from SI combustion via the spark plug to AI combustion was induced by gradually increasing

Table 1 Specifications of experimental engine

Engine type	4-stroke single-cylinder liquid-cooled
Bore x Stroke (mm)	100 x 82.6
Displacement (cm ³)	648.7
Compression ratio	10.0
Valve diameter (mm x number)	Intake : $\phi 36 \times 2$ Exhaust : $\phi 31 \times 2$
Valve drive	Hydraulic-electric (Lotus AVT)
Valve timing (1mm lift)	Exhaust opening : 35 deg BBDC
	Exhaust closure : -120 ~ 0 deg ATDC
	Intake opening : -120 ~ 0 deg ATDC
	Intake closure : 35 deg BBDC
Valve lift (mm)	6
Fuel system	Port injection
Injection pressure	350 kPa (gauge)
Fuel	Gasoline (RON 91)

the EGR rate. The EGR rate was controlled within a range from 10 to 80% by varying the degree of negative overlap. Symmetrical negative overlap (SNO), in which the exhaust valve closing timing and the inlet valve opening timing are symmetrical, was employed to simplify phenomena in the test. Figure 2 shows an example of the valve lift curve. Pressure sensors and thermocouples were positioned in the exhaust and inlet pipes to help enable study of the gas exchange process. In-cylinder pressure was measured using a piezoelectric pressure sensor fitted to the cylinder head. In order to help enable correction of in-cylinder pressure, the absolute in-cylinder pressure close to bottom dead center was measured using a semiconductor pressure sensor fitted to the side-wall of the cylinder. Combustion phenomena in the transition from SI to AI combustion were analyzed by means of analysis of pressure and exhaust gases.

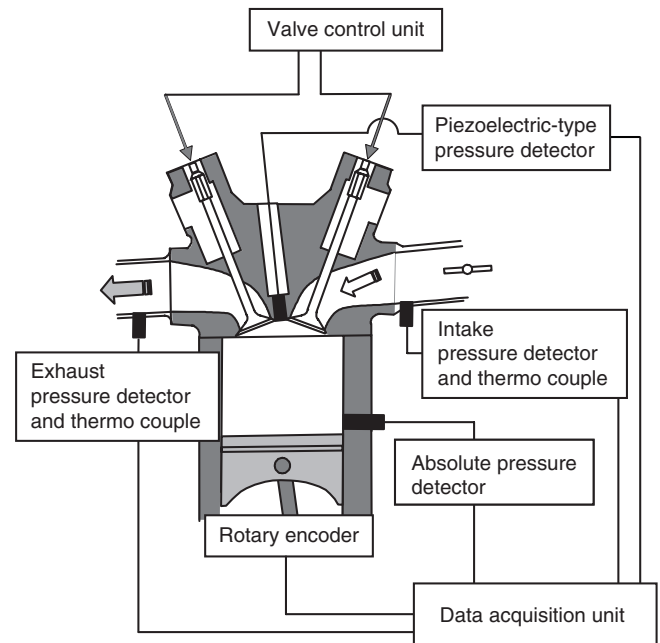


Fig. 1 Layout of detectors

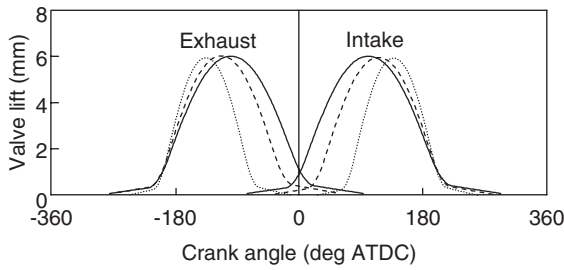
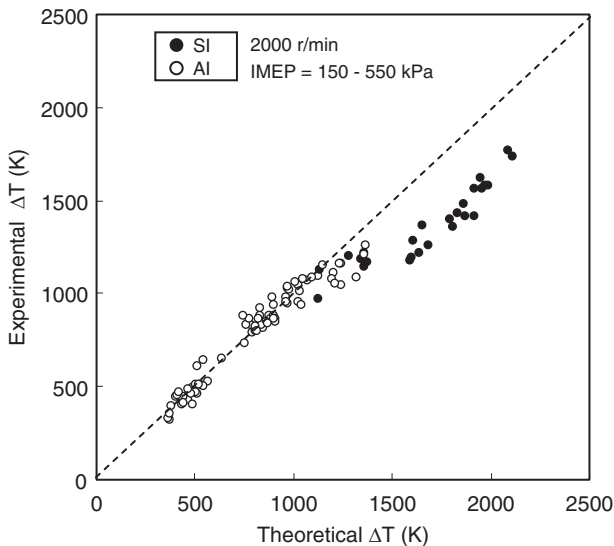


Fig. 2 Variations in valve events

4. Application of ΔT Concept to AI Combustion Control

As indicated in Chapter 2, the physical meaning of ΔT is an increase in the temperature of the in-cylinder working gas following ignition. Given this, it is possible to determine the difference between the maximum combustion temperature and the temperature at ignition timing in tests. Figure 3 shows the correlation between theoretical ΔT , as calculated using Eq. (2) and ΔT as obtained from tests. ΔT was determined in tests from an indicator diagram analysis when the indicated mean effective pressure (IMEP), equivalence ratio ϕ , and EGR rate ξ were varied at an engine speed of 2000 rpm. Theoretical ΔT was unambiguously defined based on ϕ and ξ alone, and was calculated by using figures for ϕ and ξ corresponding to test figures in Eq. (2). The solid circles in the figure show results for SI combustion, while the clear circles show those for AI combustion.

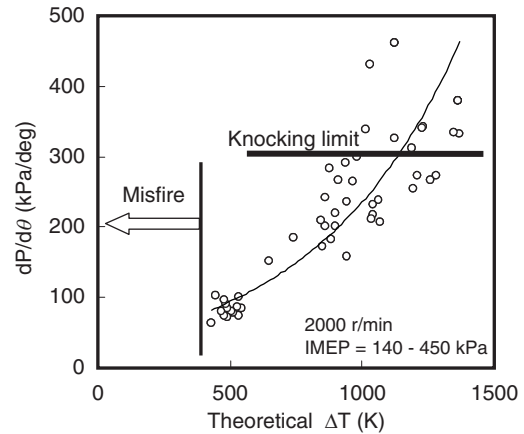
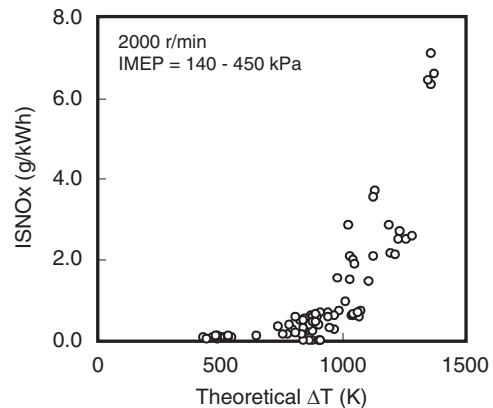
When ΔT is high, figures for test ΔT fall below figures for theoretical ΔT . This is believed to be due to the fact that the heat capacity of the in-cylinder working gas declines in proportion to the magnitude of ΔT , and the influences of dissociation and heat loss become conspicuous as the

Fig. 3 Correlation between experimental and theoretical ΔT

maximum combustion temperature increases. By contrast, when ΔT is low, there is a good correlation between theoretical ΔT and test ΔT . The correlation is particularly good in the AI combustion zone. This indicates that ΔT is effective as an index for the evaluation of heat release following ignition in AI combustion.

Figure 4 shows the relationship between theoretical ΔT and the pressure rise rate, $dP/d\theta$, during AI combustion at IMEP 140-450 kPa. When the heat capacity of the working gas declines as ΔT increases, combustion becomes rapid and $dP/d\theta$ increases, resulting in an increase in engine noise, which eventually becomes knocking. If 300 kPa/deg is defined as the knocking limit, ΔT must be 1000 K or less. Meanwhile, when ΔT is 500 K or less, it is not possible to maintain combustion due to misfires. This is believed to be because the rate of heat release slows due to an excessive increase in the heat capacity of the working gas. It would therefore be possible to control proper auto-ignition without misfires or knocking by maintaining ΔT within the range of 500-1000 K.

Figure 5 shows indicated specific NO_x emissions (ISNO_x) against theoretical ΔT during auto-ignition combustion. ISNO_x remains at a low level when ΔT is less than 1000 K, but increases rapidly when ΔT increases above 1000 K. The AI temperature, T_{AI} , is approximately 1000

Fig. 4 Effect of ΔT on pressure rise rateFig. 5 Effect of ΔT on NO_x emissions

K. And $T_{AI} + \Delta T$ is the maximum combustion temperature, T_{max} . Because T_{AI} is fixed regardless of load or engine speed⁽²⁾, ΔT corresponds to the maximum combustion temperature. The increase in ISNO_x is believed to be due to the fact that the maximum combustion temperature exceeds the thermal NO_x production temperature (approximately 1800 K) when the heat capacity of the working gas declines and ΔT increases. It is therefore necessary to employ low-temperature combustion ($\Delta T < 1000$ K) in order to control NO_x emissions to low levels during AI combustion.

5. An Approach from the Perspective of Gas Exchange

Figure 6 shows the relationship between IMEP and the in-cylinder gas exchange rate (gas charging efficiency, η_g). η_g is defined as the ratio of the mass of total gas (fresh gas and EGR gas) to the mass of fresh gas that fills the stroke volume at the reference conditions. IMEP in the figure ranges from the idle to the wide open throttle (WOT).

The solid line in the figure shows the ratio of fresh gas in the cylinder (η_c) when the equivalence ratio, ϕ , is 1. η_c is proportional to IMEP. The η_c line corresponds to zero EGR, and the IMEP in a conventional 4-cylinder engine is adjusted along this line. The curve in the upper section of the graph is the upper limit of η_g during natural aspiration, i.e., the EGR limit curve. The EGR limit curve is determined by the in-cylinder working gas temperature. Because the mass of fresh gas declines as the IMEP is reduced, the rate of EGR that can be supplied increases in proportion with the decline in IMEP. Because the in-cylinder gas temperature increases with the amount of EGR, the EGR limit curve tends to become lower as the IMEP is reduced. However, because the EGR gas temperature declines as the EGR rate increases, the EGR limit curve ceases to decline close to an IMEP of 300 kPa. The broken lines in the figure show equal EGR rates. Similar to the η_c line, the equal EGR rate lines are straight lines extending from a common point of origin. The gas exchange states between fresh gas and EGR gas that can be realized are

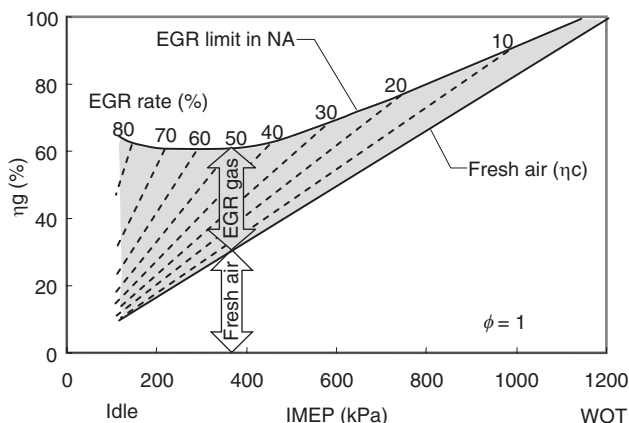


Fig. 6 Gas exchange chart

positioned between the η_c line and the EGR limit curve.

This gas exchange chart represents all the possible gas exchange states in a naturally aspirated 4-cylinder engine. Based on this chart, this paper will consider the in-cylinder gas exchange and the combustion control.

As indicated in the previous chapter, in order to realize desirable CAI, the following conditions for T_{AI} and ΔT must be satisfied: (1) $T_{AI} > 1000$ K; (2) $500 \text{ K} < \Delta T < 1000$ K. Because T_{AI} and ΔT are functions of the EGR rate and the equivalence ratio, conditions (1) and (2) can be expressed on the gas exchange chart. Figure 7 shows the CAI zone when the ΔT concept is applied to the gas exchange chart for $\phi=1$. This indicates that the upper limit of IMEP for CAI is 450 kPa, and the EGR rate is approximately 40%.

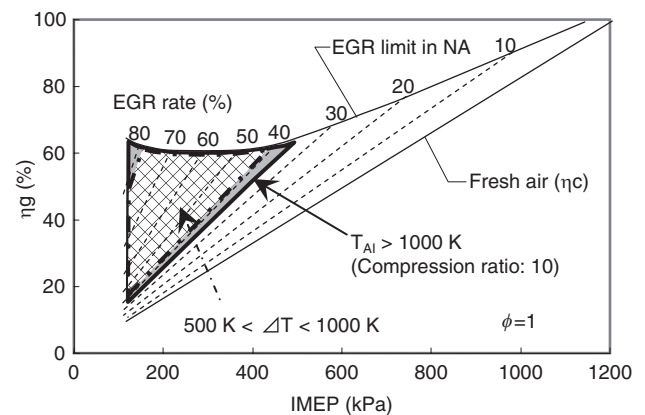


Fig. 7 Conditions for CAI obtained by applying ΔT concept to gas exchange chart

6. Positioning of Combustion Modes on Gas Exchange Chart

Figure 8 shows a plot, by combustion mode, on the gas exchange chart of the results of a firing test in which the EGR rate was varied by varying the SNO. The test was conducted at an engine speed of 2000 rpm

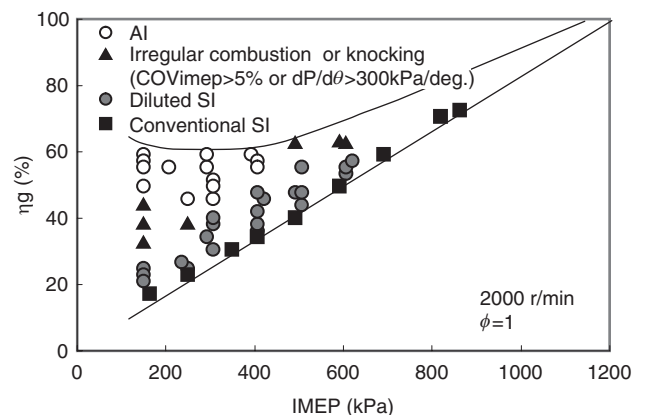


Fig. 8 Results of plotting combustion on chart

and a ϕ of 1. Clear circles show AI, solid circles show diluted SI, squares show conventional SI combustion, and triangles show irregular combustion. AI and diluted SI were distinguished by whether ignition was ON or OFF. When ignition was OFF and combustion continued, AI combustion was assumed, but when combustion could not continue and misfires occurred, diluted SI was assumed to be the combustion mode. Results for conventional SI combustion were drawn from test results at a valve timing of SNO=0 deg. Irregular combustion was defined as a combustion mode in which COVimep (Coefficient of Variation of indicated mean effective pressure) is higher than 5% or $dP/d\theta$ is higher than 300 kPa/deg. AI combustion is positioned in the low-IMEP, high- η_g zone. The possible gas exchange states in a vehicle engine should be controlled within a zone excluding the zone of irregular combustion.

7. Indicator diagrams of LTC Cycle

Figure 9 compares P-V diagrams and in-cylinder average gas temperatures for SI, diluted SI, and AI combustion at an engine speed of 2000 rpm, a ϕ of 1, and an IMEP of 300 kPa. The conventional SI results were obtained from a test conducted at a valve timing of SNO (amount of overlap) =0 deg; the EGR rate at this condition was 10%. The AI results were obtained at a valve timing of SNO=-90 deg. The EGR rate at this condition was 51%. The following characteristics are shown: (1) Rapid combustion; (2) High-pressure

combustion; (3) Low pumping loss; and (4) Low-temperature combustion (LTC). The maximum combustion temperature declines probably because of the increase in the heat capacity of the in-cylinder working gas in accordance with the increase in the EGR rate. The fact that η_g becomes high indicates that the level of LTC becomes high. The reduction in combustion temperature achieved by using a gas exchange state that promotes a high level of LTC makes it possible to reduce NOx emissions, increase thermal efficiency by reducing heat loss, and reduce pumping loss. These are the benefits of LTC.

8. Control Method using LTC

The choice of possible control of IMEP corresponds to achieving a gas exchange state in which the undesired combustion zone (irregular combustion and knocking) is avoided. Figure 10 shows three examples of IMEP control methods. The broad line shows control in conventional SI combustion. The narrow line shows a control method that passes through the diluted SI zone across the IMEP range. This can be realized at a fixed valve timing of SNO=-47 deg, but maximum IMEP is reduced by 10-20% due to the fact that EGR is trapped at a constant rate. In addition, the effect in reducing fuel consumption and NOx emissions is minimal because the level of LTC is low. The broken line shows the optimum LTC control proposed in this paper. This control method provides continuous control, offering the fuel

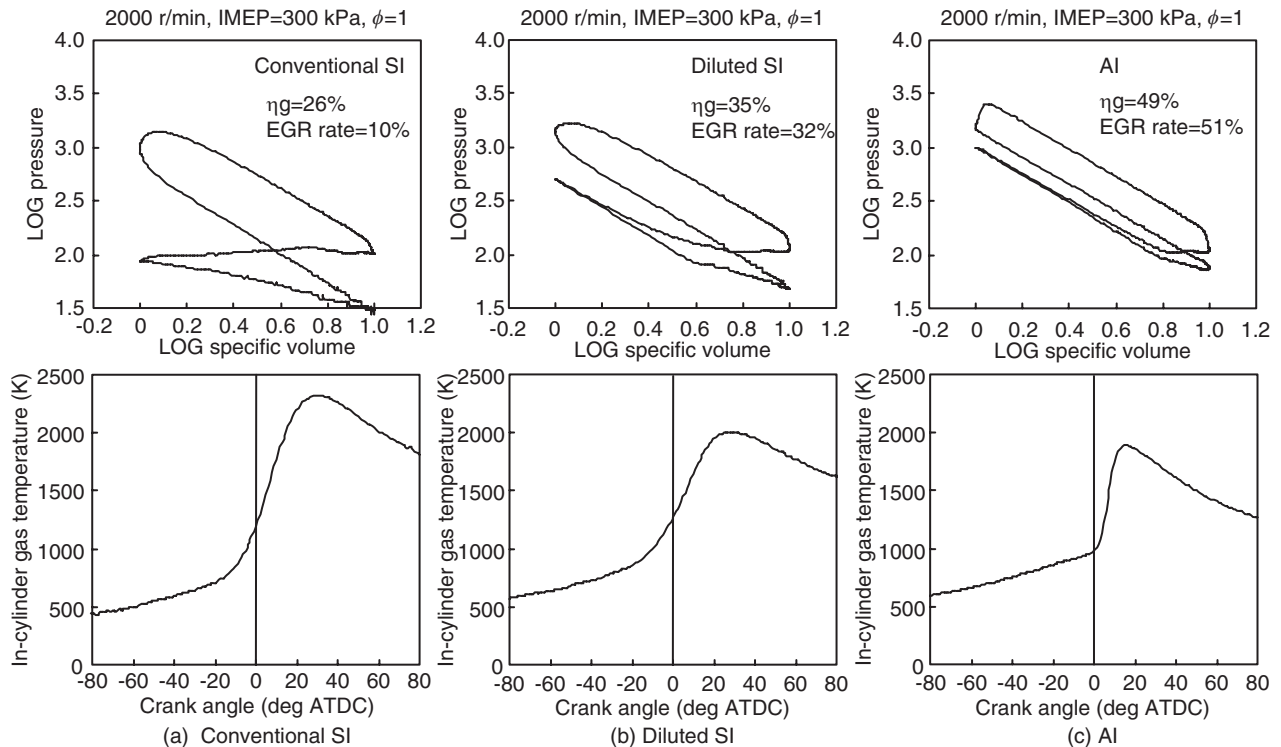


Fig. 9 Indicator diagrams for LTC cycles

consumption and emission benefits of a sequence of LTC cycles from SI combustion at the WOT to AI combustion at the idle (SI → Diluted SI → AI) throughout the IMEP range, without applying combustion mode switching. This optimum LTC control can be realized by creating the suitable gas exchange state for a certain IMEP, and does not absolutely necessitate the use of a high-speed variable valve mechanism.

Figure 11 shows SNO and degree of throttle opening against BMEP (Brake Mean Effective Pressure). At an engine speed of 2000 rpm, the optimum LTC control can be maintained by varying the valve timing so as to make SNO ranging across 0 to -90 deg. At this time CAI occurs at BMEP 300 kPa and below. The benefits of LTC, including in the diluted SI zone, can be maximized through the employment of smooth and continuous gas exchange control from maximum load (SI) to idle (AI).

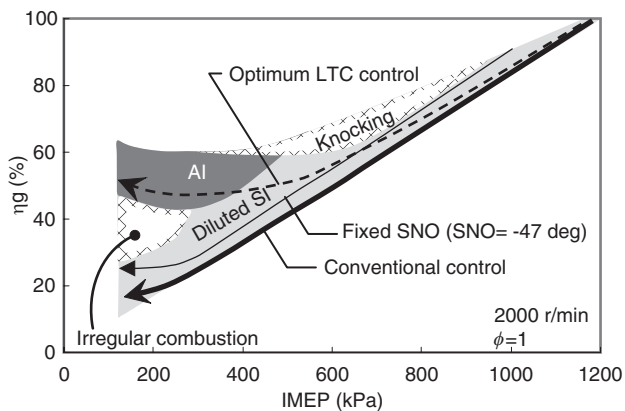


Fig. 10 Three examples of possible IMEP control

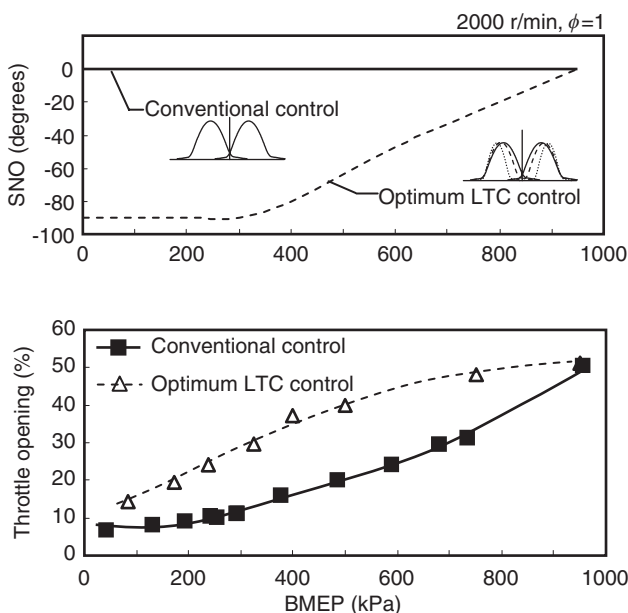


Fig. 11 SNO and throttle opening against BMEP

9. Benefits of Optimum LTC Control

Figure 12 compares BSFC (Brake Specific Fuel Consumption) and BSNO_x (Brake Specific NO_x) characteristics against BMEP for conventional SI and optimum LTC control. The bar graph at the top of the figure shows the distribution of BMEP frequency in LA-4 mode for a 1300 cm³ V-twin motorcycle. If the fuel consumption and NO_x emissions of the V-twin motorcycle are estimated from this data, reductions of approximately 20% in fuel consumption and approximately 90% in NO_x emissions can be expected in LA-4 mode.

Figure 13 shows ISFC (Indicated Specific Fuel Consumption) and PMEP (Pumping Mean Effective Pressure) for conventional SI and optimum LTC control. The use of optimum LTC control reduces ISFC in the low

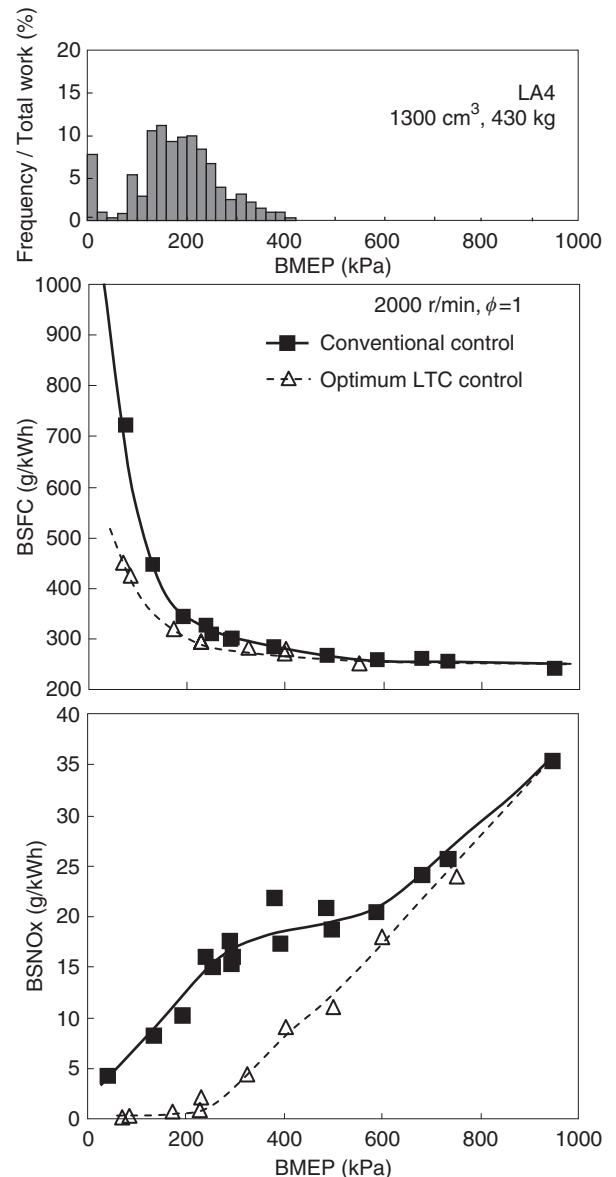


Fig. 12 BSFC and BSNO_x characteristics against BMEP

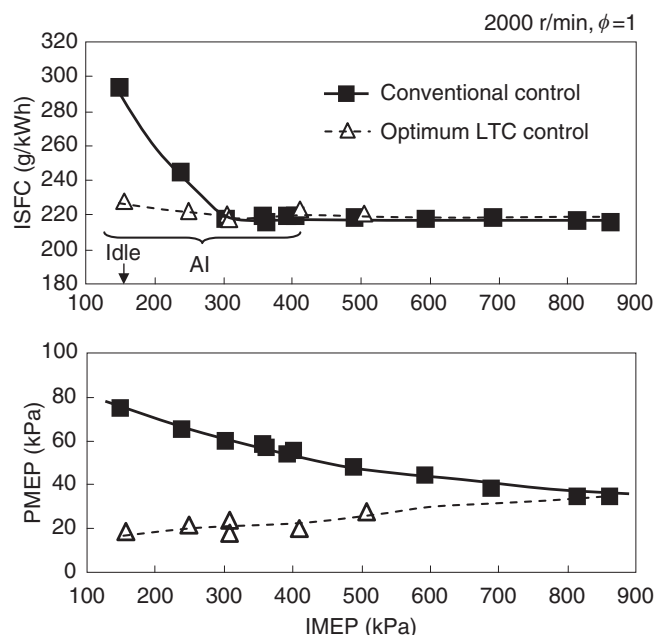


Fig. 13 ISFC and PMEP characteristics against IMEP

zone at IMEP 300 kPa and below. The combustion mode in this zone is AI. The control also reduces PMEP across the BMEP range.

10. Conclusion

A macroscopic understanding of physical parameters for the control of CAI ignition timing and heat release has produced the following experimental findings:

- (1) The auto-ignition temperature, T_{AI} , and ΔT are parameters for the realization of desirable CAI. T_{AI} expresses ignition timing, and ΔT expresses heat release following ignition.
- (2) Maintaining ΔT within the range from 500-1000 K makes it possible to control proper AI without misfires or knocking.

The adoption of an approach from the perspective of gas exchange in relation to the application of AI combustion to vehicle engines produced the following findings:

- (3) A gas exchange chart that shows all the possible gas exchange states in a 4-stroke naturally aspirated engine was proposed.
- (4) Combustion maps were formulated by plotting combustion modes including AI combustion, diluted SI combustion, and irregular combustion on this gas exchange chart.
- (5) A control method (optimum LTC control) was proposed that “continuously” selects the appropriate LTC state for each load on the combustion maps discussed in (4). This control method makes it possible to smoothly employ combustion modes from SI combustion to AI combustion without applying switching, and to maximize the fuel consumption and emission benefits of a sequence of LTC

states, including in the diluted SI zone.

- (6) It is estimated that the application of the optimum LTC control to a 1300 cm³ V-twin motorcycle would reduce fuel consumption by approximately 20% and NOx emissions by approximately 90% in LA-4 mode compared to conventional combustion methods.

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■ Author ■



Hideaki MORIKAWA

Study on Similarity of Load Path Distribution in Body Structure

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ABSTRACT

Consideration and verification of the way in which load paths will be distributed throughout the structures under various loading conditions is an important element in the design of automotive body structures. Furthermore, given the tendency that load-path distribution becomes denser where coupling rigidity is high irrespective of the loading conditions, it is conjectured that similarities exist in the load-path distributions even under differing loading conditions.

The research discussed in this paper therefore conducted a comparative analysis of two methods for identifying load paths: using U_{sum}^* , which functions as an indicator of the distribution of load paths in body structures, and using stress. The difference between the two methods was clarified, and the distribution of load paths when four different loading conditions were applied to automotive body structures was studied. As a result, it was demonstrated that using U_{sum}^* , load-path distributions can be visually analyzed employing less indicators than was the case when using stress. In addition, the level of similarity of load-path distribution to distribution in other loading conditions was high for torsional loading conditions in which load paths are distributed throughout the structure as a whole.

1. Introduction

Today, given demands for the reduction of the burden on the environment and the realization of greater dynamic performance for increased commercial appeal, the achievement of maximal weight savings is necessitated in addition to the satisfaction of the performance demands made on automotive body structures. When designing high-weight-efficiency automotive bodies, it is important to give consideration to how the loads on the body structures will be transmitted through the structures and the projected performance levels satisfied. Direct identification of where the load paths through which the loads are transferred are located in the body structures is challenging, and at present their location is indirectly estimated using stress or strain⁽¹⁾,⁽²⁾. Because of the existence of areas in which high stresses are generated even when they are not actually functioning as load paths, for example points at which stress concentrates such as areas subject to sudden changes and holes, the identification of load paths according to the magnitude of stress represents a challenge. Furthermore, it is necessary to make a comprehensive judgment on numerous indicators to estimate load paths, including principal stresses, deformation diagrams, and sectional forces.

A method of analysis of load paths within body structures using FEM was therefore proposed. Load paths

can be identified by expressing the strength of connection between load points, constraint points, and arbitrary points on body structures by means of the indicators U^* and $U_{\text{sum}}^{*(3),(4)}$. The use of these indicators offers the advantage of making the load-path distribution easy to understand visually, because there is no rapid change in values, as for example when stress concentrates at holes and points where the structure undergoes rapid change.

When various loads are applied to body structures, load paths are generated within those body structures in response to the loading conditions. At the same time, the majority of load paths tend to be distributed in areas with good coupling to surrounding areas and a high level of rigidity, while there tend to be few load paths in weak areas, irrespective of the loading conditions. In other words, some kind of similarities can be considered to exist in load-path distributions.

It can be indicated that in regions within the body structures in which the load-path distribution displays high levels of similarity, the parts that contribute to load transfer or the parts that do not contribute to load transfer are the same, irrespective of the loading conditions. As a result, if modifications were made to regions that did not contribute to load transfer under one specific loading condition to produce an effectively functioning structure, the structure would also be effective in relation to other loading conditions.

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Because conventional structural analysis involves analysis under each relevant loading condition and the modification of structures or the application of other measures under the respective loading conditions, it can be the case that separate areas of the body structures are subject to modifications. However, the ability to analyze the similarity of load-path distribution under each of the loading conditions in order to identify loading conditions and body areas displaying a high degree of similarity would help to enable the achievement of weight savings by means of the conceptualization of body structures that are effective in the case of multiple loading conditions through the application of appropriate measures in areas displaying high degrees of similarity. This method could be used across a wide range of the body development process, from analyses conducted to obtain a broad grasp of the body structures (for example, the rough framework configuration and load-path distribution within the body structures at the stage of advance review) to detailed analyses of the couplings between components (such as analyses of spot welding points and joins between panels at the detailed design stage).

The purpose of the research discussed in this paper was to demonstrate the validity of the load-path distribution analysis method using U^*_{sum} and to attempt to determine loading conditions for which the load-path distributions displayed a high degree of similarity by using the indicator U^*_{sum} in the analysis of load-path distributions when various loading conditions were applied to automotive body structures.

2. The Indicators U^* and U^*_{sum}

2.1. The Indicator U^* and Load Paths

This section will explain the definition of the indicator $U^{*(3)}$ used in the analysis of load paths within body structures employing finite element analyses. Figure 1(a) shows a schematic diagram of a body structure. Point A is a load point, Point B is a support point, and Point C is an arbitrary point within the structure. The strain energy when an imposed displacement d_A is applied to the structure, U , is expressed by Eq. (1). Here, p_A is the load necessary to displace Point A by d_A .

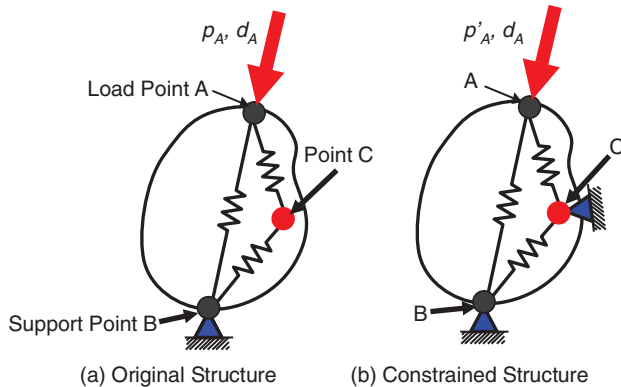


Fig. 1 Strain energy models

$$U = \frac{1}{2} p_A \cdot d_A \quad (1)$$

Figure 1(b) shows the same structure as Fig. 1(a). The arbitrary Point C is constrained, and the load necessary to displace Point A by d_A is p'_A . The strain energy in this case, U' , can therefore be expressed by Eq. (2).

$$U' = \frac{1}{2} p'_A \cdot d_A \quad (2)$$

The indicator U^* at Point C is defined by Eq. (3).

$$U^* \equiv 1 - \frac{U}{U'} \quad (3)$$

Because the value of the strain energy U' is always greater than that of U , U^* takes a value between 0 and 1, and is 1 at the load point and 0 at the support point.

Figure 2 shows a schematic diagram of the distribution of U^* in the structure. The greater the coupling rigidity with the load point within the structure, the greater the value of U^* . In other words, the ridge line of the U^* distribution can be indicated as an area of high coupling rigidity extending from the load point. It is believed that the force acting on the structure is transmitted mainly through the areas of highest rigidity to the coupling point. The main load path therefore follows the ridge line of the U^* distribution and arrives at the support point.

2.2. Load Path Distribution and the Indicator U^*_{sum}

There are multiple load paths other than the main load path. The indicator $U^{*(4)}_{\text{sum}}$, which makes it possible to quantitatively express the status of distribution of the load paths, is therefore introduced.

Figure 3(a) shows a conceptual diagram of U^* when Point A is the load point and Point B is the support point, representing U^*_1 , the value of U^* at each node. Figure 3(b) shows a conceptual diagram of U^* when the load point and the support point are transposed, representing U^*_2 , the value of U^* at each node. The indicator U^*_{sum} at each node in these cases is defined by Eq. (4).

$$U^*_{\text{sum}} = U^*_1 + U^*_2 \quad (4)$$

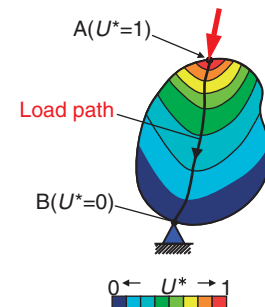


Fig. 2 Image of load path

The conceptual diagram of U_{sum}^* distribution for the whole structure is shown in Fig. 3(c).

Because the load is distributed evenly throughout the structure and the load path distribution is uniform in the one-dimensional spring model shown in Fig. 4, the value of U_{sum}^* is constant at 1.

Figure 5(a) shows an example of the calculation of U^* for a flat plate in which a hole has been opened. Support points are located on one side of the flat plate, and an imposed displacement is applied in the tensile direction on the other side of the plate. Figures 5(b) and 5(c) respectively show the stress distribution and the distribution of U_{sum}^* under the same conditions. Ridge lines which skirt the hole exist in the U^* distribution shown in Fig. 5(a), and it is possible to draw the main load paths from the load points on these ridge lines. By contrast, in the stress distribution shown in Fig. 5(b), stress concentrates at the load points and in the vicinity of the hole, and it is challenging to identify the load paths.

The load paths that extend from the load points to the support points can serve as main load paths, since greater loads can be transferred through them. However, because U^* shows the areas of highest coupling rigidity with the

load points, the magnitude of U^* is the amount evaluated in one direction from the load points, irrespective of whether or not the load paths extend to the support points. Because of this, when forms are complex the load paths become long and complex, and this makes it challenging to judge whether a load path shown by the distribution of U^* extends to a support point. In addition, because the areas identified within the structure are not the only load paths, but load paths are broadly distributed throughout the structure as a whole, an analysis using U^* that is restricted to several main load paths cannot be judged sufficient.

The distribution of U_{sum}^* shown in Fig. 5(c) shows the

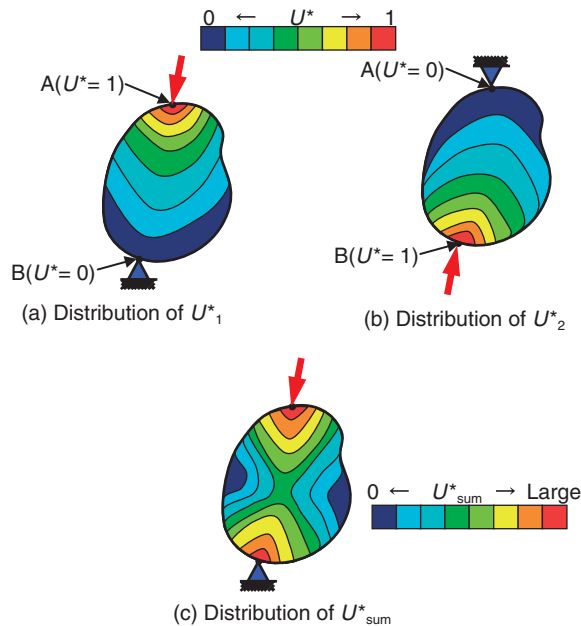


Fig. 3 Conceptual diagram of U_{sum}^*

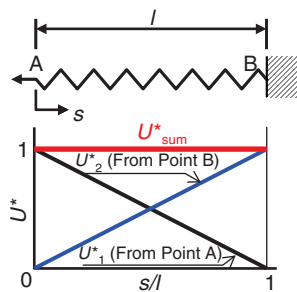


Fig. 4 Diagram for U_{sum}^*

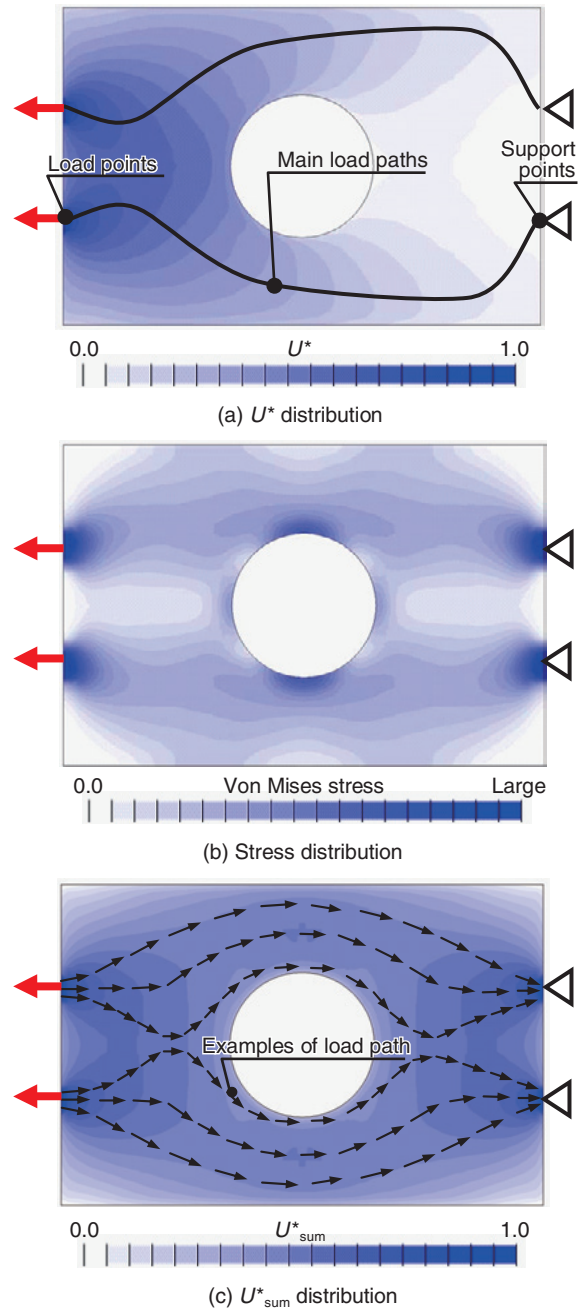


Fig. 5 U^* , stress and U_{sum}^* distribution in plate with hole

strength of coupling from the direction of both the load points and the support points. Because the coupling rigidity with both the load points and the support points is high in areas in which the value of U_{sum}^* is high, there are numerous load paths extending from the load points to the support points. On the contrary, load paths distribute sparsely in areas in which the value of U_{sum}^* is low. Figure 5(c) shows an example of load paths extending from the load points to the support points. Naturally, numerous other load paths could also be represented in this diagram. The distribution of the load paths becomes denser in areas in which the value of U_{sum}^* is high, such as close to the load points and the support points. In this way, the status of distribution of functioning load paths can be expressed visually based on the magnitude of U_{sum}^* .

3. Comparison of Stress Values and U_{sum}^* as Methods of Analysis of Load Path Distribution

3.1. Calculation Model

Figure 6 shows the loading conditions used in the analysis. Four representative loading conditions input from the suspension to the body were used: torsional loading, lateral loading, longitudinal loading, and bending loading. The front damper bases and rear damper bases were basically treated as the load points and the support points respectively, but because a load input using moment cannot be employed in the calculation of U^* in the case of bending loading, the side sill, which possesses high rigidity and displays a low level of local deformation, was treated as the

support point, and bending was reproduced by raising all the damper bases.

3.2. Comparison of Calculation Results

The distribution of stress and the distribution of U_{sum}^* were compared in order to study methods of analysis of the load-path distribution when a load is applied to an automotive body structure. Torsional loading, as mentioned in the previous section, was used as the loading condition.

Firstly, the method of analyzing load paths using stress is discussed. In this method, the flow of forces is estimated by means of distortion diagrams and the direction and magnitude of the principal stress. In the detailed diagram of the area around Part B shown in Fig. 7(b), high

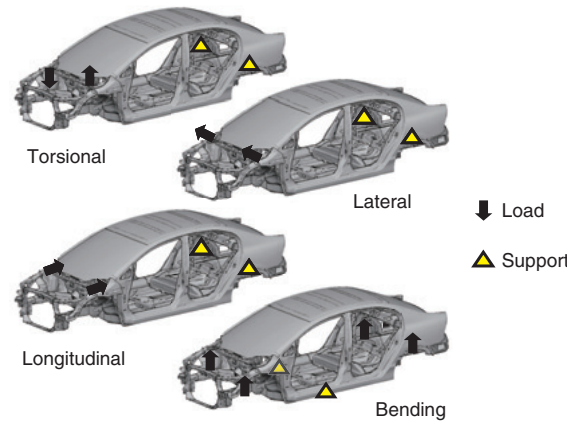


Fig. 6 Load and support conditions

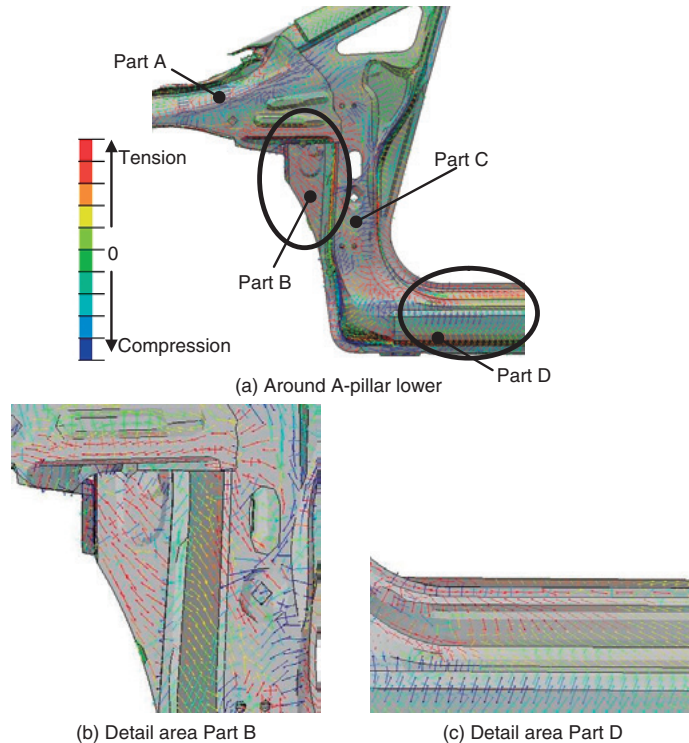


Fig. 7 Principal stress

principal stresses are acting in the tensile direction in the area extending from Part B to Part C, and their directions coincide with each other. It can therefore be conjectured that the load paths extend from Part A toward Part D, via Part B.

It should be noted that a plate has considerably lower bending stiffness in an out-of-plane direction compared to in a shearing direction. As a result, the section of the plate that is subject to out-of-plane bending load can only carry a lower load and thus does not become a main load path. In addition, there is an issue in that whether out-of-plane loads are acting or not is not easily determined by the analysis using principal stress. In order to identify the direction of load acting on the plane, it is necessary to judge whether out-of-plane deformation is occurring using the deformation

diagram or through a comparison of the direction and magnitude of the principal stress between both sides of the plane. These measures are applied to the discontinuous regions based on the understanding of plate thickness and direction of the in-plane stress. Principal stress is high in the detailed diagram of the upper surface of Part D shown in Fig. 7(c), and a strong load appears to be acting on the part. However, the deformation diagram for the cross-section of Part D shown in Fig. 9(a) shows upward displacement of the upper surface of Part D, and can be considered to indicate bending deformation in an out-of-plane direction. Figure 9(b) shows a contour diagram showing the difference between the principal stress on the surface and the reverse surface of the shell elements, formulated in order to verify this bending deformation. Where the surface

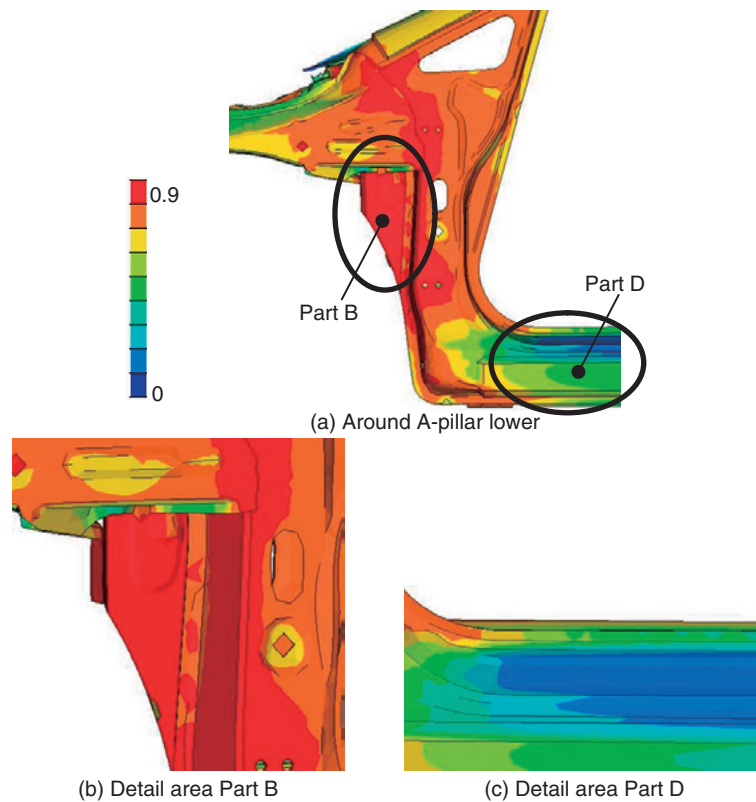


Fig. 8 U_{sum}^* distribution

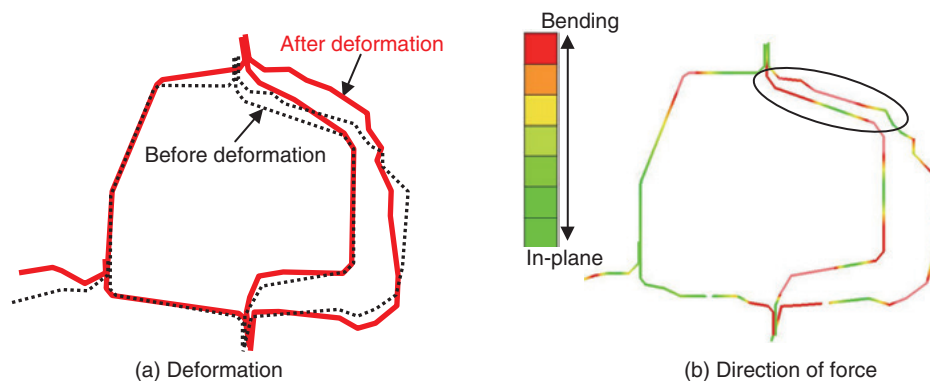


Fig. 9 Section of Part D

is compressed and the reverse surface is stretched due to an out-of-plane bending force, the difference in the principal stresses is high, and this is shown as red in the diagram. The upper surface of Part D (circled in the diagram) shows as red, indicating that a bending force is acting in an out-of-plane direction. Therefore, the upper surface of Part D does not act as a load path.

As the discussion above indicates, when the principal stress is used in an analysis, areas of high stress do not always become load paths, and it is necessary to analyze the nature of the forces by comparing multiple indicators such as deformation diagrams, in addition to judging whether there is sufficient thickness to generate load paths. Therefore, it takes practice and many labor hours to analyze load path distribution using this method and different analysis results may be derived from the same analysis object.

The use of U_{sum}^* , on the other hand, presents the advantage of helping to enable the load-path distribution in the structure to be understood at a glance due to the fact that denser load-path distribution exists at the part with higher U_{sum}^* value.

For example, the area around Part B shown in Fig. 8 (b) displays higher U_{sum}^* value, indicating that load paths distribute densely as in the case of the load path analysis using stress. In the distribution diagram of U_{sum}^* shown in Fig. 8(c), the upper surface of Part D displays a low U_{sum}^* value. The results illustrate the fact that the load path distribution is sparse as in the case of the analysis using stress.

As the discussion above indicates, similar results can be obtained by the load path distribution analysis using U_{sum}^* as those by the analysis using stress, and the analysis using U_{sum}^* can be considered to be valid. In addition, this

method also offers the advantage of making the load-path distribution easy to analyze because it is only required to judge based on the magnitude of U_{sum}^* values instead of the comprehensive judgment based on multiple information as conducted in the analysis using stress.

In addition, the use of stress also presents the issue of concentrations of stress. Figure 10(a) shows the diagram for principal stress tensor of the floor panel. Because the floor panel undergoes shearing deformation under torsional loading conditions, load paths are supposed to distribute throughout the entire floor. However, as the shape of the floor panel is complex due to elements such as beading or holes, concentrations of stress occur and the direction and strength of the principal stress vary with location. Because of this, it is not easy to analyze the distribution of load paths on the floor. By contrast, in the U_{sum}^* distribution diagram, the issue of stress concentration does not occur, and as shown in Fig. 10(b), numerous areas of the floor panel are shown as yellow, making it possible to visually grasp the fact that load paths are virtually uniformly distributed.

4. Similarity of Load-path Distribution in Automotive Structures

4.1. Similarity of Load-Path Distribution

The project went on to compare similarities in the distribution of U_{sum}^* when a variety of loads were applied to automotive body structures.

Figure 11 shows top views, Fig. 12 side views, and Fig. 13 bottom views for the distribution of U_{sum}^* under the loading conditions outlined in the previous chapter. Because the average value of U_{sum}^* differs for each loading condition, the values which are easiest to see for each loading condition have been set as the values for the contour lines.

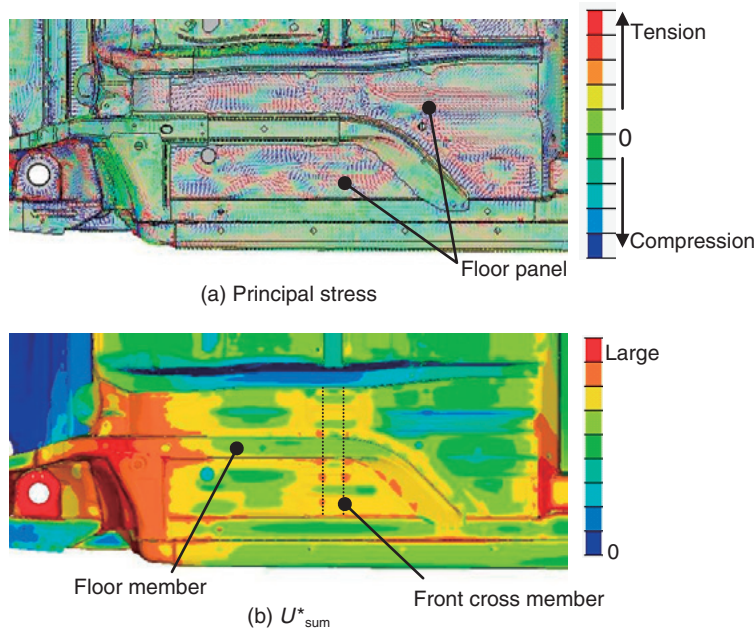


Fig. 10 Front floor

Comparing first the top views shown in Fig. 11, a similarity can be observed between the distribution of U_{sum}^* in the roof section for the torsional loading condition shown in Fig. 11(a) and the lateral loading condition shown in Fig. 11(b). The similarity appears between these two loading conditions because the roof is subjected to the same shearing forces under torsional and lateral loading conditions, while in the case of the other two loading conditions the load path distribution differs, because under longitudinal loading conditions (Fig. 11(c)) the forces act in the direction of bending and under bending loading conditions (Fig. 11(d)) the forces act in an out-of-plane direction.

Next, the side views shown in Fig. 12 will be considered. Under the bending loading condition shown in Fig. 12(d), U_{sum}^* becomes high in the side sill because there is a support point on this member. However, a comparison of the density of the total distribution in the side panel shows a similarity in the distribution of U_{sum}^*

between the torsional loading condition shown in Fig. 12(a) and the bending loading condition shown in Fig. 12(d). In addition, there is a similarity between the lateral loading condition shown in Fig. 12(b) and the longitudinal loading condition shown in Fig. 12(c), in that under both of these loading conditions the values of U_{sum}^* are lower for longitudinal members such as the front pillar lower and the center pillar than for transverse members such as the side sill. However, a detailed analysis of the side sill reinforce, as shown in Fig. 14 indicates that there are differences in the gradation of distribution between the side surface and the top surface for the lateral loading condition (Fig. 14(b)) and the longitudinal loading condition (Fig. 14(c)), and no similarity can be observed between the load-path distributions. The reason for this difference can be explained as follows: taking the direction of the loads acting on the side sill into consideration, under a lateral loading condition the side sill bends in the lateral direction of the vehicle and shearing forces act on the top and bottom

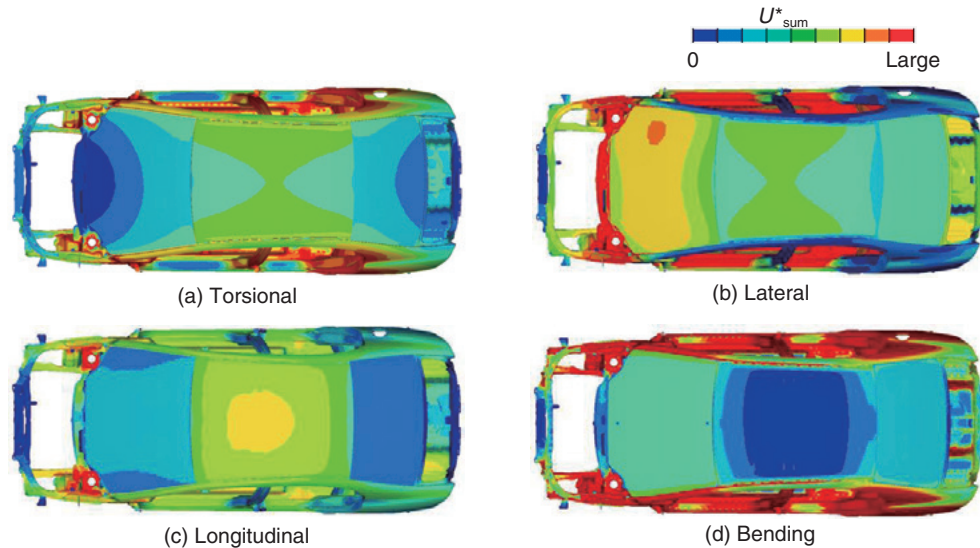


Fig. 11 Top view

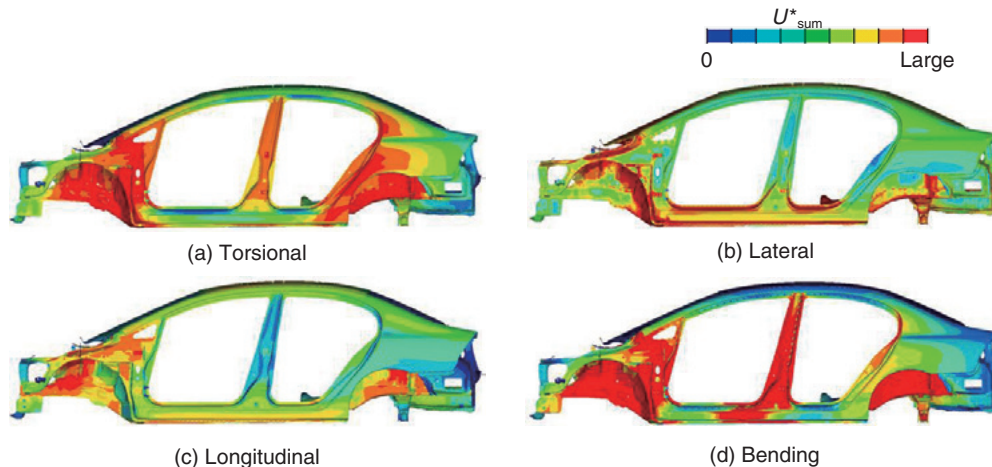


Fig. 12 Side view

surfaces of the side sill, thus generating load paths in these areas; under a longitudinal loading condition on the other hand, the load acts on the side sill in the axial direction, and there is no major difference between the value of U_{sum}^* between the side surface and the top surface of the side sill, with basically identical values displayed across the entire member.

Figure 15 shows the side-roof rail as another

perspective on the side views. Comparing these distributions shows a similarity in the distribution of U_{sum}^* between the torsional loading condition shown in Fig. 15(a) and the longitudinal loading condition shown in Fig. 15(c). Figure 16 shows a diagram of the direction of the loads acting on the side-roof rail at this time. Under a torsional loading condition, forces act on the side-roof rail in the bending and tensile directions, while under a longitudinal

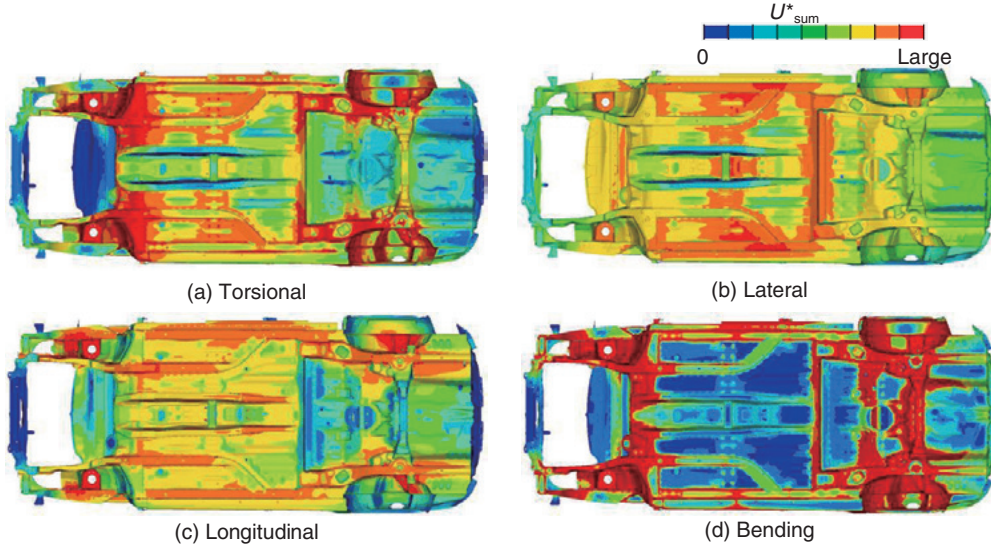


Fig. 13 Bottom view

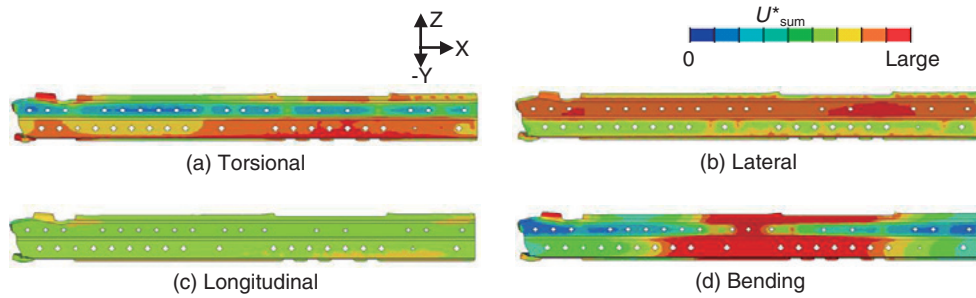


Fig. 14 Side sill reinforce

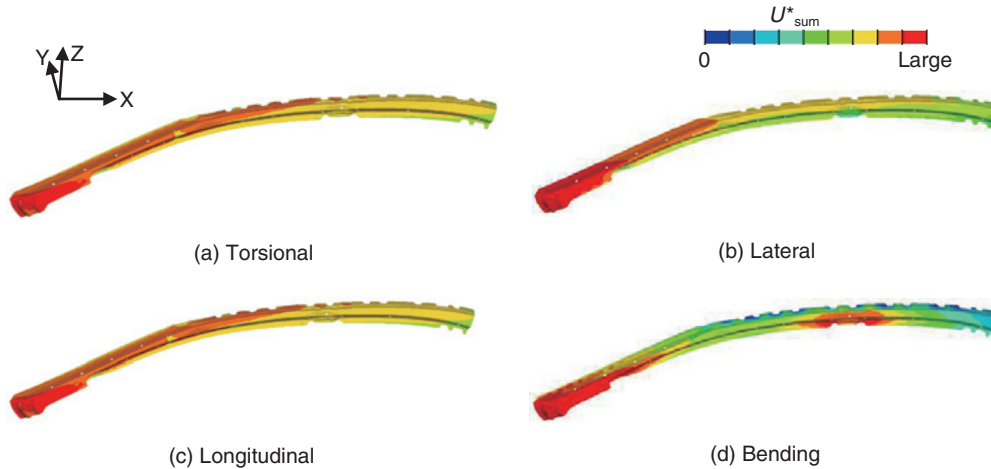


Fig. 15 Side roof rail

loading condition they act in the bending and compression directions. Despite the fact that while the bending direction is the same for both loading conditions, they differ in terms of the tensile and compression directions, because loads act in the axial direction in both cases, the dynamic conditions are identical and there is a similarity in the distribution of U_{sum}^* .

Finally, the bottom views shown in Fig. 13 will be considered. When the distribution of U_{sum}^* is compared for the four loading conditions, no conditions display a high level of similarity for the body as a whole. However, if the focus is restricted to the front floor panel, there is a comparatively high level of similarity between the torsional loading condition shown in Fig. 13(a) and the lateral loading condition shown in Fig. 13(b). A similarity appears between these two loading conditions because shearing forces act on the floor under both torsional and lateral loading conditions. Under a longitudinal loading condition in-plane compression forces and under a bending loading condition out-of-plane bending forces act on the floor, resulting in the difference in the U_{sum}^* distribution.

When the distributions of U_{sum}^* produced by the four loading conditions were compared, the distribution produced by the torsional loading condition was found to display similarities in more members than the distributions produced by the other loading conditions.

4.2. Discussion Regarding Similarities

This section will consider the reason that the torsional loading condition displays a high level of similarity with other loading conditions.

In terms of the direction in which forces act on each part when a torsional load is applied, forces act on transverse members such as the side sills in the bending direction and on longitudinal members such as the center pillars in an axial direction, and shearing loads are generated on each panel. Because of this, U_{sum}^* is high throughout the cabin under a torsional load.

By contrast, under lateral and longitudinal loading

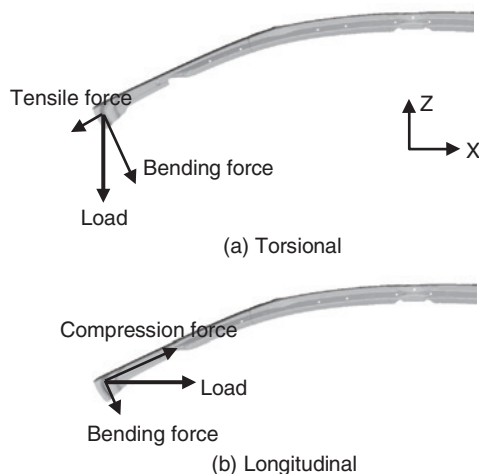


Fig. 16 Image of direction of load on side roof rail

conditions, while a shared point does exist in the fact that there is no principal load path and U_{sum}^* is low in the center pillars, the level of mutual similarity in other members is not particularly high. Taking the direction of the loads acting on the members into consideration, under lateral and longitudinal loading conditions, respectively, the direction of the in-plane forces acting on the floor panels and roof panels is transposed by 90° , and the direction of the axial forces therefore differs. As a result, members displaying high U_{sum}^* and low U_{sum}^* are transposed depending on the loading condition, as observed close to the mid-cross-members and the tunnel frame. Because of this, rather than mutual similarity, these loading conditions display a high level of similarity with the torsional loading condition, under which U_{sum}^* is high throughout the body.

The value of U_{sum}^* is low on the floor and the roof because forces act in out-of-plane directions under a bending loading condition, and load paths are not dispersed through the structures. Given this, the level of similarity between this loading condition and other conditions is low. However, this loading condition matches a torsional loading condition in the fact that principal load paths exist in longitudinal members, and for this reason its level of similarity is high only with a torsional loading condition.

5. Conclusion

- (1) It was demonstrated that similar results can be obtained by the analysis method of load path distribution using U_{sum}^* as those by comprehensive judgments based on the understanding of stress, deformation state, and plate thickness, and that the method using U_{sum}^* is valid as an analysis method for load paths.
- (2) The method based on U_{sum}^* made it possible to indicate the load path state in a structure by using only single indicator and judge the state easily compared to the abovementioned method using stress and the like. As an application example, it was shown that when U_{sum}^* is employed load path distribution throughout the structure can be judged even in a complicated automotive body structure.
- (3) Because load path distribution in the whole structure can be understood at a glance by the analysis using U_{sum}^* , it was made possible to qualitatively grasp the members which display a high similarity in load path distribution under differing loading conditions. As a result, it was shown that the members at which similar loads are acting under differing loading conditions can easily be identified in a complicated automotive body structure.
- (4) It was shown that the torsional loading condition displays a high level of similarity to other loading conditions in terms of load-path distribution. Under this condition, because forces act on each frame members in the bending or axial direction and shearing loads are generated on each panel, load paths are widely distributed throughout all the body structure.

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Development of SH-AWD System using EPS Relative Steering Angle Signal

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ABSTRACT

The Super Handling-All Wheel Drive (SH-AWD) system uses the steering angle value to determine the proper left and right rear wheel torque distribution ratio to enhance vehicle handling response. Cost reduction in the application of the SH-AWD system is realized by (1) replacing the conventional stand-alone absolute steering angle sensor with an EPS resolver sensor, which determines the position of the EPS power assist motor, (2) applying integration to the change in the EPS motor position over time to yield a steering angle value relative to the initial steering rack position, and (3) using a new midpoint learning algorithm to determine the true steering angle value by calculating the difference between the initial steering rack position and the straight-ahead driving position. Through this process, a steering angle value relative to the initial steering rack position is acquired and is equivalent to the value that would be provided by an absolute steering angle sensor. To determine the straight-ahead steering position, a learning period is required during which the vehicle needs to travel a minimum distance every time the ignition key is turned on. To meet all existing system and CBU performance requirements during this learning period, two algorithms were created. First, a steering angle estimation algorithm substitutes as the true steering angle to allow SH-AWD to perform normal torque distribution operation. This algorithm uses two independent calculations to prevent excessive steering angle estimation, which would distribute more torque than necessary. Second, a new vehicle stability enhancement algorithm reduces the SH-AWD rear torque amount when an unstable driving event occurs. Using these algorithms, the SH-AWD system can achieve comparable performance by acquiring steering angle data from an EPS resolver sensor as opposed to a more costly absolute steering angle sensor.

1. Introduction

The SH-AWD system receives data from sensors and other control systems within the vehicle to determine a proper rear differential torque amount. The steering wheel angle represents the driver's intention to turn left or right and is used by the SH-AWD system to distribute torque to the left or right rear wheel for enhanced handling response.

Two types of steering angle sensors are commonly used in the industry. An absolute-type steering angle sensor permanently stores the straight-ahead reference position to immediately output the absolute position of the steering wheel when the ignition key is turned on, but this sensor is typically more expensive than a relative type sensor. Additionally, if the absolute-type stored midpoint is incorrect, such as after changes are made to the vehicle steering or suspension, the midpoint value must be reset by a service procedure or else system malfunction may

occur and could lead to customer complaints or additional warranty cost. A relative-type steering angle sensor has a simple low cost design but no permanent reference point to indicate the straight-ahead driving steering position, which is used as a reference point to determine the absolute position of the steering wheel to the left or right. Therefore, the relative-type sensor requires a midpoint learning algorithm to determine the straight-ahead position every time the ignition key is turned on. The midpoint learning algorithm needs the vehicle to travel a minimum distance to find the center steering position, which could delay the operation of some systems. The purpose of this research was to reduce the cost of the SH-AWD system by removing the stand-alone steering angle sensor and replacing it with the EPS system's resolver sensor, which is capable of calculating relative-type steering angle data. For vehicles that use EPS, the steering angle can be calculated without adding cost.

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The influence of the steering angle value on the SH-AWD system operation was determined and objective performance requirements were created to evaluate the effectiveness of this new system configuration. The performance concerns of using the EPS relative-type steering angle signal were addressed as a part of this research.

2. Development Goals

The goal for the all-wheel-drive system's performance is to be consistent at all times, especially when climbing a steep hill or accelerating from a stop on a slippery surface. Changes made in the control system should be smooth so that the vehicle's behavior is not disturbed. Using a relative-type steering angle sensor creates two control states, where the true steering angle is initially unknown and where it is known after a learning interval. Therefore, new software algorithms are needed to enhance the performance of SH-AWD in this initial state. Once the midpoint learning is complete and the true steering angle is known, the EPS relative steering angle signal is identical to the absolute-type steering angle signal, so there is no impact to the SH-AWD system performance.

The SH-AWD system needs to meet the existing level for performance regardless of which control state the system is currently operating in, and it should operate in a way that is transparent to the driver. To achieve these targets, there are three new control algorithms that operate only during the initial control state: (1) a new midpoint learning algorithm compatible with the EPS relative-type steering

angle, which quickly and accurately finds the midpoint value to reduce the time spent in the initial control state; (2) an algorithm to allow torque vectoring of the rear axle to meet the system performance targets by estimating the steering angle value using other vehicle parameters; and (3) an algorithm to maintain vehicle stability while accelerating and turning to assist other vehicle control systems, which cannot perform their normal control until after the steering angle midpoint learning is complete.

3. EPS Relative-type Steering Angle Sensor

The EPS system uses a resolver sensor to monitor the electric power assist motor angle. Figure 1 shows the mechanical layout of the assist motor and resolver sensor. The EPS motor travels several complete revolutions while the steering rack moves from end to end.

The EPS resolver sensor uses electrical windings on a stator to measure the rotor position of the power assist motor. As shown in Fig. 2, the voltage relationship between the resolver windings determines the power assist motor angle. The motor angle value is integrated to calculate the total movement along the steering rack. The EPS system does not need to know the absolute position of the motor on the steering rack—only the driver's intention to steer left or right. The integrated resolver motor angle multiplied by the steering ratio results in a steering wheel angle value relative to the initial position of the power assist motor at the time that power was first applied to the system. Once system power is lost, the existing motor position is also lost and these values become unknown.

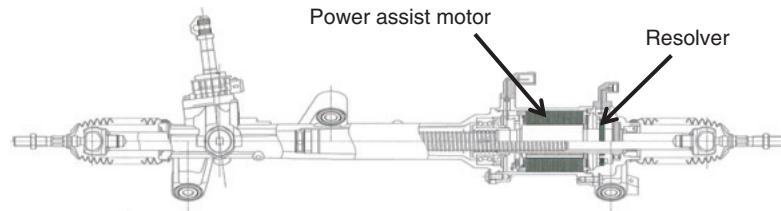


Fig. 1 Electric Power Steering rack structure

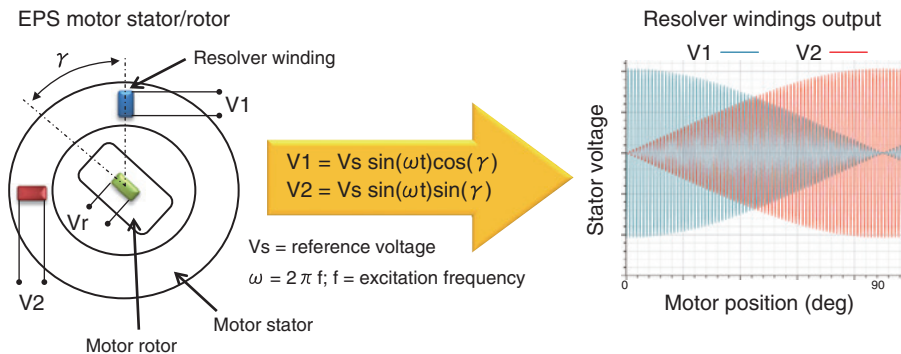


Fig. 2 EPS resolver sensor motor angle calculation

4. Midpoint Learning Algorithm

The midpoint learning algorithm is designed to take a steering angle value that has no reference point for straight-ahead driving and determine the amount of offset required to make that relationship known. Some relative-type steering angle sensors have built-in hardware that report when the steering wheel is in the straight-ahead position, in which case the midpoint learning algorithm must clarify if the wheel is truly at 0 degrees or at some multiple of 360 degrees. For the EPS relative-type steering angle, no such built-in hardware exists for the straight-ahead position, so a high accuracy algorithm is required. This algorithm can quickly determine that the vehicle is moving straight ahead and calculate the offset amount required to calibrate this position to 0 degrees.

To meet performance requirements, an absolute-type steering angle sensor must provide steering angle data at or above a required level of accuracy. To meet the same SH-AWD system performance requirements, the EPS relative steering angle signal must also achieve the same level of accuracy. To achieve this level of accuracy with a relative-type steering angle signal, many driving mode requirements were created. To allow for high accuracy and fast detection, two straight-ahead driving judgment calculations were used. The first judgment uses wheel speed requirements along with yaw rate and lateral G characteristic tables to judge that the vehicle is driving stable along a straight path. The second judgment uses different wheel speed requirements and two steering angle estimation algorithms (Eqs. 1 and 2) that must be below a threshold amount. Figures 3 and 4

show the vehicle geometry parameters for steering angle estimation and the estimation accuracy achieved while driving on a predetermined winding road course.

$$\text{Steering angle by rear wheel delta rotation (deg): } \frac{|V_o - V_i| * WB * SR * 360}{(V_o + V_i) * TR * \pi} \quad (1)$$

$$\text{Steering angle by yaw rate and speed (deg): } \frac{WB * YAW * SR}{\text{Vehicle speed}} \quad (2)$$

Using these calculations alone resulted in midpoint accuracy of ± 6 degrees, with the worst case data points

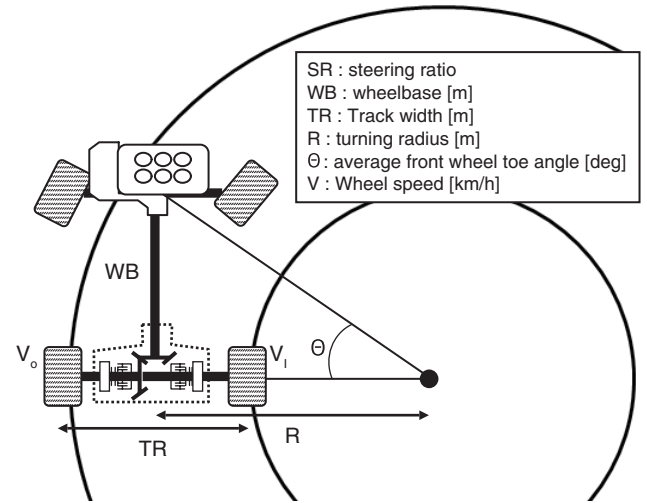


Fig. 3 Vehicle geometry parameters for steering angle estimation

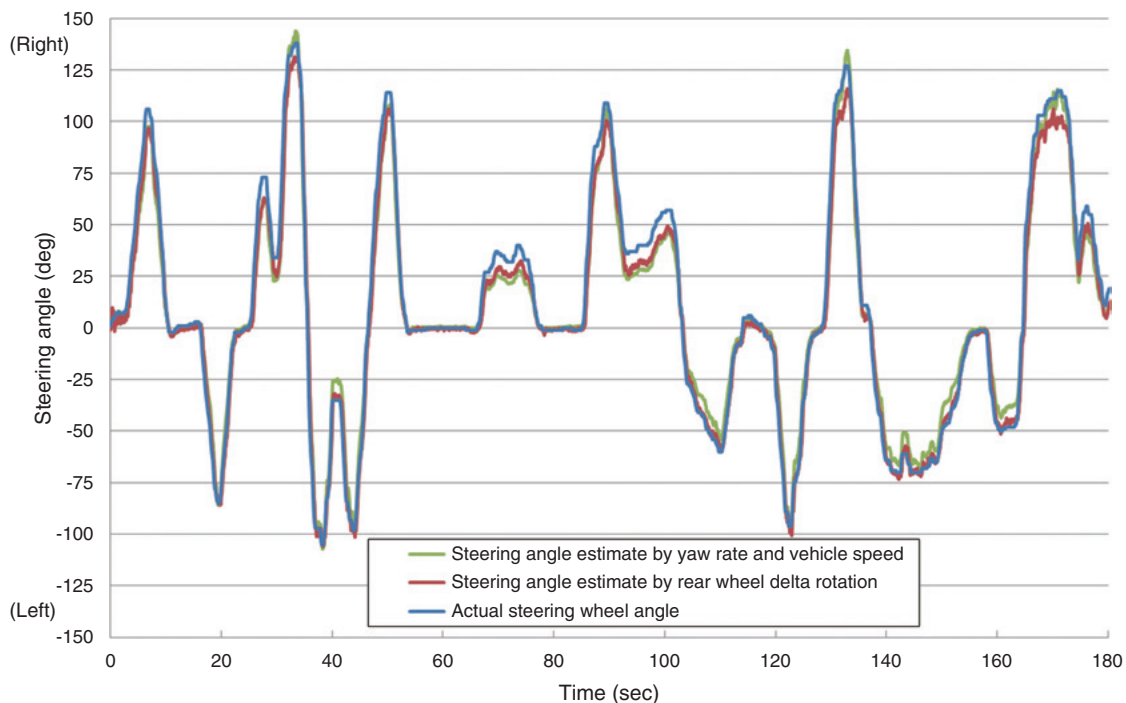


Fig. 4 Steering angle estimation accuracy

occurring during extreme driving conditions. As shown in Fig. 5, the addition of requirements for a stable EPS steering angle value along with a one second averaging algorithm achieved results of ± 3 degrees for all driving situations. This level of accuracy meets the SH-AWD system design requirement. This calibration usually finishes within the first 400 m of driving under normal driving conditions, but can be shorter or longer based on the availability of a straight road.

The total error of the steering angle signal includes the midpoint learning algorithm error, the EPS system mechanical error, and the yaw rate sensor installation angle error. The EPS system mechanical components have a base amount of error regardless of steering rack position as well as error that increases toward both ends of the steering rack.

The yaw rate and lateral G sensor have an acceptable range for zero point accuracy in order to be used by the SH-AWD system. If both sensors are at the upper limit of acceptable range, the steering angle midpoint learning algorithm has a small amount of additional error. The total overall steering angle error is ± 8.4 degrees (see Fig. 6). The EPS steering angle signal measurement point is directly on the steering rack, rather than behind the steering wheel as with the stand-alone steering angle sensor. This removes up to 5 degrees of deflection that occurs within the steering column. Therefore, the total overall steering angle error of the EPS relative type and stand-alone absolute type are at an equal level, and no SH-AWD performance impact is introduced due to using the EPS relative steering angle signal.

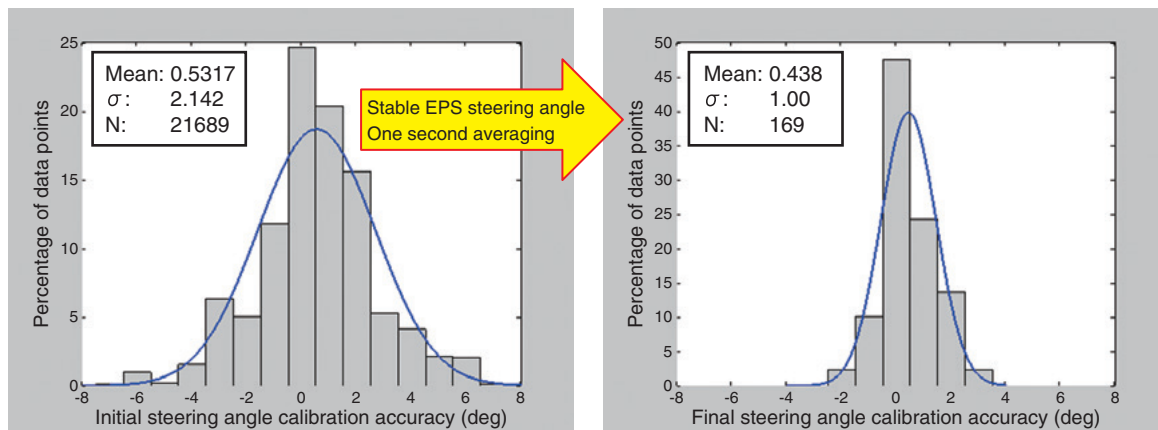


Fig. 5 Midpoint learning algorithm accuracy

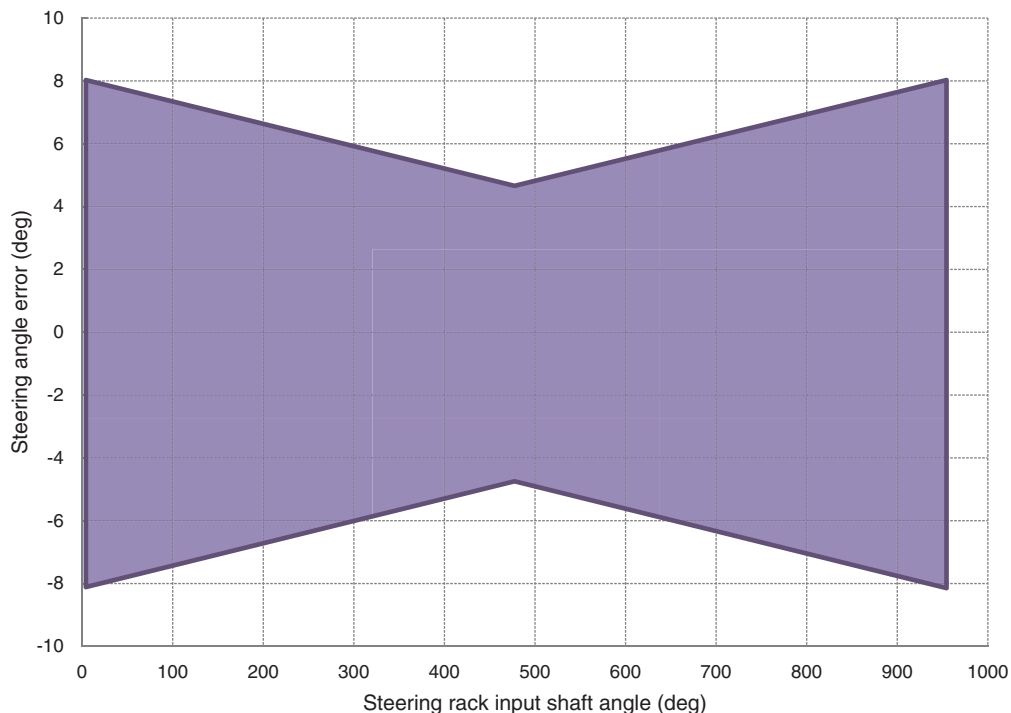


Fig. 6 EPS relative-type steering angle calculation error

5. Steering Angle Substitution

To achieve a SH-AWD system performance that is consistent and transparent to the driver, a substitute for the EPS steering angle value is required until the midpoint learning algorithm has finished, otherwise the SH-AWD will not be able to apply torque vectoring and vehicle performance will be reduced. The two steering angle estimation algorithms used for midpoint learning have very good accuracy for stable driving maneuvers, where the vehicle dynamics are linear. As shown in Fig. 7, as the driver operates the vehicle closer to the tire grip limit, the input relationships become non-linear and the steering angle estimation algorithms lose accuracy.

The design concept for steering angle substitution is to maintain the SH-AWD system performance during stable driving situations and prevent applying more rear torque than is normally distributed, which could lead to unstable vehicle dynamics. To achieve this concept, the estimated steering

angle used for SH-AWD control must be the same or lower in amplitude than if the system were using a calibrated steering angle sensor. The variables in the estimation calculations are different and, during certain driving maneuvers, one of the steering angle variables could become larger than intended, in which case the desired result is to use the other more accurate signal. Logic was made to selectively use the estimated steering angle value with the lowest amplitude and slope. This logic is effective at preventing a final estimated steering angle that is too large. During a transient maneuver it is possible for one steering angle estimate to suddenly decrease giving it the lowest amplitude, but by using the slope parameter, the signal switching does not occur until that signal stabilizes. This helps give the SH-AWD system consistent performance even during transient driving maneuvers near the tire grip limit. During unstable driving situations, one steering angle estimate can be a positive value and the other a negative value, in such case a final steering angle value of 0 degrees is used. Figure 8 shows the

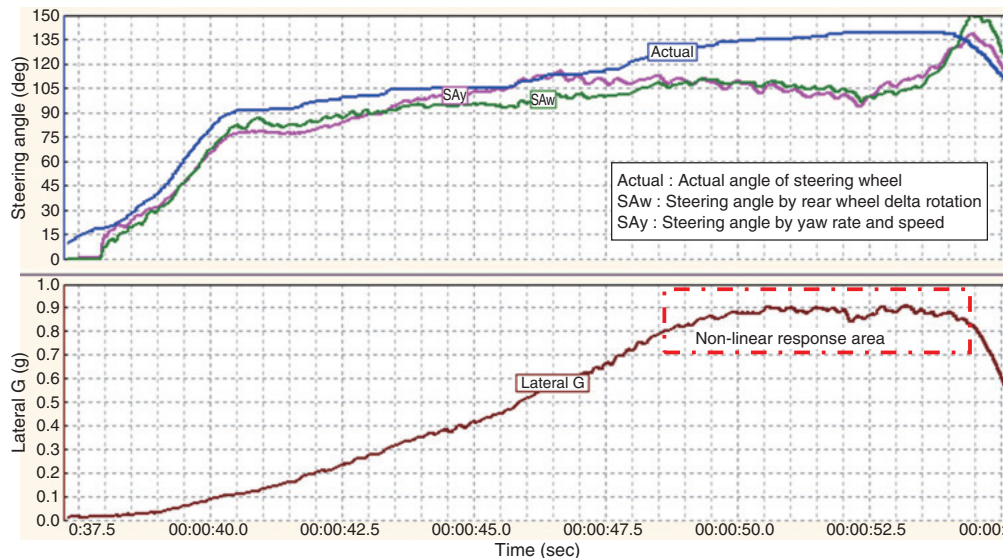


Fig. 7 Steering angle estimation accuracy versus lateral G

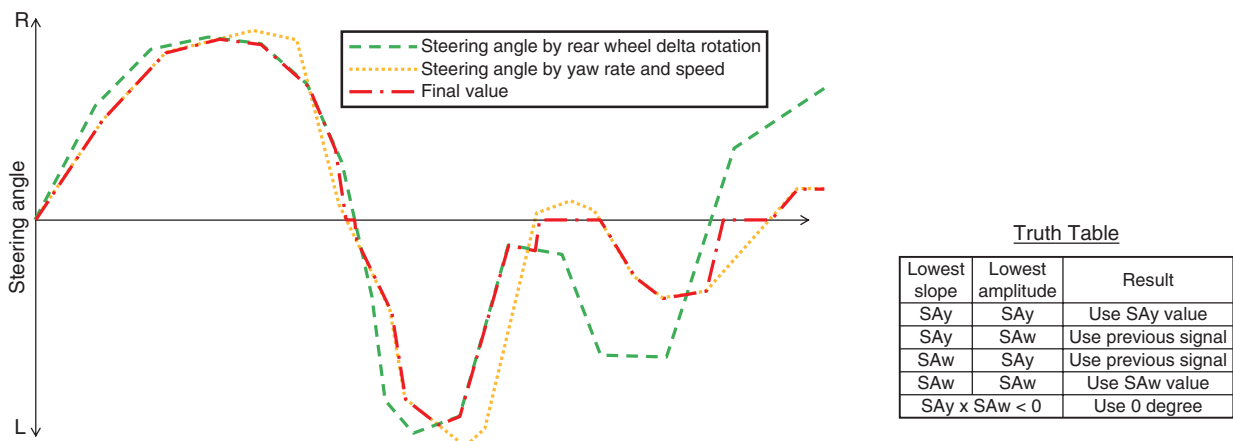


Fig. 8 Estimated steering angle signal selection logic

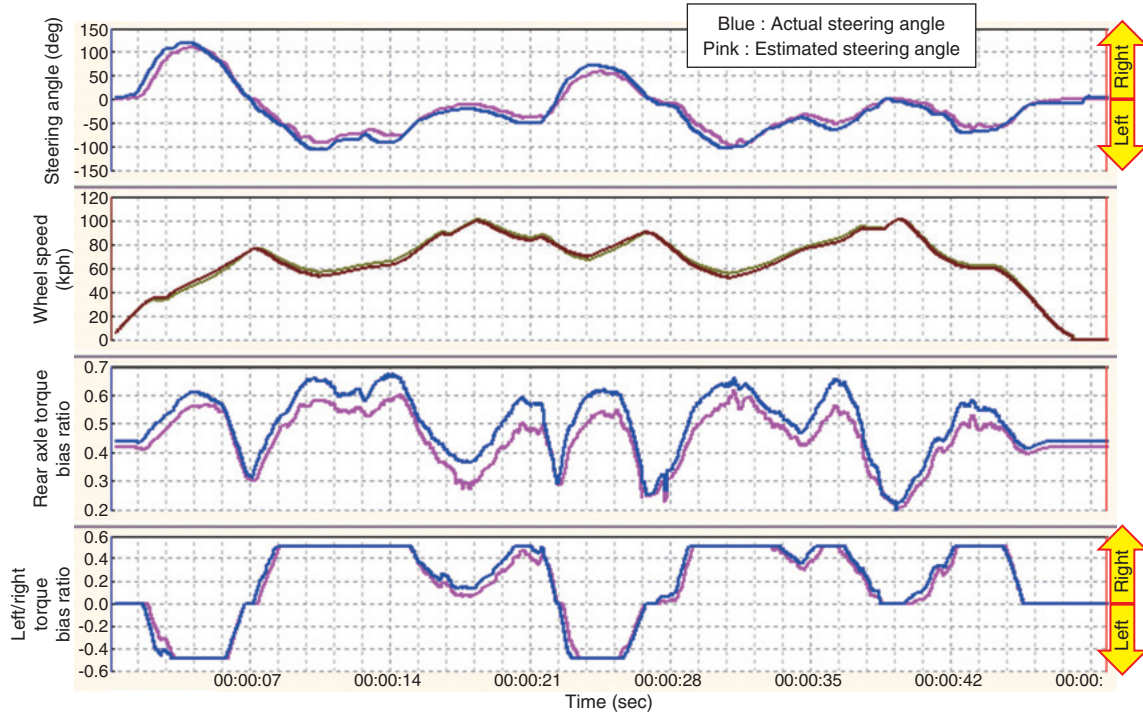


Fig. 9 SH-AWD system torque distribution change from estimated steering angle

operation of the estimated steering angle selection logic.

The change in SH-AWD system performance with the estimated steering angle was evaluated using hardware-in-the-loop simulation. The desired effect of using the estimated steering angle to allow torque vectoring to occur normally was achieved, but with slightly less rear torque and left/right distribution. As shown in Fig. 9, the effect on the front/rear torque distribution and rear axle left/right torque distribution yielded a worst case reduction of 10% compared to using the true steering angle value. The SH-AWD system performance using the estimated steering angle was evaluated by vehicle handling experts using the standard SH-AWD test courses and driving maneuvers. The SH-AWD system performance requirements were met and the system was judged acceptable. This amount of torque reduction was judged by vehicle handling experts to be similar in feeling to a slight change in road surface grip, which is typical under normal driving conditions.

6. Vehicle Stability Enhancement

Other control systems in the vehicle have limited performance capability while calculating their steering angle midpoint. During this time, the driver could experience an unstable driving situation that would normally be stabilized by one of these systems. Therefore, each control system that affects the vehicle dynamics must be capable of regulating its own performance to maintain stability during the midpoint learning time. When the absolute steering angle sensor is applied, this performance regulating function is

not needed. However, for the SH-AWD system to apply the EPS relative steering angle signal, an algorithm is required to determine when and how much to regulate performance. When accelerating and turning on road surfaces with a low coefficient of friction (low μ), the SH-AWD system can cause oversteer to occur if too much rear axle torque or left-to-right bias torque is applied. To enhance the stability in this situation, the SH-AWD system limits the rear axle torque distribution when an unstable driving event is detected (see Fig. 10). The SH-AWD system uses a normalized wheel slip calculation to determine vehicle instability and a yaw rate error calculation to determine that the car is oversteering. Using this information, the SH-AWD system sets an upper limit for the rear axle torque to stabilize the vehicle.

Understanding the speed relationship between all four wheels is critical to understanding how to detect when too much torque is applied and how to correct it. As shown in Eqs. (3)-(5) and Fig. 11, a normalized wheel slip calculation was made by comparing the wheel speed relationships between left and right along with front and rear. This wheel slip calculation has a stable driving region under which the data for all normal driving situations fall, and any data beyond this region occurs only when a vehicle loses traction.

$$\text{Left/Right Speed Delta: } V_{SLIP1} = (|V_{FL} - V_{FR}| - |V_{RL} - V_{RR}|) \quad (3)$$

$$\text{Front/Rear Speed Delta: } V_{SLIP2} = (|V_{FL} - V_{RL}| - |V_{FR} - V_{RR}|) \quad (4)$$

$$\text{Normalized Wheel Slip: } MAX(V_{SLIP1}, V_{SLIP2}) - \text{StableRegion} \quad (5)$$

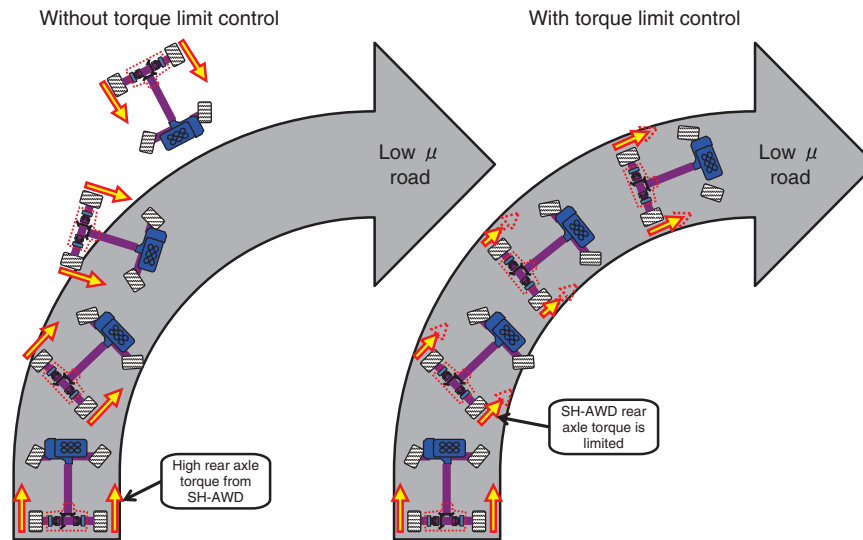


Fig. 10 SH-AWD vehicle stability enhancement operation concept

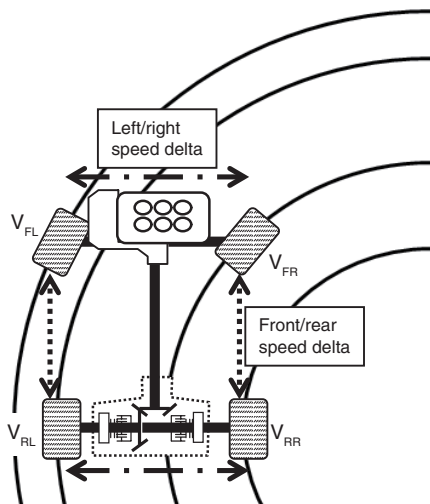


Fig. 11 Normalized wheel slip calculation

Figure 12 shows the normalized wheel slip calculation relationships during a 5 m radius turn (which has a very high speed difference between each wheel) and shows the ability of this logic to differentiate between wheel slip and tight turning. Initially the vehicle is able to do a 5 m radius turn while maintaining grip with the road surface, but as vehicle speed increases the tires eventually lose grip and the vehicle becomes unstable. The SH-AWD system removes torque from the inside rear wheel while cornering and distributes it to the outside rear wheel. This creates a situation where the inside and outside rear wheel increase speed at different rates when they lose grip.

If the vehicle loses traction driving straight, the steering angle is not needed by other systems to stabilize the vehicle; therefore, a yaw rate error algorithm was created inside the SH-AWD system to detect instability only while turning. The yaw rate error algorithm works with the normalized

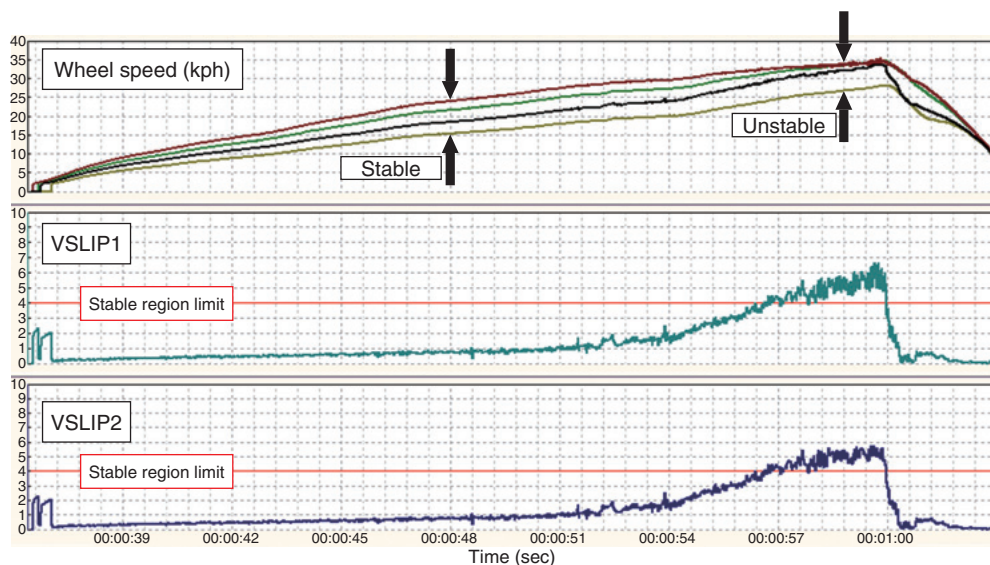


Fig. 12 Normalized wheel slip calculation during an accelerating 5 m radius turn

wheel slip calculation to determine when the SH-AWD system should limit the rear torque output. As shown in Eqs. (6) and (7), the actual yaw rate of the vehicle is compared to an estimated yaw rate calculated using the vehicle speed and lateral G sensor to determine the yaw rate error amount.

$$\text{Estimated Yaw Rate (deg/s)} = \frac{\text{Lateral } G \left(\frac{m}{s^2}\right) * 180}{\text{Vehicle speed} \left(\frac{m}{s}\right) * \pi} \quad (6)$$

$$\text{Yaw Rate Error} = |\text{YAW} - \text{Estimated YAW}| - \text{StableRegion} \quad (7)$$

During stable driving maneuvers, the actual and estimated yaw rates are nearly the same with a small amount of inaccuracy that decreases as the vehicle speed increases. As shown in Fig. 13, this inaccuracy is taken into account in order to create a stable driving region for the estimated yaw rate. When the yaw rate error goes beyond the stable driving region, the vehicle is oversteering and the SH-AWD system needs to limit the rear axle torque.

The SH-AWD system has several activation requirements before limiting the rear axle torque in order to prevent system misoperation. Situations such as driving on gravel or bumpy roads may cause the stability algorithm to temporarily go beyond the stable region, even though the vehicle is actually stable. To correct for such situations, the instability must be judged for a minimum time and the wheel speed, yaw rate, and lateral G sensor values must be within an allowable range. Using these conditions, the torque limit misoperation was eliminated.

To determine the torque limit amount, the normalized wheel slip calculation and the yaw rate error calculation are combined to create an instability index value. This value represents how unstable the vehicle is in a range from 0 to 10, with 0 representing good stability and 10 representing

very unstable driving. The SH-AWD system normally reduces the total rear axle torque as vehicle speed increases; therefore, if instability occurs at higher speeds, the torque limit amount needs to be lower to achieve the stabilizing effect. The torque limit starts at a default amount and is scaled by the instability index value and vehicle speed. As shown in Fig. 14, as the instability index value increases, the torque is limited to a lower value because more stabilization is needed.

To evaluate the torque limit control, the performance was compared to a vehicle using an absolute-type steering angle sensor with all driver aids enabled on both vehicles. The torque limit control is only needed during midpoint learning, so the vehicle with the EPS relative-type steering

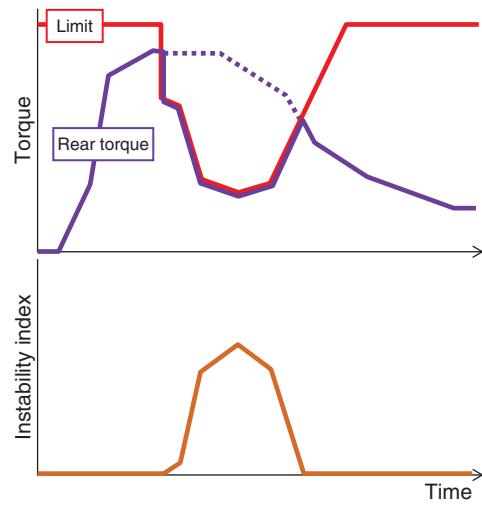


Fig. 14 Torque limit based on instability index value

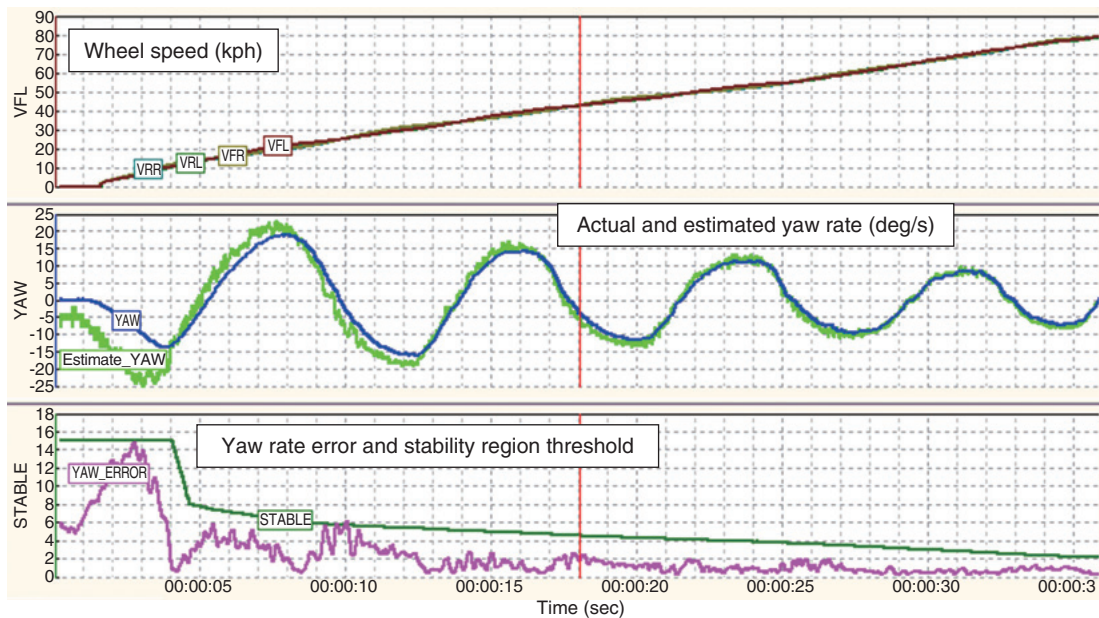


Fig. 13 Yaw rate error calculation

angle was only evaluated in this condition. The driving modes that needed torque limit control the most were a slalom or constant radius turn on snow or ice as well as driving on a snow-covered winding road.

Driving aggressively on these courses with an absolute-type steering angle sensor results in other control systems reducing the engine torque or selectively applying the brakes to stabilize the vehicle as necessary. The vehicle using the EPS relative-type steering angle does not have the same level of capability from the other control systems during midpoint learning. The SH-AWD system torque limit function redirects torque to the front axle when instability is detected, which also maximizes the benefit of the other control system capabilities that are available. The driver was easily able to maneuver through these courses in both vehicles with a high level of safety margin. The driver feeling difference between the two vehicles is small enough that changes in the road surface conditions have a similar effect. Given the opportunity for a back-to-back test, most people would not be able to perceive any difference. With this level of performance, the SH-AWD system can apply the EPS relative steering angle signal and all vehicle stability performance requirements are achieved.

7. Conclusion

The EPS relative-type steering angle sensor can be used with the SH-AWD system to meet or exceed performance requirements. A new steering angle midpoint learning algorithm was created to work with the hardware configuration of the EPS system. A steering angle substitution algorithm allows the SH-AWD system to maintain consistent performance capability during the midpoint learning time. The vehicle stability enhancement algorithm allows the SH-AWD system to determine when too much rear axle torque is applied for the current road surface conditions to stabilize the vehicle. With this new logic, using the EPS relative-type steering angle reduces overall system cost and the difference in performance compared to an absolute sensor should be imperceptible.

■ Author ■



Steve TURNER

Image Segmentation-based Moving Object Detection for Driver Assistance System

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Yiannis ALOIMONOS**

ABSTRACT

The detection of objects in front of ego-vehicle is an effective means to help prevent collisions. Since the potential of collision with moving objects, such as other vehicles and pedestrians, is relatively high, it is important to detect these moving objects in particular. Vision sensors are employed to detect the various objects encountered in actual driving situations, but because the camera and the objects are moving independently, it is not easy to detect independently moving objects using vehicle-mounted cameras. The detection of sharp boundaries for moving objects has been especially challenging. We have therefore developed two algorithms to detect sharp boundaries for moving objects. The first of these algorithms detects objects moving in a different direction to the camera by identifying inconsistency between motion in the images and the epipolar constraint. The second algorithm detects objects moving in the same direction as the camera by identifying inconsistency between stereo disparity and motion disparity. The use of image segmentation technology in these algorithms has made it possible to detect sharp boundaries for moving objects. The ability to detect sharp boundaries using these methods was verified in tests using actual vehicles.

1. Introduction

Among Intelligent Transportation Systems (ITS), Advanced Driver Assistance System (ADAS) is a field that is the subject of vigorous research. Sensors for the detection of objects in the vicinity of the subject vehicle, including other vehicles, pedestrians, lane markers, and traffic signals, are key components of ADAS. High expectations have been placed on visual sensors in the detection of these various objects around the vehicle. From the perspective of safe driving assistance, it is essential to detect independently moving objects such as vehicles and pedestrians, because these represent a higher risk of collision with the subject vehicle. However, it is not easy to detect independently moving objects from images. This is because stationary objects also appear to be moving in images captured by a camera mounted on a moving vehicle. The determination of a method of detecting the specific locations of moving objects from moving images of this kind represents an issue for research^{(1), (2)}. When the direction of motion of moving objects differs from the direction of motion of the camera, the moving objects can be detected using a single camera (a monocular camera) if epipolar constraints⁽³⁾ are applied. Epipolar constraints express the constraint conditions in relation to the apparent motion of the stationary objects. When the direction of motion of the moving object is the same as that of the camera, it cannot be detected using a

single camera. This necessitates the use of two cameras (stereo camera), a factor which also increases the complexity of the algorithms employed. The project discussed in this paper therefore developed two algorithms for application in these sensing methods. For automotive applications, it is also necessary to detect the width and size of objects. The method discussed here detects the boundaries of the object using image segmentation technology to divide the image into meaningful regions.

2. Algorithms

Ogale et al. categorized the detection of moving objects from images into three classes, based on the relationship between the movement of the camera and the movement of the object⁽⁴⁾. Objects that move in a different direction to the direction of motion of the camera are defined as belonging to Class 1. In these cases, the object can be identified as a moving object by determining whether or not the movement of the object satisfies the epipolar constraint. The epipolar constraint cannot be employed in the case of objects that move in the same direction as the direction of motion of the camera. Moving objects that move in the same direction as the camera and are partially occluded by stationary objects are categorized as Class 2. In these cases, independently moving objects can be detected by determining whether there is inconsistency between the anteroposterior

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relationship and motion disparity due to the occluding object. When its direction of motion is the same as that of the camera but no occluding objects are present, the moving object is classified as Class 3. In these cases, moving objects can be detected by determining whether any inconsistency exists between motion disparity and the anteroposterior relationship as determined from stereo disparity.

Depending on the distance of the moving object and the stationary object that partially occludes it, there are cases in which moving objects cannot be detected when using methods for the detection of Class 2 moving objects. However, objects falling into any class can be detected using methods for the detection of Class 3 objects.

Two algorithms for the detection of moving objects using vehicle-mounted cameras were developed based on these classifications and principles of detection. The first of these algorithms is employed in the detection of Class 1 moving objects using a monocular camera. The second algorithm is employed in the detection of more general moving objects, including Class 2 and 3 objects, using a stereo camera. The first algorithm is simpler than the second, but the range of moving objects that can be detected using the algorithm is limited. The second algorithm is able to detect more general moving objects, but necessitates the use of a stereo camera. The two algorithms should be used separately depending on the object for detection. In order to relate the two algorithms to actual traffic conditions, the first algorithm is referred to as an algorithm for the detection of intersecting objects, and the second algorithm is referred to as an algorithm for the detection of preceding objects. This means that the first algorithm, which is simpler and feasible using a monocular camera, can be applied to the systems for the detection of intersecting objects, and the second algorithm, which necessitates the use of stereo cameras and entails complexity in the algorithm itself and system components, can be applied to the systems for the detection of preceding objects.

Sections 2.1 and 2.2 of this chapter will discuss the essential points of the algorithms, while Sections 2.3 and 2.4 will respectively consider the algorithm for the detection of intersecting objects and the algorithm for the detection of preceding objects in greater detail.

2.1. Principle of Detection of Moving Objects

The algorithm for the detection of intersecting objects and the algorithm for the detection of preceding objects employ different principles of detection.

Figure 1 shows a representation of the principle of detection employed by the algorithm for the detection of intersecting objects. The epipolar lines⁽³⁾ when the vehicle is proceeding straight ahead are shown as blue lines on this image. In this case, the epipolar lines show the direction of motion of stationary objects. The point of intersection of the epipolar lines is the focus of expansion (FoE). In the image, all of the stationary objects are imaged as if in motion, following the directions of the epipolar lines outward from the FoE. The epipolar constraint expresses the fact that geometric constraints exist for this apparent motion of the stationary objects in the image. When motion at a specific point in the image follows an epipolar line, it is said to satisfy the epipolar constraint. By contrast, intersecting objects such as the pedestrian in the image do not move following the epipolar lines, and do not satisfy the epipolar constraint. A moving object can therefore be detected by determining whether or not the optical flow for each pixel satisfies the epipolar constraint. The optical flow is the motion at each point in the image expressed as two-dimensional vectors. When the subject vehicle is not restricted to proceeding in a straight line and turning, pitch, and roll rotation are incorporated, determination of whether the epipolar constraint is satisfied becomes more complex. Rectification, which will be discussed below, is therefore applied to cancel the effect of rotational motion by deforming the image, making it possible to detect moving objects as in the case shown in Fig. 1.

When the moving object precedes the subject vehicle, the principle of detection using the epipolar constraint cannot be employed and another principle should be used despite the fact that the object is moving, because of the effect of motion following the direction of the epipolar lines. Figure 2 shows a conceptual diagram of the principle of detection for the algorithm for the detection of preceding objects. This diagram shows the same scene, incorporating both stationary and moving objects, captured by a stereo camera (right) and a moving camera (left). Using the stereo camera, the distances to objects can be measured using

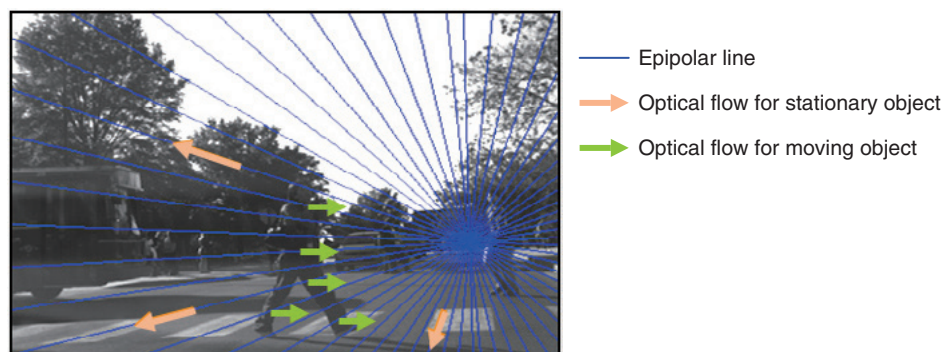


Fig. 1 Principle of detection of moving objects for intersecting objects

triangulation, irrespective of whether the objects are moving or stationary. By contrast, in the case of the moving camera, when the distances to objects are measured by means of triangulation using the images captured at time t and time $t+1$, the distance to stationary objects can be measured correctly, but the distance to moving objects cannot. This is because the object itself moves and changes position during the time that the camera is moving. As Fig. 2 shows, both the stereo camera and the moving camera measure identical distances to the stationary object (shown as the solid lines), but measure different distances to the moving object (shown as the broken lines). Therefore, if the distances to an object as measured by stereo and moving cameras differ in a comparison, the object can be identified as a moving object. Up to this point, stereo cameras and moving cameras have been discussed as separate cameras, but if a stereo camera is considered as being mounted on a vehicle and in a state of motion, one of the cameras (for example the camera on the right-hand side of the vehicle) could be considered to be a moving camera, and this principle could be applied.

2.2. Achievement of Sharper Boundaries via Image Segmentation

The research discussed here attempted to make it possible to detect sharp boundaries of moving objects using image segmentation technology. In the case of the algorithm for the detection of intersecting objects, image segmentation was employed to increase the accuracy of the optical flow; in the case of the algorithm for the detection of preceding objects, image segmentation was used to increase the accuracy of

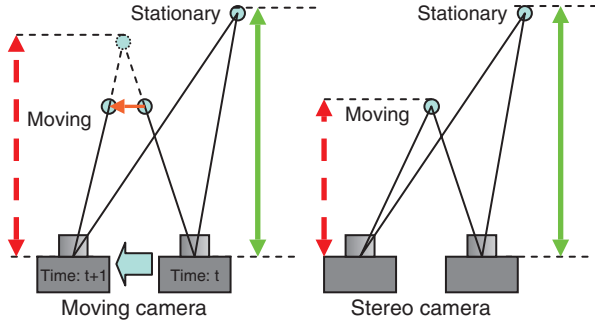
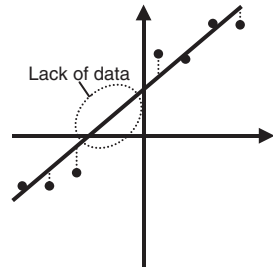
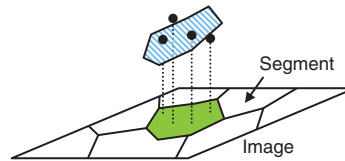


Fig. 2 Principle of detection of moving objects for preceding objects



(a) 1D model fitting



(b) Segmentation-based model fitting on image

Fig. 3 Model fitting based on segmentation

stereo disparity in addition to the accuracy of the optical flow. The concept on which the use of image segmentation is based is identical for both optical flow and stereo disparity, and a brief overview will be provided here. The background concept is model fitting, as shown in Fig. 3(a). In the figure, measured values are shown as points. It is assumed that the measured values are affected by noise, and that some values will be missing, having for various reasons been unable to be measured. Given that in principle the measured values accord with a model expressed as a linear expression, they can be assimilated to a straight line, as shown in the figure. Leaving aside the measured values themselves and taking the straight line resulting from model fitting as representing new measured values makes it possible to reduce the effect of noise and obtain more appropriate values even for missing values. Figure 3(b) shows a diagram produced by the application of this concept to measured values in an image. As the figure shows, the segmentation process divides the image into segments, and measured values can be obtained within the shaded segment (as shown by the black points). For example, if the measured values are assumed to accord with a plane model, model fitting using the measured values within the segment makes it possible to obtain the hatched plane that represents measured points. In the research discussed here, motion models were used for optical flow, and disparity models were used for stereo disparity. Conventional methods of calculating the optical flow or the stereo disparity use texture in the vicinity of the pixels to be calculated, assuming that the optical flow or the stereo disparity is uniform. Therefore, calculation could not be performed in regions with less texture and it was a challenge to determine the boundaries of objects due to increased errors in regions in which the assumption did not hold. Segmentation-based model fitting identifies regions in which the optical flow and stereo disparity change uniformly according to models in advance. Reducing the effect of noise and obtaining more appropriate values even for the regions with less texture by means of model fitting using image segmentation has made it possible to accurately determine the changes in optical flow and stereo disparity at the boundaries of the object.

2.3. Detection of Intersecting Objects

Ego-motion estimation^{(5), (6)} and optical flow estimation were the essential elements in realizing the algorithm for

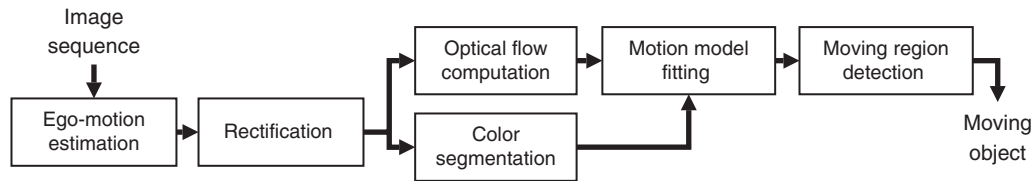


Fig. 4 Block diagram of intersecting object detection

the detection of intersecting objects. Ego-motion estimation was necessary for rectification, which will be discussed below. A method of calculation of the optical flow based on color segmentation was developed in order to obtain dense optical flows with sharp boundaries for the moving objects. Color segmentation is a method of image segmentation that divides images based on color homogeneity. Figure 4 shows the composition of the algorithm. Two continuous image frames represent the input. The motion of the camera is estimated during “Ego-motion estimation,” and “Rectification” processing is applied in order to cancel the effect of the rotation of the camera on the image. During the “Optical flow computation” process, the initial values of the optical flow are calculated, and “Color segmentation” is applied to the first image. Next, during the “Motion model fitting” process, motion model parameters that express the motion of each segment obtained from color segmentation on the image are determined from the initial optical flow by model fitting. During the “Moving region detection” process, the region occupied by the moving object is detected by identifying the pixels for which the optical flow does not follow the epipolar lines. The main elements of the algorithm will be discussed in more detail below.

2.3.1. Rectification

This sub-section will discuss the “Rectification” processing that is applied to cancel the effect of the rotation of the camera after the camera motion parameters have been determined during “Ego-motion estimation” in the processing flow shown in Fig. 4.

The optical flow is obtained from two continuous image frames. When the translational and rotational motion of the camera are determined by ego-motion estimation, it is possible to deform the images in order to shift the epipoles⁽³⁾ of the two images to the centers of the images and cancel

the motion in the images resulting from the rotational motion of the camera. As shown in Fig. 5, by this means, despite the fact that the motion of the camera incorporates both translational and rotational motion, the images can be deformed to make it appear that the camera displays only translational motion. This deformation of the images makes it easier to apply processing in the later stages to determine whether or not the optical flow follows the epipolar lines. In other words, doing away with the effect of the rotational motion of the camera makes it possible to more easily detect the moving object, as in the case shown in Fig. 1. Figure 6(b) shows an image to which rectification processing has been applied. In this example, rectification has slightly distorted the entire image in Fig. 6(b) in relation to the original image shown in Fig. 6(a).

2.3.2. Motion model fitting using segmentation

Again looking at the processing flow shown in Fig. 4, this sub-section will discuss the method of estimation of the motion model parameters for “Motion model fitting” using the initial optical flows calculated in the “Optical flow computation” process and the segments obtained in the “Color segmentation” process.

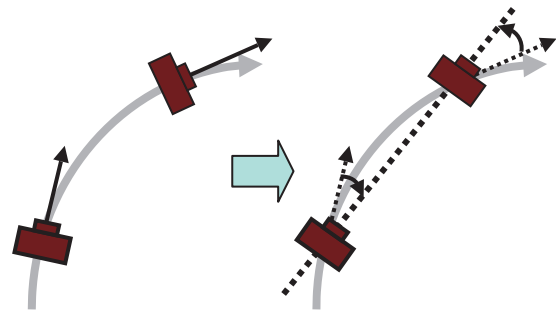


Fig. 5 Concept of rectification

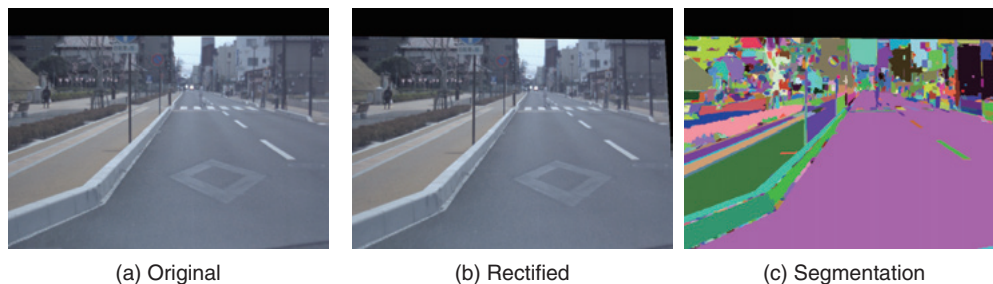


Fig. 6 Examples of rectification and segmentation

When viewed in terms of distance from a vehicle driving on the road, the objects that appear in the traffic environment – the road surface, other vehicles, signs, and so on – can be approximated to a collection of planes. When the motion in the image is determined, at a certain distance each section of even non-rigid bodies such as pedestrians can be approximated as planes. Because the motion of the planes in the two image frames is expressed as homography transformations, it is logical to model the motion on the images of each plane making up the moving object using homography transformations. A method of calculating the optical flow based on color segmentation was therefore developed, in which the images are divided into segments based on color homogeneity. Each segment is treated as part of a plane, and it is assumed that the motion of the segments can be approximated using homography transformations. First, the initial optical flows are calculated by means of a block matching method using two continuous rectified image frames. The initial optical flows are only calculated for regions within the blocks in which sufficient texture exists. Next, the first image of the two frames is segmented based on color homogeneity using the mean-shift method⁽⁷⁾, one of the color segmentation methods. Figure 6(c) shows an example of segmentation. Pixels painted the same color belong to the same segment.

Like other methods based on image segmentation⁽⁸⁾, it was assumed that the boundaries of the segments incorporate the object boundaries, and that each segment images part of a plane. Parameters are estimated by fitting in a motion model in which the initial optical flows for each segment are expressed as homography transformations. When points are imaged on a plane, the motion of a point at a specific position (x_1, y_1) in a frame when the point moves to another position (x_2, y_2) in the next frame is modeled using the following equation:

$$w \begin{pmatrix} x_1 \\ y_1 \\ 1 \end{pmatrix} = \begin{pmatrix} h_{11} & h_{12} & h_{13} \\ h_{21} & h_{22} & h_{23} \\ h_{31} & h_{32} & 1 \end{pmatrix} \begin{pmatrix} x_2 \\ y_2 \\ 1 \end{pmatrix} \quad (1)$$

where w is a quantity that represents the scale of homography transformation, obtained by determining the third factor after the transformation as 1, and $\{h_{ij}\}$ is the homography transformation parameters to be estimated. Given that the destination position (x_2, y_2) for each point at an initial position (x_1, y_1) can be determined from the initial optical flow, the substitution of the corresponding points

makes it possible to estimate the parameters. In order to reduce the effect of errors, parameters were estimated using the RANSAC method⁽⁹⁾, a robust estimation method. When the estimated parameters are used to calculate the motion of each pixel in the segments, more accurate optical flows with sharper boundaries than the initial optical flows are obtained because of using color segmentation.

2.3.3. Moving region detection

This sub-section will discuss the “Moving region detection” process in the processing flow shown in Fig. 4, in which the moving region is identified following the determination of more accurate optical flows in the “Motion model fitting” process.

Moving regions in the images are identified by determining that the direction of the optical flow for each pixel differs from the direction of the corresponding epipolar line. Because rectification has been applied to cancel the rotational motion of the camera and shift the epipoles to the centers of the images, the epipolar line corresponding to any specific point is found as a straight line connecting that point and the center of the image. If the direction of the optical flow for each pixel does not follow the epipolar lines, the object can be identified as a moving object.

2.4. Detection of Preceding Objects

In order to detect objects moving in the same direction as the subject vehicle, independently moving points are identified by comparing their distance as estimated from their motion in the image and their distance as measured by the stereo camera, and they are classified as belonging to a moving object. It is possible to identify moving objects because the distances match in the case of stationary objects, but differ in the case of moving objects. The use of color segmentation in conjunction with this method helps to enable sharp boundaries to be obtained.

Figure 7 shows the entire algorithm as a block diagram. The additional elements in relation to the algorithm for the detection of intersecting objects are stereo disparity computation based on image segmentation and distance estimation based on motion. In addition, because of the difference in the principle of detection, the method of identification of the moving object region in the image differs from that employed in the case of detection of intersecting objects. The main elements of the algorithm are discussed in greater detail below.

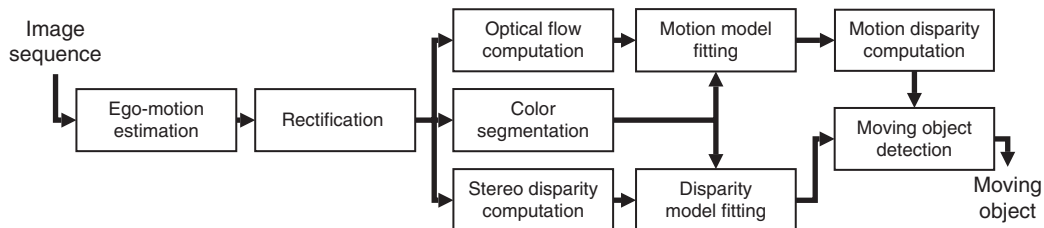


Fig. 7 Block diagram of preceding object detection

2.4.1. Stereo disparity model fitting based on image segmentation

This sub-section will discuss the method of estimating stereo model parameters during the process of “Disparity model fitting” using the initial stereo disparity as calculated during the “Stereo disparity computation” process and the image segments obtained from the “Color segmentation” process, as shown in the processing flow in Fig. 7.

As in the case of calculation of the optical flow, model fitting is employed using the initial stereo disparity in order to estimate the disparity model parameters for each image segment. Because the initial stereo disparity is determined by block matching, it can only be calculated for regions with sufficient texture. In addition, in order to minimize error due to occlusion of objects, the reference image is replaced and the disparity of the right image in relation to the left image and of the left image in relation to the right image is determined; the figures obtained are employed as disparity values only in cases in which there is a match between the two. The mean-shift method is employed to segment one of the stereo images based on color homogeneity, and the disparity model parameters are estimated by fitting initial disparity maps to each segment. At this time, the stereo images are rectified to render them as parallel stereo images. For parallel stereo images, the corresponding points on the images from the left and right cameras (x_1, y_1) and (x_2, y_2) for a plane in three-dimensional space (X, Y, Z) expressed by $\alpha X + \beta Y + \gamma Z = 1$ satisfy the following equation.

$$\begin{pmatrix} x_1 \\ y_1 \\ 1 \end{pmatrix} = \begin{pmatrix} \alpha' & \beta' & \gamma' \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{pmatrix} \begin{pmatrix} x_2 \\ y_2 \\ 1 \end{pmatrix} \quad (2)$$

Here, α' , β' and γ' are defined as follows.

$$\alpha' = B\alpha + 1 \quad (3)$$

$$\beta' = \frac{f_x}{f_y} B\beta \quad (4)$$

$$\gamma' = f_x B \left(\gamma - \frac{\alpha c_x}{f_x} - \frac{\beta c_y}{f_y} \right) \quad (5)$$

B is the length of the baseline between the stereo cameras, and f_x and f_y are the x and y components of the cameras' focal lengths in units of pixels. (C_x, C_y) is the principal point, at which the image plane and the optical axis intersect. Because (x_1, y_1) and (x_2, y_2) are determined when the initial stereo disparity is calculated, the disparity model parameters α' , β' and γ' can be obtained for each segment using Eq. (2). The calculation of disparity for each pixel using the model parameters that have been determined makes it possible to obtain more accurate disparity maps than the initial disparity, and also increases the clarity of the boundaries of the disparities.

2.4.2. Estimation of distance from motion

This sub-section will discuss the estimation of distance from motion during the “Motion disparity computation” process following the determination of high-accuracy optical flows by “Motion model fitting” in the processing flow shown in Fig. 7.

By contrast with the detection of intersecting objects, for the detection of preceding objects it is necessary to estimate distance from motion in the images in order to compare distances in the stereo images. Because rectification has been applied to cancel the rotational motion of the cameras, the distance Z can be estimated as follows.

$$Z = T \frac{x}{v_x} = T \frac{y}{v_y} \quad (6)$$

Here, T is the degree of translational motion, and is an unknown parameter. (v_x, v_y) is the optical flow at the point (x, y) . The distance Z can be determined from Eq. (6) using the x or y components of the optical flow, but the scale factor, T , is indefinite. Figure 8 shows examples of the estimation of distance from the x and y components of the optical flow. In this figure, changes in distance are rendered as changes in shading, with brighter regions representing closer distances, and darker regions further distances. Pitch-black represents regions with no calculation result. While it is possible to estimate distance using either component, there is a significant error in the vertical line passing through the FoE obtained when the x component is used and in the horizontal line passing through the FoE obtained

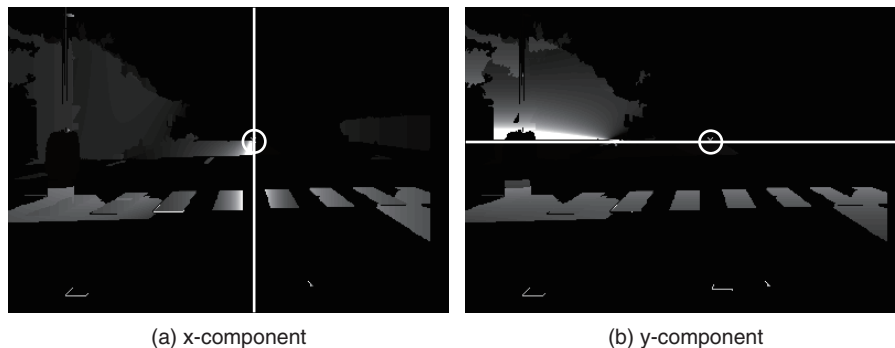


Fig. 8 Intermediate motion disparity computation results

when the y component is used. The circles in the figure show the position of the FoE. This error occurs because of the small values of each component of the optical flow. The orientation and magnitude of the optical flow can be used to minimize this error. However, making use of the fact that the errors occur in specific regions of the image, for the sake of simplicity error is minimized by taking a weighted average of two distances in order to reduce the weight of the regions in which the error is greatest for each component. The distance $Z(x, y)$ at point (x, y) in the image is calculated as follows.

$$Z(x, y) = w_x(x, y)Z_x(x, y) + w_y(x, y)Z_y(x, y) \quad (7)$$

Here, $w_x(x, y)$ and $w_y(x, y)$ are weights at position (x, y) in the image, set such that $w_x(x, y) + w_y(x, y) = 1$. $Z_x(x, y)$ and $Z_y(x, y)$ are the distances at (x, y) estimated from the x and y components of the optical flow, respectively. Figure 9 shows the distribution of the weights and the distance map obtained by weighted averages. The significant errors that occurred in the vertical and horizontal lines passing through the FoE in the case of distance maps calculated using each component, as shown in Fig. 8, have been minimized in the distance maps shown in Fig. 9.

2.4.3. Identification of moving regions

This sub-section will discuss the identification of moving regions in the “Moving object detection” process in the processing flow shown in Fig. 7, using the distance derived from motion in the “Motion disparity computation” process and the distance derived from the high-accuracy stereo disparity in the “Disparity model fitting” process.

Because the scale factor is unknown for distance estimated from motion, it cannot be directly compared with distance estimated from stereo disparity. Based on the assumption that the majority of the regions in the scene shown in the images belong to stationary objects, the

unknown scale factor, T , is calculated as follows.

$$T = \text{median}_{(x,y)} \left(\frac{D_s(x, y)}{D_m(x, y)} \right) \quad (8)$$

Here, $D_s(x, y)$ and $D_m(x, y)$ are the distances at position (x, y) in the images as derived from stereo disparity and motion, respectively. Medians are calculated for all points for which D_s and D_m are valid. Moving regions can be detected by identifying the pixels that satisfy the following condition.

$$|D_s - T \cdot D_m| > \theta \quad (9)$$

Here, θ is a positive threshold.

3. Experiments

Employing image segmentation technology, the method discussed here has increased performance in the estimation of optical flows and stereo disparity against the conventional block matching method. This has made it possible to detect the boundaries of objects during the process of moving object detection. This chapter will consider the results of experiments concerning optical flow and stereo disparity estimation performance using image segmentation, after which the results of moving object detection will be discussed.

3.1. Accuracy of Estimation of Optical Flows using Image Segmentation

The increase in the accuracy of estimation of optical flows against the initial optical flows due to the use of image segmentation was quantitatively evaluated. Here, optical flows obtained using image segmentation refer to optical flows calculated for each pixel using the model parameters obtained from the process of motion model fitting discussed

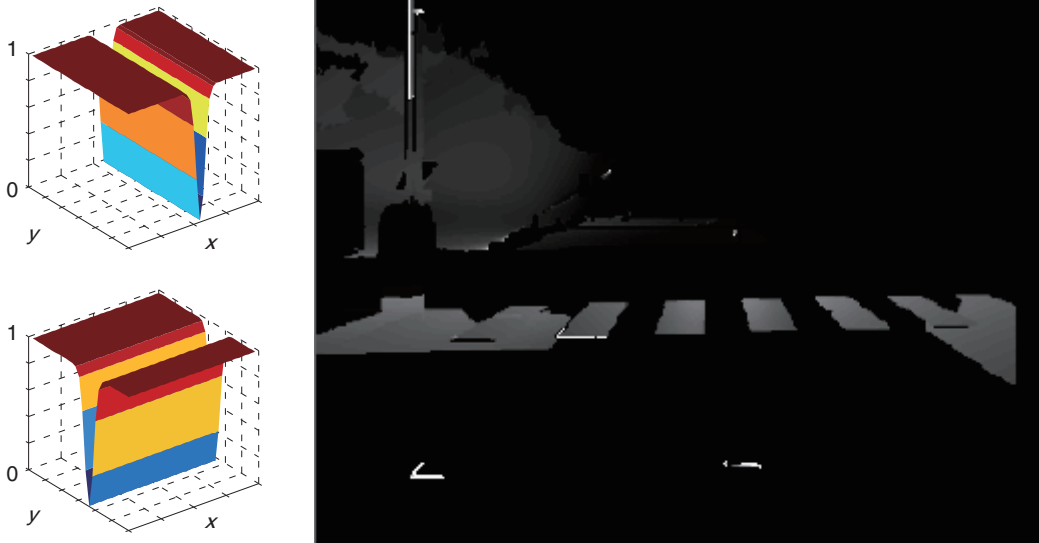


Fig. 9 Weight maps and motion disparity computation result

Table 1 Error in segmentation-based optical flow estimation

Average angular error [deg]	Sequence No.1	Sequence No.2	Sequence No.3
Initial optical flow (Block matching)	11.25 (112330)	15.57 (54162)	10.84 (81210)
Segmentation-based optical flow	8.64 (219010)	9.32 (153671)	4.60 (176364)

Average end-point error [pixel]	Sequence No.1	Sequence No.2	Sequence No.3
Initial optical flow (Block matching)	0.17 (112330)	0.22 (54162)	0.17 (81210)
Segmentation-based optical flow	0.13 (219010)	0.14 (153671)	0.07 (176364)

above. Table 1 shows the angular error and the endpoint error for initial optical flows obtained using block matching and optical flows obtained using image segmentation for each image sequence, as determined using a Middlebury data set⁽¹⁰⁾. Endpoint error is error in the length of the vectors that express the optical flows. The numbers in Table 1 represent angular error and the endpoint error in degrees and numbers of pixels, respectively. In certain cases, valid optical flows could not be obtained by block matching due to the existence of textureless regions, or could not be obtained by the image segmentation-based method due to a high degree of model fitting error. The numbers in parentheses in Table 1 show the number of pixels for which valid optical flows were obtained in each image sequence. The figures for errors in Table 1 are average values for the number of pixels in the parentheses. As the figures for error in the table make clear, estimation of optical flows based on image segmentation helps to enable more accurate optical flows to be obtained than the conventional method using block matching. In addition, the fact that there are a high number of valid

pixels indicates that it was possible to supplement the optical flows in textureless regions. This demonstrates that the initial optical flows calculated during the “Optical flow computation” process in Figs. 4 and 7 displayed increased accuracy following the “Motion model fitting” process shown in the same figures. The estimation of optical flows using image segmentation has increased the sharpness of the boundaries of motion, and has thus helped to enable clearer boundaries to be determined for objects in the images.

3.2. Accuracy of Estimation of Stereo Disparity using Image Segmentation

Figure 10 shows examples of an initial stereo disparity map obtained using block matching and a stereo disparity map obtained using image segmentation. Here, the stereo disparity based on image segmentation is disparity calculated for each pixel using the model parameters obtained during the disparity model fitting process discussed above. Figures 10(a) and (b) show the left and right images from the stereo cameras, and (c) shows the result of color

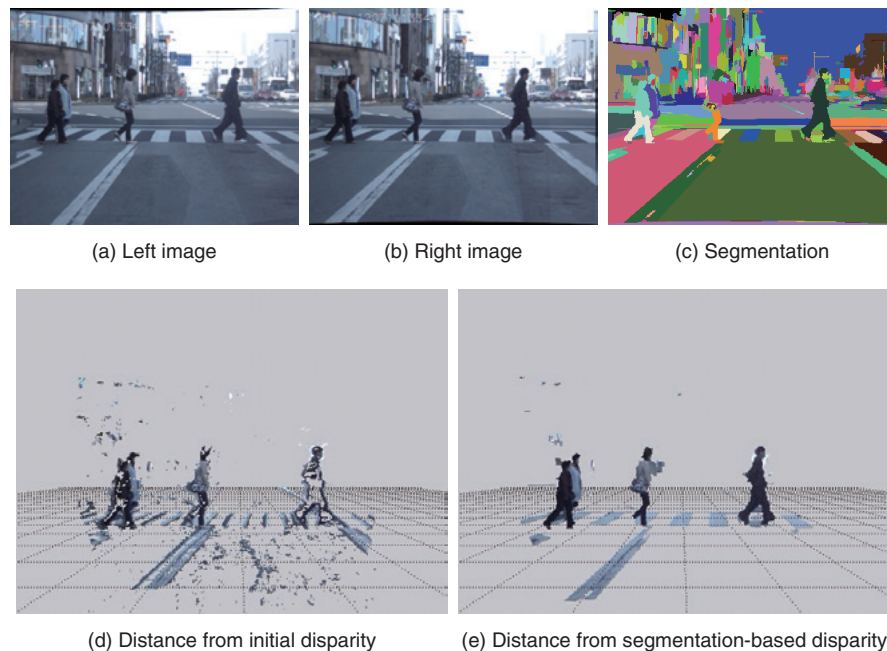


Fig. 10 Comparison of initial and segmentation-based disparity

segmentation of the right image. Figure 10(d) shows the results of estimation of the initial stereo disparity from block matching, and Fig. 10(e) shows the results of estimation of the stereo disparity from image segmentation. There is a high level of noise in the initial stereo disparity. As the torsos of the pedestrians indicate in particular, block matching is unable to measure disparity in textureless regions. In the image segmentation-based method, the use of a disparity model makes it possible to supplement the textureless regions, and the reduction of the effect of noise sharpens the boundaries of objects. This indicates that the initial stereo disparity calculated in the “Stereo disparity computation” process shown in Fig. 7 has increased in accuracy following the “Disparity model fitting” process shown in the same figure. Because there are few parameters and fitting is simple

for models in the case of stereo disparity, a quantitative evaluation like the one for the optical flows was not conducted, but in other scenes also, as in the case shown in Fig. 10, the use of the segmentation-based method helped to enable the achievement of better results for distance estimation than conventional block matching.

3.3. Experiment for Detection of Intersecting Objects

The method of detection of intersecting objects discussed above was tested using images collected on ordinary roads. The experiment used a scene in which the subject vehicle was driving towards a pedestrian crossing the road in a pedestrian crossing. Sufficient distance was maintained, and the images were captured under normal safety conditions. Figure 11 shows the results of detection

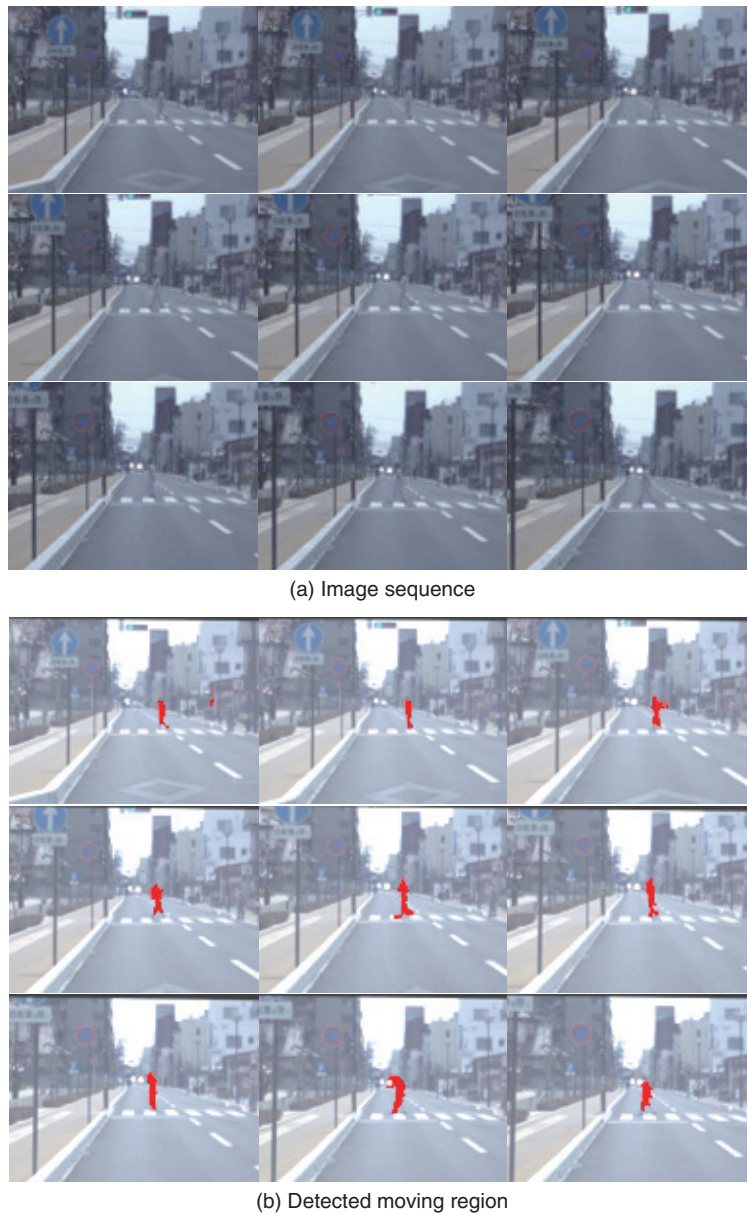


Fig. 11 Detection results for intersecting object

of the pedestrian. This figure shows the results of detection for each frame captured. The captured images are shown in Fig. 11(a). In the first frame, the pedestrian begins to enter the path of the subject vehicle, and reaches the center of the vehicle's path in the final frame. The regions painted out in Fig. 11(b) are the regions detected as belonging to a moving object. The pedestrian has been detected with no major instances of misdetection, and the boundaries of the pedestrian have also largely been correctly detected. For the sake of simplicity in this experiment, the region of the moving object was judged from horizontal motion only, and the moving objects that could be detected were limited to objects moving towards the FoE in the image. In addition,

in order to reduce error, post-processing that judged whether or not an object was moving based on size and position in the image was also employed.

3.4. Experiment for Detection of Preceding Objects

Next, the method of detection of preceding objects was applied to a scene captured on an expressway. In the scene used in this experiment, an overtaking vehicle was present in the lane next to the subject vehicle, and there was also another vehicle in front of the subject vehicle (Fig. 12). Figure 12(a) shows every eighth frame from the recorded images. Regions detected as belonging to moving objects are painted out in Fig. 12(b). The method

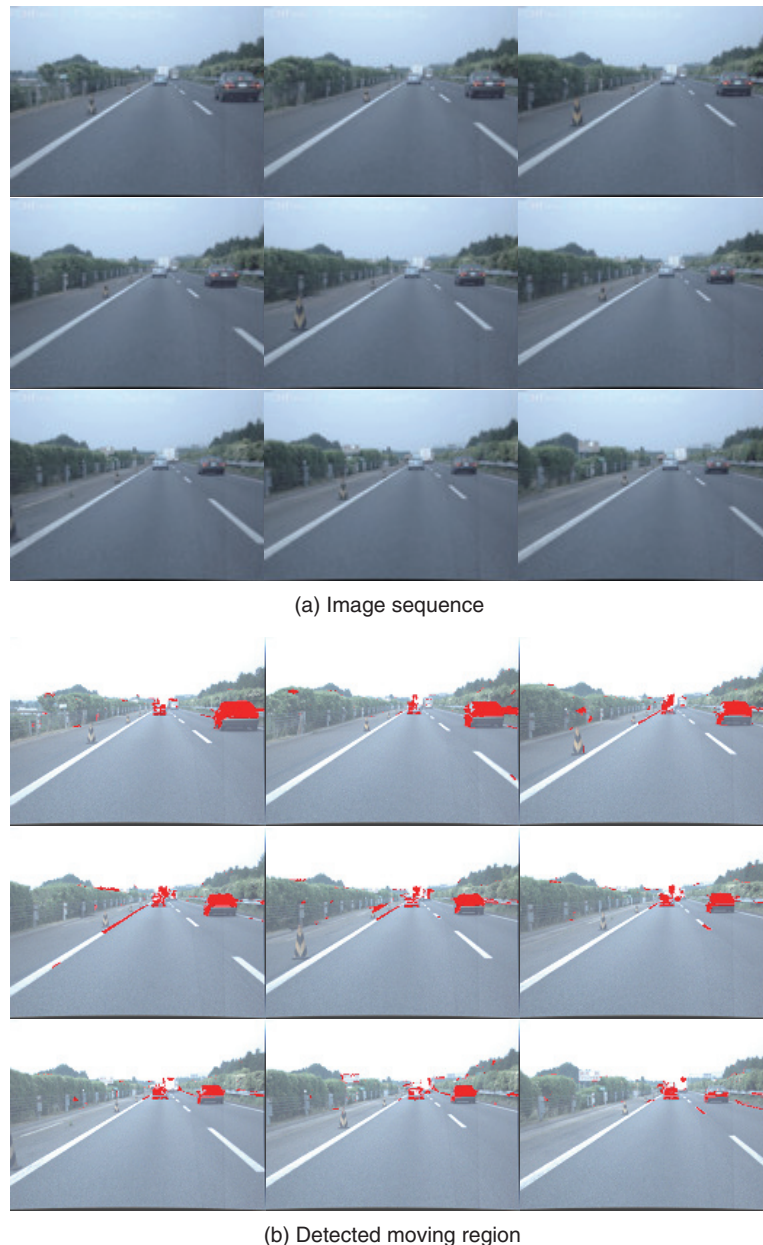


Fig. 12 Detection results for preceding object

employed made it possible to detect both the preceding vehicle and the vehicle in the next lane. In addition, other than in the shadowed areas close to the road surface, the method was able to detect the boundaries of the overtaking vehicle. In this experiment, judgment was made regarding the presence of moving objects for each frame and each pixel, and no post-processing to reduce misdetection was applied. Some misdetection can be observed, but it occurs intermittently, and it would be possible to reduce it through the employment of post-processing such as clustering or tracking.

4. Conclusion

Two methods of detecting independently moving objects from images captured by cameras mounted on vehicles were developed. The first of these algorithms is based on the use of a single camera, and can be employed when the camera and the object for detection (for example a pedestrian crossing the road or an intersecting vehicle) are moving in different directions. The second algorithm is based on the use of stereo cameras. This makes the system more complex, but enables detection of more general moving objects, including preceding objects, by comparing distance derived from stereo disparity and distance derived from motion. Experiments showed that by using this method, it was possible to obtain sharp boundaries for a moving object, and that as a result it was possible to detect the moving object.

The programs necessitate several tens of seconds of computation time for the processing of each frame. The reduction of this processing time will be an issue for the future. In addition, when employing color segmentation in the calculation of optical flows and stereo disparity, it has been assumed that the boundaries of the segments include the boundaries of the objects. However, there are cases in which misdetection or non-detection may occur when this assumption does not hold. Resolving this will also represent an issue for the future.

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Development of High-paintability Waterborne Primer Paint

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ABSTRACT

Waterborne primer paint is easily affected by temperature and has to be applied under certain temperature conditions, and so in order to save energy there is a need to expand the range of painting temperatures. To expand the range of painting temperatures, painting resin compositions were studied that can control changes in paint viscosity in relation to changes in paint solid content. By increasing the ratio of hydrophilic component of paint resin, it was discovered that the interaction of resin in the low solid-content range directly after application was strengthened and high viscosity was achieved, while resin phase transformation in the high solid-content range prior to drying was mitigated and low viscosity achieved, thus controlling changes in paint viscosity. This technique was used to develop paints, with the result that the range of painting temperatures, which was $\pm 3^{\circ}\text{C}$ with conventional paints, was expanded to $\pm 10^{\circ}\text{C}$.

1. Introduction

As Fig. 1 shows, the structure of automotive body paint consists of four films: e-coat, primer, basecoat and clearcoat. Of these, primer paint has three functions: providing a paint appearance, imparting chipping resistance and imparting weather resistance. As such, it plays a very important role in terms of the marketability and durability of the automotive body paint.

To prevent air pollution, the industry has shifted from solvent-based primer paints to waterborne primer paints in order to reduce emissions of volatile organic compounds.

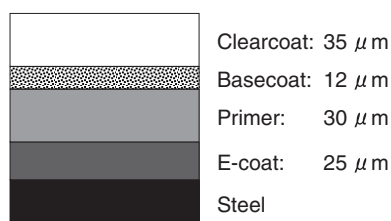


Fig. 1 Film structure of automotive body paint

However, waterborne primer paint is more easily affected than solvent-based primer paint by paint booth climate-control temperature and the coating object's temperature, so painting is done under controlled conditions in a narrow temperature range of $\pm 3^{\circ}\text{C}$. If waterborne primer is applied outside the current temperature range, sagging is likely on the low-temperature side, and poorer paint appearance is likely on the high-temperature side, leading to a loss of paint marketability under either condition. Therefore large-scale climate-control equipment is necessary in the paint booth and storage before the booth (Fig. 2), which increases energy consumption. Trial calculations indicate that energy consumption from climate control in a waterborne paint booth may be about 60% higher as compared to a case using solvent-based primer⁽¹⁾.

What is needed to save energy is a waterborne primer paint that can be applied over a wide temperature range similar to that allowed by solvent-based primer, without sacrificing paintability. Because paint's application properties vary widely according to changes in the coating object's temperature, painting materials were developed with a focus on coating object temperature.

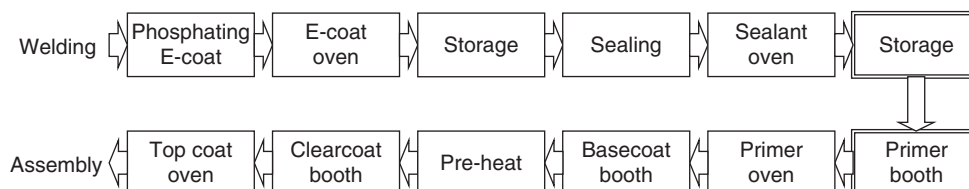


Fig. 2 Outline of body paint process

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2. Current Issues

Because solvent-based primer paints use organic solvents as thinner, one can control how much of these solvents is volatilized during painting, even if the temperatures of the paint booth and car body vary, by carefully selecting the type of organic solvent. This enables one to maintain a uniform application solid content in the paint at a wide temperature range, thus making it easy to do painting work and adjust paint appearance. Therefore, painting processes that use solvent-based paints allow a wide range of tolerable temperatures of paint booths and of car bodies, which are affected by ambient temperature and dryer heat exhaust.

Figure 3 shows body temperature changes in a condition that the temperature is not adjusted in storage before the primer paint booth. When car bodies are put inside the primer paint booth, their temperatures vary over a range of 20°C, from a low of 18°C in winter to a high of 38°C in summer.

Waterborne primer paints, on the other hand, use water as thinner, which takes greater evaporation energy than organic solvents and therefore evaporates less at low temperatures. Additionally, since there is not a choice of type as there is with organic solvents, one cannot control water evaporation at high temperatures, so it is challenging to control the paint's application solid content. Therefore, it is necessary to control water evaporation by controlling the coating object's temperature within a narrow range during painting and thereby maintain a uniform application solid content.

Figure 4 shows the results of altering the coating object's temperature and confirming changes in a current waterborne primer paint's application solid content, between one minute after spraying and five minutes after, when drying starts.

The figure shows that the difference in application solid content at 23°C - 28°C (the current waterborne primer paint's range of painting temperatures) has been minimized. When the range is expanded to 18°C - 38°C (similar to the painting temperature range of solvent-based primer), the application solid content is low at 18°C

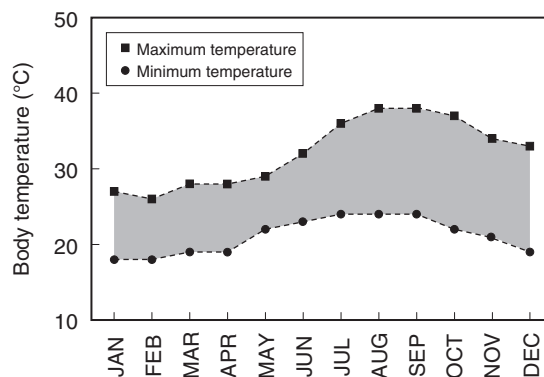


Fig. 3 Annual body temperature

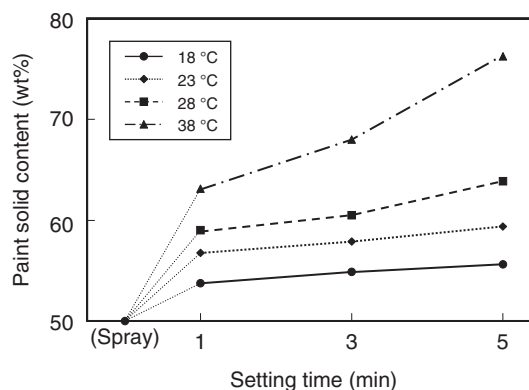


Fig. 4 Paint solid content versus setting time

immediately after painting, and there is subsequently little change in application solid content. Therefore, the state of low viscosity continues, making the paint flow easily and producing paint sagging. At 38°C, the application solid content is high starting immediately after painting, and it rises subsequently, so there is high viscosity and flow is lost; paint appearance is poorer because uneven places that occur on the paint surface during painting do not get smoothed out. Thus, waterborne primer paints do not have a stable application solid content over a wide temperature range and do not allow the marketability that solvent-based primer paints do under the same temperature conditions.

There is also a method of limiting application solid-content changes that result from temperature changes, by mixing paint and water inside the painting machine based on the coating object's monitor temperature⁽²⁾, but this is not considered to provide sufficient adjustment because the application solid-content change is too great at 38°C. Additionally, a method has been considered that alters the relationship between application solid content and paint viscosity by adding a viscosity adjuster to the paint. However, in addition to temperature changes over the course of a year, those over the course of a single day are not insignificant, making paint adjustment and control during the painting process complex and thus disadvantageous in terms of human errors.

To reduce energy consumption from climate control and achieve the same level of paintability as with solvent-based primer paint when there are fluctuating temperatures, viscosity characteristics are needed that minimize changes in viscosity even if there are great changes in application solid content.

3. Materials Development Concept

The following targets were set for materials development, accounting for environmental needs and productivity.

- (1) Be able to reduce temperature controls in the painting process
- (2) Make complex production process control unnecessary

These perspectives were taken in the study as the stance on materials development.

4. Techniques for Achieving the Targets

The relationship between current waterborne primer paint's application solid content and viscosity is shown in Fig. 5. The shaded area in the figure represents the paintable range, in which the minimum viscosity at which sagging does not occur at 18°C (application solid content 55%) and the maximum viscosity that achieves the target paint appearance at 38°C (application solid content 76%) are confirmed in advance. Because the viscosity is insufficient at 18°C and excessive at 38°C, it is necessary to adjust the viscosity for both low-application solid content and high-application solid content.

Since it uses minute particle hydrophobic resin as the main resin, waterborne primer paint has two mechanisms for increasing viscosity. One mechanism for increasing viscosity is resin particle interaction at low-application solid content (below 65%), while the other is resin phase transformation at high-application solid content (above 65%). Therefore, the study of techniques for achieving the primer paint was divided into two ranges: the low-application solid-content range and high-application solid-content range.

4.1. Increasing Viscosity in the Low-application Solid-content Range

Particulate hydrophobic resin is not uniformly distributed in paint, but rather is unevenly distributed in particulate form, and therefore when in low-application solid-content state the particles are separated from each other, the interaction is weak and viscosity is low. However, as the water evaporates and the application solid content increases, the particles draw closer to each other, stronger interaction occurs and viscosity increases (Fig. 6(a)). To help ensure sagging resistance, a method of increasing viscosity with a viscosity adjuster has been used⁽³⁾ as a technique to supplement particle interaction,

so that application solid content is increased by about 5% and sufficient viscosity is achieved. However, if the low-application solid-content viscosity is increased by viscosity adjuster, the viscosity of the high-application solid-content range discussed next is simultaneously increased, so it is challenging to get a viscosity balance between the low- and high-application solid-content ranges.

The optimization of hydrophilic resin was considered as a technique for increasing resin interaction to increase viscosity in the low-application solid-content range. Since hydrophilic resin dissolves in water and is spread out in linear form, it keeps a close distance to particulate hydrophobic resin, and it is expected this will encourage resin interaction, increasing viscosity in low-application solid content (Fig. 6(b)).

4.2. Decreasing Viscosity in the High-application Solid-content Range

An increase in the viscosity in the high-application solid-content range is the result of the main component of the paint transforming from an aqueous phase to an oil phase (phase transformation)⁽⁴⁾. An aqueous phase consists of water and hydrophilic resin dissolved in that water, whereas an oil phase contains hydrophobic resin, which does not dissolve in water, dispersed in particulate form. In the low-application solid-content range, the particles of the hydrophobic resin are distributed at low concentration in an aqueous phase. For this reason, the fluidity of the aqueous phase is not inhibited and the paint has a low viscosity, but as the water evaporates, the particle concentration increases, paint fluidity is inhibited and viscosity increases⁽⁵⁾. As more water evaporates, the main component of the paint becomes an oil phase with aqueous phase particles dispersed within it, lowering the viscosity (Fig. 7).

In current waterborne primer paints, only a narrow range of temperature conditions is allowed when painting, which controls increases in application solid content, so there has not been sufficient study of viscosity in the high-application solid-content range. The application of hydrophilic resin has been considered

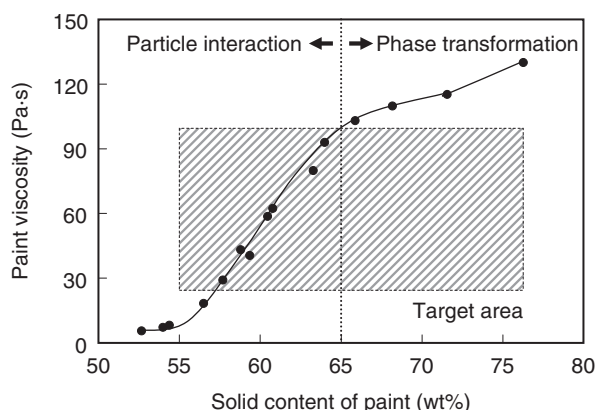


Fig. 5 Current paint viscosity versus solid content

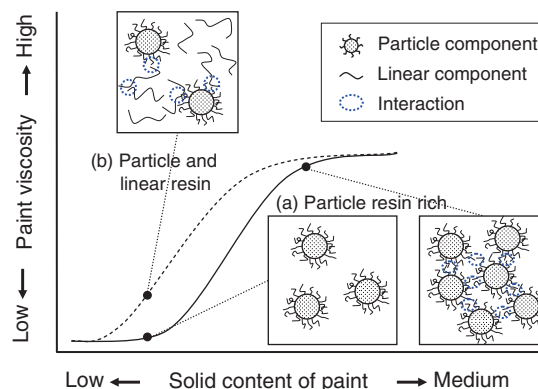


Fig. 6 Interaction between particle resin and linear resin at low solid content

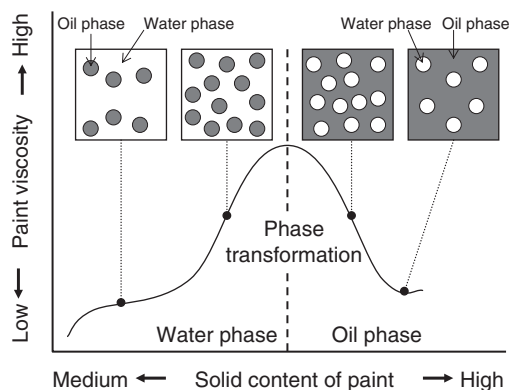


Fig. 7 Phase conversion and paint viscosity at high solid content

as a method for controlling viscosity increase in the high-application solid-content range prior to phase transformation. Hydrophilic resin dissolves in water and does not form particles, so no phase transformation occurs and viscosity is not increased. However, with hydrophilic resin alone, it is impossible to get an increase in viscosity in the low-application solid-content range necessary to ensure sagging resistance.

Because viscosity increase during phase transformation depends on particle concentration, a method was considered that reduces the hydrophobic resin that turns into particles and increases the hydrophilic resin that turns into the aqueous phase component. This makes it possible to reduce the particle component while keeping the amount of resin component uniform, so control over phase transformation viscosity can be expected.

Based on the above, the technique of increasing viscosity at the low-application solid-content range and the technique of decreasing viscosity at the high-application solid-content range coincide with each other, as they both reduce the hydrophobic resin component and increase the hydrophilic resin component of the resin.

5. Results of Viscosity Behavior when Resin Ratio Changed

Current waterborne primer paints consist of two kinds of resin: polyester resin and melamine resin. In the case of both polyester resin and melamine resin, a choice can be made between a hydrophilic and hydrophobic type. Samples of hydrophilic and hydrophobic components of each resin were produced in the ratios shown in Table 1, with the solid content varied and viscosity measured.

5.1. Polyester Resin's Hydrophilic/Hydrophobic Component Ratios

Viscosity was measured using Samples 1-4, which varied the polyester resin's hydrophilic/hydrophobic component ratios from 75:25 to 100:0. The results are given in Fig. 8.

Table 1 Resin formulation of test sample

Sample	Polyester		Melamine	
	Hydrophilic	Hydrophobic	Hydrophilic	Hydrophobic
Sample-1	75	25	—	—
Sample-2	82.5	17.5	—	—
Sample-3	90	10	—	—
Sample-4	100	0	—	—
Sample-5	56.2	18.8	7.5	17.5
Sample-6	56.2	18.8	12.5	12.5
Sample-7	56.2	18.8	16.5	8.5

In the low solid-content range, an increase in viscosity was observed as the polyester resin's ratio of hydrophilic component rose. However, when the hydrophilic component reached 100% (Sample 4), there was no increase in viscosity. The results indicate that in the low solid-content state, in the hydrophobic component that occurs as particles in the paint, there is little interaction, but by increasing the ratio of hydrophilic component, which spreads out in linear form, interaction becomes stronger and viscosity increases. Also, in the case of hydrophilic component alone, there is no strong interaction from hydrophobic component particles, and the interaction in the hydrophilic component is weak, so viscosity does not easily increase. These facts indicate that in the low solid-content range, mixing a hydrophobic component and hydrophilic component with each other and increasing the ratio of hydrophilic component can increase resin interaction and increase viscosity.

In the high solid-content range, it is recognized that increasing the ratio of hydrophilic component will reduce viscosity, but viscosity increases are not completely controlled. It is thought that, although an increase in the ratio of hydrophilic component causes the concentration of particle component to be reduced, the particle component remains, and therefore the point at which viscosity increase begins shifts to the high solid-content side, only to delay the viscosity increase.

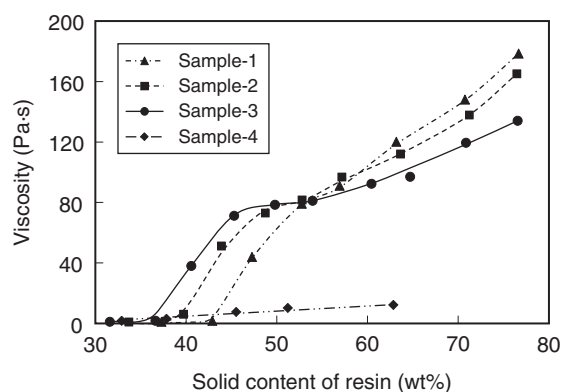


Fig. 8 Resin viscosity when polyester ratio changed

5.2. Melamine Resin Hydrophilic/Hydrophobic Component Ratios

Viscosity was measured using Samples 5-7, which varied the melamine resin hydrophilic/hydrophobic component ratios from 30:70 to 66:34. The results are given in Fig. 9.

In the low solid-content range, there was an increase in viscosity as the ratio of hydrophilic component rose, but the viscosity increase effect was small compared to that of polyester resin. The chemical structure of melamine resin is smaller than that of polyester resin and does not spread out as much in the paint, so it is thought that the viscosity increase effect is smaller because the structure has less influence on interaction between particles with a large distance between them.

In the high solid-content range, increasing the hydrophilic component markedly controls viscosity increase, and in Sample 7, with a hydrophilic/hydrophobic component ratio of 66:34, no viscosity increase was found up to a solid component of 65%. The hydrophobic component of melamine resin is stable in water and does not disperse; it is incorporated into particles that are the hydrophobic component of polyester resin, so it has no impact on particle concentration. However, the hydrophilic component can exist stably in water, so the aqueous phase increases and there is the effect of decreasing particle concentration. Also, the melamine resin's hydrophilic component has a low viscosity of 2.1 Pa·s, so it is thought that the viscosity increase is controlled by the decline in aqueous-phase viscosity.

Following the above, in the low solid-content range, paint viscosity was increased by increasing polyester resin's ratio of hydrophilic component and thereby increasing polyester resin interaction. In addition, in the high solid-content range, it was confirmed that paint viscosity is reduced by increasing melamine resin's ratio of hydrophilic component and thereby lowering particle concentration and lowering aqueous-phase viscosity. It was found that by adjusting the ratio of hydrophilic resin component, it is possible to adjust viscosity in the low and high solid-content ranges.

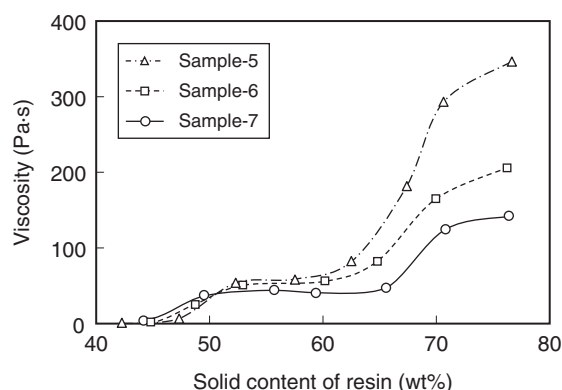


Fig. 9 Resin viscosity when melamine ratio changed

6. Developed Materials

Based on the results of confirming resin viscosity, waterborne primer paint was mixed with a hydrophilic/hydrophobic component ratio of 90:10 with polyester resin and a hydrophilic/hydrophobic component ratio of 66:34 with melamine resin.

The relationship between the developed waterborne primer paint's application solid content and viscosity is shown in Fig. 10. In this graph, the developed paint is represented by the solid line and the current paint by the broken line. Additionally, the shaded area of the figure indicates the region within the upper and lower limits of target viscosity to ensure sagging resistance and paint appearance. It was found that, compared to the current paint, the developed paint can achieve high viscosity in the low-application solid-content range and low viscosity in the high-application solid-content range. Also, whereas the current paint has a narrow-application solid-content range (57%-65%) from minimum to maximum viscosities, the developed paint has a wider range, from 55% (at body temperature 18°C) to 76% (at body temperature 38°C).

Using a test booth under actual line painting conditions, sagging resistance and paint appearance were comparatively demonstrated, using the developed paint and current paint. Sagging resistance was evaluated by applying a film of primer at an incline to panels with $\phi 10$ holes; the panel passed if it had sagging of no more than 5 mm at the bottom of the holes. The film thickness in this case was set as the sagging limit film thickness. Additionally, the paint appearance was evaluated by applying finish paint and subsequently using a Wavescan (BYK) orange peel meter, taking LW of 1.2 mm to 12 mm wavelength as the parameters.

Results of confirming sagging resistance with varied coating object temperatures are shown in Fig. 11. At 18°C, the current paint has a limit film thickness of 26 μm , making it challenging to paint, but the developed paint has outstanding painting properties, with a limit film thickness of 45 μm . Whereas the lowest paintable temperature is 23°C with the current paint, it is 18°C with the developed paint.

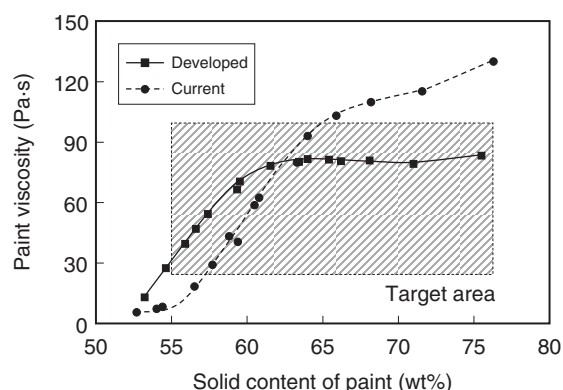


Fig. 10 Developed paint viscosity versus solid content

Next, results of confirming paint appearance with varied coating object temperatures are shown in Fig. 12. The viscosity of the current paint increases at 28°C or more, which makes smoothing properties insufficient, causing paint appearance to quickly diminish. In contrast, the developed paint has little increase in viscosity even if the application solid content is high, meaning that there is little change in paint appearance and an outstanding paint appearance can be achieved even at 38°C. Comparing the highest paintable temperature in terms of paint appearance, it is 28°C or below with the current paint, whereas the developed paint allows painting even at 38°C.

Based on the above, while it was necessary to control the current paint within a narrow temperature range in order to ensure paintability and paint appearance, it was found that the developed paint could be applied over the same temperature range as solvent-based primer paint. Further, whereas solvent-based primer paint had to be controlled by adjusting with thinning solvent, the developed paint works without any adjustment. Additionally, a check of paintability and paint appearance of the developed paint on an actual line produced good results that were similar to those in a test booth.

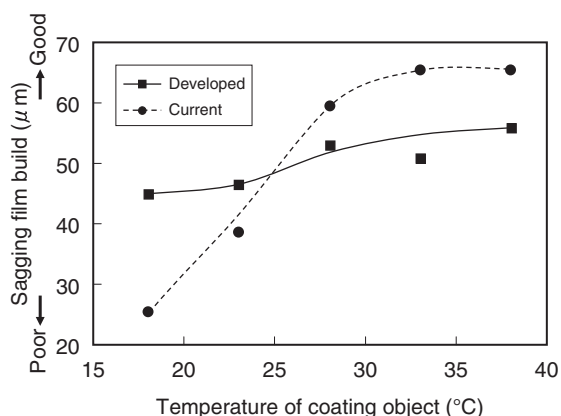


Fig. 11 Sagging film build versus temperature of coating object

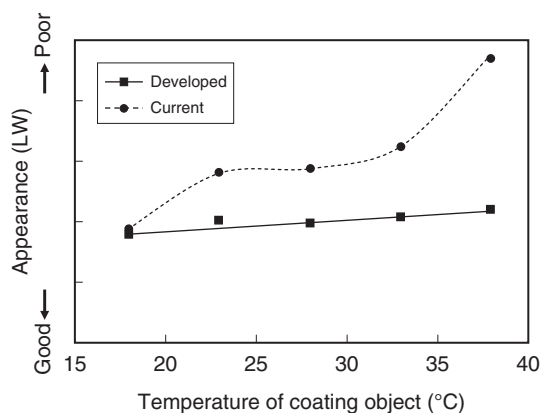


Fig. 12 Appearance versus temperature of coating object

7. Conclusion

New material was developed to ensure paintability and paint appearance of a waterborne primer paint over a wide temperature range by focusing on the ratio of hydrophilic component and hydrophobic component in polyester resin and melamine resin.

- (1) It was attempted to increase resin interaction and increase viscosity with low-application solid content and increase sagging resistance at low temperatures by increasing the ratio of hydrophilic component of polyester resin in the paint.
- (2) It was attempted to mitigate phase transformation and reduce viscosity with high-application solid content and increase paint appearance at high temperatures by increasing the ratio of hydrophilic component of melamine resin in the paint.

As a result, the temperature range of painting conditions could be expanded from the former $\pm 3^{\circ}\text{C}$ to $\pm 10^{\circ}\text{C}$. This made it possible to reduce energy consumption needed to keep the coating object at a constant temperature and reduce process control over temperature fluctuations. Additionally, since this technology can be widely applied to waterborne paints, not just waterborne primer paints, it is thought that expanding the application of the technology to basecoat, which increasingly uses waterborne paint, can further save energy in painting processes.

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Satoru IHARA

Enhancement of Vibration Analysis Accuracy of Engine Structure by Considering Nonlinear Properties between Components

Naoki KAMIMURA*

ABSTRACT

A high-accuracy and generally applicable method of predicting the frequency response functions of engine structures up to the radiated noise range has been developed.

Because the joints between engine parts possess nonlinear properties, as exemplified by the head gasket, analytic models that are linearized in relation to the state of the joints are essential in conducting frequency response analyses. The analysis discussed in this paper was divided into two stages. In the first stage, pressure data for the joint sections was obtained by means of nonlinear contact analyses. In the frequency response analyses conducted in the second stage, accuracy was increased by defining the stiffness and damping of the joint sections as linear properties based on the pressure data, and formulating vibration analysis models that considered strain due to bolt axial force.

Some properties of the joint sections cannot be determined in materials tests, and pressure conversion tables assumed to express properties dependent on pressure were therefore defined. Uncertain values were identified on the basis of test results for the base engine. The application of this method to another engine demonstrated that the frequency response analysis achieved a level of accuracy that represented no impediment to practical use, and indicated that the method possessed general applicability for frequency response analyses of engine structures.

1. Introduction

Today, demands for the realization of enhanced environmental performance from automobiles are increasing. In order to respond to these demands, automakers are seeking to realize weight-savings at the level of engine parts, in addition to proceeding with the development of combustion control that involves turbocharging. At the same time, demands for the achievement of greater commercial appeal in terms of noise and vibration performance are also increasing.

In many cases, the reduction of the weight of engine structures increases vibration response sensitivity, and turbocharging and the realization of high-speed combustion increase exciting force. Because of this, balancing the achievement of increased commercial appeal through enhanced noise and vibration performance with environmental performance is becoming increasingly challenging.

To further complicate the situation, demands for the reduction of the engine development period are also

increasing, and the formulation of technologies for applying the finite element method (FEM), which makes it possible to analyze frequency response with a high level of accuracy using the CAD models only, is essential in order to help enable the development of engine configurations able to realize a low vibration response at a reduced weight.

In order to analyze the frequency response of engine assemblies composed of a large number of parts with a high degree of accuracy, it is necessary to correctly model the stiffness and the damping of the joints between the parts. Because the head gasket and other parts display nonlinear elastic properties of the joints between the parts, the frequency response functions (FRF) of engine assemblies change depending on the bolt axial force. Figure 1 shows changes in the FRF during a test in which the cylinder head and the block were tightened with different bolt axial forces. The surface of the external wall of the cylinder block was the exciting point, and the surface of the cylinder head combustion chamber was the response point.

In the high-frequency range, the differences in FRF due to the bolt axial force become conspicuous, demonstrating

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the importance of considering nonlinear elastic properties in relation to the bolt axial force.

It has also been known that even in areas of direct contact between metal surfaces, the effective joint region changes due to the bolt axial force, and differences occur in the peak frequencies and peak levels of FRF. Therefore, accurate modeling of the stiffness and the damping of the joints between parts is essential.

Reports have previously been published concerning analyses that take the effective joint region of the bolt joints into consideration^{(1), (2)}. However, there has been insufficient discussion of the damping occurring in the joints, and in many cases the models employ modal damping for the entire structure. As an example, Fig. 2 shows the results of an analysis of an engine assembly. The same models as were employed in the development method discussed in this paper were used for the main structural parts model and the bolt model, and past empirical values were used for the modal damping of the entire structure. The camshaft drive chain-side bearing was the exciting point and the surface of the external wall of the lower block was the response point.

Given the significant deviation in the peak frequency close to 2 kHz and the significant deviation in the response level due to the fact that there is excessive modal damping, it cannot be claimed that this method achieves sufficient analytic accuracy.

This paper discusses a method of increasing the accuracy

of prediction of FRF by modeling the stiffness and damping of the joint sections based on pressure data for the sections in contact obtained from nonlinear contact analyses. Taking radiated noise into consideration, the range up to 3 kHz in an assembled state was set as the frequency range for the analysis.

2. Development Method

The analysis included the cylinder block, the lower block, and the cylinder heads, representing the main structure of the engine, and the oil pan and chain case, the parts that radiate noise.

In the engine configuration studied, joint sections with different nonlinear properties exist between the parts (head gaskets, metal contact sections, and formed in place gaskets (FIPG) sections). Consequently, joint elements were set between the parts and their nonlinear elastic properties were defined. Figure 3 shows the composition of models.

Regarding the structure in an assembled condition as displaying linear behavior and conducting a frequency response analysis is a practical method of determining the FRF within the minute amplitude range of engine noise and vibration phenomena. In addition, the modal method represents an advantage over the direct method in terms of calculation efficiency in determining the FRF up to the high-frequency range, in which response peaks of multiple

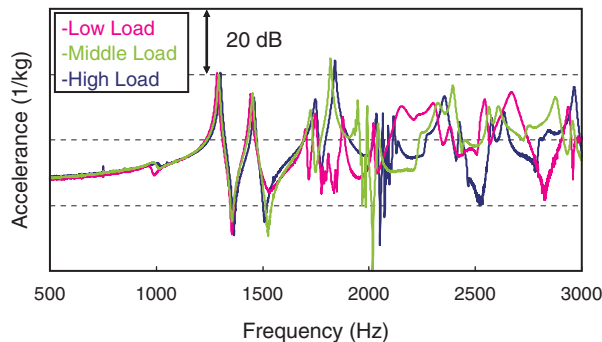


Fig. 1 Experimental FRF under different bolt axial force conditions

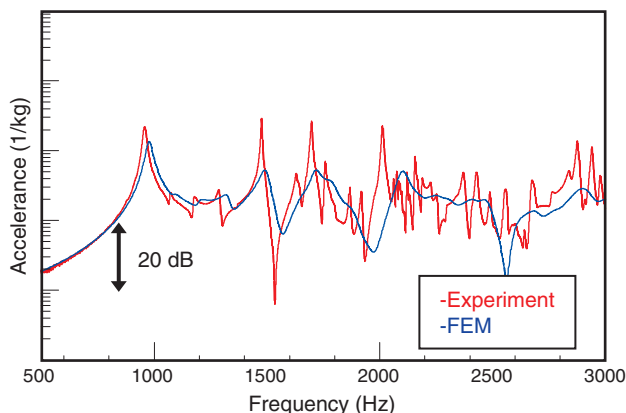
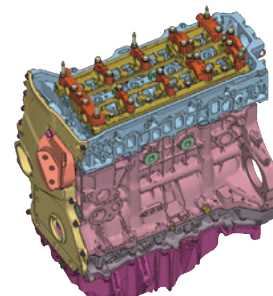
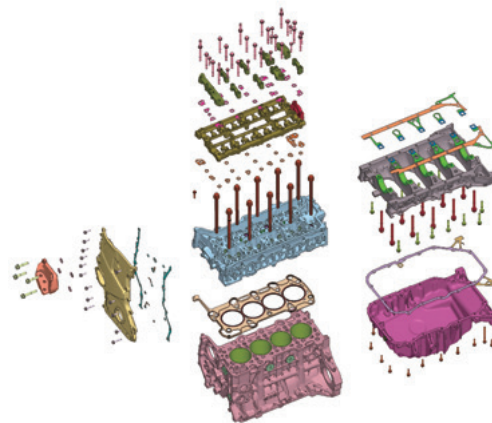


Fig. 2 FRF calculated by previous method



(a) Assembled condition

Scale of model
Nodes : 1288357
Elements : 726888



(b) Disconnected condition

Fig. 3 Composition of FE models

eigen frequencies are observed. In order to apply the modal method, it is necessary to define the coupling state between parts as stiffness rather than force.

Given these conditions, a two-stage analytic method was employed, in which a nonlinear contact analysis was conducted, following which a modal frequency response analysis was conducted with the stiffness of the joint elements due to contact pressure and the damping occurring at the contact boundaries defined as linear properties of the materials.

2.1. Hypothetical Stiffness and Damping Properties of Joint Elements

Because some properties cannot be determined in unit tests using test pieces and it is not possible to unambiguously define the joint elements as displaying linear properties, the following method was proposed.

The Young's modulus of the joint elements is a property that can basically be calculated from the nonlinear elastic properties determined in unit tests using test pieces and the thickness of the elements. However, it is believed that in a state of low contact pressure, nonlinear properties are observed even with vibrations of minute amplitudes.

The shear modulus and the damping coefficient are properties that cannot be determined in unit tests.

The conversion tables shown in Fig. 4 were introduced to define the properties of the joint elements, including properties displaying this type of uncertainty, as linear properties in response to contact pressure. The blue solid lines show certain values for properties as determined in unit tests, while the red broken lines show uncertain values for properties.

(1) Young's modulus

The conversion table shown in Fig. 4(a) was defined to show the Young's modulus of the head gasket in a clamped state, calculated from the nonlinear elastic properties and the thickness of the elements, as a linear property in relation to contact pressure.

It is believed that in a state of high contact pressure, the metal contact sections and the FIPG sections display the same Young's moduli as the base metal materials, but that in a state of low contact pressure, they are nonlinear and display lower Young's moduli. The conversion table in Fig. 4(b) was defined to linearize their properties. In addition, the metal contact sections and the FIPG sections were separately defined because the latter display differing properties due to the thickness of the FIPG film.

(2) Shear modulus

It is believed that in a state of high contact pressure, shear moduli are identical to those of the base metal material, but in a state of low contact pressure, the shear moduli are lower than those of the base metal. In addition, it is believed that infinitesimal slippages occur in a state of low contact pressure, and because of this the shear moduli are underestimated compared to the Young's moduli. Given this, the conversion table shown in Fig. 4(c) was defined, showing the shear moduli as exponential functions in relation to pressure.

(3) Damping coefficient

Most damping between parts is believed to originate in minute relative displacements between the parts when they vibrate. It is believed that in a state of low contact pressure the relative displacement will be high and a high level of damping will be produced, while in a state of high contact pressure the level of damping will be low due to the low

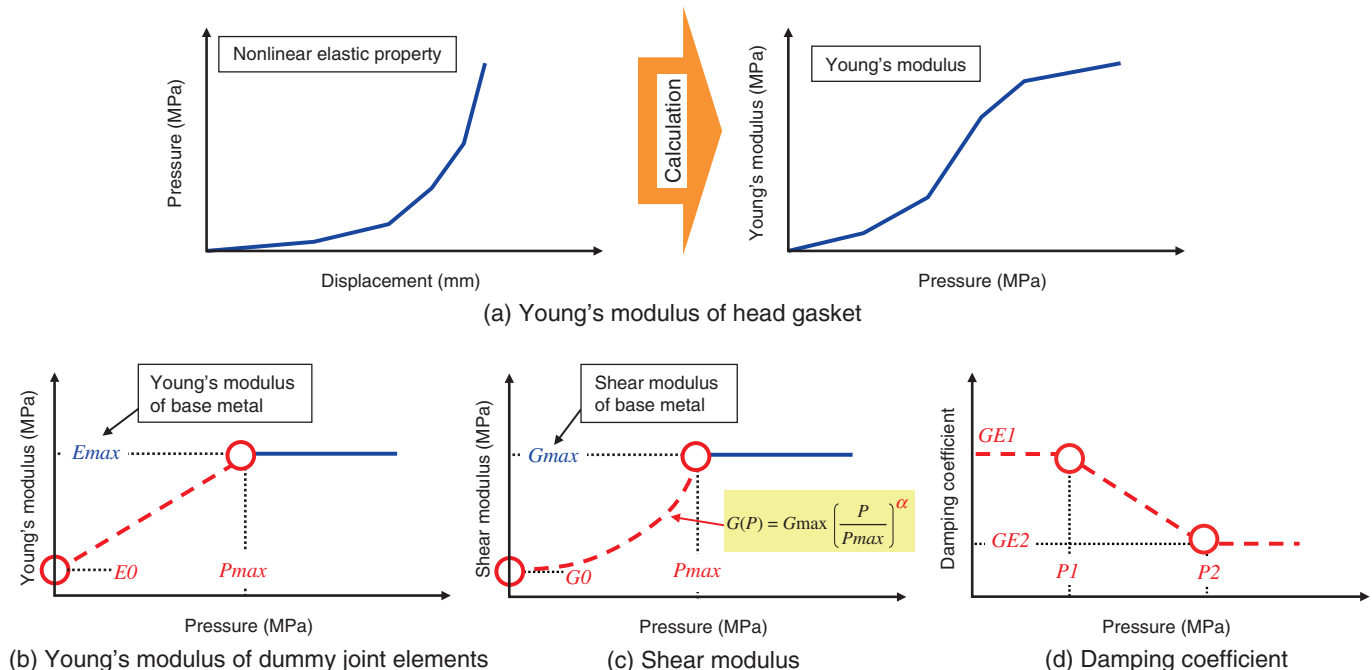


Fig. 4 Conversion tables of joint elements

relative displacement between the parts. The conversion table shown in Fig. 4(d) was defined on this basis.

2.2. Identification of Hypothesized Properties of Joint Elements

The uncertain parameters were identified so that the level of deviation from FRF as determined in excitation tests on multiple engine assemblies was minimized.

3. Nonlinear Contact Analysis

This chapter will discuss the nonlinear contact analysis (MD/NASTRAN SOL600) conducted in the first stage of the analysis.

3.1. Composition of Models

The analytic methodology was formulated with the diesel engine employed in the 2004 model ACCORD as the subject of the analysis. Here, the details of the previously mentioned structural model (Fig. 3) will be discussed.

The nonlinear contact analysis model is composed of a model of the main structural parts, such as the cylinder block; a model of the joint elements, such as the head gasket; and a bolt model.

3.1.1. Model of main structural parts

The accuracy of the modeling of individual parts is dependent on factors including shape, material properties and element size. In order to create models equivalent to parts for which figures had been verified in actual tests, the shapes of the tested parts were faithfully reproduced, and material properties measured using test pieces cut from the tested parts were employed⁽³⁾. Tetrahedral secondary elements were employed. Accuracy increases as the element size is reduced, but this results in an increased calculation load. The element size was therefore optimized by determining convergence.

Figure 5 shows a comparison of test and analysis results for the cylinder block and the oil pan as an example of the verification of the accuracy of FRF in the models of individual parts. 3 kHz was assumed in an assembled state, and the frequency range up to 5 kHz was set as the target for the analysis.

The results demonstrate that it was possible to analyze FRF with a high accuracy.

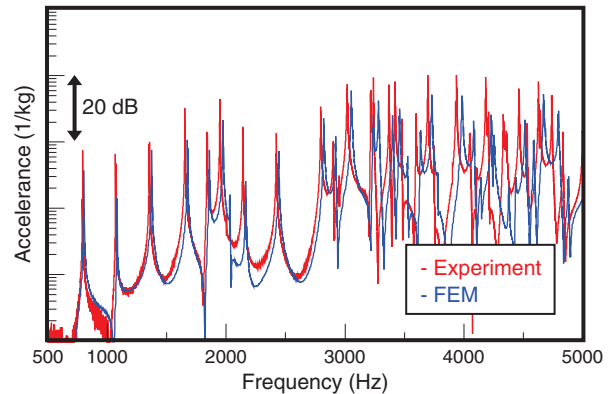
3.1.2. Joint element model

Gasket elements making it possible to define nonlinear elastic properties on one-layered hexagonal elements were employed in SOL600, one of the MD/NASTRAN nonlinear contact analyses. Contact was judged between the nodal points of the gasket elements and the element surfaces of the parts in contact, and the reaction force was calculated in relation to the distance between the nodal points. Because of this, there was no need to consider consistency in the positioning of the nodal points, and the elements also

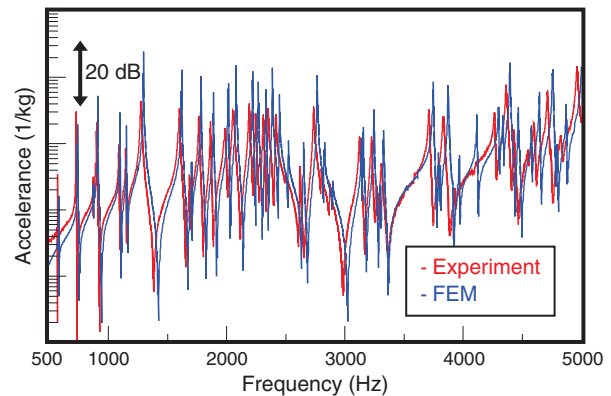
increased convenience in terms of creation and amendment of the models.

The elements employed were approximately one-third the size of the elements used in the model of the main structural parts in order to make it possible to reproduce the pressure distribution on the contact surfaces.

Figure 6 shows the head gasket model. Each part of the head gasket displays differing nonlinear elastic properties due to the bead shapes and the thickness of the structural sheets, and property values were therefore defined on the basis of values obtained from unit tests on test pieces for



(a) FRF of cylinder block



(b) FRF of oilpan

Fig. 5 FRF of component

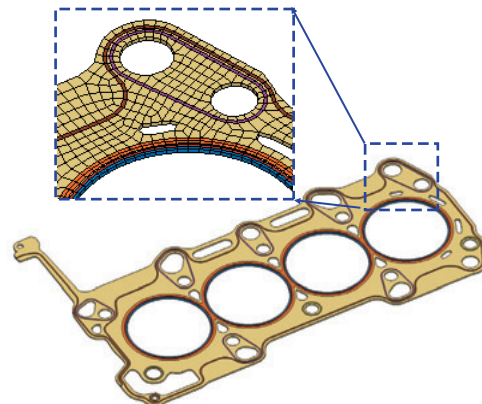


Fig. 6 Head gasket model

each region. As an example of this process, Fig. 7 shows the definition of nonlinear elastic properties for seven regions.

To make it possible to treat the metal contact sections and the FIPG sections in the same way as the head gasket, the hypothetical thin joint elements shown in Fig. 8 were introduced, and defined as gasket elements.

The nonlinear elastic properties were defined on the basis of values calculated from the Young's moduli of the metals in the contacting sections. Because the reactive force of the FIPG due to viscoelasticity was much lower in comparison to the joint stiffness of the main structural parts, it was considered possible to ignore this parameter. As examples, Fig. 9 shows the properties of the cast iron,

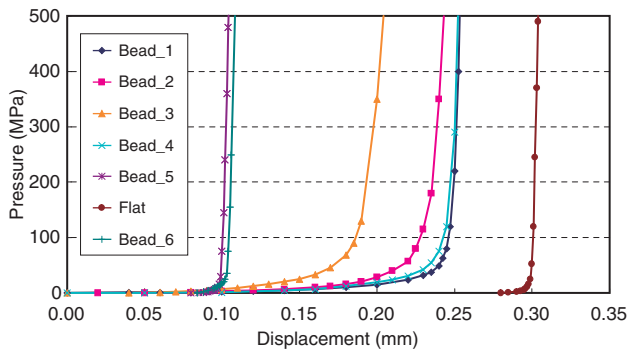


Fig. 7 Elastic properties of head gasket

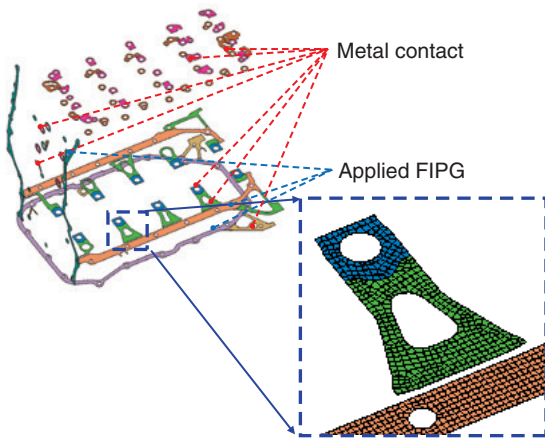


Fig. 8 Joint elements model

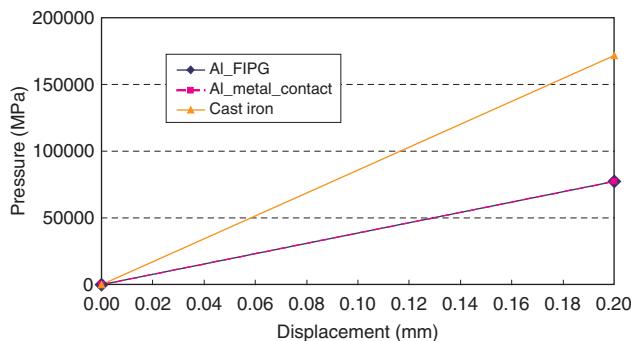


Fig. 9 Elastic properties of joint elements

aluminum, and FIPG sections of the lower block. Because the Young's moduli of the metals are employed, the properties display linear gradients.

3.1.3. Bolt model

In order to correctly reproduce the distribution of mass and help to enable an accurate analysis of the distribution of pressure in the joints, the bolt model employed hexagonal elements that reproduced the shapes of the heads of the bolts (Fig. 10). The not-in-contact sections of the bolt shafts were divided at the middle, and drawing direction values for bolt axial force were imparted to the separated sections.

3.2. Engine Configurations for Nonlinear Contact Analysis

In order to verify the ability of the analytic method to respond to a variety of parts structures, analyses were conducted of the four engine assembly configurations with differing composition of parts shown in Fig. 11.

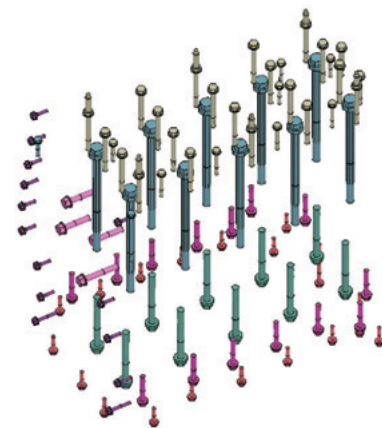


Fig. 10 Bolts model

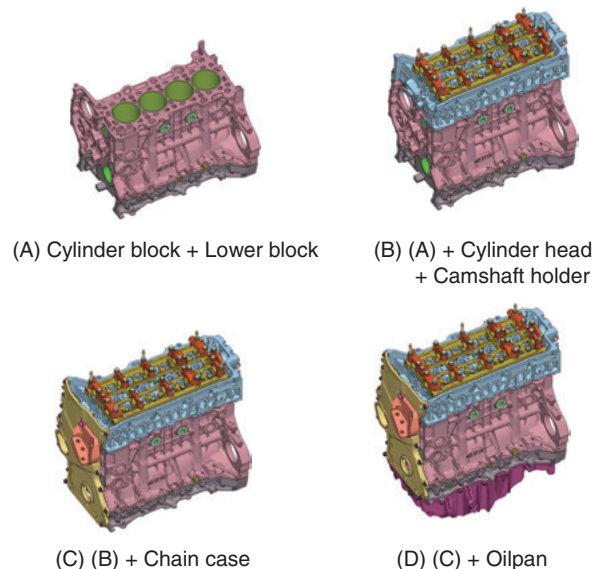


Fig. 11 Assembled models

3.3. Results of Nonlinear Contact Analysis

Figure 12 shows (a) the Mises stress distribution and (b) the contact pressure distribution in the joint elements for assembly (D) as an example of the results obtained from the nonlinear contact analysis.

The contact pressure displays different values in each part due to the effect of factors including the bolt axial force, the stiffness of the part, and the properties of the gasket beads. This demonstrates that nonlinear contact analysis was essential in obtaining this data.

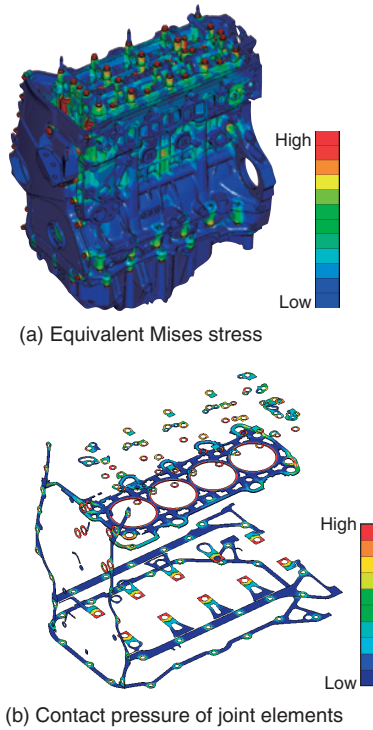


Fig. 12 Nonlinear contact analysis results

4. Frequency Response Analysis

This section will discuss the frequency response analysis (MD/NASTRAN SOL111) conducted in the second stage of the analysis.

The states of connection of the main structural parts and the joint elements and bolts in the nonlinear contact analysis are linked by the force between the nodal points judged to be in contact. In order to maintain the same linkage, the nodal points judged to be in contact were connected by means of multi-point constraints (MPC), and a frequency response analysis was conducted.

Because the eigen frequencies of the bolt shafts depend on the strain due to the axial forces and therefore vary, axial forces were imparted to the bolts in the analysis.

4.1. Identification of Hypothetical Property Values

Based on the concepts discussed in Chapter 2, the uncertain parameters were identified using the four assembly configurations with different parts structures (see Section 3.2) and employing optimization methods, design of experiments, and other mechanisms to minimize deviation between the values for FRF determined in the analysis and the values determined in excitation tests.

The number of parameters which it was necessary to identify is shown as X^* in Fig. 13. The number of parameters to be identified was minimized by considering the values for the head gasket sections, the metal contact sections, and the FIPG sections to be identical.

4.2. Joint Element Properties

Figure 14 shows results for the distribution of property values for the joint elements defined in accordance with the identified conversion tables. Figure 14(a) shows the distribution of the Young's moduli, Fig. 14(b) the

	Young's modulus		Shear modulus			Damping coefficient			
	$E0$	P_{max}	$G0$	P_{max}	a	$GE1$	$P1$	$GE2$	$P2$
Head gasket	—	—	X1	X2	X3	X4	X5	X6	X7
Metal contact	X8	X9	X10	X9	X11	X12	X13	X14	X15
FIPG	X16	X9	X17	X9	X11	X18	X13	X14	X15

Fig. 13 Parameters of identification

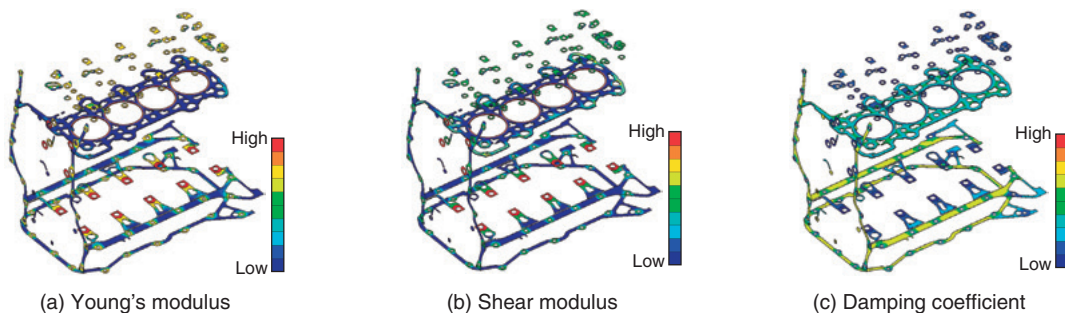


Fig. 14 Properties of joint elements for frequency response analysis

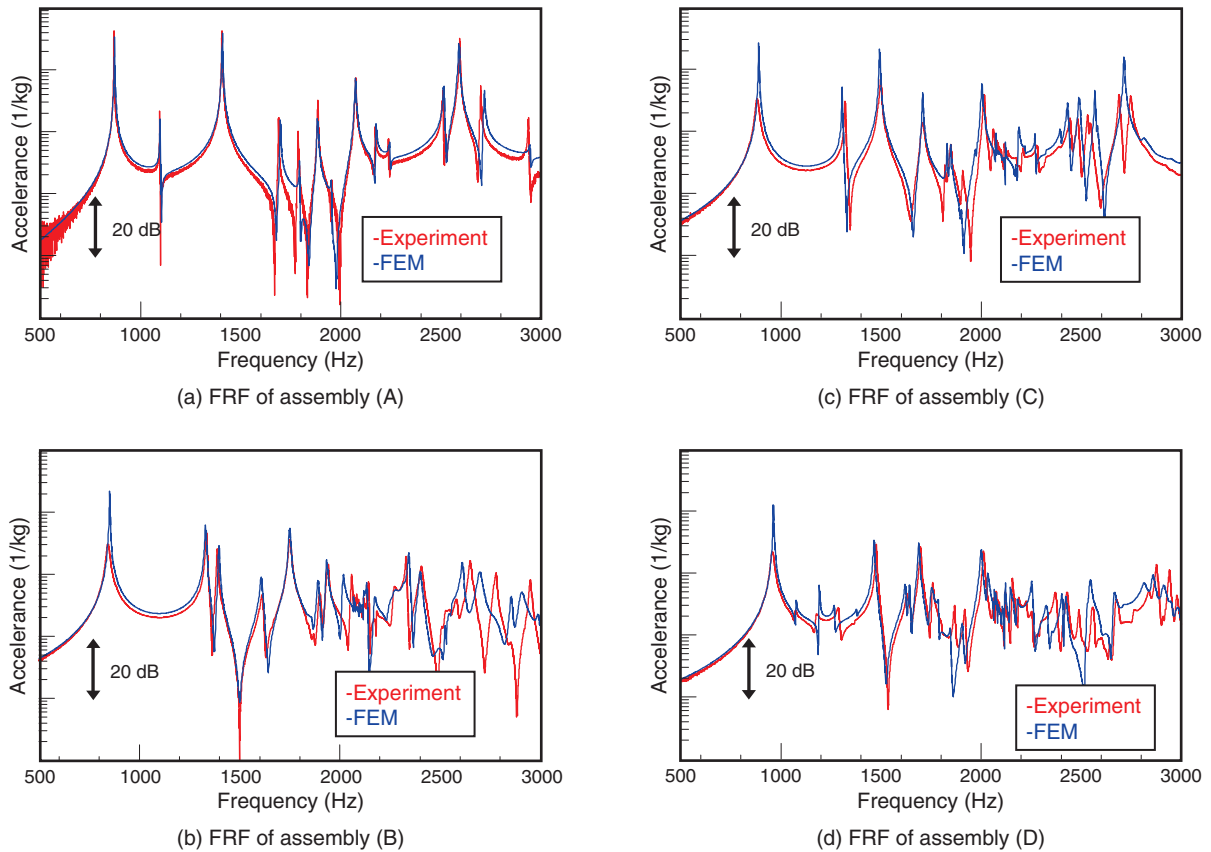


Fig. 15 FRF of assembled models

distribution of the shear moduli, and Fig. 14(c) the distribution of the damping coefficients.

4.3. Results of Frequency Response Analysis

Figure 15 shows a comparison of the results of frequency response analyses and tests for the four assembly compositions. The camshaft drive chain-side bearing was the exciting point, and the surface of the external wall of the lower block was the response point.

FRF has been determined with a high accuracy for each of the assembly compositions, indicating that the identified stiffness and damping of the joint elements were accurately modeled.

5. Verification of Method

5.1. Engine used in Verification

Tests were conducted using a different engine (the diesel engine employed in the 2008 model year ACCORD), as shown in Fig. 16, in order to verify the validity of the developed method. Because the analysis was to be conducted using only CAD data, the FEM model was formulated using 3D-CAD data, and deviations in the shapes of the parts in relation to the shapes of the parts used in tests were not corrected. The same material properties as were employed in the model discussed in Chapter 3 were used.

Figures 17(a) and 17(b) respectively show results for the cylinder block and the lower block as examples of verification for individual parts.

In the case of the lower block, deviations are observed which are thought to originate in factors including deviations of shape in relation to the parts used in tests⁽³⁾, but this corresponds to the eigen frequency deviation of around 3%, indicating sufficient accuracy for practical use for a model formulated from CAD data.

5.2. Results of Frequency Response Analysis

Figure 18 shows an example of the results of frequency response analyses conducted using the pressure conversion tables for identified joint element properties discussed in Section 4.1. The transmission-side bearing was the exciting

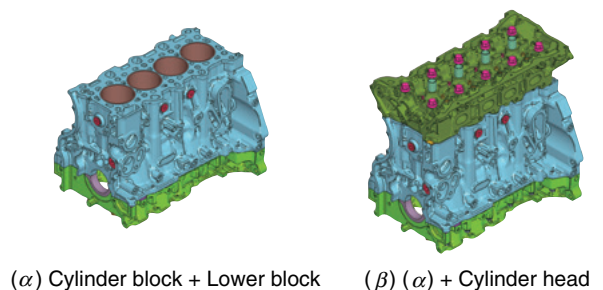
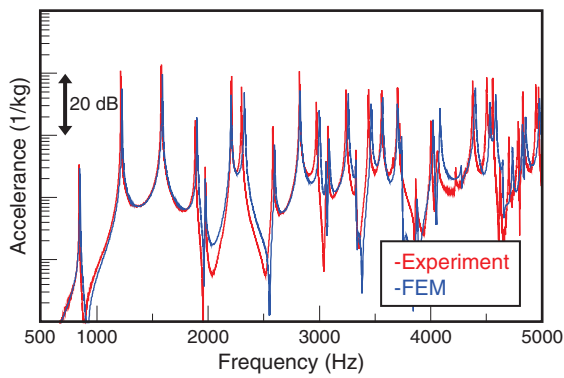
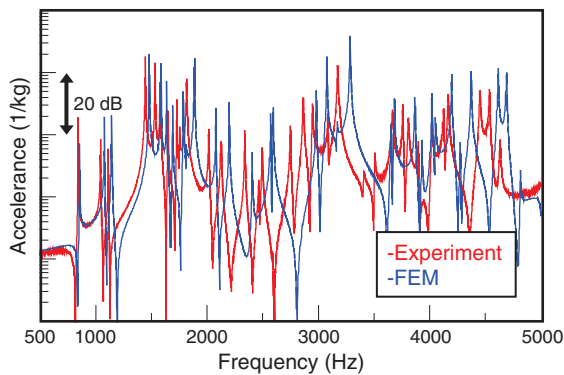


Fig. 16 Models for verification



(a) FRF of cylinder block



(b) FRF of lower block

Fig. 17 FRF of component for verification

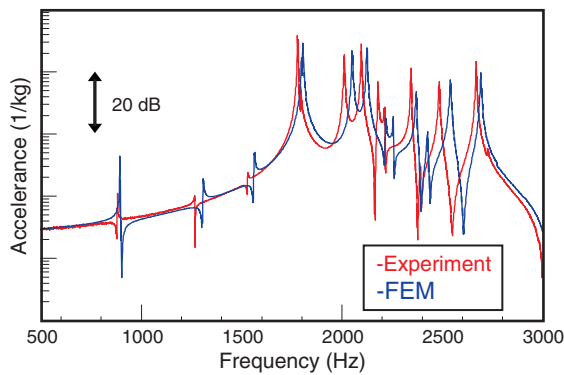
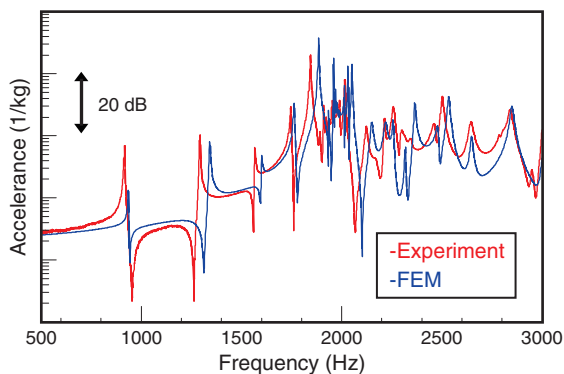
(a) FRF of assembly (α)(b) FRF of assembly (β)

Fig. 18 FRF of assembled models for verification

point, and the surface of the external wall of the cylinder block was the response point.

The level of deviation is basically identical to that for the individual parts, demonstrating the achievement of a level of accuracy suitable for practical use, and verifying the validity and general applicability of the developed method.

6. Discussion

The stiffness and damping of the joint elements used in the frequency response analyses discussed here were identified in order to correspond to test FRF values on the basis of conversion tables assumed to express properties dependent on pressure when parts are clamped. They were not physically verified as local property values. However, the results of employing the same method and the same conversion tables to analyze another engine indicated the achievement of sufficient accuracy for practical use. These results indicate that the method is basically valid and generally applicable as a method of expressing the stiffness and damping of the joints in engine structure vibration analysis models.

The method can be increased in accuracy and made more generally applicable in future by accumulating further examples of its application and making minor corrections in the conversion tables, which contain uncertain elements. It is also believed that it will be possible to expand the application of the method to joints offering different conditions, for example rubber seals.

7. Conclusion

A high-accuracy and generally applicable method of predicting frequency response functions of engine structures up to the radiated noise range was developed.

The method makes it possible to conduct highly accurate analyses using only 3D-CAD data and material properties for the main structural parts, the nonlinear elastic properties of the head gasket and other parts, and the bolt axial force.

Because the developed method makes it possible to conduct high-accuracy analyses at the drawing stage, it is expected that its use in future engine development will contribute to the reduction of the development period, the reduction of prototype numbers, and the realization of engine structures displaying low vibration response characteristics at reduced weights.

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Study of Suspension System for Reduction of Road Noise

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Toshimitsu MAKI*

Toshitaka SUGIMOTO*

ABSTRACT

The front strut / rear torsion beam suspension systems employed in the majority of compact cars were considered from the perspective of road noise performance. The low-frequency band from 100 to 200 Hz was the subject of the analysis. Tests using 14 vehicles employing front strut / rear torsion beam suspension systems demonstrated the existence of shared suspension vibration modes. FE model analyses clarified the mechanisms of the respective vibration modes, helping to enable the identification of design guidelines for the enhancement of road noise performance.

1. Introduction

The enhancement of fuel efficiency and the achievement of weight-savings in the vehicle body in order to realize this goal are today two of the most important technological elements in vehicle development. The achievement of weight-savings in the vehicle body and NV performance are normally antagonistic goals, and balancing the two represents a significant issue. Not only are vehicles very prone to be affected by road noise when the tires, wheels, suspension, subframe and other parts are reduced in weight, but there are numerous cases in compact cars in particular in which low-frequency (100 Hz to 200 Hz) road noise in the cabin represents an issue when driving on rough roads.

Numerous parts are implicated in low-frequency road noise, but vibration generated by the natural mode of the suspension is known to have a significant effect on cabin noise^{(1), (2)}. In order to accurately predict road noise, it is necessary to study the characteristics of the suspension with consideration of their combination with the natural frequencies of the tires and the acoustic characteristics of the body⁽³⁾. However, given that the basic specifications of the suspension are decided at an early stage of vehicle development, at which it is not possible to sufficiently predict the characteristics of the tires or the body, techniques for studying the NV performance simply by using the suspension system are also necessary. The research discussed in this paper focused exclusively on the suspension, and identified the vibration modes which made the greatest contribution to road noise and clarified the vibration mechanisms of those vibration modes.

The suspensions focused on in the study were front strut/rear torsion beam systems (Fig. 1), of the type generally used in compact cars.

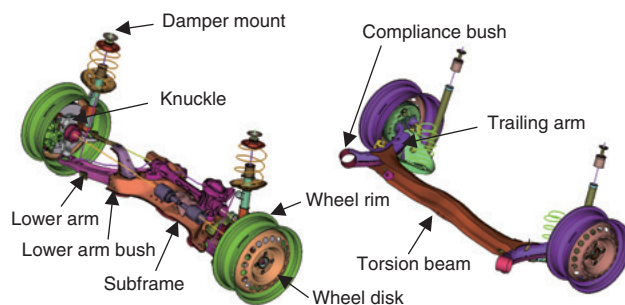


Fig. 1 Front strut and rear torsion beam suspension

2. Research Methodology

The arrangement of the suspension support points and the composition of the parts are virtually identical in front strut/rear torsion beam suspension systems, even when used in different vehicles. In addition, when the analysis is restricted to compact cars, there is no significant difference in the weight or stiffness of the wheels or the suspension parts. It can therefore be conjectured that the distribution of the spring and mass elements that determine the suspension vibration modes are similar for all these vehicles, and that each type of suspension displays vibration modes shared with other suspensions of the same type.

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First, the mechanisms of these vibration modes will be analyzed. Next, the degree to which the mechanisms of the vibration modes are shared will be considered by means of comparative tests using vehicles employing the same suspension type, and factors generating differences in vehicle performance will be analyzed. The results of studies of methods for road noise reduction derived from these vibration mechanisms will further be discussed.

3. Suspension Vibration Mechanisms

Figure 2 shows measurements of the cabin sound pressure level at the position of the driver's ears while driving on road surfaces for the evaluation of cabin road noise in a representative vehicle, and the lateral vibration response of the lower part of the knuckle when the bottom edges of the front wheels undergo lateral excitation. Both sound pressure and vibration display large peaks at close to 120 and 180 Hz.

Figure 3 shows measurements of the sound pressure level at the position of the rear passenger's ears and the vertical vibration response of the compliance bush connecting points when the bottom edges of the rear wheels undergo lateral excitation. Measurements for both sound pressure and vibration display large peaks at close to 120 Hz.

Below, the mechanism of the suspension vibration mode generating these vibration peaks will be classified by means of a parametric study using an FE model.

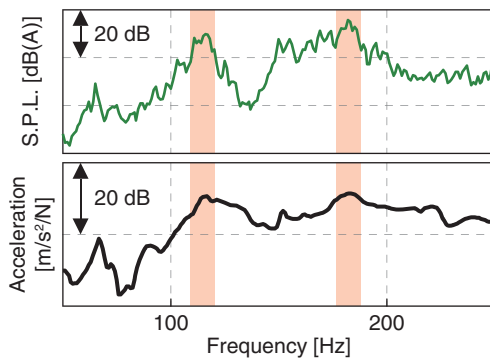


Fig. 2 Road noise and front suspension vibration

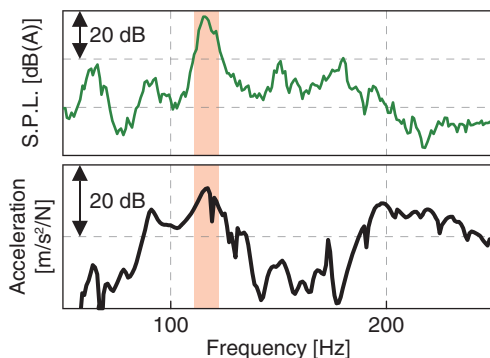


Fig. 3 Road noise and rear suspension vibration

3.1. Vibration Mode of Front Strut Suspension

Figure 4 shows the vibration mode close to 120 Hz. This is a mode in which the damper case, the knuckle, the wheel, and the lower arm rotate basically as a single unit in the direction of the camber. Because the suspension as a whole appears to rotate as a rigid body, this mode was termed “rigid camber mode” in the present research.

Figure 5 shows the vibration mode close to 180 Hz. In this mode, the wheel disk, the hub bearing, the knuckle and other parts undergo elastic deformation, while the wheel rotates in the direction of the camber. For this reason, this mode was termed “elastic camber mode” in the present research.

3.1.1. Vibration mechanism in rigid camber mode

Figure 6 shows the model generated when the rigid camber mode is expressed as a simple spring-mass system.

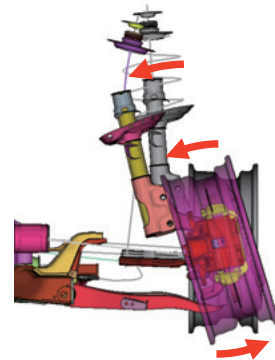


Fig. 4 Rigid camber mode

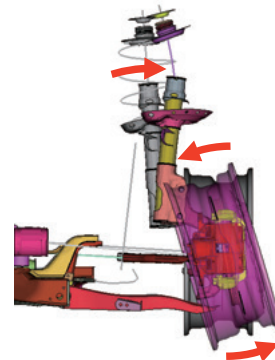


Fig. 5 Elastic camber mode

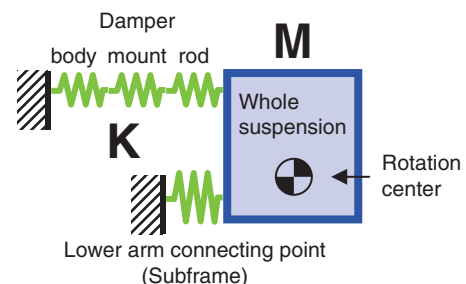


Fig. 6 System concept of rigid camber mode

M represents mass elements, while K represents spring elements. This can be regarded as a configuration in which the large mass of the entire suspension is supported by two springs, the upper section by the damper rod, the damper mount, and the damper connecting points, representing series springs, and the lower section by the stiffness at the lower arm connecting point. These upper and lower springs are the elements that control this mode.

3.1.2. Vibration mechanism in elastic camber mode

Figure 7 shows a simple spring-mass model of elastic camber mode. This can be regarded as a configuration in which the wheel rim, the main mass element, is supported at its center by two rotating springs, the prying stiffness of the hub bearing and the bending stiffness of the wheel disks, which represent the main spring elements. The swan neck section of the knuckle, the damper rod, and the damper mount also have an effect on this vibration mode as spring elements.

3.2. Vibration Mode of Rear Torsion Beam Suspension

Because the heavy, laterally coupled structure is supported by a comparatively flexible compliance bush in torsion beam suspensions, this configuration displays good isolation from vibration input, and offers advantages from the perspective of road noise⁽⁴⁾. However, because the torsion beam functions as a torsion spring connecting the left and right trailing arms, this member is prone to elastic deformation, and has a comparatively significant effect on road noise.

Figure 8 shows the vibration mode at 120 Hz. The U cross-section of a representative automotive torsion beam is open towards the front. While the wheels rotate in the direction of the camber, the center of the beam deforms elastically, generating pitching.

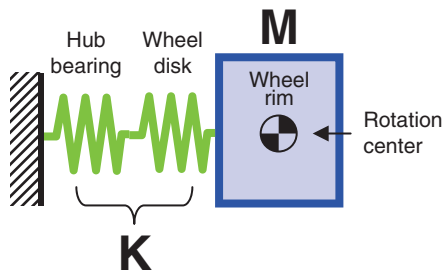


Fig. 7 System concept of elastic camber mode

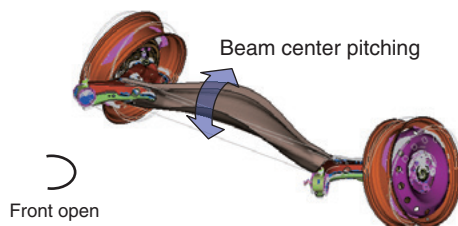


Fig. 8 Torsion beam vibration mode, with front open section

3.2.1. Vibration mechanism in torsion beam elastic deformation mode

As the conceptual diagram shown in Fig. 9 indicates, the wheel rims and the center of the beam represent the mass elements, while the trailing arm torsion, the torsion beam, and the compliance bush are the main spring elements.

As in the front suspension, the wheel rims rotate in the direction of the camber. A force producing vertical bending is therefore applied to the beam. Because the shear center of the beam U cross-section is positioned at the rear of the cross-section when the cross-section is open at the front, twisting occurs at the same time as vertical bending.

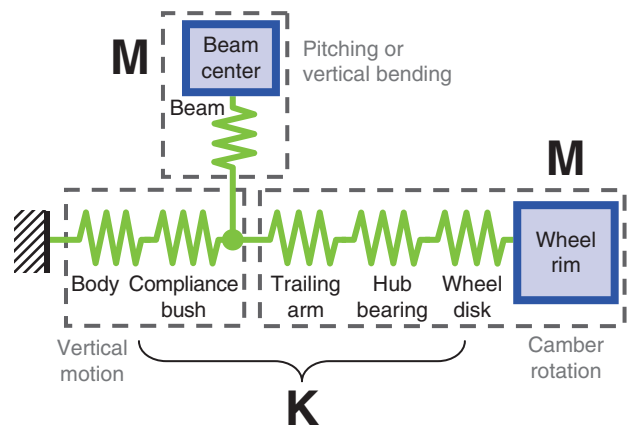


Fig. 9 System concept of torsion beam mode

4. Test Method

Excitation tests were performed on multiple vehicles in order to determine the extent to which the suspension vibration modes discussed in Chapter 3 were shared. All the test vehicles were drawn from the compact car category. Excitation tests were performed on the front strut suspension in ten vehicles, for a total of 25 types when differences in wheel materials and sizes are taken into consideration. Excitation tests were conducted on the rear torsion beam suspension in 14 vehicles. Transfer functions were measured in each section of the suspensions with the lateral direction of the bottom edges of the wheels as the points of excitation, and vibration modes and vibration transfer characteristics at representative points were compared.

5. Test Results

5.1. Results of Comparative Analyses of Front Suspensions

Rigid and elastic camber modes were studied in all the test vehicles. These two modes are believed to exist in almost all front strut suspensions. Figure 10 shows the vibration transfer functions for the lateral direction of the bottom edges of the knuckles in five vehicles. In rigid camber mode, there are minimal differences between the natural frequencies

for each vehicle, but significant differences in their levels of vibration response. In elastic camber mode, there are significant differences between natural frequencies, but minimal differences between levels of vibration response.

Figure 11 shows the relationship between the lateral vibration response of the knuckles and the vertical stiffness of the lower arm connecting point at rigid camber mode frequencies. These results indicate that the vibration of the suspension will decrease, and road noise performance will consequently be enhanced, in proportion to the stiffness of the connecting point of the lower arm, the main spring element.

5.2. Results of Comparative Analyses of Rear Suspensions

Figure 12 shows the arrangement of all measured modes. Each vehicle shares a large number of vibration modes from 100 to 200 Hz, but the vibration modes in which vertical vibration of the compliance bush (the main input point) is high and becomes the major factor in road noise differ with the direction of the opening of the beam U cross-section.

When the U cross-section is open at the front, the vibration mode generates pitching of the center of the beam, as shown in Fig. 8; when the U cross-section is open at the bottom, the vibration mode generates vertical vibration of the entire suspension while the beam bends vertically (Fig. 13).

The vibration response level in this mode was compared in the vertical direction at the body side of the compliance bush connecting point. Figure 14 shows the relationship

between the degree of isolation of the compliance bush from vibration in the vertical direction and vibration on the body side. Significant differences can be observed between each vehicle, and the magnitude of these differences is strongly correlated with the degree of isolation of the compliance bush from vertical vibration. It may be indicated that the greater the degree of isolation of the compliance bush from vibration, i.e., the lower the vertical stiffness of the bush, the lower the degree of vibration on the body side, and hence the better the road noise performance.

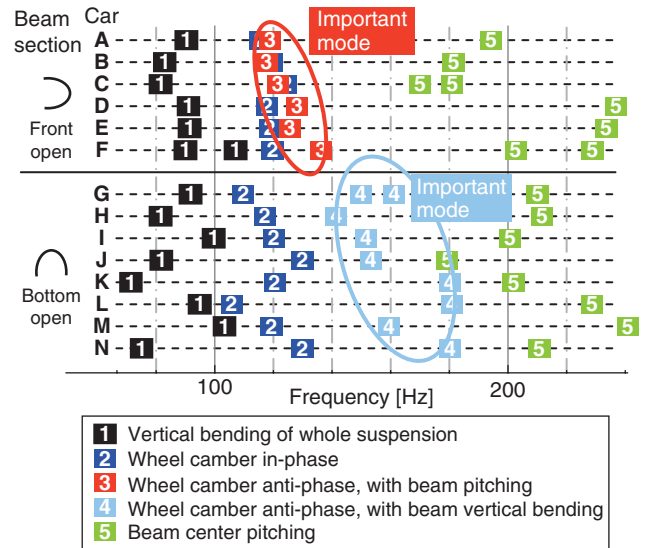


Fig. 12 Rear torsion beam suspension modes

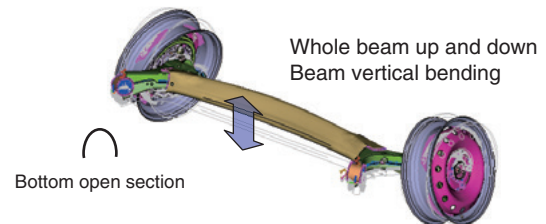


Fig. 13 Torsion beam vibration mode, with bottom open section

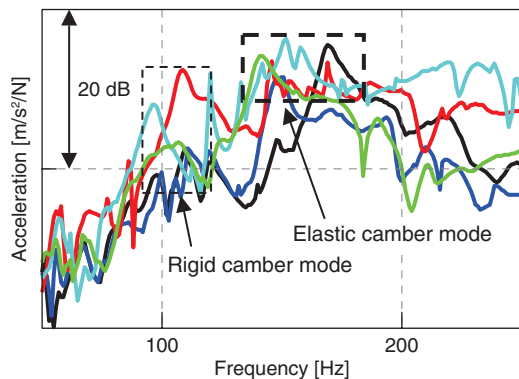


Fig. 10 Camber modes of various cars

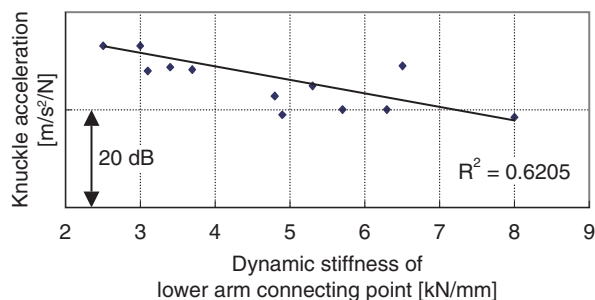


Fig. 11 Lower arm connecting point vertical dynamic stiffness and knuckle acceleration

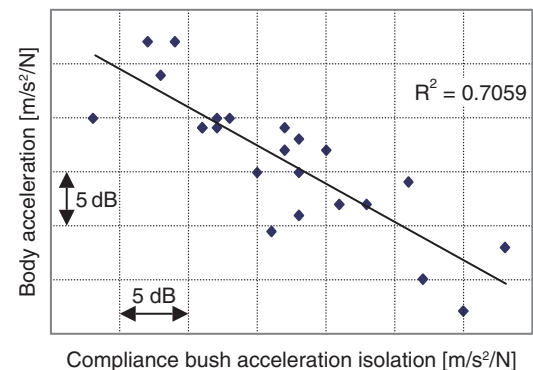


Fig. 14 Compliance bush isolation and body acceleration

6. Study of Methods for Reduction of Road Noise

This chapter will discuss methods for the reduction of the sound pressure level of road noise originating in the suspension vibration modes.

6.1. Front Rigid Camber Mode

Methods of reducing vibration at the lower arm connecting point, a transfer path that makes a significant contribution to cabin noise, were examined. Because in this suspension configuration the large mass rotating in the camber direction is supported by two distant spring elements, the rotational center of the mass can be controlled by means of the ratio between the two springs. It was possible to reduce the vibrational input to the lower arm by reducing the lateral spring of the damper mount in order to lower the rotational center, which had been positioned close to the center of the wheel center in the base state. Figure 15 shows a conceptual diagram, while Fig. 16 shows the results of testing of the measure in an actual vehicle.

6.2. Front Elastic Camber Mode

6.2.1. Correlation between elastic camber mode natural frequencies and cabin noise

As indicated in Section 5.1, in elastic camber mode, there are minimal differences in vibration levels between vehicles, but significant differences between natural frequencies.

A study of the relationship between natural frequencies in the elastic camber mode and cabin noise during lateral excitation of the lower edge of the wheels revealed the loose correlation shown in Fig. 17. The results showed that vibration sound pressure would tend to be lower the lower the natural frequency.

In most cases, a low natural frequency in the elastic camber mode is achieved by means of increasing the weight

of the wheel rims, the main mass element, and a reduction in vibration sound pressure is a natural outcome. However, measurements conducted in the present project showed that almost identical results were obtained when comparing wheels of basically identical mass (Fig. 18), indicating that irrespective of the weight of the wheels, lower natural

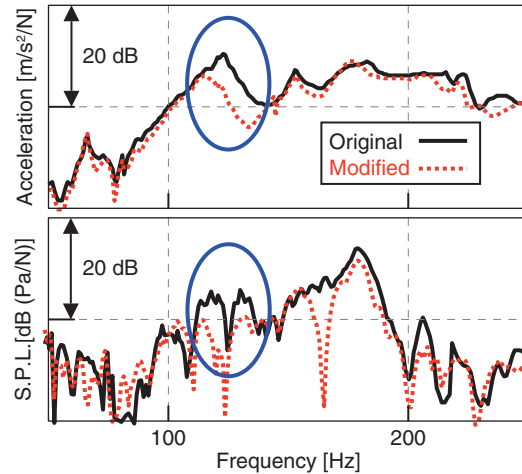


Fig. 16 Enhancement of rigid camber mode by decreasing damper mount lateral stiffness

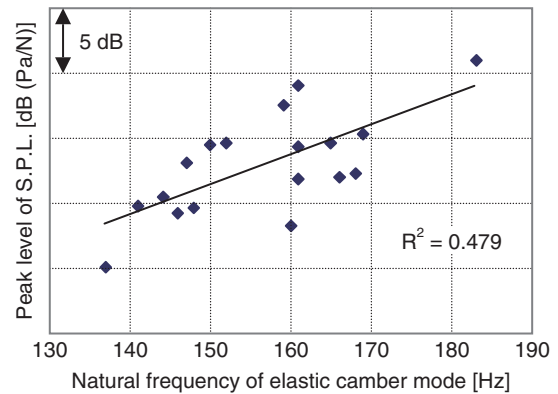


Fig. 17 Elastic camber mode natural frequency and S.P.L.

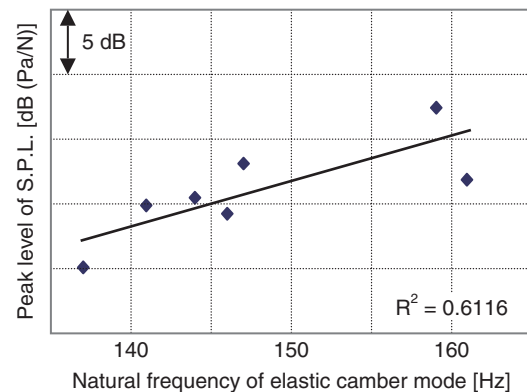


Fig. 18 Elastic camber mode natural frequency and S.P.L. around 8 kg wheel only

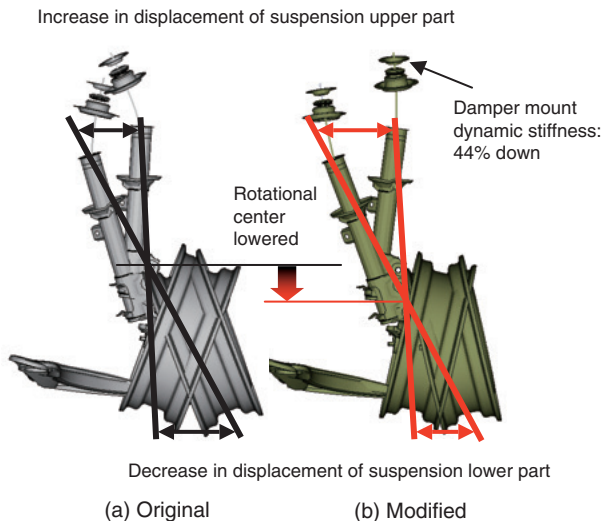


Fig. 15 Lower rotational center of rigid camber mode

frequencies could be considered to tend to enhance road noise performance. Based on these results, it was conjectured that the lower the natural frequency of the suspension elastic camber mode, the less coupled it is with that of the body side response points, and that all the vehicles would display this point of weakness in relation to the body side response points at frequencies close to the elastic camber mode.

6.2.2. Considerations regarding body acoustic sensitivity

This sub-section will discuss acoustic sensitivity from the suspension connecting points, the body side response points, to the sound measurement position in the cabin (body acoustic sensitivity). Because the body acoustic sensitivity is determined by a number of elements, including the inertance of the suspension connecting point, the body framework mode, and the vibration characteristics of the various panels, it is unlikely that the point of weakness in relation to the body side exists in the same frequency band in every vehicle. However, because the basic characteristics of the cabin acoustic mode – one of the elements which determines the body acoustic sensitivity – is determined by the shape and volume of the cabin, it was possible to consider that the acoustic mode peak would occur in similar frequency bands in the present research, which employed vehicles of almost identical size.

Figure 19 shows measurements of cabin acoustic mode and body acoustic sensitivity. The transfer function (sound pressure / volume acceleration) to a microphone set up at the position of the driver's ears when random noise was generated in the same position via a speaker was employed as an index to express the cabin acoustic mode. The average value of body acoustic sensitivity from all the front suspension connecting points was employed as an index to express body acoustic sensitivity. As Fig. 19 shows, the average acoustic sensitivity and the cabin acoustic mode both peak at the same frequencies, between 160 and

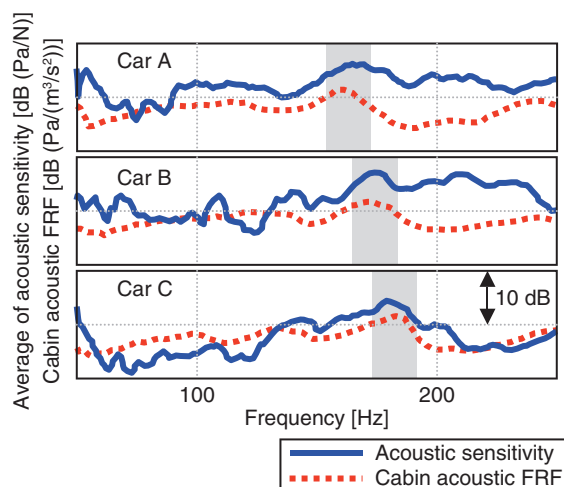


Fig. 19 Cabin acoustic mode and body acoustic sensitivity

180 Hz, close to the elastic camber mode frequency band. Given this, it may be considered that the natural frequency of cabin acoustic mode will also appear in body acoustic sensitivity as a peak. Because this acoustic mode frequency can be predicted when the cabin shape is determined at the initial stage of development, it would be possible to take measures to shift the natural frequency of the elastic camber mode of the suspension.

Figure 20 converts the horizontal axis of Fig. 17 into the difference between the frequencies of elastic camber mode and cabin acoustic mode. It is clear that the correlation becomes higher.

6.2.3. Verification of effect in reducing road noise

The reduction in road noise produced by modifying the natural frequency of the elastic camber mode was studied. The prying stiffness of the bearings and the bending stiffness of the wheel disks, the main spring elements, were modified.

Figure 21 shows results for wheel excitation tests with the original and modified hub bearing stiffness, and

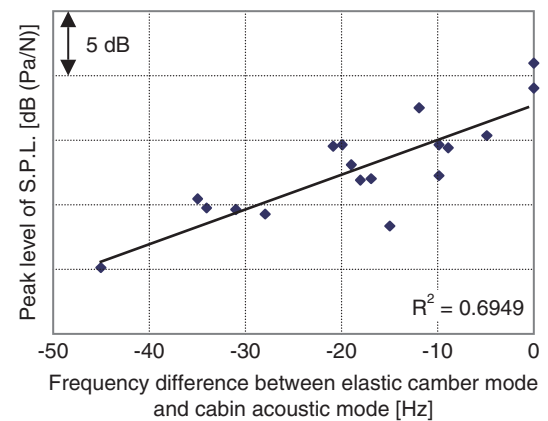


Fig. 20 Frequency difference between elastic camber mode and cabin acoustic mode, and S.P.L.

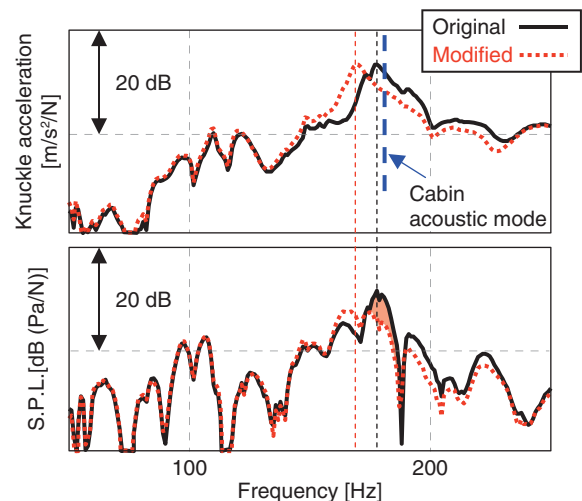


Fig. 21 Enhancement of road noise by decreasing stiffness of hub bearing

Fig. 22 shows results for the original and modified wheel disk stiffness. In both cases, the natural frequency of the elastic camber mode has been lowered, and cabin noise has been ameliorated.

6.3. Rear Torsion Beam Elastic Deformation Mode

As indicated in Section 5.2, reducing the vertical stiffness of the compliance bush is an effective means of enhancing road noise performance. In the present project, the reduction of the vertical vibration of the compliance bush connecting point by modifying the shape of the beam itself was attempted as another method. The study employed an analysis using an FE model of a torsion beam for a mass-production vehicle with a front opening.

As indicated above, in a torsion beam with an opening at the front, the beam twists as it undergoes vertical bending, and the central section displays pitching behavior with the rear extremity of the cross-section as the center. There is therefore significant vertical vibration of the front section of the center of the beam, and this increases the vibration of the connecting point of the compliance bush.

Because the compliance bush connecting point is located on the underside of the rear side frame, which extends longitudinally, from a structural perspective the stiffness of the connecting points is higher in the longitudinal direction than the vertical direction, and body acoustic sensitivity is lower in the longitudinal direction. Therefore, if it is possible to reduce vertical vibration of the suspension even if longitudinal vibration increases to some extent, road noise will be reduced. In order to realize this vibration characteristic, the angle of beam opening was slanted. Figure 23 shows the results, and Fig. 24 shows the difference in motion. The vertical vibration of the compliance bush connecting point has been reduced by modifying the vibration of the center of the beam from the vertical direction to the longitudinal direction. When

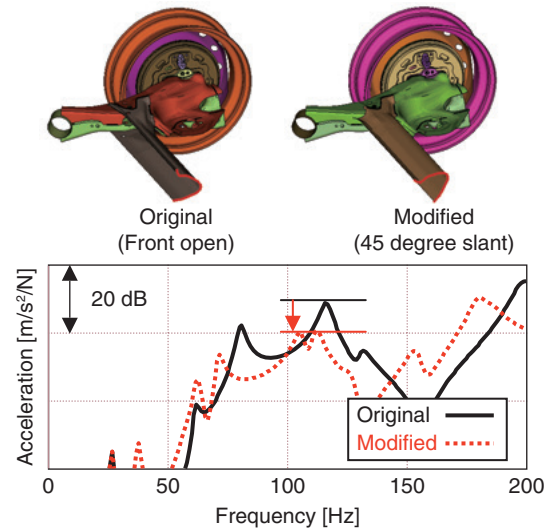


Fig. 23 Enhancement of torsion beam pitching mode

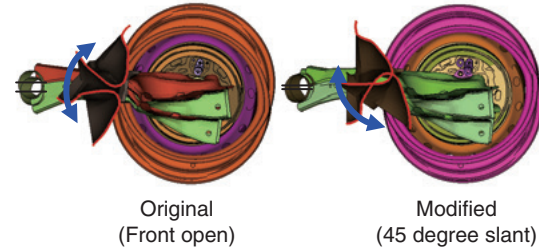


Fig. 24 Enhancement of torsion beam mode shape

the direction of the opening was shifted to the bottom of the beam using the same model, the characteristic bottom opening mode described in Fig. 13 became prominent, and road noise performance was affected. In the analysis discussed here, 45° was the optimum opening angle.

7. Conclusion

The following conclusions were drawn from a parametric study of a suspension FE model and a survey of 14 compact cars.

- (1) It was demonstrated that front strut and rear torsion beam suspensions each possess shared vibration modes that are the chief factors in road noise. The main modes are front rigid and elastic camber modes, and a rear wheel camber mode that involves elastic deformation of the rear torsion beam.
- (2) The main spring and mass elements for control of the vibration mode were identified for each mode.
- (3) The potential for enhancement of road noise performance through the application of suspension vibration mode control derived from the mechanisms of generation of road noise was verified in vehicle tests and computational predictions.

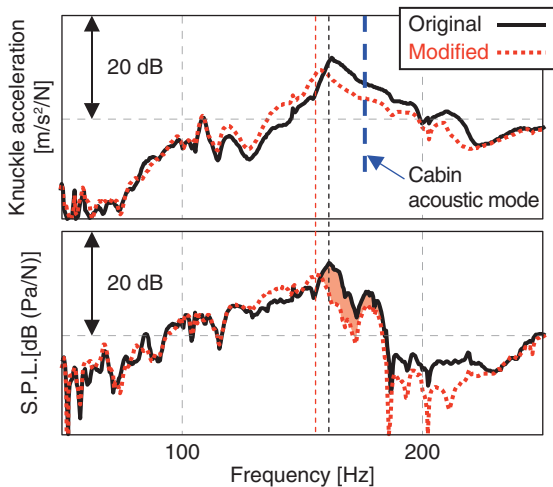
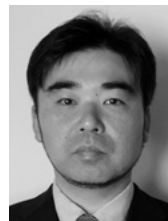


Fig. 22 Enhancement of road noise by decreasing stiffness of wheel disk

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— Editorial Afterword —

In striving to continuously advance the reputation of the Honda Technical Review in North America, and explain its value to fellow engineers, I always try to reflect on my own experiences in the development and promotion of technologies and products, and the research of human motivation.

My humble opinion is that in product development we strive to create the perfect harmony of technologies to bring joy to our customers. But often, within that harmony, the individual beauty of each technology cannot be readily appreciated. By creating a technical paper, we can shine a light on one unique technology and reveal its individual beauty by going beyond the “what” and into the “how” and the “why.” This brings our philosophy to life, separates Honda from others, and creates greater appreciation for the product in which the technology is incorporated.

The effort to shine that light - the required process of methodical consideration and explanation - further strengthens the understanding and capability of the author. Furthermore, that effort ultimately leads to the satisfaction of the instinctive human desires to be recognized and appreciated, and to contribute to the greater good. That satisfaction then becomes the energy that powers us and fuels our motivation.

As the beauty of each individual technology is understood, the expectation of excellence expands to elevate the stature and technological presence of the entire brand. Our technical papers then become the continuously advancing standard for others to aspire to surpass and become the rungs on which our colleagues will stand in the future to reach even higher.

— (John E. Dirrig) —

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